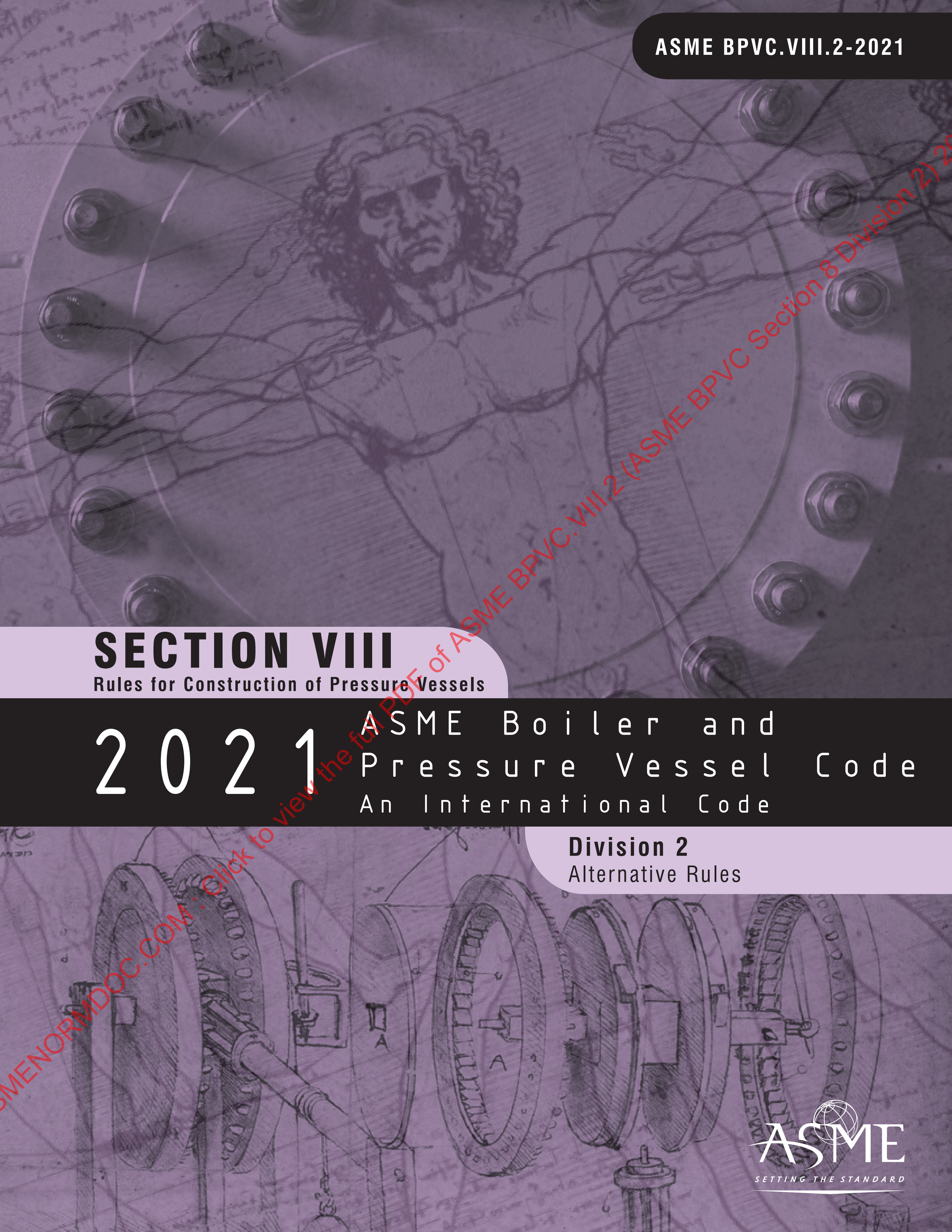
ASME BPVC.VIII.2-2021



SECTION VIII

Rules for Construction of Pressure Vessels

2021 ASME Boiler and Pressure Vessel Code

An International Code

Division 2

Alternative Rules

ASME
SETTING THE STANDARD

AN INTERNATIONAL CODE

2021 ASME Boiler & Pressure Vessel Code

2021 Edition July 1, 2021

VIII

RULES FOR CONSTRUCTION OF PRESSURE VESSELS

Division 2

Alternative Rules

ASME Boiler and Pressure Vessel Committee
on Pressure Vessels

The American Society of
Mechanical Engineers

Two Park Avenue • New York, NY • 10016 USA

Date of Issuance: July 1, 2021

This international code or standard was developed under procedures accredited as meeting the criteria for American National Standards and it is an American National Standard. The Standards Committee that approved the code or standard was balanced to assure that individuals from competent and concerned interests have had an opportunity to participate. The proposed code or standard was made available for public review and comment that provides an opportunity for additional public input from industry, academia, regulatory agencies, and the public-at-large.

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The endnotes and preamble in this document (if any) are part of this American National Standard.

ASME Collective Membership Mark

ASME Single Certification Mark

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INTERPRETATIONS

Interpretations are issued in real time in ASME's Interpretations Database at http://go.asme.org/Interpretations. Historical BPVC interpretations may also be found in the Database.

CODE CASES

The Boiler and Pressure Vessel Code committees meet regularly to consider proposed additions and revisions to the Code and to formulate Cases to clarify the intent of existing requirements or provide, when the need is urgent, rules for materials or constructions not covered by existing Code rules. Those Cases that have been adopted will appear in the appropriate 2021 Code Cases book: "Boilers and Pressure Vessels" or "Nuclear Components." Each Code Cases book is updated with seven Supplements. Supplements will be sent or made available automatically to the purchasers of the Code Cases books up to the publication of the 2023 Code. Annulments of Code Cases become effective six months after the first announcement of the annulment in a Code Case Supplement or Edition of the appropriate Code Case book. Code Case users can check the current status of any Code Case at http://go.asme.org/BPVCCDatabase. Code Case users can also view an index of the complete list of Boiler and Pressure Vessel Code Cases and Nuclear Code Cases at http://go.asme.org/BPVCC.

FOREWORD*

(21)

In 1911, The American Society of Mechanical Engineers established the Boiler and Pressure Vessel Committee to formulate standard rules for the construction of steam boilers and other pressure vessels. In 2009, the Boiler and Pressure Vessel Committee was superseded by the following committees:

(a) Committee on Power Boilers (I)
(b) Committee on Materials (II)
(c) Committee on Construction of Nuclear Facility Components (III)
(d) Committee on Heating Boilers (IV)
(e) Committee on Nondestructive Examination (V)
(f) Committee on Pressure Vessels (VIII)
(g) Committee on Welding, Brazing, and Fusing (IX)
(h) Committee on Fiber-Reinforced Plastic Pressure Vessels (X)
(i) Committee on Nuclear Inservice Inspection (XI)
(j) Committee on Transport Tanks (XII)
(k) Committee on Overpressure Protection (XIII)
(l) Technical Oversight Management Committee (TOMC)

Where reference is made to "the Committee" in this Foreword, each of these committees is included individually and collectively.

The Committee's function is to establish rules of safety relating only to pressure integrity, which govern the construction** of boilers, pressure vessels, transport tanks, and nuclear components, and the inservice inspection of nuclear components and transport tanks. The Committee also interprets these rules when questions arise regarding their intent. The technical consistency of the Sections of the Code and coordination of standards development activities of the Committees is supported and guided by the Technical Oversight Management Committee. This Code does not address other safety issues relating to the construction of boilers, pressure vessels, transport tanks, or nuclear components, or the inservice inspection of nuclear components or transport tanks. Users of the Code should refer to the pertinent codes, standards, laws, regulations, or other relevant documents for safety issues other than those relating to pressure integrity. Except for Sections XI and XII, and with a few other exceptions, the rules do not, of practical necessity, reflect the likelihood and consequences of deterioration in service related to specific service fluids or external operating environments. In formulating the rules, the Committee considers the needs of users, manufacturers, and inspectors of pressure vessels. The objective of the rules is to afford reasonably certain protection of life and property, and to provide a margin for deterioration in service to give a reasonably long, safe period of usefulness. Advancements in design and materials and evidence of experience have been recognized.

This Code contains mandatory requirements, specific prohibitions, and nonmandatory guidance for construction activities and inservice inspection and testing activities. The Code does not address all aspects of these activities and those aspects that are not specifically addressed should not be considered prohibited. The Code is not a handbook and cannot replace education, experience, and the use of engineering judgment. The phrase *engineering judgment* refers to technical judgments made by knowledgeable engineers experienced in the application of the Code. Engineering judgments must be consistent with Code philosophy, and such judgments must never be used to overrule mandatory requirements or specific prohibitions of the Code.

The Committee recognizes that tools and techniques used for design and analysis change as technology progresses and expects engineers to use good judgment in the application of these tools. The designer is responsible for complying with Code rules and demonstrating compliance with Code equations when such equations are mandatory. The Code neither requires nor prohibits the use of computers for the design or analysis of components constructed to the

* The information contained in this Foreword is not part of this American National Standard (ANS) and has not been processed in accordance with ANSI's requirements for an ANS. Therefore, this Foreword may contain material that has not been subjected to public review or a consensus process. In addition, it does not contain requirements necessary for conformance to the Code.

** *Construction*, as used in this Foreword, is an all-inclusive term comprising materials, design, fabrication, examination, inspection, testing, certification, and overpressure protection.

requirements of the Code. However, designers and engineers using computer programs for design or analysis are cautioned that they are responsible for all technical assumptions inherent in the programs they use and the application of these programs to their design.

The rules established by the Committee are not to be interpreted as approving, recommending, or endorsing any proprietary or specific design, or as limiting in any way the manufacturer's freedom to choose any method of design or any form of construction that conforms to the Code rules.

The Committee meets regularly to consider revisions of the rules, new rules as dictated by technological development, Code Cases, and requests for interpretations. Only the Committee has the authority to provide official interpretations of this Code. Requests for revisions, new rules, Code Cases, or interpretations shall be addressed to the Secretary in writing and shall give full particulars in order to receive consideration and action (see Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees). Proposed revisions to the Code resulting from inquiries will be presented to the Committee for appropriate action. The action of the Committee becomes effective only after confirmation by ballot of the Committee and approval by ASME. Proposed revisions to the Code approved by the Committee are submitted to the American National Standards Institute (ANSI) and published at http://go.asme.org/BPVCPublicReview to invite comments from all interested persons. After public review and final approval by ASME, revisions are published at regular intervals in Editions of the Code.

The Committee does not rule on whether a component shall or shall not be constructed to the provisions of the Code. The scope of each Section has been established to identify the components and parameters considered by the Committee in formulating the Code rules.

Questions or issues regarding compliance of a specific component with the Code rules are to be directed to the ASME Certificate Holder (Manufacturer). Inquiries concerning the interpretation of the Code are to be directed to the Committee. ASME is to be notified should questions arise concerning improper use of the ASME Single Certification Mark.

When required by context in this Section, the singular shall be interpreted as the plural, and vice versa, and the feminine, masculine, or neuter gender shall be treated as such other gender as appropriate.

The words "shall," "should," and "may" are used in this Standard as follows:

– *Shall* is used to denote a requirement.
– *Should* is used to denote a recommendation.
– *May* is used to denote permission, neither a requirement nor a recommendation.

STATEMENT OF POLICY ON THE USE OF THE ASME SINGLE CERTIFICATION MARK AND CODE AUTHORIZATION IN ADVERTISING

ASME has established procedures to authorize qualified organizations to perform various activities in accordance with the requirements of the ASME Boiler and Pressure Vessel Code. It is the aim of the Society to provide recognition of organizations so authorized. An organization holding authorization to perform various activities in accordance with the requirements of the Code may state this capability in its advertising literature.

Organizations that are authorized to use the ASME Single Certification Mark for marking items or constructions that have been constructed and inspected in compliance with the ASME Boiler and Pressure Vessel Code are issued Certificates of Authorization. It is the aim of the Society to maintain the standing of the ASME Single Certification Mark for the benefit of the users, the enforcement jurisdictions, and the holders of the ASME Single Certification Mark who comply with all requirements.

Based on these objectives, the following policy has been established on the usage in advertising of facsimiles of the ASME Single Certification Mark, Certificates of Authorization, and reference to Code construction. The American Society of Mechanical Engineers does not "approve," "certify," "rate," or "endorse" any item, construction, or activity and there shall be no statements or implications that might so indicate. An organization holding the ASME Single Certification Mark and/or a Certificate of Authorization may state in advertising literature that items, constructions, or activities "are built (produced or performed) or activities conducted in accordance with the requirements of the ASME Boiler and Pressure Vessel Code," or "meet the requirements of the ASME Boiler and Pressure Vessel Code." An ASME corporate logo shall not be used by any organization other than ASME.

The ASME Single Certification Mark shall be used only for stamping and nameplates as specifically provided in the Code. However, facsimiles may be used for the purpose of fostering the use of such construction. Such usage may be by an association or a society, or by a holder of the ASME Single Certification Mark who may also use the facsimile in advertising to show that clearly specified items will carry the ASME Single Certification Mark.

STATEMENT OF POLICY ON THE USE OF ASME MARKING TO IDENTIFY MANUFACTURED ITEMS

The ASME Boiler and Pressure Vessel Code provides rules for the construction of boilers, pressure vessels, and nuclear components. This includes requirements for materials, design, fabrication, examination, inspection, and stamping. Items constructed in accordance with all of the applicable rules of the Code are identified with the ASME Single Certification Mark described in the governing Section of the Code.

Markings such as "ASME," "ASME Standard," or any other marking including "ASME" or the ASME Single Certification Mark shall not be used on any item that is not constructed in accordance with all of the applicable requirements of the Code.

Items shall not be described on ASME Data Report Forms nor on similar forms referring to ASME that tend to imply that all Code requirements have been met when, in fact, they have not been. Data Report Forms covering items not fully complying with ASME requirements should not refer to ASME or they should clearly identify all exceptions to the ASME requirements.

(21) # SUBMITTAL OF TECHNICAL INQUIRIES TO THE BOILER AND PRESSURE VESSEL STANDARDS COMMITTEES

1 INTRODUCTION

(a) The following information provides guidance to Code users for submitting technical inquiries to the applicable Boiler and Pressure Vessel (BPV) Standards Committee (hereinafter referred to as the Committee). See the guidelines on approval of new materials under the ASME Boiler and Pressure Vessel Code in Section II, Part D for requirements for requests that involve adding new materials to the Code. See the guidelines on approval of new welding and brazing materials in Section II, Part C for requirements for requests that involve adding new welding and brazing materials ("consumables") to the Code.

Technical inquiries can include requests for revisions or additions to the Code requirements, requests for Code Cases, or requests for Code Interpretations, as described below:

(1) Code Revisions. Code revisions are considered to accommodate technological developments, to address administrative requirements, to incorporate Code Cases, or to clarify Code intent.

(2) Code Cases. Code Cases represent alternatives or additions to existing Code requirements. Code Cases are written as a Question and Reply, and are usually intended to be incorporated into the Code at a later date. When used, Code Cases prescribe mandatory requirements in the same sense as the text of the Code. However, users are cautioned that not all regulators, jurisdictions, or Owners automatically accept Code Cases. The most common applications for Code Cases are as follows:

(-a) to permit early implementation of an approved Code revision based on an urgent need

(-b) to permit use of a new material for Code construction

(-c) to gain experience with new materials or alternative requirements prior to incorporation directly into the Code

(3) Code Interpretations

(-a) Code Interpretations provide clarification of the meaning of existing requirements in the Code and are presented in Inquiry and Reply format. Interpretations do not introduce new requirements.

(-b) Interpretations will be issued only if existing Code text is ambiguous or conveys conflicting requirements. If a revision of the requirements is required to support the Interpretation, an Intent Interpretation will be issued in parallel with a revision to the Code.

(b) Code requirements, Code Cases, and Code Interpretations established by the Committee are not to be considered as approving, recommending, certifying, or endorsing any proprietary or specific design, or as limiting in any way the freedom of manufacturers, constructors, or Owners to choose any method of design or any form of construction that conforms to the Code requirements.

(c) Inquiries that do not comply with the following guidance or that do not provide sufficient information for the Committee's full understanding may result in the request being returned to the Inquirer with no action.

2 INQUIRY FORMAT

Submittals to the Committee should include the following information:

(a) Purpose. Specify one of the following:

(1) request for revision of present Code requirements

(2) request for new or additional Code requirements

(3) request for Code Case

(4) request for Code Interpretation

(b) Background. The Inquirer should provide the information needed for the Committee's understanding of the Inquiry, being sure to include reference to the applicable Code Section, Division, Edition, Addenda (if applicable), paragraphs, figures, and tables. This information should include a statement indicating why the included paragraphs, figures, or tables are ambiguous or convey conflicting requirements. Preferably, the Inquirer should provide a copy of, or relevant extracts from, the specific referenced portions of the Code.

(c) Presentations. The Inquirer may desire to attend or be asked to attend a meeting of the Committee to make a formal presentation or to answer questions from the Committee members with regard to the Inquiry. Attendance at a BPV Standards Committee meeting shall be at the expense of the Inquirer. The Inquirer's attendance or lack of attendance at a meeting will not be used by the Committee as a basis for acceptance or rejection of the Inquiry by the Committee. However, if the Inquirer's request is unclear, attendance by the Inquirer or a representative may be necessary for the Committee to understand the request sufficiently to be able to provide an Interpretation. If the Inquirer desires to make a presentation at a Committee meeting, the Inquirer should provide advance notice to the Committee Secretary, to ensure time will be allotted for the presentation in the meeting agenda. The Inquirer should consider the need for additional audiovisual equipment that might not otherwise be provided by the Committee. With sufficient advance notice to the Committee Secretary, such equipment may be made available.

3 CODE REVISIONS OR ADDITIONS

Requests for Code revisions or additions should include the following information:

(a) Requested Revisions or Additions. For requested revisions, the Inquirer should identify those requirements of the Code that they believe should be revised, and should submit a copy of, or relevant extracts from, the appropriate requirements as they appear in the Code, marked up with the requested revision. For requested additions to the Code, the Inquirer should provide the recommended wording and should clearly indicate where they believe the additions should be located in the Code requirements.

(b) Statement of Need. The Inquirer should provide a brief explanation of the need for the revision or addition.

(c) Background Information. The Inquirer should provide background information to support the revision or addition, including any data or changes in technology that form the basis for the request, that will allow the Committee to adequately evaluate the requested revision or addition. Sketches, tables, figures, and graphs should be submitted, as appropriate. The Inquirer should identify any pertinent portions of the Code that would be affected by the revision or addition and any portions of the Code that reference the requested revised or added paragraphs.

4 CODE CASES

Requests for Code Cases should be accompanied by a statement of need and background information similar to that described in 3(b) and 3(c), respectively, for Code revisions or additions. The urgency of the Code Case (e.g., project underway or imminent, new procedure) should be described. In addition, it is important that the request is in connection with equipment that will bear the ASME Single Certification Mark, with the exception of Section XI applications. The proposed Code Case should identify the Code Section and Division, and should be written as a Question and a Reply, in the same format as existing Code Cases. Requests for Code Cases should also indicate the applicable Code Editions and Addenda (if applicable) to which the requested Code Case applies.

5 CODE INTERPRETATIONS

(a) Requests for Code Interpretations should be accompanied by the following information:

(1) Inquiry. The Inquirer should propose a condensed and precise Inquiry, omitting superfluous background information and, when possible, composing the Inquiry in such a way that a "yes" or a "no" Reply, with brief limitations or conditions, if needed, can be provided by the Committee. The proposed question should be technically and editorially correct.

(2) Reply. The Inquirer should propose a Reply that clearly and concisely answers the proposed Inquiry question. Preferably, the Reply should be "yes" or "no," with brief limitations or conditions, if needed.

(3) Background Information. The Inquirer should include a statement indicating why the included paragraphs, figures, or tables are ambiguous or convey conflicting requirements. The Inquirer should provide any need or background information, such as described in 3(b) and 3(c), respectively, for Code revisions or additions, that will assist the Committee in understanding the proposed Inquiry and Reply.

If the Inquirer believes a revision of the Code requirements would be helpful to support the Interpretation, the Inquirer may propose such a revision for consideration by the Committee. In most cases, such a proposal is not necessary.

(b) Requests for Code Interpretations should be limited to an Interpretation of a particular requirement in the Code or in a Code Case. Except with regard to interpreting a specific Code requirement, the Committee is not permitted to consider consulting-type requests such as the following:

(1) a review of calculations, design drawings, welding qualifications, or descriptions of equipment or parts to determine compliance with Code requirements

(2) a request for assistance in performing any Code-prescribed functions relating to, but not limited to, material selection, designs, calculations, fabrication, inspection, pressure testing, or installation

(3) a request seeking the rationale for Code requirements

6 SUBMITTALS

(a) Submittal. Requests for Code Interpretation should preferably be submitted through the online Interpretation Submittal Form. The form is accessible at http://go.asme.org/InterpretationRequest. Upon submittal of the form, the Inquirer will receive an automatic e-mail confirming receipt. If the Inquirer is unable to use the online form, the Inquirer may mail the request to the following address:

Secretary
ASME Boiler and Pressure Vessel Committee
Two Park Avenue
New York, NY 10016-5990

All other Inquiries should be mailed to the Secretary of the BPV Committee at the address above. Inquiries are unlikely to receive a response if they are not written in clear, legible English. They must also include the name of the Inquirer and the company they represent or are employed by, if applicable, and the Inquirer's address, telephone number, fax number, and e-mail address, if available.

(b) Response. The Secretary of the appropriate Committee will provide a written response, via letter or e-mail, as appropriate, to the Inquirer, upon completion of the requested action by the Committee. Inquirers may track the status of their Interpretation Request at http://go.asme.org/Interpretations.

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(21)

January 1, 2021

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SUMMARY OF CHANGES

Errata to the BPV Code may be posted on the ASME website to provide corrections to incorrectly published items, or to correct typographical or grammatical errors in the BPV Code. Such Errata shall be used on the date posted.

Information regarding Special Notices and Errata is published by ASME at http://go.asme.org/BPVCerrata.

Changes given below are identified on the pages by a margin note, **(21)**, placed next to the affected area.

Page	*Location*	*Change*
xvii	List of Sections	*(1)* Listing for Section III updated *(2)* Section XIII added *(3)* Code Case information updated
xix	Foreword	*(1)* Subparagraph (k) added and subsequent subparagraph redesignated *(2)* Second footnote revised *(3)* Last paragraph added
xxii	Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees	Paragraphs 1(a)(3)(-b), 2(b), and 5(a)(3) revised
xxv	Personnel	Updated
1	1.1	Paragraphs 1.1.1.1, 1.1.2.1(i), and 1.1.2.3 revised
4	1.2.7	Revised
5	Table 1.1	Revised
8	1-B.2.16	Revised
15	2.2	Former 2.2.2.1 and 2.2.2.2 redesignated as 2.2.1.1 and 2.2.1.2, respectively
15	2.2.3.1	Subparagraphs (f)(4)(-a), (f)(4)(-b), and (i)(2)(-b) revised
17	2.3.1	Paragraph 2.3.1.3 added and subsequent paragraph redesignated
18	2.3.3	*(1)* In 2.3.3.1, subpara. (b) revised *(2)* In 2.3.3.3, former subparagraphs below (d) moved under subpara. (c)
19	2.3.7	Revised in its entirety
21	Annex 2-A	*(1)* In 2-A.1(b), first cross-reference updated *(2)* In 2-A.2.1, first paragraph, cross-references updated
23	2-B.3	Revised
28	Table 2-D.1	For Note (56), instructions revised
31	Form A-1	Line 18 revised
34	Form A-1P	Lines 9, 12, and 13 revised
36	Form A-2	Line 20 revised
42	2-E.1	In 2-E.1.1 and 2-E.1.4, last sentence deleted
44	2-E.16	Revised in its entirety

Page	*Location*	*Change*
46	2-F.4.1	Last paragraph added
46	2-F.5	Subparagraph (c)(1) revised in its entirety
48	2-F.7.1	Revised
49	Figure 2-F.1	Under "Certified by," text below fourth line revised
50	2-G.1	First line revised
50	2-G.6.3	Deleted
53	Annex 2-J	*(1)* Title revised *(2)* Paragraphs 2-J.1, 2-J.2(a), 2-J.3.1(b), 2-J.3.2, 2-J.3.3, 2-J.4.1(b), and 2-J.4.2(a) revised
57	3.2.5.2	Subparagraph (b)(4)(-b) revised
63	3.3.4.1	First paragraph and subpara. (b) revised
66	3.4.4.1	Last sentence deleted
76	3.11.2.1	In subpara. (a), last sentence deleted
78	3.11.2.5	*(1)* In subpara. (a), Step 4, second sentence added *(2)* In supbara. (a), Step 5(b), last two sentences revised *(3)* Subparagraph (e) added
79	3.11.2.6	In subpara. (a), last sentence added
81	3.11.4.1	Subparagraphs (c) and (c)(1) revised
83	3.11.4.5	Subparagraphs (a) and (d)(1) revised
85	3.11.7.6	Subparagraphs (b)(1)(-a) and (b)(2) revised
86	3.11.8.1	In subpara. (b), third cross-reference updated
115	Figure 3.10	Revised and Note (2) added
124	Table 3-A.1	Material Specification SA/NF A36-215 deleted
130	Table 3-A.3	For SA-249, UNS No. S31266 added below TP310S
145	Annex 3-C	Deleted
146	3-D.1	Revised
146	3-D.3	Equation (3-D.2) added and subsequent equations renumbered
147	3-D.4	Equations renumbered and cross-references to equations updated
148	3-D.5.1	Equations renumbered and cross-reference to equation updated
170	4.1.5.3	In subpara. (b), cross-references to ASCE/SEI 7 updated
172	4.1.11.1	Revised
174	Table 4.1.2	In Note (4), cross-reference to ASCE/SEI 7 updated
244	4.4.12.3	In subpara. (a), t corrected by errata to t_c
257	4.5.2.1	Last sentence deleted
258	4.5.5.1	Step 9 and eqs. (4.5.32) and (4.5.58) revised
263	4.5.10.1	Equation (4.5.120) revised
268	4.5.12.1	Equation (4.5.164) editorially revised and eq. (4.5.164a) added
272	4.5.15	Revised

Page	*Location*	*Change*
274	4.5.17	*(1)* Title revised *(2)* In 4.5.17.2, last sentence added *(3)* Paragraph 4.5.17.4 merged with 4.5.17.3 *(4)* Former 4.5.17.5 redesignated as 4.5.17.4 and revised
274	4.5.18	Definitions of A_5 and S revised
279	Figure 4.5.2	Legend revised
300	4.7.1.4	Added
303	4.7.5.3	Equation (4.7.32) revised
306	Table 4.7.1	Under "Operating Conditions" and "Gasket Seating Conditions" equations for S_{hlbi} and S_{hlbo} revised
318	4.11.1.3	Revised
319	4.11.4	Paragraph 4.11.4.1 revised in its entirety, 4.11.4.2 added, and subsequent paragraph redesignated
320	4.11.6.4	Equation (4.11.5) revised
320	4.11.7	Definition of C_{ul} added
327	Table 4.11.2	For Detail 1, requirements revised
400	4.15.3.1	Subparagraph (c) revised and eqs. (4.15.1) and (4.15.2) deleted
401	4.15.3.2	Equations renumbered and revised
401	4.15.3.3	Equations renumbered and their cross-references updated
402	4.15.3.4	*(1)* Equations renumbered and their cross-references updated *(2)* In subpara. (d)(2), eqs. (4.15.15) and (4.15.16) [former eqs. (4.15.17) and (4.15.18)] revised
402	4.15.3.5	*(1)* Equations renumbered and their cross-references updated *(2)* Subparagraph (c)(3) revised in its entirety
406	4.15.5.2	Subparagraph (b) revised
406	4.15.6	*(1)* Definitions of h_2 and R_i deleted *(2)* Definitions of h_m and R_{mh} added
409	Figure 4.15.1	Revised
410	Figure 4.15.2	Revised
411	Figure 4.15.3	Revised
418	4.16.4.3	Subparagraphs (b) and (c) deleted by errata
421	4.16.13	Definitions of h_n and h_p added
427	Table 4.16.4	Equation for X_h revised for both flange type
431	Table 4.16.7	Revised
443	4.17.3.5	Metric value revised
449	Table 4.17.1	For Flange, last row of equations revised
456	4.18.7.3	Subparagraph (a)(2) revised in its entirety
456	4.18.7.4	In Step 9(a), in-line equation revised
460	4.18.8.3	Subparagraph (a)(2) revised

Page	*Location*	*Change*
461	4.18.8.4	*(1)* In Step 8(a), in-line equation revised *(2)* In Step 9(c)(2), cross-references updated
473	4.18.9.1	Subparagraph (d) added
473	4.18.9.3	Subparagraph (b)(2) revised
474	4.18.9.4	*(1)* In Step 8(a), in-line equation revised *(2)* In Step 9(c)(2), cross-references updated
482	4.18.10	Deleted
485	4.18.15	Subparagraph (e) deleted
505	Figure 4.18.13	Deleted
519	4.19.8.6	In eq. (4.19.62), $\Delta q_{e,1}$> corrected by errata to $\Delta q_{e,1}$
544	Form 4.19.1	Line 24 and Note (1) revised
545	Form 4.19.2	Line 24 and Note (1) revised
550	4.21	Added
558	Figure 4.21.1	Added
559	Figure 4.21.2	Added
560	Figure 4.21.3	Added
565	Annex 4-C	Deleted
580	5.1.3.2	Cross-references to ASCE/SEI 7 updated
609	Table 5.3	In Note (2), cross-reference to ASCE/SEI 7 updated
699	6.2.4.9	Subparagraph (b) revised
705	6.4.5.2	Subparagraph (f)(1) revised in its entirety
714	6.7.6.3	Subparagraph (b) revised in its entirety
717	6.7.12	Revised
719	6.8.10	In subpara. (b)(3), cross-reference to table updated
721	Table 6.1	Second column head revised
724	Table 6.5	Revised
724	Table 6.6	Second column head revised
726	Table 6.8	In subpara. (a)(3)(-e) revised
727	Table 6.9	In subparas. (b)(1), (c)(4), (d), and (e) revised
728	Table 6.10	In subpara. (c), cross-reference updated
729	Table 6.11	In subparas. (b) and (c), cross-reference updated
731	Table 6.12	In subpara. (b), cross-reference updated
732	Table 6.13	For Materials P-No. 7, cross-reference updated in subpara. (b)
733	Table 6.14	*(1)* For Materials P-No. 9A, subparas. (a)(2), (b)(5), (c), and (e) revised *(2)* For Materials P-No. 9B, subparas. (a)(1), (b)(3), (d), and (f) revised
735	Table 6.15	Revised

Page	*Location*	*Change*
738	Table 6.17	Under Plate Steels, thickness revised for SA-553
764	7.5.5.2	In first paragraph and subpara. (c), second cross-reference updated
764	7.5.5.3	Cross-references to figures updated
768	7.6.1	In second line, "(if nonferromagnetic)" corrected by errata to "(if ferromagnetic)"
769	Table 7.1	Note (3) revised
770	Table 7.2	*(1)* Under Type of Weld, entries for Joint Categories D and E revised *(2)* Former Note (11) deleted, and subsequent Note renumbered
778	Table 7.10	General Note (f) revised and general notes (g) and (h) added
779	Table 7.11	General Note (e) revised and general notes (f) through (i) added
788	Figure 7.11	Revised and Note (1) added
789	Figure 7.12	Added
791	Figure 7.13	Added
793	Figure 7.14	Former Figure 7.12 redesignated
794	Figure 7.15	Former Figure 7.13 redesignated
795	Figure 7.16	Former Figure 7.14 redesignated
796	Figure 7.17	*(1)* Former Figure 7.15 redesignated *(2)* General Note added and Note (3) revised in its entirety
797	Figure 7.18	Former Figure 7.16 redesignated
797	Figure 7.19	Former Figure 7.17 redesignated
800	Table 7-A.1	For "Certification of qualification of nondestructive radiographic, ultrasonic, magnetic particle, liquid penetrant, and eddy current test examiners," paragraph reference for Procedure, corrected by errata from "7-A.3.2.4" to "7.3"
804	8.1.3.3	In subpara. (b), cross-reference to equation number corrected by errata from "(8.2)" to "(8.1)"
805	8.1.5	Added
808	Part 9	Revised in its entirety
813	9-A.2	Revised in its entirety
817	Annex 9-B	Added

LIST OF CHANGES IN RECORD NUMBER ORDER

DELETED

CROSS-REFERENCING AND STYLISTIC CHANGES IN THE BOILER AND PRESSURE VESSEL CODE

There have been structural and stylistic changes to BPVC, starting with the 2011 Addenda, that should be noted to aid navigating the contents. The following is an overview of the changes:

Subparagraph Breakdowns/Nested Lists Hierarchy

- First-level breakdowns are designated as (a), (b), (c), etc., as in the past.
- Second-level breakdowns are designated as (1), (2), (3), etc., as in the past.
- Third-level breakdowns are now designated as (-a), (-b), (-c), etc.
- Fourth-level breakdowns are now designated as (-1), (-2), (-3), etc.
- Fifth-level breakdowns are now designated as (+a), (+b), (+c), etc.
- Sixth-level breakdowns are now designated as (+1), (+2), etc.

Footnotes

With the exception of those included in the front matter (roman-numbered pages), all footnotes are treated as endnotes. The endnotes are referenced in numeric order and appear at the end of each BPVC section/subsection.

Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees

Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees has been moved to the front matter. This information now appears in all Boiler Code Sections (except for Code Case books).

Cross-References

It is our intention to establish cross-reference link functionality in the current edition and moving forward. To facilitate this, cross-reference style has changed. Cross-references within a subsection or subarticle will not include the designator/identifier of that subsection/subarticle. Examples follow:

- *(Sub-)Paragraph Cross-References.* The cross-references to subparagraph breakdowns will follow the hierarchy of the designators under which the breakdown appears.
  - If subparagraph (-a) appears in X.1(c)(1) and is referenced in X.1(c)(1), it will be referenced as (-a).
  - If subparagraph (-a) appears in X.1(c)(1) but is referenced in X.1(c)(2), it will be referenced as (1)(-a).
  - If subparagraph (-a) appears in X.1(c)(1) but is referenced in X.1(e)(1), it will be referenced as (c)(1)(-a).
  - If subparagraph (-a) appears in X.1(c)(1) but is referenced in X.2(c)(2), it will be referenced as X.1(c)(1)(-a).
- *Equation Cross-References.* The cross-references to equations will follow the same logic. For example, if eq. (1) appears in X.1(a)(1) but is referenced in X.1(b), it will be referenced as eq. (a)(1)(1). If eq. (1) appears in X.1(a)(1) but is referenced in a different subsection/subarticle/paragraph, it will be referenced as eq. X.1(a)(1)(1).

PART 1
GENERAL REQUIREMENTS

1.1 GENERAL (21)

1.1.1 INTRODUCTION

1.1.1.1 This Division contains mandatory requirements, specific prohibitions, and nonmandatory guidance for the design, materials, fabrication, examination, inspection, testing, overpressure protection, and certification of pressure vessels.

1.1.1.2 The Code does not address all aspects of these activities. Those aspects that are not specifically addressed should not be considered prohibited and shall be addressed by appropriate engineering judgment. Engineering judgment shall be consistent with the philosophy of this Division, and such judgments shall never be used to overrule mandatory requirements or specific prohibitions of this Division.

1.1.2 ORGANIZATION

1.1.2.1 The requirements of this Division are contained in the nine Parts listed below. Each of these Parts and Annexes is composed of paragraphs that are identified by an alphanumeric numbering system in accordance with the ISO Standard Template for the Preparation of Normative-Type Documents. References to paragraphs are made directly by reference to the paragraph number. For example, the Scope is referenced as 1.2.

(a) Part 1 – General Requirements, provides the scope of this division and establishes the extent of coverage
(b) Part 2 – Responsibilities and Duties, sets forth the responsibilities of the user and Manufacturer, and the duties of the Inspector
(c) Part 3 – Materials Requirements, provides the permissible materials of construction, applicable material specification and special requirements, physical properties, allowable stresses, and design fatigue curves
(d) Part 4 – Design by Rule Requirements, provides requirements for design of vessels and components using rules
(e) Part 5 – Design by Analysis Requirements, provides requirements for design of vessels and components using analytical methods
(f) Part 6 – Fabrication Requirements, provides requirements governing the fabrication of vessels and parts
(g) Part 7 – Examination and Inspection Requirements, provides requirements governing the examination and inspection of vessels and parts
(h) Part 8 – Pressure Testing Requirements, provides pressure testing requirements
(i) Part 9 – Pressure Vessel Overpressure Protection, provides overpressure protection requirements

1.1.2.2 Mandatory and nonmandatory requirements are provided as normative and informative annexes, respectively, to the specific Part under consideration. The Normative Annexes address specific subjects not covered elsewhere in this Division and their requirements are mandatory when the subject covered is included in construction under this Division. Informative Annexes provide information and suggested good practices.

1.1.2.3 The materials, design, fabrication, examination, inspection, testing, overpressure protection, and certification of pressure vessels shall satisfy all applicable Parts and Normative Annexes shown above in order to qualify the construction in accordance with this Division.

1.1.3 DEFINITIONS

The definitions for the terminology used in this Part are contained in Annex 1-B.

1.2 SCOPE

1.2.1 OVERVIEW

1.2.1.1 In the scope of this Division, pressure vessels are containers for the containment of pressure, either internal or external. This pressure may be obtained from an external source or by the application of heat from a direct or indirect source as a result of a process, or any combination thereof.

1.2.1.2 Vessels with an internal or external design pressure not exceeding 103 kPa (15 psi) and multichambered vessels of which the design pressure on the common elements does not exceed 103 kPa (15 psi) were not considered when the rules of this Division were developed and are not considered within the scope.

1.2.1.3 The rules of this Division may be used for the construction of the following pressure vessels. These vessels shall be designated as either a Class 1 or Class 2 vessel in conformance with the User's Design Specification required in Part 2.

(a) Vessels to be installed at a fixed (stationary) location for a specific service where operation and maintenance control is retained during the useful life of the vessel by the user and is in conformance with the User's Design Specification required by Part 2.

(b) Pressure vessels installed in ocean-going ships, barges, and other floating craft or used for motor vehicle or rail freight. For these applications it is necessary that prior written agreement with the jurisdictional authority be established covering operation and maintenance control for a specific service. This operation and maintenance control must be retained during the useful life of the pressure vessel by the user in conformance with the User's Design Specification required in Part 2. Such a pressure vessel as described above may be constructed and stamped within the scope of this Division, provided it meets all other requirements as specified with the following additional provisions.

(1) Loading conditions imposed by movement of the pressure vessel during operation and by relocation of the pressure vessel between work sites or due to loading and discharge, as applicable, shall be considered in the design.

(2) The User's Design Specification shall include the agreements that define those aspects of operation and maintenance control unique to the particular pressure vessel.

(c) Pressure vessels or parts subject to direct firing from the combustion of fuel (solid, liquid, or gaseous), that are not within the scope of Section I, III, or IV may be constructed in accordance with the rules of this Division.

(d) Unfired steam boilers shall be constructed in accordance with the rules of Section I or Section VIII, Division 1.

(e) The following pressure vessels in which steam is generated shall be constructed in accordance with the rules of Section VIII, Division 1 or this Division:

(1) Vessels known as evaporators or heat exchangers;

(2) Vessels in which steam is generated by the use of heat resulting from operation of a processing system containing a number of pressure vessels such as used in the manufacture of chemical and petroleum products; and

(3) Vessels in which steam is generated but not withdrawn for external use.

1.2.1.4 The scope of this Division has been established to identify components and parameters considered in formulating the rules given in this Division. Laws or regulations issued by municipality, state, provincial, federal, or other enforcement or regulatory bodies having jurisdiction at the location of an installation establish the mandatory applicability of the Code rules, in whole or in part, within the jurisdiction. Those laws or regulations may require the use of this Division of the Code for vessels or components not considered to be within its scope. These laws or regulations should be reviewed to determine size or service limitations of the coverage which may be different or more restrictive than those given here.

1.2.2 ADDITIONAL REQUIREMENTS FOR VERY HIGH PRESSURE VESSELS

1.2.2.1 The rules of this Division do not specify a limitation on pressure but are not all-inclusive for all types of construction. For very high pressures, some additions to these rules may be necessary to meet the design principles and construction practices essential to vessels for such pressures. However, only in the event that, after application of additional design principles and construction practices, the vessel still complies with all of the requirements of the Code, may it be stamped with the Certification Mark.

1.2.2.2 As an alternative to this Division, Section VIII, Division 3 should be considered for the construction of vessels intended for operating pressures exceeding 68.95 MPa (10,000 psi).

1.2.3 GEOMETRIC SCOPE OF THIS DIVISION

The scope of this Division is intended to include only the vessel and integral communicating chambers, and shall include the following:

(a) Where external piping, other pressure vessels including heat exchangers, or mechanical devices (i.e., pumps, mixers, or compressors) are to be connected to the vessel:

(1) The welding end connection for the first circumferential joint for welded connections (see 4.2.5.9).

(2) The first threaded joint for screwed connections.

(3) The face of the first flange for bolted and flanged connections. Optionally, when the first flange is welded to the nozzle neck, the weld connecting the flange to the nozzle neck may be considered as the first circumferential joint, provided this construction is documented in the User's Design Specification and is properly described on the vessel drawing and the Manufacturer's Data Report Form.

(4) The first sealing surface for proprietary connections or fittings.

(b) Where non-pressure parts are welded directly to either the internal or external pressure-retaining surface of a pressure vessel, the scope of this Division shall include the design, fabrication, testing, and material requirements established for non-pressure-part attachments by the applicable paragraphs of this Division (see 4.2.5.6).

(c) Pressure-retaining covers and their fasteners (bolts and nuts) for vessel openings, such as manhole and handhole covers.

(d) The first sealing surface for proprietary connections, fittings or components that are designed to rules that are not provided by this Division, such as gages, instruments, and nonmetallic components.

1.2.4 CLASSIFICATIONS OUTSIDE THE SCOPE OF THIS DIVISION

1.2.4.1 The scope of this Division has been established to identify the components and parameters considered in formulating the rules given in this Division. Laws or regulations issued by a Jurisdictional Authority at the location of an installation establish the mandatory applicability of the Code rules, in whole or in part, within that jurisdiction. Those laws or regulations may require the use of this Division of the Code for vessels or components not considered to be within its Scope. These laws or regulations should be reviewed to determine size or service limitations that may be more restrictive than those given here.

1.2.4.2 The following vessels are not included in the scope of this Division. However, any pressure vessel, with the exception of (a) below, that is not excluded from the scope of this Division by 1.2.1.3 and that meets all applicable requirements of this Division may be stamped with the Certification Mark with the U2 Designator and vessel class.

(a) Vessels within the scope of other Sections.

(b) Fired process tubular heaters as defined in API RP560.

(c) Pressure containers that are integral parts or components of rotating or reciprocating mechanical devices, such as pumps, compressors, turbines, generators, engines, and hydraulic or pneumatic cylinders where the primary design considerations and/or stresses are derived from the functional requirements of the device.

(d) Structures consisting of piping components, such as pipe, flanges, bolting, gaskets, valves, expansion joints, and fittings whose primary function is the transport of fluids from one location to another within a system of which it is an integral part, that is, piping systems, including the piping system between a pressure relief device and the vessel it protects, see Part 9.

(e) Pressure-containing parts of components, such as strainers and devices that serve such purposes as mixing, separating, snubbing, distributing, and metering or controlling flow, provided that pressure-containing parts of such components are generally recognized as piping components or accessories.

(f) A vessel for containing water under pressure, including those containing air the compression of which serves only as a cushion, when none of the following limitations are exceeded:

(1) A design pressure of 2.07 MPa (300 psi)

(2) A design temperature of 99°C (210°F)

(g) A hot water supply storage tank heated by steam or any other indirect means when none of the following limitations is exceeded:

(1) A heat input of 58.6 kW (200,000 Btu/hr)

(2) A water temperature of 99°C (210°F)

(3) A nominal water containing capacity of 454 L (120 gal)

(h) Vessels with an internal or external design pressure not exceeding 103 kPa (15 psi) with no limitation on size, for multi-chambered vessels, the design pressure on the common elements shall not exceed 103 kPa (15 psi).

(i) Vessels with an inside diameter, width, height, or cross section diagonal not exceeding 150 mm (6 in.), with no limitation on length of vessel or pressure.

(j) Pressure vessels for human occupancy (requirements for pressure vessels for human occupancy are covered in ASME PVHO-1).

1.2.5 COMBINATION UNITS

When a pressure vessel unit consists of more than one pressure chamber, only the chambers that come within the scope of this Division need be constructed in compliance with its provisions (see 4.1.8).

1.2.6 FIELD ASSEMBLY OF VESSELS

1.2.6.1 Field assembly of vessels constructed to this Division may be performed as follows.

(a) The Manufacturer of the vessel completes the vessel in the field, completes the Form A-1 or Form A-1P Manufacturer's Data Report, and stamps the vessel.

(b) The Manufacturer of parts of a vessel to be completed in the field by some other party stamps these parts in accordance with Code rules and supplies the Form A-2 Manufacturer's Partial Data Report to the other party. The other party, who must hold a valid U2 Certificate of Authorization, makes the final assembly, performs the required NDE, performs the final pressure test, completes the Form A-1 or Form A-1P Manufacturer's Data Report, and stamps the vessel.

(c) The field portion of the work is completed by a holder of a valid U2 Certificate of Authorization other than the vessel Manufacturer. The Certificate holder performing the field work is required to supply a Form A-2 Manufacturer's Partial Data Report covering the portion of the work completed by his organization (including data on the pressure test if conducted by the Certificate holder performing the field work) to the Manufacturer responsible for the Code vessel. The vessel Manufacturer applies his Certification Mark with U2 Designator in the presence of a representative from his Inspection Agency and completes the Form A-1 or Form A-1P Manufacturer's Data Report with his Inspector.

1.2.6.2 In all three alternatives, the party completing and signing the Form A-1 or Form A-1P Manufacturer's Data Report assumes full Code responsibility for the vessel. In all three cases, each Manufacturer's Quality Control System shall describe the controls to assure compliance by each Certificate holder.

(21) 1.2.7 OVERPRESSURE PROTECTION

The scope of this Division includes provisions for overpressure protection necessary to satisfy the requirements of Part 9.

1.3 STANDARDS REFERENCED BY THIS DIVISION

(a) Throughout this Division, references are made to various standards, such as ASME standards, which describe parts or fittings or which establish dimensional limits for pressure vessel parts. These standards, with the year of the acceptable edition, are listed in Table 1.1.

(b) Rules for the use of these standards are stated elsewhere in this Division.

1.4 UNITS OF MEASUREMENT

(a) Either U.S. Customary, SI, or any local customary units may be used to demonstrate compliance with requirements of this edition related to materials, fabrication, examination, inspection, testing, certification, and overpressure protection.

(b) A single system of units shall be used for all aspects of design except where otherwise permitted by this Division. When components are manufactured at different locations where local customary units are different than those used for the general design, the local units may be used for the design and documentation of that component within the limitations given in (c). Similarly, for proprietary components or those uniquely associated with a system of units different than that used for the general design, the alternate units may be used for the design and documentation of that component within the limitations given in (c).

(c) For any single equation, all variables shall be expressed in a single system of units. Calculations using any material data published in this Division or Section II, Part D (e.g., allowable stresses, physical properties, external pressure design factor B) shall be carried out in one of the standard units given in Table 1.2. When separate equations are provided for U.S. Customary and SI units, those equations shall be executed using variables in the units associated with the specific equation. Data expressed in other units shall be converted to U.S. Customary or SI units for use in these equations. The result obtained from execution of these equations or any other calculations carried out in either U.S. Customary or SI units may be converted to other units.

(d) Production, measurement and test equipment, drawings, welding procedure specifications, welding procedure and performance qualifications, and other fabrication documents may be in U.S. Customary, SI or local customary units in accordance with the fabricator's practice. When values shown in calculations and analysis, fabrication documents or measurement and test equipment are in different units, any conversions necessary for verification of Code compliance and to ensure that dimensional consistency is maintained shall be in accordance with the following:

(1) Conversion factors shall be accurate to at least four significant figures

(2) The results of conversions of units shall be expressed to a minimum of three significant figures

(e) Conversion of units, using the precision specified above shall be performed to assure that dimensional consistency is maintained. Conversion factors between U.S. Customary and SI units may be found in Annex 1-C. Whenever local customary units are used the Manufacturer shall provide the source of the conversion factors which shall be subject to verification and acceptance by the Authorized Inspector or Certified Individual.

(f) Dimensions shown in the text, tables and figures, whether given as a decimal or a fraction, may be taken as a decimal or a fraction and do not imply any manufacturing precision or tolerance on the dimension.

(g) Material that has been manufactured and certified to either the U.S. Customary or SI material specification (e.g., SA-516 or SA-516M) may be used regardless of the unit system used in design. Standard fittings (e.g., flanges, elbows, etc.) that have been certified to either U.S. Customary units or SI units may be used regardless of the units system used in design.

(h) All entries on a Manufacturer's Data Report and data for Code-required nameplate marking shall be in units consistent with the fabrication drawings for the component using U. S. Customary, SI, or local customary units. Units (either primary or alternative) may be shown parenthetically. Users of this Code are cautioned that the receiving Jurisdiction should be contacted to ensure the units are acceptable.

1.5 TOLERANCES

The Code does not fully address tolerances. When dimensions, sizes, or other parameters are not specified with tolerances, the values of these parameters are considered nominal, and allowable tolerances or local variances may be considered acceptable when based on engineering judgment and standard practices as determined by the designer.

1.6 TECHNICAL INQUIRIES

A procedure for submittal of Technical Inquiries to the ASME Boiler and Pressure Vessel Code Committee is contained in the front matter.

1.7 TABLES

Table 1.1
Year of Acceptable Edition of Referenced Standards in This Division (21)

Title	Number	Year
Marking and Labeling Systems	ANSI/UL-969	Latest edition
Fitness-For-Service	API 579-1/ASME FFS-1	2016
Materials and Fabrication of $2\frac{1}{4}$Cr–1Mo, $2\frac{1}{4}$Cr–1Mo–$\frac{1}{4}$V, 3Cr–1Mo, and 3Cr–1Mo–$\frac{1}{4}$V Steel Heavy Wall Pressure Vessels for High-Temperature, High-Pressure Hydrogen Service	API RP 934-A	2019
Fired Heaters for General Refinery Service	API Standard 560	Latest edition
Minimum Design Loads and Associated Criteria for Buildings and Other Structures	ASCE/SEI 7	2016
Nuts for General Applications: Machine Screw Nuts, Hex, Square, Hex Flange, and Coupling Nuts (Inch Series)	ASME/ANSI B18.2.2	Latest edition
Unified Inch Screw Threads (UN and UNR Thread Form)	ASME B1.1	Latest edition
Metric Screw Threads — M Profile	ASME B1.13M	Latest edition
Pipe Threads, General Purpose, Inch	ASME B1.20.1	Latest edition
Metric Screw Threads — MJ Profile	ASME B1.21M	Latest edition
Pipe Flanges and Flanged Fittings, NPS $\frac{1}{2}$ Through NPS 24 Metric/Inch Standard	ASME B16.5	2017
Factory-Made Wrought Buttwelding Fittings	ASME B16.9	Latest edition
Forged Fittings, Socket-Welding and Threaded	ASME B16.11	Latest edition
Cast Copper Alloy Threaded Fittings, Classes 125 and 250	ASME B16.15	Latest Edition
Metallic Gaskets for Pipe Flanges	ASME B16.20	Latest edition
Cast Copper Alloy Pipe Flanges, Flanged Fittings, and Valves, Classes 150, 300, 600, 900, 1500, and 2500	ASME B16.24	2016
Large Diameter Steel Flanges, NPS 26 Through NPS 60 Metric/Inch Standard	ASME B16.47	2017
Metric Heavy Hex Screws	ASME B18.2.3.3M	Latest edition
Metric Hex Bolts	ASME B18.2.3.5M	Latest edition
Metric Heavy Hex Bolts	ASME B18.2.3.6M	Latest edition
Metric Fasteners for Use in Structural Applications	ASME B18.2.6M	Latest edition
Conformity Assessment Requirements	ASME CA-1	Latest edition
Guidelines for Pressure Boundary Bolted Flange Joint Assembly	ASME PCC-1	2019
Repair of Pressure Equipment and Piping	ASME PCC-2	2018
Qualifications for Authorized Inspection	ASME QAI-1	Latest edition
Standard Practice for Quantitative Measurement and Reporting of Hypoeutectoid Carbon and Low-Alloy Steel Phase Transformations	ASTM A1033	Latest edition

Table 1.1
Year of Acceptable Edition of Referenced Standards in This Division (Cont'd)

Title	Number	Year
Standard Reference Photographs for Magnetic Particle Indications on Ferrous Castings	ASTM E125	1963 (R2018) [Note (1)]
Standard Practice for Fabricating and Checking Aluminum Alloy Ultrasonic Standard Reference Blocks	ASTM E127	2019
Standard Test Methods for Conducting Creep, Creep-Rupture, and Stress-Rupture Tests of Metallic Materials	ASTM E139	Latest edition
Standard Hardness Conversion Tables for Metals Relationship Among Brinell Hardness, Vickers Hardness, Rockwell Hardness, Superficial Hardness, Knoop Hardness, and Scleroscope Hardness	ASTM E140	Latest edition
Standard Reference Radiographs for Heavy-Walled [2 to $4\frac{1}{2}$ in. (50.8 to 114 mm)] Steel Castings	ASTM E186	2015 [R2019] [Note (1)]
Standard Test Method of Conducting Drop Weight Test to Determine Nil Ductility Transition Temperature of Ferritic Steel	ASTM E208	2019
Standard Reference Radiographs for High-Strength Copper-Base and Nickel-Copper Alloy Castings	ASTM E272	2019
Standard Reference Radiographs for Heavy-Walled [$4\frac{1}{2}$ to 12 in. (114 to 305 mm)] Steel Castings	ASTM E280	2015 [R2019] [Note (1)]
Standard Reference Radiographs for Steel Castings up to 2 in. (51 mm) in Thickness	ASTM E446	2015
Standard Procedures for Calibrating Magnetic Instruments to Measure the Delta Ferrite Content of Austenitic and Duplex Austenitic-Ferrite Stainless Steel Weld Metal	AWS 4.2M	2006
Metallic materials — Charpy pendulum impact test — Part 1: Test method	ISO 148-1	2009
Metallic materials — Charpy pendulum impact test — Part 2: Verification of testing machines	ISO 148-2	2008
Metallic materials — Charpy pendulum impact test — Part 3: Preparation and characterization of Charpy V-notch test pieces for indirect verification of pendulum impact machines	ISO 148-3	2008
Petroleum, petrochemical and natural gas industries — Fired heaters for general refinery service	ISO 13705	Latest edition
Standard Practice for Ultrasonic Examination of Steel Forgings	SA-388/SA-388M	Latest edition

NOTE:
(1) "R" indicates reaffirmed.

Table 1.2
Standard Units for Use in Equations

Quantity	SI Units	U.S. Customary Units
Linear dimensions (e.g., length, height, thickness, radius, diameter)	millimeters (mm)	inches (in.)
Area	square millimeters (mm^2)	square inches ($in.^2$)
Volume	cubic millimeters (mm^3)	cubic inches ($in.^3$)
Section modulus	cubic millimeters (mm^3)	cubic inches ($in.^3$)
Moment of inertia of section	millimeters^4 (mm^4)	inches^4 ($in.^4$)
Mass (weight)	kilograms (kg)	pounds mass (lbm)
Force (load)	newtons (N)	pounds force (lbf)
Bending moment	newton-millimeters (N·mm)	inch-pounds (in.-lb)
Pressure, stress, stress intensity, and modulus of elasticity	megapascals (MPa)	pounds per square inch (psi)
Energy (e.g., Charpy impact values)	joules (J)	foot-pounds (ft-lb)
Temperature	degrees Celsius (°C)	degrees Fahrenheit (°F)
Absolute temperature	kelvin (K)	Rankine (°R)
Fracture toughness	MPa square root meters ($MPa\sqrt{m}$)	ksi square root inches ($ksi\sqrt{in.}$)
Angle	degrees or radians	degrees or radians
Boiler capacity	watts (W)	Btu/hr

ANNEX 1-B DEFINITIONS

(Normative)

1-B.1 INTRODUCTION

This Annex contains definitions of terms generally used in this Division. Definitions relating to specific applications may also be found in related Parts of this Division.

1-B.2 DEFINITION OF TERMS

1-B.2.1 Acceptance by the Inspector, accepted by the Inspector - an indication that the Inspector has reviewed a subject in accordance with his duties as required by the rules of this Division and after such review is able to sign the Certificate of Inspection for the applicable Manufacturer's Data Report Form.

1-B.2.2 ASME Designated Organization – see ASME CA-1.

1-B.2.3 ASME designee – see ASME CA-1.

1-B.2.4 Certificate of Compliance – a document that states that the material represented has been manufactured, sampled, tested and inspected in accordance with the requirements of the material specification (including year of issue) and any other requirements specified in the purchase order or contract shown on the certificate and has been found to meet such requirements. This document may be combined with the Materials Test Report (see 1-B.2.19) as a single document.

1-B.2.5 Certificate of Authorization – a document issued by the Society that authorizes the use of the ASME Certification Mark and appropriate designator for a specified time and for a specified scope of activity.

1-B.2.6 Certification Mark – an ASME symbol identifying a product as meeting Code requirements.

1-B.2.7 Certification Mark Stamp – a metallic stamp issued by the Society for use in impressing the Certification Mark.

1-B.2.8 Certification Designator (Designator) – the symbol used in conjunction with the Certification Mark for the scope of activity described in a Manufacturer's Certificate of Authorization.

1-B.2.9 Certifying Engineer – an engineer or other technically competent professional duly accredited and qualified to practice engineering as required by this Division.

1-B.2.10 Class 1 Vessel – a vessel that is designed using the allowable stresses from Section II, Part D, Subpart 1, Table 2A or Table 2B.

1-B.2.11 Class 2 Vessel – a vessel that is designed using the allowable stresses from Section II, Part D, Subpart 1, Table 5A or Table 5B.

1-B.2.12 Communicating Chambers – appurtenances to a vessel that intersect the shell or heads of a vessel and form an integral part of the pressure-containing enclosure.

1-B.2.13 Construction – an all-inclusive term comprising materials, design, fabrication, examination, inspection, testing, certification, and pressure relief.

1-B.2.14 Designer – an individual who is qualified to design pressure vessels in accordance with the rules of this Division by demonstrated knowledge in Code requirements and proficiency in selecting correct design formulas and appropriate values to be used when preparing the design of a pressure vessel.

1-B.2.15 Local Jurisdictional Authority – an agency enforcing laws or regulations applicable to pressure vessels.

(21) **1-B.2.16** Manufacturer – the organization responsible for construction of a pressure vessel, vessel component, or part in accordance with the rules of this Division and who holds an ASME Certificate of Authorization to apply the Certification Mark to such an item.

1-B.2.17 Material – any substance or product form covered by a material specification in Section II Part A, B, or C or any other substance or product form permitted for use in pressure vessel construction by this Division.

1-B.2.18 Material Manufacturer – the organization responsible for the production of products meeting the requirements of the material specification and accepting the responsibility for any statements or data in any required Certificate of Compliance or Material Test Report representing the material.

1-B.2.19 Material Test Report – a document in which the results of tests, examinations, repairs, or treatments required by the material specification to be reported are recorded, including those of any supplementary requirements or other requirements stated in the order for the material. This document may be combined with a Certificate of Compliance (see 1-B.2.4) as a single document.

1-B.2.20 User – the organization that purchases the finished pressure vessel for its own use or as an agent for the owner. The user's designated agent may be either a design agency specifically engaged by the user, the Manufacturer of a system for a specific service which includes a pressure vessel as a part and which is purchased by the user, or an organization which offers pressure vessels for sale or lease for specific services.

ANNEX 1-C
GUIDANCE FOR THE USE OF U.S. CUSTOMARY AND SI UNITS IN THE ASME BOILER AND PRESSURE VESSEL CODES

(Informative)

1-C.1 USE OF UNITS IN EQUATIONS

The equations in this Division are suitable for use only with either the SI or U.S. Customary units provided in Table 1.2 or with the units provided in the nomenclatures associated with the equations. It is the responsibility of the individual and organization performing the calculations to ensure that appropriate units are used. Either SI or U.S. Customary units may be used as a consistent set. When necessary to convert from one system to another, the units shall be converted to at least four significant figures for use in calculations and other aspects of construction.

1-C.2 GUIDELINES USED TO DEVELOP SI EQUIVALENTS

(a) U.S. Customary units are placed in parenthesis after the SI unit in the text.

(b) In general, both SI and U.S. Customary tables are provided if interpolation is expected. The table designation (e.g., table number) is the same for both the SI and the U.S. Customary tables, with the addition of an M after the table number for the SI Table. In the text, references to a Table use only the primary table number (i.e., without the M). For some small tables, where interpolation is not required, U.S. Customary units are placed in parenthesis after the SI unit.

(c) Separate SI and U.S. Customary versions of graphical information (charts) are provided, except that if both axes are dimensionless a single figure (chart) is used.

(d) In most cases, conversions of units in the text were done using hard SI conversion practices, with some soft conversions on a case-by-case basis as appropriate. This was implemented by rounding the SI values to the number of significant figures of implied precision in the existing U.S. Customary units. For example, 3000 psi has an implied precision of one significant figure. Therefore, the conversion to SI units would typically be to 20,000 kPa. This is a difference of about 3% from the "exact" or soft conversion of 20,684.27 kPa. However, the precision of the conversion was determined by the Committee on a case-by-case basis. More significant digits were included in the SI equivalent if there was any question. The values of allowable stress in Section II, Part D generally include three significant figures.

(e) Minimum thickness and radius values that are expressed in fractions of an inch were generally converted according to Table 1-C.1.

(f) For nominal sizes that are in even increments of inches, even multiples of 25 mm were generally used. Intermediate values were interpolated rather than converting and rounding to the nearest mm. See examples in Table 1-C.2. Note that this table does not apply to nominal pipe sizes (NPS), which are covered in Table 1-C.4.

(g) For nominal pipe sizes, the relationships shown in Table 1-C.4 were used.

(h) Areas in square inches (in.^{2}) were converted to square millimeters (mm^2), and areas in square feet (ft^2) were converted to square meters (m^2), see examples in Table 1-C.5.

(i) Volumes in cubic inches (in.^{3}) were converted to cubic millimeters (mm^3), and volumes in cubic feet (ft^3) were converted to cubic meters (m^3), see examples in the Table 1-C.6.

(j) Although the pressure should always be in MPa or psi for calculations, there are cases where other units are used in the text. For example, kPa is sometimes used for low pressures and ksi is sometimes used for high pressures and stresses. Also, rounding was to one significant figure (two at the most) in most cases, see examples in Table 1-C.7. Note that 14.7 psi converts to 101 kPa, while 15 psi converts to 100 kPa. While this may seem at first glance to be an anomaly, it is consistent with the rounding philosophy.

(k) Material properties that are expressed in psi or ksi (e.g., allowable stress, yield and tensile strength, elastic modulus) were generally converted to MPa to three significant figures. See example in Table 1-C.8.

(l) In most cases, temperatures (e.g., for PWHT) were rounded to the nearest 5°C. Depending on the implied precision of the temperature, some were rounded to the nearest 1°C or 10°C or even 25°C. Temperatures colder than 0°F (negative values) were generally rounded to the nearest 1°C. The examples in Table 1-C.9 were created by rounding to the nearest 5°C, with one exception.

1-C.3 SOFT CONVERSION FACTORS

Table 1-C.10 of "soft" conversion factors is provided for convenience. Multiply the U.S. Customary value by the factor given to obtain the SI value. Similarly, divide the SI value by the factor given to obtain the U.S. Customary value. In most cases it is appropriate to round the answer to three significant figures.

1-C.4 TABLES

Table 1-C.1
Typical Size or Thickness Conversions for Fractions

Fraction in U.S. Customary Units, in.	Proposed SI Conversion, mm	Difference, %
$^1/_{32}$	0.8	−0.8
$^3/_{64}$	1.2	−0.8
$^1/_{16}$	1.5	5.5
$^3/_{32}$	2.5	−5.0
$^1/_8$	3	5.5
$^5/_{32}$	4	−0.8
$^3/_{16}$	5	−5.0
$^7/_{32}$	5.5	1.0
$^1/_4$	6	5.5
$^5/_{16}$	8	−0.8
$^3/_8$	10	−5.0
$^7/_{16}$	11	1.0
$^1/_2$	13	−2.4
$^9/_{16}$	14	2.0
$^5/_8$	16	−0.8
$^{11}/_{16}$	17	2.6
$^3/_4$	19	0.3
$^7/_8$	22	1.0
1	25	1.6

Table 1-C.2
Typical Size or Thickness Conversions

Size, in.	Size, mm
1	25
$1^1/_8$	29
$1^1/_4$	32
$1^1/_2$	38
2	50
$2^1/_4$	57
$2^1/_2$	64
3	75
$3^1/_2$	89
4	100
$4^1/_2$	114

Table 1-C.2
Typical Size or Thickness Conversions (Cont'd)

Size, in.	Size, mm
5	125
6	150
8	200
12	300
18	450
20	500
24	600
36	900
40	1 000
54	1 350
60	1 500
72	1 800

Table 1-C.3
Typical Size or Length Conversions

Size or Length, ft	Size or Length, m
3	1
5	1.5
200	60

Table 1-C.4
Typical Nominal Pipe Size Conversions

U.S. Customary Practice	SI Practice	U. S. Customary Practice	SI Practice
NPS $^1/_8$	DN 6	NPS 20	DN 500
NPS $^1/_4$	DN 8	NPS 22	DN 550
NPS $^3/_8$	DN 10	NPS 24	DN 600
NPS $^1/_2$	DN 15	NPS 26	DN 650
NPS $^3/_4$	DN 20	NPS 28	DN 700
NPS 1	DN 25	NPS 30	DN 750
NPS $1^1/_4$	DN 32	NPS 32	DN 800
NPS $1^1/_2$	DN 40	NPS 34	DN 850
NPS 2	DN 50	NPS 36	DN 900
NPS $2^1/_2$	DN 65	NPS 38	DN 950
NPS 3	DN 80	NPS 40	DN 1 000
NPS $3^1/_2$	DN 90	NPS 42	DN 1 050
NPS 4	DN 100	NPS 44	DN 1 100
NPS 5	DN 125	NPS 46	DN 1 150
NPS 6	DN 150	NPS 48	DN 1 200
NPS 8	DN 200	NPS 50	DN 1 250
NPS 10	DN 250	NPS 52	DN 1 300
NPS 12	DN 300	NPS 54	DN 1 350
NPS 14	DN 350	NPS 56	DN 1 400
NPS 16	DN 400	NPS 58	DN 1 450
NPS 18	DN 450	NPS 60	DN 1 500

Table 1-C.5
Typical Area Conversions

Area in U.S. Customary	Area in SI
1 in.^{2}	650 mm^2
6 in.^{2}	4 000 mm^2
10 in.^{2}	6 500 mm^2
5 ft^2	0.5 m^2

Table 1-C.6
Typical Volume Conversions

Volume in U.S. Customary	Volume in SI
1 in.^{3}	16 000 mm^3
6 in.^{3}	100 000 mm^3
10 in.^{3}	160 000 mm^3
5 ft^3	0.14 m^3

Table 1-C.7
Typical Pressure Conversions

Pressure in U.S. Customary	Pressure in SI
0.5 psi	3 kPa
2 psi	15 kPa
3 psi	20 kPa
10 psi	70 kPa
14.7 psi	101 kPa
15 psi	100 kPa
30 psi	200 kPa
50 psi	350 kPa
100 psi	700 kPa
150 psi	1 MPa
200 psi	1.5 MPa
250 psi	1.7 MPa
300 psi	2 MPa
350 psi	2.5 MPa
400 psi	3 MPa
500 psi	3.5 MPa
600 psi	4 MPa
1,200 psi	8 MPa
1,500 psi	10 MPa

Table 1-C.8
Typical Strength Conversions

Strength in U.S. Customary, psi	Strength in SI, MPa
30,000	205
38,000	260
60,000	415
70,000	480
95,000	655

Table 1-C.9
Typical Temperature Conversions

Temperature, °F	Temperature, °C
70	20
100	38
120	50
150	65
200	95
250	120
300	150
350	175
400	205
450	230
500	260
550	290
600	315
650	345
700	370
750	400
800	425
850	455
900	480
925	495
950	510
1,000	540
1,050	565
1,100	595
1,150	620
1,200	650
1,250	675
1,800	980
1,900	1 040
2,000	1 095
2,050	1 120

Table 1-C.10
Conversion Factors

U.S. Customary	SI	Conversion Factor	Notes
in.	mm	25.4	...
ft	m	0.3048	...
in.^{2}	mm^2	645.16	...
ft^2	m^2	0.09290304	...
in^3	mm^3	16,387.064	...
ft^3	m^3	0.02831685	...
US Gal.	m^3	0.003785412	...
psi	MPa	0.0068948	Used exclusively in equations
psi	kPa	6.894757	Used only in text and for nameplate
ft-lb	J	1.355818	...
°F	°C	$\frac{5}{9}$(°F − 32)	Not for temperature difference
°F	°C	$\frac{5}{9}$(°F)	For temperature differences only
R	K	$\frac{5}{9}$	Absolute temperature
lbm	kg	0.4535924	...
lbf	N	4.448222	...
in.-lb	N·mm	112.98484	Use exclusively in equations
ft-lb	N·m	1.3558181	Use only in text
ksi$\sqrt{in}$	MPa$\sqrt{m}$	1.0988434	...
Btu/hr	W	0.2930711	Use for Boiler rating and heat transfer
lb/ft^3	kg/m^3	16.018463	...

PART 2
RESPONSIBILITIES AND DUTIES

2.1 GENERAL

2.1.1 INTRODUCTION

The user, Manufacturer, and Inspector involved in the production and certification of vessels in accordance with this Division have definite responsibilities or duties in meeting the requirements of this Division. The responsibilities and duties set forth in the following relate only to compliance with this Division, and are not to be construed as involving contractual relations or legal liabilities.

2.1.2 DEFINITIONS

The definitions for the terminology used in this Part are contained in Annex 1-B.

2.1.3 CODE REFERENCE

The Code Edition year on the User's Design Specification and Manufacturer's Design Report shall be the same as the Code Edition year on the Manufacturer's Data Report.

2.2 USER RESPONSIBILITIES (21)

2.2.1 GENERAL

It is the responsibility of the user or an agent acting on behalf of the user to provide a User's Design Specification for each pressure vessel to be constructed in accordance with this Division. The User's Design Specification shall contain sufficient detail to provide a complete basis for design and construction in accordance with this Division. It is the user's responsibility to specify, or cause to be specified, the effective Code edition and vessel class to be used for construction.

2.2.1.1 Class 1. The User's Design Specification shall be certified by a Certifying Engineer meeting the requirements described in Annex 2-A when the user provides the data required by 2.2.3.1(f)(1) and 2.2.3.1(f)(2) to perform a fatigue analysis.

2.2.1.2 Class 2. The User's Design Specification shall be certified by a Certifying Engineer in accordance with Annex 2-A.

2.2.2 MULTIPLE IDENTICAL VESSELS

A single User's Design Specification may be prepared to support the design of more than one pressure vessel that is to be located in a single, specific jurisdiction provided that the environmental requirements and jurisdictional regulatory authority applied for each installation location are clearly specified and are the same or more conservative than required.

2.2.3 USER'S DESIGN SPECIFICATION

2.2.3.1 The User's Design Specification shall include but not necessarily be limited to the following: (21)

(a) Installation Site
  (1) Location
  (2) Jurisdictional authority if applicable
  (3) Environmental conditions
    (-a) Wind design loads including relevant factors (i.e., design wind speed, exposure, gust factors)
    (-b) Earthquake design loads
    (-c) Snow loads
    (-d) Lowest one day mean temperature for location

(b) Vessel Identification
  (1) Vessel number or identification

(2) Service fluid for proprietary fluids specific properties needed for design, e.g., gas, liquid, density, etc.

(c) Vessel Configuration and Controlling Dimensions

(1) Outline drawings

(2) Vertical or horizontal

(3) Openings, connections, closures including quantity, type and size, and location (i.e., elevation and orientation)

(4) Principal component dimensions in sufficient detail so that volume capacities can be determined

(5) Support method

(d) Design Conditions

(1) Specified design pressure. The specified design pressure is the design pressure, see 4.1.5.2(a), required at the top of the vessel in its operating position. It shall include suitable margins required above the maximum anticipated operating pressure to ensure proper operation of the pressure relief devices. The MAWP of the vessel may be set equal to this specified design pressure. If the actual MAWP of the vessel is calculated, it shall not be less than the specified design pressure.

(2) Design temperature and coincident specified design pressure (see 4.1.5.2(d)).

(3) Minimum Design Metal Temperature (MDMT) and coincident specified design pressure (see 4.1.5.2(e)).

(4) Dead loads, live loads, and other loads required to perform the load case combinations required in Parts 4 and 5.

(e) Operating Conditions

(1) Operating pressure and pressure load factor for occasional load combinations in Tables 4.1.2 and 5.3

(2) Operating temperature

(3) Fluid transients and flow and sufficient properties for determination of steady-state and transient thermal gradients across the vessel sections, if applicable (see 5.5.2)

(4) Dead loads, live loads, and other operating loads required to perform the load case combinations required in Part 5

(f) Design Fatigue Life

(1) Cyclic operating conditions and whether or not a fatigue analysis of the vessel as required shall be determined in accordance with 4.1.1.4. When a fatigue analysis is required, provide information in sufficient detail so that an analysis of the cyclic operation can be carried out in accordance with 5.5.

(2) When a vessel is designed for cyclic conditions, the number of design cycles per year and the required vessel design life in years shall be stated.

(3) When cyclic operating conditions exist and a fatigue analysis is not required based on comparable equipment experience, this shall be stated. The possible harmful effects of the design features listed in 5.5.2.2(a) through 5.5.2.2(f) shall be evaluated when contemplating comparable equipment experience.

(4) Corrosion Fatigue

(-a) The design fatigue cycles given by eqs. (3-F.21) and (3-F.22) do not include any allowances for corrosive conditions and may be modified to account for the effects of environment other than ambient air that may cause corrosion or subcritical crack propagation. If corrosion fatigue is anticipated, a factor should be chosen on the basis of experience or testing, by which the calculated design fatigue cycles (fatigue strength) should be reduced to compensate for the corrosion.

(-b) When using (3-F.22) an environmental modification factor shall be specified in the User's Design Specification.

(-c) If due to lack of experience it is not certain that the chosen stresses are low enough, it is advisable that the frequency of inspection be increased until there is sufficient experience to justify the factor used. This need for increased frequency should be stated in the User's Design Specification.

(g) Materials of Construction

(1) Material specification requirements shall be in accordance with one or more of the following criteria.

(-a) Specification of materials of construction in accordance with Part 3.

(-b) Generic material type (i.e., carbon steel or Type 304 Stainless Steel). The user shall specify requirements that provide an adequate basis for selecting materials to be used for the construction of the vessel. The Manufacturer shall select the appropriate material from Part 3, considering information provided by the user per (3).

(2) The user shall specify the corrosion and/or erosion allowance.

(3) The user, when selecting the materials of construction, shall consider the following:

(-a) Damage mechanisms associated with the service fluid at design conditions. Informative and nonmandatory guidance regarding metallurgical phenomena is provided in Section II, Part D, Nonmandatory Appendix A; API RP 571; and WRC Bulletins 488, 489, and 490.

(-b) Minimum Design Metal Temperature and any additional toughness requirements.

(-c) The need for specific weld filler material to meet corrosion resistance requirements, see 6.2.5.8.

(h) Loads and Load Cases
(1) The user shall specify all expected loads and load case combinations as listed in 4.1.5.3.
(2) These loading data may be established by:
(-a) Calculation
(-b) Experimental methods
(-c) Actual experience measurement from similar units
(-d) Computer analysis
(-e) Published data
(i) Overpressure Protection
(1) The user shall be responsible for the design, construction and installation of the overpressure protection system unless it is delegated to the Manufacturer. This system shall meet the requirements of Part 9.
(2) The type of over pressure protection intended for the vessel shall be documented in the User's Design Specification as follows (see 9.1):
(-a) Type of overpressure protection system (e.g., type of pressure relief valve, rupture disc, etc.)
(-b) System design [see 9.5(e)]
(3) The user shall state if jurisdictional acceptance is required prior to operation of the vessel.

2.2.3.2 Additional Requirements. The user shall state what additional requirements are appropriate for the intended vessel service such as:
(a) Additional requirements such as non-destructive examination, restricted chemistry, or heat treatments
(b) Type of weld joints and the extent of required nondestructive examinations
(c) Nonmandatory or optional provisions of this Division that are considered to be mandatory for the subject vessel
(d) Any special requirements for marking and their location (see 4.1 and Annex 2-F)
(e) Requirements for seals and/or bolting for closures and covers
(f) Additional requirements relating to erection loadings
(g) Any agreements which resolve the problems of operation and maintenance control unique to the particular pressure vessel. See also 2.2.3.1(f)(4)(-c).
(h) Specific additional requirements relating to pressure testing such as:
(1) Fluid properties and test temperature limits
(2) Position of vessel and support/foundation adequacy if field hydrostatic testing is required
(3) Location: Manufacturer's facility or on-site
(4) Cleaning and drying
(5) Selection of pressure test method, see 8.1.1
(6) Application of paints, coatings and linings, see 8.1.2(e)

2.3 MANUFACTURER'S RESPONSIBILITIES

2.3.1 CODE COMPLIANCE (21)

2.3.1.1 The Manufacturer is responsible for the structural and pressure-retaining integrity of a vessel or part thereof, as established by conformance with the requirements of the rules of this Division and the requirements in the User's Design Specification.

2.3.1.2 The Manufacturer completing any vessel or part marked with the Certification Mark with the U2 Designator and class or the Certification Mark with the PRT Designator in accordance with this Division has the responsibility to comply with all the applicable requirements of this Division and, through proper certification, to ensure that any work by others also complies with the requirements of this Division. The Manufacturer shall certify compliance with these requirements by completing a Manufacturer's Data Report (see 2.3.4).

2.3.1.3 The PRT Certificate Holder is not permitted to assume full Code responsibility for the completed vessel. The PRT Certificate Holder shall only assume responsibility for the construction and marking of completed parts.

2.3.1.4 A single Manufacturer's Design Report may be completed and certified to document more than one pressure vessel that is to be located in a single, specific jurisdiction, provided that the details of design and construction demonstrate that the environmental requirements and jurisdictional regulatory authority applied for each installation location are the same or more conservative than required.

2.3.2 MATERIALS SELECTION

2.3.2.1 When generic material types (i.e., carbon steel or Type 304 Stainless Steel) are specified, the Manufacturer shall select the appropriate material from Part 3, considering information provided by the user per 2.2.3.1(g)(3).

2.3.2.2 Any material substitutions by the Manufacturer are subject to approval of the user.

(21) 2.3.3 MANUFACTURER'S DESIGN REPORT

2.3.3.1 Certification of a Manufacturer's Design Report for Class 1.

(a) The Manufacturer's Design Report shall be certified by a Certifying Engineer in accordance with Annex 2-B when any of the following are performed:

(1) fatigue analysis

(2) use of Part 5 to determine thickness of pressure parts when design rules are not provided in Part 4

(3) use of Part 4.8 to design a quick-actuating closure

(4) a dynamic seismic analysis

(b) The Manufacturer's Design Report may be certified by an engineer or a designer in accordance with Annex 2-B when none of the conditions of (a)(1) through (a)(4) applies.

2.3.3.2 Certification of a Manufacturer's Design Report for Class 2. The Manufacturer's Design Report shall be certified by a Certifying Engineer in accordance with Annex 2-B.

2.3.3.3 Contents of the Manufacturer's Design Report. The Manufacturer shall provide a Manufacturer's Design Report that includes:

(a) Final as-built drawings.

(b) The actual material specifications used for each component.

(c) Design calculations and analysis that establish that the design as shown on the drawings complies with the requirements of this Division for the design conditions that have been specified in the User's Design Specification.

(1) Documentation of design-by-rule calculations in Part 4 shall include the following:

(-a) The name and version of computer software, if applicable

(-b) Loading conditions and boundary conditions used to address the load cases in the User's Design Specification

(-c) Material models utilized for all required physical properties (i.e., stress–strain data, modulus of elasticity, Poisson's ratio, thermal expansion coefficient, thermal conductivity, thermal diffusivity), strength parameters (i.e., yield and tensile strength), and allowable stresses

(-d) Detailed calculations, including results from all of the applicable steps in the calculations, showing the acceptance criteria utilized to meet the requirements of this Division.

(-e) A summary of the calculation results

(2) Documentation of design-by-analysis calculations in Part 5 shall include the following:

(-a) A detailed description of the numerical method used, including the name and version of computer software, if applicable

(-b) Description of model geometry (including element type for finite element analysis)

(-c) Loading conditions and boundary conditions used to address the load cases in the User's Design Specification

(-d) Material models utilized for all required physical properties (i.e., modulus of elasticity, Poisson's ratio, thermal expansion coefficient, thermal conductivity, thermal diffusivity), strength parameters (i.e., yield and tensile strength), strain limits, if applicable, and the design membrane stress intensity per Part 3

(-e) Description of whether material nonlinearity is utilized in the analysis including a description of the material model (i.e., stress–strain curve and cyclic stress–strain curve)

(-f) Description of the numerical analysis procedure (i.e., static analysis, thermal analysis (temperature and stress), buckling analysis, natural frequency analysis, dynamic analysis) and whether a geometrically linear or nonlinear option is invoked

(-g) Graphical display of relevant results (i.e., numerical model, deformed plots, and contour plots of thermal and stress results)

(-h) Method used to validate the numerical model (i.e., mesh sensitivity review and equilibrium check for finite element analysis, e.g., check of hoop stress in a component away from structural discontinuity and a check to ensure that global equilibrium is achieved between applied loads and reactions at specified boundary conditions)

(-i) Description of results processing performed to establish numerical analysis results (i.e., stress linearization method, use of centroidal or nodal values for stress, strain, and temperature results)

(-j) A summary of the numerical analysis results showing the acceptance criteria utilized to meet the requirements of this Division

(-k) Electronic storage of analysis results including input files and output files that contain numerical analysis results utilized to demonstrate compliance with the requirements of this Division

(d) Any methods of design used that are not covered by the rules of this Division.

(e) The results of any fatigue analyses according to 5.5, as applicable.
(f) Any assumptions used by the Manufacturer to perform the vessel design.

2.3.4 MANUFACTURER'S DATA REPORT

The Manufacturer shall certify compliance to the requirements of this Division by the completion of the appropriate Manufacturer's Data Report as described in Annex 2-C and Annex 2-D.

2.3.5 MANUFACTURER'S CONSTRUCTION RECORDS

The Manufacturer shall prepare, collect and maintain construction records and documentation as fabrication progresses, to show compliance with the Manufacturer's Design Report (e.g., NDE reports, repairs, deviations from drawings, etc.). An index of the construction records files, in accordance with the Manufacturer's Quality Control system, shall be maintained current (see 2-C.3). These construction records shall be maintained by the Manufacturer for the duration as specified in 2-C.3.

2.3.6 QUALITY CONTROL SYSTEM

The Manufacturer shall have and maintain a Quality Control System in accordance with Annex 2-E.

2.3.7 MANUFACTURER'S DESIGN PERSONNEL (21)

2.3.7.1 The Manufacturer has the responsibility of ensuring all personnel performing and/or evaluating design activities are competent in the area of design (see Annexes 2-C and 2-J).

2.3.7.2 The Manufacturer shall maintain a controlled document, referenced in the Quality Control System, identifying the persons who may exercise control of the design work performed by others.

2.3.8 CERTIFICATION OF SUBCONTRACTED SERVICES

2.3.8.1 The Quality Control system shall describe the manner in which the Manufacturer (Certificate Holder) controls and accepts the responsibility for the subcontracting of activities. The Manufacturer shall ensure that all contracted activities meet the requirements of this Division.

2.3.8.2 Work such as forming, nondestructive examination, heat treating, etc., may be performed by others (for welding, see 6.1.4.2). It is the vessel Manufacturer's responsibility to ensure that all work performed complies with all the applicable requirements of this Division. After ensuring compliance, and obtaining concurrence of the Inspector, the vessel may be stamped with the Certification Mark.

2.3.8.3 Subcontracts that involve welding on the pressure boundary components for construction under the rules of this Division, other than as provided in 6.1.4.2 and for repair welds permitted by the ASME material specifications, shall be made only to subcontractors holding a valid U2 Certificate of Authorization. All such subcontracted welding shall be documented on Form A-2; see Annex 2-D.

2.3.8.4 A Manufacturer may engage individuals by contract for their services as Welders or Welding Operators, at shop or site locations shown on his Certification of Authorization, provided all of the following conditions are met:
(a) The work to be done by Welders or Welding Operators is within the scope of the Certificate of Authorization.
(b) The use of such Welders or Welding Operators is described in the Quality Control system of the Manufacturer. The Quality Control System shall include a requirement for direct supervision and direct technical control of the Welders and Welding operators, acceptable to the Manufacturer's accredited Authorized Inspection Agency.
(c) The Welding Procedures have been properly qualified by the Manufacturer, according to Section IX.
(d) The Welders and Welding Operators are qualified by the Manufacturer according to Section IX to perform these procedures.
(e) Code responsibility and control is retained by the Manufacturer.

2.3.9 INSPECTION AND EXAMINATION

The Manufacturer's responsibility for inspection and examination is summarized in Annex 7-A.

2.3.10 APPLICATION OF CERTIFICATION MARK

Vessels or parts shall be stamped in accordance with the requirements in Annex 2-F. The procedure to obtain and use a Certification Mark is described in Annex 2-G.

2.4 THE INSPECTOR

2.4.1 IDENTIFICATION OF INSPECTOR

All references to Inspectors throughout this Division mean the Authorized Inspector as defined in this paragraph. All inspections required by this Division shall be by an Inspector regularly employed by an ASME accredited Authorized Inspection Agency, as defined in ASME QAI-1, or by a company that manufacturers pressure vessels exclusively for its own use and not for resale that is defined as a User-Manufacturer. This is the only instance in which an Inspector may be in the employ of the Manufacturer.

2.4.2 INSPECTOR QUALIFICATION

All Inspectors shall have been qualified in accordance with ASME QAI-1.

2.4.3 INSPECTOR'S DUTIES

2.4.3.1 It is the duty of the Inspector to make all the inspections specified by the rules of this Division. In addition, the Inspector shall make other such inspections as considered necessary in order to ensure that all requirements have been met. Some typical required inspections and verifications that are defined in the applicable rules are included in the Inspector's responsibility for inspection and examination as summarized in Annex 7-A.

2.4.3.2 The Inspector of the completed vessel does not have the duty of establishing the accuracy of the design calculations but has the duty of verifying that the required design calculations have been performed. The Inspector has the duty of verifying that the Manufacturer of the completed vessel has the User's Design Specification on file and that the requirements specified therein have been addressed in the Manufacturer's Design Report. The Inspector shall verify that both the User's Design Specification and the Manufacturer's Design Report are certified in accordance with the requirements of this Division.

2.4.3.3 The Inspector shall verify that the Manufacturer has a valid Certificate of Authorization and is working according to an approved Quality Control System including having a system in place to maintain the documentation for the Manufacturer's construction records current with production, and the reconciliation of any deviations from the Manufacturer's Design Report.

2.4.3.4 The Inspector shall certify the Manufacturer's Data Report. When the Inspector has certified by signing the Manufacturer's Data Report, this indicates acceptance by the Inspector. This acceptance does not imply assumption by the Inspector of any responsibilities of the Manufacturer.

ANNEX 2-A GUIDE FOR CERTIFYING A USER'S DESIGN SPECIFICATION

(21)

(Normative)

2-A.1 GENERAL

(a) When required in 2.2.1, one or more individuals in responsible charge of the specification of the vessel and the required design conditions shall certify that the User's Design Specification meets the requirements of this Division and any additional requirements needed for adequate design. Such certification requires the signature(s) of one or more Certifying Engineers as described in (b). One or more individuals may sign the documentation based on information they reviewed and the knowledge and belief that the objectives of this Division have been satisfied.

(b) One or more individuals in responsible charge of the specification of the vessel and the required design conditions shall certify that the User's Design Specification meets the requirements in 2.2.3. Such certification requires the signature(s) of one or more Certifying Engineers with the requisite technical stature and, when applicable, jurisdictional authority to sign such a document. One or more individuals shall sign the documentation based on information they reviewed and the knowledge and belief that the objectives of this Division have been satisfied. In addition, these individuals shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see 2-A.2.3).

2-A.2 CERTIFICATION OF THE USER'S DESIGN SPECIFICATION

2-A.2.1 When required by 2.2.1.1 or 2.2.1.2, certification of the User's Design Specification requires the signature(s) of one or more Certifying Engineers with requisite experience and qualifications as defined in Annex 2-J. The Certifying Engineer(s) shall certify that the User's Design Specification meets the requirements of 2.2.3.

(a) The Certifying Engineer(s) shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see 2-A.2.3).

(b) This Certifying Engineer shall be other than the Certifying Engineer who certifies the Manufacturer's Design Report, although both may be employed by or affiliated with the same organization.

(c) The Certifying Engineer shall identify the location and authority under which he or she has received the authority to perform engineering work stipulated by the user in the User's Design Specification.

2-A.2.2 When more than one Certifying Engineer certifies and signs the User's Design Specification the area of expertise shall be noted next to their signature under "areas of responsibilities" (e.g., design, metallurgy, pressure relief, fabrication). In addition, one of the Certifying Engineers signing the User's Design Specification shall certify that all elements required by this Division are included in the Specification.

2-A.2.3 An example of a typical User's Design Specification Certification Form is shown in Table 2-A.1.

2-A.3 TABLES

Table 2-A.1
Typical Certification of Compliance of the User's Design Specification

CERTIFICATION OF COMPLIANCE OF THE USER'S DESIGN SPECIFICATION
I (We), the undersigned, being experienced and competent in the applicable field of design related to pressure vessel requirements relative to this User's Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Service Conditions given and provides a complete basis for construction in accordance with Part 2, 2.2.3 and other applicable requirements of the ASME Section VIII, Division 2 Pressure Vessel Code, Class ______, ______Edition, and Code Case(s) ________________. This certification is made on behalf of the organization that will operate these vessels (company name)________________________ .
Certified by: ________________________ Title and areas of responsibility: ________________________ Date: ________________ Certified by: ________________________ Title and areas of responsibility: ________________________ Date: ________________ Certifying Engineer Seal: (As required) ________________________ Date: ________________ Engineer's registration authority: ________________________ Registration authority location: ________________________ Engineer's registration number (if applicable): ________________________

ANNEX 2-B
GUIDE FOR CERTIFYING A MANUFACTURER'S DESIGN REPORT

(Normative)

2-B.1 GENERAL

(a) As required in 2.3.3, one or more individuals in responsible charge of the design and construction of the vessel(s) shall certify that the Manufacturer's Design Report is complete, accurate, and in accordance with the User's Design Specification, and that all the requirements of this Division and any additional requirements needed for adequate design have been met. Such certification requires the signature(s) of one or more individuals as described in (b). One or more individuals may sign the documentation based on information they reviewed and the knowledge and belief that the requirements of this Division have been satisfied.

(b) One or more individual(s) experienced in pressure vessel design shall certify that the Manufacturer's Design Report meets the requirements in 2.3.3. Such certification requires the signature(s) of one or more individuals with the requisite technical and corporate authority needed for such a document. These responsible individuals shall sign the documentation based on information they have reviewed and the knowledge and belief that the objectives of this Division have been satisfied. In addition, these individuals shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see 2-B.2.4).

(c) The Inspector shall review the Manufacturer's Design Report and ensure that the requirements of 2.4.3 have been satisfied.

2-B.2 CERTIFICATION OF MANUFACTURER'S DESIGN REPORT BY A CERTIFYING ENGINEER

2-B.2.1 When required by either 2.3.3.1(a) or 2.3.3.2, certification of the Manufacturer's Design Report requires the signature(s) of one or more Certifying Engineers with requisite experience and qualifications as defined in Annex 2-J. The Certifying Engineer(s) shall certify that the Manufacturer's Design Report meets the requirements of 2.3.3.

(a) The Certifying Engineer(s) shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see 2-B.4).

(b) This Certifying Engineer shall be other than the Certifying Engineer who certifies the User's Design Specification, although both may be employed by or affiliated with the same organization.

(c) The Certifying Engineer shall identify the location and authority under which he or she has reached the authority to perform engineering work stipulated by the user in the User's Design Specification.

2-B.2.2 When more than one Certifying Engineer certifies and signs the Manufacturer's Design Report, the area of expertise shall be noted next to their signature under "areas of responsibilities" (e.g., design, metallurgy, pressure relief, fabrication). In addition, one of the Certifying Engineers signing the Manufacturer's Design Report shall certify that all elements required by this Division are included in the Report.

2-B.3 CERTIFICATION OF A MANUFACTURER'S DESIGN REPORT BY AN ENGINEER OR A DESIGNER (21)

When permitted by 2.3.3.1(b), certification of the Manufacturer's Design Report requires the signature(s) of one or more engineers or designers with requisite experience and qualifications as defined in Annex 2-J. The engineer(s) or designer(s) shall certify that the Manufacturer's Design report meets the requirements of 2.3.3. The Inspector shall review the Manufacturer's Design Report and ensure that the requirements of 2.4.3 have been satisfied.

(a) The engineer or designer shall prepare a statement to be affixed to the document attesting to its compliance with the applicable requirements of the Code (see 2-B.4).

(b) When more than one engineer or designer certifies and signs the Manufacturer's Design Report, the area of expertise shall be noted next to their signature under "areas of responsibilities" (e.g., design, metallurgy, pressure relief, fabrication). In addition, one of the engineers or designers signing the Manufacturer's Design Report shall certify that all elements required by this Division are included in the report.

2-B.4 MANUFACTURER'S DESIGN REPORT CERTIFICATION FORM

An example of a typical Manufacturer's Design Report Certification Form is shown in Table 2-B.1.

2-B.5 TABLES

Table 2-B.1
Typical Certification of Compliance of the Manufacturer's Design Report

CERTIFICATION OF COMPLIANCE OF
THE MANUFACTURER'S DESIGN REPORT

I (We), the undersigned, being experienced and competent in the applicable field of design related to pressure vessel construction relative to the certified User's Design Specification, certify that to the best of my knowledge and belief the Manufacturer's Design Report is complete, accurate and complies with the User's Design Specification and with all the other applicable construction requirements of the ASME Section VIII, Division 2 Pressure Vessel Code, Class ______, ______________ Edition, and Code Case(s) ____________________.

This certification is made on behalf of the Manufacturer (company name)____ .

Certified by: ____

Title and areas of responsibility: ________________________________

Date: ____________________

Certified by: ____

Title and areas of responsibility: ________________________________

Date: ____________________

Certifying Engineer Seal: (As required)

____

Date: ____________________

Engineer's registration authority: ________________________________

Registration authority location: ________________________________

Engineer's registration number (if applicable): ____________________________

Authorized Inspector Review: ________________________________

Date: ____________________

ANNEX 2-C
REPORT FORMS AND MAINTENANCE OF RECORDS

(Normative)

2-C.1 MANUFACTURER'S DATA REPORTS

2-C.1.1 A Manufacturer's Data Report shall be completed by the Manufacturer for each pressure vessel to be stamped with the Certification Mark.

(a) For sample report forms and guidance in preparing Manufacturer's Data Reports, see Annex 2-D.

(b) A Manufacturer's Data Report shall be filled out on Form A-1 or Form A-1P by the Manufacturer and shall be signed by the Manufacturer and the Inspector for each pressure vessel stamped with the Certification Mark with the U2 Designator and class.

(c) A Manufacturer's Data Report shall be filled out on Form A-2 by the Manufacturer and shall be signed by the Manufacturer and the Inspector for each part stamped with the Certification Mark with the U2 or PRT Designator, as applicable. Same-day production of vessel parts may be reported on a single parts-documenting Form A-2, provided all of the following requirements are met:

(1) Vessel parts are identical.

(2) Vessel parts are manufactured for stock or for the same user or his designated agent.

(3) Serial numbers are in uninterrupted sequence.

(4) The Manufacturer's written Quality Control System includes procedures to control the development, distribution, and retention of the Manufacturer's Data Reports.

(d) Horizontal spacing for information on each page may be altered as necessary. All information must be addressed; however, footnotes described in the "Remarks" block are acceptable, e.g., for multiple cases of "none" or "not applicable."

(e) The method of completing the Manufacturer's Data Report shall be consistent. The report shall be typed or handwritten using legible printing. Handwritten additions or corrections shall be initialed and dated by the Manufacturer's representative and Inspector.

(f) Forms shall not contain advertising slogans, logos, or other commercial matter.

(g) Manufacturer's Data Report Forms may be preprinted or computer generated. Forms shall be identical in size, arrangement, and content, as shown in this Appendix, except that additional lines may be added or Form A-3 or Form A-4 may be used.

When using forms that result in multiple pages, each page shall be marked to be traceable to the first page of the form. For Forms A-1 and A-2, each page shall contain, at the top of the page, as a minimum, the Manufacturer's name, Manufacturer's serial number, CRN (as applicable), and National Board number (as applicable), as shown on the first page of the form.

Additionally, on all forms, each sheet shall contain the page number and total number of pages that compose the complete form. These requirements do not apply to Forms A-3 and A-4, which are intended to be single-page forms attached to another form.

2-C.1.2 Special Requirements for Layered Vessels. A description of the layered shell and/or layered heads shall be given on the Manufacturer's Data Report, describing the number of layers, their thickness or thicknesses, and type of construction (see Table 2-D.2 for the use of Form A-3, Manufacturer's Data Report Supplementary Sheet). An example of the use of Form A-3 illustrating the minimum required data for layered construction is given in Form A-3L.

2-C.1.3 Special Requirements for Combination Units.

(a) Those chambers included within the scope of this Division shall be described on the same Manufacturer's Data Report. This includes the following, as applicable:

(1) for differential pressure design, the maximum differential design pressure for each common element and the name of the higher pressure chamber

(2) for mean metal temperature design, the maximum mean metal design temperature for each common element

(3) for a common element adjacent to a chamber not included within the scope of this Division, the common element design conditions from that chamber

(b) It is recommended that those chambers not included within the scope of this Division be described in the "Remarks" section of the Manufacturer's Data Report.

(c) For fixed tubesheet heat exchangers, Form A-4 shall be completed in conjunction with Form A-1.

2-C.1.4 The Manufacturer shall distribute the Manufacturer's Data Report as indicated below.

(a) Furnish a copy of the Manufacturer's Data Report to the user and, upon request, to the Inspector;

(b) Submit a copy of the Manufacturer's Data Report to the appropriate enforcement authority in the jurisdiction in which the vessel is to be installed where required by law;

(c) Keep a copy of the Manufacturer's Data Report on file in a safe repository for at least 3 yr;

(d) In lieu of (b) or (c) above, the vessel may be registered and the Manufacturer's Data Reports filed with the National Board of Boiler and Pressure Vessel Inspectors, 1055 Crupper Ave., Columbus, Ohio 43229, USA, where permitted by the jurisdiction in which the vessel is to be installed.

2-C.2 MANUFACTURER'S PARTIAL DATA REPORTS

2-C.2.1 The parts Manufacturer shall indicate under "Remarks" the extent the Manufacturer has performed any or all of the design functions. For guidance in preparing Manufacturer's Partial Data Reports, see Annex 2-D.

2-C.2.2 Manufacturer's Partial Data Reports for pressure vessel parts requiring examination under this Division, which are furnished to the Manufacturer responsible for the completed vessel, shall be executed by the parts Manufacturer's Inspector in accordance with this Division (see 2.3.1.2). All Manufacturer's Partial Data Reports, Form A-2, shall be attached to the Manufacturer's Data Report, Form A-1 or Form A-1P.

2-C.2.3 A Manufacturer with multiple locations, each holding its own Certificate of Authorization, may transfer pressure vessel parts from one of its locations to another without Manufacturer's Partial Data Reports, provided the Quality Control System describes the method of identification, transfer, and receipt of the parts. For cases in which a Manufacturer has multiple locations that include both shop and field locations, and the field assembly of the vessel is completed by one Manufacturer's location that is different from the part Manufacturer's location(s), the name of the Manufacturer responsible for field assembly shall be shown on Line 1 of the Manufacturer's Data Report. The Manufacturer responsible for field assembly shall complete and sign both the Shop and Field portions of the Manufacturer's Data Report.

2-C.3 MAINTENANCE OF RECORDS

2-C.3.1 The Manufacturer shall maintain a file for three years after stamping of the vessel, and furnish to the user and, upon request, to the Inspector, the reports and records shown below. It is noted that items that are included in the Manufacturer's Quality Control System meet the requirements of these subparagraphs.

(a) User's Design Specification (see 2.2.3)

(b) Manufacturer's Design Report (see 2.3.3)

(c) Manufacturer's Data Report (see 2.3.4)

(d) Manufacturer's Construction Records and Manufacturer's Partial Data Reports (see 2.3.5)

(1) Tabulated list of all material used for fabrication with Materials Certifications and Material Test Reports, and a record of any repairs to pressure-retaining material that require a radiographic examination by the rules of this Division. The record of the repairs shall include the location of the repair, examination results, and the repair procedures.

(2) Fabrication information including all heat treatment requirements, forming and rolling procedure when prepared, an inspection and test plan identifying all inspection points required by the user, and signed inspection reports

(3) List of any subcontracted services or parts, if applicable

(4) Welding Procedure Specifications (WPS), Procedure Qualification Records (PQR), weld map and Welder/Welding Operator Performance Qualification Records for each welder who welded on the vessel

(5) Pressure parts documentation and certifications

(6) Record of all heat treatments including post weld heat treatment (these records may be either the actual heat treatment charts or a certified summary description of heat treatment time and temperature)

(7) Results of production test plates, if applicable

(8) NDE procedures, records of procedure demonstrations, and records of personnel certifications

(9) All reports stating the results of inspection, nondestructive examinations, and testing, including radiographic examination, ultrasonic examination, magnetic particle examination, liquid dye penetrant examination, and hardness tests; and documentation of the Manufacturer's acceptance of the examination results

(10) All nonconformance reports including resolution and a detailed description of any repairs including repair procedures, a sketch, photo, or drawing indicating the location and size of the repaired area

(11) Charts or other records of required hydrostatic, pneumatic, or other tests. Test logs shall include the test date, testing fluid, duration of the test, temperature of the test fluid, and test pressure

(12) Dimensional drawings of the as-built condition

2-C.3.2 The Manufacturer shall maintain a complete set of radiographs until the signing of the Manufacturer's Data Report, and furnish upon request to the user and, upon request to the Inspector [see 7.5.3.1(a)].

ANNEX 2-D
GUIDE FOR PREPARING MANUFACTURER'S DATA REPORTS

(Informative)

2-D.1 INTRODUCTION

2-D.1.1 The instructions in this Annex provide general guidance to the Manufacturer in preparing the Manufacturer's Data Reports as required in 2.3.4.

2-D.1.2 Manufacturer's Data Reports required by this Division are not intended for pressure vessels that do not meet the provisions of this Division, including those of special design or construction that require and receive approval by jurisdictional authorities under, laws, rules, and regulations of the respective state or municipality in which the vessel is to be installed.

2-D.1.3 The instructions for completing the Manufacturer's Data Reports are identified by numbers corresponding to numbers on the sample forms in this Annex (see Forms A-1, A-1P, A-2, A-3, and A-4).

2-D.1.4 Where more space is needed than has been provided on the form for any item, indicate in the space "See Remarks," "See attached Form A-3," or "See attached Form A-4," as appropriate.

2-D.1.5 For fixed tubesheet heat exchangers, Form A-4 shall be completed.

2-D.1.6 It is not intended that these Manufacturer's Data Reports replace in any way the required Manufacturer's Design Report (2.3.3) or the Manufacturer's Construction Records (2.3.5). It is intended that the Manufacturer's Data Reports be used for identifying the vessel, retrieval of records, and certification of compliance with this Division and with the User's Design Specification, by the Manufacturer and by the Inspector.

2-D.2 TABLES

(21)

Table 2-D.1
Instructions for the Preparation of Manufacturer's Data Reports

Applies to Form					Note	
A-1	A-1P	A-2	A-3	A-4	No.	Instructions
X	X	X	X	X	1	Name, street address, city, state or province (as applicable), and country of Manufacturer.
X	X	...	X	X	2	Name and address of purchaser.
X	X	...	X	X	3	Name of user, and address where vessel is to be installed.
...	...	X	...	...	4	Name and address of Manufacturer who will use the vessel part in making the complete vessel
X	X	X	...	X	5	Type of vessel, such as horizontal or vertical, tank, separator, heat exchanger, reactor.
...	...	X	...	...	6	Brief description of vessel part (i.e., shell, two-piece head, tube, bundle).
X	X	X	X	X	7	An identifying Manufacturer's serial number marked on the vessel (or vessel part) (see Annex 2-F).
X	X	X	X	X	8	Applicable Jurisdiction Registration No.
X	X	X	...	X	9	Indicate drawing numbers, including revision numbers, which cover general assembly and list materials. For Canadian registration, the number of the drawing approved by the applicable jurisdictional authority.
...	...	X	...	...	10	Organization that prepared drawing.
X	X	X	X	X	11	Where applicable, National Board Number from Manufacturer's Series of National Board Numbers.
X	X	X	...	...	12	Issue date of Section VIII, Division 2 under which vessel was manufactured, and vessel class.
X	X	X	...	...	13	All Code Case numbers when the vessel is manufactured to any Code Cases.

Table 2-D.1
Instructions for the Preparation of Manufacturer's Data Reports (Cont'd)

Applies to Form					Note	
A-1	A-1P	A-2	A-3	A-4	No.	Instructions
X	...	...	...	...	14	To be completed when one or more parts of the vessel are furnished by others and certified on Manufacturer's Data Report Form A-2 as required by Annex 2-F. The part manufacturer's name and serial number should be indicated.
X	X	X	...	...	15	Show the complete ASME Specification number and grade of the actual material used in the vessel part. Material is to be as designated in Section VIII, Division 2 (e.g., "SA-285 C"). Exceptions: A specification number for a material not identical to an ASME Specification may be shown only if such material meets the criteria in the Foreword of this Section. When material is accepted through a Code Case, the applicable Case Number shall be shown.
X	X	X	...	...	16	Thickness is the nominal thickness of the material used in the fabrication of the vessel. It includes corrosion allowance.
X	X	X	...	...	17	State corrosion allowance on thickness.
X	...	X	...	...	18	Indicate whether the diameter is inside diameter or outside diameter.
X	...	X	...	...	19	The shell length shall be shown as the overall length between closure or transition section welds, for a shell of a single diameter. In other cases, define length, as appropriate.
X	...	X	...	...	20	Type of longitudinal joint in cylindrical section, or any joint in a sphere (e.g., Type No.1 butt, or seamless) per 4.2.
X	X	X	...	...	21	State the temperature and time if heat treatment is performed by the Manufacturer (i.e., postweld heat treatment, annealing, or normalizing). Explain any special cooling procedure under "Remarks."
X	...	X	...	...	22	Indicate examination applied to longitudinal seams. Any additional examinations should be included under "Remarks."
X	...	X	...	...	23	Type of welding used in girth joints in the cylindrical section (see 20).
X	...	X	...	...	24	Indicate examination applied to girth joints (see 22).
X	...	X	...	...	25	Number of cylindrical courses, or belts, required to make one shell.
X	...	X	...	...	26	Show specified minimum thickness of head after forming. State dimensions that define the head shape.
X	X	X	...	...	27	Bolts used to secure removable head or heads of vessel and vessel sections.
X	...	X	...	...	28	For jacketed vessels, explain the type of jacket closures used.
X	X	X	...	X	29	Show the internal maximum allowable working pressure and the external maximum allowable working pressure.
X	X	X	...	X	30	Show the coincident temperatures that correspond to the internal maximum allowable working pressure and the external maximum allowable working pressure, as applicable.
X	X	X	...	...	31	Show minimum Charpy V-notch impact value required and impact test temperature. If exempted, indicate under "Remarks" paragraph under which exemption was taken.
X	X	X	...	...	32	Show minimum design metal temperature.
X	X	X	...	...	33	Show hydrostatic or other tests made with specified test pressure at top of vessel in the test position. Cross out words (pneumatic, hydrostatic, or combination test pressure) that do not apply. Indicate under "Remarks" if vessel was tested in the vertical position. See Part 8 for special requirements for combination units.
X	X	X	...	...	34	Indicate nozzle or other opening that is designated for pressure relief.
X	X	X	...	...	35	Show other nozzles and openings by size and type (see 50).
X	X	X	...	...	36	Show opening designated for inspection. Show location.
X	X	X	...	...	37	Indicate provisions for support of the vessel and any attachments for superimposed equipment.
X	X	X	...	...	38	Indicate whether fatigue analysis is required per Part 4.
X	X	X	...	...	39	Describe contents or service of the vessel.
X	X	X	...	...	40	Space for additional comments, including any Code restrictions on the vessel or any unusual Code requirements that have been met, such as those noted in 21, 22, 24, 31, and 33, or in 1.2.1 and 1.2.2; 2-C.1.3; or 5.10. Indicate stiffening rings, if used.
X	X	X	...	...	41	Certificate of Compliance block is to show the name of the Manufacturer as shown on his ASME Code Certificate of Authorization. This should be signed in accordance with organizational authority defined in the Quality Control System (see Annex 2-E).
X	X	X	...	...	42	This certificate is to be completed by the Manufacturer to show the disposition of the User's Design Specification and the Manufacturer's Design Report, and to identify the individuals who certify them per 2.2.3 and 2.3.3, respectively (see 49).

Table 2-D.1
Instructions for the Preparation of Manufacturer's Data Reports (Cont'd)

Applies to Form					Note	
A-1	A-1P	A-2	A-3	A-4	No.	Instructions
X	X	X	X	X	43	This certificate is to be completed by the Manufacturer and signed by the Authorized Inspector who performs the shop inspection.
X	X	X	X	X	44	This National Board Authorized Inspector Commission number must be shown.
X	X	...	...	...	45	This certificate is for the Authorized Inspector to sign for any field construction or assembly work (see 44 for National Board Authorized Inspector Commission number requirements). Indicate the method used to pressure test the vessel.
...	...	...	X	X	46	Fill in information identical to that shown on the Data Report to which this sheet is supplementary.
...	...	...	X	X	47	Fill in information for which there was insufficient space for a specific item on the Data Report Form as identified by the notation "See attached Form A-3" or "See attached Form A-4" on the Data Report. Identify the information by the applicable Data Report Item Number.
...	...	X	...	...	48	Indicate data, if known.
X	X	X	...	...	49	Registration locale (as required per 2.2.3 and 2.3.3).
X	X	X	...	...	50	Data entries with descriptions acceptable to Inspector. Abbreviations, coded identification, or reference to Code Figure and sketch number may be used to define any generic name. For ASME B16.5 flanges, the class should be identified. Flange facing and attachment to neck is not required. Some typical abbreviations are shown below. • Flanged fabricated nozzle: CI. 300 flg • Long weld neck flange: CI. 300 Iwn • Weld end fabricated nozzle: w.e.
X	X	X	...	...	51	Material for nozzle neck. Flange material not necessary.
X	X	X	...	...	52	Nominal nozzle neck thickness. For ASME B16.ll and similar parts, class designation may be substituted for thickness.
...	...	...	...	X	53	Fill in data required by 4.18.14.3(b).
...	X	...	...	...	54	Indicate whether the heat transfer plates are gasketed, semiwelded, or brazed.
...	X	...	...	...	55	Indicate the endplate width and length dimensions.
...	X	...	...	...	56	Describe *(a)* heat transfer plate model name *(b)* heat transfer plate nominal thickness *(c)* minimum and maximum number of heat transfer plates for the given frame configuration *(d)* quantity of heat transfer plates installed at time of pressure test *(e)* minimum and maximum tightening dimension of installed heat transfer plates at the time of pressure test

Table 2-D.2
Supplementary Instructions for the Preparation of Manufacturer's Data Reports for Layered Vessels

Note Letter	Instructions
A	Letter symbols indicate instructions that supplement the instructions of Table 2-D.1.
B	The Form A-3L is not available preprinted as shown. It is intended as an example of suggested use of Form A-3 for reporting data for a vessel of layered construction. It is intended that the Manufacturer develop his own arrangement to provide supplementary data that describes his vessel.
C	Note the NDE performed (RT, PT, MT, UT).
D	Applies only when heads are of layered construction.
E	Indicates if seamless or welded.
F	When more than one layer thickness is used, add lines as needed.
G	Indicate diameter of vent holes in the layers.
H	Indicate whether vent holes are in random locations in each layer, or are drilled through all layers.
I	Indicate locations of nozzles and openings; layered shell; layered head.
J	Indicate method of attachment and reinforcement of nozzles and openings in layered shells and layered heads. Refer to figure number if applicable.

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FORM A-1 MANUFACTURER'S DATA REPORT FOR PRESSURE VESSELS

As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

Page ____ of ____

1. Manufactured and certified by ______ 1 ______
(Name and address of manufacturer)

2. Manufactured for ______ 2 ______
(Name and address of purchaser)

3. Location of installation ______ 3 ______
(Name and address)

4. Type ___ 5 ___ Horiz. or vert. tank ___ 7 ___ Mfr.'s serial no. ___ 8 ___ CRN ___ 9 ___ Drawing no. ___ 11 ___ Nat'l. Bd. no. ______ Year built

5. The chemical and physical properties of all parts meet the requirements of material specifications of the ASME BOILER AND PRESSURE VESSEL CODE. The design, construction, and workmanship conform to ASME Code, Section VIII, Division 2.
___ 12 ___ Year ___ 12 ___ Class ___ 13 ___ Code case no.

Items 6 to 11 incl. to be completed for single wall vessels, jackets of jacketed vessels, or shells of heat exchangers

6. Shell ___ 15 ___ Material (spec. no., grade) ___ 16 ___ Nom. thk. ___ 17 ___ Corr. allow. ___ 18 ___ Diameter ___ 19 ___ Length (overall)

7. Seams ___ 20 ___ Longitudinal ___ 21 ___ Heat treatment ___ 22 ___ Nondestructive examination
___ 23 ___ Girth ___ 21 ___ Heat treatment ___ 24 ___ Nondestructive examination ___ 25 ___ No. of courses

8. Heads: (a) Matl. ___ 15 20 21 22 ___ Spec. no., grade (b) Matl. ___ 15 20 21 22 ___ Spec. no., grade

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)		26	17							
(b)										

9. If removable, bolts used (describe other fastenings): ___ 27 ___
Matl. spec. no., grade, size, number

10. Jacket closure ___ 28 ___ Describe as ogee and weld, bar, etc. If bar, give dimensions ______. If bolted, describe or sketch.

11. MAWP ___ 29 ___ (Internal) ___ 29 ___ (External) at max. temp. ___ 30 ___ (Internal) ___ 30 ___ (External) Min. design metal temp. ___ 32 ___ at ___ 32 ___

Impact test ___ 31 ___ At test temperature of ___ 31 ___

Hydro., pneu., or comb test pressure ___ 33 ___

Items 12 and 13 to be completed for tube sections

12. Tubesheets ___ 15 ___ Stationary matl. (spec. no., grade) ______ Diam. (subject to pressure) ___ 16 ___ Nom. thk. ___ 17 ___ Corr. allow. ______ Attach. (wld., bolted)
___ 15 ___ Floating matl. (spec. no., grade) ______ (Diam.) ___ 16 ___ Nom. thk. ___ 17 ___ Corr. allow. ______ Attach. (wld., bolted)

13. Tubes ___ 15 ___ Matl. (spec. no., grade) ______ O.D. ______ Nom. thk. ______ Number ______ Type (straight or "U")

Items 14 to 18 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers

14. Shell ___ 15 ___ Material (spec. no., grade) ___ 16 ___ Nom. thk. ___ 17 ___ Corr. allow. ______ Diameter ___ 19 ___ Length (overall)

15. Seams ___ 20 ___ Longitudinal ___ 21 ___ Heat treatment ___ 22 ___ Nondestructive examination
___ 23 ___ Girth ___ 21 ___ Heat treatment ___ 24 ___ Nondestructive examination ______ No. of courses

16. Heads: (a) Matl. ______ Spec. no., grade (b) Matl. ______ Spec. no., grade

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)										
(b)										

17. If removable, bolts used (describe other fastenings): ______
Matl. spec. no., grade, size, number

18. MAWP ___ 29 ___ (Internal) ___ 29 ___ (External) at max. temp. ___ 30 ___ (Internal) ___ 30 ___ (External) Min. design metal temp. ___ 32 ___ at ___ 32 ___

Impact test ___ 31 ___ At test temperature of ___ 31 ___

Hydro., pneu., or comb test pressure ___ 33 ___

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FORM A-1

Page ______ of ______

Manufactured by ______ (41)

Manufacturer's Serial No. ______ (7) ______ CRN ______ (8) ______ National Board No. ______ (11) ______

Items below to be completed for all vessels where applicable.

19. Nozzles inspection and pressure relief device openings

Purpose (Inlet, Outlet, Drain, etc.)	No.	Diam. or Size	Type	Material	Nom. Thk.	Reinforcement Material	How Attached	Location
(34) (35) (36)		(35)	(35) (50)	(15) (51)	(16) (52)			(36)

20. Body Flanges

Body Flanges on Shells

No.	Type	ID	OD	Flange Thk	Min Hub Thk	Material	How Attached	Location	Bolting: Num & Size	Bolting: Bolting Material	Bolting: Washer (OD, ID, thk)	Bolting: Washer Material
						(15)			(27)	(15)		(15)

Body Flanges on Heads

No.	Type	ID	OD	Flange Thk	Min Hub Thk	Material	How Attached	Location	Bolting: Num & Size	Bolting: Bolting Material	Bolting: Washer (OD, ID, thk)	Bolting: Washer Material
						(15)			(27)	(15)		(15)

21. Support Skirt ______ (37) ______ (Yes or no) Lugs ______ (No.) Legs ______ (No.) Other ______ (Describe) Attached ______ (Where and how)

22. Service: Fatigue analysis required ______ (38) ______ (Yes or no) and ______ (39) ______ (Describe contents or service)

Remarks: (21) (22) (24) (31) (33) (37) (40) (47)

______

______

______

(42) **CERTIFICATION OF DESIGN**

User's Design Specification on file at ______

Manufacturer's Design Report on file at ______

User's Design Specification certified by ______ PE State ______ (42) (49) ______ Reg. No. ______

Manufacturer's Design Report certified by ______ PE State ______ (42) (49) ______ Reg. No. ______

(41) **CERTIFICATE OF SHOP COMPLIANCE**

We certify that the statements in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Pressure Vessels, Section VIII, Division 2.

"U2" Certificate of Authorization No. ______ (41) ______ expires ______

Date ______ Co. name ______ (41) ______ (Manufacturer) Signed ______ (41) ______ (Representative)

(43) **CERTIFICATE OF SHOP INSPECTION**

Vessel made by ______ at ______

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by ______ of ______,

have inspected the pressure vessel described in this Manufacturer's Data Report on ______,

and state that, to the best of my knowledge and belief, the Manufacturer has constructed this pressure vessel in accordance with ASME Code, Section VIII, Division 2. By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this Manufacturer's' Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed ______ (43) ______ (Authorized Inspector) Commissions ______ (44) ______ (National Board Authorized Inspector Commission number)

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FORM A-1

Page ______ of ______

Manufactured by ______ (41)

Manufacturer's Serial No. ______ (7) CRN ______ (8) National Board No. ______ (11)

(41)

CERTIFICATE OF FIELD ASSEMBLY COMPLIANCE

We certify that the field assembly construction of all parts of this vessel conforms with the requirements of Section VIII, Division 2 of the ASME BOILER AND PRESSURE VESSEL CODE.

"U2" Certificate of Authorization No. ______ expires ______

Date ______ Co. name ______ Signed ______

Assembler that certified and constructed field assembly   Representative

(45)

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by ______ of ______

have compared the statements in this Manufacturer's Data Report with the described pressure vessel and state that parts referred to as data items ______

not included in the certificate of shop inspection, have been inspected by me and that, to the best of my knowledge and belief, the Manufacturer has constructed and assembled this pressure vessel in accordance with the ASME Code, Section VIII, Division 2.

The described vessel was inspected and subjected to a hydrostatic test of ______ .

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this Manufacturer's Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed ______ (43) Commissions ______ (44)

Authorized Inspector   National Board Authorized Inspector Commission number

(07/17)

(21)

FORM A-1P MANUFACTURER'S DATA REPORT FOR PLATE HEAT EXCHANGERS
As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

Page ____ of ____

1. Manufactured and certified by ______ (1) ______
(Name and address of Manufacturer)

2. Manufactured for ______ (2) ______
(Name and address of Purchaser)

3. Location of installation ______ (3) ______
(Name and address)

4. Type (5) (Horizontal or vertical) (54) (Gasketed, semiweld, brazed) (7) (Manufacturer's serial no.) (8) (CRN) (9) (Drawing no.)
______ (National Board no.) ______ Year built

5. The chemical and physical properties of all parts meet the requirements of material specifications of the ASME BOILER AND PRESSURE VESSEL CODE. The design, construction, and workmanship conform to ASME Code, Section VIII, Division 2.
(12) [Edition (Year)] (13) Code case no.

6. Endplates: (a) (15) (Fixed material) (b) (15) (Movable material) (c) (15) (Other material)

No.	Quantity	Width	Length	Thickness	Corr. Allow.	Heat Treat	Temp.	Time
		(55)	(55)	(16)	(17)	(21)	(21)	(21)

7. Frame compression bolts and nuts ______ (27) ______
(Quantity, diameter, material specification, and grade)

8. Impact test ______ (31) ______
[Indicate YES and the component(s) impact tested, or NO]

9. Heat transfer plates (56) (Plate model) (15) (Material specification and grade) (56) (Thickness) (56) (Minimum/maximum quantity of plates for frame)
(56) (Quantity of plates pressure tested) (56) (Minimum tightening dimension) (56) (Maximum tightening dimension)

10. Chamber 1, MAWP (29) at max. temp (30), (32) MDMT at (29) Hydro/pneu. test press. (33)

11. Chamber 2, MAWP (29) at max. temp (30), (32) MDMT at (29) Hydro/pneu. test press. (33)

12. Nozzles, connections, inspections, and pressure relief device openings:

Purpose (Inlet, Outlet, Drain, etc.)	Qty.	Dia. or Size	Type	Material		Flange Rating	Nozzle Thickness		How Attached		Location (Insp./Open.)
				Nozzle	Flange		Nom.	C.A.	Nozzle	Flange	
(34) (35) (36)		(35)	(35) (50)	(15) (51)	(15)	(50)	(16) (52)	(17)			(36)

13. Supports: Lugs (37) (Quantity) Legs Feet (37) (Quantity) Others (37) (Describe) Attached (37) (Where and how)

14. Service: Fatigue analysis required (38) (Yes or No) (39) (Describe contents or service)

15. Remarks: (21) (31) (40) (47)

(07/21)

FORM A-1P

Page ______ of ______

Manufactured by ______ (41) ______

Manufacturer's Serial No. ______ (7) ______ CRN ______ (8) ______ National Board No. ______ (11) ______

(42)

CERTIFICATION OF DESIGN

User's Design Specification on file at ______

Manufacturer's Design Report on file at ______

User's Design Specification certified by ______ PE State ______ (42) (49) ______ Reg. No. ______

Manufacturer's Design Report certified by ______ PE State ______ (42) (49) ______ Reg. No. ______

(41)

CERTIFICATE OF SHOP COMPLIANCE

We certify that the statements in this report are correct and that all details of design, material, construction, and workmanship of this plate heat exchanger conform to the ASME Code for Pressure Vessels, Section VIII, Division 2.

"U2" Certificate of Authorization No. ______ (41) ______ expires ______

Date ______ Name ______ (41) ______ Signed ______ (41) ______

Manufacturer Representative

(43)

CERTIFICATE OF SHOP INSPECTION

Plate heat exchanger made by ______ at ______

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by ______ of ______,

have inspected the plate heat exchanger described in this Manufacturer's Data Report on ______

and state that, to the best of my knowledge and belief, the Manufacturer has constructed this plate heat exchanger in accordance with ASME Code, Section VIII, Division 2. By signing this certificate neither the Inspector nor his/her employer makes any warranty, expressed or implied, concerning the plate heat exchanger described in this Manufacturer's' Data Report. Furthermore, neither the Inspector nor his/her employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed ______ (43) ______ Commissions ______ (44) ______

Authorized Inspector National Board Authorized Inspector Commission number

(41)

CERTIFICATE OF FIELD ASSEMBLY COMPLIANCE

We certify that the field assembly construction of all parts of this plate heat exchanger conforms with the requirements of Section VIII, Division 2 of the ASME BOILER AND PRESSURE VESSEL CODE.

"U2" Certificate of Authorization No. ______ expires ______

Date ______ Name ______ Signed ______

Assembler Representative

(45)

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by ______ of ______

have compared the statements in this Manufacturer's Data Report with the described plate heat exchanger and state that parts referred to as data items ______,

not included in the certificate of shop inspection, have been inspected by me and that, to the best of my knowledge and belief, the Manufacturer has constructed and assembled this pressure vessel in accordance with the ASME Code, Section VIII, Division 2.

The described plate heat exchanger was inspected and subjected to a hydrostatic test of ______.

By signing this certificate neither the Inspector nor his/her employer makes any warranty, expressed or implied, concerning the plate heat exchanger described in this Manufacturer's Data Report. Furthermore, neither the Inspector nor his/her employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed ______ (43) ______ Commissions ______ (44) ______

Authorized Inspector National Board Authorized Inspector Commission number

(07/17)

(21)

FORM A-2 MANUFACTURER'S PARTIAL DATA REPORT

Page ____ of ____

A PART OF A pressure Vessel Fabricated by One Manufacturer for Another Manufacturer
As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by ______ 1 ______
(Name and address of manufacturer)

2. Manufactured for ______ 4 ______
(Name and address of purchaser)

3. Location of installation ______ 3 ______
(Name and address)

4. Type ______ 5 Horiz. or vert. tank ______ 7 Mfr's. Serial No. ______ 8 CRN ______ 9 Drawing No. ______ 11 Nat'l Board No. ______ Year built

5. The chemical and physical properties of all parts meet the requirements of material specifications of the ASME BOILER AND PRESSURE VESSEL CODE. The design, construction, and workmanship conform to ASME Code, Section VIII, Division 2.
______ 12 Year ______ 12 Class ______ 13 Code case No.

6. Constructed to: ______ Drawing No. ______ Drawing Prepared by ______ 6 Description of part inspected

Items 7 to 12 incl. to be completed for single wall vessels, jackets of jacketed vessels, or shells of heat exchangers

7. Shell ______ 15 Material (Spec. No., Grade) ______ 16 Nom. thk. ______ 17 Corr. allow. ______ 18 diameter ______ 19 Length (overall)

8. Seams ______ 20 Longitudinal ______ 21 Heat treatment ______ 22 Nondestructive Examination
______ 23 Girth ______ 21 Heat treatment ______ 24 Nondestructive Examination ______ 25 No. of Courses

9. Heads: (a) Matl. ______ 15 20 21 22 Spec., No., Grade ______ (b) Matl. ______ 15 20 21 22 Spec., No., Grade

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)		26	17							
(b)										

10. If removable, bolts used (describe other fastenings): ______ 27 Matl. Spec. No. Grade Size Number

11. Jacket closure ______ 28 Describe as ogee and weld, bar. etc ______ If bar, give dimensions ______ If bolted, describe or sketch.

12. MAWP ______ 29 (internal) ______ 29 (external) at max. temp. ______ 30 (internal) ______ 30 (external) Min. design metal temp. ______ 32 at ______ 32
Impact test ______ 31 ______ At test temperature of ______ 31
Hydro., pneu., or comb test pressure ______ 33 ______

Items 13 and 14 to be completed for tube sections.

13. Tubesheets ______ 15 Stationary matl. (Spec. No., Grade) ______ 18 Diam. (Subject to pressure) ______ 16 Nom. thk. ______ 17 Corr. Allow. ______ Attach. (wld., bolted)
______ 15 Floating matl. (Spec. No., Grade) ______ (Diam.) ______ 16 Nom. thk. ______ 17 Corr. Allow. ______ Attach. (wld., bolted)

14. Tubes ______ 15 Matl. (Spec. No.. Grade) ______ O.D. ______ Nom. thk. ______ Number ______ Type (straight or "U")

Items 15 to 18 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers

15. Shell ______ 15 Material (Spec. No., Grade) ______ 16 Nom. thk. ______ 17 Corr. allow. ______ 18 diameter ______ 19 Length (overall)

16. Seams ______ 20 Longitudinal ______ 21 Heat treatment ______ 22 Nondestructive Examination
______ 23 Girth ______ 21 Heat treatment ______ 24 Nondestructive Examination ______ No. of Courses

17. Heads: (a) Matl. ______ Spec., No., Grade ______ (b) Matl. ______ Spec., No., Grade

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)										
(b)										

18. If removable, bolts used (describe other fastenings): ______ Matl. Spec. No. Grade Size Number

19. Design press. ______ 29 ______ at max. temp. ______ 30 ______. Charpy impact ______ 31 ______
at test temp. of ______ 31 ______ . Min. design metal temp. ______ 32 at ______
Pneu., hydro., or comb. pressure test ______ 33 ______

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Manufactured by ______ 41 ______

Manufacturer's Serial No. ______ 7 ______ CRN ______ 8 ______ National Board No. ______ 11 ______

Items below to be completed for all vessels where applicable

20. Nozzles inspection and pressure relief device openings

Purpose (Inlet, Outlet, Drain, etc)	No.	Diam. or Size	Type	Material	Nom. Thk.	Reinforcement Material	How Attached	Location
34		35	35	15	16			36
35			50	51	52			
36								
48								

21. Body Flanges

Body Flanges on Shells

No.	Type	ID	OD	Flange Thk	Min Hub Thk	Material	How Attached	Location	Bolting: Num & Size	Bolting Material	Washer (OD, ID, thk)	Washer Material
						15			27	15		15

Body Flanges on Heads

No.	Type	ID	OD	Flange Thk	Min Hub Thk	Material	How Attached	Location	Bolting: Num & Size	Bolting Material	Washer (OD, ID, thk)	Washer Material
						15			27	15		15

22. Support Skirt ______ 37 ______ Lugs ______ Legs ______ Other ______ Attached ______
Yes or No No. No Describe Where and how

Remarks: 21 22 24 31 33 37 40 47

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Page ______ of ______

Manufactured by ______ (41) ______

Manufacturer's Serial No. ______ (7) ______ CRN ______ (8) ______ National Board No. ______ (11) ______

(42) **CERTIFICATION OF DESIGN**

User's Design Specifcation on file at ______

Manufacturer's Design Report on file at ______

User's Design Specification certified by ______ PE State ______ (42) (49) ______ Reg. No. ______

Manufacturer's Design Report certified by ______ PE State ______ (42) (49) ______ Reg. No. ______

(41) **CERTIFICATE OF SHOP COMPLIANCE**

We certify that the statements in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Pressure Vessels, Section VIII, Division 2.

"U2" or "PRT"Certificate of Authorization No ______ (41) ______ expires ______

Date ______ Co. name ______ (41) ______ Signed ______ (41) ______

Manufacturer Representative

(43) **CERTIFICATE OF SHOP INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by ______ of ______,

have inspected the part of a pressure vessel described in this Manufacturer's Data Report on ______,

and state that, to the best of my knowledge and belief, the Manufacturer has constructed this part in accordance with ASME Code, Section VIII, Division 2. By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the part described in this Manufacturer's' Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed ______ (43) ______ Commissions ______ (44) ______

Authorized Inspector National Board Authorized Inspector Commission number

(07/17)

FORM A-3 MANUFACTURER'S DATA REPORT SUPPLEMENTARY SHEET
As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by ______ 1 ______
(Name and address of manufacturer)

2. Manufactured for ______ 2 ______
(Name and address of purchaser)

3. Location of installation ______ 3 ______
(Name and address)

4. Type

5	7	8	9	11	
Horiz. or vert. tank	Mfr's. Serial No.	CRN	Drawing No.	Nat'l Board No.	Year built

Data Report Item Number	Remarks
46	47

Date	______	Co. name	43 46 Manufacturer	Signed	43 46 Representative
Date	______	Signed	43 46 Authorized Inspector	Commissions	44 National Board Authorized Inspector Commission number

(07/17)

FORM A-3L MANUFACTURER'S DATA REPORT SUPPLEMENTARY SHEET
As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by ______ 1 ______
(Name and address of manufacturer)

2. Manufactured for ______ 2 ______
(Name and address of purchaser)

3. Location of installation ______ 3 ______
(Name and address)

4. Type ___ 5 ___ ___ 7 ___ ___ 8 ___ ___ 9 ___ ___ 11 ___ ______
Horiz. or vert. tank | Mfr's. Serial No. | CRN | Drawing No. | Nat'l Board No. | Year built

Data Report Item Number 46	Remarks A B 47					
Item 6 or 7	(a) layered construction type: (Concentric, wrapped, spiral, coil wound, shrink fit, etc.) Nom. Layer					
(Shell)	Location	Mat'l.	Layer Thk.	Nom. Thk.Tot.	No. Courses	NDE
	(b) Inner Shell					
	(c) Dummy Layer	15	F	16	25	C
	(d) Layers:					
	(e) Overwraps:					
Item 8	(a) Layered Construction Type: *(Formed, Machined, Segmental, etc.)*					
(Heads)	(b) Inner Head					
	(c) Dummy Layer	15	F	16	E 20	C
	(d) Layers:					
	(a) Layered Construction Type:					
	(1) Inner Head					
	(2) Dummy Layer					
	(3) Layers:					
Item 21		Diam Hole	Staggered Layers or Radial Through			
(Vent holes in layers)	(a) Layered Shell		H			
	(b) Layered Head	G				
Item 24	Gaps Have Been Controlled According to the Provisions of Paragraph:					
(Remarks)	(See 4.13.12.1, 14.13.12.2, and 14.13.12.3)					
	I J					
	B					

Date ______ Co. name ______ Signed ______
Manufacturer | Representative

Date ______ Signed ______ Commissions ______ 44
Authorized Inspector | National Board Authorized Inspector Commission number

(07/17)

FORM A-4 MANUFACTURER'S DATA REPORT SUPPLEMENTARY SHEET SHELL-AND-TUBE HEAT EXCHANGERS

As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by ______ (1) ______
(Name and address of manufacturer)

2. Manufactured for ______ (2) ______
(Name and address of purchaser)

3. Location of installation ______ (3) ______
(Name and address)

4. Type ______ (5) ______ Horizontal, vertical, or sloped; ______ (7) ______ Mfr's. Serial No.; ______ (8) ______ CRN; ______ (9) ______ Drawing No.; ______ (11) ______ Nat'l Board No.; ______ Year built

FIXED TUBESHEET HEAT EXCHANGERS

Name of Condition	Design/Operating Pressure Ranges: Shell Side Min. (units)	Design/Operating Pressure Ranges: Shell Side Max. (units)	Design/Operating Pressure Ranges: Tube Side Min. (units)	Design/Operating Pressure Ranges: Tube Side Max. (units)	Design/Operating Metal Temperatures: Shell (units)	Design/Operating Metal Temperatures: Channel (units)	Design/Operating Metal Temperatures: Tubes (units)	Design/Operating Metal Temperatures: Tubesheet (units)	Allowable Axial Differential Thermal Expansion Range: Min. (units)	Allowable Axial Differential Thermal Expansion Range: Max. (units)
Design	(29)	(29)	(29)	(29)	(30)	(30)	(30)	(30)	(30)	(30)
(53)	(53)	(53)	(53)	(53)	(53)	(53)	(53)	(53)	(53)	(53)

Data Report Item Number	Remarks
(46)	(47)

Date ______ Co. Name ______ (43) (46) ______ Manufacturer Signed ______ (43) (46) ______ Representative

Date ______ Signed ______ (43) (46) ______ Authorized Inspector Commissions ______ (43) (44) (46) ______ National Board Authorized Inspector Commission number

(07/19)

ANNEX 2-E
QUALITY CONTROL SYSTEM

(Normative)

(21) ## 2-E.1 GENERAL

2-E.1.1 The Manufacturer shall have and maintain a Quality Control System that will establish that all Code requirements, including material, design, fabrication, examination (by the Manufacturer), and inspection of vessels and vessel parts (by the Inspector), will be met. Provided that Code requirements are suitably identified, the system may include provisions for satisfying any requirements by the Manufacturer or user that exceed minimum Code requirements and may include provisions for quality control of non-Code work. In such systems, the Manufacturer of vessels and vessel parts may make changes in parts of the system that do not affect the Code requirements without securing acceptance by the Inspector (see 2.1.1).

2-E.1.2 The system that the Manufacturer uses to meet the requirements of this Division shall be one suitable for the Manufacturer's circumstances. The necessary scope and detail of the system shall depend on the complexity of the work performed and on the size and complexity of the Manufacturer's organization. A written description of the system the Manufacturer will use to produce a Code item shall be available for review. Depending upon the circumstances, the description may be brief or extensive.

2-E.1.3 The written description may contain information of a proprietary nature relating to the Manufacturer's processes. Therefore, the Code does not require any distribution of this information except for the Inspector's or ASME designee's copy as covered by 2-E.15.3. It is intended that information learned about the system in connection with the evaluation will be treated as confidential and that all loaned descriptions will be returned to the Manufacturer upon completion of the evaluation.

2-E.2 OUTLINE OF FEATURES INCLUDED IN THE QUALITY CONTROL SYSTEM

The following is a guide to some of the features which should be covered in the written description of the Quality Control System and is equally applicable to both shop and field work.

(a) The information associated with 2.3 and Annex 7-A.

(b) The complexity of the work includes factors such as design simplicity versus complexity, the types of materials and welding procedures used, the thickness of materials, the types of nondestructive examinations applied, and whether heat treatments are applied.

(c) The size and complexity of the Manufacturer's organization includes factors such as the number of employees, the experience level of employees, the number of vessels produced, and whether the factors defining the complexity of the work cover a wide or narrow range.

2-E.3 AUTHORITY AND RESPONSIBILITY

The authority and responsibility of those in charge of the Quality Control System shall be clearly established. Persons performing quality control functions shall have sufficient and well-defined responsibility, the authority, and the organizational freedom to identify quality control problems and to initiate, recommend, and provide solutions.

2-E.4 ORGANIZATION

An organization chart showing the relationship between management and engineering, purchasing, manufacturing, field construction, inspection, and quality control is required to reflect the actual organization. The purpose of this chart is to identify and associate the various organizational groups with the particular function for which they are responsible. The Code does not intend to encroach on the Manufacturer's right to establish, and from time to time to alter, whatever form of organization the Manufacturer considers appropriate for its Code work.

2-E.5 DRAWINGS, DESIGN CALCULATIONS, AND SPECIFICATION CONTROL

2-E.5.1 The Manufacturer's Quality Control System shall provide procedures which will ensure that the latest applicable drawings, design calculations, specifications, and instructions, required by the Code, as well as authorized changes, are used for manufacture, assembly, examination, inspection, and testing. The system shall ensure that authorized changes are included, when appropriate, in the User's Design Specification and/or in the Manufacturer's Design Report.

2-E.5.2 The Manufacturer's or Assembler's Quality Control System shall provide procedures that will ensure Certifying Engineers and Designers performing design activities are competent for each activity they perform (see Annex 2-J).

2-E.6 MATERIAL CONTROL

The Manufacturer shall include a system of receiving control that will ensure that the material received is properly identified and has documentation including required material certifications or material test reports to satisfy Code requirements as ordered. The system material control shall ensure that only the intended material is used in Code construction.

2-E.7 EXAMINATION AND INSPECTION PROGRAM

The Manufacturer's Quality Control System shall describe the fabrication operations, including examination, sufficiently to permit the Inspector or ASME designee to determine at what stages specific inspections are to be performed.

2-E.8 CORRECTION OF NONCONFORMITIES

There shall be a system agreed upon with the Inspector for correction of nonconformities. A nonconformity is any condition which does not comply with the applicable rules of this Division. Nonconformities must be corrected or eliminated in some way before the completed component can be considered to comply with this Division.

2-E.9 WELDING

The Quality Control System shall include provisions for indicating that welding conforms to requirements of Section IX as supplemented by this Division.

2-E.10 NONDESTRUCTIVE EXAMINATION

The Quality Control System shall include provisions for identifying nondestructive examination procedures the Manufacturer or Assembler will apply to conform to the requirements of this Division.

2-E.11 HEAT TREATMENT

The Quality Control System shall provide controls to ensure that heat treatments as required by the rules of this Division are applied. Means shall be indicated by which the Inspector or ASME designee will be ensured that these Code heat treatment requirements are met. This may be by review of furnace time-temperature records or by other methods as appropriate.

2-E.12 CALIBRATION OF MEASUREMENT AND TEST EQUIPMENT

The Manufacturer shall have a system for the calibration of examination, measuring, and test equipment used in fulfillment of requirements of this Division.

2-E.13 RECORDS RETENTION

The Manufacturer shall have a system for the maintenance of Data Reports and records as required by this Division. Requirements for maintenance of records are given in 2-C.3. Additionally, retained records as required by this Division and the Quality Control System shall be made available to the Authorized Inspector Supervisors or to review teams designated by ASME.

2-E.14 SAMPLE FORMS

The forms used in this Quality Control System and any detailed procedures for their use shall be available for review. The written description shall make necessary references to these forms.

2-E.15 INSPECTION OF VESSELS AND VESSEL PARTS

2-E.15.1 Inspection of vessels and vessel parts shall be by the Inspector as defined in 2.4.

2-E.15.2 The written description of the Quality Control System shall include reference to the Inspector.

2-E.15.3 The Manufacturer shall make available to the Inspector, at the Manufacturer's plant or construction site, a current copy of the written description of the Quality Control System.

2-E.15.4 The Manufacturer's Quality Control System shall provide for the Inspector at the Manufacturer's plant to have access to the User's Design Specification, the Manufacturer's Design Report, and all drawings, calculations, specifications, procedures, process sheets, repair procedures, records, test results, and other documents as necessary for the Inspector to perform his duties in accordance with this Division. The Manufacturer may provide such access either to his own files of such documents or by providing copies to the Inspector.

(21) 2-E.16 INSPECTION OF PRESSURE RELIEF VALVES

Moved to Section XIII. See Annex 9-B for a complete cross-reference list.

ANNEX 2-F
CONTENTS AND METHOD OF STAMPING

(Normative)

2-F.1 REQUIRED MARKING FOR VESSELS

Each pressure vessel to which the Certification Mark with the U2 Designator and class is applied shall be marked with the following:

(a) The official Certification Mark with the U2 Designator and class, as shown in Figure 2-F.1, sketch (a), which shall be stamped on vessels certified in accordance with this Division.

(b) The name of the Manufacturer of the pressure vessel as it is shown on the Certificate of Authorization or an abbreviation accepted by ASME, preceded by "Certified by." A trademark is not considered to be sufficient identification for vessels or parts constructed to this Division.

(c) The Manufacturer's serial number (MFG SER).

(d) The MAWP (Maximum Allowable Working Pressure), internal or external, at the coincident maximum design metal temperature. When a vessel is specified to operate at more than one pressure and temperature condition, such values of coincident pressure and design temperature shall be added to the required markings. The maximum allowable working pressure (external) is required only when specified as a design condition.

(e) The MDMT (minimum design metal temperature) at coincident MAWP in accordance with Part 3.

(f) The year built.

(g) Code Edition (see 2.1.3).

(h) The construction type, i.e., all of the applicable construction types shall be marked under the Certification Mark and U2 Designator and class or, if marking is by a fabricator of pressure vessel parts only, under the Certification Mark and PRT Designator. WL (welded layered) is the only construction type that is required to be marked on the vessel.

(i) Heat treatment markings shall be as follows:

(1) The letters HT shall be applied under the Certification Mark and U2 Designator and class and under the Certification Mark and PRT Designator, as applicable, when the complete vessel has been postweld heat treated in accordance with Part 3.

(2) The letters PHT shall be applied under the Certification Mark and U2 Designator and class or under the Certification Mark and PRT Designator, as applicable, when only part of the complete vessel has been postweld heat treated in accordance with Part 3.

(j) When the complete vessel or vessel parts are inspected by a user's Inspector as provided in 2.4.1, the word USER shall be marked above the

(1) Certification Mark and U2 Designator and class for complete pressure vessels or vessel parts, as applicable, or

(2) Certification Mark and PRT Designator for parts only, which have been fabricated by a Manufacturer holding a valid PRT Certificate of Authorization

2-F.2 REQUIRED MARKING FOR COMBINATION UNITS

(a) Those chambers included within the scope of this Division shall be marked. The marking shall include the name of each chamber (e.g., process chamber, jacket, tubes, channel) and its corresponding data. The markings shall be grouped in one location on the combination unit or applied to each individual chamber. Each detachable chamber shall be marked to identify it with the combination unit. When required, the marking shall include the following:

(1) for differential pressure design, the maximum differential design pressure for each common element and the name of the higher pressure chamber

(2) for mean metal temperature design, the maximum mean metal design temperature for each common element

(3) for a common element adjacent to a chamber not included within the scope of this Division, the common element design conditions from that chamber

(b) It is recommended that the design conditions for those chambers not included within the scope of this Division be marked on the combination unit. The markings may be on the applicable chamber or grouped as described in 2-F.2(a), provided they are not included in the markings covered by the Certification Mark.

2-F.3 APPLICATION OF STAMP

The Certification Mark with the U2 Designator and class or the Certification Mark with the PRT Designator shall be applied by the Manufacturer only with the approval of the Inspector, and after the hydrostatic test and all other required inspection and testing has been satisfactorily completed. Such application of the Certification Mark with the U2 Designator and class or the Certification Mark with the PRT Designator, together with final certification in accordance with the rules of this Division, shall confirm that all applicable requirements of this Division and the User's Design Specification have been satisfied.

2-F.4 PART MARKING

(21) **2-F.4.1** Parts of pressure vessels for which Partial Data Reports are required shall be marked by the parts Manufacturer with the following:

(a) the official Certification Mark with, as applicable, the

(1) U2 Designator and class, as shown in Figure 2-F.1, above the word "PART," or

(2) PRT Designator, as shown in Figure 2-F.1

(b) the name of the Manufacturer of the part, preceded by the words "Certified by"

(c) the Manufacturer's serial number assigned to the part

(d) the MAWP and coincident maximum design metal temperature (see Part 2)

(e) the MDMT (minimum design metal temperature) at the MAWP (see Part 3)

When stamping with the Certification Mark with the PRT Designator, the word "PART" may be eliminated from the stamping.

2-F.4.2 The requirements for part marking in accordance with 2-F.4.1(d) and 2-F.4.1(e) do not apply for the following:

(a) parts for which the parts Manufacturer does not prepare a Manufacturer's Design Report

(b) overpressure relief devices that are covered in Part 9

(21) 2-F.5 APPLICATION OF MARKINGS

Markings required in 2-F.1 through 2-F.4 shall be applied by one of the following methods:

(a) Nameplate – A separate metal nameplate, of a metal suitable for the intended service, at least 0.5 mm (0.02 in.) thick, shall be permanently attached to the vessel or to a bracket that is permanently attached to the vessel. The nameplate and attachment shall be such that removal shall require willful destruction of the nameplate or its attachment system. The attachment weld to the vessel shall not adversely affect the integrity of the vessel. Attachment by welding shall not be permitted on materials enhanced by heat treatment or on vessels that have been pre-stressed.

(1) Only the Certification Mark need be stamped on the nameplate.

(2) All other data may be stamped, etched, or engraved on the nameplate (see 2-F.7).

(3) The nameplate for the vessel may be attached to a component other than the pressure-retaining shell under the following conditions:

(-a) The UDS shall state the need for not directly attaching the nameplate on the vessel shell.

(-b) The nameplate shall be located in a clearly visible location and welded to the vessel skirt or other component that is permanently attached to the vessel.

(-c) The nameplate location shall be indicated in the remarks on the Data report.

(b) Directly on Vessel Shell

(1) Markings shall be stamped, with low stress type stamps, directly on the vessel, located on an area designated as a low stress area by the Manufacturer in the Manufacturer's Design Report (see 2.3.3).

(2) Markings, including the Certification Mark, may be electrochemically etched on the external surfaces on the vessel under the following conditions:

(-a) The markings are acceptable to the user as indicated in the User's Design Specification.

(-b) The data shall be in characters not less than 8 mm ($^5/_{16}$ in.) high.

(-c) The materials shall be limited to high alloy steels and nonferrous materials.

(-d) The process controls for electrochemical etching shall be described in the Quality Control System and shall be acceptable to the Authorized Inspector. The process controls shall be established so that it can be demonstrated that the characters will be at least 0.1 mm (0.004 in.) deep.

(-e) The external vessel surface condition where electrochemical etching is acceptable shall be clean, uncoated, and unpainted.

(-f) The electrochemical etching shall not result in any detrimental effect to the materials of the vessel.

(c) Adhesive Attachment – Nameplates may be attached with pressure-sensitive acrylic adhesive systems in accordance with the following requirements:

(1) Adhesive systems for the attachment of nameplates are permitted under the following conditions:

(-a) The adhesive used is a pressure-sensitive acrylic adhesive that has been preapplied by the nameplate manufacturer to a nominal thickness of at least 0.13 mm (0.005 in.).

(-b) The adhesive is protected with a moisture-stable liner.

(-c) The vessel(s) to which the nameplate is being attached has a design temperature within the range of −40°C to 150°C (−40°F to 300°F), inclusive.

(-d) The nameplate is applied to a clean, bare metal surface with attention being given to removal of anti-weld-spatter compound that may contain silicone.

(-e) The nameplate application procedure is qualified as outlined in (2).

(-f) The preapplied adhesive is used within 2 yr after initial adhesive application.

(2) Nameplate Application Procedure Qualification

(-a) The Manufacturer's Quality Control System (see Annex 2-E) shall define that written procedures, acceptable to the Inspector, for the application of adhesive-backed nameplates shall be prepared and qualified.

(-b) The application procedure qualification shall include the following essential variables, using the adhesive and nameplate manufacturers' recommendations where applicable:

(-1) Description of the pressure-sensitive acrylic adhesive system employed, including generic composition

(-2) The qualified temperature range, the cold box test temperature shall be −40°C (−40°F) for all applications

(-3) Materials of nameplate and substrate when the mean coefficient of expansion at design temperature of one material is less than 85% of that for the other material

(-4) Finish of the nameplate and substrate surfaces

(-5) The nominal thickness and modulus of elasticity at application temperature of the nameplate when nameplate preforming is employed — a change of more than 25% in the quantity: [(nameplate nominal thickness)^2 × nameplate modulus of elasticity at application temperature] will require requalification

(-6) The qualified range of preformed nameplate and companion substrate contour combinations when preforming is employed

(-7) Cleaning requirements for the substrate

(-8) Application temperature range and application pressure technique

(-9) Application steps and safeguards

(-c) Each procedure used for nameplate attachment by pressure-sensitive acrylic adhesive systems shall be qualified for outdoor exposure in accordance with Standard UL-969, Marking and Labeling Systems, with the following additional requirements.

(-1) Width of nameplate test strip shall not be less than 25 mm (1 in.).

(-2) Nameplates shall have an average adhesion of not less than 1.4 N/mm (8 lb/in.) of width after all exposure conditions, including low temperature.

(-3) Any change in (-b) shall require requalification.

(-4) Each lot or package of nameplates shall be identified with the adhesive application date.

2-F.6 DUPLICATE NAMEPLATE

A duplicate nameplate may be attached on the support, jacket, or other permanent attachment to the vessel. All data on the duplicate nameplate, including the Certification Mark with U2 Designator and class, shall be cast, etched, engraved, or stamped. The Inspector need not witness this marking. The duplicate nameplate shall be marked "DUPLICATE." The use of duplicate nameplates, and the stamping of the Certification Mark on the duplicate nameplate, shall be controlled as described in the Manufacturer's Quality Control System.

2-F.7 SIZE AND ARRANGEMENTS OF CHARACTERS FOR NAMEPLATE AND DIRECT STAMPING OF VESSELS

(21) **2-F.7.1** The data shall be arranged substantially as shown in Figure 2-F.1. The characters for direct stamping of the vessel shall be not less than 8 mm ($^5/_{16}$ in.) high. The characters for nameplate stamping shall be not less than 4 mm ($^5/_{32}$ in.) high. The characters shall be either indented or raised at least 0.10 mm (0.004 in.) and shall be legible and readable.

2-F.7.2 Where space limitations do not permit the requirements of 2-F.7.1 to be met, such as for parts with outside diameters of 89 mm (3.5 in.) or smaller, the required character size to be stamped directly on the vessel may be 3 mm ($^1/_8$ in.).

2-F.8 ATTACHMENT OF NAMEPLATE OR TAG

If all or part of the data is marked on the nameplate or tag before it is attached to the vessel, the Manufacturer shall ensure that the nameplate with the correct marking has been attached to the vessel to which it applies as described in their Quality Control System. The Inspector shall verify that this has been done.

2-F.9 FIGURES

Figure 2-F.1
Form of Stamping (21)

USER (when inspected by user's Inspector as provided in 2.4.1)

U2
CLASS (1 or 2)

Letters Denoting the Construction Type [see 2-F.1(h), (i), and (j), and 2-F.4.1(a)]

(a) [Note (1)]

USER (when inspected by user's Inspector as provided in 2.4.1)

PRT

Letters Denoting the Construction Type [see 2-F.1(h), (i), and (j), and 2-F.4.1(a)]

(b) [Note (1)]

Certified by

____
(Name of Manufacturer)

__________ at __________
Maximum Allowable Working Pressure (Internal)

__________ at __________
Maximum Allowable Working Pressure (External) [Note (2)]

__________ at __________
Minimum Design Metal Temperature

____
Manufacturer's Serial Number

____
Year Built

____
Code Edition

GENERAL NOTES:

(a) For cases where the MAWP (internal) and MAWP (external) have the same designated coincident temperature, the values may be combined on a single line as follows:

$$P_{int}/FV \text{ (psi) at Temp (°F)}$$

(b) The letters "FV" may be used to designate a full vacuum condition, e.g., 150 psi/FV at 300°F.

NOTES:

(1) Information within parentheses or brackets is not part of the required marking. Phrases identifying data may be abbreviated; minimum abbreviations shall be MAWP, MDMT, S/N, and year, respectively.

(2) The maximum allowable working pressure (external) required only when specified as a design condition.

ANNEX 2-G
OBTAINING AND USING CERTIFICATION MARK STAMPS

(Normative)

(21) ## 2-G.1 CERTIFICATION MARK

A Certificate of Authorization to use the Certification Mark with the U2 or PRT Designator (see https://www.asme.org/shop/certification-accreditation) shown in Annex 2-F will be granted by ASME pursuant to the provisions of the following paragraphs. Stamps for applying the Certification Mark shall be obtained from ASME.

2-G.2 APPLICATION FOR CERTIFICATE OF AUTHORIZATION

Any organization desiring a Certificate of Authorization shall apply to ASME in accordance with the certification process of ASME CA-1. Authorization to use Certification Marks may be granted, renewed, suspended, or withdrawn as specified in ASME CA-1.

2-G.3 ISSUANCE OF AUTHORIZATION

A Certificate of Authorization shall be issued in accordance with ASME CA-1.

2-G.4 DESIGNATED OVERSIGHT

The Manufacturer shall comply with the requirements of ASME CA-1 for designated oversight by use of an Authorized Inspection Agency.

2-G.5 QUALITY CONTROL SYSTEM

Any Manufacturer holding or applying for a Certificate of Authorization shall demonstrate a Quality Control System that meets the requirements of ASME CA-1 and Annex 2-E.

2-G.6 EVALUATION OF THE QUALITY CONTROL SYSTEM

2-G.6.1 The issuance or renewal of a Certificate of Authorization is based upon ASME's evaluation and approval of the Quality Control System, and shall be in accordance with ASME CA-1.

2-G.6.2 Before issuance or renewal of a Certificate of Authorization for use of the Certification Mark with the U2 Designator and class or the Certification Mark with the PRT Designator, the Manufacturer's facilities and organization are subject to a joint review by a representative of his Authorized Inspection Agency and an individual certified as an ASME designee who is selected by the concerned legal jurisdiction. For those areas where there is no jurisdiction or where a jurisdiction does not choose to select an ASME designee to review a Manufacturer's facility, an ASME designee selected by ASME shall perform that function. Where the jurisdiction is the Manufacturer's Inspection Agency, the jurisdiction and the ASME designee shall make the joint review and joint report.

(21) **2-G.6.3**

DELETED

2-G.7 CODE CONSTRUCTION BEFORE RECEIPT OF CERTIFICATE OF AUTHORIZATION

A Manufacturer may start fabricating Code items before receipt of a Certificate of Authorization to use a Certification Mark and Designator under the conditions specified in ASME CA-1.

ANNEX 2-I ESTABLISHING GOVERNING CODE EDITIONS AND CASES FOR PRESSURE VESSELS AND PARTS

(Normative)

2-I.1 GENERAL

(a) After Code revisions are approved by ASME, they may be used beginning with the date of issuance shown on the Code. Except as noted below, revisions become mandatory six months after the date of issuance. Code Cases are permissible and may be used beginning with the date of approval by ASME. Only Code Cases that are specifically identified as being applicable to this Section may be used. At the time a Code Case is applied, only the latest revision may be used. Code Cases that have been incorporated into this Section or have been annulled shall not be used.

(b) Changes in the Code and Code Cases that have been published prior to completion of the pressure vessel or part may include details critical to the intended service conditions of the pressure vessel, which should be considered by the Manufacturer. Application of such changes shall be a matter of agreement between the Manufacturer and the user. Specific incorporated Code provisions from later editions that have been applied to construction shall be noted in the "Remarks" section of the Manufacturer's Data Report.

2-I.2 CONSTRUCTION

(a) The Manufacturer of any complete vessel or part that is to be stamped with the ASME Certification Mark required by this Section (see Annex 2-C) has the responsibility of ensuring through proper Code certification that all work performed complies with the effective Code Edition as follows:

(1) Vessels. The Code Edition used for construction of a pressure vessel shall be either the Edition that is mandatory on the date the pressure vessel is contracted for by the Manufacturer, or a published Edition issued by ASME prior to the contract date that is not yet mandatory [see 2-I.1(a)].

(2) Subcontracted Parts. When a vessel Manufacturer subcontracts some of the construction to another Certificate Holder, the part Manufacturer shall construct the part to the Code Edition established for the entire pressure vessel.

(3) Parts Built for Stock. Parts built for stock shall be constructed to either the Edition that is mandatory at the time of Code certification or a published Edition issued by ASME prior to Code certification that is not yet mandatory [see 2-I.1(a)].

(4) Parts Used From Stock. When a vessel Manufacturer uses a part from stock, the vessel Manufacturer shall ensure that the part fully satisfies all applicable Code requirements for the Code Edition used for construction of the complete vessel.

(b) It is permitted to use overpressure protection requirements from the Edition in effect when the vessel is placed in service.

2-I.3 MATERIALS

For parts subject to stress due to pressure, the Manufacturer shall use material conforming to one of the specifications listed as approved for use in the Edition specified for construction, or listed as approved for use in the Guideline for Acceptable ASTM Editions or in the Guideline for Acceptable Non-ASTM Editions in Section II, Part A or Part B.

ANNEX 2-J QUALIFICATIONS AND REQUIREMENTS FOR CERTIFYING ENGINEERS AND DESIGNERS (21)

(Normative)

2-J.1 INTRODUCTION

(a) Persons engaged in design activity shall be competent in the topic of each design activity performed and shall be able to show evidence of this competency as described in 2-J.2.

(b) When a Certifying Engineer is required by 2.3.3.1 to certify the Manufacturer's Design Report, it is permissible for an engineer or designer to perform the design activity, provided all the following requirements are met:

(1) The individual has evidence of competence in the topic of design under consideration.

(2) The individual is working under the responsible charge[1] of a Certifying Engineer.

2-J.2 COMPETENCY REQUIREMENTS

(a) The engineer or designer may engage in any activity required by this Division or any supplemental requirements from the User's Design Specification except for Code activities listed in Table 2-J.1, unless the requirements of 2-J.1(b) are met.

(b) The Certifying Engineer may engage in any design activity required by this Division or any supplemental requirements from the User's Design Specification.

2-J.3 QUALIFICATION REQUIREMENTS

2-J.3.1 GENERAL

(a) One or more persons within the Manufacturer's organization shall be qualified to perform design work in accordance with the requirements of this Annex for any design activity listed in the Manufacturer's Quality Control System (see 2-E.5).

(b) The qualifications of 2-J.3.2 and 2-J.3.3 shall also apply to Certifying Engineers, engineers, and designers that are engaged by the Manufacturer by contract or agreement for their services.

2-J.3.2 CERTIFYING ENGINEERS

(a) The Certifying Engineer shall attest in writing that they understand and meet the requirements of the ASME Code of Ethics and shall meet the requirements of (b) and (c).

(b) The Certifying Engineer may perform any design activity required by this Division for which the engineer has a minimum of 4 yr of experience in the design of pressure vessels.

(1) For Certifying Engineers who certify the Manufacturer's Design Report, this experience shall be demonstrated through documentation certified by a Manufacturer.

(2) For Certifying Engineers who certify the User's Design Specification, this experience shall be demonstrated through documentation maintained by the Certifying Engineer.

(c) The Certifying Engineer shall be chartered, registered, or licensed in accordance with one or more of the following:

(1) a registered Professional Engineer in at least one state of the United States or province of Canada

(2) the International Register of Professional Engineers by an authorized member of the International Professional Engineers Agreement (IPEA)

[1] For further information regarding responsible charge, see the National Society of Professional Engineers Position Statement No. 1778.

Table 2-J.1
Design Activities Requiring a Certifying Engineer

Design Activities	Code Location
Performance of numerical analysis	5.1.2
Fatigue assessments	
Elastic stress analysis	5.5.3
Elastic–plastic stress analysis	5.5.4
Elastic analysis and structural stress	5.5.5
Design due to seismic reactions	
Linear response history procedure	5.1.3
Nonlinear response history procedure	5.1.3
Quick-actuating closures	4.8

(3) an authorized member of the Asia Pacific Economic Cooperation (APEC)

(4) an authorized member of the European Federation of National Engineering Associations (FEANI)

2-J.3.3 ENGINEERS AND DESIGNERS

(a) Education

(1) The engineer shall have a degree from an accredited university or college in engineering, science, or technology requiring an equivalent of 4 yr of full-time study of higher education.

(2) The designer shall have completed an accredited engineering technician or associates degree, requiring the equivalent of at least 2 yr of study.

(b) Personnel Experience

(1) An engineer engaged in and/or having responsible oversight for pressure vessel design shall as a minimum hold the qualification described in (c)(1)(-b).

(2) A designer meeting the education requirements of (a)(2) engaged in and/or having responsible oversight for pressure vessel design shall as a minimum hold the qualification described in (c)(1)(-c).

(3) A designer that does not meet the minimum education requirements of (a)(2) engaged in and/or having responsible oversight for pressure vessel design shall as a minimum hold the qualification described in (c)(1)(-d).

(4) The engineer or designer may also hold any of the additional qualifications described in (c)(2) through (c)(4).

(c) Practical Experience

(1) General Pressure Vessel Design

(-a) This qualification includes all design activity required for this Division that the individual engages in, or as listed in the Manufacturer's Quality Control System, except as provided for in (2) through (4).

(-b) The engineer shall be able to demonstrate through documentation that he or she has a minimum of 4 yr of experience in pressure vessel design.

(-c) A designer meeting the education requirements of (a)(2) shall be able to demonstrate through documentation that he or she has a minimum of 6 yr of experience in pressure vessel design.

(-d) A designer that does not meet the minimum education requirements of (a)(2) shall be able to demonstrate that he or she has a minimum of 10 yr of experience in pressure vessel design.

(2) Heat Exchanger Design. The engineer or designer shall be able to demonstrate through documentation that he or she has a minimum of 2 yr of experience in each of the following design Code activities for heat exchangers that they practice:

(-a) tubesheets (see 4.18)

(-b) bellows expansion joints (see 4.19)

(-c) flexible shell element expansion joints (see 4.20)

(3) Numerical Analysis

(-a) The engineer or designer shall be able to demonstrate through documentation that he or she has a minimum of 2 yr of experience performing design calculations not specifically addressed in this Division, including numerical analysis.

(-b) An engineer or designer engaged in the performance of numerical analysis shall be able to demonstrate through documentation that he or she has received instruction in the use and understanding of any numerical analysis computer program(s). This documentation shall be provided to the engineer or designer by one of the following:

(-1) the developer of the computer program (e.g., the software vendor)

(-2) a training course acceptable to or licensed by the developer

(-3) a Certifying Engineer with requisite knowledge of the computer program and qualifications to train others on its use

(4) Quick-Actuating Closures. The engineer or designer shall be able to demonstrate through documentation that he or she has a minimum of 2 yr of experience in design activity for quick-actuating closures (see 4.8).

(5) The experience requirements of (2) through (4) may be acquired concurrently.

(6) The engineer or designer's qualification(s) remain valid if the individual can demonstrate through documentation design activity completed within a continuous period of 36 months for each of their qualifications in (1), (2), (3), or (4).

2-J.4 CERTIFICATION REQUIREMENTS

2-J.4.1 CERTIFYING ENGINEERS

(a) The Manufacturer who employs (directly or by contract) the engineer who certifies the Manufacturer's Design Report shall prepare a statement, in conjunction with 2-J.3.2(b)(1), that the Certifying Engineer is qualified to perform the design activities used.

(b) Certifying Engineers who certify the User's Design Specification shall indicate their qualification as shown in 2-J.3.2(b)(2).

(c) Unless otherwise modified by the Manufacturer as stated in their Quality Control Manual, certification of pressure vessel design competence qualification expires for all design activities when no single design activity has occurred within a continuous period of 36 months.

2-J.4.2 ENGINEERS AND DESIGNERS

(a) The Manufacturer who employs (directly or by contract) the engineer or designer who certifies the Manufacturer's Design Report shall prepare a statement that the individual is qualified to perform the design activities used.

(b) Unless otherwise modified by the Manufacturer as stated in their Quality Control Manual, certification of pressure vessel design competence qualification expires for all Code activities when no single design activity has occurred within a continuous period of 36 months.

2-J.4.3 REACTIVATION

Certification may be reactivated by either of the following methods:

(a) continuity of the design activity for a 6-month period

(b) completion of eight or more professional development hours (PDHs) consisting of one or more of the following activities:

(1) taught or attended an appropriate course, training program, or seminar covering the design topic

(2) attended a technical society meeting related to the topic

PART 3
MATERIALS REQUIREMENTS

3.1 GENERAL REQUIREMENTS

The requirements for materials used in the construction of pressure vessel parts according to the rules of this Division are defined in this Part. General rules and supplemental requirements are defined for different material types and product forms. In cases of conflicts, the requirements stipulated in the paragraphs containing "Supplemental Requirements" shall govern.

3.2 MATERIALS PERMITTED FOR CONSTRUCTION OF VESSEL PARTS

3.2.1 MATERIALS FOR PRESSURE PARTS

3.2.1.1 Materials used for the construction of pressure parts shall conform to one of the specifications given in Section II, and shall be limited to those material specifications shown in the allowable design stress tables in Annex 3-A unless specifically allowed by other rules of this Division.

3.2.1.2 Materials outside the limits of size, thickness, or weight limits stipulated in the title or scope clause of the material specification given in Section II and permitted by 3.2.1.1 may be used if the material is in compliance with the other requirements of the specification and a size, thickness, or weight limitation is not given in the allowable design stress table (see Annex 3-A) or in Table 7.2. For specifications in which chemical composition or mechanical properties vary with size or thickness, materials outside the range shall be required to conform to the composition and mechanical properties shown for the nearest specified range.

3.2.1.3 Materials shall be proven of weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof.

3.2.1.4 Materials for which fatigue curves are provided (see 3.15) shall be used in construction of vessels or vessel parts subject to fatigue unless the fatigue analysis exemption criteria of 5.5.2 are satisfied.

3.2.1.5 Materials other than those allowed by this Division shall not be used unless data therein are submitted to and approved by the Boiler and Pressure Vessel Committee in accordance with Section II, Part D, Mandatory Appendix 5.

3.2.1.6 The rules in this Division do not provide detailed requirements for selection of an alloy suitable for the intended service or the amount of corrosion allowance to be provided. It is required that the user or his designated agent assure the materials used for the construction of vessels or vessel parts are suitable for the intended service conditions with respect to mechanical properties, resistance to corrosion, erosion, oxidation, and other damage mechanisms anticipated during service life. Informative and nonmandatory guidance regarding metallurgical phenomena that occur in material subject to certain process environments is provided in Section II, Part D, Nonmandatory Appendix A.

3.2.1.7 The material specifications listed in Annex 3-A of this Division include a column of UNS (Unified Numbering System) numbers assigned to identify the various alloy compositions. These numbers are used in the rules of this Division whenever reference is made to materials of approximately the same chemical composition that are furnished under more than one approved specification or in more than one product form.

3.2.2 MATERIALS FOR ATTACHMENTS TO PRESSURE PARTS

3.2.2.1 Except as permitted in 3.2.2.2, materials for non-pressure parts which are welded to pressure parts shall meet all the requirements of 3.2.1 and all supplemental requirements stipulated in this Part [see 2.2.3.1(g)].

3.2.2.2 Except as limited in 3.5 for quenched and tempered steels, or by 6.7 for forged vessel construction where welding is not permitted, minor attachments may be of a non-ASME material and may be welded directly to the pressure part, provided the criteria listed below are satisfied. In this context, minor attachments are parts of small size [i.e., not over 10 mm ($^3/_8$ in.) thick or 80 cm^3 (5 in.^{3}) volume] that support no load or insignificant loads (i.e., stress calculations are not required in the Manufacturer's judgment), such as name plates, insulation supports, and locating lugs.

(a) The material is identified and is suitable for welding. Satisfactory qualification of welding procedure under Section IX is considered as proof.

(b) The material is compatible insofar as welding is concerned with that to which the attachment is to be made.

(c) The welds are postweld heat treated when required by 6.4.2 of this Division.

3.2.3 WELDING MATERIALS

3.2.3.1 Welding materials used for the construction of pressure parts shall comply with the requirements of this Division, those of Section IX, and the applicable qualified welding procedure specification.

3.2.3.2 When the welding materials comply with one of the specifications in Section II, Part C, the marking or tagging of the material, containers, or packages as required by the applicable Section II specification may be adopted for identification in lieu of a Test Report or a Certificate of Compliance. When the welding materials do not comply with one of the specifications of Section II, the marking or tagging shall be identifiable with the welding materials set forth in the welding procedure specification, and may be acceptable in lieu of a Test Report or a Certificate of Compliance.

3.2.4 DISSIMILAR MATERIALS

3.2.4.1 The user or his designated agent shall ensure that the coupling of dissimilar materials will not have a detrimental effect on the corrosion rate or service life of the vessel (see Section II, Part D, Nonmandatory Appendix A).

3.2.4.2 The requirements for the base metals, heat-affected zones (HAZ), and weld metals of weldments between metals having different impact testing requirements and acceptance criteria shall be applied in accordance with the rules of this Division.

3.2.5 PRODUCT SPECIFICATIONS

3.2.5.1 The term plate as used in this Division also includes sheet and strip.

3.2.5.2 See below. (**21**)

(a) Rods and Bars Used for Pressure Parts. Rods and bars may be used in pressure vessel construction for pressure parts such as flange rings [see 4.16.4.3(a)], stiffening rings, frames for reinforced openings, stays and staybolts, and similar parts.

(b) Parts Machined From Rod and Bar. Pressure parts such as hollow, cylindrically shaped parts, heads, caps, flanges, elbows, return bends, tees, and header tees may be machined directly from rod or bar as provided below.

(1) Examination by the magnetic particle or liquid penetrant method in accordance with the requirements of Part 7 shall be as follows:

(-a) for flanges: the back of the flange and outer surface of the hub

(-b) for heads, caps, elbows, return bends, tees, and header tees: all surfaces

(-c) for hollow, cylindrically shaped parts: no surface examination needed

(2) Parts may be machined from rod or bar having a hot-worked diameter not greater than 140 mm (5.50 in.), provided that the axial length of the part is approximately parallel to the metal flow lines of the stock.

(3) Parts may be machined from rod or bar having a hot-worked diameter greater than 140 mm (5.50 in.), but not greater than 205 mm (8.00 in.), provided the axial length of the part is approximately parallel to the metal flow lines of the stock, and the minimum required thickness of the component is calculated following the rules of this Division using 50% of the specified allowable stress.

(4) As an alternative to (3) and for rod or bar having a hot-worked diameter greater than 205 mm (8.00 in.), parts may be machined from such rod or bar if the following requirements are met:

(-a) The longitudinal axis of the part shall be parallel to the longitudinal axis of the rod or bar.

(-b) In addition to the tension test specimens required by the material specification, at least two transverse tension test specimens shall be taken from each lot (as defined in the material specification) of rod or bar material, and having the same diameter.

(-1) The second specimen shall be taken at 90 deg around the perimeter from the first specimen.

(-2) The axis of the tension test specimen shall be located, as nearly as practicable, midway between the center thickness and the surface of the rod or bar.

(-3) Both specimens shall meet the mechanical property requirements of the material specification.

(-4) For Table 3-A.1 materials, the reduction of area shall be not less than 30%.

(-c) Each rod or bar, before machining, shall be 100% ultrasonically examined perpendicular to the longitudinal axis by the straight beam technique in accordance with SA-388. The rod or bar shall be unacceptable if either of the following occurs:

(-1) The examination results show one or more indications accompanied by loss of back reflection larger than 60% of the reference back reflection.

(-2) The examination results show indications larger than 40% of the reference back reflection when accompanied by a 40% loss of back reflection.

(-d) For heads and the flat portion of caps, the examinations of (-c) shall also be performed in the axial direction.

(-e) Before welding, the cut surfaces of the part adjacent to the weld shall be examined by magnetic particle or liquid penetrant methods in accordance with Part 7.

3.2.5.3 When a material specification is not listed in this Division covering a particular wrought product of a grade, but there is an approved specification listed in this Division covering some other wrought product of that grade, the product for which there is no specification listed may be used, provided:

(a) The chemical and mechanical properties, heat treating requirements, and requirements for deoxidation, or grain size requirements conform to the approved specification listed in this Division. The stress values for that specification given in Annex 3-A shall be used.

(b) The material specification is published Section II and covers that grade.

(c) For the case of welded product forms without the addition of filler metal, the appropriate stress intensity values are multiplied by 0.85.

(d) The product is not fabricated by fusion welding with the addition of filler metal unless it is fabricated in accordance with the rules of this Division as a pressure part.

(e) The mill test reports reference the specifications used in producing the material and in addition make reference to this paragraph.

3.2.5.4 Forgings certified to SA-105, SA-181, SA-182, SA-350, SA-403, and SA-420 may be used as tubesheets and hollow cylindrical forgings for pressure vessel shells that otherwise meet all the rules of this Division, provided that the following additional requirements are met:

(a) Forgings certified to SA-105 or SA-181 shall be subject to one of the austenitizing heat treatments permitted by these specifications.

(b) One tension test specimen shall be taken from each forging weighing more than 2 250 kg (5,000 lb). The largest obtainable tension test specimen as specified by the test methods referenced in the applicable specification shall be used. Except for upset-disk forgings, the longitudinal axis of the test specimen shall be taken parallel to the direction of major working of the forging. For upset-disk forgings, the longitudinal axis of the test specimen shall be taken in the tangential direction. When agreed to by the Manufacturer, and when not prohibited by the material specification, test specimens may be machined from specially forged test blocks meeting the provisions for such as provided in SA-266 or other similar specifications for large forgings.

(c) For quenched and tempered forgings weighing more than 4 500 kg (10,000 lb) at the time of heat treatment, two tension test specimens shall be taken from each forging. These shall be offset 180 deg from each other, except if the length of the forging, excluding test prolongations, exceeds 3.7 m (12 ft); then one specimen shall be taken from each end of the forging.

3.2.6 CERTIFICATION

3.2.6.1 Certificate of Compliance and Material Test Report.

(a) The Manufacturer shall ensure all requirements of the material specification, and all special requirements of Part 3 of this Division, that are to be fulfilled by the materials manufacturer have been complied with. The Manufacturer shall accomplish this by obtaining Certificates of Compliance or Material Test Reports. These documents shall include results of all required tests and examinations, evidence of compliance with the material specifications and additional requirements as applicable. When the specification permits certain specific requirements to be completed later, those incomplete items shall be noted on the material documentation. When these specific requirements have been completed by someone other than the material manufacturer, this completion shall be documented and attached to the material documentation.

(b) For plates, the Manufacturer shall receive a copy of the test report or reports as prepared by the material manufacturer or by the material manufacturer and subsequent processors, if any, responsible for the data, and shall maintain the reports as part of his construction records.

(c) For all other product forms, the Manufacturer shall receive a copy of the test report as prepared by the material manufacturer. When preparing a test report, a material manufacturer may transcribe data produced by other organizations, provided he accepts responsibility for the accuracy and authenticity of the data.

(d) All conflicts between the material specification and the supplemental requirements stipulated in this Part shall be noted, and compliance with the supplemental requirements shall be certified.

3.2.6.2 Certificate of Compliance and Material Test Reports by Other Than Materials Manufacturer.

(a) Except as otherwise provided in 3.2.5.3 and 3.2.7, if the requirements in a material specification listed in Annex 3-A have been completed by other than the materials manufacturer, then the vessel Manufacturer shall obtain supplementary material test reports and the Inspector shall examine these documents and determine that they represent the material and meet the requirements of the material specification.

(b) The vessel Manufacturer shall certify compliance with all the supplemental requirements stipulated in this Part for any of the treatments or examinations specified herein. The certification shall include certified reports of results of all tests and examinations performed on the materials by the vessel Manufacturer.

3.2.7 PRODUCT IDENTIFICATION AND TRACEABILITY

3.2.7.1 General Requirements.

(a) Material for pressure parts shall be organized so that when the vessel is completed, one complete set of the original identification markings required in the specifications for all materials of construction will be clearly visible. In case the original identification markings are unavoidably cut out or the material is divided into two or more parts, the vessel Manufacturer shall assure identification of each piece of material during fabrication and subsequent identification of the markings on the completed vessel by using the methods listed below.

(1) Accurate transfer of the original identification markings to a location where the markings will be visible on the completed vessel.

(2) Identification by coded marking, described in the Quality System Manual, acceptable to the Inspector and traceable to the original required marking.

(b) An as-built sketch or tabulation of materials shall be made, identifying each piece of material with a test report or, where permitted by this Part, with a Certificate of Compliance and the coded marking that ensure identification of each piece of material during fabrication and subsequent identification in the completed vessel.

(c) When plate specification heat treatments are not performed by the material manufacturer, they shall be performed by, or under the control of, the vessel Manufacturer who shall then place the letter "T" following the letter "G" in the Mill plate marking (see SA-20) to indicate that the heat treatments required by the material specification have been performed. The fabricator shall also document in accordance with 3.2.6.2(b) that the specified heat treatments have been performed in accordance with the material manufacturer's recommendation.

3.2.7.2 Method of Transferring Markings by the Manufacturer.

(a) Transfer of markings shall be made prior to cutting except that the Manufacturer may transfer markings immediately after cutting, provided the control of these transfers is described in the Manufacturer's written Quality Control System. The Inspector need not witness the transfer of the marks but shall be satisfied that it has been done correctly.

(b) The material may be marked by any method acceptable to the Inspector; however, all steel stamping shall be done with commercially available "low stress" dies.

(c) Where the service conditions prohibit die-stamping for material identification, and when so specified by the user, the material manufacturer and the vessel Manufacturer shall mark the required data on the plates in a manner which will allow positive identification upon delivery. The markings must be recorded so that each plate will be positively identified in its position in the completed vessel to the satisfaction of the Inspector.

3.2.7.3 Transfer of Markings by Other Than the Manufacturer.

(a) When material is to be formed into shapes by anyone other than the Manufacturer of the completed pressure vessel and the original markings as required by the applicable material specification are unavoidably cut out, or the material is divided into two or more parts, the manufacturer of the shape shall either:

(1) Transfer the original identification markings to another location on the shape.

(2) Provide for identification by the use of a coded marking traceable to the original required marking, using a marking method agreed upon and described in the Quality Control System of the Manufacturer of the completed pressure vessel.

(b) The mill certification of the mechanical and chemical properties requirements of the material formed into shapes, in conjunction with the above modified marking requirements, shall be considered sufficient to identify these shapes. Manufacturer's Partial Data Reports and parts stamping are not required unless there has been fabrication of the shapes that include welding, except as exempted by 3.2.8.2.

3.2.7.4 Marking of Plates. The material manufacturer's identification marking required by the material specification shall not be stamped on plate material less than 6 mm ($^1/_4$ in.) in thickness unless the following requirements are met.

(a) The materials shall be limited to P-No. 1 Group Nos. 1 and 2.

(b) The minimum nominal plate thickness shall be 5 mm ($^3/_{16}$ in.) or the minimum nominal pipe wall thickness shall be 4 mm (0.154 in.).

(c) The MDMT shall be no colder than −29°C (−20°F).

3.2.8 PREFABRICATED OR PREFORMED PRESSURE PARTS FURNISHED WITHOUT A CODE STAMP

3.2.8.1 General Requirements.

(a) Prefabricated or preformed pressure parts for pressure vessels that are subject to stresses due to pressure and that are furnished by others or by the Manufacturer of the completed vessel shall conform to all applicable requirements of this Division except as permitted in 3.2.8.2, 3.2.8.3, 3.2.8.4, and 3.2.8.5.

(b) When the prefabricated or preformed parts are furnished with a nameplate that contains product identifying marks and the nameplate interferes with further fabrication or service, and where stamping on the material is prohibited, the Manufacturer of the completed vessel, with the concurrence of the Authorized Inspector, may remove the nameplate. The removal of the nameplate shall be noted in the "Remarks" section of the vessel Manufacturer's Data Report. The nameplate shall be destroyed.

(c) The rules of 3.2.8.2, 3.2.8.3, 3.2.8.4, and 3.2.8.5 below shall not be applied to welded shells or heads or to quick-actuating closures (4.8).

(d) Parts furnished under the provisions of 3.2.8.2, 3.2.8.3, or 3.2.8.4 need not be manufactured by a Certificate of Authorization Holder.

(e) Prefabricated or preformed pressure parts may be supplied as follows:

(1) cast, forged, rolled, or die-formed nonstandard pressure parts

(2) cast, forged, rolled, or die-formed standard pressure parts, either welded or nonwelded, that comply with an ASME product standard

(3) cast, forged, rolled, or die-formed standard pressure parts, either welded or nonwelded, that comply with a standard other than an ASME product standard

3.2.8.2 Cast, Forged, Rolled, or Die-Formed Nonstandard Pressure Parts.

(a) Pressure parts such as shells, heads, removable doors, and pipe coils that are wholly formed by casting, forging, rolling, or die forming may be supplied basically as materials. All such parts shall be made of materials permitted under this Division, and the Manufacturer of the part shall furnish identification in accordance with 3.2.6.1.

Such parts shall be marked with the name or trademark of the parts manufacturer and with such other markings as will serve to identify the particular parts with accompanying material identification.

(b) The Manufacturer of the completed vessel shall be satisfied that the part is suitable for the design conditions specified for the completed vessel in accordance with the rules of this Division.

3.2.8.3 Cast, Forged, Rolled, or Die-Formed Standard Pressure Parts, Either Welded or Nonwelded, That Comply With an ASME Product Standard.

(a) Pressure parts that comply with an ASME product standard accepted by reference in 4.1.11. The ASME product standard establishes the basis for the pressure–temperature rating and marking unless modified in 4.1.11.

(b) Flanges and flanged fittings may be used at the pressure–temperature ratings specified in the appropriate standard listed in this Division.

(c) Materials for standard pressure parts shall be as follows:

(1) as permitted by this Division or

(2) as specifically listed in the ASME product standard

(d) When welding is performed it shall meet the following:

(1) the requirements of 6.2.2.1(a) and 6.2.2.2 through 6.2.2.5, or;

(2) the welding requirements of SA-234

(e) Pressure parts, such as welded standard pipe fittings, welding caps, and flanges that are fabricated by one of the welding processes recognized by this Division do not require inspection, material certification in accordance with 3.2.6, or Partial Data Reports, provided the requirements of 3.2.8.3 are met.

(f) If postweld heat treatment is required by the rules of this Division, it may be performed either in the location of the parts manufacturer or in the location of the Manufacturer of the vessel to be marked with the Certification Mark.

(g) If radiography or other volumetric examination is required by the rules of this Division, it may be performed at one of the following locations:

(1) the location of the Manufacturer of the completed vessel

(2) the location of the pressure parts manufacturer

(h) Parts made to an ASME product standard shall be marked as required by the ASME product standard.

(i) The Manufacturer of the completed vessels shall have the following responsibilities when using standard pressure parts that comply with an ASME product standard:

(1) Ensure that all standard pressure parts comply with applicable rules of this Division.

(2) Ensure that all standard pressure parts are suitable for the design conditions of the completed vessel.

(3) When volumetric examination is required by the rules of this Division, obtain the completed radiographs, properly identified, with a radiographic inspection report, and any other applicable volumetric examination report.

(j) The Manufacturer shall fulfill these responsibilities by obtaining when necessary, documentation as provided below, provide for retention of this documentation, and have such documentation available for examination by the Inspector when requested. The documentation shall contain at a minimum:

(1) material used

(2) the pressure–temperature rating of the part

(3) The basis for establishing the pressure–temperature rating

3.2.8.4 Cast, Forged, Rolled, or Die-Formed Standard Pressure Parts, Either Welded or Nonwelded, That Comply With a Standard Other Than an ASME Product Standard.

(a) Standard pressure parts that are either welded or nonwelded and comply with a manufacturer's proprietary standard or a standard other than an ASME product standard may be supplied by

(1) a Certificate of Authorization holder

(2) a pressure parts manufacturer

(b) Parts of small size falling within this category for which it is impossible to obtain identified material or that may be stocked and for which material certification in accordance with 3.2.6 cannot be obtained and are not customarily furnished, may be used for parts as described in 3.2.2.2.

(c) Materials for these parts shall be as permitted by this Division only.

(d) When welding is performed, it shall meet the requirements of 6.2.2.1(a) and 6.2.2.2 through 6.2.2.5.

(e) Pressure parts, such as welded standard pipe fittings, welding caps, and flanges that are fabricated by one of the welding processes recognized by this Division do not require inspection, material certification in accordance with 3.2.6, or Partial Data Reports provided the requirements of 3.2.8.4 are met.

(f) If postweld heat treatment is required by the rules of this Division, it may be performed either in the location of the parts manufacturer or in the location of the Manufacturer of the completed vessel.

(g) If radiography or other volumetric examination is required by the rules of this Division, it may be performed at one of the following locations:

(1) The location of the Manufacturer of the completed vessel

(2) The location of the parts Manufacturer

(3) The location of the pressure parts manufacturer

(h) Marking for these parts shall be as follows:

(1) with the name or trademark of the Certificate Holder or the pressure part manufacturer and any other markings as required by the proprietary standard or other standard used for the pressure part.

(2) with a permanent or temporary marking that will serve to identify the part with the Certificate Holder or the pressure parts manufacturer's written documentation of the particular items, and that defines the pressure–temperature rating of the part.

(i) The Manufacturer of the completed vessels shall have the following responsibilities when using standard pressure parts:

(1) Ensure that all standard pressure parts comply with applicable rules of this Division

(2) Ensure that all standard pressure parts are suitable for the design conditions of the completed vessel.

(3) When volumetric examination is required by the rules of this Division, obtain the completed radiographs, properly identified, with a radiographic inspection report, and any other applicable volumetric examination report.

(j) The Manufacturer of the completed vessel shall fulfill the responsibilities of (i) by one of the following methods:

(1) Obtain when necessary, documentation as provided below, provide for retention of this documentation, and have such documentation available for examination by the Inspector when requested or

(2) Perform an analysis of the pressure part in accordance with the rules of this Division. This analysis shall be included in the documentation and shall be made available for examination by the Inspector when requested.

(k) The documentation shall contain at a minimum the following:

(1) material used

(2) the pressure–temperature rating of the part

(3) the basis for establishing the pressure–temperature rating

(4) a written certification by the pressure parts manufacturer that all welding complies with Code requirements

3.2.8.5 The Code recognizes that a Certificate of Authorization Holder may fabricate parts in accordance with 3.2.8.4, and that are marked in accordance with 3.2.8.4(h). In lieu of the requirement in 3.2.8.4(d), the Certificate of Authorization Holder may subcontract to an individual or organization not holding an ASME Certificate of Authorization standard pressure parts that are fabricated to a standard other than an ASME product standard, provided all the following conditions are met:

(a) The activities to be performed by the subcontractor are included within the Certificate Holder's Quality Control System.

(b) The Certificate Holder's Quality Control System provides for the following activities associated with subcontracting of welding operations, and these provisions shall be acceptable to the Manufacturer's Authorized Inspection Agency.

(1) The welding processes permitted by this Division that are permitted to be subcontracted.

(2) Welding operations

(3) Authorized Inspection activities

(4) Placement of the Certificate of Authorization Holders marking in accordance with (d).

(c) The Certificate Holder's Quality Control System provides for the requirements of 7.2.2 to be met at the subcontractor's facility.

(d) The Certificate Holder shall be responsible for reviewing and accepting the Quality Control Programs of the subcontractor.

(e) The Certificate Holder shall ensure that the subcontractor uses written procedures and welding operations that have been qualified as required by this Division.

(f) The Certificate Holder shall ensure that the subcontractor uses personnel that have been qualified as required by this Division.

(g) The Certificate Holder and the subcontractor shall describe in their Quality Control Systems the operational control of procedure and personnel qualifications of the subcontracted welding operations.

(h) The Certificate Holder shall be responsible for controlling the quality and ensuring that all materials and parts that are welded by subcontractors and submitted to the Inspector for acceptance, conform to all applicable requirements of this Division.

(i) The Certificate Holder shall describe in their Quality Control Systems the operational control for maintaining traceability of materials received from the subcontractor.

(j) The Certificate Holder shall receive approval for subcontracting from the Authorized Inspection Agency prior to commencing of activities.

3.2.9 DEFINITION OF PRODUCT FORM THICKNESS

3.2.9.1 The requirements in this Division make reference to thickness. When the material specification does not specify thickness, the following definitions of nominal thickness apply.

(a) Plate – the thickness is the dimension of the short transverse dimension.

(b) Forgings – the thickness is the dimension defined as follows:

(1) Hollow Forgings – the nominal thickness is measured between the inside and the outside surfaces (radial thickness).

(2) Disk Forgings– the nominal thickness is the axial length (axial length ≤ outside the diameter).

(3) Flat Ring Forgings – for axial length less than or equal to 50 mm (2 in.), the axial length is the nominal thickness; for axial length greater than 50 mm (2 in.), the radial thickness is the nominal thickness (axial length less than the radial thickness).

(4) Rectangular Solid Forgings – the least rectangular dimension is the nominal thickness.

(5) Round, Hexagonal and Octagonal Solid Forgings - the nominal thickness is the diameter or distance across the flats (axial length > diameter or distance across the flats).

(c) Castings – for castings of the general shapes described for forgings, the same definitions apply. For other castings, the maximum thickness between two cast coincidental surfaces is the nominal thickness.

3.2.9.2 The definition of nominal thickness for postweld heat treat requirements is covered in 6.4.2.7.

3.2.10 PRODUCT FORM TOLERANCES

3.2.10.1 Plate. Plate material shall be ordered not thinner than the design thickness. Vessels made of plate furnished with an undertolerance of not more than the smaller value of 0.3 mm (0.01 in.) or 6% of the ordered thickness may be used at the full design pressure for the thickness ordered if the material specification permits such an undertolerance. If the specification to which the plate is ordered allows a greater undertolerance, the ordered thickness of the material shall be sufficiently greater than the design thickness so that the thickness of the material furnished is not more than the smaller of 0.3 mm (0.01 in.) or 6% under the design thickness.

3.2.10.2 Pipe and Tube. If pipe or tube is ordered by its nominal wall thickness, the manufacturing undertolerance on wall thickness shall be taken into account. After the minimum required wall thickness is determined, it shall be increased by an amount sufficient to provide the manufacturing undertolerance allowed in the pipe or tube specification.

3.2.11 PURCHASE REQUIREMENTS

3.2.11.1 A summary of the pertinent requirements in 3.2 through 3.8 is provided in Annex 3-B.

3.2.11.2 Special chemical compositions, heat treatment procedures, fabrication requirements, and supplementary tests may be required to assure that the vessel will be in the most favorable condition for the intended service.

3.2.12 MATERIAL IDENTIFIED WITH OR PRODUCED TO A SPECIFICATION NOT PERMITTED BY THIS DIVISION

3.2.12.1 Identified Material With Complete Certification From the Material Manufacturer. Material identified with a specification not permitted by this Division and identified to a single production lot as required by a permitted specification may be accepted as satisfying the requirements of a specification permitted by this Division, provided the following conditions are satisfied:

(a) Documentation is provided to the Certificate Holder demonstrating that all applicable requirements (including, but not limited to, melting method, melting practice, deoxidation, chemical analysis, mechanical properties, quality, and heat treatment) of the specification permitted by this Division to which the material is to be recertified, including the requirements of this Division (see 3.2.6), have been met.

(b) The material has marking, acceptable to the Inspector, for identification to the documentation.

(c) When the conformance of the material with the permitted specification has been established, the material shall be marked as required by the permitted specification.

3.2.12.2 Identified Material Recertification. Only the vessel or Part Manufacturer is permitted to recertify material per 3.2.12.1.

3.3 SUPPLEMENTAL REQUIREMENTS FOR FERROUS MATERIALS

3.3.1 GENERAL

All forms of ferrous products listed in Table 3-A.1 and Table 3-A.3 shall meet the supplemental requirements of 3.3. The high strength quenched and tempered steels listed in Table 3-A.2, shall meet the supplemental requirements of 3.4.

3.3.2 CHEMISTRY REQUIREMENTS

Carbon and low alloy steel having carbon content of more than 0.35% by heat analysis shall not be used in welded construction or be shaped by oxygen cutting (except as provided elsewhere in this Division).

3.3.3 ULTRASONIC EXAMINATION OF PLATES

3.3.3.1 Except as permitted in 3.3.3.2, all plate 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined in accordance with the requirements of SA-578. The acceptance standard shall be Level B of SA-578.

3.3.3.2 When the design rules permit credit for thickness of cladding on plate conforming to SA-263, SA-264, and SA-265, ultrasonic examination shall be made of the base plate and the bond between the cladding and the base plate in accordance with the requirements of SA-578. The acceptance standard shall be at least Level B of SA-578. Alternatively, the acceptance standard of Level C may be used to satisfy this requirement.

3.3.4 ULTRASONIC EXAMINATION OF FORGINGS

3.3.4.1 All forgings 50 mm (2 in.) and over in nominal thickness shall be examined ultrasonically as follows: (**21**)

(a) Rings, flanges, and other hollow forgings shall be examined using the angle beam technique. For other forgings, the straight beam technique shall be used.

(b) Reference specimens shall have the same nominal thickness, composition, and P-number grouping as the forgings to be examined in order to have substantially the same structure.

(c) Tables 3.1, 3-A.1, and 3-A.2 steels shall be examined in accordance with Section V, SA-388.

(d) Table 3-A.3 steels shall be examined in accordance with Section V, SA-388 or Section V, SA-745, as applicable.

3.3.4.2 Forgings are unacceptable if:

(a) The straight beam examination results show one or more discontinuities which produce indications accompanied by a complete loss of back reflection not associated with or attributable to the geometric configuration.

(b) Angle beam examination results show one or more discontinuities which produce indications exceeding in amplitude the indication from the calibration notch.

3.3.4.3 In the case of straight beam examination, the following conditions shall be reported to the purchaser for his consideration and approval prior to shipment of the forging:

(a) Forgings containing one or more indications with amplitudes exceeding adjacent back reflections.

(b) Forgings containing one or more discontinuities which produce traveling indications accompanied by reduced back reflections. A traveling indication is defined as an indication that displays sweep movement of the oscilloscope screen at constant amplitudes as the transducer is moved.

3.3.4.4 In the case of angle beam examination, the following conditions shall be reported to the purchaser for his consideration and approval prior to shipment of the forging:

(a) Indications having an amplitude exceeding 50% of the calibration block amplitude.

(b) Clusters of indications located in a small area of the forging with amplitudes less than 50% of the calibration notch amplitude. A cluster of indications is defined as three or more indications exceeding 10% of the standard calibration notch amplitude and located in any volume approximately a 50 mm (2 in.) or smaller cube.

3.3.4.5 Additional nondestructive examination procedures or trepanning may be employed to resolve questions of interpretation of ultrasonic indications.

3.3.5 MAGNETIC PARTICLE AND LIQUID PENETRANT EXAMINATION OF FORGINGS

3.3.5.1 Following final machining by the manufacturer, all accessible surfaces of forgings having a nominal thickness greater than 100 mm (4 in.), such as contour and variable-thickness nozzles, integrally hubbed tubesheets, standard or custom flanges, and other forgings that are contour shaped or machined to essentially the finished product configuration prior to heat treatment, shall be examined by the magnetic particle method in accordance with ASTM A275/A275M or by the liquid penetrant method in accordance with ASTM E165. The evaluation of indications detected by the magnetic particle method or by the liquid penetrant method and the acceptance standards shall be in accordance with Part 7 of this Division.

3.3.5.2 Unacceptable imperfections shall be removed and the areas shall be reexamined to ensure complete removal of the unacceptable imperfection. Unless prohibited by the material specification, the forgings may be repair welded with the approval of the vessel Manufacturer. Repairs shall be made utilizing welding procedures that have been qualified in accordance with Section IX. The repaired forging shall meet all requirements of this Division.

3.3.6 INTEGRAL AND WELD METAL OVERLAY CLAD BASE METAL

3.3.6.1 Applied Linings. Material used for applied corrosion resistant lining may be any metallic material of weldable quality, provided all applicable requirements of this Division are satisfied.

3.3.6.2 Design Calculations Based on Total Thickness.

(a) Base material with corrosion resistant integral or weld metal overlay cladding used in construction in which the design calculations are based on total thickness including cladding (4.1.9) shall consist of base plate listed in one of the material tables in Part 3 and shall conform to one of the following specifications or utilize weld metal overlay cladding meeting the requirements of this Division.

(1) SA-263, Specification for Corrosion-Resisting Chromium-Steel Clad Plate, Sheet and Strip;

(2) SA-264, Specification for Corrosion-Resisting Chromium-Nickel Steel Clad Plate, Sheet and Strip; or

(3) SA-265, Specification for Nickel and Nickel-Base Alloy Clad Steel Plate.

(b) Base material with corrosion resistant integral cladding in which any part of the cladding is included in the design calculations, as permitted in (a), that is constructed of multiple cladding plates welded together prior being bonded to the base material shall have the cladding-alloy-to-cladding-alloy welding that is performed prior to bonding to the base material:

(1) performed by a Manufacturer holding a Certificate of Authorization.

(2) radiographically examined for their full length in the manner prescribed in 7.5.3. In place of radiographic examination, welds may be ultrasonically examined for their full length (see 7.5.5).

(3) be supplied with a Partial Data Report if that welding is not performed by the vessel Manufacturer.

3.3.6.3 Design Calculations Based on Base-Plate Thickness. Clad plate used in constructions in which the design calculations are based on the base-plate thickness, exclusive of the thickness of the cladding material, may consist of any base-plate material satisfying the requirements of Part 3 and any metallic integral or weld metal overlay cladding material of weldable quality that meets the requirements of 6.5 of this Division.

3.3.6.4 Shear Strength of Bond of Integrally Clad Plates. Integrally clad plates in which any part of the cladding is included in the design calculations, as permitted in 4.1.9, shall show a minimum shear strength of 140 MPa (20 ksi) when tested in the manner described in the plate specification. One shear test shall be made on each such clad plate and the results shall be reported on the test report. A shear or bond strength test is not required for weld metal overlay cladding.

3.3.6.5 Removal of Cladding for Mill Tension Tests. When any part of the cladding thickness is specified an allowance for corrosion, such added thickness shall be removed before mill tension tests.

3.3.7 CLAD TUBESHEETS

3.3.7.1 Tube-to-tubesheet welds in the cladding of either integral or weld metal overlay clad tubesheets may be considered strength welds (full or partial), provided the welds meet the design requirements of 4.18.10. In addition, when the strength welds are to be made in the clad material of integral clad tubesheets, the integral clad material to be used for tubesheets shall meet the requirements in (a) and (b) for any combination of clad and base materials. The shear strength test and ultrasonic examination specified in (a) and (b) are not required for weld metal overlay clad tubesheets.

(a) Integral clad material shall be shear strength tested in accordance with SA-263. One shear test shall be made on each integral clad plate or forging, and the results shall be reported on the material test report.

(b) Integral clad material shall be ultrasonically examined for bond integrity in accordance with SA-578, including Supplementary Requirement S1, and shall meet the acceptance criteria given in SA-263 for Quality Level Class 1.

3.3.7.2 When the design calculations for clad tubesheets are based on the total thickness including the cladding, the clad material shall meet any additional requirements specified in 3.3.6.

3.3.7.3 When tubesheets are constructed using linings or integral cladding that does not meet the requirements of 3.3.7.1(a) and 3.3.7.1(b), the strength of the tube-to-tubesheet joint shall not be dependent upon the connection between the tubes and the lining or integral cladding, as applicable.

3.3.7.4 When the tubes are strength welded (full or partial) to integral or weld metal overlay clad tubesheets, S_t shall be the allowable stress value of the integral cladding or the wrought material whose chemistry most closely approximates that of the weld metal overlay cladding. The thickness of the integral or weld metal clad overlay material shall be sufficient to prevent any of the strength weld from extending into the base material.

3.4 SUPPLEMENTAL REQUIREMENTS FOR Cr–Mo STEELS

3.4.1 GENERAL

3.4.1.1 The rules in 3.4 include supplemental requirements for fabrication and testing for Cr-Mo steels. The materials and appropriate specifications covered by this paragraph are listed in Table 3.1.

3.4.1.2 Certification that the requirements of 3.4 have been satisfied shall be shown on the Manufacturer's Data Report Form.

3.4.2 POSTWELD HEAT TREATMENT

The final postweld heat treatment shall be in accordance with the requirements of 6.4.2 of this Division.

3.4.3 TEST SPECIMEN HEAT TREATMENT

3.4.3.1 Two sets of tension specimens and one set of Charpy impact specimens shall be tested. One set each of the tension specimens shall be exposed to heat treatment Condition A. The second set of tension specimens and the set of Charpy specimens shall be exposed to heat treatment Condition B.

(a) Condition A – Temperature shall be no lower than the actual maximum vessel-portion temperature, less 14°C (25°F). Time at temperature shall be no less than 80% of the actual holding time of the vessel portion exposed to the maximum vessel-portion temperature.

(b) Condition B – Temperature shall be no higher than the actual minimum vessel-portion temperature, plus 14°C (25°F). Time at temperature shall be no more than 120% of the actual hold time of the vessel portion exposed to the minimum vessel-portion temperature.

3.4.3.2 The suggested procedure for establishing the test specimen heat treatment parameters are shown below.

(a) Establish maximum and minimum temperatures and hold times for the vessel/component heat treatment based on experience/equipment;

(b) Determine Conditions A and B for the test specimen heat treatments;

(c) Vessel heat treatment temperature and hold time limitations, and test specimen Conditions A and B, are shown in Figure 3.1.

3.4.4 WELDING PROCEDURE QUALIFICATIONS AND WELDING CONSUMABLES TESTING

(**21**) **3.4.4.1** Welding procedure qualifications using welding consumables of the same classification or trade designation as those to be used in production shall be made for material welded to itself or to other materials. The qualifications shall conform to the requirements of Section IX, and the maximum tensile strength at room temperature shall be 760 MPa (110 ksi) (for heat treatment Conditions A and B).

3.4.4.2 Weld metal from each heat or lot of electrodes and filler-wire–flux combination shall be tested, unless specific heat- or lot-traceable test reports meeting the additional requirements of 3.4 related to welding consumables testing have been provided by the welding consumables manufacturer. The minimum and maximum tensile properties shall be met in postweld heat treated (PWHT) Conditions A and B. The minimum Charpy V-notch impact properties shall be met in PWHT Condition B. Testing shall be in general conformance with SFA-5.5 for covered electrodes and SFA-5.23 for filler wire-flux combinations.

3.4.4.3 Duplicate testing in PWHT Condition A and PWHT Condition B (see 3.4.3) is required. The minimum tensile strength and Charpy impact properties for the base metal shall be met. Charpy impact testing is only required for Condition B.

3.4.4.4 For $2^1/_4$Cr–1Mo–$^1/_4$V material, the weld metal shall meet the compositional requirements listed in Table 3.2. For all other materials, the minimum carbon content of the weld metal shall be 0.05%.

3.4.4.5 In addition for $2^1/_4$Cr–1Mo and $2^1/_4$Cr–1Mo–$^1/_4$V material, Category A welds intended for design temperatures above 440°C (825°F), each heat of filler wire and flux combination used in production shall also be qualified by a weld metal stress-rupture test performed in accordance with ASTM E139 or other equivalent national or international test standard on specimens machined parallel (all weld metal specimens) and transverse to the weld axis (one specimen each), applying the following testing parameters and acceptance criteria:

(a) The specimen diameter within the gage length shall be 13 mm ($^1/_2$ in.) or greater. The specimen centerline shall be located at the 0.25-t thickness location (or closer to the center) for material 19 mm ($^3/_4$ in.) and greater in thickness.

(b) The gage length for the transverse specimen shall include the weld and at least 19 mm ($^3/_4$ in.) of base metal adjacent to the fusion line.

(c) The test material shall be postweld heat treated to Condition A.

(d) For $2^1/_4$Cr–1Mo material, the condition of the stress-rupture test shall be 210 MPa (30 ksi) at 510°C (950°F). The time of failure shall exceed 650 hr.

(e) For $2^1/_4$Cr–1Mo–$^1/_4$V material, the condition of the stress-rupture test shall be 210 MPa (30 ksi) at 540°C (1000°F). The time of failure shall exceed 900 hr.

3.4.5 TOUGHNESS REQUIREMENTS

The minimum toughness requirements for base metal, weld metal, and heat-affected zone, after exposure to the simulated postweld heat treatment Condition B, are shown in Table 3.3. If the material specification or other parts of this Division have more demanding toughness requirements, they shall be met.

3.5 SUPPLEMENTAL REQUIREMENTS FOR Q&T STEELS WITH ENHANCED TENSILE PROPERTIES

3.5.1 GENERAL

3.5.1.1 The supplemental requirements in 3.5 apply to ferritic steels with tensile properties enhanced by quenching and tempering and shall be used in conjunction with the other requirements of this Division. The material specifications for these steels are shown in Table 3-A.2.

3.5.1.2 The requirements of this paragraph are not intended to apply to steels listed in Table 3-A.1 that are furnished in such thicknesses that heat treatment, involving the use of accelerated cooling, including liquid quenching, is used to obtain structures comparable to those attained by normalizing thinner sections.

3.5.2 PARTS FOR WHICH Q&T STEELS MAY BE USED

High strength quenched and tempered steels shown in Table 3-A.2, may be used for the entire vessel or for individual components of vessels that are joined to other grades of quenched and tempered steels, or to other steels conforming to specifications listed in Tables 3-A.1, 3-A.3, and 3-A.6, subject to the requirements and limitations of this Division.

3.5.3 STRUCTURAL ATTACHMENTS

3.5.3.1 Except as permitted in 3.5.3.2 below, all permanent structural attachments other than minor attachments specified in 3.5.3.3 and stiffening rings that are welded directly to pressure parts shall be made of material whose specified minimum yield strength is within ±20% of that of the material to which they are attached.

3.5.3.2 All permanent structural attachments welded directly to a shell or head constructed of a material conforming to SA-333, Grade 8, SA-334, Grade 8, SA-353, SA-522, SA-553, and SA-645 Grade A shall be made from a material covered by these same specifications, or nickel alloys UNS N06625 or N10276, or from wrought non-hardenable austenitic stainless steels. If an austenitic stainless steel is used, consideration should be given to the additional weld stresses resulting from the difference in thermal expansion between the attachment and the shell.

3.5.3.3 If the following conditions are met, the material of minor attachments given in 4.2.5.6(c) may be used. The definition of minor attachments is given in 4.2.5.1(h).

(a) The specified minimum tensile strength of quenched and tempered steel for pressure parts shall be less than 690 MPa (100 ksi).

(b) The specified minimum yield strength of minor attachments shall be within +20% and −60% of that of the material to which they are attached.

(c) If the minor attachment is welded in the area less than $2.5\sqrt{R_m t}$ from any gross structural discontinuity, where R_m is the mean radius of the shell, and t is the thickness of the shell, the stress evaluation in accordance with Part 5 shall be performed.

(d) If the continuous fillet weld is used, the leg dimension of fillet weld shall not be less than $0.25t$, where t is the thickness of the minor attachment.

(e) The effect of differential thermal expansion shall be considered when the thermal expansion coefficient of the minor attachment differs from that of the pressure part to which it is attached.

(f) Welding materials with room-temperature tensile strength equivalent to that of quenched and tempered steels shall be used.

(g) The welds shall be postweld heat treated when required by Part 6.

3.6 SUPPLEMENTAL REQUIREMENTS FOR NONFERROUS MATERIALS

3.6.1 GENERAL

Nonferrous materials covered by 3.6 shall conform to one of the specifications listed in Tables 3-A.4, 3-A.5, 3-A.6, and 3-A.7, and shall be used in conjunction with the other requirements of this Division.

3.6.2 ULTRASONIC EXAMINATION OF PLATES

All plates 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined in accordance with the applicable requirements of the ASTM standards and ASME specifications listed below:

(a) SE-114, Ultrasonic Testing by Reflection Method Using Pulsed Longitudinal Waves Induced by Direct Contact;

(b) E214, Immersed Ultrasonic Testing by the Reflection Method Using Pulsed Longitudinal Waves;

(c) E127, Fabricating and Checking Aluminum Alloy Ultrasonic Standard Reference Blocks;

(d) SB-548, Ultrasonic Testing of Aluminum Plate.

3.6.3 ULTRASONIC EXAMINATION OF FORGINGS

3.6.3.1 Insofar as practicable, all solid rectangular forgings shall be examined by the straight beam technique from two directions at approximately right angles. Hollow forgings including flanges and rings 50 mm (2 in.) and over in nominal thickness shall be examined using the angle beam technique by either the contact method or the immersion method. Reference specimens and acceptance criteria shall be examined from one face or surface normal to the axis in the circumferential direction unless the wall thickness or geometric configuration makes angle beam examination impracticable. Disk forgings shall be examined from one flat side and from the circumferential surface.

3.6.3.2 The entire volume of metal shall be ultrasonically examined at some state of manufacture. For heat-treated material, examination after final heat treatment is preferred, but if the contour of the forging precludes complete examination at this stage, the maximum possible volume of the forging shall be reexamined after the final heat treatment.

3.6.3.3 The method used in the examination of forgings shall conform to the following requirements.

(a) In straight beam examination, the transducers shall be 19 mm to 29 mm ($^3/_4$ in. to $1^1/_8$ in.) in diameter or 25 mm (1 in.) square. The nominal frequency shall be appropriate for the material being examined. The instrument shall be set so that the first back reflection is 75% ± 5% of the screen height when the transducer is placed on the indication-free area of the forging.

(b) In angle beam examination by the contact method, a 25 mm × 25 mm (1 in. × 1 in.) or 25 mm × 38 mm (1 in. × $1^1/_2$ in.), 45 deg. transducer shall be used at an appropriate frequency.

(c) In angle beam examination by the immersion method, a 19 mm ($^3/_4$ in.) diameter transducer oriented at an approximate angle of inclination shall be used at an appropriate frequency.

(d) Angle beam examination shall be calibrated with a notch of a depth equal to the smaller of 10 mm ($^3/_8$ in.) or 3% of the nominal section thickness, a length of approximately 25 mm (1 in.) and width not greater than two times the depth.

3.6.3.4 The material shall be unacceptable (unless repaired in accordance with the rules of this Division) if straight beam examination shows one or more discontinuities which produce indications accompanied by a complete loss of back reflection not associated with or attributable to the geometric configuration, or if angle beam examination results show one or more discontinuities which produce indications exceeding that of the calibration notch.

3.6.4 LIQUID PENETRANT EXAMINATION OF FORGINGS

3.6.4.1 Following final machining by the manufacturer all accessible surfaces of thick and complex forgings, such as contour nozzles, thick tubesheets, flanges, and other complex forgings that are contour shaped or machined to essentially the finished product configuration prior to heat treatment, shall be examined by the liquid penetrant method in accordance with Practice E165.

3.6.4.2 The evaluation of indications detected by the liquid penetrant method and the acceptance standards shall be in accordance with Part 7 of this Division.

3.6.4.3 Unacceptable imperfections shall be removed and the areas shall be reexamined to ensure complete removal of the unacceptable imperfection. Unless prohibited by the material specification, the forgings may be repair welded with the approval of the vessel Manufacturer. Repairs shall be made utilizing welding procedures that have been qualified in accordance with Section IX. The repaired forging shall meet all requirements of this Division.

3.6.5 CLAD PLATE AND PRODUCTS

Clad plate or products used in construction for which the design calculations are based on total thickness, including cladding, shall consist of base plate listed in one of the material tables in this Division and shall conform to one of the following specifications:

(a) SB-209, Specification for Aluminum Alloy Sheet and Plate.

(b) SB-211, Specification for Aluminum Alloy Extruded Bars, Rods, Shapes, and Tubes.

3.6.6 CLAD TUBESHEETS

Clad tubesheets that will contain strength welded tube-to-tubesheet joints in the cladding shall meet the requirements of 3.3.7 and any applicable requirements specified in 3.6.5.

3.7 SUPPLEMENTAL REQUIREMENTS FOR BOLTING

3.7.1 GENERAL

The supplemental requirements in 3.7 are required for all bolts, studs, and nuts supplied with vessels constructed to this Division.

3.7.2 EXAMINATION OF BOLTS, STUDS, AND NUTS

Bolts, studs, and nuts covered by the material specifications listed in Annex 3-A shall be subjected to the following examinations:

(a) All areas of threads, shanks, and heads of final machined parts shall be visually examined. Discontinuities, such as laps, seams, cracks are unacceptable.

(b) All bolts, studs, and nuts over 25 mm (1 in.) nominal bolt size shall be examined by the magnetic particle method or by the liquid penetrant method in accordance with Part 7 of this Division. This examination shall be performed on the finished component after threading or on the material stock at approximately the finished diameter before threading and after heading (if involved). Linear non-axial indications are unacceptable. Linear indications greater than 25 mm (1 in.) in length are unacceptable.

(c) All bolts, studs, and nuts greater than 50 mm (2 in.) nominal bolt size shall be ultrasonically examined over the entire surface prior to threading in accordance with the following requirements:

(1) Examination shall be carried out by the straight beam, radial scan method.

(2) Examination shall be performed at a nominal frequency of 2.25 MHz with the search unit not to exceed 645 mm^2 (1 $in.^2$) in area.

(3) Calibration sensitivity shall be established by adjustment of the instrument so that the first back screen reflection is 75% to 90% of full screen height.

(4) Any discontinuity which causes an indication in excess of 20% of the height of the first back reflection or any discontinuity which prevents the production of the first back reflection of 50% of the calibration amplitude is not acceptable.

(d) All bolts, studs, and nuts greater than 100 mm (4 in.) nominal bolt size shall be ultrasonically examined over an entire end surface before or after threading in accordance with the following requirements:

(1) Examination shall be carried out by the straight beam, longitudinal scan method.

(2) Examination shall be performed at a nominal frequency of 2.25 MHz with the search unit not to exceed 320 mm^2 (0.5 $in.^2$) in area.

(3) Calibration shall be established on a test bar of the same nominal composition and diameter as the production part and a minimum of one half of the length. A 10 mm ($^3/_8$ in.) diameter × 76 mm (3 in.) deep flat bottom hole shall be drilled in one end of the bar and plugged to full depth. A distance amplitude correction curve shall be established by scanning from both ends of the test bar.

(4) Any discontinuity which causes an indication in excess of that produced by the calibration hole in the reference specimen as corrected by the distance amplitude correction curve is not acceptable.

3.7.3 THREADING AND MACHINING OF STUDS

3.7.3.1 Studs shall be threaded the full length, or shall be machined down to the root diameter of the thread in the unthreaded portion, provided that the threaded portions are at least 1.5 diameters in length.

3.7.3.2 Studs greater than 8 diameters in length may have an unthreaded portion which has the nominal diameter of the thread, provided the following requirements are met:

(a) The threaded portion shall be at least 1.5 diameters in length.

(b) The stud shall be machined down to the root diameter of the thread for a minimum distance of 0.5 diameters adjacent to the threaded portion.

(c) Suitable transition shall be provided between the root diameter and the unthreaded portion.

(d) Particular consideration shall be given to any dynamic loadings.

3.7.4 USE OF WASHERS

When washers are used in conjunction with torquing methods (e.g., the use of manual or hydraulic torque wrenches) for the purpose of bolt tightening, they shall be designed to provide a smooth and low-friction contact surface for the nuts, which are important considerations when torquing methods are used for bolt tightening.

NOTE: Flat washers typically should be 6 mm ($^1/_4$ in.) thick and made of through-hardened, wrought low alloy steel. See ASME PCC-1 for more information.

3.7.5 FERROUS BOLTING

3.7.5.1 Material for Ferrous Bolting.

(a) Approved specifications for ferrous bolting are given in Annex 3-A, Tables 3-A.8, 3-A.9, 3-A.10 and 3-A.11.

(b) High alloy steel studs, bolts, and nuts may be used with carbon and low alloy steel components, provided they are suitable for the application (see Section II, Part D, Nonmandatory Appendix A, A-300).

(c) Nonferrous nuts and washers may be used with ferrous bolts and studs, provided they are suitable for the application. Consideration shall be given to the differences in thermal expansion and possible corrosion resulting from combination of dissimilar materials.

3.7.5.2 Material for Ferrous Nuts and Washers.

(a) Material for nuts and washers shall conform to SA-194, SA-563, or to the requirements for nuts in the specification for the bolting material with which they are to be used.

(b) Materials for ferrous nuts and washers shall be selected as follows:

(1) Carbon or low alloy steel nuts and carbon or low alloy steel washers of approximately the same hardness as the nuts may be used for metal temperatures not exceeding 480°C (900°F).

(2) Alloy steel nuts shall be used for metal temperatures exceeding 480°C (900°F). Washers, if used, shall be of alloy steel equivalent to the nut material.

3.7.5.3 Requirements for Ferrous Nuts.

(a) Nuts shall be semifinished, chamfered, and trimmed. Nuts shall be threaded to Class 2B or finer tolerances according to ASME B1.1.

(b) For use with flanges conforming to ASME/ANSI B16.5, nuts shall conform to at least to the dimensions given in ASME/ANSI B18.2.2 for Heavy Series Nuts.

(c) For use with connections designed in accordance with rules in 4.16, nuts may be of the American National Standard Heavy Series or they may be of other dimensions provided their strength is equal to that of the bolting, giving due consideration to the bolt hole clearance, bearing area, thread form and class of it, thread shear, and radial thrust from threads.

(d) Nuts shall engage the threads for the full depth of the nut or, in the case of cap nuts, to a depth equivalent to the depth of a standard nut.

(e) Nuts of special design may be used, provided their strength is equal to that of the bolting.

3.7.6 NONFERROUS BOLTING

3.7.6.1 Material for Nonferrous Bolting. Approved specifications for Nonferrous bolting are given in Annex 3-A, Tables 3-A.9 and 3-A.10, and 3-A.11.

3.7.6.2 Condition of Material Selected and Allowable Stress Value.

(a) When nonferrous bolts are machined from heat-treated, hot-rolled, or cold-worked material and are not subsequently hot worked or annealed, the allowable design stress values in Section II, Part D, Subpart 1, Table 3 to be used in design shall be based on the condition of material selected.

(b) When nonferrous bolts are fabricated by hot heading, the allowable design stress values for annealed materials in Section II, Part D, Subpart 1, Table 3 shall apply unless the manufacturer can furnish adequate control data to show that the tensile properties of hot-rolled or heat-treated bars or hot-finished or heat-treated forgings are being met, in which case the allowable stress values for the material in the hot finished condition may be used.

(c) When nonferrous bolts are fabricated by cold heading, the allowable design stress values for annealed materials in Section II, Part D, Subpart 1, Table 3 shall apply unless the manufacturer can furnish adequate control data to show that higher design stresses, as agreed upon may be used. In no case shall such stresses exceed the allowable stress values given in Section II, Part D, Subpart 1, Table 3 for cold-worked bar stock.

3.7.6.3 Materials for Nonferrous Nuts and Washers.

(a) Materials for ferrous nuts used with nonferrous bolting shall conform to 3.7.5.3.

(b) Nonferrous nuts and washers may be made of any suitable material listed in Tables 3-A.5, 3-A.6, and 3-A.7.

3.7.6.4 Requirements for Nonferrous Nuts. Nonferrous nuts shall meet the requirements in 3.7.5.3.

3.7.7 MATERIALS FOR FERROUS AND NONFERROUS NUTS OF SPECIAL DESIGN

Nuts of special design, such as wing nuts, may be made of any suitable wrought material permitted by this Division, and shall be either: hot or cold forged; or machined from hot-forged, hot-rolled, or cold-drawn bars.

3.8 SUPPLEMENTAL REQUIREMENTS FOR CASTINGS

3.8.1 GENERAL

3.8.1.1 Each casting shall be marked with the name, trademark, or other traceable identification of the manufacturer and the casting identification, including material designation. The casting manufacturer shall furnish certification that each casting conforms to all the applicable requirements in the casting specification and the requirements of this Division. The certification of castings shall also indicate the nature, location, and extent of any repairs.

3.8.1.2 All castings to be welded shall be of weldable grade.

3.8.2 REQUIREMENTS FOR FERROUS CASTINGS

3.8.2.1 Centrifugal Steel Castings. In addition to the minimum requirements of the material specification, all surfaces of centrifugal castings shall be machined after heat treatment to a finish not coarser than 6.35 μmm (250 μin.) arithmetic average deviation.

3.8.2.2 Nondestructive Examination of Ferrous Castings.

(a) General – Castings shall be examined by radiographic, ultrasonic, magnetic particle and liquid penetrant methods examination as provided herein and shall meet the requirements of (a) through (d), inclusive. Radiographic examination, and when required ultrasonic examination, of castings shall be made after at least one austenitizing heat treatment, except austenitic castings not requiring heat treatment may have radiographic and ultrasonic examination performed at any stage of manufacture. Magnetic particle or liquid penetrant examinations shall be made after final heat treatment and after final machining of machined areas.

(b) Radiographic Examination – All parts of ferrous castings regardless of thickness shall be fully radiographed in accordance with the procedures of Section V, Article 2. The radiographs shall be compared to the appropriate Radiographic Standard listed below, and the maximum acceptable severity levels for imperfection shall be as follows:

(1) For castings having radiographed thickness of less than 50 mm (2 in.), ASTM E446, Standard Reference Radiographs for Steel Castings up to 2 in. (50 mm) in Thickness, and with maximum severity levels as shown in Table 3.9.

(2) For castings having radiographed thickness from 50 mm to 305 mm (2 in. to 12 in.), ASTM E186, Standard Reference Radiographs for Heavy-Walled [2 to $4\frac{1}{2}$ in. (50.8 to 114 mm)] Steel Castings, or ASTM E280, Standard Reference Radiographs for Heavy-Walled [$4\frac{1}{2}$ to 12 in. (114 to 305 mm)] Steel Castings, as appropriate, and with maximum severity levels as shown in Table 3.10.

(c) Ultrasonic Examination – All parts of ferrous castings over 305 mm (12 in.) thick shall be examined by ultrasonic methods in accordance with the procedures of Section V, Article 5. Castings with imperfections shown by discontinuities whose reflections exceed the height equal to 20% of the normal back reflection, or which reduce the height of the back reflections by more than 30% during movement of the transducer 50 mm (2 in.) in any direction are unacceptable unless other methods of nondestructive testing, such as radiographic examination, demonstrate to the satisfaction of the vessel Manufacturer and the Inspector that the indications are acceptable or unless such imperfections are removed and the casting is repaired.

(d) Magnetic Particle Examination – Castings of ferromagnetic material shall be examined on all surfaces by a magnetic particle method in accordance with Part 7 of this Division. Castings with imperfections shown by Type I indications or by indications exceeding Degree I of Types II, III, IV, and V of ASTM E125, Reference Photographs for Magnetic Particle Indications on Ferrous Castings, are unacceptable unless the imperfections are removed and casting is repaired.

(e) Liquid Penetrant Examination – Castings of nonferromagnetic material shall be examined on all surfaces by a liquid penetrant method in accordance with Part 7 of this Division. Castings with cracks and linear imperfections exceeding the following limits are unacceptable:

(1) Linear indications resulting in more than six indications in any 40 mm × 150 mm ($1\frac{1}{2}$ in. × 6 in.) rectangle or 90 mm (3.5 in.) diameter circle with these taken in the most unfavorable location relative to the indications being evaluated.

(2) Linear imperfections resulting in indications more than 6 mm ($\frac{1}{4}$ in.) in length for thicknesses up to 19 mm ($\frac{3}{4}$ in.), one third of the thickness in length for thicknesses from 19 mm ($\frac{3}{4}$ in.) to 57 mm (2.25 in.), and 19 mm ($\frac{3}{4}$ in.) in length for thicknesses over 57 mm (2.25 in.). Aligned acceptable imperfections separated from one another by a distance equal to the length of the longer imperfection are acceptable.

(3) All nonlinear imperfections which are indicated to have any dimension which exceeds 2.5 mm (0.0938 in.).

3.8.2.3 Repairing of Ferrous Castings.

(a) Castings with unacceptable imperfections may be repaired. Whenever an imperfection is removed and subsequent repair by welding is not required, the affected area shall be blended into the surrounding surface so as to avoid sharp notches, crevices, or corners.

(b) Repairing of Ferrous Castings by Welding – Castings having imperfections in excess of the maximum sizes permitted in 3.8.2.2 may be repaired by welding if the imperfections are removed and providing prior approval is obtained from the vessel Manufacturer. To ensure complete removal of such imperfections prior to making repairs the base metal shall be reexamined by either magnetic particle or liquid penetrant examination, if it is ferromagnetic, or by liquid penetrant examination, if it is nonferromagnetic.

(1) Requirements for Examining Repairs in Castings – All weld repairs of depth exceeding 10 mm ($\frac{3}{8}$ in.) or 20% of the section thickness, whichever is the lesser, shall be examined by radiography and by magnetic particle examination or liquid penetrant examination, if the material is magnetic, or by liquid penetrant examination, if it is nonferromagnetic, in accordance with 3.8.2.2. Where the depth of the repairs is less than 20% of the section thickness or 25 mm (1 in.), whichever is the lesser, and where the repaired section cannot be radiographed effectively, the first layer of each 6 mm ($\frac{1}{4}$ in.)

thickness of deposited weld metal and the finished weld surface shall be examined, as indicated previously by magnetic particle or liquid penetrant examination. The finished surface examination shall be made after any heat treating operations that are applied to the casting. Weld repairs resulting from ultrasonic examination shall be examined by ultrasonic methods.

(2) Postweld Heat Treatment of Repaired Castings – When repair welding is done after heat treatment of the casting, the casting shall be postweld heat treated after repair welding of the casting.

(3) Required Welding Procedure and Welder Qualifications – All welding shall be performed with a welding procedure qualified in accordance with Section IX. The procedure qualification tests shall be performed on specimens of cast material of the same specification and subject to the same heat treatment before and after welding as will be applied to the work. All welders and operators performing this welding shall also be qualified in accordance with Section IX.

(4) Certification of Weld Repairs – The location and extent of the weld repairs together with the repair procedure and examination results shall be recorded and transmitted as part of the certification.

3.8.3 REQUIREMENTS FOR NONFERROUS CASTINGS

3.8.3.1 Examination of Nonferrous Castings. All nonferrous castings shall be examined in accordance with the following:

(a) Each casting shall be subjected to 100% visual examination and to liquid penetrant examination on all surfaces in accordance with 3.8.2.2(e). These examinations shall be performed following the final heat treatment applied to the casting.

(b) All parts of castings shall be subjected to complete radiographic examination and the radiographs shall be compared with the radiographic standards of ASTM E272, Reference Radiographs for Inspection of High Strength Copper Base and Nickel-Copper Castings. Acceptable castings shall meet Class 1 standards, if the wall thickness is less than 25 mm (1 in.) or Class 2 standards if the wall thickness is greater than or equal to 25 mm (1 in.) as defined in the Specification.

(c) All parts of castings with a thickness greater than 305 mm (12 in.) shall be ultrasonically examined in accordance with the procedures given in SE-114. Any imperfections whose reflections do not exceed a height equal to 20% of the normal back reflection or do not reduce the height of the back reflection by more than 30% during movement of the transducer 50 mm (2 in.), in any direction, shall be considered acceptable. The above limits are established for the use of transducers having approximately 645 mm^2 (1 in.^{2}) of area.

3.8.3.2 Repairing of Nonferrous Castings by Welding. Upon approval by the vessel Manufacturer, castings subject to rejection because of these examinations may be repaired in accordance with the following requirements.

(a) Castings having imperfections in excess of the maximum sizes permitted in 3.8.3.1 may be repaired by welding, if the imperfections are removed and provided prior approval is obtained from the vessel Manufacturer. To assure complete removal of such imperfections, prior to making repairs, the base metal shall be reexamined by liquid penetrant examination.

(b) All weld repairs of depth exceeding 10 mm ($^3/_8$ in.), or 20% of the section thickness, whichever is the lesser, shall be examined by radiography and by liquid penetrant examination in accordance with 3.8.3.1. Where the depth of repairs is less than 20% of the section thickness or 25 mm (1 in.), whichever are the lesser, and where the repaired section cannot be radiographed effectively, the first layer of each 6 mm ($^3/_4$ in.) thickness of deposited weld metal and the finished weld surface shall be examined, as indicated previously, by liquid penetrant examination. The finished surface examination shall be made after any heat treating operation that is applied to the casting. Weld repairs resulting from ultrasonic examination shall be examined by ultrasonic methods.

(c) When repair welding is done after heat treatment of the casting, the casting shall be postweld heat treated after repair welding.

(d) All welding shall be performed using welding procedures qualified in accordance with Section IX. The procedure qualifications shall be performed on test specimens of cast material of the same specification and subject to the same heat treatments before and after welding as will be applied to the work. All welders and welding operators performing this welding shall be qualified in accordance with Section IX.

(e) The location and extent of the weld repairs together with the repair procedure and examination results shall be recorded and transmitted as part of the certification.

3.9 SUPPLEMENTAL REQUIREMENTS FOR HUBS MACHINED FROM PLATE

3.9.1 GENERAL

The supplemental requirements of 3.9 are required for plate materials that are used in the fabrication of hubs for tubesheets, lap joint stub ends, and flat heads machined from plate when the hub length is in the through thickness direction of the plate.

3.9.2 MATERIAL REQUIREMENTS

3.9.2.1 Plate shall be manufactured by a process that produces material having through thickness properties which are at least equal to those specified in the material specification. Such plate can be but is not limited to that produced by methods such as electroslag (ESR) and vacuum arc remelt (VAR). The plate must be tested and examined in accordance with the requirements of the material specification and the additional requirements specified in the following paragraphs.

3.9.2.2 Test specimens, in addition to those required by the material specifications, shall be taken in a direction parallel to the axis of the hub and as close to the hub as practical, as shown in Figure 3.2. At least two tensile test specimens shall be taken from the plate in the proximity of the hub, with one specimen taken from the center third of the plate width as rolled, and the second specimen taken at 90 deg around the circumference from the other specimen. Both specimens shall meet the mechanical property requirements of the material specification. For carbon and low alloy steels, the reduction of area shall not be less than 30%; for those materials for which the material specification requires a reduction of area value greater than 30%, the higher value shall be met.

3.9.2.3 Subsize test specimens conforming to the requirements of SA-370, Figure 5 may be used if necessary, in which case the value for percent elongation in 50 mm (2 in.), required by the material specification, shall apply to the gage length specified in SA-370, Figure 5.

3.9.2.4 Tension test specimen locations are shown in Figure 3.2.

3.9.3 EXAMINATION REQUIREMENTS

3.9.3.1 After machining the part, regardless of thickness, shall be ultrasonically examined by the straight beam technique in accordance with SA-388. The examination shall be in two directions approximately at right angles, i.e., from the cylindrical or flat rectangular surfaces of the hub and in the axial direction of the hub. The part shall be unacceptable if

(a) the examination results show one or more indications accompanied by loss of back reflection larger than 60% of the reference back reflection, and

(b) the examination results show indications larger than 40% of the reference back reflection when accompanied by a 40% loss of back reflection.

3.9.3.2 Before welding the hub of the tubesheet flange or flat head to the adjacent shell, the hub shall be examined by magnetic particle or liquid penetrant methods in accordance with Part 7.

3.9.3.3 After welding, the weld and the area of the hub for at least 13 mm ($^1/_2$ in.) from the edge of the weld shall be 100% radiographed in accordance with Part 7. As an alternate, the weld and hub area adjacent to the weld may be ultrasonically examined in accordance with Part 7.

3.9.4 DATA REPORTS AND MARKING

Whenever the provisions of this supplemental requirement are used, they shall be indicated on the Data Report. Special markings are not required.

3.10 MATERIAL TEST REQUIREMENTS

3.10.1 GENERAL

Material tests required by this Division shall be performed in accordance with 3.10.

3.10.2 REQUIREMENTS FOR SAMPLE TEST COUPONS

3.10.2.1 Heat Treatment. Heat treatment as used in this Division shall include all thermal treatments during fabrication at 480°C (900°F) and above.

3.10.2.2 Provisions of Sample Test Coupons. When material is subjected to heat treatment during fabrication, the test specimens required by this Division shall be obtained from sample coupons which have been heat treated in the same manner as the material, including such heat treatments as were applied by the material producer before shipment. The required tests may be performed by the material producer or the fabricator.

3.10.2.3 Heat Treating of Sample Test Coupons.

(a) The material used in the vessel shall be represented by test specimens that have been subjected to the same manner of heat treatment, including postweld heat treatment. The kind and number of tests and test results shall be as required by the material specification. The vessel Manufacturer shall specify the temperature, time, and cooling rates to which the material will be subject during fabrication. Material from which the specimens are prepared shall be heated at the specified temperature within the tolerance established by the manufacturer for use in actual fabrication. The total time at temperature shall be within at least 80% of the total time at temperature during actual heat treatment of the product and may be performed in a single cycle. Simulation of postweld heat treatment may be applied to the test specimen blanks.

(b) Heat treatment of material is not intended to include such local heating as flame or arc cutting, preheating, welding, or heating below the critical range of tubing or pipe for bending or sizing.

3.10.3 EXEMPTIONS FROM REQUIREMENT OF SAMPLE TEST COUPONS

3.10.3.1 Standard Pressure Parts. An exception to the requirements of 3.10.2.2 and 3.10.2.3 shall apply to standard nonwelded items such as described in 3.2.8.3 and 3.2.8.4. These may be subjected to postweld heat treatment with the vessel or vessel part without the same treatment being required of the test specimens. This exception shall not apply to castings that are specially designed or to cast wrought fittings.

3.10.3.2 For Materials When PWHT to Table 6.16. Materials listed in Section IX, Table QW/QB-422 as P-No. 1 Group 3 and P-No. 3, Groups 1 and 2 that are certified in accordance with 3.10.2.2 and 3.10.2.3 from test specimens subjected to the PWHT requirements of Table 6.8 or Table 6.9 need not be recertified if subjected to the alternative PWHT conditions permitted in Table 6.16.

3.10.3.3 Re-Austenitized Materials. All thermal treatments which precede a thermal treatment that fully austenitizes the material need not be accounted for by the specimen heat treatments, provided the austenitizing temperature is at least as high as any of the preceding thermal treatments.

3.10.4 PROCEDURE FOR OBTAINING TEST SPECIMENS AND COUPONS

3.10.4.1 Plates.

(a) Unless otherwise specified, test specimens shall be taken in accordance with the requirements of the applicable material specification, except for the provisions in (b), (c), and (d) below. Tension test specimens and Charpy V-notch specimens shall be orientated in the direction perpendicular to the final direction of the plate rolling.

(b) When the plate is heat treated with a cooling rate faster than still-air cooling from the austenitizing temperature, the specimens shall be taken in accordance with requirements of applicable material specifications and $1t$ from any heat-treated edge, where t is the nominal thickness of the material.

(c) Where a separate test coupon is used to represent the vessel material, it shall be of sufficient size to ensure that the cooling rate of the region from which the test specimens are removed represents the cooling rate of the material at least $^1/_4t$ deep and $1t$ from any edge of the product. Unless cooling rates applicable to the bulk pieces or product are simulated in accordance with 3.10.5, the dimensions of the coupon shall be not less than $3t \times 3t \times 1t$, where t is the nominal thickness of the material.

(d) When flat heads, tubesheets, and flanges with integral hubs for butt welding are to be machined from plate, additional specimens shall be taken in the locations as shown in Figure 3.2.

3.10.4.2 Forgings.

(a) Test specimens shall be taken in accordance with the applicable material specification, except for the provisions in (b), (c), and (d) below.

(b) When the forging is heat treated with a cooling rate faster than still-air cooling from the austenitizing temperature the specimens shall be taken at least $^1/_4t$ of the maximum heat-treated thickness from one surface and $1t$ from a second surface. This is normally referred to as $^1/_4t \times 1t$, where t is the maximum heat-treated thickness. A thermal buffer may be used to achieve these conditions unless cooling rates applicable to the bulk forgings are simulated in accordance with 3.10.5.

(c) For thick and complex forgings, such as contour nozzles, thick tubesheets, flanges, and other complex forgings that are contour shaped or machined to essentially the finished product configuration prior to heat treatment, the registered engineer who prepares the Design Report shall designate the surfaces of the finished product subject to high tensile stresses in service. Test specimens for these products shall be removed from prolongations or other stock provided on the product. The specimens shall be removed as follows:

(1) The distance from the longitudinal axis of the specimen to the nearest heat-treated surface shall be no less than the distance from the location where the maximum tensile stress is expected to the nearest heat-treated surface. This distance shall be at least 19 mm ($^{3}/_{4}$ in.).

(2) The distance from the mid-length of the specimen to a second heat-treated surface shall be at least twice the distance in (1). This distance shall be at least 38 mm (1.5 in.).

(d) With prior approval of the vessel Manufacturer, test specimens for flat ring and simple ring forgings may be taken from a separately forged piece under the following conditions.

(1) The separate test forging shall be of the same heat of material and shall be subjected to substantially the same reduction and working as the production forgings it represents.

(2) The separate test forging shall be heat treated in the same furnace charge and under the same conditions as the production forgings.

(3) The separate test forging shall be of the same nominal thickness as the production forgings. Test specimen material shall be removed as required in (a) and (b).

3.10.4.3 Tubular Products. Specimens shall be taken in accordance with the requirements of the applicable material specification.

3.10.4.4 Bars and Bolting Materials.

(a) Test specimens shall be taken in accordance with the requirements of the applicable material specification, except for the provisions of (b) below.

(b) Test specimens shall be at least $^{1}/_{4}t$ from the outside or rolled surface and with the end of the specimen no closer than one diameter or thickness from the heat-treated end.

(c) For bolting, the specimens shall be taken in conformance with the applicable material specification and with the end of the specimen no closer than one diameter or thickness from a heat-treated end.

3.10.4.5 Castings.

(a) The conventional separately cast test coupon meets the intent of 3.10.5 where normalizing or accelerated cooling heat treatments are employed on castings having a maximum thickness of less than 50 mm (2 in.).

(b) For castings having a thickness of 50 mm (2 in.) and over, the specimens shall be taken from the casting (or the extension of it) at least $^{1}/_{4}t$ of the maximum heat-treated thickness from one surface and $1t$ from a second surface. A thermal buffer may be used.

(c) For massive castings that are cast or machined to essentially the finished product configuration prior to heat treatment, the registered engineer who prepares the Design Report shall designate the surfaces of the finished product subject to high tensile stresses in service. Test specimens for these products shall be removed from prolongations or other stock provided on the product. The specimens shall be removed as follows:

(1) The distance from the longitudinal axis of the specimen to the nearest heat-treated surface shall be no less than the distance from the location where the maximum tensile stress is expected to the nearest heat-treated surface. This distance shall be at least 19 mm ($^{3}/_{4}$ in.).

(2) The distance from the mid-length of the specimen to a second heat-treated surface shall be at least twice the distance in (1). This distance shall be at least 38 mm (1.5 in.).

(d) With prior approval of the vessel Manufacturer, test specimens may be taken from a separately cast test coupon under the following conditions:

(1) The separate test coupon shall be of the same heat of material and shall be subjected to substantially the same casting practices as the production casting it represents.

(2) The separate test coupon shall be heat treated in the same furnace charge and under the same conditions as the production casting, unless cooling rates applicable to bulk castings are simulated in accordance with 3.10.5.

(3) The separate test coupon shall be of the same nominal thickness as the production casting. Test specimen material shall be removed from the region midway between mid-thickness and the surface and shall not be nearer than on thickness to a second surface.

3.10.5 PROCEDURE FOR HEAT TREATING TEST SPECIMENS FROM FERROUS MATERIALS

3.10.5.1 General requirements for heat treating of sample test coupons are covered in 3.10.2.3.

3.10.5.2 When ferritic steel products are subjected to normalizing or accelerated cooling from the austenitizing temperature, the test specimens representing those products shall be cooled at a rate similar to and no faster than the main body of the product except in the case of certain forgings and castings [see 3.10.4.2(c) and 3.10.4.5(c)]. This rule shall apply for specimens taken directly from the product as well as those taken from separate test coupons representing the product. The following general techniques may be applied to all product forms or test coupons representing the product.

(a) Any procedure may be applied which can be demonstrated to produce a cooling rate in the test specimen that matches the cooling rate of the main body of the product at the region midway between mid-thickness and surface ($^1/_4t$) and no nearer any heat-treated edge than a distance equal to the nominal thickness, t, being cooled within 14°C (25°F) and 20 sec at all temperatures after cooling begins from the austenitizing temperature.

(b) Faster cooling rates at product edges may be compensated for by:

(1) Taking the test specimens at least $1t$ from a quenched edge where t equals the product thickness.

(2) Attaching a steel pad at least $1t$ wide by a partial penetration weld to the product edge where specimen are to be removed.

(3) Using thermal buffers or insulation at the product edge where specimens are to be removed.

(c) If cooling rate data for the product and cooling rate device control devices for the test specimens are available, the test specimens may be heat treated in the device to represent the product, provided that the provisions of (a) are met.

(d) When the material is clad or weld deposit overlayed by the product prior to normalizing or accelerated cooling from the austenitizing temperature, the full thickness samples shall be clad or the weld deposit overlayed before such heat treatments.

3.10.6 TEST COUPON HEAT TREATMENT FOR NONFERROUS MATERIALS

3.10.6.1 Fabrication heat treatments of nonferrous material are normally not necessary. If heat treatment is performed, it shall be by agreement between the user and the vessel Manufacturer.

3.10.6.2 Materials where the mechanical properties are affected by fabrication heat treatments shall be represented by test specimens that have been subjected to the simulated fabrication heat treatments. The vessel Manufacturer shall specify the pertinent fabrication heat treatment parameters to the material manufacturer.

3.10.6.3 The requirements of 3.10.6.2 above exclude annealing and stress relieving.

3.11 MATERIAL TOUGHNESS REQUIREMENTS

3.11.1 GENERAL

3.11.1.1 Charpy V-notch impact tests shall be made for materials used for shells, heads, nozzles, and other pressure-containing parts, as well as for the structural members essential to structural integrity of the vessel, unless exempted by the rules of 3.11.

(a) Toughness requirements for materials listed in Table 3-A.1 (carbon and low alloy steel materials except bolting materials) are given in 3.11.2.

(b) Toughness requirements for materials listed in Table 3-A.2 (quenched and tempered steels with enhanced tensile properties) are given in 3.11.3.

(c) Toughness requirements for materials listed in Table 3-A.3 (high alloy steels except bolting materials) are given in 3.11.4.

(d) Toughness requirements for materials listed in Table 3-A.4 through 3-A.7 (nonferrous alloys) are given in 3.11.5.

(e) Toughness requirements for all bolting materials are given in 3.11.6.

3.11.1.2 Toughness testing procedures and requirements for impact testing of welds and vessel test plates of ferrous materials are given in 3.11.7 and 3.11.8, respectively.

3.11.1.3 Throughout 3.11, reference is made to the Minimum Design Metal Temperature (MDMT). The MDMT is part of the design basis of the vessel and is defined in 4.1.5.2(e). The rules in 3.11 are used to establish an acceptable MDMT for the material based on the materials of construction, product form, wall thickness, stress state, and heat treatment.

3.11.2 CARBON AND LOW ALLOY STEELS EXCEPT BOLTING

(21) **3.11.2.1 Toughness Requirements for Carbon and Low Alloy Steels.**

(a) Impact tests shall be performed on carbon and low alloy materials listed in Table 3-A.1 for all combinations of materials and MDMTs except as exempted by 3.11.2.3, 3.11.2.4, 3.11.2.5, or 3.11.2.8.

(b) When impact testing is necessary, the following toughness values are required.

(1) If the specified minimum tensile strength is less than 655 MPa (95 ksi), then the required minimum energy for all specimen sizes shall be that shown in Figure 3.3 and Figure 3.4 for vessel parts not subject to postweld heat treatment (PWHT) and vessel parts subject to PWHT or nonwelded parts, respectively, multiplied by the ratio of the actual specimen width along the notch to the width of a full-size specimen, except as otherwise provided in 3.11.7.2(b).

(2) If the specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the minimum lateral expansion (see Figure 3.5) opposite the notch for all specimen sizes shall not be less than the values shown in Figure 3.6.

3.11.2.2 Required Impact Testing Based on the MDMT, Thickness, and Yield Strength.

(a) If the governing thickness (see 3.11.2.3(b) at any welded joint or of any nonwelded part exceeds 100 mm (4 in.) and the MDMT is colder than 43°C (110°F), then impact testing is required.

(b) Materials having a specified minimum yield strength greater than 450 MPa (65 ksi) shall be impact tested.

3.11.2.3 Exemption From Impact Testing Based on the MDMT, Thickness, and Material Specification.

(a) Figure 3.7 for parts not subject to PWHT or Figure 3.8 for parts subject to PWHT shall be used to establish impact testing exemptions based on the impact test exemption curve for the subject material specification and grade or class of the steel, MDMT, and governing thickness of a welded part. If an MDMT and thickness combination for the subject material is on or above the applicable impact test exemption curve in Figure 3.7 or Figure 3.8, then impact testing is not required except as required by 3.11.8 for weld metal and heat-affected zones.

(b) The governing thickness, t_g, of a welded part is determined using the following criteria. Examples of the governing thickness for some typical vessel details are shown in Figures 3.9, 3.10, and 3.11.

(1) For all product forms except castings:

(-a) For butt joints except those in flat heads and tubesheets, the nominal thickness of the thickest welded joint [see Figure 3.9, sketch (a)],

(-b) For corner, fillet, or lap-welded joints, including attachments as defined in 3.11.1.1, the thinner of the two parts joined,

(-c) For flat heads or tubesheets, the larger of (-b) above or the flat component thickness divided by 4.

(2) The governing thickness of a casting shall be its largest nominal thickness.

(3) The governing thickness of flat nonwelded parts, such as bolted flanges, tubesheets, and flat heads, is the flat component thickness divided by 4.

(4) The governing thickness of a nonwelded dished head is the greater of the flat flange thickness divided by 4 or the minimum thickness of the dished portion.

(c) Components such as shells, heads, nozzles, manways, reinforcing pads, stiffening rings, flanges, tubesheets, flat cover plates, backing strips, and attachments that are essential to the structural integrity of the vessel when welded to pressure-retaining components shall be treated as separate components. Each component shall be evaluated for impact test requirements based on its individual material classification, governing thickness [see (b)], and the MDMT. For welded assemblies comprised of more than two components (e.g., nozzle-to-shell joint with reinforcing pad), the governing thickness and permissible MDMT of each of the individual welded joints of the assembly shall be determined, and the warmest MDMT shall be used as the permissible MDMT of the welded assembly.

(d) Figure 3.7 limits the maximum nominal governing thickness for welded parts not subject to postweld heat treatment to 38 mm ($1^1/_2$ in.). Some vessels may have welded non-postweld-heat-treated pressure parts whose thickness exceeds the nominal governing thickness of 38 mm ($1^1/_2$ in.). Examples of such welded and non-postweld-heat-treated pressure parts are thick tubesheets, flat heads, and thick insert plates (with beveled edges) with nozzles or load-carrying structural attachments. Such welded non-postweld-heat-treated pressure parts shall be impact tested and shall meet the impact test requirements of this Division.

(e) Impact testing is not required for materials with a thickness of 2.5 mm (0.099 in.) and thinner, but such exempted materials shall not be used at design metal temperatures colder than −48°C (−55°F). For components made from DN 100 (NPS 4) pipe or smaller and for equivalent size of tubes of P-No. 1 materials, the following exemptions from impact testing are also permitted as a function of the specified minimum yield strength (SMYS) of the material for metal temperatures of −104°C (−155°F) and warmer:

(1) For SMYS between 140 MPa and 240 MPa (20 ksi and 35 ksi), inclusive, the thickness exemption for impact testing is 6 mm ($^1/_4$ in.).

(2) For SMYS between 250 MPa and 310 MPa (36 ksi and 45 ksi), inclusive, the thickness exemption for impact testing is 3.2 mm ($^1/_8$ in.).

(3) For SMYS higher than 315 MPa (46 ksi), inclusive, the thickness exemption for impact testing is 2.5 mm (0.099 in.).

(f) Note that the rules in this paragraph for the exemption of impact testing do not provide assurance that all test results for these materials will satisfy the impact test acceptance criteria of 3.11.2.1(b).

3.11.2.4 Exemption From Impact Testing Based on Material Specification and Product Form.

(a) Impact testing is not required for the ferritic steel flanges shown below when produced to fine grain practice and supplied in heat-treated condition (normalized, normalized and tempered, or quenched and tempered after forging) when used at design temperatures no colder than −29°C (−20°F) and no colder than −18°C (0°F) when supplied in the as-forged condition:

(1) ASME B16.5 flanges.

(2) ASME B16.47 flanges.

(3) Long weld neck flanges, defined as forged nozzles that meet the dimensional requirements of a flanged fitting given in ASME B16.5 but have a straight hub/neck. The neck inside diameter shall not be less than the nominal size of the flange, and the outside diameter of the neck and any nozzle reinforcement shall not exceed the diameter of the hub as specified in ASME B16.5.

(b) Materials produced and impact tested in accordance with the requirements of the specifications shown below are exempt from impact testing by the rules of this Division at MDMTs not more than 3°C (5°F) colder than the test temperature required by the specification.

(1) SA-320
(2) SA-333
(3) SA-334
(4) SA-350
(5) SA-352
(6) SA-420
(7) SA-437
(8) SA-508 Grade 5 Class 2
(9) SA-540 except for materials produced under Table 2, note 4 in this specification
(10) SA-765

(21) 3.11.2.5 Exemption From Impact Testing Based on Design Stress Values.

(a) A colder MDMT for a component than that derived from 3.11.2.2 or 3.11.2.3 may be determined in accordance with the procedure outlined below.

Step 1. For the welded part under consideration, determine the nominal thickness of the part, t_n, and the required governing thickness of the part, t_g, using 3.11.2.3(b).

Step 2. Determine the applicable material toughness curve to be used in Figure 3.7 for parts not subject to PWHT or Figure 3.8 for parts subject to PWHT. See 3.11.2.2(b) for materials having a specified minimum yield strength greater than 450 MPa (65 ksi).

Step 3. Determine the MDMT from Figure 3.7 for parts not subject to PWHT or Figure 3.8 for parts subject to PWHT based on the applicable toughness curve and the governing thickness, t_g. For materials having a specified minimum yield strength greater than 450 MPa (65 ksi), the MDMT shall be determined by impact testing per 3.11.2.2(b).

Step 4. Based on the design loading conditions at the MDMT, determine the stress ratio, R_{ts}, using one of the equations below. For pressure vessel attachments that are exposed to tensile stresses from internal pressure (e.g., nozzle reinforcement pads, horizontal vessel saddle attachments, and stiffening rings), the coincident ratio shall be that of the shell or head to which each component is attached. Note that this ratio can be computed in terms of required design thickness and nominal thickness, applied stress and allowable design stress, or applied pressure and maximum allowable working pressure based on the design rules in this Division or ASME/ANSI pressure–temperature ratings.

$$R_{ts} = \frac{t_r E^*}{t_n - CA} \qquad \text{(Thickness Basis)} \tag{3.1}$$

$$R_{ts} = \frac{S^* E^*}{SE} \qquad \text{(Stress Basis)} \tag{3.2}$$

$$R_{ts} = \frac{P_a}{P_{\text{rating}}} \qquad \text{(Pressure–Temperature Rating Basis)} \tag{3.3}$$

Step 5. Determine the final value of the MDMT and evaluate results.

(a) If the computed value of the R_{ts} ratio from Step 4 is less than or equal to 0.3 for Class 1, or 0.24 for Class 2, then set the MDMT to −104°C (−155°F).

(b) If the computed value of the R_{ts} ratio from Step 4 is greater than 0.3 for Class 1, or 0.24 for Class 2, then determine the temperature reduction, T_R. If the specified minimum yield strength is less than or equal to 450 MPa (65 ksi), then determine T_R from Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT based on the R_{ts}

ratio from Step 4. If the specified minimum yield strength is greater than 450 MPa (65 ksi) for parts subject to PWHT, then determine the temperature reduction, T_R from eq. (3.4). The final computed value of the MDMT is determined using eq. (3.5). The reduction in the MDMT given by eq. (3.5) shall not exceed 55°C (100°F). Impact testing is not required if the specified MDMT is warmer than the computed MDMT. However, if the specified MDMT is colder than −48°C (−55°F), impact testing is required.

$$T_R = \frac{\left(\begin{array}{c} -27.20656 - 76.98828 R_{ts} + \\ 103.0922 R_{ts}^2 + 7.433649(10)^{-3} S_y \end{array}\right)}{\left(\begin{array}{c} 1 - 1.986738 R_{ts} - 1.758474(10)^{-2} S_y + \\ 6.479033(10)^{-5} S_y^2 \end{array}\right)} \qquad (°\text{F, ksi}) \tag{3.4}$$

$$\text{MDMT} = \text{MDMT}_{\text{STEP3}} - T_R \tag{3.5}$$

(b) The procedure in 3.11.2.5(a) above is repeated for each welded part, and the warmest MDMT of all welded parts is the MDMT for the vessel.

(c) For a flange attached by welding, the procedure in 3.11.2.5(a) above can be used by determining the temperature reduction as determined for the neck or shell to which the flange is attached. The bolt-up condition need not be considered when determining the temperature reduction for flanges.

(d) For components not stressed in primary membrane tensile stress such as flat heads, covers, tubesheets, and flanges, the MDMT shall not be colder than the MDMT derived from 3.11.2.3 or the impact test temperature less the allowable temperature reduction as determined in 3.11.2.5(a). The ratio used in 3.11.2.5(a) shall be the ratio of the maximum design pressure at the MDMT to the maximum allowable pressure (MAP) of the component at the MDMT.

(e) Longitudinal tensile stress in the vessel due to net-section bending that results in general primary membrane tensile stress (e.g., due to wind or earthquake in a vertical vessel, at mid-span and in the plane of the saddles of a saddle-supported horizontal vessel) shall be considered when calculating the R_{ts} ratio in Step 4.

3.11.2.6 Adjusting the MDMT for Impact Tested Materials. (21)

(a) For components that are impact tested, the components may be used at a MDMT colder than the impact test temperature, provided the stress ratio defined in 3.11.2.5(a), Step 4 is less than one and the MDMT is not colder than −104°C (−155°F). For such components, the MDMT shall not be colder than the impact test temperature less the allowable temperature reduction as determined from 3.11.2.5 (i.e., the starting point for the MDMT calculation in 3.11.2.5(a), Step 3, is the impact test temperature). For pressure vessel attachments that are exposed to tensile stresses from internal pressure (e.g., nozzle reinforcement pads, horizontal vessel saddle attachments, and stiffening rings), the coincident ratio shall be that of the shell or head to which each component is attached. [See 3.11.2.4(b)].

(b) One common usage of the exemptions in 3.11.2.5 and 3.11.2.6 will be for vessels in which the pressure is dependent on the vapor pressure of the contents (e.g., vessels in refrigeration, or hydrocarbon processing plants with operating systems that do not permit immediate repressurization). For such services, the primary thickness calculations (shell and head) normally will be made for the maximum design pressure coincident with the design (MDMT) temperature expected. The ratio of required thickness/nominal thickness as defined in 3.11.2.5(a), Step 4, for the design condition is then calculated. Thickness calculations are also made for other expected pressures at coincident temperature, along with the ΔT difference from the MDMT [see 3.11.2.5(a), Step 3], and the thickness ratio defined in 3.11.2.5(a), Step 4. Ratio/ΔT points that are on or below the line in Figure 3.12 (for as-welded parts) or Figure 3.13 (for PWHT or nonwelded parts), as applicable, are acceptable, but in no case may the operating temperature be colder than −104°C (−155°F). Comparison of pressure–temperature coincident ratios or stress coincident ratios may also be used as illustrated in 3.11.2.5(a), Step 4.

3.11.2.7 Vessel or Components Operating Below the MDMT. Vessels or components may be operated at temperatures colder than the MDMT stamped on the nameplate if:

(a) The provisions of 3.11.2 are met when using the reduced (colder) operating temperature as the MDMT, but in no case shall the operating temperature be colder than −104°C (−155°F); or

(b) For vessels or components whose thicknesses are based on pressure loading only, the coincident operating temperature may be as cold as the MDMT stamped on the nameplate less the allowable temperature reduction as determined from 3.11.2.5. The ratio used in 3.11.2.5(a), Step 4, of the procedure in 3.11.2.5 shall be the ratio of maximum pressure at the coincident operating temperature to the design pressure of the vessel at the stamped MDMT, but in no case shall the operating temperature be colder than −104°C (−155°F).

3.11.2.8 Establishment of the MDMT Using a Fracture Mechanics Methodology.

(a) In lieu of the procedures in 3.11.2.1 through 3.11.2.7, the MDMT may be established using a fracture mechanics approach. The fracture mechanics procedures shall be in accordance with API 579-1/ASME FFS, Part 9, Level 2 or Level 3.

(b) The assessment used to determine the MDMT shall include a systematic evaluation of all factors that control the susceptibility to brittle fracture, e.g., stresses from the applied loadings including thermal stresses, flaw size, fracture toughness of the base metal and welded joints, heat treatment, and the loading rate.

(c) The reference flaw size used in the fracture mechanics evaluation shall be a surface flaw with a depth of $a = \min[t/4, 25\text{ mm (1 in.)}]$ and a length of $2c = 6a$ where t is the thickness of the plate containing the reference flaw. If approved by the user, an alternative reference flaw size may be used based on the weld joint geometry and the NDE that will be used and demonstrated for qualification of the vessel (see Part 7).

(d) The material fracture toughness shall be established using the exemption curve for the material (see Notes to Figures 3.7 and 3.8) and MPC Charpy impact energy correlation described in API 579-1/ASME FFS-1, Appendix F, F.4. If approved by the user, an alternative material fracture toughness may be used based on fracture toughness test results.

(e) The MDMT established using a fracture mechanics approach shall not be colder than that given in 3.11.2.3(e).

3.11.2.9 Postweld Heat Treatment Requirements for Materials in Low Temperature Service.

(a) If the MDMT is colder than −48°C (−55°F) and the stress ratio defined in 3.11.2.5(a), Step 4 is greater than or equal to 0.3 for Class 1, or 0.24 for Class 2, then welded joints shall be subject to PWHT in accordance with the requirements of 6.4.2.

(b) The requirement in (a) above does not apply to the welded joints listed in (1) and (2) below in vessel or vessel parts fabricated of P-No. 1 materials that are impact tested at the MDMT or colder in accordance with 3.11.2.1. The minimum average energy requirement for base metal, weld metal, and heat-affected zones shall be 41 J (30 ft-lb) instead of the values shown in Figure 3.3 for parts not subject to PWHT or Figure 3.4 for parts subject to PWHT or for nonwelded parts.

(1) Type 1 Category A and B joints, not including cone-to-cylinder junctions, that have been 100% radiographed. Category A and B joints attaching sections of unequal thickness shall have a transition with a slope not exceeding 3:1.

(2) Fillet welds having leg dimensions not exceeding 10 mm ($^3/_8$ in.) attaching lightly loaded attachments, provided the attachment material and the attachment weld meet the requirements of 3.11.2 and 3.11.8. Lightly loaded attachments, for this application, are defined as attachments in which the stress in the attachment weld does not exceed 25% of the allowable stress. All such welds shall be examined by liquid penetrant or magnetic particle examination in accordance with Part 7 of this Division.

3.11.2.10 Impact Tests of Welding Procedures.

(a) For welded construction, the welding procedure qualification shall include impact testing of weld metals in accordance with 3.11.2.1 when required by (b) or (c).

(b) Welds made with filler metal shall be deposited using welding procedures qualified with impact testing when

(1) either base metal is required to be impact tested by the rules of this Division; or

(2) any individual weld pass exceeds 13 mm ($^1/_2$ in.) in thickness and the MDMT is colder than 21°C (70°F); or

(3) joining base metals exempt from impact testing by 3.11.2.3, 3.11.2.4, and 3.11.2.5 when the MDMT is colder than −48°C (−55°F); or

(4) joining base metals from Figure 3.7 or Figure 3.8, Curves C or D, or metals exempted from impact testing by 3.11.2.4(b), and the MDMT is colder than −29°C (−20°F) but not colder than −48°C (−55°F). Qualification of the welding procedure with impact testing is not required when no individual weld pass in the fabrication weld exceeds 6 mm ($^1/_4$ in.) in thickness, and each heat and/or lot of filler metal or combination of heat and/or lot of filler metal and batch of flux has been classified by their manufacturer through impact testing per the applicable SFA specification at a temperature not warmer than the MDMT. Additional testing beyond the scope of the SFA specification may be performed by the filler metal and/or flux manufacturer to expand their classification for a broader range of temperatures.

(c) Except for welds made as part of the material specification, welds made without the use of filler metal shall be completed using welding procedures qualified with impact testing when

(1) either base metal is required to be impact tested by the rules of this Division; or

(2) the thickness at the weld exceeds 13 mm ($^1/_2$ in.) for all MDMTs, or 8 mm ($^5/_{16}$ in.) when the MDMT is colder than 10°C (50°F); or

(3) joining base metals exempt from testing by 3.11.2.4(b) when the MDMT is colder than −48°C (−55°F).

3.11.3 QUENCHED AND TEMPERED STEELS

3.11.3.1 Toughness Requirements for Quenched and Tempered Ferritic Steels.

(a) All quenched and tempered steels listed in Table 3-A.2 shall be subject to Charpy V-notch testing.

(b) Impact tests shall be conducted at a temperature not warmer than the MDMT determined in 4.1.5.2(d). However, in no case shall the impact test temperature be warmer than 0°C (32°F).

(c) Materials may be used at temperatures colder than the MDMT as permitted below.

(1) When the stress ratio defined in 3.11.2.5(a), Step 4 is 0.3 or less for Class 1, or 0.24 or less for Class 2, the corresponding MDMT shall not be colder than −104°C (−155°F).

(2) When the stress ratio defined in 3.11.2.5(a), Step 4 is greater than 0.3 for Class 1, or 0.24 for Class 2, the corresponding MDMT shall not be colder than the impact test temperature less the allowable temperature reduction as determined in 3.11.2.5(a) and shall in no case be colder than −104°C (−155°F).

3.11.3.2 Impact Testing.

(a) Preparation of Test Specimens – All test specimens shall be prepared from the material in its final heat-treated condition according to the requirements of 3.11.7.2.

(b) Number of Impact Tests and Test Specimens – One Charpy V-notch impact test shall consist of three test specimens. For as-rolled plates, one Charpy V-notch test shall be made from each as-rolled plate. For heat-treated plates (normalized, normalized and tempered, or quenched and tempered), one Charpy V-notch test shall be made from each plate-as-heat-treated. One Charpy V-notch test shall be made from each heat of bars, pipe, tubing, rolled sections, forged parts or castings included in any one heat treatment lot. The number of impact tests shall not be less than required by the material specification.

(c) Locations and Orientation of Test Specimens – The location and orientation of the specimens shall be the same as required for Charpy type impact tests by 3.11.7.2 and 3.11.7.3 except that specimens from plates shall be transverse to the final direction of rolling and for forgings and pipe, transverse to the direction of major work (see Figure 3.14).

(d) The minimum lateral expansion shall be in accordance with 3.11.2.1(b)(2).

(e) Retesting shall be in accordance with 3.11.7.6.

3.11.3.3 Drop-Weight Tests.

(a) When the MDMT is colder than −29°C (−20°F), drop-weight tests as defined by ASTM E208, Conducting Drop-Weight Test to Determine Nil-Ductility Transition Temperature of Ferritic Steels, shall be made on all materials listed in Table 3-A.2, with the following exceptions:

(1) SA-522 for any MDMT;

(2) SA-353 and SA-553 when the temperature is not colder than −196°C (−320°F);

(3) SA-645 Grade A when the temperature is not colder than −170°C (−275°F).

(b) Number of Tests for Plates – For plates 16 mm ($^5/_8$ in.) thick and greater, one drop-weight test (two specimens) shall be made for each plate in the as-heat-treated condition (see 3.11.3.2).

(c) Number of Tests for Forgings and Castings – For forgings and castings of all thicknesses, one drop-weight test (two specimens) shall be made for each heat in any one heat treatment lot. The sampling procedure shall comply with the requirements of ASTM E208. Specimen locations for forgings shall be the same as specified in SA-350 for location of impact test specimens (SA-350, paragraph 7.2.3).

(d) Required Test Results – Each of the two test specimens shall meet the “no-break” criterion, as defined by ASTM E208, at the test temperature.

3.11.4 HIGH ALLOY STEELS EXCEPT BOLTING

3.11.4.1 Toughness Requirements for High Alloy Steels. (21)

(a) Impact tests shall be performed on high alloy materials listed in Table 3-A.3 for all combinations of materials and MDMTs except as exempted by 3.11.4.3 or 3.11.4.5. Impact testing is required for UNS S17400 materials. Impact tests shall be made from sets of three specimens: one set from the base metal, one set from the weld metal, one set from the heat-affected zone (HAZ). Specimens shall be subjected to the same thermal treatments as the part or vessel that the specimens represent.

(b) When the MDMT is −196°C (−320°F) and warmer, impact tests shall be conducted at the MDMT or colder, and the minimum lateral expansion opposite the notch shall be no less than 0.38 mm (0.015 in.) for MDMTs of −196°C (−320°F) and warmer.

(c) When the MDMT is colder than −196°C (−320°F), production welding processes shall be limited to shielded metal arc welding (SMAW), flux-cored arc welding (FCAW), gas metal arc welding (GMAW), submerged arc welding (SAW), plasma arc welding (PAW), and gas tungsten arc welding (GTAW). Each heat, lot, or batch of filler metal and filler

metal/flux combination shall be pre-use tested as required by 3.11.4.5(d)(1) through 3.11.4.5(d)(3). Exemption from pre-use testing as allowed by 3.11.4.5(d)(4) and 3.11.4.5(d)(5) is not applicable. Toughness testing shall be performed as specified in (1) or (2) below, as appropriate.

(1) If using Type 316L weld filler metal, or Type 308L filler metal welded with the GTAW, FCAW, or GMAW process,

(-a) Weld metal deposited from each heat of Type 316L filler metal shall have a Ferrite Number (FN) no greater than 10, and a weld metal deposited from each heat of Type 308L filler metal shall have a FN in the range of 4 to 14, as measured by a ferritescope or magna gauge calibrated in accordance with AWS A4.2, or as determined by applying the chemical composition from the test weld to Figure 3.15.

(-b) Toughness tests shall be conducted at −196°C (−320°F) on three sets of three specimens: one set from the base metal, one set from the weld metal, one set from the HAZ.

(-c) Each of the three specimens from each test set shall have a lateral expansion opposite the notch not less than 0.53 mm (0.021 in.).

(2) When the qualifying conditions of (1) cannot be met:

(-a) Weld metal deposited from each heat or lot of austenitic stainless steel filler metal used in production shall have a FN no greater than the FN determined for the test weld.

(-b) Toughness tests shall be conducted at −196°C (−320°F) on a set of three specimens from the base metal. Each of three specimens shall have a lateral expansion opposite the notch no less than 0.53 mm (0.021 in.).

(-c) ASTM E1820 J_{Ic} tests shall be conducted on two sets of two specimens: one set from the HAZ, one set from the weld metal, at a test temperature no warmer than MDMT. The HAZ specimen orientation shall be T-L. A K_{Ic} (J) value no less than $132\,\mathrm{MPa}\sqrt{\mathrm{m}}\ \left(120\,\mathrm{ksi}\sqrt{\mathrm{in.}}\right)$ is required for all specimens tested.

(3) When the required toughness test specimens do not meet the lateral expansion requirements in (1)(-c) or (2)(-b), ASTM E1820 J_{Ic} tests shall be conducted on an additional set of two specimens representing the failed set of toughness test specimens at a test temperature no warmer than MDMT. The specimen orientation for the base metal and HAZ shall be T-L. A K_{Ic} (J) value no less than $132\,\mathrm{MPa}\sqrt{\mathrm{m}}\ \left(120\,\mathrm{ksi}\sqrt{\mathrm{in.}}\right)$ is required for all specimens tested.

3.11.4.2 Required Impact Tests When Thermal Treatments Are Performed. Impact tests are required at the test temperature in accordance with 3.11.4.1 but no warmer than 21°C (70°F) whenever thermal treatments within the temperature ranges listed for the following materials are applied.

(a) Austenitic stainless steels thermally treated between 480°C and 900°C (900°F and 1650°F), except for Types 304, 304L, 316, and 316L which are thermally treated at temperatures between 480°C and 705°C (900°F and 1300°F), are exempt from impact testing, provided the MDMT is −29°C (−20°F) and warmer and vessel production impact tests of the thermally treated weld metal are performed for Category A and B joints.

(b) Austenitic-ferritic duplex stainless steels thermally treated at temperatures between 315°C and 955°C (600°F and 1750°F).

(c) Ferritic chromium stainless steels and martensitic chromium stainless steels thermally treated at temperatures between 425°C and 730°C (800°F and 1350°F).

Thermal treatments of materials do not include thermal cutting.

3.11.4.3 Exemptions From Impact Testing for Base Materials and HAZs. Impact testing is not required for the following combinations of base metals and HAZs (if welded) and MDMT, except as modified in 3.11.4.2.

(a) For austenitic chromium-nickel stainless steels as follows:

(1) Those having a carbon content not exceeding 0.10% at MDMTs of −196°C (−320°F) and warmer. (The value of the carbon content may be specified by the purchaser, or must be within the limits of the material specification.);

(2) Those types having a carbon content exceeding 0.10% (the value of the carbon content may be as specified by the purchaser) at MDMTs of −48°C (−55°F) and warmer;

(3) For castings at MDMTs of −29°C (−20°F) and warmer.

(b) For austenitic chromium-manganese-nickel stainless steels (200 series) as follows:

(1) Having a carbon content not exceeding 0.10% at MDMTs of −196°C (−320°F) and warmer;

(2) Having a carbon content exceeding 0.10% at MDMTs of −48°C (−55°F) and warmer;

(3) For castings at MDMTs of −29°C (−20°F) and warmer.

(c) For the following steels in all product forms at MDMTs of −29°C (−20°F) and warmer:

(1) Austenitic-ferritic duplex steels with a nominal material thickness of 10 mm ($^3/_8$ in.) and thinner;

(2) Ferritic chromium stainless steels with a nominal material thickness of 3 mm ($^1/_8$ in.) and thinner;

(3) Martensitic chromium stainless steels with a nominal material thickness of 6 mm ($^1/_4$ in.) and thinner.

(d) Impact tests are not required where the maximum obtainable Charpy specimen has a width along the notch less than 2.5 mm (0.099 in.).

(e) Impact testing of materials is not required, except as modified by 3.11.4.2, when the coincident ratio of applied stress in tension to allowable tensile stress is less than 0.3 for Class 1, or 0.24 for Class 2. The applied stress is the stress from pressure and non-pressure loadings, including those listed in Table 4.1.1 which result in general primary membrane tensile stress.

3.11.4.4 Exemptions From Impact Testing for Welding Procedure Qualifications. For welding procedure qualifications, impact testing is not required for the following combinations of weld metals and MDMT except as modified by 3.11.4.2.

(a) For austenitic chromium-nickel stainless steel base materials having a carbon content not exceeding 0.10%, welded without the addition of filler metal, at MDMTs of −104°C (−155°F) and warmer.

(b) For austenitic weld metal:

(1) Having a carbon content not exceeding 0.10% and produced with filler metals conforming to SFA-5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 at MDMTs of −104°C (−155°F) and warmer;

(2) Having a carbon content exceeding 0.10% and produced with filler metals conforming, to SFA- 5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 at MDMTs of −48°C (−55°F) and warmer.

(c) For the following weld metal, if the base metal of similar chemistry is exempt as stated in 3.11.4.3(c) above, then the weld metal shall also be exempt at MDMTs of −29°C (−20°F) and warmer:

(1) Austenitic-ferritic duplex steels;

(2) Ferritic chromium stainless steels; and

(3) Martensitic chromium stainless steels.

3.11.4.5 Required Impact Testing for Austenitic Stainless Steel Welding Consumables With MDMT Colder Than −104°C (−155°F). For production welds at MDMTs colder than −104°C (−155°F), all of the following conditions shall be satisfied: (21)

(a) The welding processes are limited to SMAW, SAW, FCAW, GMAW, GTAW, and PAW;

(b) The applicable Welding Procedure Specifications (WPSs) are supported by Procedure Qualification Records (PQRs) with impact testing in accordance with the requirements of 3.11.7 and 3.11.4.1, or when the applicable PQR is exempted from impact testing by other provisions of this Division;

(c) The weld metal (produced with or without the addition of filler metal) has a carbon content not exceeding 0.10%;

(d) The weld metal is produced by filler metal conforming to Section II, Part C, SFA-5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 as modified below.

(1) Each heat and/or lot of welding consumables to be used in production welding with the SMAW, FCAW, and GMAW processes shall be pre-use tested by conducting impact tests in accordance with 3.11.4.1. Test coupons shall be prepared in accordance with Section II, Part C, SFA-5.4, A9.3.5 utilizing the WPS to be used in production welding.

(2) Each heat of filler metal and batch of flux combination to be used in production welding with the SAW process shall be pre-use tested by conducting impact tests in accordance with 3.11.4.1. Test coupons shall be prepared in accordance with Section II, Part C, SFA-5.4, A9.3.5 utilizing the WPS to be used in production welding.

(3) Combining more than one welding process or more than one heat, lot, and/or batch of welding material into a single test coupon is unacceptable. Pre-use testing in accordance with 3.11.4.1 may be conducted by the welding consumable manufacturer, provided mill test reports are furnished with the consumables.

(4) The following filler metals may be used without pre-use testing of each heat, lot, and/or batch provided that the procedure qualification impact testing in accordance with 3.11.8 at the MDMT or colder is performed using the same manufacturer brand and type filler metal: ENiCrFe-2; ENiCrFe-3; ENiCrMo-3; ENiCrMo-4; ENiCrMo-6; ERNiCr-3; ERNiCrMo-3; ERNiCrMo-4; SFA-5.4, E310-15 or 16.

(5) The following filler metals may be used without pre-use testing of each heat and/or lot provided that procedure qualification impact testing in accordance with 3.11.8 at the MDMT or colder is performed: ER308L, ER316L, and ER310 used with the GTAW or PAW processes.

3.11.4.6 Required Impact Testing for Vessel Production Test Plates.

(a) For welded construction, of duplex stainless steels, ferritic stainless steels and martensitic stainless steels, vessel production impact tests in accordance with 3.11.8.4 are required if the welding procedure qualification requires impact testing, unless otherwise exempted by the rules of this Division.

(b) For welded construction of austenitic stainless steels, vessel (production) impact tests in accordance with 3.11.8.4 are required unless exempted as follows in (1) and (2):

(1) At MDMTs of −104°C (−155°F) and warmer, vessel (production) impact tests are exempted, provided the impact test exemption requirements for the applicable Weld Procedure Qualification in 3.11.4.4 are satisfied.

(2) At MDMTs colder than −104°C (−155°F) but no colder than −196°C (−320°F), vessel (production) impact tests are exempted, provided the pre-use test requirements in 3.11.4.5 are satisfied.

(3) At MDMTs colder than −196°C (−320°F), vessel (production) impact tests or ASTM E1820 J_{Ic} tests shall be conducted in accordance with 3.11.4.1(c).

(c) Vessel Production Impact Testing for Autogeneous Welds in Austenitic Stainless Steels – For autogenous welds (welded without filler metal) in austenitic stainless steels, vessel (production) impact tests are not required when all of the following conditions are satisfied:

(1) The material is solution annealed after welding.

(2) The MDMT is not colder than −196°C (−320°F).

3.11.5 NONFERROUS ALLOYS

3.11.5.1 Nonferrous materials listed in Tables 3-A.4 through 3-A.7, together with deposited weld metal within the range of composition for material in that Table, do not undergo a marked drop in impact resistance at subzero temperature. Therefore, additional requirements are not specified for:

(a) Wrought aluminum alloys when they are used at temperature down to −269°C (−452°F);

(b) Copper and copper alloys, nickel and nickel alloys, and cast aluminum alloys when they are used at temperatures down to −198°C (−325°F); and

(c) Titanium or zirconium and their alloys used at temperatures down to −59°C (−75°F).

3.11.5.2 The nonferrous materials listed in Tables 3-A.4 through 3-A.7 may be used at lower temperatures than those specified herein and for other weld metal compositions, provided the user satisfies himself by suitable test results such as determinations of tensile elongation and sharp-notch tensile strength (compared to unnotched tensile strength) that the material has suitable ductility at the design temperature.

3.11.6 BOLTING MATERIALS

3.11.6.1 Bolting Materials for Use With Flanges Designed to 4.16.

(a) Impact tests are not required for bolting materials listed in Tables 3.4, 3.5, 3.6, and 3.7 when used at MDMTs equal to or warmer than those shown in these Tables.

(b) Bolting materials to be used for colder temperatures than those shown in Tables 3.4 through 3.7 shall conform to SA-320, except that the toughness criterion shall be Charpy V-notch with acceptance criteria in accordance with 3.11.2 or 3.11.4, as applicable.

3.11.6.2 Bolting Materials for Use With Flanges Designed to Part 5 of This Division. Impact testing is required for the ferrous bolting materials listed in Table 3-A.11 for use with flanges designed in accordance with Part 5 of this Division. The average for three Charpy V-notch impact specimens shall be at least 41 J (30 ft-lb), with the minimum value for any individual specimen not less than 34 J (25 ft-lb).

3.11.7 TOUGHNESS TESTING PROCEDURES

3.11.7.1 Test Procedures.

(a) Impact test procedures and apparatus shall conform to the applicable paragraphs of SA-370 or ISO 148 (Parts 1, 2, and 3).

(b) The impact test temperature shall not be warmer than the MDMT (see 4.1.5.2(e)).

3.11.7.2 Test Specimens.

(a) Each set of impact tests shall consist of three specimens.

(b) The impact test specimens shall be of the Charpy V-notch type and shall conform in all respects to the specimen requirements of SA-370 (for Type A specimens). The standard full-size (10 mm × 10mm) specimen, when obtainable, shall be used, except that for materials that normally have absorbed energy in excess of 244 J (180 ft-lb) when tested using full size specimens at the specified testing temperature, subsize (10 mm × 6.7 mm) specimens may be used in lieu of full-size specimens. However, when this option is used, the acceptance value shall be 102 J (75 ft-lb) minimum for each specimen.

(c) For material from which full-size specimens cannot be obtained, either due to the material shape or thickness, the specimens shall be either the largest possible subsize specimen obtainable or specimens of full material thickness which may be machined to remove surface irregularities [the test temperature criteria of 3.11.7.5 shall apply for carbon and low alloy materials having a specified minimum tensile strength less than 655 MPa (95 ksi) when the width along the notch is less than 80% of the material thickness]. Alternatively, such material may be reduced in thickness to produce the largest possible Charpy subsize specimen. Toughness tests are not required where the maximum obtainable Charpy specimen has a width along the notch less than 2.5 mm (0.099 in.), but carbon steels too thin to impact test shall not be used for design temperatures colder than −48°C (−55°F), subject to the exemptions provided by 3.11.2.9.

3.11.7.3 Product Forms.

(a) Impact test specimens of each product form shall be located and oriented in accordance with the requirements of 3.10.4.

(b) The manufacturer of small parts, either cast or forged, may certify a lot of not more than 20 duplicate parts by reporting the results of one set of impact specimens taken from one such part selected at random, provided the same specification and heat of material and the same process of production, including heat treatment, were used for all of the lot. When the part is too small to provide the three specimens of at least minimum size indicated in 3.11.7.2, then impact test do not need to be performed (see 3.11.7.2(c)).

3.11.7.4 Certification of Compliance With Impact Test Requirements.

(a) Certified reports of impact tests by the materials manufacturer will be acceptable evidence that the material meets the requirements of this paragraph, provided:

(1) The specimens taken are representative of the material delivered [see 3.11.7.3(a)] and the material is not subjected to heat treatment during or following fabrication that will materially reduce its impact properties; or

(2) The materials from which the specimens are removed are heat treated separately such that they are representative of the material in the finished vessel.

(b) The Manufacturer of the vessel may have impact tests made to prove the suitability of a material which the materials manufacturer has not impact tested, provided the number of tests and the method of taking the test specimens shall be as specified for the materials manufacturer.

3.11.7.5 Impact Test Temperature Criteria. For all Charpy impact tests, the following test temperature criteria shall be observed.

(a) Materials of Thickness Equal to or Greater Than 10 mm (0.394 in.) – Where the largest obtainable Charpy V-notch specimen has a width along the notch of at least 8 mm (0.315 in.), the Charpy test of such a specimen shall be conducted at a temperature not warmer than the MDMT. Where the largest possible test specimen has a width along the notch less than 8 mm (0.315 in.), the test shall be conducted at a temperature colder than the MDMT by the amount shown in Table 3.11 for the specimen width. Note that this requirement does not apply when the option of 3.11.7.2(b) is used.

(b) Materials With Thickness Less Than 10 mm (0.394 in.) – Where the largest obtainable Charpy V-notch specimen has a width along the notch of at least 80% of the material thickness, the Charpy test of such a specimen shall be conducted at a temperature not warmer than the MDMT. Where the largest possible test specimen has a width along the notch of less than 80% of the material thickness, the test for carbon steel and low alloy materials having a specified minimum tensile strength of less than 655 MPa (95 ksi) shall be conducted at a temperature colder than the MDMT by an amount equal to the difference, see Table 3.11, between the temperature reduction corresponding to the actual material thickness and the temperature reduction corresponding to the Charpy specimen width actually tested. This requirement does not apply when the option of 3.11.7.2(b) is used. For Table 3-A.2, carbon and low alloy materials having a specified minimum tensile strength greater than or equal to 655 MPa (95 ksi), for high alloy materials and quenched and tempered material with enhanced tensile properties, the test shall be conducted at a temperature not warmer than the MDMT.

3.11.7.6 Retests. (**21**)

(a) Absorbed Energy Criteria – If the absorbed energy criteria are not met, retesting in accordance with the applicable procedures of SA-370 shall be permitted.

(b) Lateral Expansion Criteria – retests shall be performed as follows:

(1) Retesting is permitted if the average value for three specimens equals or exceeds the value required.

(-a) For materials of Table 3-A.1 (carbon and low alloy steels) having specified minimum tensile strengths of 655 MPa (95 ksi) or greater, if the measured value of lateral expansion for one specimen in a group of three is less than that required in Figure 3.6.

(-b) For materials of Table 3-A.3 (high alloy steels) for MDMTs no colder than −196°C (−320°F), if the measured value of lateral expansion for one specimen in a group of three is less than 0.38 mm (0.015 in.), but not less than two-thirds of the value required.

(-c) For materials of Table 3-A.3 (high alloy steels) for for MDMTs colder than −196°C (−320°F), if the value of lateral expansion for one specimen of a set is less than 0.53 mm (0.021 in.).

(-d) For materials of Table 3-A.2 (Q&T steel with enhanced strength properties), if the measured value of lateral expansion for one specimen in a group of three is less than that required in Figure 3.6 but not less than two-thirds of the required value.

(2) The retest shall consist of three additional specimens. For materials of Table 3-A.1 (carbon and low alloy steels) having specified minimum tensile strengths of 655 MPa (95 ksi) or greater and for Table 3-A.2 (Q&T steels with enhanced strength properties) materials, the retest value for each specimen must equal or exceed the value required in

Figure 3.6. For materials of Table 3-A.3 (high alloy steels), the retest value for each specimen must equal or exceed 0.38 mm (0.015 in.) for MDMTs no colder than −196°C (−320°F). For MDMTs colder than −196°C (−320°F), see 3.11.2.1(b)(2) and 3.11.4.1(b).

(3) In the case of materials with properties enhanced by heat treatment, the material may be reheat treated and retested if the required values are not obtained in the retest or if the values in the initial test are less than the values required for retest. After reheat treatment, a set of three specimens shall be made; for acceptance, the lateral expansion of each of the specimens must equal or exceed the value required in Figure 3.6.

(c) When an erratic result is caused by a defective specimen or there is uncertainty in the test procedure, a retest will be allowed. When the option of 3.11.7.2(b) is used for the initial test and the acceptance of 102 J (75 ft-lb) minimum is not attained, a retest using full-size (10 mm × 10 mm) specimens will be allowed.

3.11.8 IMPACT TESTING OF WELDING PROCEDURES AND TEST PLATES OF FERROUS MATERIALS

(21) **3.11.8.1 Impact Tests.**

(a) For steel vessels of welded construction, the impact toughness of welds and heat-affected zones of procedure qualification test plates and vessel test plates (production impact test plates) shall be determined as required in this paragraph.

(b) All test plates shall be subjected to heat treatment, including cooling rates and aggregate time at temperature or temperatures as established by the manufacturer for use in actual manufacture. Heat treatment requirements of 6.4.2, 3.10.2, and 3.10.4 shall apply to test plates, except that the provisions of 3.10.3.3 are not applicable to test plates for welds joining P-No. 3, Groups 1 and 2 materials. For P-No. 1, Groups 1, 2, and 3 materials, impact testing of the welds and heat-affected zones of the weld procedure qualification and production test plates need not be repeated when the fabrication heat treatment differs from the heat treatment applied to the test plates, provided the PWHT or simulated heat treatment cycles applied to the test plates and the production welds were applied observing the holding temperatures and times specified in Table 6.8 or the holding temperatures and times permitted in Table 6.16.

3.11.8.2 Location, Orientation, Temperature, and Values of Weld Impact Tests.

(a) All weld impact tests shall comply with the following requirements.

(b) Each set of weld metal impact specimens shall be taken across the weld with the notch in the weld metal. Each specimen shall be oriented so that the notch is normal to the surface of the material, and one face of the specimen shall be within 1.5 mm ($^1/_{16}$ in.) of the surface of the material. When procedure tests are made on material over 38 mm ($1^1/_2$ in.) in thickness, two sets of impact specimens shall be taken from the weld with one set located within 1.5 mm ($^1/_{16}$ in.) of the surface of one side of the material and one set taken as near as practical midway between the surface and the center of thickness of the opposite side as described above [see Section IX, QW-200.4(a)].

(c) Each set of heat-affected zone impact specimens shall be taken across the weld and of sufficient length to locate, after etching, the notch in the affected zone. The number of heat-affected zone impact specimen sets to be removed, and the location of the centerline in the prepared test specimens, shall be as shown in Table 3.18 and Figure 3.16. Test specimens that are full sized or the largest obtainable subsized test specimens that have been removed and prepared with the width along the notch located fully within the specified range of removal depth are acceptable. The notch shall be cut approximately normal to the material surface in such a manner as to include as much heat-affected zone material as possible in the resulting fracture. Where the material thickness permits, the axis of the notch may be inclined to allow the root of the notch to align parallel to the fusion line.

(d) For welds made by a solid-state welding process, such as for electric resistance-welded (ERW) pipe, the weld impact tests shall consist only of one set of three specimens taken across the weld with the notch at the weld centerline. Each specimen shall be oriented so that the notch is normal to the surface of the material and one face of the specimen shall be within 1.5 mm ($^1/_{16}$ in.) of the surface of the material.

(e) The test temperature for welds and heat-affected zones shall not be higher than for the base materials.

(f) Impact values shall be at least as high as those required for the base materials (see 3.11.2, 3.11.3, and 3.11.4, as applicable).

(g) When qualifying a WPS for welding base metals having different impact testing requirements and acceptance criteria, the following shall apply:

(1) The weld metal impact test specimens shall meet the acceptance criteria for either base metal.

(2) When HAZ tests are required, separate test specimens shall be removed from the HAZ of each base metal that requires impact testing, and those specimens shall meet the acceptance criteria for the base metal from which they were removed.

3.11.8.3 Impact Tests for Welding Procedures.

(a) Welding procedure impact tests shall be made on welds and heat-affected zones when base materials are required to be impact tested, except as exempted by 3.11.4.4 and 3.11.2.10.

(b) If impact tests are required for the deposited weld, but the base material is exempted from impact tests, welding procedure test plates shall be made. The test plate material shall be material of the same P-Number and Group Number used in the vessel. One set of impact specimens shall be taken with the notch approximately centered in the weld metal and perpendicular to the surface; the heat-affected zone need not be impact tested.

(c) When the welding procedure employed for production welding is used for fillet welds only, it shall be qualified by a groove weld qualification test. The qualification test plate or pipe material shall meet the requirements of 3.11.7 when impact testing is a requirement. This welding procedure test qualification is in addition to the requirements of Section IX, QW-202.2 for P-No. 11 materials.

(d) The supplementary essential variables specified in Section IX, QW-250, for impact testing are required.

(e) For test plates or pipe receiving a postweld heat treatment in which the lower critical temperature is exceeded, the maximum thickness qualified is the thickness of the test plate or pipe.

(f) For materials of Table 3-A.1 (carbon steel and low alloy steel), the test plate material shall satisfy all of the following requirements relative to the material to be used in production:

(1) Be of the same P-Number and Group Number;

(2) Be in the same heat-treated condition, and the heat-treated condition shall be noted on the PQR and WPS used for construction;

(3) Meet the minimum toughness requirements 3.11.2, 3.11.3, and 3.11.4, as applicable for the thickest material of the range of base material qualified by the procedure.

(g) Multiple-Process Welding Procedures. When qualifying a welding procedure with impact testing that employs multiple welding processes, or multiple sets of essential and supplementary essential variables for a welding process, the welding procedure shall be qualified by testing separate sets of impact test specimens removed from the weld metal and heat-affected zone, as follows:

(1) The requirements of 3.11.8.1 shall be met.

(2) The requirements of 3.11.8.2 specifying the location, number, and orientation of test specimens to be removed for each welding process or set of variables shall be modified as follows:

(-a) The weld metal thickness shall be considered the base metal thickness.

(-b) The surface of the last deposited layer of weld metal shall be considered the weld surface.

(-c) The root side of the first deposited layer of weld metal shall be considered the root surface.

(3) If the weld thickness for a welding process or set of variables is small enough that the maximum obtainable Charpy specimen has a width along the notch less than 2.5 mm (0.099 in.), toughness testing of the weld metal and heat-affected zone is not required for that welding process or set of variables.

3.11.8.4 Impact Tests of Vessel Test Plates.

(a) When the base material or welding procedure qualification requires impact testing, impact tests of welds and heat-affected zones shall be made for Category A and B joints in accordance with 3.11.8.2 and 3.11.8.3 for each qualified welding procedure used on each vessel. The test plate shall be from one of the heats of steel used for the vessel or group of vessels and shall be welded as an extension to the end of a production Category A joint where practicable, or welded as close to the start of production welding as practicable, utilizing equipment, welding materials, and procedures which are to be used on the production joint. The test plate shall also represent each welding process or welding process combination used in production on Category A joints.

(b) For Category B joints that are welded using a different welding procedure than used on Category A joints, a test plate shall be welded under the production welding conditions used for the vessel, using the same type of equipment and at the same location and using the same procedures as used for the joint, and it shall be welded concurrently with the production welds or as close to the start of production welding as practicable. The test plate shall also represent each welding process or welding process combination used in production on Category B joints.

(c) Number of Vessel Impact Test Plates Required

(1) For each vessel, one test plate shall be made for each welding procedure used for joints of Categories A and B, unless the vessel is one of several as defined in (2). In addition, for Category A and B joints, the following requirements shall apply:

(-a) If automatic, machine, or semiautomatic welding is performed, a test plate shall be made in each position employed in the vessel welding.

(-b) If manual welding is also employed, a test plate shall be made in the flat position only, except if welding is to be performed in other positions a test plate need be made in the vertical position only (where the major portions of the layers of welds are deposited in the vertical upward direction). The vertically welded test plate will qualify the manual welding in all positions.

(-c) The vessel test plate shall qualify the impact requirements for vessel materials thickness in accordance with Section IX, Tables QW-451.1 and QW-451.2 (including Notes), except that, if the thickness is less than 16 mm ($^5/_8$ in.), the thickness of the test material is the minimum thickness qualified.

(2) For several vessels or parts of vessels, welded within any 3 month period at one location, the plate thickness of which does not vary by more than 6 mm ($^1/_4$ in.) or 25%, whichever is greater, and of the same specification and grade of material, a test plate shall be made for each 122 m (400 ft) of joints welded by the same procedure.

(d) If the vessel test plate fails to meet the impact requirements, the welds represented by the test plate shall be unacceptable. Reheat treatment and retesting, or retesting only, are permitted.

3.12 ALLOWABLE DESIGN STRESSES

The design stresses for materials permitted by this Division are given in Annex 3-A.

3.13 STRENGTH PARAMETERS

The strength parameters for materials permitted by this Division are given in Annex 3-D.

3.14 PHYSICAL PROPERTIES

The following physical properties for all permissible materials of construction are given in the tables referenced in Annex 3-E.

(a) Young's Modulus
(b) Thermal Expansion Coefficient
(c) Thermal Conductivity
(d) Thermal Diffusivity

3.15 DESIGN FATIGUE CURVES

Design fatigue curves for nonwelded and for welded construction are provided in Annex 3-F. As an alternative, the adequacy of a part to withstand cyclic loading may be demonstrated by means of fatigue test following the requirements of Annex 5-F. However, the fatigue test shall not be used as justification for exceeding the allowable values of primary or primary plus secondary stresses.

3.16 DESIGN VALUES FOR TEMPERATURES COLDER THAN −30°C (−20°F)

For design temperatures colder than −30°C (−20°F), the allowable design stress values and strength parameter values to be used in design shall not exceed those given in the pertinent tables in Section II, Part D for −30°C to 40°C (−20°F to 100°F), unless specifically addressed elsewhere in this Division.

3.17 NOMENCLATURE

a = reference flaw depth.
$2c$ = reference flaw length.
E = joint efficiency (see Part 7) used in the calculation of t_r. For castings, the quality factor or joint efficiency E, whichever governs design, shall be used.
E^* = E^* equal to E except that E^* shall not be less than 0.80, or $E^* = \max[E, 0.80]$.
CA = corrosion allowance
MDMT = minimum design metal temperature.
P_a = applied pressure for the condition under consideration.
P_{rating} = maximum allowable working pressure based on the design rules in this Division of ASME/ANSI pressure–temperature ratings.
R_{ts} = stress ratio defined as the stress for the operating condition under consideration divided by the stress at the design minimum temperature. The stress ratio may also be defined in terms of required and actual thicknesses, and for components with pressure–temperature ratings, the stress ratio is computed as the applied pressure for the condition under consideration divided by the pressure rating at the MDMT.

S = allowable stress from Annex 3-A
S_y = specified minimum yield strength.
S^* = applied general primary stress.
t = reference flaw plate thickness.
t_g = governing thickness.
t_n = nominal uncorroded thickness. For welded pipe where a mill undertolerance is allowed by the material specification, the thickness after mill undertolerance has been deducted shall be taken as the nominal thickness. Likewise, for formed heads, the minimum specified thickness after forming shall be used as the nominal thickness.
t_r = required thickness of the part under consideration in the corroded condition for all applicable loadings
T_R = reduction in MDMT based on available excess thickness.

3.18 DEFINITIONS

The definitions for the terminology used in this Part are contained in Annex 1-B.

3.19 TABLES

Table 3.1
Material Specifications

Nominal Composition	Type/Grade	Specification	Product Form
$2^1/_4$Cr–1Mo	Grade 22, Cl. 3	SA-508	Forgings
	Grade 22, Cl. 3	SA-541	Forgings
	Type B, Cl. 4	SA-542	Plates
	Grade 10CrMo9–10	SA/EN 10028-2	Plates
$2^1/_4$Cr–1Mo–$^1/_4$V	Grade F22V	SA-182	Forgings
	Grade F22V	SA-336	Forgings
	Grade 22V	SA-541	Forgings
	Type D, Cl. 4a	SA-542	Plates
	Grade 22V	SA-832	Plates
3Cr–1Mo–$^1/_4$V–Cb–Ca	Grade F3VCb	SA-182	Forgings
	Grade F3VCb	SA-336	Forgings
	Grade 3VCb	SA-508	Forgings
	Grade 3VCb	SA-541	Forgings
	Type E, Cl. 4a	SA-542	Plates
	Grade 23V	SA-832	Plates
3Cr–1Mo–$^1/_4$V–Ti–B	Grade F3V	SA-182	Forgings
	Grade F3V	SA-336	Forgings
	Grade 3V	SA-508	Forgings
	Grade 3V	SA-541	Forgings
	Type C, Cl. 4a	SA-542	Plates
	Grade 21 V	SA-832	Plates

Table 3.2
Composition Requirements for 2.25Cr–1Mo–0.25V Weld Metal

Welding Process	C	Mn	Si	Cr	Mo	P	S	V	Cb
SAW	0.05–0.15	0.50–1.30	0.05–0.35	2.00–2.60	0.90–1.20	0.015 max	0.015 max	0.20–0.40	0.010–0.040
SMAW	0.05–0.15	0.50–1.30	0.20–0.50	2.00–2.60	0.90–1.20	0.015 max	0.015 max	0.20–0.40	0.010–0.040
GTAW	0.05–0.15	0.30–1.10	0.05–0.35	2.00–2.60	0.90–1.20	0.015 max	0.015 max	0.20–0.40	0.010–0.040
GMAW	0.05–0.15	0.30–1.10	0.20–0.50	2.00–2.60	0.90–1.20	0.015 max	0.015 max	0.20–0.40	0.010–0.040

Table 3.3
Toughness Requirements for 2.25Cr–1Mo Materials

Number of Specimens	Impact Energy, J (ft-lb)
Average of 3	54 (40)
Only one in the set	48 (35)

GENERAL NOTE: Full size Charpy V-notch, transverse, tested at the MDMT.

Table 3.4
Low Alloy Bolting Materials for Use With Flanges Designed to 4.16

Material Specification	Material Type/Grade	Diameter, mm (in.)	MDMT Without Impact Testing, °C (°F)
Low Alloy Bolting			
SA-193	B5	Up to 102 (4), inclusive	−29 (−20)
	B7	64 (2½) and under	−48 (−55)
		Over 64 to 102 (2½ to 4), inclusive	−40 (−40)
		Over 102 to 178 (4 to 7), inclusive	−40 (−40)
	B7M	64 (2½) and under	−48 (−55)
	B16	64 (2½) and under	−29 (−20)
		Over 64 to 102 (2½ to 4), inclusive	−29 (−20)
		Over 102 to 178 (4 to 7), inclusive	−29 (−20)
SA-320	L7	64 (2½) and under	See 3.11.2.4(b)
	L7 A	Up to 64 (2½), inclusive	See 3.11.2.4(b)
	L7M	64 (2½) and under	See 3.11.2.4(b)
	L43	25 (1) and under	See 3.11.2.4(b)
SA-325	1	13 to 38 (½ to 1½), inclusive	−29 (−20)
SA-354	BC	Up to 102 (4),	−18 (0)
	BD	Up to 102 (4), inclusive	−7 (+20)
SA-437	B4B, B4C	All diameters	See 3.11.2.4(b)
SA-449	...	Up to 76 (3), inclusive	−29 (−20)
SA-508	5 Cl. 2	All diameters	See 3.11.2.4(b)
SA-540	B21	All diameters	Impact test is required
	B23 Cl. 1 & 2	All diameters	Impact test is required
	B23 Cl. 3 & 4	Up to 152 (6), inclusive	See 3.11.2.4(b)
		Over 152 to 241 (6 to 9½), inclusive	Impact test is required
	B23 Cl. 5	Up to 203 (8), inclusive	See 3.11.2.4(b)
		Over 203 to 241 (8 to 9½), inclusive	Impact test is required
	B24 Cl. 1	Up to 152 (6), inclusive	See 3.11.2.4(b)
		Over 152 to 203 (6 to 8), inclusive	Impact test is required
	B24 Cl. 2	Up to 178 (7), inclusive	See 3.11.2.4(b)
		Over 178 to 241 (7 to 9½), inclusive	Impact test is required
	B24 Cl. 3 & 4	Up to 203 (8), inclusive	See 3.11.2.4(b)
		Over 203 to 241 (8 to 9½), inclusive	Impact test is required
	B24 Cl. 5	Up to 241 (9½), inclusive	See 3.11.2.4(b)
	B24V Cl. 3	All diameters	See 3.11.2.4(b)
Low Alloy Steel Nuts			
SA-194	2, 2H, 2HM, 3, 4, 7, 7M, 16	All diameters	−48 (−55)
SA-540	B21, B23, B24, B24V	All diameters	−48 (−55)

Table 3.5
High Alloy Bolting Materials for Use With Flanges Designed to 4.16

Material Specification	Material Type/Grade	Diameter, mm (in.)	MDMT Without Impact Testing, °C (°F)
SA-193	B6	102 (4) and under	−29 (−20)
	B8 Cl. 1	All diameters	−254 (−425)
	B8 Cl. 2	Up to 38 ($1\frac{1}{2}$), inclusive	Impact test is required
	B8C Cl. 1	All diameters	−254 (−425)
	B8C Cl. 2	19 to 38 (0.75 to $1\frac{1}{2}$), inclusive	Impact test is required
SA-193	B8M Cl. 1	All diameters	−254 (−425)
	B8M2	51 to 64 (2 to $2\frac{1}{2}$), inclusive	Impact test is required
	B8MNA Cl. 1A	All diameters	−196 (−320)
	B8NA Cl. 1A	All diameters	−196 (−320)
	B8P Cl. 1	All diameters	Impact test is required
	B8P Cl. 2	Up to 38 ($1\frac{1}{2}$), inclusive	Impact test is required
	B8S, 88SA	All diameters	Impact test is required
	B8T Cl. 1	All diameters	−254 (−425)
	B8T Cl, 2	19 to 25 ($\frac{3}{4}$ to 1), inclusive	Impact test is required
SA-320	B8 Cl. 1	All diameters	See 3.11.2.4(b)
	B8 Cl. 2	Up to 25 (1), inclusive	See 3.11.2.4(b)
	B8A Cl. 1A	All diameters	See 3.11.2.4(b)
	B8C Cl. 1 & 1A	All diameters	See 3.11.2.4(b)
	B8C Cl. 2	Up to 25 (1), inclusive	See 3.11.2.4(b)
	B8CA Cl. 1A	All diameters	See 3.11.2.4(b)
	B8F Cl. 1	All diameters	See 3.11.2.4(b)
	B 8FA Cl. 1A	All diameters	See 3.11.2.4(b)
	B8M Cl. 1	All diameters	See 3.11.2.4(b)
	B8M Cl. 2	Up to 38 ($1\frac{1}{2}$), inclusive	See 3.11.2.4(b)
	B8MA Cl. 1A	All diameters	See 3.11.2.4(b)
	B8T Cl. 1	All diameters	See 3.11.2.4(b)
	B8T Cl. 2	Up to 38 ($1\frac{1}{2}$), inclusive	See 3.11.2.4(b)
	B8TA Cl. 1A	All diameters	See 3.11.2.4(b)
SA-453	651 Cl. A & B, 660 Cl. A & B	All diameters	Impact test is required
SA-479	XM-19	Up to 8 (203), inclusive	Impact test is required
SA-564	630	Up to 8 (203), inclusive.	Impact test is required
SA-705	630	Up to 8 (203), inclusive.	Impact test is required

Table 3.6
Aluminum Alloy, Copper, and Copper Alloy Bolting Materials for Use With Flanges Designed to 4.16

Material Specification	UNS
SB-98	C65100, C65500, C66100
SB-150	C61400, C62300, C63000, C64200
SB-187	C10200, C11000
SB-211	A92014, A92024, A96061

GENERAL NOTE: The MDMT for all bolting material listed in this Table is −196°C (-320°F).

Table 3.7
Nickel and Nickel Alloy Bolting Materials for Use With Flanges Designed to 4.16

Material Specification	UNS
SB-160	N02200, N02201
SB-164	N04400 N04405
SB-166	N06600
SB-335	N10001, N10665
SB-408	N08800, N08810
SB-425	N08825
SB-446	N06625
SB-572	N06002, R30556
SB-573	N10003
SB-574	N06022, N06455, N10276
SB-581	N06007, N06030, N06975
SB-621	N08320
SB-637	N07718, N07750

GENERAL NOTE: The MDMT for all bolting material listed in this Table is −196°C (−320°F).

Table 3.8
Bolting Materials for Use With Flanges Designed to Part 5

Material Specification	Material Grade
SA-193	B5, B6, B7, B7M, B8, B8C, B8M, B8MNA, B8NA, B8R, B8RA, B8S, B8SA, B8T, B16
SA-320	L43
SA-437	B4B, B4C
SA-453	651, 660
SA-540	B21, B22, B23, B24, B24V
SA-564	630
SA-705	630
SB-164	N04400, N04405
SB-637	N07718, N07750

GENERAL NOTE: See 3.11.6.2 for impact testing requirements.

Table 3.9
Maximum Severity Levels for Castings With a Thickness of Less Than 50 mm (2 in.)

Imperfection Category	Thickness <25 mm (1 in.)	Thickness 25 mm < 50 mm (1 in. < 2 in.)
A – Gas porosity	1	2
B – Sand and slag	2	3
C – Shrinkage (four types)	1	3
D – Cracks	0	0
E – Hot tears	0	0
F - Inserts	0	0
G – Mottling	0	0

Table 3.10
Maximum Severity Levels for Castings With a Thickness of 50 mm to 305 mm (2 in. to 12 in.)

Imperfection Category	Thickness 50 mm to 115 mm (2 in. to $4\frac{1}{2}$ in.)	Thickness >115 mm to 305 mm (>$4\frac{1}{2}$ in. to 12 in.)
A - Gas porosity	2	2
B – Sand and slag inclusions	2	2
C – Shrinkage – Type 1	1	2
C – Shrinkage – Type 2	2	2
C – Shrinkage – Type 3	3	2
D – Cracks	0	0
E – Hot tears	0	0

Table 3.11
Charpy Impact Test Temperature Reduction Below the Minimum Design Metal Temperature

Actual Material Thickness [See 3.11.7.5(b)] or Charpy Impact Specimen Width Along the Notch		Temperature Reduction	
mm	in.	°C	°F
10 (full-size standard bar)	0.394	0	0
9	0.354	0	0
8	0.315	0	0
7.5 ($\frac{3}{4}$ size bar)	0.295	3	5
7	0.276	4	8
6.65 ($\frac{2}{3}$ size bar)	0.262	6	10
6	0.236	8	15
5 ($\frac{1}{2}$ size bar)	0.197	11	20
4	0.158	17	30
3.33 ($\frac{1}{3}$ size bar)	0.131	19	35
3	0.118	22	40
2.5 ($\frac{1}{4}$ size bar)	0.099	28	50

GENERAL NOTES:
(a) Straight line interpolation for intermediate values is permitted.
(b) For carbon and low alloy materials having a specified minimum tensile strength of less than 655 MPa (95 ksi) when the subsize Charpy impact width is less than 80% of the material thickness.

Table 3.12
Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels as a Function of the Minimum Specified Yield Strength — Parts Not Subject to PWHT (See Figures 3.3 and 3.3M)

Thickness, mm	CVN, (J) Specified Minimum Yield Strength, MPa					Thickness, in.	CVN, (ft-lb) Specified Minimum Yield Strength, ksi				
	205	260	345	450	550		30	38	50	65	80
6	27	27	27	27	27	0.25	20	20	20	20.	20
10	27	27	27	27	31	0.375	20	20	20	20	23
13	27	27	27	27	36	0.5	20	20	20	20	27
16	27	27	27	29	43	0.625	20	20	20	21	32
19	27	27	27	34	51	0.75	20	20	20	25	37
25	27	27	27	45	62	1	20	20	20	33	46
32	27	27	34	53	72	1.25	20	20	25	39	53
38	27	27	40	61	82	1.5	20	20	30	45	60

GENERAL NOTE: The Charpy V-notch values given in this table represent a smooth curve in Figures 3.3 and 3.3M.

Table 3.13
Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels as a Function of the Minimum Specified Yield Strength — Parts Subject to PWHT or Nonwelded Parts (See Figures 3.4 and 3.4M)

Thickness, mm	CVN, J: Specified Minimum Specified Yield Strength, MPa					Thickness, in.	CVN, ft-lb: Specified Minimum Specified Yield Strength, ksi				
	205	260	345	450	550		30	38	50	65	80
6	27	27	27	27	27	0.25	20	20	20	20	20
10	27	27	27	27	27	0.375	20	20	20	20	20
13	27	27	27	27	27	0.5	20	20	20	20	20
16	27	27	27	27	27	0.625	20	20	20	20	20
19	27	27	27	27	27	0.75	20	20	20	20	20
25	27	27	27	27	27	1	20	20	20	20	20
32	27	27	27	27	34	1.25	20	20	20	20	25
38	27	27	27	27	40	1.5	20	20	20	20	30
44	27	27	27	31	47	1.75	20	20	20	23	35
51	27	27	27	35	52	2	20	20	20	26	38
57	27	27	27	40	56	2.25	20	20	20	29	41
64	27	27	27	43	60	2.5	20	20	20	32	44
70	27	27	29	46	64	2.75	20	20	21	34	47
76	27	27	31	49	68	3	20	20	23	36	50
83	27	27	33	52	71	3.25	20	20	25	38	52
89	27	27	35	54	74	3.5	20	20	26	40	54
95	27	27	37	56	76	3.75	20	20	27	42	56
102	27	27	38	58	78	4	20	20	28	43	58.
108	27	27	39	59	80	4.25	20	20	29	44	59
114	27	27	40	60	81	4.5	20	20	29	45	60
121	27	27	40	61	82	4.75	20	20	30	45	60
127	27	27	41	61	82	5	20	20	30	45	61
133	27	27	41	61	82	5.25	20	20	30	45	61
140	27	27	41	61	82	5.5	20	20	30	45.	61
146	27	27	41	61	82	5.75	20	20	30	45	61
152	27	27	41	61	82	6	20	20	30	45	61
159	27	27	41	61	82	6.25	20	20	30	45	61
165	27	27	41	61	82	6.5	20	20	30	45	61
171	27	27	41	61	82	6.75	20	20	30	45	61
178	27	27	41	61	82	7	20	20	30	45	61

GENERAL NOTE: The Charpy V-notch values given in this table represent a smooth curve in Figures 3.4 and 3.4M.

Table 3.14
Impact Test Exemption Curves — Parts Not Subject to PWHT (See Figures 3.7 and 3.7M)

Thickness, mm	Exemption Curve, °C A	B	C	D	Thickness, in.	Exemption Curve, °F A	B	C	D
0	20.5	−0.6	−21.7	−36.1	0	68.9	30.9	−7.1	−33.1
10	20.5	−0.6	−21.7	−36.1	0.394	68.9	30.9	−7.1	−33.1
13	22.9	1.8	−19.3	−33.7	0.5	73.3	35.3	−2.7	−28.7
16	26.3	5.1	−16.0	−30.4	0.625	79.3	41.3	3.3	−22.7
19	29.6	8.5	−12.6	−27.1	0.75	85.3	47.3	9.3	−16.7
25	35.2	14.1	−7.0	−21.4	1	95.4	57.4	19.4	−6.6
32	39.7	18.6	−2.6	−17.0	1.25	103.4	65.4	27.4	1.4
38	43.4	22.3	1.2	−13.2	1.5	110.2	72.2	34.2	8.2

GENERAL NOTE: The Charpy V-notch values given in this table represent a smooth curve in Figures 3.7 and 3.7M.

Table 3.15
Impact Test Exemption Curves — Parts Subject to PWHT and Nonwelded Parts (See Figures 3.8 and 3.8M)

Thickness, mm	Exemption Curve, °C A	B	C	D	Thickness, in.	Exemption Curve, °F A	B	C	D
0	0.6	−20.5	−41.6	−48.3	0	33.2	−4.8	−42.8	−55.0
10	0.6	−20.5	−41.6	−48.3	0.394	33.2	−4.8	−42.8	−55.0
13	3.8	−17.3	−38.4	−48.3	0.5	38.9	0.9	−37.1	−55.0
16	7.9	−13.2	−34.3	−48.3	0.625	46.2	8.2	−29.8	−55.0
19	11.7	−9.4	−30.5	−45.0	0.75	53.0	15.0	−23.0	−49.0
25	17.5	−3.6	−24.7	−39.2	1	63.5	25.5	−12.5	−38.5
32	21.7	0.5	−20.6	−35.0	1.25	71.0	33.0	−5.0	−31.0
38	24.9	3.8	−17.3	−31.8	1.5	76.8	38.8	0.8	−25.2
44	27.7	6.6	−14.6	−29.0	1.75	81.8	43.8	5.8	−20.2
51	30.1	9.0	−12.1	−26.5	2	86.2	48.2	10.2	−15.8
57	32.4	11.3	−9.9	−24.3	2.25	90.3	52.3	14.3	−11.7
64	34.4	13.3	−7.8	−22.3	2.5	93.9	55.9	17.9	−8.1
70	36.2	15.1	−6.0	−20.5	2.75	97.2	59.2	21.2	−4.8
76	37.8	16.7	−4.4	−18.9	3	100.0	62.0	24.0	−2.0
83	39.2	18.1	−3.0	−17.5	3.25	102.6	64.6	26.6	0.6
89	40.4	19.3	−1.8	−16.3	3.5	104.7	66.7	28.7	2.7
95	41.4	20.3	−0.8	−15.3	3.75	106.5	68.5	30.5	4.5
102	42.2	21.1	−0.1	−14.5	4	107.9	69.9	31.9	5.9

GENERAL NOTE: The Charpy V-notch values given in this table represent a smooth curve in Figures 3.8 and 3.8M.

Table 3.16
Reduction in the MDMT, T_R, Without Impact Testing — Parts Not Subject to PWHT (See Figures 3.12 and 3.12M)

Stress or Thickness Ratio	T_R, °C Specified Minimum Yield Strength, MPa ≤345 MPa	T_R, °C Specified Minimum Yield Strength, MPa >345 MPa ≤450 MPa	T_R, °F Specified Minimum Yield Strength, ksi ≤50 ksi	T_R, °F Specified Minimum Yield Strength, ksi >50 ksi ≤65 ksi
1.000	0.0	0.0	0.0	0.0
0.940	2.7	2.5	4.9	4.5
0.884	5.2	4.7	9.3	8.4
0.831	7.3	6.6	13.2	11.9
0.781	9.3	8.4	16.7	15.1
0.734	11.1	10.0	20.0	18.1
0.690	12.8	11.5	23.0	20.8
0.648	14.3	13.0	25.8	23.3
0.610	15.8	14.3	28.5	25.7
0.573	17.2	15.5	31.0	27.9
0.539	18.5	16.7	33.3	30.0
0.506	19.7	17.7	35.5	31.9
0.476	20.9	18.8	37.6	33.8
0.447	22.0	19.7	39.6	35.5
0.421	23.1	20.6	41.5	37.1
0.395	24.0	21.5	43.3	38.7
0.372	25.0	22.3	45.0	40.1
0.349	25.9	23.1	46.6	41.5
0.328	26.7	23.8	48.1	42.8
0.309	27.5	24.5	49.6	44.0
0.2908	28.3	25.1	50.9	45.2
0.273	29.0	25.7	52.2	46.3
0.256	29.7	26.3	53.5	47.3
0.241	30.4	26.8	54.6	48.3

GENERAL NOTE: The temperature reduction values given in this table represent a smooth curve in Figures 3.12 and 3.12M.

Table 3.17
Reduction in the MDMT, T_R, Without Impact Testing — Parts Subject to PWHT and Nonwelded Parts (See Figures 3.13 and 3.13M)

Stress or Thickness Ratio	T_R, °C		T_R, °F	
	Specified Minimum Yield Strength, MPa		Specified Minimum Yield Strength, ksi	
	≤345 MPa	>345 MPa ≤450 MPa	≤50 ksi	>50 ksi ≤65 ksi
1.000	0.0	0.0	0.0	0.0
0.940	3.0	2.6	5.4	4.6
0.884	5.9	5.0	10.6	8.9
0.831	8.7	7.3	15.6	13.1
0.781	11.5	9.5	20.7	17.2
0.734	14.3	11.7	25.8	21.1
0.690	17.3	13.9	31.1	25.0
0.648	20.3	16.1	36.5	29.0
0.610	23.5	18.3	42.2	32.9
0.573	26.9	20.5	48.4	36.8
0.539	30.6	22.7	55.0	40.9
0.506	34.7	25.0	62.5	45.0
0.476	39.5	27.3	71.1	49.2
0.447	45.3	29.8	81.6	53.6
0.421	52.9	32.3	95.2	58.1
0.395	...	35.0	...	62.9
0.372	...	37.8	...	68.1
0.349	...	40.9	...	73.6
0.328	...	44.3	...	79.7
0.309	...	48.0	...	86.4
0.290	...	52.3	...	94.2
0.273	...	...	...	...
0.256	...	...	...	...
0.241	...	...	...	...

GENERAL NOTE: The temperature reduction values given in this table represent a smooth curve in Figures 3.13 and 3.13M.

Table 3.18
Required HAZ Impact Test Specimen Set Removal

Base Metal Thickness, t	Number of Specimen Sets and the Locations of Their Approximate Centerline	
	Single-Sided Weld	Two-Sided Weld
$t \le 19$ mm ($^3/_4$ in.)	One set, $^1/_4t$ to $^1/_2t$	One set, middle $^1/_2t$ [Note (1)]
$t > 19$ mm ($^3/_4$ in.)	One set, $^1/_4t$ to $^1/_2t$	Two sets, $^1/_4t$ to $^1/_2t$ [Note (2)]

GENERAL NOTE: Testing shall be performed on sets of three impact test specimens as required by 3.11.7.2(a). Each specimen shall be full size or the largest subsize specimen that may be removed from the available material thickness. The specimen sets shall be removed at the indicated depth from the weld surface, as described in Notes (1) and (2).

NOTES:
(1) For two-sided welds in base metal thicknesses of 19 mm ($^3/_4$ in.) or less, a single specimen set prepared with the centerline of the width along the notch falling within the middle $^1/_2t$ shall represent the HAZ of the welds applied to both surfaces.
(2) For two-sided welds in base metal thicknesses greater than 19 mm ($^3/_4$ in.), specimen sets shall be prepared with the centerline of the width along the notch falling between $^1/_4t$ and $^1/_2t$ from each weld surface.

3.20 FIGURES

Figure 3.1
Cr–Mo Heat Treatment Criteria

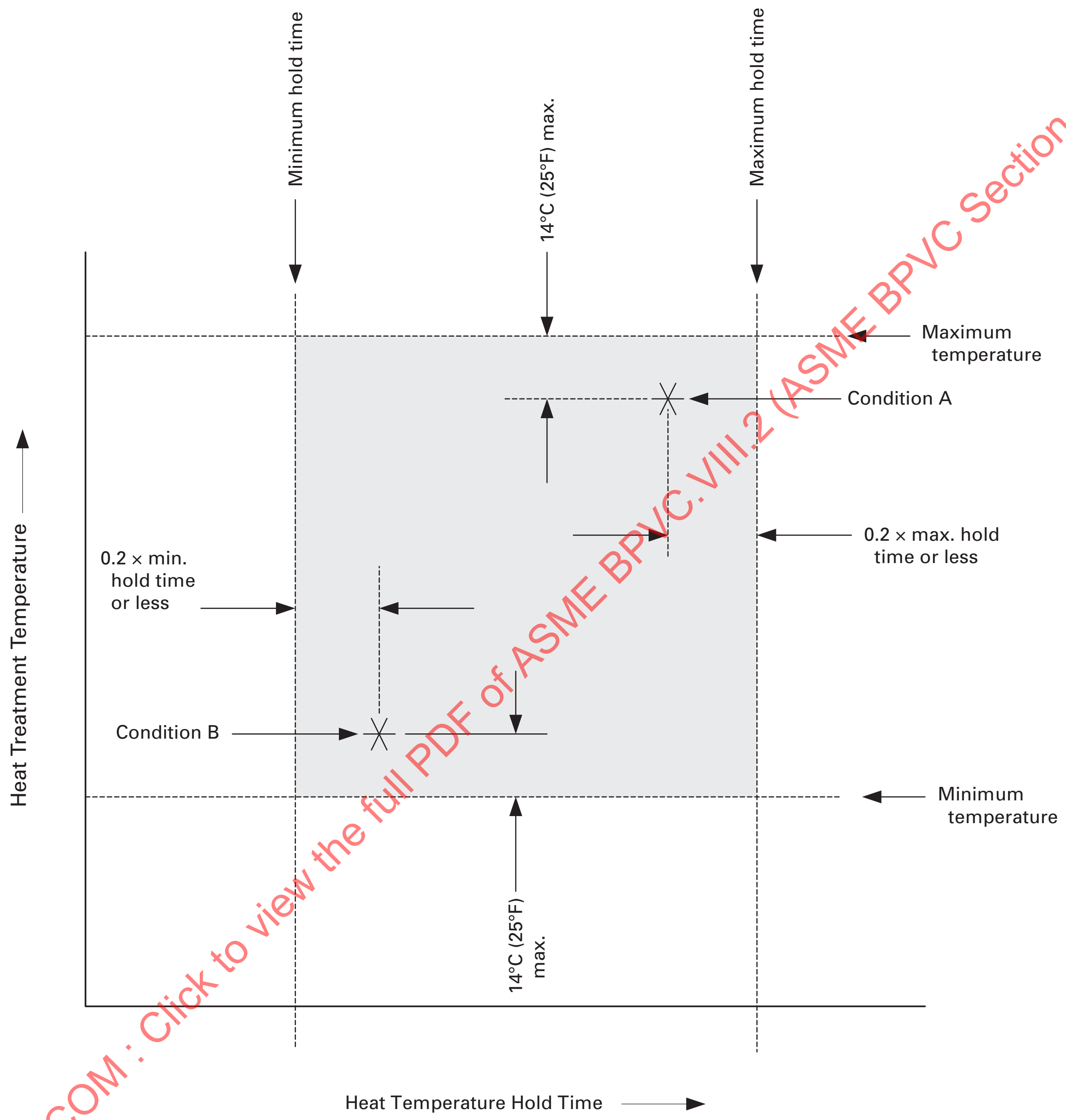
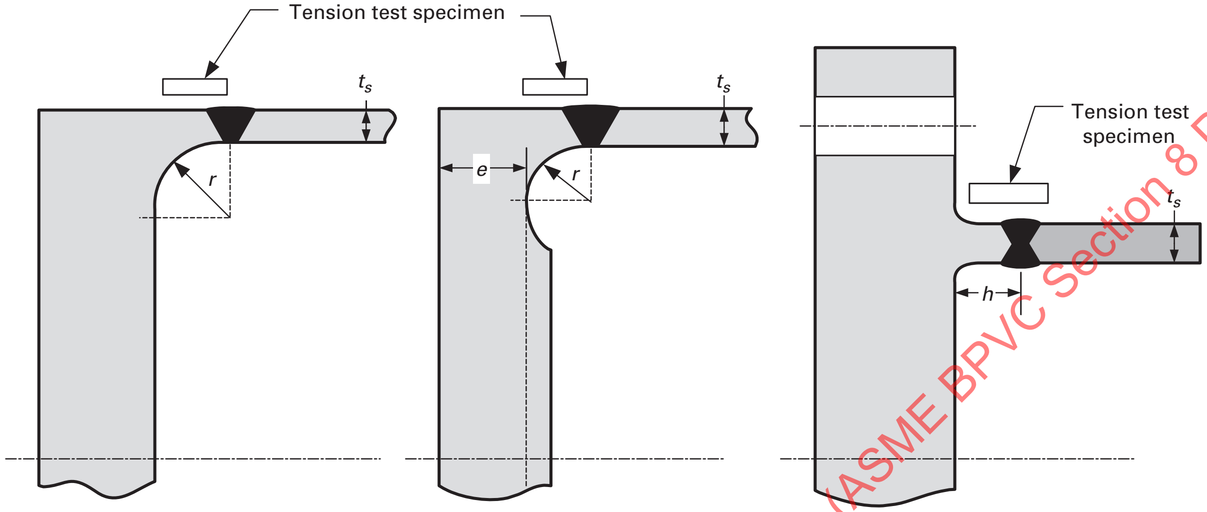
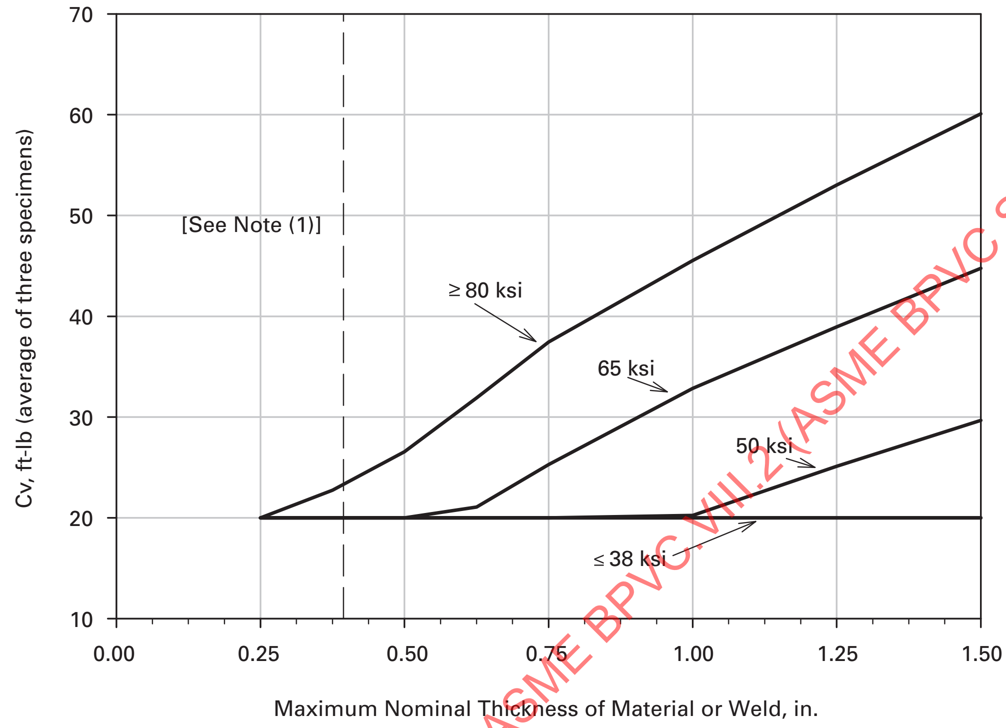


Figure 3.2
Typical Locations for Tensile Specimens



GENERAL NOTE: These details are not permissible if machined from plate unless the requirements of 3.9 are satisfied.

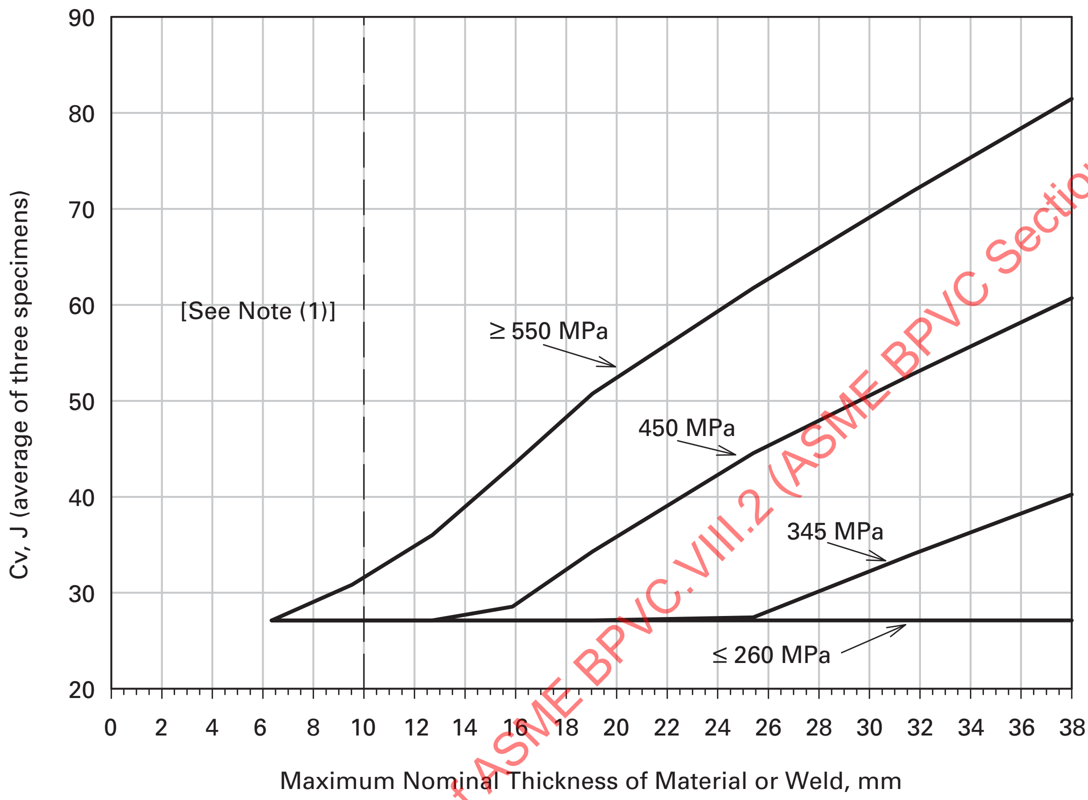
Figure 3.3
Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels as a Function of the Minimum Specified Yield Strength — Parts Not Subject to PWHT



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The minimum impact energy for one specimen shall not be less than two-thirds of the average impact energy required for three specimens.
(c) Materials produced and impact tested in accordance with SA-320, SA-333, SA-334, SA-350, SA-352, SA-420, SA-437, SA-508 Grade 5 Class 2, SA-540 (except for materials produced under Table 2, Note 4 in the specification), SA-723, and SA-765 do not have to satisfy these energy values. Materials produced to these specifications are acceptable for use at a minimum design metal temperature not colder than the test temperature when the energy values required by the applicable specification are satisfied.
(d) If the material specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the material toughness requirements shall be in accordance with 3.11.2.1(b)(2).
(e) Data of Figures 3.3 and 3.3M are shown in Table 3.12.

NOTE:
(1) See 3.11.2.1(b)(1) for Charpy V-notch specimen thicknesses less than 10 mm (0.394 in.).

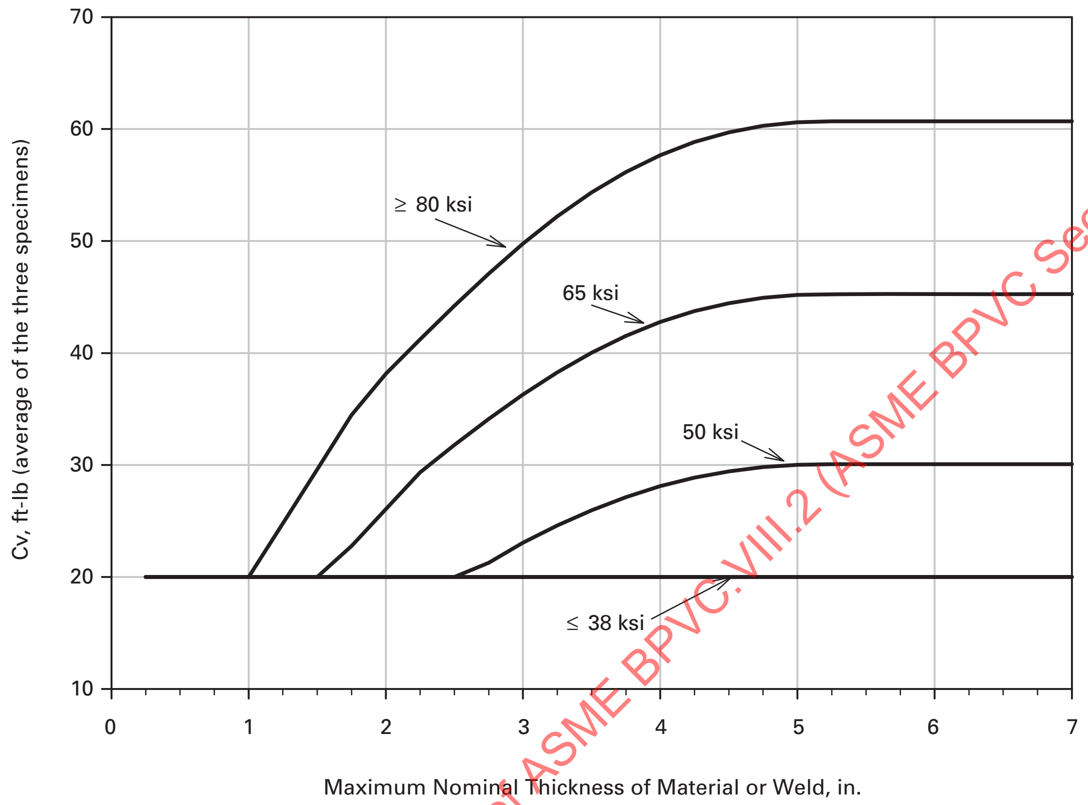
Figure 3.3M
Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels as a Function of the Minimum Specified Yield Strength — Parts Not Subject to PWHT



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The minimum impact energy for one specimen shall not be less than two-thirds of the average impact energy required for three specimens.
(c) Materials produced and impact tested in accordance with SA-320, SA-333, SA-334, SA-350, SA-352, SA-420, SA-437, SA-508 Grade 5 Class 2, SA-540 (except for materials produced under Table 2, Note 4 in the specification), SA-723, and SA-765 do not have to satisfy these energy values. Materials produced to these specifications are acceptable for use at a minimum design metal temperature not colder than the test temperature when the energy values required by the applicable specification are satisfied.
(d) If the material specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the material toughness requirements shall be in accordance with 3.11.2.1(b)(2).
(e) Data of Figures 3.3 and 3.3M are shown in Table 3.12.

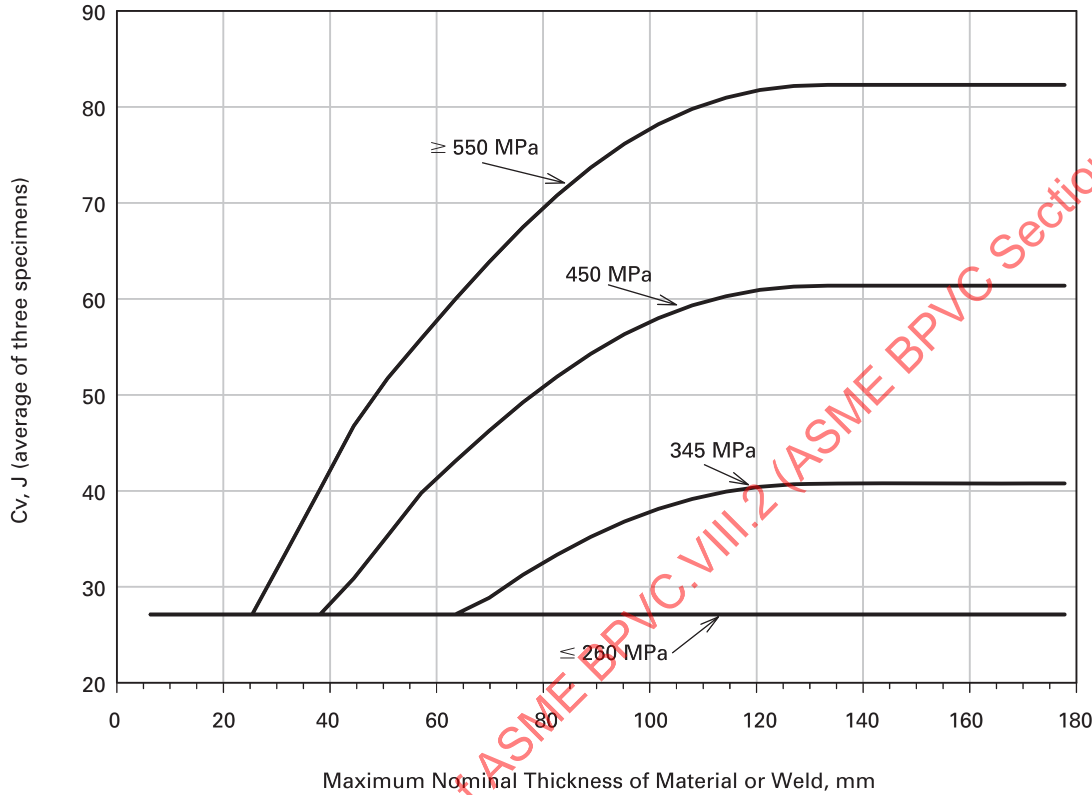
NOTE:
(1) See 3.11.2.1(b)(1) for Charpy V-notch specimen thicknesses less than 10 mm (0.394 in.).

Figure 3.4
Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels as a Function of the Minimum Specified Yield Strength — Parts Subject to PWHT or Nonwelded Parts



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The minimum impact energy for one specimen shall not be less than two-thirds of the average impact energy required for three specimens.
(c) Materials produced and impact tested in accordance with SA-320, SA-333, SA-334, SA-350, SA-352, SA-420, SA-437, SA-508 Grade 5 Class 2, SA-540 (except for materials produced under Table 2, Note 4 in the specification), SA-723, and SA-765 do not have to satisfy these energy values. Materials produced to these specifications are acceptable for use at a minimum design metal temperature not colder than the test temperature when the energy values required by the applicable specification are satisfied.
(d) If the material specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the material toughness requirements shall be in accordance with 3.11.2.1(b)(2).
(e) Data of Figures 3.4 and 3.4M are shown in Table 3.13.
(f) See 3.11.2.1(b)(1) for Charpy V-notch specimen thicknesses less than 10 mm (0.394 in.).

Figure 3.4M
Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels as a Function of the Minimum Specified Yield Strength — Parts Subject to PWHT or Nonwelded Parts



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The minimum impact energy for one specimen shall not be less than two-thirds of the average impact energy required for three specimens.
(c) Materials produced and impact tested in accordance with SA-320, SA-333, SA-334, SA-350, SA-352, SA-420, SA-437, SA-508 Grade 5 Class 2, SA-540 (except for materials produced under Table 2, Note 4 in the specification), SA-723, and SA-765 do not have to satisfy these energy values. Materials produced to these specifications are acceptable for use at a minimum design metal temperature not colder than the test temperature when the energy values required by the applicable specification are satisfied.
(d) If the material specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the material toughness requirements shall be in accordance with 3.11.2.1(b)(2).
(e) Data of Figures 3.4 and 3.4M are shown in Table 3.13.
(f) See 3.11.2.1(b)(1) for Charpy V-notch specimen thicknesses less than 10 mm (0.394 in.).

Figure 3.5
Illustration of Lateral Expansion in a Broken Charpy V-Notch Specimen

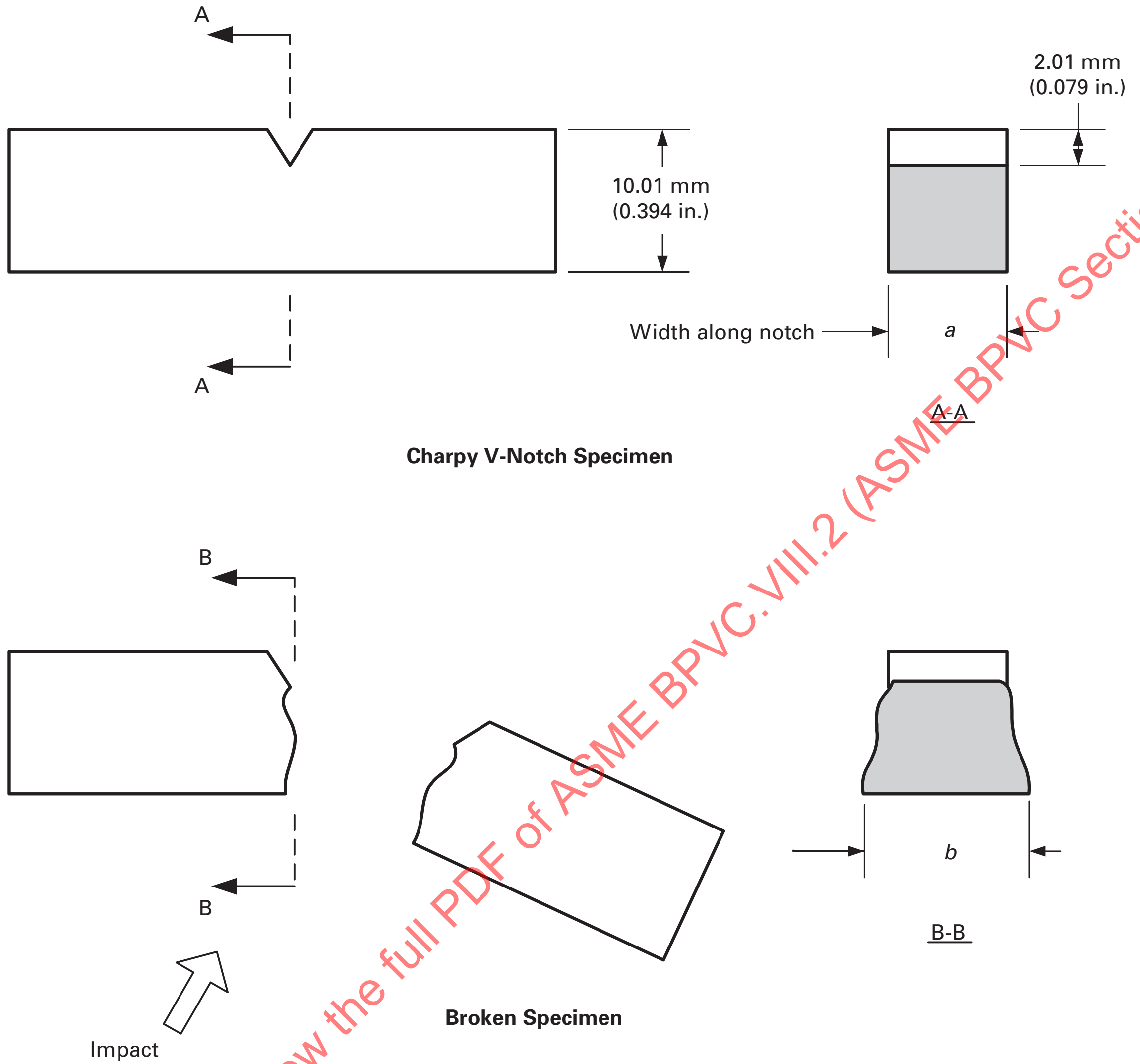


Figure 3.6
Lateral Expansion Requirements

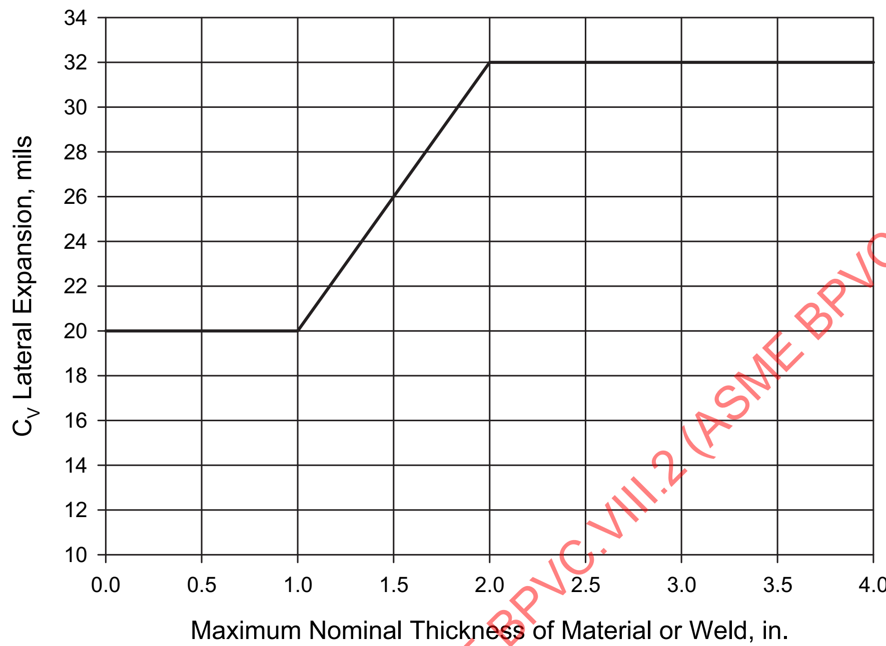


Figure 3.6M
Lateral Expansion Requirements

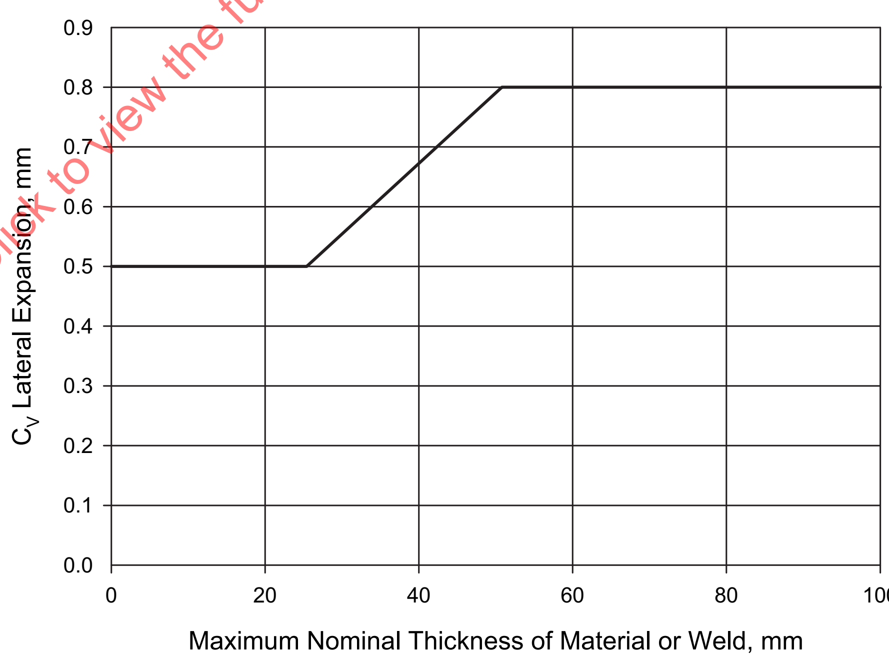
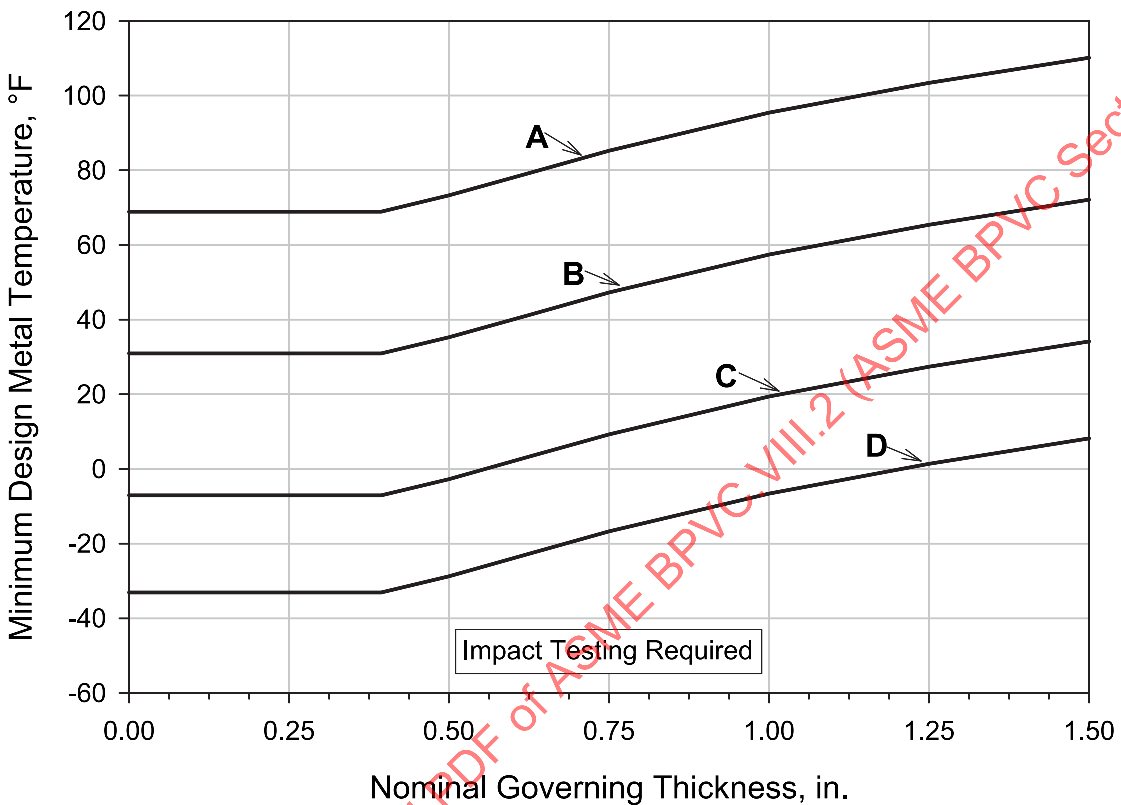


Figure 3.7
Impact Test Exemption Curves — Parts Not Subject to PWHT



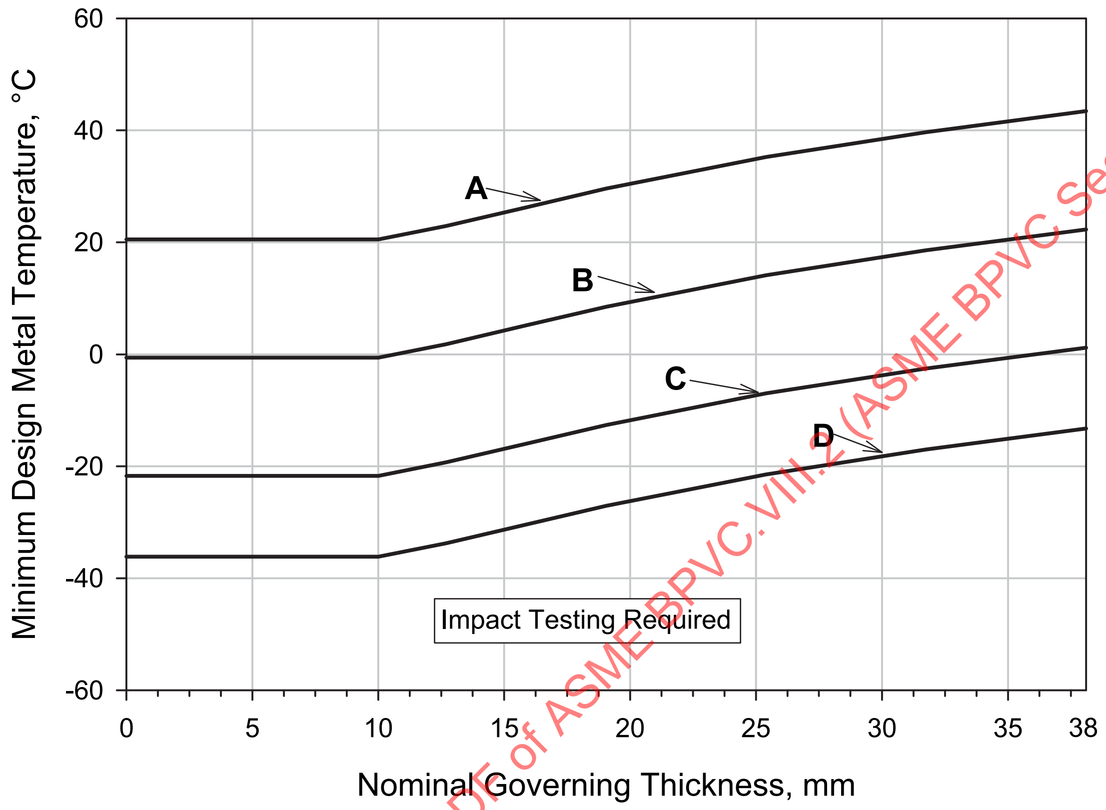
Curve	Material Assignment
A	*(a)* All carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below *(b)* SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA -217 Grade WC6 if normalized and tempered or water-quenched and tempered *(c)* A/SA-105 forged flanges supplied in the as-forged condition
B	*(a)* SA-216 Grades WCA if normalized and tempered or water-quenched and tempered; Grades WCB and WCC for thicknesses not exceeding 50 mm (2 in.) if produced to a fine grain practice and water-quenched and tempered *(b)* SA-217 Grade WC9 if normalized and tempered *(c)* SA-285 Grades A and B *(d)* SA-299 *(e)* SA-414 Grade A *(f)* SA-515 Grades 60 *(g)* SA-516 Grades 65 and 70 if not normalized *(h)* SA-662 Grade B if not normalized *(i)* SA/EN 10028-2 Grade P355GH as-rolled *(j)* Except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curve C and D below *(k)* Pipe; fittings; forgings; A/SA-105 forged flanges that are produced to fine grain practice and normalized, normalized and tempered, or quenched and tempered after forging; and tubing not listed for Curves C and D below *(l)* Parts permitted from 3.2.8 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.
C	*(a)* SA-182 Grades F21 and F22 if normalized and tempered *(b)* SA-302 Grades C and D

Figure 3.7
Impact Test Exemption Curves — Parts Not Subject to PWHT (Cont'd)

Curve	Material Assignment
	(c) SA-336 Grades F21 and F22 if normalized and tempered, or liquid quenched and tempered
	(d) SA-387 Grades 21 and 22 if normalized and tempered, or liquid quenched and tempered
	(e) SA-516 Grades 55 and 60 if not normalized
	(f) SA-533 Types B and C, Class 1
	(g) SA-662 Grade A
	(h) SA/EN 10028-2 Grade 10CrMo9–10 if normalized and tempered
	(i) All materials listed in (a) through (i) and in (k) for Curve B if produced to fine grain practice and normalized, normalized and tempered, or liquid quenched and tempered as permitted in the material specification, and not listed for Curve D below
D	*(a)* SA-203
	(b) SA-299 if normalized
	(c) SA-508 Class 1
	(d) SA-516 if normalized
	(e) SA-524 Classes 1 and 2
	(f) SA-537 Classes 1, 2, and 3
	(g) SA-612 if normalized; except that the increased Cb limit in the footnote of Table 1 of SA-20 is not permitted
	(h) SA-662 if normalized
	(i) SA-738 Grade A
	(j) SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than −29°C (−20°F)
	(k) SA-738 Grade B not colder than −29°C (−20°F)
	(l) SA/EN 10028-2 Grade P355GH if normalized [See General Note (d)(3)]

GENERAL NOTES:
(a) Castings not listed as Curve A and B shall be impact tested.
(b) For bolting see 3.11.6.
(c) When a class or grade is not shown in a material assignment, all classes and grades are indicated.
(d) The following apply to all material assignments:
(1) Cooling rates faster than those obtained in air, followed by tempering, as permitted by the material specification, are considered equivalent to normalizing and tempering heat treatments.
(2) Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.
(3) Normalized rolling condition is not considered as being equivalent to normalizing.
(e) Data of Figures 3.7 and 3.7M are shown in Table 3.14.

Figure 3.7M
Impact Test Exemption Curves — Parts Not Subject to PWHT



Curve	Material Assignment
A	*(a)* All carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below *(b)* SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA -217 Grade WC6 if normalized and tempered or water-quenched and tempered *(c)* A/SA-105 forged flanges supplied in the as-forged condition
B	*(a)* SA-216 Grades WCA if normalized and tempered or water-quenched and tempered; Grades WCB and WCC for thicknesses not exceeding 50 mm (2 in.) if produced to a fine grain practice and water-quenched and tempered *(b)* SA-217 Grade WC9 if normalized and tempered *(c)* SA-285 Grades A and B *(d)* SA-299 *(e)* SA-414 Grade A *(f)* SA-515 Grades 60 *(g)* SA-516 Grades 65 and 70 if not normalized *(h)* SA-662 Grade B if not normalized *(i)* SA/EN 10028-2 Grade P355GH as-rolled *(j)* Except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curve C and D below *(k)* Pipe; fittings; forgings; A/SA-105 forged flanges that are produced to fine grain practice and normalized, normalized and tempered, or quenched and tempered after forging; and tubing not listed for Curves C and D below *(l)* Parts permitted from 3.2.8 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.
C	*(a)* SA-182 Grades F21 and F22 if normalized and tempered *(b)* SA-302 Grades C and D *(c)* SA-336 Grades F21 and F22 if normalized and tempered, or liquid quenched and tempered

Figure 3.7M
Impact Test Exemption Curves — Parts Not Subject to PWHT (Cont'd)

Curve	Material Assignment
	(d) SA-387 Grades 21 and 22 if normalized and tempered, or liquid quenched and tempered
	(e) SA-516 Grades 55 and 60 if not normalized
	(f) SA-533 Types B and C, Class 1
	(g) SA-662 Grade A
	(h) SA/EN 10028-2 Grade 10CrMo9–10 if normalized and tempered
	(i) All materials listed in (a) through (i) and in (k) for Curve B if produced to fine grain practice and normalized, normalized and tempered, or liquid quenched and tempered as permitted in the material specification, and not listed for Curve D below
D	*(a)* SA-203
	(b) SA-299
	(c) SA-508 Class 1
	(d) SA-516 if normalized
	(e) SA-524 Classes 1 and 2
	(f) SA-537 Classes 1, 2, and 3
	(g) SA-612 if normalized; except that the increased Cb limit in the footnote of Table 1 of SA-20 is not permitted
	(h) SA-662 if normalized
	(i) SA-738 Grade A
	(j) SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than −29°C (−20°F)
	(k) SA-738 Grade B not colder than −29°C (−20°F)
	(l) SA/EN 10028-2 Grade P355GH if normalized [See General Note (d)(3)]

GENERAL NOTES:

(a) Castings not listed as Curve A and B shall be impact tested.

(b) For bolting see 3.11.6.

(c) When a class or grade is not shown in a material assignment, all classes and grades are indicated.

(d) The following apply to all material assignments:

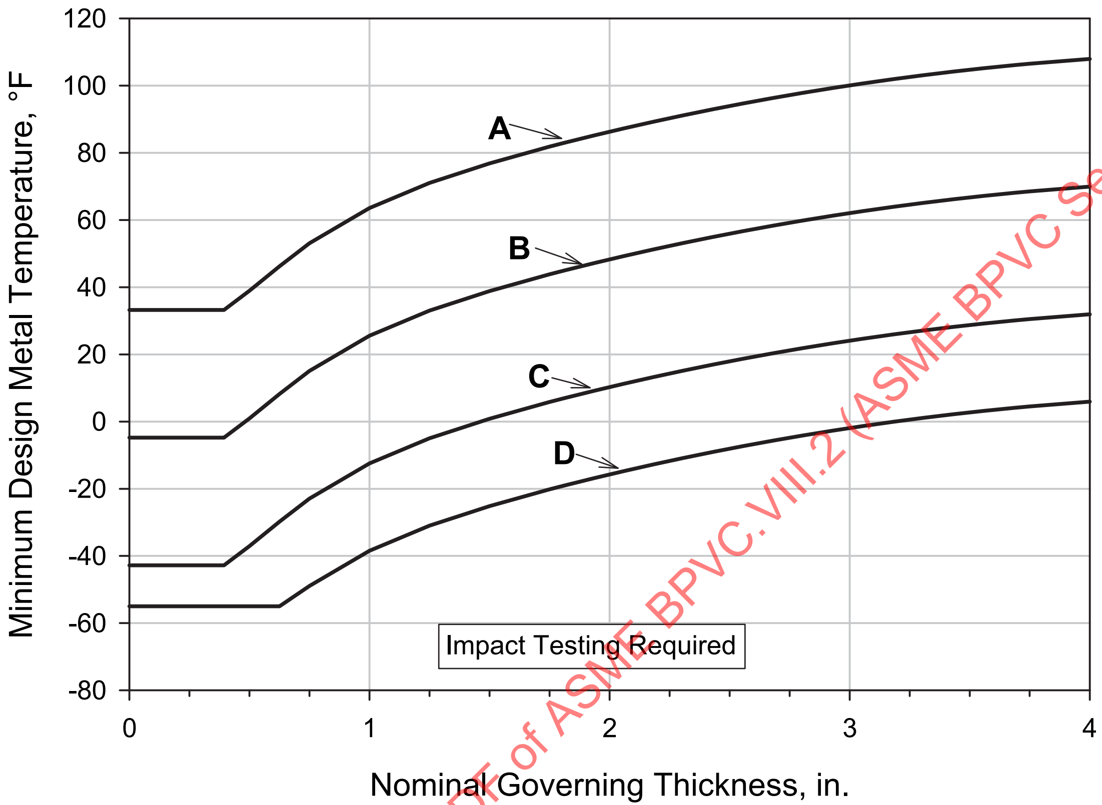
(1) Cooling rates faster than those obtained in air, followed by tempering, as permitted by the material specification, are considered equivalent to normalizing and tempering heat treatments.

(2) Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.

(3) Normalized rolling condition is not considered as being equivalent to normalizing.

(e) Data of Figures 3.7 and 3.7M are shown in Table 3.14.

Figure 3.8
Impact Test Exemption Curves — Parts Subject to PWHT and Nonwelded Parts



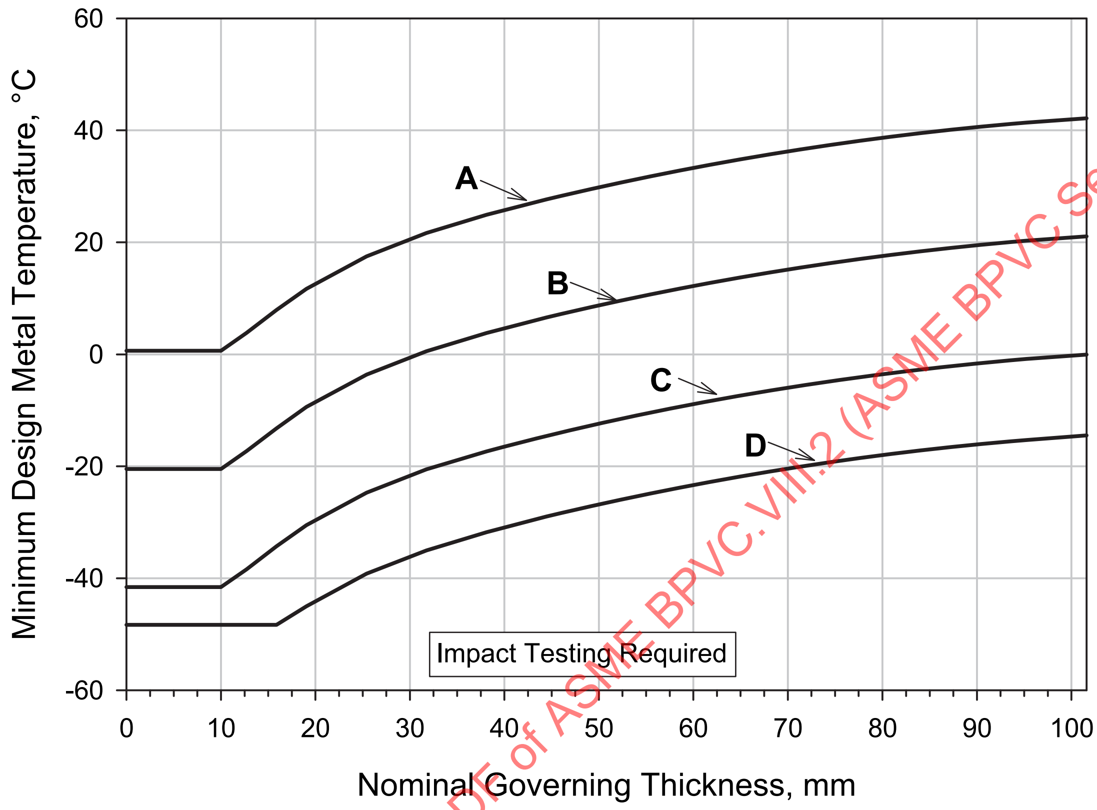
Curve	Material Assignment
A	*(a)* All carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below *(b)* SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA -217 Grade WC6 if normalized and tempered or water-quenched and tempered *(c)* A/SA-105 forged flanges supplied in the as-forged condition
B	*(a)* SA-216 Grades WCA if normalized and tempered or water-quenched and tempered; Grades WCB and WCC for thicknesses not exceeding 50 mm (2 in.) if produced to a fine grain practice and water-quenched and tempered *(b)* SA-217 Grade WC9 if normalized and tempered *(c)* SA-285 Grades A and B *(d)* SA-299 *(e)* SA-414 Grade A *(f)* SA-515 Grades 60 *(g)* SA-516 Grades 65 and 70 if not normalized *(h)* SA-662 Grade B if not normalized *(i)* SA/EN 10028-2 Grade P355GH as-rolled *(j)* Except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curve C and D below *(k)* Pipe; fittings; forgings; A/SA-105 forged flanges that are produced to fine grain practice and normalized, normalized and tempered, or quenched and tempered after forging; and tubing not listed for Curves C and D below *(l)* Parts permitted from 3.2.8 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.
C	*(a)* SA-182 Grades F21 and F22 if normalized and tempered *(b)* SA-302 Grades C and D *(c)* SA-336 Grades F21 and F22 if normalized and tempered, or liquid quenched and tempered

Figure 3.8
Impact Test Exemption Curves — Parts Subject to PWHT and Nonwelded Parts (Cont'd)

Curve	Material Assignment
	(d) SA-387 Grades 21 and 22 if normalized and tempered, or liquid quenched and tempered
	(e) SA-516 Grades 55 and 60 if not normalized
	(f) SA-533 Types B and C, Class 1
	(g) SA-662 Grade A
	(h) SA/EN 10028-2 Grade 10CrMo9–10 if normalized and tempered
	(i) All materials listed in (a) through (i) and in (k) for Curve B if produced to fine grain practice and normalized, normalized and tempered, or liquid quenched and tempered as permitted in the material specification, and not listed for Curve D below
D	*(a)* SA-203
	(b) SA-299 if normalized
	(c) SA-508 Class 1
	(d) SA-516 if normalized
	(e) SA-524 Classes 1 and 2
	(f) SA-537 Classes 1, 2, and 3
	(g) SA-612 if normalized; except that the increased Cb limit in the footnote of Table 1 of SA-20 is not permitted
	(h) SA-662 if normalized
	(i) SA-738 Grade A
	(j) SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than −29°C (−20°F)
	(k) SA-738 Grade B not colder than −29°C (−20°F)
	(l) SA/EN 10028-2 Grade P355GH if normalized [See General Note (d)(3)]

GENERAL NOTES:
(a) Castings not listed as Curve A and B shall be impact tested.
(b) For bolting see 3.11.6.
(c) When a class or grade is not shown in a material assignment, all classes and grades are indicated.
(d) The following apply to all material assignments:
(1) Cooling rates faster than those obtained in air, followed by tempering, as permitted by the material specification, are considered equivalent to normalizing and tempering heat treatments.
(2) Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.
(3) Normalized rolling condition is not considered as being equivalent to normalizing.
(e) Data of Figures 3.8 and 3.8M are shown in Table 3.15.

Figure 3.8M
Impact Test Exemption Curves — Parts Subject to PWHT and Nonwelded Parts



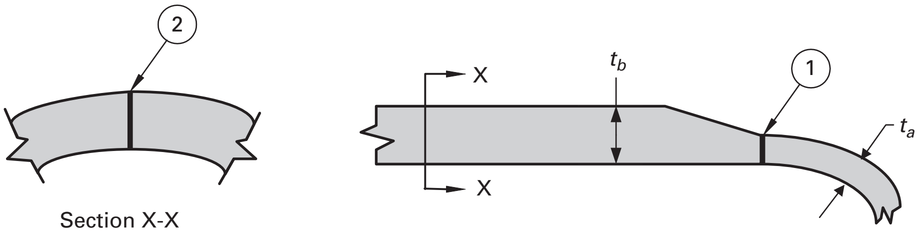
Curve	Material Assignment
A	*(a)* All carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below *(b)* SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA -217 Grade WC6 if normalized and tempered or water-quenched and tempered *(c)* A/SA-105 forged flanges supplied in the as-forged condition
B	*(a)* SA-216 Grades WCA if normalized and tempered or water-quenched and tempered; Grades WCB and WCC for thicknesses not exceeding 50 mm (2 in.) if produced to a fine grain practice and water-quenched and tempered *(b)* SA-217 Grade WC9 if normalized and tempered *(c)* SA-285 Grades A and B *(d)* SA-299 *(e)* SA-414 Grade A *(f)* SA-515 Grades 60 *(g)* SA-516 Grades 65 and 70 if not normalized *(h)* SA-662 Grade B if not normalized *(i)* SA/EN 10028-2 Grade P355GH as-rolled *(j)* Except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curve C and D below *(k)* Pipe; fittings; forgings; A/SA-105 forged flanges that are produced to fine grain practice and normalized, normalized and tempered, or quenched and tempered after forging; and tubing not listed for Curves C and D below *(l)* Parts permitted from 3.2.8 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.
C	*(a)* SA-182 Grades F21 and F22 if normalized and tempered *(b)* SA-302 Grades C and D *(c)* SA-336 Grades F21 and F22 if normalized and tempered, or liquid quenched and tempered

Figure 3.8M
Impact Test Exemption Curves — Parts Subject to PWHT and Nonwelded Parts (Cont'd)

Curve	Material Assignment
	(d) SA-387 Grades 21 and 22 if normalized and tempered, or liquid quenched and tempered
	(e) SA-516 Grades 55 and 60 if not normalized
	(f) SA-533 Types B and C, Class 1
	(g) SA-662 Grade A
	(h) SA/EN 10028-2 Grade 10CrMo9–10 if normalized and tempered
	(i) All materials listed in (a) through (i) and in (k) for Curve B if produced to fine grain practice and normalized, normalized and tempered, or liquid quenched and tempered as permitted in the material specification, and not listed for Curve D below
D	*(a)* SA-203
	(b) SA-299 if normalized
	(c) SA-508 Class 1
	(d) SA-516 if normalized
	(e) SA-524 Classes 1 and 2
	(f) SA-537 Classes 1, 2, and 3
	(g) SA-612 if normalized; except that the increased Cb limit in the footnote of Table 1 of SA-20 is not permitted
	(h) SA-662 if normalized
	(i) SA-738 Grade A
	(j) SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than −29°C (−20°F)
	(k) SA-738 Grade B not colder than −29°C (−20°F)
	(l) SA/EN 10028-2 Grade P355GH if normalized [See General Note (d)(3)]

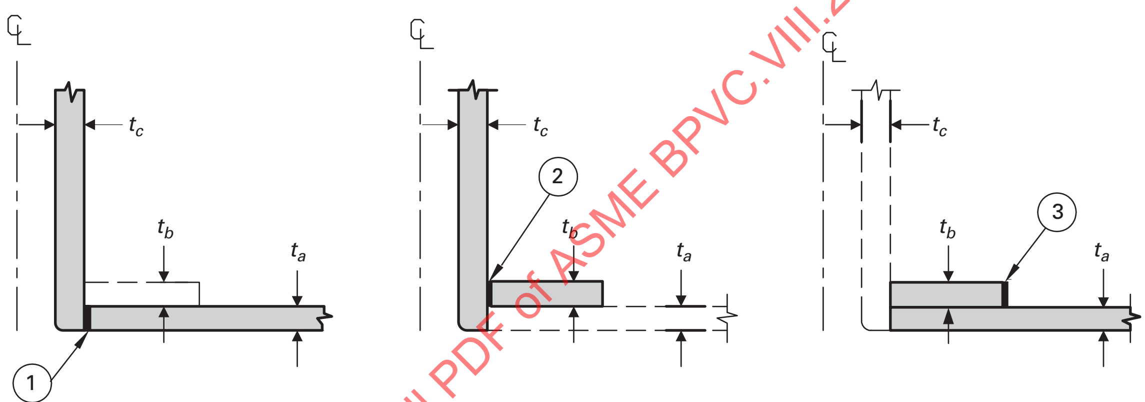
GENERAL NOTES:
(a) Castings not listed as Curve A and B shall be impact tested.
(b) For bolting see 3.11.6.
(c) When a class or grade is not shown in a material assignment, all classes and grades are indicated.
(d) The following apply to all material assignments:
(1) Cooling rates faster than those obtained in air, followed by tempering, as permitted by the material specification, are considered equivalent to normalizing and tempering heat treatments.
(2) Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.
(3) Normalized rolling condition is not considered as being equivalent to normalizing.
(e) Data of Figures 3.8 and 3.8M are shown in Table 3.15.

Figure 3.9
Typical Vessel Details Illustrating the Governing Thickness



$t_{g1} = t_a$
$t_{g2} = t_a$ (seamless) or t_b (welded)

(a) Butt-Welded Components



t_{g1} = min. (t_a, t_c)

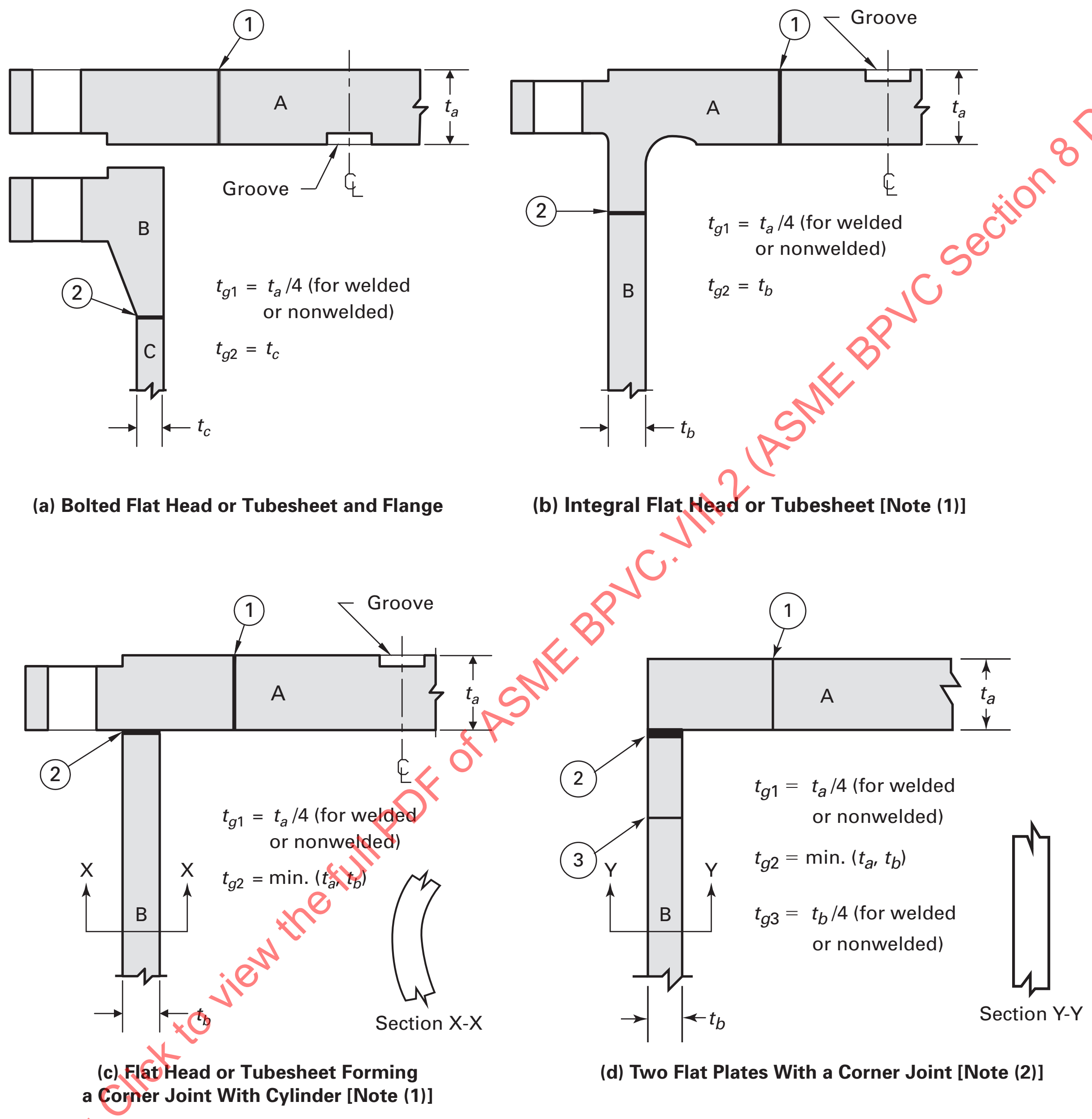
t_{g2} = min. (t_b, t_c)

t_{g3} = min. (t_a, t_b)

(b) Welded Connection With or Without a Reinforcing Plate

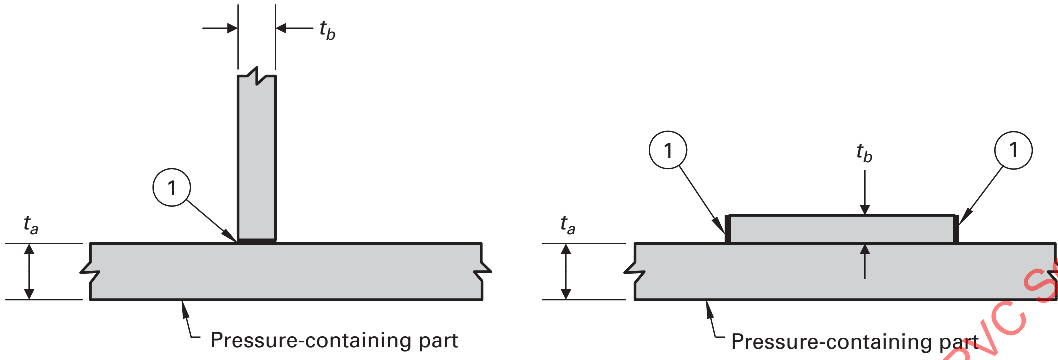
(21)

Figure 3.10
Typical Vessel Details Illustrating the Governing Thickness



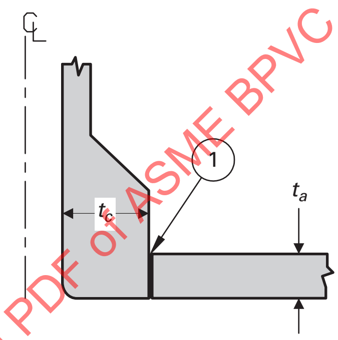
NOTES:
(1) The governing thickness of the integral flat head or tubesheet is $\max(t_{g1}, t_{g2})$. The governing thickness of the shell is t_{g2}.
(2) The governing thickness of component A is $\max(t_{g1}, t_{g2})$. The governing thickness of component B is $\max(t_{g2}, t_{g3})$.

Figure 3.11
Typical Vessel Details Illustrating the Governing Thickness



t_{g1} = min. (t_a, t_b)

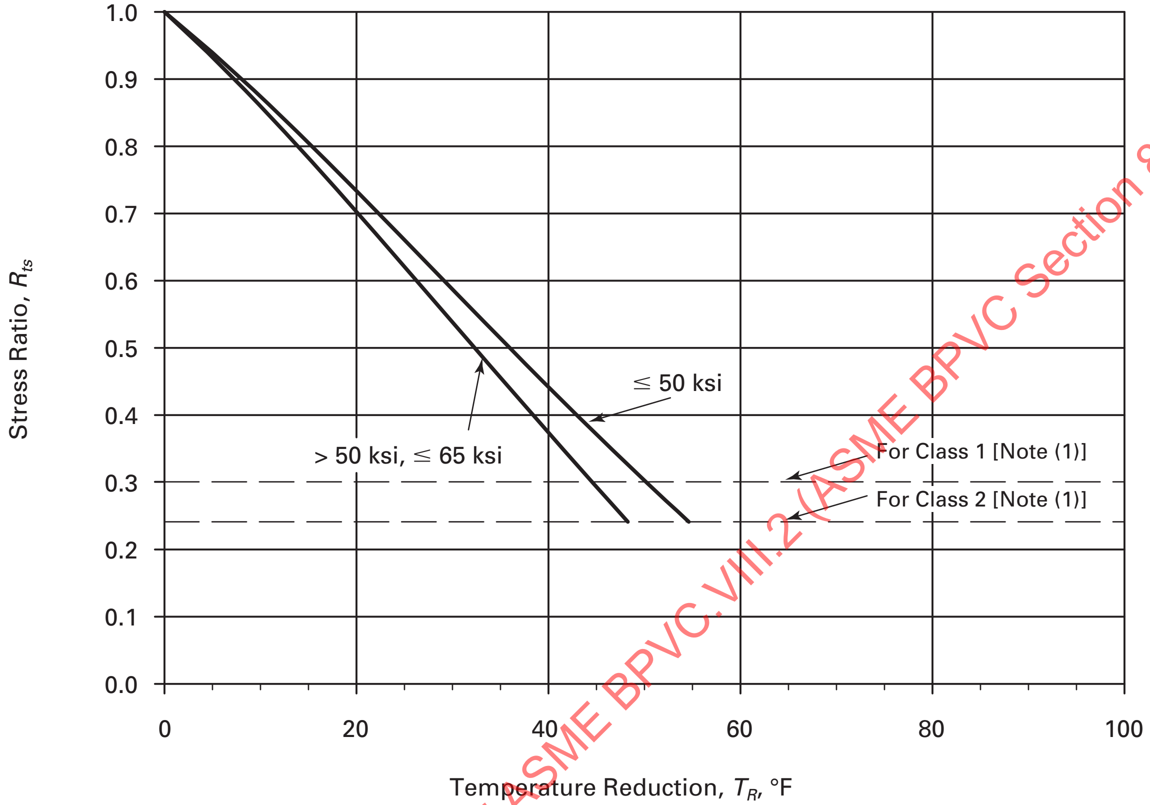
(a) Welded Attachments



t_{g1} = min. (t_a, t_c)

(b) Integrally Reinforced Welded Connection

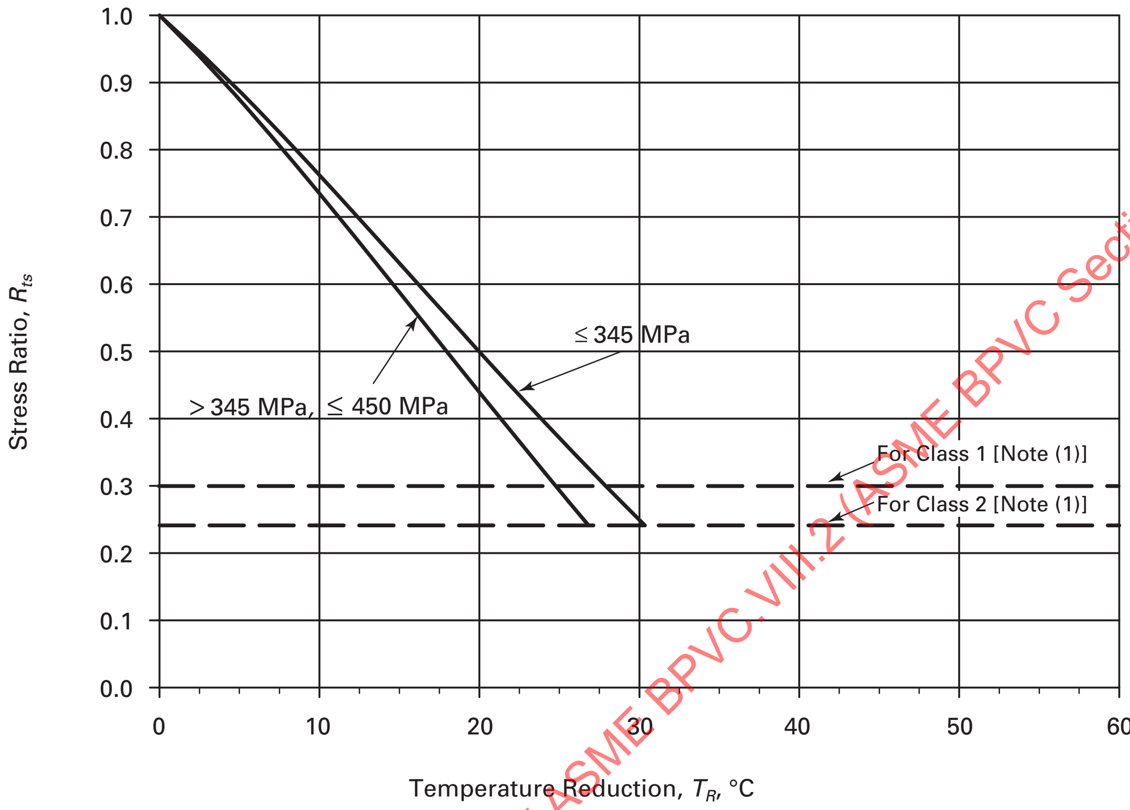
Figure 3.12
Reduction in the MDMT Without Impact Testing — Parts Not Subject to PWHT



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The reduction in MDMT shall not exceed 55°C (100°F), except as permitted by 3.11.2.5(a), Step 5(b).
(c) Data of Figures 3.12 and 3.12M are shown in Table 3.16.

NOTE:
(1) See 3.11.2.5(a), Step 5(a) when R_{ts} is less than or equal to 0.3 for Class 1, or 0.24 for Class 2.

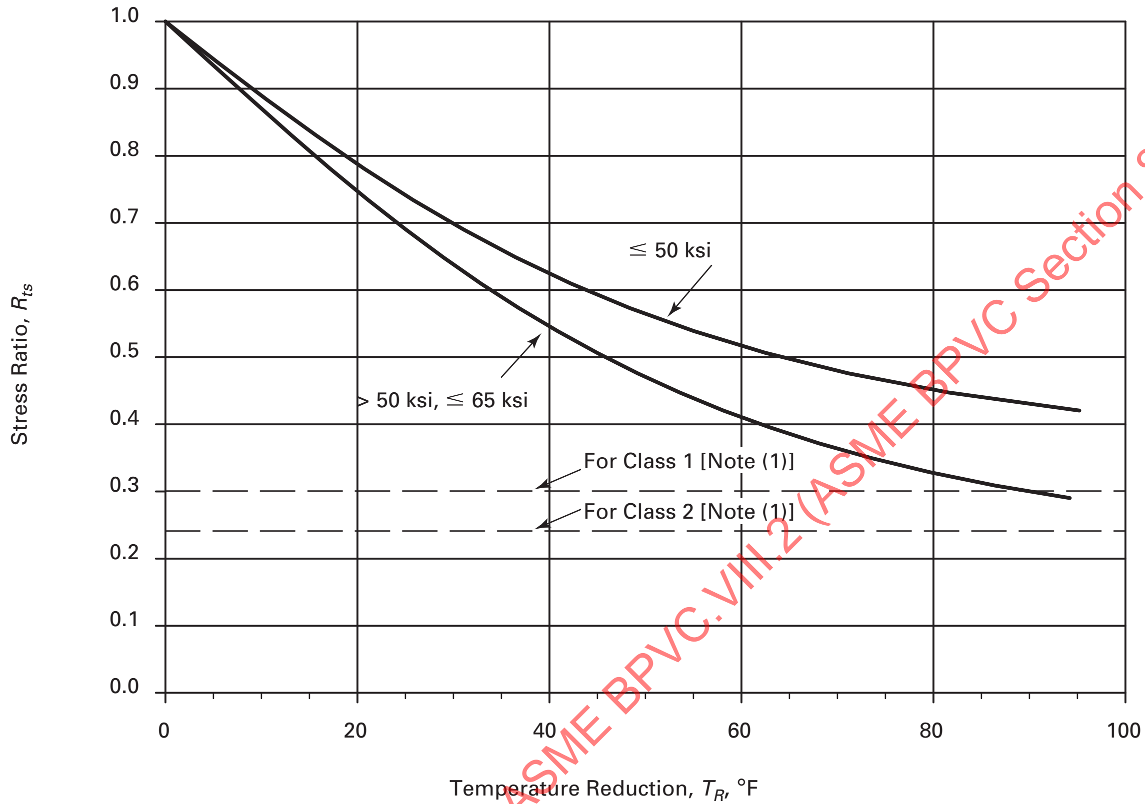
Figure 3.12M
Reduction in the MDMT Without Impact Testing — Parts Not Subject to PWHT



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The reduction in MDMT shall not exceed 55°C (100°F), except as permitted by 3.11.2.5(a), Step 5(b).
(c) Data of Figures 3.12 and 3.12M are shown in Table 3.16.

NOTE:
(1) See 3.11.2.5(a), Step 5(a) when R_{ts} is less than or equal to 0.3 for Class 1, or 0.24 for Class 2.

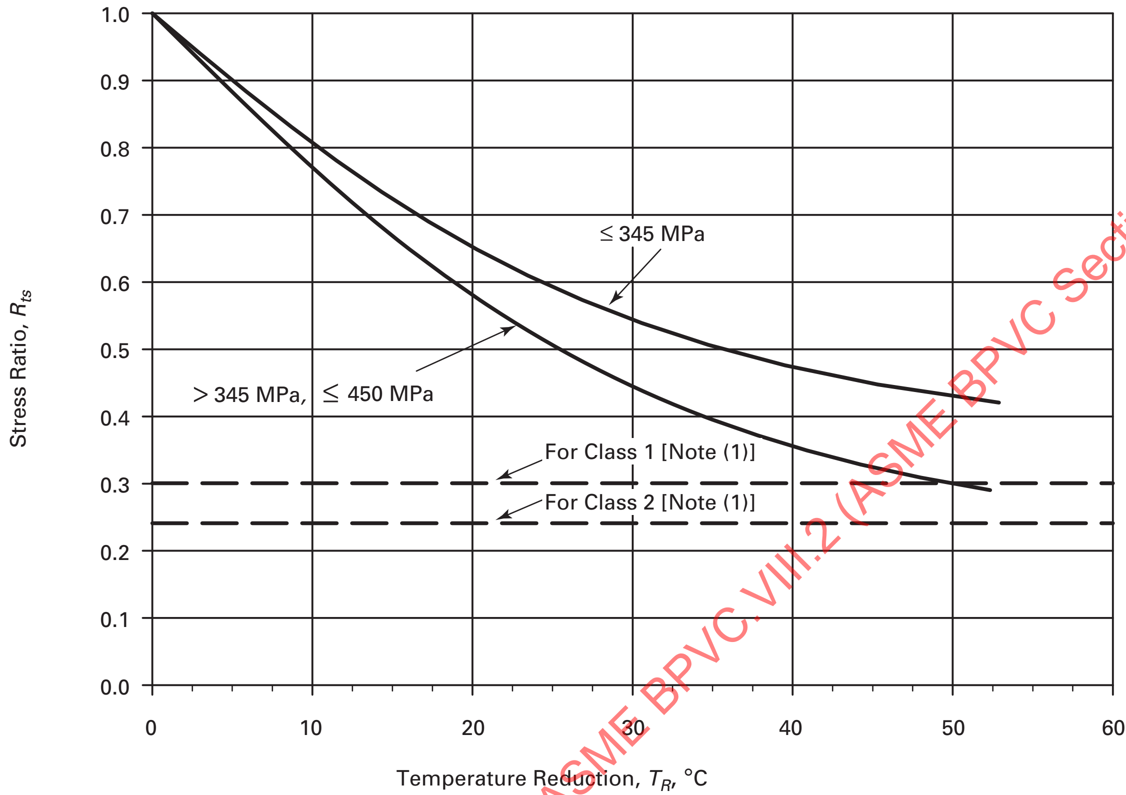
Figure 3.13
Reduction in the MDMT Without Impact Testing — Parts Subject to PWHT and Nonwelded Parts



GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The reduction in MDMT shall not exceed 55°C (100°F), except as permitted by 3.11.2.5(a), Step 5(b).
(c) Data of Figures 3.13 and 3.13M are shown in Table 3.17.

NOTE:
(1) See 3.11.2.5(a), Step 5(a) when R_{ts} is less than or equal to 0.3 for Class 1, or 0.24 for Class 2.

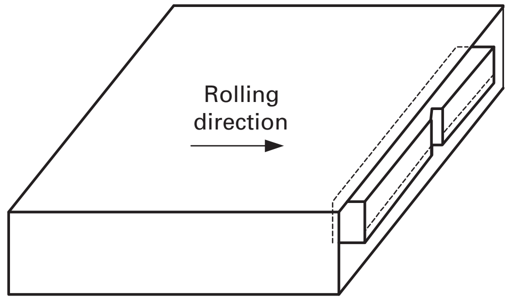
Figure 3.13M
Reduction in the MDMT Without Impact Testing — Parts Subject to PWHT and Nonwelded Parts



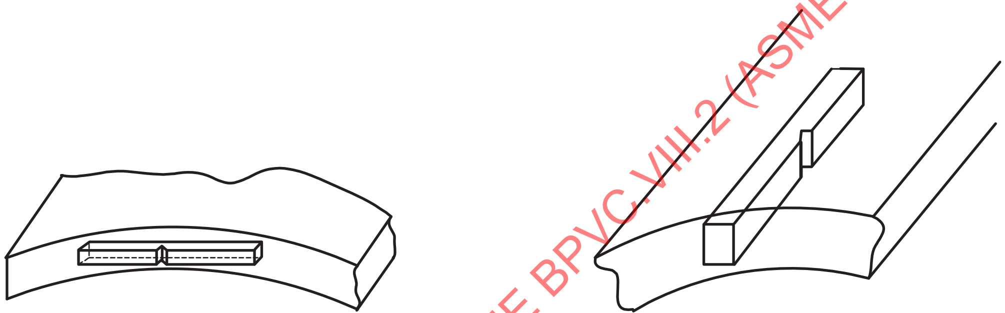
GENERAL NOTES:
(a) Interpolation between yield strength values is permitted.
(b) The reduction in MDMT shall not exceed 55°C (100°F), except as permitted by 3.11.2.5(a), Step 5(b).
(c) Data of Figures 3.13 and 3.13M are shown in Table 3.17.

NOTE:
(1) See 3.11.2.5(a), Step 5(a) when R_{ts} is less than or equal to 0.3 for Class 1, or 0.24 for Class 2.

Figure 3.14
Orientation and Location of Transverse Charpy V-Notch Specimens

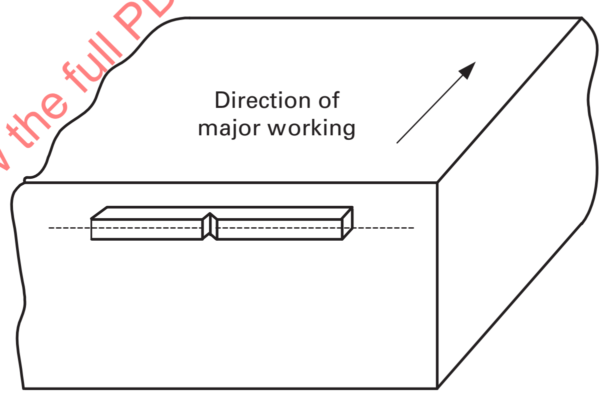


(a) Charpy V-Notch Specimens From Plate



(b-1) (b-2)

(b) Charpy V-Notch Specimens From Pipe



(c) Charpy V-Notch Specimens From Forgings

GENERAL NOTE: The transverse Charpy V-Notch specimen orientation in the pipe shall be as shown in sketch (b-1). If this transverse specimen cannot be obtained from the pipe geometry, then the alternate orientation shown in sketch (b-2) shall be used.

Figure 3.15
Weld Metal Delta Ferrite Content

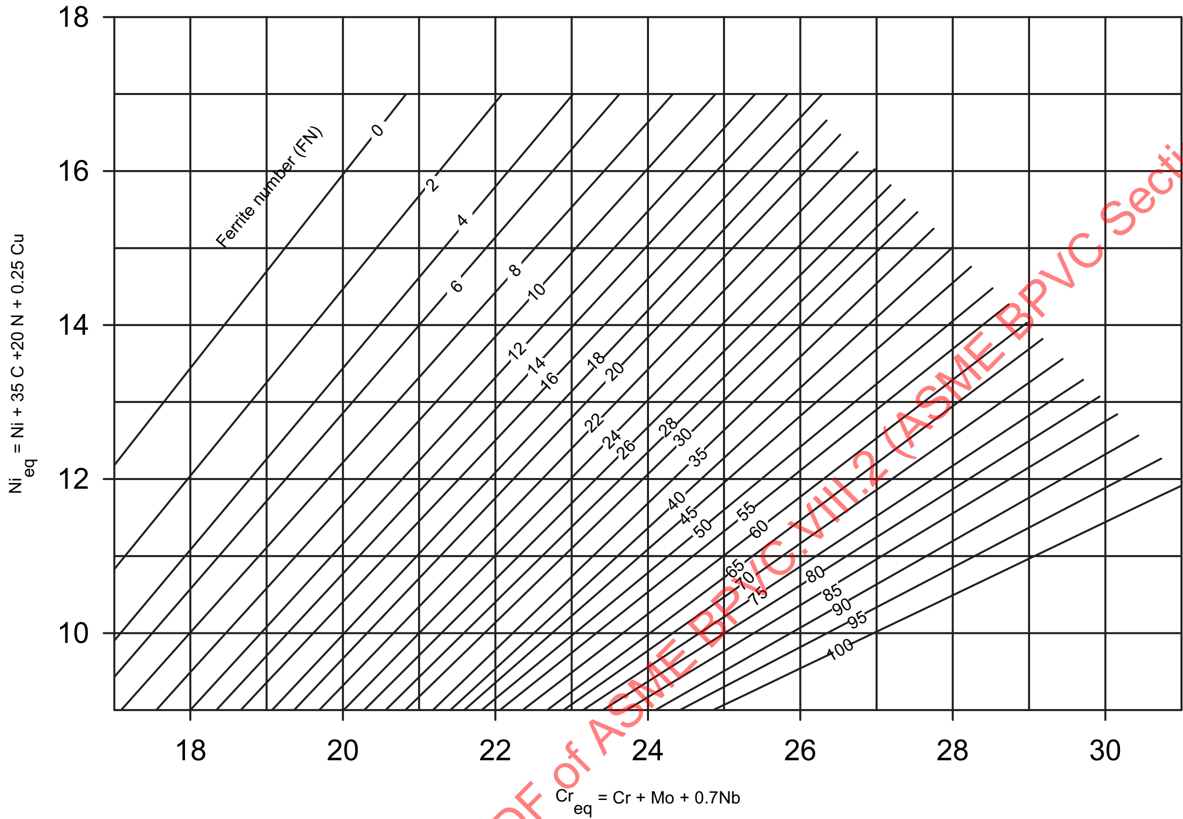
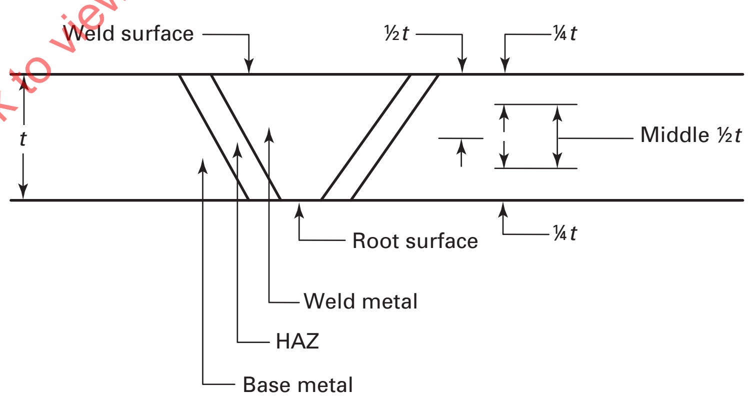


Figure 3.16
HAZ Impact Specimen Removal



ANNEX 3-A
ALLOWABLE DESIGN STRESSES

(Normative)

3-A.1 ALLOWABLE STRESS BASIS — ALL MATERIALS EXCEPT BOLTING

3-A.1.1

The materials that may be used in this Division for all product forms except bolting are shown below.

(a) Carbon Steel and Low Alloy Steel – Table 3-A.1

(b) Quenched and Tempered High Strength Steels – Table 3-A.2

(c) High Alloy Steel – Table 3-A.3

(d) Aluminum and Aluminum Alloys – Table 3-A.4

(e) Copper and Copper Alloys – Table 3-A.5

(f) Nickel and Nickel Alloys – Table 3-A.6

(g) Titanium and Titanium Alloys – Table 3-A.7

3-A.1.2

The allowable stresses to be used in this Division for all product forms except bolting are provided in the following tables of Section II, Part D.

(a) Carbon Steel and Low Alloy Steel

(1) Class 1 – Section II, Part D, Subpart 1, Table 2A

(2) Class 2 – Section II, Part D, Subpart 1, Table 5A

(b) Quenched and Tempered High Strength Steels

(1) Class 1 – Section II, Part D, Subpart 1, Table 2A

(2) Class 2 – Section II, Part D, Subpart 1, Table 5A

(c) High Alloy Steel

(1) Class 1 – Section II, Part D, Subpart 1, Table 2A

(2) Class 2 – Section II, Part D, Subpart 1, Table 5A

(d) Aluminum and Aluminum Alloys

(1) Class 1 – Section II, Part D, Subpart 1, Table 2B

(2) Class 2 – Section II, Part D, Subpart 1, Table 5B

(e) Copper and Copper Alloys

(1) Class 1 – Section II, Part D, Subpart 1, Table 2B

(2) Class 2 – Section II, Part D, Subpart 1, Table 5B

(f) Nickel and Nickel Alloys

(1) Class 1 – Section II, Part D, Subpart 1, Table 2B

(2) Class 2 – Section II, Part D, Subpart 1, Table 5B

(g) Titanium and Titanium Alloys

(1) Class 1 – Section II, Part D, Subpart 1, Table 2B

(2) Class 2 – Section II, Part D, Subpart 1, Table 5B

3-A.2 ALLOWABLE STRESS BASIS — BOLTING MATERIALS

3-A.2.1

The materials that may be used in this Division for bolting are shown below.

(a) Ferrous Bolting Materials for Design in Accordance With Part 4 of this Division – Table 3-A.8

(b) Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4 of this Division – Table 3-A.9

(c) Nickel and Nickel Alloy Bolting Materials Bolting Materials for Design in Accordance With Part 4 of this Division – Table 3-A.10

(d) Bolting Materials for Design in Accordance With Part 5 of this Division – Table 3-A.11

3-A.2.2

The allowable stresses to be used in this Division for bolting are provided in the following tables of Section II, Part D.

(a) Ferrous Bolting Materials for Design in Accordance With Part 4 of this Division – Section II, Part D, Subpart 1, Table 3

(b) Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4 of this Division – Section II, Part D, Subpart 1, Table 3

(c) Nickel and Nickel Alloy Bolting Materials Bolting Materials for Design in Accordance With Part 4 of this Division – Section II, Part D, Subpart 1, Table 3

(d) Bolting Materials for Design in Accordance With Part 5 of this Division – Section II, Part D, Subpart 1, Table 4

3-A.3 TABLES

(21)

Table 3-A.1
Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-36	...	K02600	Carbon steel	Str. plate
SA-105	...	K03504	Carbon steel	Forgings
SA-106	A	K02501	Carbon steel	Smls. pipe
SA-106	B	K03006	Carbon steel	Smls. pipe
SA-106	C	K03501	Carbon steel	Smls. pipe
SA-178	C	K03503	Carbon steel	Wld. tube
SA-181	60	K03502	Carbon steel	Forgings
SA-181	70	K03502	Carbon steel	Forgings
SA-182	F1	K12822	C–$^1/_2$Mo	Forgings
SA-182	F2	K12122	$^1/_2$Cr–$^1/_2$Mo	Forgings
SA-182	F3VCb	K31390	3Cr–1Mo–$^1/_4$V–Cb–Ca	Forgings
SA-182	F3V	K31830	3Cr–1Mo–$^1/_4$V–Ti–B	Forgings
SA-182	F5	K41545	5Cr–$^1/_2$Mo	Forgings
SA-182	F5a	K42544	5Cr–$^1/_2$Mo	Forgings
SA-182	F9	K90941	9Cr–1Mo	Forgings
SA-182	F12, Cl. 1	K11562	1Cr–$^1/_2$Mo	Forgings
SA-182	F12, Cl. 2	K11564	1Cr–$^1/_2$Mo	Forgings
SA-182	F11, Cl. 1	K11597	1$^1/_4$Cr–$^1/_2$Mo–Si	Forgings
SA-182	F11, Cl. 2	K11572	1$^1/_4$Cr–$^1/_2$Mo–Si	Forgings
SA-182	F21	K31545	3Cr–1Mo	Forgings
SA-182	F22, Cl. 1	K21590	2$^1/_4$Cr–1Mo	Forgings
SA-182	F22, Cl. 3	K21590	2$^1/_4$Cr–1Mo	Forgings
SA-182	F22V	K31835	2$^1/_4$Cr–1Mo–$^1/_4$V	Forgings
SA-182	F91	K90901	9Cr–1Mo–V	Forgings
SA-182	FR	K22035	2Ni–1Cu	Forgings
SA-203	A	K21703	2 $^1/_2$Ni	Plate
SA-203	B	K22103	2 $^1/_2$Ni	Plate
SA-203	D	K31718	3$^1/_2$Ni	Plate
SA-203	E	K32018	3$^1/_2$Ni	Plate

Table 3-A.1
Carbon Steel and Low Alloy Materials (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-203	F	...	$3^1/_2$Ni	Plate
SA-204	A	K11820	C–$^1/_2$Mo	Plate
SA-204	B	K12020	C–$^1/_2$Mo	Plate
SA-204	C	K12320	C–$^1/_2$Mo	Plate
SA-209	T1	K11522	C–$^1/_2$Mo	Smls. tube
SA-209	T1a	K12023	C–$^1/_2$Mo	Smls. tube
SA-209	T1b	K11422	C–$^1/_2$Mo	Smls. tube
SA-210	A-1	K02707	Carbon steel	Smls. tube
SA-210	C	K03501	Carbon steel	Smls. tube
SA-213	T2	K11547	$^1/_2$Cr–$^1/_2$Mo	Smls. tube
SA-213	T5	K41545	5Cr–$^1/_2$Mo	Smls. tube
SA-213	T5b	K41545	5Cr–$^1/_2$Mo–Si	Smls. tube
SA-213	T5c	K41245	5Cr–$^1/_2$Mo–Ti	Smls. tube
SA-213	T9	K90941	9Cr–1Mo	Smls. tube
SA-213	T11	K11597	$1^1/_4$Cr–$^1/_2$Mo–Si	Smls. tube
SA-213	T12	K11562	1Cr–$^1/_2$Mo	Smls. tube
SA-213	T21	K31545	3Cr–1Mo	Smls. tube
SA-213	T22	K21590	$2^1/_2$Cr–1Mo	Smls. tube
SA-213	T91	K90901	9Cr–1Mo–V	Smls. tube
SA-216	WCA	J02502	Carbon steel	Castings
SA-216	WCB	J03002	Carbon steel	Castings
SA-216	WCC	K02503	Carbon steel	Castings
SA-217	C5	J42045	5Cr–$^1/_2$Mo	Castings
SA-217	C12	J82090	9Cr–1Mo	Castings
SA-217	WC1	J12524	C–$^1/_2$Mo	Castings
SA-217	WC4	J12082	1Ni–$^1/_2$Cr–$^1/_2$Mo	Castings
SA-217	WC5	J22000	$^3/_4$Ni–1Mo–$^3/_4$Cr	Castings
SA-217	WC6	J12072	$1^1/_4$Cr–$^1/_2$Mo	Castings
SA-217	WC9	J21890	$2^1/_4$Cr–1Mo	Castings
SA-225	C	K12524	Mn–$^1/_2$Ni–V	Plate
SA-234	WPB	K03006	Carbon steel	Fittings
SA-234	WPC	K03501	Carbon steel	Fittings
SA-234	WP1	K12821	C–$^1/_2$Mo	Fittings
SA-234	WP5	K41515	5Cr–$^1/_2$Mo	Fittings
SA-234	WP9	K90941	9Cr–1Mo	Fittings
SA-234	WP11, Cl. 1	...	$1^1/_4$Cr–$^1/_2$Mo–Si	Fittings
SA-234	WP12, Cl. 1	K12062	1Cr–$^1/_2$Mo	Fittings
SA-234	WP22, Cl. 1	K21590	$2^1/_4$Cr–1Mo	Fittings
SA-266	1	K03506	Carbon steel	Forgings
SA-266	2	K03506	Carbon steel	Forgings
SA-266	3	K05001	Carbon steel	Forgings
SA-266	4	K03017	Carbon steel	Forgings
SA-283	B	...	Carbon steel	Str. plate
SA-283	D	...	Carbon steel	Str. plate
SA-285	A	K01700	Carbon steel	Plate
SA-285	B	K02200	Carbon steel	Plate
SA-285	C	K02801	Carbon steel	Plate
SA-299	A	K02803	Carbon steel	Plate
SA-299 [Note (1)]	B	K02803	Carbon steel	Plate
SA-302	A	K12021	Mn–$^1/_2$Mo	Plate
SA-302	B	K12022	Mn–$^1/_2$Mo	Plate
SA-302	C	K12039	Mn–$^1/_2$Mo–$^1/_2$Ni	Plate
SA-302	D	K12054	Mn–$^1/_2$Mo–$^3/_4$Ni	Plate
SA-333	1	K03008	Carbon steel	Smls. pipe

Table 3-A.1
Carbon Steel and Low Alloy Materials (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-333	3	K31918	$3^1/_2$Ni	Smls. pipe
SA-333	4	K11267	$^3/_4$Cr–$^3/_4$Ni–Cu–Al	Smls. pipe
SA-333	6	K03006	Carbon steel	Smls. pipe
SA-333	9	K22035	2Ni–1Cu	Smls. pipe
SA-333	1	K03008	Carbon steel	Wld. pipe
SA-334	1	K03008	Carbon steel	Wld. tube
SA-334	1	K03008	Carbon steel	Smls. tube
SA-334	3	K31918	$3^1/_2$Ni	Smls. tube
SA-334	9	K22035	2Ni–1Cu	Smls. tube
SA-335	P1	K11522	C–$^1/_2$Mo	Smls. pipe
SA-335	P2	K11547	$^1/_2$Cr–$^1/_2$Mo	Smls. pipe
SA-335	P5	K41545	5Cr–$^1/_2$Mo	Smls. pipe
SA-335	P5b	K51545	5Cr–$^1/_2$Mo–Si	Smls. pipe
SA-335	P5c	K41245	5Cr–$^1/_2$Mo–Ti	Smls. pipe
SA-335	P9	K90941	9Cr–1Mo	Smls. pipe
SA-335	P11	K11597	$1^1/_4$Cr–$^1/_2$Mo–Si	Smls. pipe
SA-335	P12	K11562	1Cr–$^1/_2$Mo	Smls. pipe
SA-335	P21	K31545	3Cr–1Mo	Smls. pipe
SA-335	P22	K21590	$2^1/_4$Cr–1Mo	Smls. pipe
SA-335	P91	K90901	9Cr–1Mo–V	Smls. pipe
SA-336	F1	K11564	1Cr–$^1/_2$Mo	Forgings
SA-336	F3VCb	K31390	3Cr–1Mo–$^1/_4$V–Cb–Ca	Forgings
SA-336	F3V	K31830	3Cr–1Mo–$^1/_4$V–Ti–B	Forgings
SA-336	F5	K41545	5Cr–1Mo	Forgings
SA-336	F5A	K42544	5Cr–1Mo	Forgings
SA-336	F9	K90941	9Cr–1Mo	Forgings
SA-336	F11, Cl. 2	K11572	$1^1/_4$Cr–$^1/_2$Mo–Si	Forgings
SA-336	F11, Cl. 3	K11572	$1^1/_4$Cr–$^1/_2$Mo–Si	Forgings
SA-336	F12	K11564	1Cr–$^1/_2$Mo	Forgings
SA-336	F21, Cl. 1	K31545	3Cr–1Mo	Forgings
SA-336	F21, Cl. 3	K31545	3Cr–1Mo	Forgings
SA-336	F22, Cl. 1	K21590	$2^1/_4$Cr–1Mo	Forgings
SA-336	F22, Cl. 3	K21590	$2^1/_4$Cr–1Mo	Forgings
SA-336	F22V	K31835	$2^1/_4$Cr–1Mo–$^1/_4$V	Forgings
SA-336 [Note (1)]	F91	K90901	9Cr–1Mo–V	Forgings
SA-350	LF1	K03009	Carbon steel	Forgings
SA-350	LF2	K03011	Carbon steel	Forgings
SA-350	LF3	K32025	$3^1/_2$Ni	Forgings
SA-350	LF9	K22036	2Ni–1Cu	Forgings
SA-352	LCB	J03003	Carbon steel	Castings
SA-352	LC1	J12522	C–$^1/_2$Mo	Castings
SA-352	LC2	J22500	$2^1/_2$Ni	Castings
SA-352	LC3	J31550	$3^1/_2$Ni	Castings
SA-369	FP1	K11522	C–$^1/_2$Mo	Forged pipe
SA-369	FP2	K11547	$^1/_2$Cr–$^1/_2$Mo	Forged pipe
SA-369	FP5	K41545	5Cr–$^1/_2$Mo	Forged pipe
SA-369	FP9	K90941	9Cr–1Mo	Forged pipe
SA-369	FP11	K11597	$1^1/_4$Cr–$^1/_2$Mo–Si	Forged pipe
SA-369	FP12	K11562	1Cr–$^1/_2$Mo	Forged pipe
SA-369	FP21	K31545	3Cr–1Mo	Forged pipe
SA-369	FP22	K21590	$2^1/_4$Cr–1Mo	Forged pipe
SA-372	A	K03002	Carbon steel	Forgings
SA-372	B	K04001	Carbon steel	Forgings
SA-372	C	K04801	Carbon steel	Forgings

Table 3-A.1
Carbon Steel and Low Alloy Materials (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-372	D	K10508	Mn–$\frac{1}{4}$Mo	Forgings
SA-387	2, Cl. 1	K12143	$\frac{1}{2}$Cr–$\frac{1}{2}$Mo	Plate
SA-387	2, Cl. 2	K12143	$\frac{1}{2}$Cr–$\frac{1}{2}$Mo	Plate
SA-387	5, Cl. 1	K41545	5Cr–$\frac{1}{2}$Mo	Plate
SA-387	5, Cl. 2	K41545	5Cr–$\frac{1}{2}$Mo	Plate
SA-387	11, Cl. 1	K11789	1$\frac{1}{4}$Cr–$\frac{1}{2}$Mo–Si	Plate
SA-387	11, Cl. 2	K11789	1$\frac{1}{4}$Cr–$\frac{1}{2}$Mo–Si	Plate
SA-387	12, Cl. 1	K11757	1Cr–$\frac{1}{2}$Mo	Plate
SA-387	12, Cl. 2	K11757	1Cr–$\frac{1}{2}$Mo	Plate
SA-387	21, Cl. 1	K31545	3Cr–1Mo	Plate
SA-387	21, Cl. 2	K31545	3Cr–1Mo	Plate
SA-387	22, Cl. 1	K21590	2$\frac{1}{4}$Cr–1Mo	Plate
SA-387	22, Cl. 2	K21590	2$\frac{1}{4}$Cr–1Mo	Plate
SA-387	91	K90901	9Cr–1Mo–V	Plate
SA-420	WPL3	...	3$\frac{1}{2}$Ni	Fittings
SA-420	WPL6	...	Carbon steel	Fittings
SA-420	WPL9	K22035	2Ni–Cu	Fittings
SA-423	1	K11535	$\frac{1}{4}$Cr–$\frac{1}{2}$Ni–Cu	Smls. tube
SA-423	2	K11540	$\frac{3}{4}$Ni–$\frac{3}{4}$Cu–Mo	Smls. tube
SA-487	1, Cl. A	J13002	Mn–V	Castings
SA-487	4, Cl. A	J13047	$\frac{1}{2}$Ni–$\frac{1}{2}$Cr–$\frac{1}{4}$Mo–V	Castings
SA-487	8, Cl. A	J22091	2$\frac{1}{2}$Cr–1Mo	Castings
SA-508	1	K13502	Carbon steel	Forgings
SA-508	1A	K13502	Carbon steel	Forgings
SA-508	2, Cl. 1	K12766	$\frac{3}{4}$Ni–$\frac{1}{2}$Mo–$\frac{1}{3}$Cr–V	Forgings
SA-508	2, Cl. 2	K12766	$\frac{3}{4}$Ni–$\frac{1}{2}$Mo–$\frac{1}{3}$Cr–V	Forgings
SA-508	3, Cl. 1	K12042	$\frac{3}{4}$Ni–$\frac{1}{2}$Mo–Cr–V	Forgings
SA-508	3, Cl. 2	K12042	$\frac{3}{4}$Ni–$\frac{1}{2}$Mo–Cr–V	Forgings
SA-508	3VCb	K31390	3Cr–1Mo–$\frac{1}{4}$V–Cb–Ca	Forgings
SA-508	3V	K31830	3Cr–1Mo–$\frac{1}{4}$V–Ti–B	Forgings
SA-508	4N, Cl. 3	K22375	3$\frac{1}{2}$Ni–1$\frac{3}{4}$Cr–$\frac{1}{2}$Mo–V	Forgings
SA-508	22, Cl. 3	K215909	2$\frac{1}{4}$Cr–1Mo	Forgings
SA-515	60	K02401	Carbon steel	Plate
SA-515	65	K02800	Carbon steel	Plate
SA-515	70	K03101	Carbon steel	Plate
SA-516	55	K01800	Carbon steel	Plate
SA-516	60	K02100	Carbon steel	Plate
SA-516	65	K02403	Carbon steel	Plate
SA-516	70	K02700	Carbon steel	Plate
SA-524	I	K02104	Carbon steel	Smls. pipe
SA-524	II	K02104	Carbon steel	Smls. pipe
SA-533	A, Cl. 1	K12521	Mn–$\frac{1}{2}$Mo	Plate
SA-533	A, Cl. 2	K12521	Mn–$\frac{1}{2}$Mo	Plate
SA-533	B, Cl. 1	K12539	Mn–$\frac{1}{2}$Mo–$\frac{1}{2}$Ni	Plate
SA-533	B, Cl. 2	K12539	Mn–$\frac{1}{2}$Mo–$\frac{1}{2}$Ni	Plate
SA-533	C, Cl. 1	K12554	Mn–$\frac{1}{2}$Mo–$\frac{3}{4}$Ni	Plate
SA-533	C, Cl. 2	K12554	Mn–$\frac{1}{2}$Mo–$\frac{3}{4}$Ni	Plate
SA-533	D, Cl. 2	K12529	Mn–$\frac{1}{2}$Mo–$\frac{1}{4}$Ni	Plate
SA-533 [Note (1)]	E, Cl. 1	K12554	Mn–$\frac{1}{2}$Mo–$\frac{3}{4}$Ni	Plate
SA-533 [Note (1)]	E, Cl. 2	K12554	Mn–$\frac{1}{2}$Mo–$\frac{3}{4}$Ni	Plate
SA-537	Cl. 1	K12437	Carbon steel	Plate
SA-537	Cl. 2	K12437	Carbon steel	Plate
SA-537	Cl. 3	K12437	Carbon steel	Plate
SA-541	1	K03506	Carbon steel	Forgings

Table 3-A.1
Carbon Steel and Low Alloy Materials (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-541	1A	K03020	Carbon steel	Forgings
SA-541	2, Cl. 1	K12765	$\frac{3}{4}$Ni–$\frac{1}{2}$Mo–$\frac{1}{3}$Cr–V	Forgings
SA-541	2, Cl. 2	K12765	$\frac{3}{4}$Ni–$\frac{1}{2}$Mo–$\frac{1}{3}$Cr–V	Forgings
SA-541	3, Cl. 1	K12045	$\frac{1}{2}$Ni–$\frac{1}{2}$Mo–V	Forgings
SA-541	3, Cl. 2	K12045	$\frac{1}{2}$Ni–$\frac{1}{2}$Mo–V	Forgings
SA-541	3VCb	K31390	3Cr–1Mo–$\frac{1}{4}$V–Cb–Ca	Forgings
SA-541	3V	K31830	3Cr–1Mo–$\frac{1}{4}$V–Ti–B	Forgings
SA-541	22, Cl. 3	K21390	$2\frac{1}{4}$Cr–1Mo	Forgings
SA-541	22V	K31835	$2\frac{1}{4}$Cr–1Mo–$\frac{1}{4}$V	Forgings
SA-542	B, Cl. 4	...	$2\frac{1}{4}$Cr–1Mo	Plate
SA-542	E, Cl. 4a	K31390	3Cr–1Mo–$\frac{1}{4}$V–Cb–Ca	Plate
SA-542	C, Cl. 4a	...	3Cr–1Mo–$\frac{1}{4}$V–Ti–B	Plate
SA-542	D, Cl. 4a	...	$2\frac{1}{4}$Cr–1Mo–$\frac{1}{4}$V	Plate
SA-612	...	K02900	Carbon steel	Plate
SA-662	A	K10701	Carbon steel	Plate
SA-662	B	K02203	Carbon steel	Plate
SA-662	C	K02007	Carbon steel	Plate
SA-675	45	...	Carbon steel	Bar, shapes
SA-675	50	...	Carbon steel	Bar, shapes
SA-675	55	...	Carbon steel	Bar, shapes
SA-675	60	...	Carbon steel	Bar, shapes
SA-675	65	...	Carbon steel	Bar, shapes
SA-675	70	...	Carbon steel	Bar, shapes
SA-727	...	K02506	Carbon steel	Forgings
SA-737	B	K12001	C–Mn–Si–Cb	Plate
SA-737	C	K12202	C–Mn–Si–V	Plate
SA-738	A	K12447	Carbon steel	Plate
SA-738	B	K12007	Carbon steel	Plate
SA-738	C	...	Carbon steel	Plate
SA-739	B11	K11797	$1\frac{1}{4}$Cr–$\frac{1}{2}$Mo	Bar
SA-739	B22	K21390	$2\frac{1}{4}$Cr–1Mo	Bar
SA-765	I	K03046	Carbon steel	Forgings
SA-765	II	K03047	Carbon steel	Forgings
SA-765	III	K32026	$3\frac{1}{2}$Ni	Forgings
SA-765	IV	K02009	Carbon steel	Forgings
SA-832	23V	K31390	3Cr–1Mo–$\frac{1}{4}$V–Cb–Ca	Plate
SA-832	21V	K31830	3Cr–1Mo–$\frac{1}{4}$V–Ti–B	Plate
SA-832	22V	K31835	$2\frac{1}{4}$Cr–1Mo–V	Plate
SA-841 [Note (1)]	A, Cl. 1	...	Carbon steel	Plate
SA-841 [Note (1)]	B, Cl. 2	...	Carbon steel	Plate
SA/EN 10028-2 [Note (1)]	P355GH	...	Carbon steel	Plate
SA/EN 10028-2 [Note (1)]	13CrMo4–5	...	1Cr–$\frac{1}{2}$Mo	Plate
SA/EN 10028-2 [Note (1)]	10CrMo9–10	...	$2\frac{1}{4}$Cr–1Mo	Plate
SA/EN 10028-2 [Note (1)]	13CrMoSi5–5 +QT	...	$1\frac{1}{4}$Cr–$\frac{1}{2}$Mo–Si	Plate
SA/EN 10222-2 [Note (1)]	P280GH	...	Carbon steel	Forgings
SA/EN 10222-2 [Note (1)]	P30SGH	...	Carbon steel	Forgings
SA/EN 10222-2 [Note (1)]	13CrMo4–5	...	1Cr–$\frac{1}{2}$Mo	Forgings
SA/EN 10222-2 [Note (1)]	11CrMo9–10	...	$2\frac{1}{4}$Cr–1Mo	Forgings

NOTE:
(1) For Class 2 construction only.

Table 3-A.2
Quenched and Tempered High Strength Steels

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-333	8	K81340	9Ni	Smls. pipe
SA-334	8	K81340	9Ni	Smls. tube
SA-353	...	K81340	9Ni	Plate
SA-372	D	K14508	Mn–$^{1}/_{4}$Mo	Forgings
SA-372	E, Cl. 70	K13047	1Cr–$^{1}/_{5}$Mo	Forgings
SA-372	F, Cl. 70	G41350	1Cr–$^{1}/_{5}$Mo	Forgings
SA-372	G, Cl. 70	K13049	$^{1}/_{2}$Cr–$^{1}/_{5}$Mo	Forgings
SA-372	H, Cl. 70	K13547	$^{1}/_{2}$Cr–$^{1}/_{5}$Mo	Forgings
SA-372	J, Cl. 70	K13548	1Cr–$^{1}/_{5}$Mo	Forgings
SA-372	J, Cl. 110	G41370	1Cr–$^{1}/_{5}$Mo	Forgings
SA-420	WPL8	K81340	9Ni	Smls. pipe
SA-508	4N, Cl. 1	K22375	3$^{1}/_{2}$Ni–1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-508	4N, Cl. 2	K22375	3$^{1}/_{2}$Ni–1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-517	A	K11856	$^{1}/_{2}$Cr–$^{1}/_{4}$Mo–Si	Plate
SA-517	B	K11630	$^{1}/_{2}$Cr–$^{1}/_{5}$Mo–V	Plate
SA-517	E	K21604	1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo–Ti	Plate
SA-517	F	K11576	$^{3}/_{4}$Ni–$^{1}/_{2}$Cr–$^{1}/_{2}$Mo–V	Plate
SA-517	J	K11625	C–$^{1}/_{2}$Mo	Plate
SA-517	P	K21650	1$^{1}/_{4}$Ni–1Cr–$^{1}/_{2}$Mo	Plate
SA-522	I	K81340	9Ni	Forgings
SA-533	B, Cl. 3	K12554	Mn–$^{1}/_{2}$Mo–$^{3}/_{4}$Ni	Plate
SA-533	D, Cl. 3	K12529	Mn–$^{1}/_{2}$Mo–$^{1}/_{4}$Ni	Plate
SA-543	B, Cl. 1	K42339	3Ni–1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo	Plate
SA-543	B, Cl. 2	K42339	3Ni–1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo	Plate
SA-543	B, Cl. 3	K42339	3Ni–1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo	Plate
SA-543	C, Cl. 1	...	2$^{3}/_{4}$Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo	Plate
SA-543	C, CL. 2	...	2$^{3}/_{4}$Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo	Plate
SA-543	C, CL. 3	...	2$^{3}/_{4}$Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo	Plate
SA-553	I	K81340	9Ni	Plate
SA-553	III	K61365	7Ni	Plate
SA-592	A	K11856	$^{1}/_{2}$Cr–$^{1}/_{4}$Mo–Si	Forgings
SA-592	E	K11695	1$^{3}/_{4}$Cr–$^{1}/_{2}$Mo–Cu	Forgings
SA-592	F	K11576	$^{3}/_{4}$Ni–$^{1}/_{2}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-645	A	K41583	5Ni–$^{1}/_{4}$Mo	Plate
SA-723	1, Cl. 1	K23550	2Ni–1$^{1}/_{2}$Cr–$^{1}/_{4}$Mo–V	Forgings
SA-723	1, Cl. 2	K23550	2Ni–1$^{1}/_{2}$Cr–$^{1}/_{4}$Mo–V	Forgings
SA-723	2, Cl. 1	K34035	2$^{3}/_{4}$Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-723	2, Cl. 2	K34035	2$^{3}/_{4}$Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-723	3, Cl. 1	K44045	4Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-723	3, Cl. 2	K44045	4Ni–1$^{1}/_{2}$Cr–$^{1}/_{2}$Mo–V	Forgings
SA-724	A	K11831	Carbon steel	Plate
SA-724	B	K12031	Carbon steel	Plate
SA-724	C	K12037	Carbon steel	Plate

(21)

Table 3-A.3
High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-182	FXM-11	S21904	21Cr–6Ni–9Mn	Forgings
SA-182	FXM-19	S20910	22Cr–13Ni–5Mn	Forgings
SA-182	F6a, Cl. 1	S41000	13Cr	Forgings
SA-182	F6a, Cl. 2	S41000	13Cr	Forgings
SA-182	F51	S31803	22Cr–5Ni–3Mo–N	Forgings
SA-182	F58	S31266	24Cr–22.5Ni–5.7Mo–Cu–W	Forgings
SA-182 [Note (1)]	F60	S32205	22Cr–5.5Ni–3Mo–N	Forgings
SA-182	F304	S30400	18Cr–8Ni	Forgings
SA-182	F304H	S30409	18Cr–8Ni	Forgings
SA-182	F304L	S30403	18Cr–8Ni	Forgings
SA-182	F310	S31000	25Cr–20Ni	Forgings
SA-182 [Note (1)]	F310MoLN	S31050	25Cr–22Ni–2Mo–N	Forgings
SA-182	F316	S31600	16Cr–12Ni–2Mo	Forgings
SA-182	F316H	S31609	16Cr–12Ni–2Mo	Forgings
SA-182	F316L	S31603	16Cr–12Ni–2Mo	Forgings
SA-182	F321	S32100	18Cr–10Ni–Ti	Forgings
SA-182	F321H	S32109	18Cr–10Ni–Ti	Forgings
SA-182	F347	S34700	18Cr–10Ni–Cb	Forgings
SA-182	F347H	S34909	18Cr–10Ni–Cb	Forgings
SA-182	F348	S34800	18Cr–10Ni–Cb	Forgings
SA-182	F348H	S34809	18Cr–10Ni–Cb	Forgings
SA-213	TP304	S30400	18Cr–8Ni	Smls. tube
SA-213	TP304H	S30409	18Cr–8Ni	Smls. tube
SA-213	TP304L	S30403	18Cr–8Ni	Smls. tube
SA-213	TP304N	S30451	18Cr–8Ni–N	Smls. tube
SA-213	TP309Cb	S30940	23Cr–12Ni–Cb	Smls. tube
SA-213	TP309H	S30909	23Cr–12Ni	Smls. tube
SA-213	TP309S	S30908	23Cr–12Ni	Smls. tube
SA-213	TP310H	S31009	25Cr–20Ni	Smls. tube
SA-213	TP310MoLN	S31050	25Cr–22Ni–2Mo–N	Smls. tube
SA-213	TP310S	S31008	25Cr–20Ni	Smls. tube
SA-213	TP316	S31600	16Cr–12Ni–2Mo	Smls. tube
SA-213	TP316H	S31609	16Cr–12Ni–2Mo	Smls. tube
SA-213	TP316L	S31603	16Cr–12Ni–2Mo	Smls. tube
SA-213	TP316N	S31651	16Cr–12Ni–2Mo–N	Smls. tube
SA-213	TP321	S32100	18Cr–10Ni–Ti	Smls. tube
SA-213	TP321H	S32109	18Cr–10Ni–Ti	Smls. tube
SA-213	TP347	S34700	18Cr–10Ni–Cb	Smls. tube
SA-213	TP347H	S34709	18Cr–10Ni–Cb	Smls. tube
SA-213	TP348	S34800	18Cr–10Ni–Cb	Smls. tube
SA-213	TP348H	S34809	18Cr–10Ni–Cb	Smls. tube
SA-213	XM-15	S38100	18Cr–18Ni–2Si	Smls. tube
SA-217	CA15	J91150	13Cr	Castings
SA-240	XM-15	S38100	18Cr–18Ni–2Si	Plate
SA-240	XM19	S20910	22Cr–13Ni–5Mn	Plate
SA-240	XM-29	S24000	18Cr–3Ni–12Mn	Plate
SA-240	XM-29	S24000	18Cr–3Ni–12Mn	Sheet and strip
SA-240	201LN	S20153	16Cr–4Ni–6Mn	Plate
SA-240 [Note (1)]	255	S32550	25Cr–5Ni–3Mo–2Cu	Plate
SA-240	302	S30200	18Cr–8Ni	Plate

Table 3-A.3
High Alloy Steel (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-240	304	S30400	18Cr–8Ni	Plate
SA-240	304H	S30409	18Cr–8Ni	Plate
SA-240	304L	S30403	18Cr–8Ni	Plate
SA-240	304N	S30451	18Cr–8Ni–N	Plate
SA-240	...	S30601	17.5Cr–17.5Ni–5.3Si	Plate
SA-240	...	S31266	24Cr–22.5Ni–5.7Mo–Cu–W	Plate
SA-240	309Cb	S30940	23Cr–12Ni–Cb	Plate
SA-240	309H	S30909	23Cr–12Ni	Plate
SA-240	309S	S30908	23Cr–12Ni	Plate
SA-240	310H	S31009	25Cr–20Ni	Plate
SA-240	310MoLN	S31050	25Cr–22Ni–2Mo–N	Plate
SA-240	310S	S31008	25Cr–20Ni	Plate
SA-240	316	S31600	16Cr–12Ni–2Mo	Plate
SA-240	316L	S31603	16Cr–12Ni–2Mo	Plate
SA-240	316N	S31651	16Cr–12Ni–2Mo–N	Plate
SA-240	317	S31700	18Cr–13Ni–3Mo	Plate
SA-240	317L	S31703	18Cr–13Ni–3Mo	Plate
SA-240	321	S32100	18Cr010Ni–Ti	Plate
SA-240	321H	S32109	18Cr–10Ni–Ti	Plate
SA-240	347	S34700	18Cr–10Ni–Cb	Plate
SA-240	347H	S34709	18Cr–10Ni–Cb	Plate
SA-240	348	S34800	18Cr–10Ni–Cb	Plate
SA-240	405	S40500	13Cr–1Al	Plate
SA-240	410	S41000	13Cr	Plate
SA-240	410S	S41008	13Cr	Plate
SA-240	429	S42900	15Cr	Plate
SA-240	430	S43000	17Cr	Plate
SA-240	...	S31803	22Cr–5Ni–3Mo–N	Plate
SA-240 [Note (1)]	2205	S32205	22Cr–5.5Ni–3Mo–N	Plate
SA-240	26-3-3	S44660	26Cr–3Ni–3Mo	Plate
SA-240 [Note (1)]	...	S32906	29Cr–6.5Ni–2Mo–N	Plate, sheet, and strip
SA-249	TPXM-15	S38100	18Cr–18Ni–2Si	Wld. tube
SA-249	TPXM-19	S20910	22Cr–13Ni–5Mn	Wld. tube
SA-249	TP304	S30400	18Cr–8Ni	Wld. tube
SA-249	TP304H	S30409	18Cr–8Ni	Wld. tube
SA-249	TP304L	S30403	18Cr–8Ni	Wld. tube
SA-249	TP304N	S30451	18Cr–8Ni–N	Wld. tube
SA-249	TP309Cb	S30940	23Cr–12Ni–Cb	Wld. tube
SA-249	TP309H	S30909	23Cr–12Ni	Wld. tube
SA-249	TP309S	S30908	23Cr–12Ni	Wld. tube
SA-249	TP310Cb	S31040	25Cr–20Ni–Cb	Wld. tube
SA-249	TP310H	S31009	23Cr–12Ni	Wld. tube
SA-249	TP310MoLN	S31050	25Cr–22Ni–2Mo–N	Wld. tube
SA-249	TP310S	S31008	23Cr–12Ni	Wld. tube
SA-249	...	S31266	24Cr–22Ni–6Mo–2W–Cu–N	Wld. tube
SA-249	TP316	S31600	16Cr–12Ni–2Mo	Wld. tube
SA-249	TP316H	S31609	16Cr–12Ni–2Mo	Wld. tube
SA-249	TP316L	S31603	16Cr–12Ni–2Mo	Wld. tube
SA-249	TP316N	S31651	16Cr–12Ni–2Mo–N	Wld. tube
SA-249	TP317	S31700	18Cr–3Ni–3Mo	Wld. tube
SA-249	TP321	S32100	18Cr–10Ni–Ti	Wld. tube

Table 3-A.3
High Alloy Steel (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-249	TP321H	S32109	18Cr–10Ni–Ti	Wld. tube
SA-249	TP347	S34700	18Cr–10Ni–Cb	Wld. tube
SA-249	TP347H	S34709	18Cr–10Ni–Cb	Wld. tube
SA-249	TP348	S34800	18Cr–10Ni–Cb	Wld. tube
SA-249	TP348H	S34809	18Cr–10Ni–Cb	Wld. tube
SA-268	TP405	S40500	12Cr–1Al	Smls. tube
SA-268	TP410	S41000	13Cr	Smls. tube
SA-268	TP429	S42900	15Cr	Smls. tube
SA-268	TP430	S43000	17Cr	Smls. tube
SA-268	26-3-3	S44660	26Cr–3Ni–3Mo	Smls. tube
SA-268	26-3-3	S44660	26Cr–3Ni–3Mo	Wld. tube
SA-312	TPXM-11	S21904	21Cr–6Ni–9Mn	Smls. pipe
SA-312	TPXM-15	S38100	18Cr–18Ni–2Si	Smls. pipe
SA-312	TPXM-19	S20910	22Cr–13Ni–5Mn	Smls. pipe
SA-312	TP304	S30400	18Cr–8Ni	Smls. pipe
SA-312	TP304H	S30409	18Cr–8Ni	Smls. pipe
SA-312	TP304L	S30403	18Cr–8Ni	Smls. pipe
SA-312	TP304N	S30451	18Cr–8Ni–N	Smls. pipe
SA-312	TP309Cb	S30940	23Cr–12Ni–Cb	Smls. pipe
SA-312	TP309H	S30909	23Cr–12Ni	Smls. pipe
SA-312	TP309S	S30908	23Cr–12Ni	Smls. pipe
SA-312	TP310H	S31009	23Cr–12Ni	Smls. pipe
SA-312	TP310S	S31008	23Cr–12Ni	Smls. pipe
SA-312	TP316	S31600	16Cr–12Ni–2Mo	Smls. pipe
SA-312	TP316H	S31609	16Cr–12Ni–2Mo	Smls. pipe
SA-312	TP316L	S31603	16Cr–12Ni–2Mo	Smls. pipe
SA-312	TP316N	S31651	16Cr–12Ni–2Mo–N	Smls. pipe
SA-312	TP317	S31700	18Cr–3Ni–3Mo	Smls. pipe
SA-312	TP321	S32100	18Cr–10Ni–Ti	Smls. pipe
SA-312	TP321	S32100	18Cr–10Ni–Ti	Smls. pipe
SA-312	TP321H	S32109	18Cr–10Ni–Ti	Smls. pipe
SA-312	TP321H	S32109	18Cr–10Ni–Ti	Smls. pipe
SA-312	TP347	S34700	18Cr–10Ni–Cb	Smls. pipe
SA-312	TP347H	S34709	18Cr–10Ni–Cb	Smls. pipe
SA-312	TP348	S34800	18Cr–10Ni–Cb	Smls. pipe
SA-312	TP348H	S34809	18Cr–10Ni–Cb	Smls. pipe
SA-312	TPXM-11	S21904	21Cr–6Ni–9Mn	Wld. pipe
SA-312	TPXM-15	S38100	18Cr–18Ni–2Si	Wld. pipe
SA-312	TPXM-19	S20910	22Cr–13Ni–5Mn	Wld. pipe
SA-312	TP304	S30400	18Cr–8Ni	Wld. pipe
SA-312	TP304H	S30409	18Cr–8Ni	Wld. pipe
SA-312	TP304L	S30403	18Cr–8Ni	Wld. pipe
SA-312	TP304N	S30451	18Cr–8Ni–N	Wld. pipe
SA-312	TP309Cb	S30940	23Cr–12Ni–Cb	Wld. pipe
SA-312	TP309H	S30909	23Cr–12Ni	Wld. pipe
SA-312	TP309S	S30908	23Cr–12Ni	Wld. pipe
SA-312	TP310Cb	S31040	25Cr–20Ni–Cb	Wld. pipe
SA-312	TP310H	S31009	23Cr–12Ni	Wld. pipe
SA-312	TP310MoLN	S31050	25Cr–22Ni–2Mo–N	Wld. pipe
SA-312	TP310MoLN	S31050	25Cr–22Ni–2Mo–N	Wld. pipe

Table 3-A.3
High Alloy Steel (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-312	TP310S	S31008	23Cr–12Ni	Wld. pipe
SA-312	TP316	S31600	16Cr–12Ni–2Mo	Wld. pipe
SA-312	TP316H	S31609	16Cr–12Ni–2Mo	Wld. pipe
SA-312	TP316L	S31603	16Cr–12Ni–2Mo	Wld. pipe
SA-312	TP316N	S31651	16Cr–12Ni–2Mo–N	Wld. pipe
SA-312	TP317	S31700	18Cr–3Ni–3Mo	Wld. pipe
SA-312	TP321	S32100	18Cr–10Ni–Ti	Wld. pipe
SA-312	TP321H	S32109	18Cr–10Ni–Ti	Wld. pipe
SA-312	TP347	S34700	18Cr–10Ni–Cb	Wld. pipe
SA-312	TP347H	S34709	18Cr–10Ni–Cb	Wld. pipe
SA-312	TP348	S34800	18Cr–10Ni–Cb	Wld. pipe
SA-312	TP348H	S34809	18Cr–10Ni–Cb	Wld. pipe
SA-336	FXM-11	S21904	21Cr–6Ni–9Mn	Forgings
SA-336	FXM-19	S20910	22Cr–13Ni–5Mn	Forgings
SA-336	F6	S41000	13Cr	Forgings
SA-336	F304	S30400	18Cr–8Ni	Forgings
SA-336	F304H	S30409	18Cr–8Ni	Forgings
SA-336	F304L	S30403	18Cr–8Ni	Forgings
SA-336	F304N	S30451	18Cr–8Ni–N	Forgings
SA-336	F310	S31000	25Cr–20Ni	Forgings
SA-336	F316	S31600	16Cr–12Ni–2Mo	Forgings
SA-336	F316H	S31609	16Cr–12Ni–2Mo	Forgings
SA-336	F316L	S31603	16Cr–12Ni–2Mo	Forgings
SA-336	F316N	S31651	16Cr–12Ni–2Mo–N	Forgings
SA-336	F321	S32100	18Cr–10Ni–Ti	Forgings
SA-336	F321H	S32109	18Cr–10Ni–Ti	Forgings
SA-336	F347	S34700	18Cr–10Ni–Cb	Forgings
SA-336	F347H	S34709	18Cr–10Ni–Cb	Forgings
SA-351	CF3	J92500	18Cr–8Ni	Castings
SA-351	CF8	J92600	18Cr–8Ni	Castings
SA-351	CF8C	J92710	18Cr–10Ni–Cb	Castings
SA-351	CF8M	J92900	18Cr–12Ni–2Mo	Castings
SA-351	CF10	J92590	19Cr–9Ni–0.5Mo	Castings
SA-351	CH8	J93400	25Cr–12Ni	Castings
SA-351	CH20	J93402	25Cr–12Ni	Castings
SA-351	CK20	J94202	25Cr–20Ni	Castings
SA-376	TP304	S30400	18Cr–8Ni	Smls. pipe
SA-376	TP304H	S30409	18Cr–8Ni	Smls. pipe
SA-376	TP304N	S30451	18Cr–8Ni–N	Smls. pipe
SA-376	TP316	S31600	16Cr–12Ni–2Mo	Smls. pipe
SA-376	TP316H	S31609	16Cr–12Ni–2Mo	Smls. pipe
SA-376	TP316N	S31651	16Cr–12Ni–2Mo–N	Smls. pipe
SA-376	TP321	S32100	18Cr–10Ni–Ti	Smls. pipe
SA-376	TP321	S32100	18Cr–10Ni–Ti	Smls. pipe
SA-376	TP321H	S32109	18Cr–10Ni–Ti	Smls. pipe
SA-376	TP321H	S32109	18Cr–10Ni–Ti	Smls. pipe
SA-376	TP347	S34700	18Cr–10Ni–Cb	Smls. pipe
SA-376	TP347H	S34709	18Cr–10Ni–Cb	Smls. pipe
SA-376	TP348	S34800	18Cr–10Ni–Cb	Smls. pipe
SA-403	XM-19	S20910	22Cr–13Ni–5Mn	Fittings

Table 3-A.3
High Alloy Steel (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-403	304	S30400	18Cr–8Ni	Fittings
SA-403	304H	S30409	18Cr–8Ni	Fittings
SA-403	304L	S30403	18Cr–8Ni	Fittings
SA-403	304N	S30451	18Cr–8Ni–N	Fittings
SA-403	309	S30900	23Cr–12Ni	Fittings
SA-403	310	S31000	25Cr–20Ni	Fittings
SA-403	316	S31600	16Cr–12Ni–2Mo	Fittings
SA-403	316L	S31603	16Cr–12Ni–2Mo	Fittings
SA-403	316N	S31651	16Cr–12Ni–2Mo–N	Fittings
SA-403	317	S31700	18Cr–13Ni–3Mo	Fittings
SA-403	317L	S31700	18Cr–13Ni–3Mo	Fittings
SA-403	321	S32100	18Cr–10Ni–Ti	Fittings
SA-403	321H	S32109	18Cr–10Ni–Ti	Fittings
SA-403	347	S34700	18Cr–10Ni–Cb	Fittings
SA-403	347H	S34709	18Cr–10Ni–Cb	Fittings
SA-403	348	S34800	18Cr–10Ni–Cb	Fittings
SA-403	348H	S34809	18Cr–10Ni–Cb	Fittings
SA-403	XM-19	S20910	22Cr–13Ni–5Mn	Wld. fittings
SA-403	304	S30400	18Cr–8Ni	Wld. fittings
SA-403	304H	S30409	18Cr–8Ni	Wld. fittings
SA-403	304N	S30451	18Cr–8Ni–N	Wld. fittings
SA-403	309	S30900	23Cr–12Ni	Wld. fittings
SA-403	310	S31000	25Cr–20Ni	Wld. fittings
SA-403	316L	S31603	16Cr–12Ni–2Mo	Wld. fittings
SA-403	316N	S31651	16Cr–12Ni–2Mo–N	Wld. fittings
SA-403	321	S32100	18Cr–10Ni–Ti	Wld. fittings
SA-403	321H	S32109	18Cr–10Ni–Ti	Wld. fittings
SA-403	347	S34700	18Cr–10Ni–Cb	Wld. fittings
SA-403	347H	S34709	18Cr–10Ni–Cb	Wld. fittings
SA-403	348	S34800	18Cr–10Ni–Cb	Wld. fittings
SA-403	348H	S34809	18Cr–10Ni–Cb	Wld. fittings
SA-479 [Note (1)]	...	S32906	29Cr–6.5Ni–2Mo–N	Bar
SA-479	XM-19	S20910	22Cr–13Ni–5Mn	Bar
SA-479	309H	S30909	23Cr–12Ni	Bar
SA-564	630	S17400	17Cr–4Ni–4Cu	Bar
SA-666	XM-11	S21904	21Cr–6Ni–9Mn	Plate
SA-688	TP304	S30400	18Cr–8Ni	Wld. tube
SA-688	TP304L	S30403	18Cr–8Ni	Wld. tube
SA-688	TP316	S31600	16Cr–12Ni–2Mo	Wld. tube
SA-688	TP316L	S31603	16Cr–12Ni–2Mo	Wld. tube
SA-693	630	S17400	17Cr–4Ni–4Cu	Plate, sheet, and strip
SA-705	630	S17400	17Cr–4Ni–4Cu	Forgings
SA-789	...	S31500	18Cr–5Ni–3Mo–N	Smls. tube
SA-789	...	S31803	22Cr–5Ni–3Mo–N	Smls. tube
SA-789	...	S31500	18Cr–5Ni–3Mo–N	Wld. tube
SA-789	...	S31803	22Cr–5Ni–3Mo–N	Wld. tube
SA-789 [Note (1)]	...	S32205	22Cr–5.5Ni–3Mo–N	Smls. tube
SA-789 [Note (1)]	...	S32205	22Cr–5.5Ni–3Mo–N	Wld. tube
SA-789 [Note (1)]	...	S32906	29Cr–6.5Ni–2Mo–N	Smls. tube
SA-790 [Note (1)]	...	S32205	22Cr–5.5Ni–3Mo–N	Smls. tube

Table 3-A.3
High Alloy Steel (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-790 [Note (1)]	...	S32205	22Cr–5.5Ni–3Mo–N	Wld. pipe
SA-790 [Note (1)]	...	S32906	29Cr–6.5Ni–2Mo–N	Smls. tube
SA-790	...	S31500	18Cr–5Ni–3Mo–N	Smls. pipe
SA-790	...	S31803	22Cr–5Ni–3Mo–N	Smls. pipe
SA-790	...	S31500	18Cr–5Ni–3Mo–N	Wld. pipe
SA-790	...	S31803	22Cr–5Ni–3Mo–N	Wld. pipe
SA-790	...	S32906	29Cr–6.5Ni–2Mo–N	Smls. pipe
SA-803	26-3-3	S44660	26Cr–3Ni–3Mo	Wld. tube
SA-813	TP309Cb	S30940	23Cr–12Ni–Cb	Wld. pipe
SA-813	TP309S	S30908	23Cr–12Ni	Wld. pipe
SA-813	TP310Cb	S31040	25Cr–20Ni–Cb	Wld. pipe
SA-813	TP310S	S31008	25Cr–20Ni	Wld. pipe
SA-814	TP309Cb	S30940	23Cr–12Ni–Cb	Wld. pipe
SA-814	TP309S	S30908	23Cr–12Ni	Wld. pipe
SA-814	TP310Cb	S31040	25Cr–20Ni–Cb	Wld. pipe
SA-814	TP310S	S31008	25Cr–20Ni	Wld. pipe
SA-965	FXM-11	S21904	21Cr–6Ni–9Mn	Forgings
SA-965	FXM-19	S20910	22Cr–13Ni–5Mn	Forgings
SA-965	F6	S41000	13Cr	Forgings
SA-965	F304	S30400	18Cr–8Ni	Forgings
SA-965	F304H	S30409	18Cr–8Ni	Forgings
SA-965 [Note (1)]	F304L	S30403	18Cr–8Ni	Forgings
SA-965	F304N	S30451	18Cr–8Ni–N	Forgings
SA-965	F310	S31000	25Cr–20Ni	Forgings
SA-965	F316	S31600	16Cr–12Ni–2Mo	Forgings
SA-965	F316H	S31609	16Cr–12Ni–2Mo	Forgings
SA-965	F316L	S31603	16Cr–12Ni–2Mo	Forgings
SA-965	F316N	S31651	16Cr–12Ni–2Mo–N	Forgings
SA-965	F321	S32100	18Cr–10Ni–Ti	Forgings
SA-965	F321H	S32109	18Cr–10Ni–Ti	Forgings
SA-965	F347	S34700	18Cr–10Ni–Cb	Forgings
SA-965	F347H	S34909	18Cr–10Ni–Cb	Forgings
SA/EN 10028-7 [Note (2)]	X2CrNi18-9	...	18Cr– 8Ni	Plate and strip
SA/EN 10028-7 [Note (2)]	X2CrNiMo17-12-2	...	16Cr–12Ni–2Mo	Plate and strip
SA/EN 10028-7 [Note (2)]	X2CrNiMoN17-11-2	...	16Cr–12Ni–2Mo–N	Plate and strip
SA/EN 10028-7 [Note (2)]	X2CrNiMoN17-13-3	...	16Cr–12Ni–2Mo–N	Plate and strip
SA/EN 10028-7 [Note (2)]	X2CrNiN18-10	...	18Cr–8Ni–N	Plate and strip
SA/EN 10028-7 [Note (2)]	X5CrNi18-10	...	18Cr–8Ni	Plate and strip
SA/EN 10028-7 [Note (2)]	X5CrNiMo17-12-2	...	16Cr–12Ni–2Mo	Plate and strip
SA/EN 10028-7 [Note (2)]	X5CrNiN19-9	...	18Cr–8Ni–N	Plate and strip
SA/EN 10028-7 [Note (2)]	X6CrNiTi18-10	...	18Cr-8Ni-Ti	Plate and strip

NOTES:
(1) For Class 2 construction only.
(2) SA/EN 10028-7 materials shall be considered as SA-240 materials with the following corresponding grades:
(a) Grade X2CrNi18-9 shall be considered as Type 304L.
(b) Grade X2CrNiMo17-12-2 shall be considered as Type 316L.
(c) Grades X2CrNiMoN17-11-2 and X2CrNiMoN17-13-3 shall be considered as Type 316LN.
(d) Grade X2CrNiN18-10 shall be considered as Type 304LN.
(e) Grade X5CrNi18-10 shall be considered as Type 304.
(f) Grade X5CrNiMo17-12-2 shall be considered as Type 316.
(g) Grade X5CrNiN19-9 shall be considered as Type 304N.
(h) Grade X6CrNiTi18-10 shall be considered as Type 321.

Table 3-A.4
Aluminum Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-209	3003	A93003	Al–Mn–Cu	Plate, sheet
SB-209	3004	A93004	Al–Mn–Mg	Plate, sheet
SB-209	5052	A95052	Al–2.5Mg	Plate, sheet
SB-209	5083	A95083	Al–4.4Mg–Mn	Plate, sheet
SB-209	5086	A95086	Al–4.0Mg–Mn	Plate, sheet
SB-209	5454	A95454	Al–2.7Mg–Mn	Plate, sheet
SB-209	6061	A96061	Al–Mg–Si–Cu	Plate, sheet
SB-210	Allclad 3003	...	Al–Mn–Cu	Smls. drawn tube
SB-210	3003	A93003	Al–Mn–Cu	Smls. drawn tube
SB-210	6061	A96061	Al–Mg–Si–Cu	Smls. drawn tube
SB-210	6063	A96063	Al–Mg–Si	Smls. drawn tube
SB-221	3003	A93003	Al–Mn–Cu	Bar, rod, shapes
SB-221	5083	A95083	Al–4.4Mg–Mn	Bar, rod, shapes
SB-221	5454	A95454	Al–2.7Mg–Mn	Bar, rod, shapes
SB-221	6061	A96061	Al–Mg–Si–Cu	Bar, rod, shapes
SB-221	6063	A96063	Al–Mg–Si	Bar, rod, shapes
SB-241	Allclad 3003	...	Al–Mn–Cu	Smls. extr. tube
SB-241	3003	A93003	Al–Mn–Cu	Smls. extr. tube
SB-241	3003	A93003	Al–Mn–Cu	Smls. pipe
SB-241	5083	A95083	Al–4.4Mg–Mn	Smls. extr. tube
SB-241	5454	A95454	Al–2.7Mg–Mn	Smls. extr. tube
SB-241	6061	A96061	Al–Mg–Si–Cu	Smls. extr. tube/pipe
SB-241	6061	A96061	Al–Mg–Si–Cu	Smls. drawn pipe
SB-241	6063	A96063	Al–Mg–Si	Smls. extr. tube/pipe
SB-308	6061	A96061	Al–Mg–Si–Cu	Shapes

Table 3-A.5
Copper Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-96	...	C65500	97Cu–3.3Si	Plate, sheet
SB-98	...	C65100	98.5Cu–1.5Si	Rod, bar & shapes
SB-98	...	C65500	97Cu–3Si	Rod, bar & shapes
SB-98	...	C66100	94Cu–3Si	Rod, bar & shapes
SB-111	...	C28000	60Cu–20Zn	Smls. tube
SB-111	...	C44300	71Cu–28Zn–1Sn–0.06As	Smls. tube
SB-111	...	C44400	71Cu–28Zn–1Sn–0.06Sb	Smls. tube
SB-111	...	C44500	71Cu–28Zn–1Sn–0.06P	Smls. tube
SB-111	...	C60800	95Cu–5Al	Smls. tube
SB-111	...	C70600	90Cu–10Ni	Cond. tube
SB-111	...	C71500	70Cu–30Ni	Cond. tube
SB-169	...	C61400	90Cu–7Al–3Fe	Plate, sheet
SB-171	...	C46400	60Cu–39Zn–Sn	Plate
SB-171	...	C70600	90Cu–10Ni	Plate
SB-171	...	C71500	70Cu–30Ni	Plate
SB-187	...	C10200	99.95Cu–P	Rod & bar
SB-187	...	C11000	99.9Cu	Rod & bar
SB-395	...	C70600	90Cu–10Ni	Smls. U–bend tube
SB-395	...	C71500	70Cu–30Ni	Smls. U–bend tube

Table 3-A.6
Nickel and Nickel Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-127	...	N04400	67Ni–30Cu	Plate
SB-160	...	N02200	99Ni	Bar, rod
SB-160	...	N02201	99Ni–Low C	Bar, rod
SB-161	...	N02200	99Ni	Smls. pipe & tube
SB-161	...	N02201	99Ni–Low C	Smls. pipe & tube
SB-162	...	N02200	99Ni	Plate, sheet, strip
SB-162	...	N02201	99Ni–Low C	Plate, sheet, strip
SB-163	...	N02200	99Ni	Smls. tube
SB-163	...	N02201	99Ni–Low C	Smls. tube
SB-163	...	N04400	67Ni–30Cu	Smls. tube
SB-163	...	N06600	72Ni–15Cr–8Fe	Smls. tube
SB-163	...	N08800	33Ni–42Fe–21Cr	Smls. tube
SB-163	...	N08810	33Ni–42Fe–21Cr	Smls. tube
SB-163	...	N08825	42Ni–21.5Cr–3Mo–2.3Cu	Smls. tube
SB-164	...	N04400	67Ni–30Cu	Bar, rod
SB-164	...	N04405	67Ni–30Cu–S	Bar, rod
SB-165	...	N04400	67Ni–30Cu	Smls. pipe & tube
SB-166	...	N06600	72Ni–15Cr–8Fe	Bar, rod
SB-167	...	N06600	72Ni–15Cr–8Fe	Smls. pipe & tube
SB-168	...	N06600	72Ni–15Cr–8Fe	Plate
SB-333	...	N10001	62Ni–28Mo–5Fe	Plate, strip
SB-333	...	N10665	65Ni–28Mo–2Fe	Plate, strip
SB-335	...	N10001	62Ni–28Mo–5Fe	Rod
SB-335	...	N10665	65Ni–28Mo–2Fe	Rod
SB-366	...	N06022	55Ni–21Cr–13.5Mo	Smls. & wld. fittings
SB-366	...	N06059	59Ni–23Cr–16Mo	Smls. fittings
SB-366	...	N10276	54Ni–16Mo–15Cr	Smls. fittings
SB-366	...	N10665	65Ni–28Mo–2Fe	Smls. fittings
SB-366	...	N06059	59Ni–23Cr–16Mo	Wld. fittings
SB-366	...	N10276	54Ni–16Mo–15Cr	Wld. fittings
SB-366	...	N10665	65Ni–28Mo–2Fe	Wld. fittings
SB-407	...	N08800	33Ni–42Fe–21Cr	Smls. pipe & tube
SB-407	...	N08810	33Ni–42Fe–21Cr	Smls. pipe & tube
SB-408	...	N08800	33Ni–42Fe–21Cr	Bar, rod
SB-408	...	N08810	33Ni–42Fe–21Cr	Bar, rod
SB-409	...	N08800	33Ni–42Fe–21Cr	Plate
SB-409	...	N08810	33Ni–42Fe–21Cr	Plate
SB-423	...	N08825	42Ni–21.5Cr–3Mo–2.3Cu	Smls. pipe & tube
SB-424	...	N08825	42Ni–21.5Cr–3Mo–2.3Cu	Plate, sheet, strip
SB-425	...	N08825	42Ni–21.5Cr–3Mo–2.3Cu	Bar, rod
SB-434	...	N10003	70Ni–16Mo–7Cr–5Fe	Plate, sheet, strip
SB-435	...	N06002	47Ni–22Cr–9Mo–18Fe	Sheet
SB-435	...	N06002	47Ni–22Cr–9Mo–18Fe	Plate
SB-462	...	N06022	55Ni–21Cr–13.5Mo	Forgings
SB-462	...	N06059	59Ni–23Cr–16Mo	Forgings
SB-462	...	N10276	54Ni–16Mo–15Cr	Forgings
SB-462	...	N10665	65Ni–28Mo–2Fe	Forgings
SB-511	...	N08330	35Ni–19Cr–1.25Si	Bar
SB-514	...	N08800	33Ni–42Fe–21Cr	Welded pipe
SB-514	...	N08810	33Ni–42Fe–21Cr	Welded pipe
SB-515	...	N08800	33Ni–42Fe–21Cr	Welded tube
SB-515	...	N08810	33Ni–42Fe–21Cr	Welded tube

Table 3-A.6
Nickel and Nickel Alloys (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-516	...	N06600	72Ni–15Cr–8Fe	Welded tube
SB-517	...	N06600	72Ni–15Cr–8Fe	Welded tube
SB-535	...	N08330	35Ni–19Cr–1$^1/_4$Si	Smls. & welded pipe
SB-536	...	N08330	35Ni–19Cr–1$^1/_4$Si	Plate, sheet, strip
SB-564	...	N04400	67Ni–30Cu	Forgings
SB-564	...	N06022	55Ni–21Cr–13.5Mo	Forgings
SB-564	...	N06059	59Ni–23Cr–16Mo	Forgings
SB-564	...	N06600	72Ni–15Cr–8Fe	Forgings
SB-564	...	N08800	33Ni–42Fe–21Cr	Forgings
SB-564	...	N08810	33Ni–42Fe–21Cr	Forgings
SB-564 [Note (1)]	...	N08825	42Ni–21.5Cr–3Mo–2.3Cu	Forgings
SB-572	...	N06002	47Ni–22Cr–9Mo–18Fe	Rod
SB-573	...	N10003	70Ni–16Mo–7Cr–5Fe	Rod
SB-574	...	N06022	55Ni–21Cr–13.5Mo	Rod
SB-574	...	N06059	59Ni–23Cr–16Mo	Rod
SB-574	...	N06455	61Ni–16Mo–16Cr	Rod
SB-574	...	N10276	54Ni–16Mo–15Cr	Rod
SB-575	...	N06022	55Ni–21Cr–13.5Mo	Plate, sheet & strip
SB-575	...	N06059	59Ni–23Cr–16Mo	Plate, sheet & strip
SB-575	...	N06455	61Ni–16Mo–16Cr	Plate, sheet & strip
SB-575	...	N10276	54Ni–16Mo–15Cr	Plate, sheet & strip
SB-581	...	N06007	47Ni–22Cr–19Fe–6Mo	Rod
SB-582	...	N06007	47Ni–22Cr–19Fe–6Mo	Plate, sheet, strip
SB-619	...	N06002	47Ni–22Cr–9Mo–18Fe	Welded pipe
SB-619	...	N06007	47Ni–22Cr–19Fe–6Mo	Welded pipe
SB-619	...	N06022	55Ni–21Cr–13.5Mo	Welded pipe
SB-619	...	N06059	59Ni–23Cr–16Mo	Welded pipe
SB-619	...	N06455	61Ni–16Mo–16Cr	Welded pipe
SB-619	...	N10001	62Ni–28Mo–5Fe	Welded pipe
SB-619	...	N10276	54Ni–16Cr–16Mo–5.5Fe	Welded pipe
SB-619	...	N10665	65Ni–28Mo–2Fe	Welded pipe
SB-622	...	N06002	47Ni–22Cr–9Mo–18Fe	Smls. pipe & tube
SB-622	...	N06007	47Ni–22Cr–19Fe–6Mo	Smls. pipe & tube
SB-622	...	N06022	55Ni–21Cr–13.5Mo	Smls. pipe & tube
SB-622	...	N06059	59Ni–23Cr–16Mo	Smls. pipe & tube
SB-622	...	N06455	61Ni–16Mo–16Cr	Smls. pipe & tube
SB-622	...	N10001	62Ni–28Mo–5Fe	Smls. pipe & tube
SB-622	...	N10276	54Ni–16Cr–16Mo–5.5Fe	Smls. pipe & tube
SB-622	...	N10665	65Ni–28Mo–2Fe	Smls. pipe & tube
SB-626	...	N06002	47Ni–22Cr–9Mo–18Fe	Welded tube
SB-626	...	N06007	47Ni–22Cr–19Fe–6Mo	Welded tube
SB-626	...	N06022	55Ni–21Cr–13.5Mo	Welded tube
SB-626	...	N06059	59Ni–23Cr–16Mo	Welded tube
SB-626	...	N06455	61Ni–16Mo–16Cr	Welded tube
SB-626	...	N10001	62Ni–28Mo–5Fe	Welded tube
SB-626	...	N10276	54Ni–16Cr–16Mo–5.5Fe	Welded tube
SB-626	...	N10665	65Ni–28Mo–2Fe	Welded tube

NOTE:
(1) For Class 2 construction only.

Table 3-A.7
Titanium and Titanium Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-265	1	R50250	Ti	Plate, sheet, strip
SB-265	2	R50400	Ti	Plate, sheet, strip
SB-265	3	R50550	Ti	Plate, sheet, strip
SB-265	7	R52400	Ti–Pd	Plate, sheet, strip
SB-265	16	R52402	Ti–Pd	Plate, sheet, strip
SB-265	12	R53400	Ti–0.3Mo–0.8Ni	Plate, sheet, strip
SB-338	1	R50250	Ti	Smls. tube
SB-338	2	R50400	Ti	Smls. tube
SB-338	3	R50550	Ti	Smls. tube
SB-338	7	R52400	Ti–Pd	Smls. tube
SB-338 [Note (1)]	16	R52402	Ti–Pd	Smls. tube
SB-338	12	R53400	Ti–0.3Mo–0.8Ni	Smls. tube
SB-338	1	R50250	Ti	Wld. tube
SB-338	2	R50400	Ti	Wld. tube
SB-338	3	R50550	Ti	Wld. tube
SB-338	7	R52400	Ti–Pd	Wld. tube
SB-338 [Note (1)]	16	R52402	Ti–Pd	Wld. tube
SB-338	12	R53400	Ti–0.3Mo–0.8Ni	Wld. tube
SB-348	1	R50250	Ti	Bar, billet
SB-348	2	R50400	Ti	Bar, billet
SB-348	3	R50550	Ti	Bar, billet
SB-348	7	R52400	Ti–Pd	Bar, billet
SB-348 [Note (1)]	16	R52402	Ti–Pd	Bar, billet
SB-348	12	R53400	Ti–0.3Mo–0.8Ni	Bar, billet
SB-381	F1	R50250	Ti	Forgings
SB-381	F2	R50400	Ti	Forgings
SB-381	F3	R50550	Ti	Forgings
SB-381	F7	R52400	Ti–Pd	Forgings
SB-381 [Note (1)]	F16	R52402	Ti–Pd	Forgings
SB-381	F12	R53400	Ti–0.3Mo–0.8Ni	Forgings
SB-861	1	R50250	Ti	Smls. pipe
SB-861	2	R50400	Ti	Smls. pipe
SB-861	3	R50550	Ti	Smls. pipe
SB-861	7	R52400	Ti–Pd	Smls. pipe
SB-861	12	R53400	Ti–0.3Mo–0.8Ni	Smls. pipe
SB-862	1	R50250	Ti	Wld. pipe
SB-862	2	R50400	Ti	Wld. pipe
SB-862	3	R50550	Ti	Wld. pipe
SB-862	7	R52400	Ti–Pd	Wld. pipe
SB-862	12	R53400	Ti–0.3Mo–0.8Ni	Wld. pipe

NOTE:
(1) For Class 2 construction only.

Table 3-A.8
Ferrous Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
Low Alloy Steel Bolts				
SA-193	B5	K50100	5Cr –$^1/_2$Mo	Bolting
SA-193	B7	G41400	1Cr–$^1/_5$Mo	Bolting
SA-193	B7M	G41400	1Cr–$^1/_5$Mo	Bolting
SA-193	B16	K14072	1Cr–$^1/_2$Mo–V	Bolting
SA-320	L7	G41400	1Cr–$^1/_5$Mo	Bolting
SA-320	L7A	G40370	C–$^1/_4$Mo	Bolting
SA-320	L7M	G41400	1Cr–$^1/_5$Mo	Bolting
SA-320	L43	G43400	1$^3/_4$Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting
SA-325	1	K02706	Carbon steel	Bolting
SA-354	BC	K04100	Carbon steel	Bolting
SA-354	BD	K04100	Carbon steel	Bolting
SA-437	B4B	K91352	12Cr–1Mo–V–W	Bolting
SA-437	B4C	K91352	12Cr–1Mo–V–W	Bolting
SA-449	...	K04200	Carbon steel	Bolting
SA-449	...	K04200	Carbon steel	Bolting
SA-449	...	K04200	Carbon steel	Bolting
SA-508	5, Cl. 2	K42365	3$^1/_2$Ni–1$^3/_4$Cr–$^1/_2$Mo–V	Bolting
SA-540	B21, Cl. 1	K14073	1Cr–$^1/_2$Mo–V	Bolting
SA-540	B21, Cl. 2	K14073	1Cr–$^1/_2$Mo–V	Bolting
SA-540	B21, Cl. 3	K14073	1Cr–$^1/_2$Mo–V	Bolting
SA-540	B21, Cl. 4	K14073	1Cr–$^1/_2$Mo–V	Bolting
SA-540	B21, Cl. 5	K14073	1Cr–$^1/_2$Mo–V	Bolting
SA-540	B23, Cl. 1	H43400	2Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting
SA-540	B23, Cl. 2	H43400	2Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting
SA-540	B23, Cl. 3	H43400	2Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting
SA-540	B23, Cl. 4	H43400	2Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting
SA-540	B23, Cl. 5	H43400	2Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting
SA-540	B24, Cl. 1	K24064	2Ni–$^3/_4$Cr–$^1/_3$Mo	Bolting
SA-540	B24, Cl. 2	K24064	2Ni–$^3/_4$Cr–$^1/_3$Mo	Bolting
SA-540	B24, Cl. 3	K24064	2Ni–$^3/_4$Cr–$^1/_3$Mo	Bolting
SA-540	B24, Cl. 4	K24064	2Ni–$^3/_4$Cr–$^1/_3$Mo	Bolting
SA-540	B24, Cl. 5	K24064	2Ni–$^3/_4$Cr–$^1/_3$Mo	Bolting
SA-540	B24V, Cl. 3	K24070	2Ni–$^3/_4$Cr–$^1/_3$Mo–V	Bolting
Low Alloy Steel Nuts				
SA-194	2	...	...	Nuts
SA-194	2H	...	...	Nuts
SA-194	2HM	...	...	Nuts
SA-194	3	...	...	Nuts
SA-194	4	...	...	Nuts
SA-194	7	...	...	Nuts
SA-194	7M	...	...	Nuts
SA-194	16	...	...	Nuts
SA-540	B21	...	...	Nuts
SA-540	B23	...	...	Nuts
SA-540	B24	...	...	Nuts
SA-540	B24V	...	...	Nuts
High Alloy Steel Bolts				
SA-193	B6	S41000	13Cr	Bolting
SA-193	B8, Cl. 1	S30400	18Cr–8Ni	Bolting
SA-193	B8, Cl. 2	S30400	18Cr–8Ni	Bolting

Table 3-A.8
Ferrous Bolting Materials for Design in Accordance With Part 4 (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
High Alloy Steel Bolts (Cont'd)				
SA-193	B8C, Cl. 1	S34700	18Cr–10Ni–Cb	Bolting
SA-193	B8C, Cl. 2	S34700	18Cr–10Ni–Cb	Bolting
SA-193	B8M, Cl. 1	S31600	16Cr–12Ni–2Mo	Bolting
SA-193	B8M2	S31600	16Cr–12Ni–2Mo	Bolting
SA-193	B8M2	S31600	16Cr–12Ni–2Mo	Bolting
SA-193	B8M2	S31600	16Cr–12Ni–2Mo	Bolting
SA-193	B8MNA, Cl. 1A	S31651	16Cr–12Ni–2Mo–N	Bolting
SA-193	B8NA, Cl. 1A	S30451	18Cr–8Ni–N	Bolting
SA-193	B8P, Cl. 1	S30500	18Cr–11Ni	Bolting
SA-193	B8P, Cl. 2	S30500	18Cr–11Ni	Bolting
SA-193	B8S	S21800	18Cr–8Ni–4Si–N	Bolting
SA-193	B8SA	S21800	18Cr–8Ni–4Si–N	Bolting
SA-193	B8T, Cl. 1	S32100	18Cr–10Ni–Ti	Bolting
SA-193	B8T, Cl. 2	S32100	18Cr–10Ni–Ti	Bolting
SA-320	B8, Cl. 1	S30400	18Cr–8Ni	Bolting
SA-320	B8, Cl. 2	S30400	18Cr–8Ni	Bolting
SA-320	B8A, Cl. 1A	S30400	18Cr–8Ni	Bolting
SA-320	B8C, Cl. 1	S34700	18Cr–10Ni–Cb	Bolting
SA-320	B8C, Cl. 2	S34700	18Cr–10Ni–Cb	Bolting
SA-320	B8CA, Cl. 1A	S34700	18Cr–10Ni–Cb	Bolting
SA-320	B8F, Cl. 1	S30323	18Cr–8Ni–S	Bolting
SA-320	B8FA, Cl. 1A	S30323	18Cr–8Ni–S	Bolting
SA-320	B8M, Cl. 1	S31600	16Cr–12Ni–2Mo	Bolting
SA-320	B8M, Cl. 2	S31600	16Cr–12Ni–2Mo	Bolting
SA-320	B8MA, Cl. 1A	S31600	16Cr–12Ni–2Mo	Bolting
SA-320	B8T, Cl. 1	S32100	18Cr–10Ni–Ti	Bolting
SA-320	B8T, CL. 2	S32100	18Cr–10Ni–Ti	Bolting
SA-320	B8TA, Cl. 1A	S32100	18Cr–10Ni–Ti	Bolting
SA-453	651, Cl. A	S63198	19Cr–9Ni–Mo–W	Bolting
SA-453	651, Cl. B	S63198	19Cr–9Ni–Mo–W	Bolting
SA-453	660, Cl. A	S66286	25Ni–15Cr–2Ti	Bolting
SA-453	660, Cl. B	S66286	25Ni–15Cr–2Ti	Bolting
SA-479	XM–19	S20910	22Cr–13Ni–5Mn	Bolting
SA-564	630	S17400	17Cr–4Ni–4Cu	Bolting
SA-705	630	S17400	17Cr–4Ni–4Cu	Bolting

Table 3-A.9
Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-211	2014	A92014	...	Bolting
SB-211	2024	A92024	...	Bolting
SB-211	6061	A96061	...	Bolting
SB-98	...	C65100	98.5Cu–1.5Si	Rod
SB-98	...	C65500	97Cu–3Si	Rod
SB-98	...	C66100	94Cu–3Si	Rod
SB-150	...	C61400	90Cu–7Al–3Fe	Bar,
SB-150	...	C61400	90Cu–7Al–3Fe	Rod

Table 3-A.9
Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4 (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-150	...	C62300	81Cu–10Al–5Ni–3Fe	Bar
SB-150	...	C63000	81Cu–10Al–5Ni–3Fe	Rod
SB-150	...	C63000	81Cu–10Al–5Ni–3Fe	Bar
SB-150	...	C64200	91Cu–7Al–2Si	Bar
SB-150	...	C64200	91Cu–7Al–2Si	Rod
SB-187	...	C10200	99.95Cu–P	Rod
SB-187	...	C11000	99.9Cu	Rod

Table 3-A.10
Nickel and Nickel Alloy Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-160	...	N02200	99 Ni	Bolting
SB-160	...	N02201	99Ni–Low C	Bolting
SB-164	...	N04400	67Ni–30Cu	Bolting
SB-164	...	N04405	67Ni–30Cu	Bolting
SB-166	...	N06600	72Ni–15Cr–8Fe	Bolting
SB-335	...	N10001	62Ni–28Mo–5Fe	Bolting
SB-335	...	N10665	65Ni–28Mo–2Fe	Bolting
SB-408	...	N08800	33Ni–42Fe–21Cr	Bolting
SB-408	...	N08810	33Ni–42Fe–21Cr	Bolting
SB-425	...	N08825	42Ni–21.5Cr–3Mo–2.3Cu	Bolting
SB-446	1	N06625	60Ni–22Cr–9Mo–3.5Cb	Bolting
SB-572	...	N06002	47Ni–22Cr–9Mo–18Fe	Bolting
SB-572	...	R30556	21Ni–30Fe–22Cr–18Co–3Mo–3W	Bolting
SB-573	...	N10003	70Ni–16Mo–7Cr–5Fe	Bolting
SB-574	...	N06022	55Ni–21Cr–13.5Mo	Bolting
SB-574	...	N06455	61Ni–16Mo–16Cr	Bolting
SB-574	...	N10276	54Ni–16Mo–15Cr	Bolting
SB-581	...	N06007	47Ni–22Cr–19Fe–6Mo	Bolting
SB-581	...	N06030	40Ni–29Cr–15Fe–5Mo	Bolting
SB-581	...	N06975	49Ni–25Cr–18Fe–6Mo	Bolting
SB-621	...	N08320	26Ni–43Fe–22Cr–5Mo	Bolting
SB-637	...	N07718	53Ni–19Cr–19Fe–Cb–Mo	Bolting
SB-637	2	N07750	70Ni–16Cr–7Fe–Ti–Al	Bolting

Table 3-A.11
Bolting Materials for Design in Accordance With Part 5

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
		Low Alloy Steel Bolts		
SA-193	B5	K50100	5Cr –$^1/_2$Mo	Bolting
SA-193	B7	G41400	1Cr–$^1/_5$Mo	Bolting
SA-193	B7M	G41400	1Cr–$^1/_5$Mo	Bolting
SA-193	B16	K14072	1Cr–$^1/_2$Mo–V	Bolting
SA-320	L43	G43400	1$^3/_4$Ni–$^3/_4$Cr–$^1/_4$Mo	Bolting

Table 3-A.11
Bolting Materials for Design in Accordance With Part 5 (Cont'd)

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
Low Alloy Steel Bolts (Cont'd)				
SA-437	B4B	K91352	12Cr–1Mo–V–W	Bolting
SA-437	B4C	K91352	12Cr–1Mo–V–W	Bolting
SA-540	B21 Cl. 1	K14073	1Cr–$\frac{1}{2}$Mo–V	Bolting
SA-540	B21 Cl. 2	K14073	1Cr–$\frac{1}{2}$Mo–V	Bolting
SA-540	B21 Cl. 3	K14073	1Cr–$\frac{1}{2}$Mo–V	Bolting
SA-540	B21 Cl. 4	K14073	1Cr–$\frac{1}{2}$Mo–V	Bolting
SA-540	B21 Cl. 5	K14073	1Cr–$\frac{1}{2}$Mo–V	Bolting
SA-540	B22 Cl. 1	K41420	1Cr–1Mn–$\frac{1}{4}$Mo	Bolting
SA-540	B22 Cl. 2	K41420	1Cr–1Mn–$\frac{1}{4}$Mo	Bolting
SA-540	B22 Cl. 3	K41420	1Cr–1Mn–$\frac{1}{4}$Mo	Bolting
SA-540	B22 Cl. 4	K41420	1Cr–1Mn–$\frac{1}{4}$Mo	Bolting
SA-540	B22 Cl. 5	K41420	1Cr–1Mn–$\frac{1}{4}$Mo	Bolting
SA-540	B23 Cl. 1	H43400	2Ni–$\frac{3}{4}$Cr–$\frac{1}{4}$Mo	Bolting
SA-540	B23 Cl. 2	H43400	2Ni–$\frac{3}{4}$Cr–$\frac{1}{4}$Mo	Bolting
SA-540	B23 Cl. 3	H43400	2Ni–$\frac{3}{4}$Cr–$\frac{1}{4}$Mo	Bolting
SA-540	B23 Cl. 4	H43400	2Ni–$\frac{3}{4}$Cr–$\frac{1}{4}$Mo	Bolting
SA-540	B23 Cl. 5	H43400	2Ni–$\frac{3}{4}$Cr–$\frac{1}{4}$Mo	Bolting
SA-540	B24 Cl. 1	K24064	2Ni–$\frac{3}{4}$Cr–$\frac{1}{3}$Mo	Bolting
SA-540	B24 Cl. 2	K24064	2Ni–$\frac{3}{4}$Cr–$\frac{1}{3}$Mo	Bolting
SA-540	B24 Cl. 3	K24064	2Ni–$\frac{3}{4}$Cr–$\frac{1}{3}$Mo	Bolting
SA-540	B24 Cl. 4	K24064	2Ni–$\frac{3}{4}$Cr–$\frac{1}{3}$Mo	Bolting
SA-540	B24 Cl. 5	K24064	2Ni–$\frac{3}{4}$Cr–$\frac{1}{3}$Mo	Bolting
SA-540	B24V Cl. 3	K24070	2Ni–$\frac{3}{4}$Cr–$\frac{1}{3}$Mo–V	Bolting
High Alloy Steel Bolts				
SA-193	B6	S41000	13Cr	Bolting
SA-193	B8 Cl. 1	S30400	18Cr–8Ni	Bolting
SA-193	B8C Cl. 1	S34700	18Cr–10Ni–Cb	Bolting
SA-193	B8M Cl. 1	S31600	16Cr–12Ni–2Mo	Bolting
SA-193	B8MNA Cl. 1A	S31651	16Cr–12Ni–2Mo–N	Bolting
SA-193	B8NA Cl. 1A	S30451	18Cr–8Ni–N	Bolting
SA-193	B8S	S21800	18Cr–8Ni–4Si–N	Bolting
SA-193	B8SA	S21800	18Cr–8Ni–4Si–N	Bolting
SA-193	B8T Cl. 1	S32100	18Cr–10Ni–Ti	Bolting
SA-193	B8R, Cl. 1C	S20910	22Cr–13Ni–5Mn	Bolting
SA-193	B8RA	S20910	22Cr–13Ni–5Mn	Bolting
SA-453	651 Cl. A	S63198	19Cr–9Ni–Mo–W	Bolting
SA-453	651 Cl. B	S63198	19Cr–9Ni–Mo–W	Bolting
SA-453	660 Cl. A	S66286	25Ni–15Cr–2Ti	Bolting
SA-453	660 Cl. B	S66286	25Ni–15Cr–2Ti	Bolting
SA-564	630	S17400	17Cr–4Ni–4Cu	Bolting
SA-564	Temper H1100	S17400	17Cr–4Ni–4Cu	Bolting
SA-705	630	S17400	17Cr–4Ni–4Cu	Bolting
SA-705	Temper H1100	S17400	17Cr–4Ni–4Cu	Bolting
Nickel Alloy Bolts				
SB-164	...	N04400	67Ni–30Cu	Bolting
SB-164	...	N04405	67Ni–30Cu–S	Bolting
SB-637	...	N07718	53Ni–19Cr–19Fe–Cb–Mo	Bolting
SB-637	2	N07750	70Ni–16Cr–7Fe–Ti–Al	Bolting

ANNEX 3-B
REQUIREMENTS FOR MATERIAL PROCUREMENT

(Currently Not Used)

ANNEX 3-C ISO MATERIAL GROUP NUMBERS

(21)

DELETED

ANNEX 3-D STRENGTH PARAMETERS

(Normative)

(21) 3-D.1 YIELD STRENGTH

Values for the yield strength as a function of temperature are provided in Section II, Part D, Subpart 1, Table Y-1.

If the material being used is not listed in Table Y-1, while being listed in other tables of Section II, Part D, Subpart 1, or the specified temperature exceeds the highest temperature for which a value is provided, the yield strength may be determined as in (a) and (b) for use in the design equations in Part 4. S is the maximum allowable stress of the material at the temperature specified [see Annex 3-A] and f is the factor (e.g., weld factor) used to determine the allowable stress as indicated in the notes for the stress line. If the value of f is not provided, set f equal to 1.

(a) If the allowable design stress is established based on the 66$^2/_3$% yield criterion, then the yield strength, S_Y, shall be taken as $1.5S/f$.

(b) If the allowable design stress is established based on yield criterion between 66$^2/_3$% and 90%, then the yield strength, S_Y, shall be taken as $1.1S/f$.

NOTE: For temperatures where the allowable stress, S, is based on time dependent properties, the yield strength obtained by these formulas may be overly conservative .

3-D.2 ULTIMATE TENSILE STRENGTH

Values for the ultimate tensile strength as a function of temperature are provided in Section II, Part D, Subpart 1, Table U.

(21) 3-D.3 STRESS STRAIN CURVE

The following model for the stress–strain curve shall be used in design calculations where required by this Division when the strain hardening characteristics of the stress–strain curve are to be considered. The yield strength and ultimate tensile strength in 3-D.1 and 3-D.2 may be used in this model to determine a stress–strain curve at a specified temperature.

$$\varepsilon_t = \frac{\sigma_t}{E_y} + \gamma_1 + \gamma_2 \tag{3-D.1}$$

When $\gamma_1 + \gamma_2 \le \varepsilon_p$, eq. (3-D.1) shall be reduced to

$$\varepsilon_t = \frac{\sigma_t}{E_y} \tag{3-D.2}$$

where

$$\gamma_1 = \frac{\varepsilon_1}{2}\left(1.0 - \tanh[H]\right) \tag{3-D.3}$$

$$\gamma_2 = \frac{\varepsilon_2}{2}\left(1.0 + \tanh[H]\right) \tag{3-D.4}$$

$$\varepsilon_1 = \left(\frac{\sigma_t}{A_1}\right)^{\frac{1}{m_1}} \tag{3-D.5}$$

$$A_1 = \frac{\sigma_{ys}\left(1+\varepsilon_{ys}\right)}{\left(\ln\left[1+\varepsilon_{ys}\right]\right)^{m_1}} \tag{3-D.6}$$

$$m_1 = \frac{\ln\left[R\right] + \left(\varepsilon_p - \varepsilon_{ys}\right)}{\ln\left[\frac{\ln\left[1+\varepsilon_p\right]}{\ln\left[1+\varepsilon_{ys}\right]}\right]} \tag{3-D.7}$$

$$\varepsilon_2 = \left(\frac{\sigma_t}{A_2}\right)^{\frac{1}{m_2}} \tag{3-D.8}$$

$$A_2 = \frac{\sigma_{uts}\exp\left[m_2\right]}{{m_2}^{m_2}} \tag{3-D.9}$$

$$H = \frac{2\left[\sigma_t - \left(\sigma_{ys} + K\left\{\sigma_{uts} - \sigma_{ys}\right\}\right)\right]}{K\left(\sigma_{uts} - \sigma_{ys}\right)} \tag{3-D.10}$$

$$R = \frac{\sigma_{ys}}{\sigma_{uts}} \tag{3-D.11}$$

$$\varepsilon_{ys} = 0.002 \tag{3-D.12}$$

$$K = 1.5R^{1.5} - 0.5R^{2.5} - R^{3.5} \tag{3-D.13}$$

The parameters m_2, and ε_p are provided in Table 3-D.1. The development of the stress strain curve should be limited to a value of true ultimate tensile stress at true ultimate tensile strain. The stress strain curve beyond this point should be perfectly plastic. The value of true ultimate tensile stress at true ultimate tensile strain is calculated as follows:

$$\sigma_{uts,t} = \sigma_{uts}\exp\left[m_2\right] \tag{3-D.14}$$

3-D.4 CYCLIC STRESS STRAIN CURVE (21)

The cyclic stress–strain curve of a material (i.e., strain amplitude versus stress amplitude) may be represented by eq. (3-D.15). The material constants for this model are provided in Table 3-D.2.

$$\varepsilon_{ta} = \frac{\sigma_a}{E_y} + \left[\frac{\sigma_a}{K_{css}}\right]^{\frac{1}{n_{css}}} \tag{3-D.15}$$

The hysteresis loop stress–strain curve of a material (i.e., strain range versus stress range) obtained by scaling the cyclic stress–strain curve by a factor of two is represented by eq. (3-D.16). The material constants provided in Table 3-D.2 are also used in this equation.

$$\varepsilon_{tr} = \frac{\sigma_r}{E_y} + 2\left[\frac{\sigma_r}{2K_{css}}\right]^{\frac{1}{n_{css}}} \tag{3-D.16}$$

3-D.5 TANGENT MODULUS

(21) 3-D.5.1 TANGENT MODULUS BASED ON THE STRESS–STRAIN CURVE MODEL

The tangent modulus based on the stress–strain curve model in 3-D.3 is given by the following equation.

$$E_t = \frac{\partial \sigma_t}{\partial \varepsilon_t} = \left(\frac{\partial \varepsilon_t}{\partial \sigma_t}\right)^{-1} = \left(\frac{1}{E_y} + D_1 + D_2 + D_3 + D_4\right)^{-1} \quad (3\text{-D.17})$$

where

$$D_1 = \frac{\sigma_t^{\left(\frac{1}{m_1} - 1\right)}}{2 m_1 A_1^{\left(\frac{1}{m_1}\right)}} \quad (3\text{-D.18})$$

$$D_2 = -\frac{1}{2}\left(\frac{1}{A_1^{\left(\frac{1}{m_1}\right)}}\right) \cdot \left(\sigma_t^{\left(\frac{1}{m_1}\right)} \left\{\frac{2}{K\left(\sigma_{uts} - \sigma_{ys}\right)}\right\} \left\{1 - \tanh^2[H]\right\} + \frac{1}{m_1}\sigma_t^{\left(\frac{1}{m_1} - 1\right)} \tanh[H]\right) \quad (3\text{-D.19})$$

$$D_3 = \frac{\sigma_t^{\left(\frac{1}{m_2} - 1\right)}}{2 m_2 A_2^{\left(\frac{1}{m_2}\right)}} \quad (3\text{-D.20})$$

$$D_4 = \frac{1}{2}\left(\frac{1}{A_2^{\left(\frac{1}{m_2}\right)}}\right) \cdot \left(\sigma_t^{\left(\frac{1}{m_2}\right)} \left\{\frac{2}{K\left(\sigma_{uts} - \sigma_{ys}\right)}\right\} \left\{1 - \tanh^2[H]\right\} + \frac{1}{m_2}\sigma_t^{\left(\frac{1}{m_2} - 1\right)} \tanh[H]\right) \quad (3\text{-D.21})$$

The parameter K is given by eq. (3-D.13).

3-D.5.2 TANGENT MODULUS BASED ON EXTERNAL PRESSURE CHARTS

An acceptable alternative for calculating the Tangent Modulus is to use the External Pressure charts in Section II, Part D, Subpart 3, including the notes to Subpart 3. The appropriate chart for the material under consideration is assigned in the column designated External Pressure Chart Number given in Table 5A or 5B. The tangent modulus, E_t, is equal to $2B/A$, where A is the strain given on the abscissa and B is the stress value on the ordinate of the chart.

3-D.6 NOMENCLATURE

A = Section II, Part D, Subpart 3 external pressure chart A-value.
A_1 = curve fitting constant for the elastic region of the stress–strain curve.
A_2 = curve fitting constant for the plastic region of the stress–strain curve.
B = Section II, Part D, Subpart 3 external pressure chart B-value.
D_1 = coefficient used in the tangent modulus.
D_2 = coefficient used in the tangent modulus.
D_3 = coefficient used in the tangent modulus.
D_4 = coefficient used in the tangent modulus.
ε_p = stress–strain curve fitting parameter.
ε_t = total true strain
ε_{ta} = total true strain amplitude.
ε_{tr} = total true strain range.
ε_{ys} = 0.2% engineering offset strain.
ε_1 = true plastic strain in the micro-strain region of the stress–strain curve.
ε_2 = true plastic strain in the macro-strain region of the stress–strain curve.
E_t = tangent modulus of elasticity evaluated at the temperature of interest.
E_y = modulus of elasticity evaluated at the temperature of interest, see Annex 3-E.
γ_1 = true strain in the micro-strain region of the stress–strain curve.
γ_2 = true strain in the macro-strain region of the stress–strain curve.
H = stress–strain curve fitting parameter.

K = material parameter for stress–strain curve model
K_{css} = material parameter for the cyclic stress–strain curve model.
m_1 = curve fitting exponent for the stress–strain curve equal to the true strain at the proportional limit and the strain hardening coefficient in the large strain region.
m_2 = curve fitting exponent for the stress–strain curve equal to the true strain at the true ultimate stress.
n_{css} = material parameter for the cyclic stress–strain curve model.
σ_a = total stress amplitude.
σ_r = total stress range.
σ_t = true stress at which the true strain will be evaluated, may be a membrane, membrane plus bending, or membrane, membrane plus bending plus peak stress depending on the application.
σ_{ys} = engineering yield stress evaluated at the temperature of interest, see 3-D.1.
σ_{uts} = engineering ultimate tensile stress evaluated at the temperature of interest, see 3-D.2.
$\sigma_{uts,t}$ = true ultimate tensile stress evaluated at the true ultimate tensile strain
R = engineering yield to engineering tensile ratio.

3-D.7 TABLES

Table 3-D.1
Stress–Strain Curve Parameters

Material	Temperature Limit	m_2	ε_p
Ferritic Steel	480°C (900°F)	0.60 (1.00 − R)	2.0E-5
Stainless Steel and Nickel Base Alloys	480°C (900°F)	0.75 (1.00 − R)	2.0E-5
Duplex Stainless Steel	480°C (900°F)	0.70 (0.95 − R)	2.0E-5
Precipitation Hardenable Nickel Base	540°C (1,000°F)	1.09 (0.93 − R)	2.0E-5
Aluminum	120°C (250°F)	0.52 (0.98 − R)	5.0E-6
Copper	65°C (150°F)	0.50 (1.00 − R)	5.0E-6
Titanium and Zirconium	260°C (500°F)	0.50 (0.98 − R)	2.0E-5

Table 3-D.2
Cyclic Stress–Strain Curve Data

Material Description	Temperature, °F	n_{css}	K_{css}, ksi
Carbon Steel (0.75 in. — base metal)	70	0.128	109.8
	390	0.134	105.6
	570	0.093	107.5
	750	0.109	96.6
Carbon Steel (0.75 in. — weld metal)	70	0.110	100.8
	390	0.118	99.6
	570	0.066	100.8
	750	0.067	79.6
Carbon Steel (2 in. — base metal)	70	0.126	100.5
	390	0.113	92.2
	570	0.082	107.5
	750	0.101	93.3
Carbon Steel (4 in. — base metal)	70	0.137	111.0
	390	0.156	115.7
	570	0.100	108.5
	750	0.112	96.9
1Cr–$^1/_2$Mo (0.75 in. — base metal)	70	0.116	95.7
	390	0.126	95.1
	570	0.094	90.4
	750	0.087	90.8

Table 3-D.2
Cyclic Stress–Strain Curve Data (Cont'd)

Material Description	Temperature, °F	n_{css}	K_{css}, ksi
1Cr–½Mo (0.75 in. — weld metal)	70	0.088	96.9
	390	0.114	102.7
	570	0.085	99.1
	750	0.076	86.9
1Cr–½Mo (0.75 in. — base metal)	70	0.105	92.5
	390	0.133	99.2
	570	0.086	88.0
	750	0.079	83.7
1Cr–1Mo–¼V	70	0.128	156.9
	750	0.128	132.3
	930	0.143	118.2
	1,020	0.133	100.5
	1,110	0.153	80.6
2¼Cr–½Mo	70	0.100	115.5
	570	0.109	107.5
	750	0.096	105.9
	930	0.105	94.6
	1,110	0.082	62.1
9Cr–1Mo	70	0.177	141.4
	930	0.132	100.5
	1,020	0.142	88.3
	1,110	0.121	64.3
	1,200	0.125	49.7
Type 304	70	0.171	178.0
	750	0.095	85.6
	930	0.085	79.8
	1,110	0.090	65.3
	1,290	0.094	44.4
Type 304 (Annealed)	70	0.334	330.0
800H	70	0.070	91.5
	930	0.085	110.5
	1,110	0.088	105.7
	1,290	0.092	80.2
	1,470	0.080	45.7
Aluminum (Al–4.5Zn–0.6Mn)	70	0.058	65.7
Aluminum (Al–4.5Zn–1.5Mg)	70	0.047	74.1
Aluminum (1100-T6)	70	0.144	22.3
Aluminum (2014-T6)	70	0.132	139.7
Aluminum (5086)	70	0.139	96.0
Aluminum (6009-T4)	70	0.124	83.7
Aluminum (6009-T6)	70	0.128	91.8
Copper	70	0.263	99.1

Table 3-D.2M
Cyclic Stress–Strain Curve Data

Material Description	Temperature, °C	n_{css}	K_{css}, MPa
Carbon Steel (20 mm — base metal)	20	0.128	757
	200	0.134	728
	300	0.093	741
	400	0.109	666
Carbon Steel (20 mm — weld metal)	20	0.110	695
	200	0.118	687
	300	0.066	695
	400	0.067	549
Carbon Steel (50 mm — base metal)	20	0.126	693
	200	0.113	636
	300	0.082	741
	400	0.101	643
Carbon Steel (100 mm — base metal)	20	0.137	765
	200	0.156	798
	300	0.100	748
	400	0.112	668
1Cr–$^1/_2$Mo (20 mm — base metal)	20	0.116	660
	200	0.126	656
	300	0.094	623
	400	0.087	626
1Cr–$^1/_2$Mo (20 mm — weld metal)	20	0.088	668
	200	0.114	708
	300	0.085	683
	400	0.076	599
1Cr–$^1/_2$Mo (50 mm — base metal)	20	0.105	638
	200	0.133	684
	300	0.086	607
	400	0.079	577
1Cr–1Mo–$^1/_4$V	20	0.128	1082
	400	0.128	912
	500	0.143	815
	550	0.133	693
	600	0.153	556
2$^1/_4$Cr–$^1/_2$Mo	20	0.100	796
	300	0.109	741
	400	0.096	730
	500	0.105	652
	600	0.082	428
9Cr–1Mo	20	0.117	975
	500	0.132	693
	550	0.142	609
	600	0.121	443
	650	0.125	343
Type 304	20	0.171	1227
	400	0.095	590
	500	0.085	550
	600	0.090	450
	700	0.094	306
Type 304 (Annealed)	20	0.334	2275

Table 3-D.2M
Cyclic Stress–Strain Curve Data (Cont'd)

Material Description	Temperature, °C	n_{css}	K_{css}, MPa
800H	20	0.070	631
	500	0.085	762
	600	0.088	729
	700	0.092	553
	800	0.080	315
Aluminum (Al–4.5Zn–0.6Mn)	20	0.058	453
Aluminum (Al–4.5Zn–1.5Mg)	20	0.047	511
Aluminum (1100-T6)	20	0.144	154
Aluminum (2014-T6)	20	0.132	963
Aluminum (5086)	20	0.139	662
Aluminum (6009-T4)	20	0.124	577
Aluminum (6009-T6)	20	0.128	633
Copper	20	0.263	683

ANNEX 3-E
PHYSICAL PROPERTIES

(Normative)

3-E.1 YOUNG'S MODULUS

Values for the Young's Modulus as a function of temperature are provided in Section II, Part D.

3-E.2 THERMAL EXPANSION COEFFICIENT

Values for the thermal expansion coefficient as a function of temperature are provided in Section II, Part D.

3-E.3 THERMAL CONDUCTIVITY

Values for the thermal conductivity as a function of temperature are provided in Section II, Part D.

3-E.4 THERMAL DIFFUSIVITY

Values for the thermal diffusivity as a function of temperature are provided in Section II, Part D.

ANNEX 3-F
DESIGN FATIGUE CURVES

(Normative)

3-F.1 SMOOTH BAR DESIGN FATIGUE CURVES

3-F.1.1

Fatigue analysis performed through direct interpretation of the smooth bar fatigue curves found in 3-F.5 requires the calculated stress amplitude, S_a, be corrected for temperature by the ratio of the modulus of elasticity of the given fatigue curve to the value used in the analysis. The equations used to correct S_a for the temperature effect based upon the different material fatigue curves are provided in Table 3-F.1. The temperature-corrected stress amplitude, S_{ac}, is then used to enter the smooth bar fatigue curves to determine the number of allowable cycles, N.

NOTES:

(1) For Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F), the fatigue curve values may be interpolated for intermediate values of the ultimate tensile strength.

(2) For Wrought 70–30 Copper–Nickel for temperatures not exceeding 371°C (700°F), the fatigue curve values may be interpolated for intermediate values of the minimum specified yield strength.

3-F.1.2

Fatigue analysis performed using smooth bar fatigue curve models in equation form is provided below. The fatigue curves and the associated equations for different materials are also shown below.

(a) Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F). The fatigue curve values may be interpolated for intermediate values of the ultimate tensile strength.

$$Y = \log\left[28.3\,\mathrm{E3}\left(\frac{S_a}{E_T}\right)\right] \tag{3-F.1}$$

(1) For $\sigma_{uts} \le 552$ MPa (80 ksi) (see Figures 3-F.1M and 3-F.1) and for 48 MPa (7 ksi) $\le S_a \le$ 3 999 MPa (580 ksi)

$$X = -4706.5245 + 1813.6228Y + \frac{6785.5644}{Y} - 368.12404Y^2 - \frac{5133.7345}{Y^2} + 30.708204Y^3 + \frac{1596.1916}{Y^3} \quad \text{for } 10^Y \ge 20 \tag{3-F.2}$$

$$X = \frac{38.1309 - 60.1705Y^2 + 25.0352Y^4}{1 + 1.80224Y^2 - 4.68904Y^4 + 2.26536Y^6} \quad \text{for } 10^Y < 20 \tag{3-F.3}$$

(2) For σ_{uts} = 793 MPa to 892 MPa (115 ksi to 130 ksi) (see Figures 3-F.2M and 3-F.2) and for 77.2 MPa (11.2 ksi) $\le S_a \le$ 2 896 MPa (420 ksi)

$$X = \frac{5.37689 - 5.25401Y + 1.14427Y^2}{1 - 0.960816Y + 0.291399Y^2 - 0.0562968Y^3} \quad \text{for } 10^Y \ge 43 \tag{3-F.4}$$

$$X = \frac{-9.41749 + 14.7982Y - 5.94Y^2}{1 - 3.46282Y + 3.63495Y^2 - 1.21849Y^3} \quad \text{for } 10^Y < 43 \tag{3-F.5}$$

(b) Series 3XX High Alloy Steels, Nickel–Chromium–Iron Alloy, Nickel–Iron–Chromium Alloy, and Nickel–Copper Alloy for temperatures not exceeding 427°C (800°F) (see Figures 3-F.3M and 3-F.3) and for 93.7 MPa (13.6 ksi) ≤ S_a ≤ 6 000 MPa (870 ksi)

$$Y = \log\left[28.3\,\mathrm{E3}\left(\frac{S_a}{E_T}\right)\right] \tag{3-F.6}$$

$$X = \frac{17.0181 - 19.8713Y + 4.21366Y^2}{1 - 0.1720606Y - 0.633592Y^2} \quad \text{for } 10^Y \geq 14.4 \tag{3-F.7}$$

$$X = \frac{1}{-0.331096 + \dfrac{4.3261\ln(Y)}{Y^2}} \quad \text{for } 10^Y < 14.4 \tag{3-F.8}$$

(c) Wrought 70–30 Copper–Nickel for temperatures not exceeding 371°C (700°F). The fatigue curve values may be interpolated for intermediate values of the minimum specified yield strength.

$$Y = \log\left[20.0\,\mathrm{E3}\left(\frac{S_a}{E_T}\right)\right] \tag{3-F.9}$$

(1) For σ_{ys} ≤ 134 MPa (18 ksi) (see Figures 3-F.4M and 3-F.4) and for 83 MPa (12 ksi) ≤ S_a ≤ 1 793 MPa (260 ksi)

$$X = -2.632 + \frac{0.1186}{\ln(Y)} + \frac{15.12\ln(Y)}{Y^2} + \frac{7.087}{Y^2} \tag{3-F.10}$$

(2) For σ_{ys} = 207 MPa (30 ksi) (see Figures 3-F.5M and 3-F.5) and for 62 MPa (9 ksi) ≤ S_a ≤ 1 793 MPa (260 ksi)

$$X = 8.580044 - 1.889784Y - \frac{8.261383\ln(Y)}{Y} \quad \text{for } 10^Y \geq 24.5 \tag{3-F.11}$$

$$X = 5.89029 - 0.2280247Y - \frac{6.649501\ln(Y)}{Y} \quad \text{for } 10^Y < 24.5 \tag{3-F.12}$$

(3) For σ_{ys} = 310 MPa (45 ksi) (see Figures 3-F.6M and 3-F.6) and for 34 MPa (5 ksi) ≤ S_a ≤ 1 793 MPa (260 ksi)

$$X = -884.9989 + \frac{8936.214}{Y} - \frac{36034.88}{Y^2} + \frac{72508.69}{Y^3} - \frac{72703.36}{Y^4} + \frac{29053.66}{Y^5} \quad \text{for } 10^Y \geq 46 \tag{3-F.13}$$

$$X = -17.50197 + \frac{109.168}{Y} - \frac{236.7921}{Y^2} + \frac{257.9938}{Y^3} - \frac{137.1654}{Y^4} + \frac{28.55546}{Y^5} \quad \text{for } 10^Y < 46 \tag{3-F.14}$$

(d) Nickel–Chromium–Molybdenum–Iron, Alloys X, G, C-4, and C-276 for temperatures not exceeding 427°C (800°F) (see Figures 3-F.7M and 3-F.7) and for 103 MPa (15 ksi) ≤ S_a ≤ 4 881 MPa (708 ksi)

$$Y = \log\left[28.3\,\mathrm{E3}\left(\frac{S_a}{E_T}\right)\right] \tag{3-F.15}$$

$$X = \frac{-42.08579 + 12.514054Y}{1 - 4.3290016Y + 0.60540862Y^2} \quad \text{for } 10^Y \geq 35.9 \tag{3-F.16}$$

$$X = \frac{9.030556 - 8.1906623Y}{1 - 0.36077181Y - 0.47064984Y^2} \quad \text{for } 10^Y < 35.9 \tag{3-F.17}$$

(e) High strength bolting for temperatures not exceeding 371°C (700°F)

$$Y = \log\left[30.0\,\mathrm{E3}\left(\frac{S_a}{E_T}\right)\right] \tag{3-F.18}$$

(1) For a maximum nominal stress ≤ $2.7S_M$ (see Figures 3-F.8M and 3-F.8) and for 93 MPa (13.5 ksi) ≤ S_a ≤ 7 929 MPa (1,150 ksi)

$$X = 3.75565644 - \frac{75.58638}{Y} + \frac{403.70774}{Y^2} - \frac{830.40346}{Y^3} + \frac{772.53426}{Y^4} - \frac{267.75105}{Y^5} \tag{3-F.19}$$

(2) For a maximum nominal stress > $2.7S_M$ (see Figures 3-F.9M and 3-F.9) and for 37 MPa (5.3 ksi) ≤ S_a ≤ 7 929 MPa (1,150 ksi)

$$X = -9.0006161 + \frac{51.928295}{Y} - \frac{86.121576}{Y^2} + \frac{73.1573}{Y^3} - \frac{29.945507}{Y^4} + \frac{4.7332046}{Y^5} \tag{3-F.20}$$

3-F.1.3

The design number of design cycles, N, can be computed from eq. (3-F.21) based on the parameter X calculated for the applicable material.

$$N = 10^X \tag{3-F.21}$$

3-F.2 WELDED JOINT DESIGN FATIGUE CURVES

3-F.2.1

Subject to the limitations of 5.5.5, the welded joint design fatigue curves in 3-F.5 can be used to evaluate welded joints for the following materials and associated temperature limits:

(a) Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F)

(b) Series 3XX High Alloy Steels, Nickel–Chromium–Iron Alloy, Nickel–Iron–Chromium Alloy, and Nickel–Copper Alloy for temperatures not exceeding 427°C (800°F)

(c) Wrought 70 Copper–Nickel for temperatures not exceeding 371°C (700°F)

(d) Nickel–Chromium–Molybdenum–Iron, Alloys X, G, C-4, and C-276 for temperatures not exceeding 427°C (800°F)

(e) Aluminum Alloys

3-F.2.2

The number of allowable design cycles for the welded joint fatigue curve shall be computed as follows.

(a) The design number of allowable design cycles, N, can be computed from eq. (3-F.22) based on the equivalent structural stress range parameter, $\Delta S_{ess,k}$, determined in accordance with 5.5.5 of this Division. The constants C and h for use in eq. (3-F.22) are provided in Table 3-F.2. The lower 99% Prediction Interval (-3σ) shall be used for design unless otherwise agreed to by the Owner-User and the Manufacturer.

$$N = \frac{f_I}{f_E}\left(\frac{f_{MT} \cdot C}{\Delta S_{ess,k}}\right)^{\frac{1}{h}} \tag{3-F.22}$$

(b) If a fatigue improvement method is performed that exceeds the fabrication requirements of this Division, then a fatigue improvement factor, f_I, may be applied. The fatigue improvement factors shown below may be used. An alternative factor determined may also be used if agreed to by the user or user's designated agent and the Manufacturer.

(1) For burr grinding in accordance with Figure 6.2

$$f_I = 1.0 + 2.5 \cdot (10)^q \tag{3-F.23}$$

(2) For TIG dressing

$$f_I = 1.0 + 2.5 \cdot (10)^q \tag{3-F.24}$$

(3) For hammer peening

$$f_I = 1.0 + 4.0 \cdot (10)^q \tag{3-F.25}$$

In the above equations, the parameter is given by the following equation:

$$q = -0.0016 \cdot \left(\frac{\Delta S_{ess,k}}{C_{usm}} \right)^{1.6} \tag{3-F.26}$$

(c) The design fatigue cycles given by eq. (3-F.22) do not include any allowances for corrosive conditions and may be modified to account for the effects of environment other than ambient air that may cause corrosion or subcritical crack propagation. If corrosion fatigue is anticipated, a factor should be chosen on the basis of experience or testing by which the calculated design fatigue cycles (fatigue strength) should be reduced to compensate for the corrosion. The environmental modification factor, f_E, is typically a function of the fluid environment, loading frequency, temperature, and material variables such as grain size and chemical composition. The environmental modification factor, f_E, shall be specified in the User's Design Specification.

(d) A temperature adjustment is required to the fatigue curve for materials other than carbon steel and/or for temperatures above 21°C (70°F). The temperature adjustment factor is given by eq. (3-F.27).

$$f_{MT} = \frac{E_T}{E_{ACS}} \tag{3-F.27}$$

3-F.3 NOMENCLATURE

C_{usm} = conversion factor, C_{usm} = 1.0 for units of stress in ksi and C_{usm} = 14.148299 for units of stress in MPa
E_{ACS} = modulus of elasticity of carbon steel at ambient temperature or 21°C (70°F)
E_T = modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated
f_E = environmental correction factor to the welded joint fatigue curve
f_I = fatigue improvement method correction factor to the welded joint fatigue curve
f_{MT} = material and temperature correction factor to the welded joint fatigue curve
q = parameter used to determine the effect equivalent structural stress range on the fatigue improvement factor
N = number of allowable design cycles
S_a = computed stress amplitude from Part 5
S_{ac} = temperature-corrected stress amplitude
X = exponent used to compute the permissible number of cycles
Y = stress amplitude temperature correction factor used to compute X
$\Delta S_{ess,k}$ = computed equivalent structural stress range parameter from Part 5
σ_{uts} = minimum specified ultimate tensile strength

3-F.4 TABLES

Table 3-F.1
Smooth Bar Fatigue Curve Stress Amplitude Correction Equations

Fatigue Curve	Temperature-Corrected Stress Amplitude, S_{ac}	
	MPa	ksi
Figure 3-F.1 Figure 3-F.2 Figure 3-F.3	$195.0\text{ E3}\left(\frac{S_a}{E_T}\right)$	$28.3\text{ E3}\left(\frac{S_a}{E_T}\right)$
Figure 3-F.4 Figure 3-F.5 Figure 3-F.6	$138.0\text{ E3}\left(\frac{S_a}{E_T}\right)$	$20.0\text{ E3}\left(\frac{S_a}{E_T}\right)$
Figure 3-F.7	$195.0\text{ E3}\left(\frac{S_a}{E_T}\right)$	$28.3\text{ E3}\left(\frac{S_a}{E_T}\right)$
Figure 3-F.8 Figure 3-F.9	$206.0\text{ E3}\left(\frac{S_a}{E_T}\right)$	$30.0\text{ E3}\left(\frac{S_a}{E_T}\right)$

Table 3-F.2
Coefficients for the Welded Joint Fatigue Curves

Statistical Basis	Ferritic and Stainless Steels C	Ferritic and Stainless Steels h	Aluminum C	Aluminum h
Mean Curve	1,408.7	0.31950	247.04	0.27712
Upper 68% Prediction Interval (+1σ)	1,688.3	0.31950	303.45	0.27712
Lower 68% Prediction Interval (−1σ)	1,175.4	0.31950	201.12	0.27712
Upper 95% Prediction Interval (+2σ)	2,023.4	0.31950	372.73	0.27712
Lower 95% Prediction Interval (−2σ)	980.8	0.31950	163.73	0.27712
Upper 99% Prediction Interval (+3σ)	2,424.9	0.31950	457.84	0.27712
Lower 99% Prediction Interval (−3σ)	818.3	0.31950	133.29	0.27712

GENERAL NOTE: In U.S. Customary units, the equivalent structural stress range parameter, $\Delta S_{ess,k}$, in 3-F.2.2 and the structural stress effective thickness, t_{ess}, defined in 5.5.5 are in ksi/(inches)$^{(2-m_{ss})/2m_{ss}}$ and inches, respectively. The parameter m_{ss} is defined in 5.5.5.

Table 3-F.2M
Coefficients for the Welded Joint Fatigue Curves

Statistical Basis	Ferritic and Stainless Steels C	Ferritic and Stainless Steels h	Aluminum C	Aluminum h
Mean Curve	19 930.2	0.31950	3 495.13	0.27712
Upper 68% Prediction Interval (+1σ)	23 885.8	0.31950	4 293.19	0.27712
Lower 68% Prediction Interval (−1σ)	16 629.7	0.31950	2 845.42	0.27712
Upper 95% Prediction Interval (+2σ)	28 626.5	0.31950	5 273.48	0.27712
Lower 95% Prediction Interval (−2σ)	13 875.7	0.31950	2 316.48	0.27712
Upper 99% Prediction Interval (+3σ)	34 308.1	0.31950	6 477.60	0.27712
Lower 99% Prediction Interval (−3σ)	11 577.9	0.31950	1 885.87	0.27712

GENERAL NOTE: In SI units, the equivalent structural stress range parameter, $\Delta S_{ess,k}$, in 3-F.2.2 and the structural stress effective thickness, t_{ess}, defined in 5.5.5 are in MPa/(mm)$^{(2-m_{ss})/2m_{ss}}$ and millimeters, respectively. The parameter m_{ss} is defined in 5.5.5.

3-F.5 FIGURES

Figure 3-F.1
Fatigue Curve for Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for Temperatures Not Exceeding 700°F — $\sigma_{uts} \leq 80$ ksi

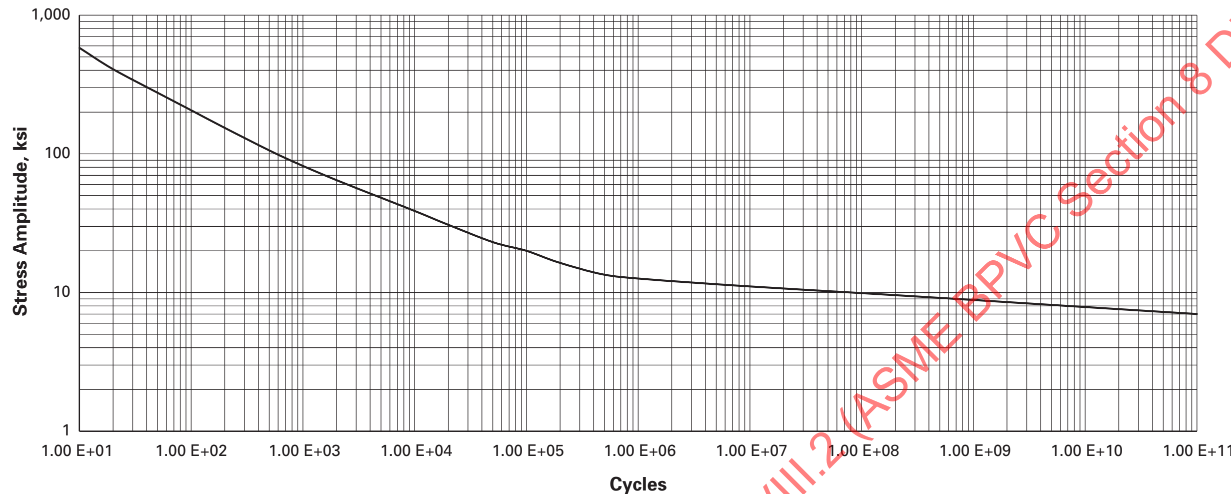


Figure 3-F.1M
Fatigue Curve for Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for Temperatures Not Exceeding 371°C — $\sigma_{uts} \leq 552$ MPa

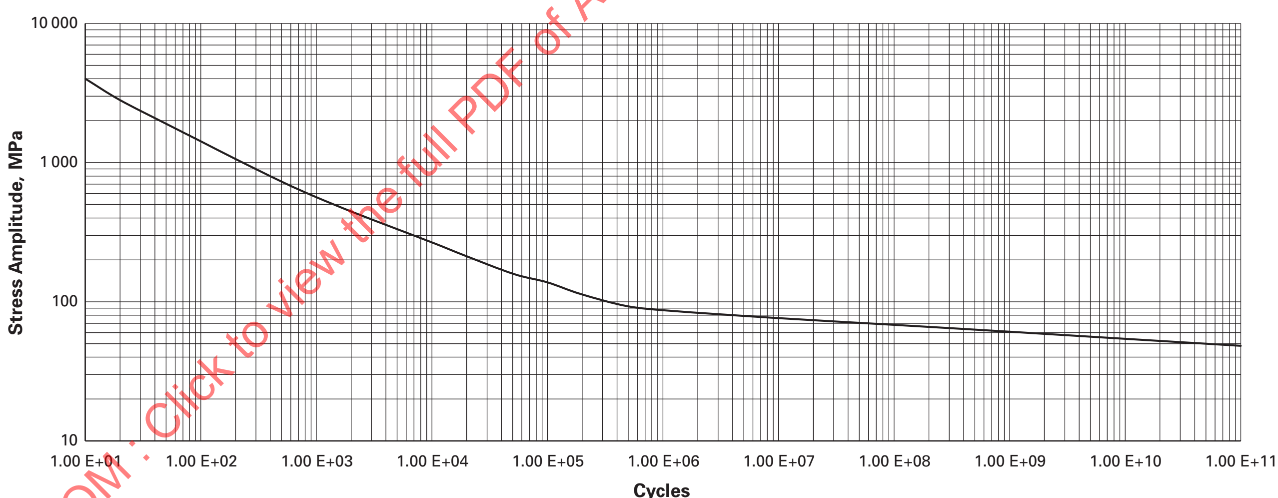


Figure 3-F.2
Fatigue Curve for Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for Temperatures Not Exceeding 700°F — σ_{uts} = 115 ksi to 130 ksi

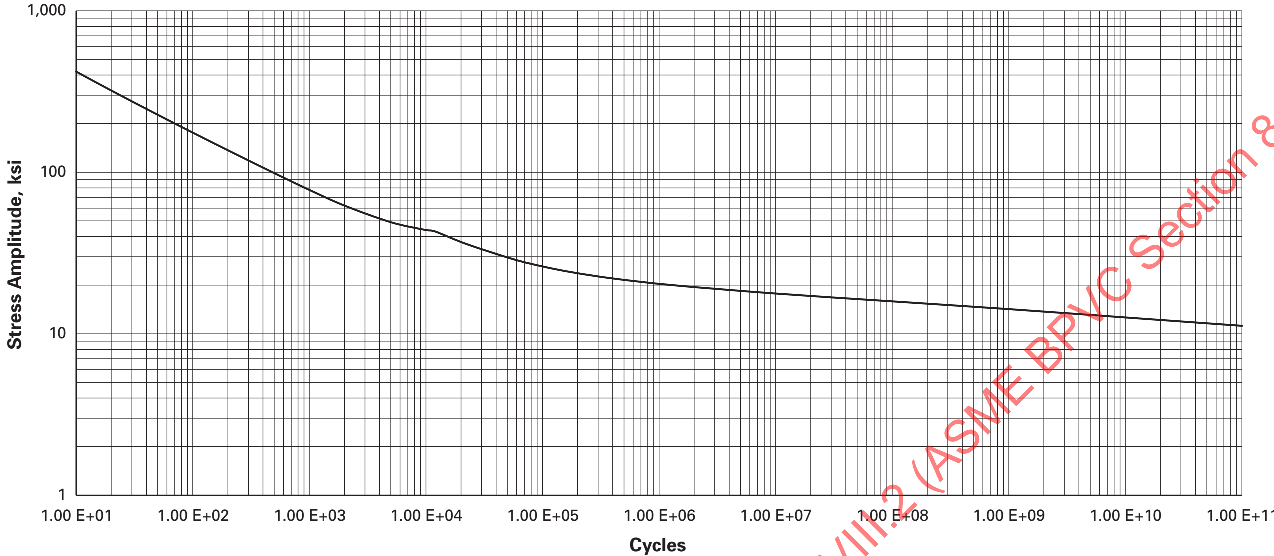


Figure 3-F.2M
Fatigue Curve for Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for Temperatures Not Exceeding 371°C — σ_{uts} = 793 MPa to 892 MPa

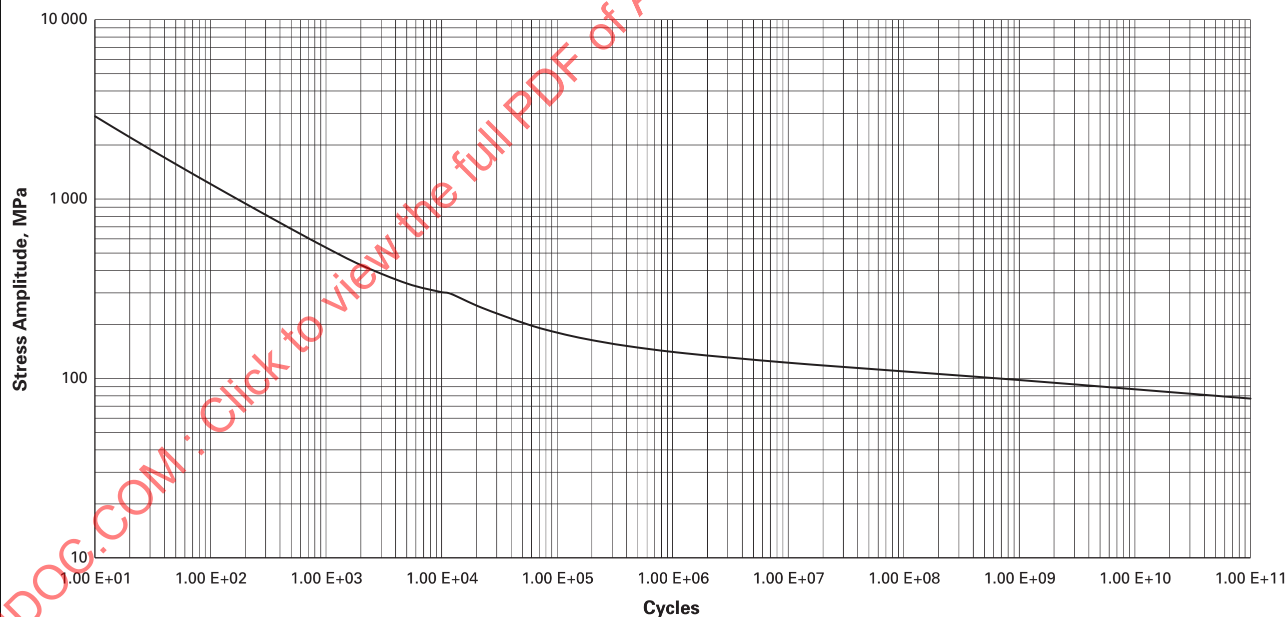


Figure 3-F.3
Fatigue Curve for Series 3XX High Alloy Steels, Nickel–Chromium–Iron Alloy, Nickel–Iron–Chromium Alloy, and Nickel–Copper Alloy for Temperatures Not Exceeding 800°F

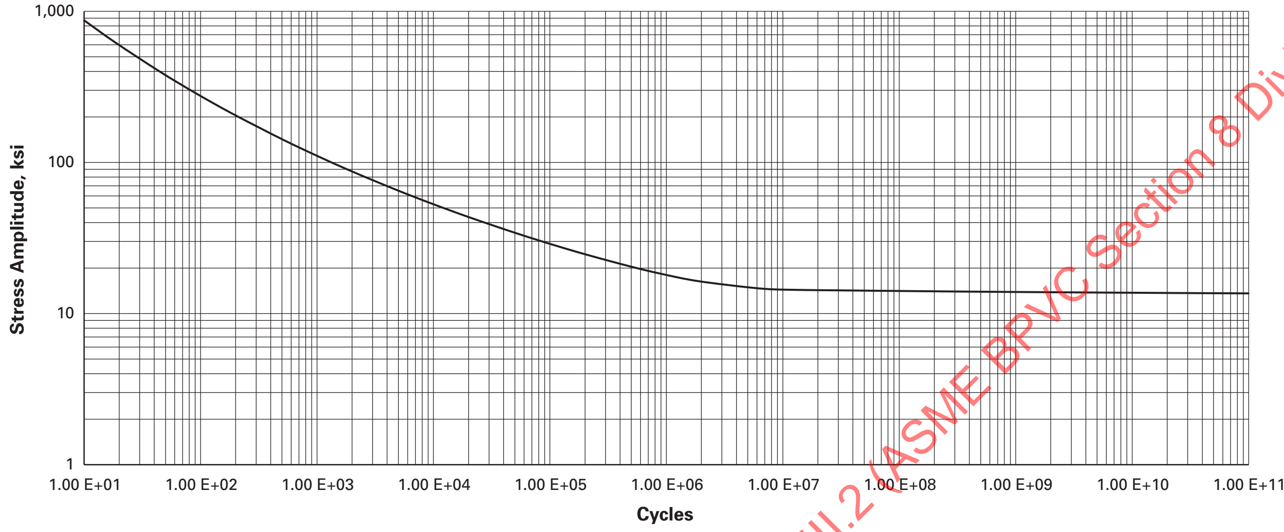


Figure 3-F.3M
Fatigue Curve for Series 3XX High Alloy Steels, Nickel–Chromium–Iron Alloy, Nickel–Iron–Chromium Alloy, and Nickel–Copper Alloy for Temperatures Not Exceeding 427°C

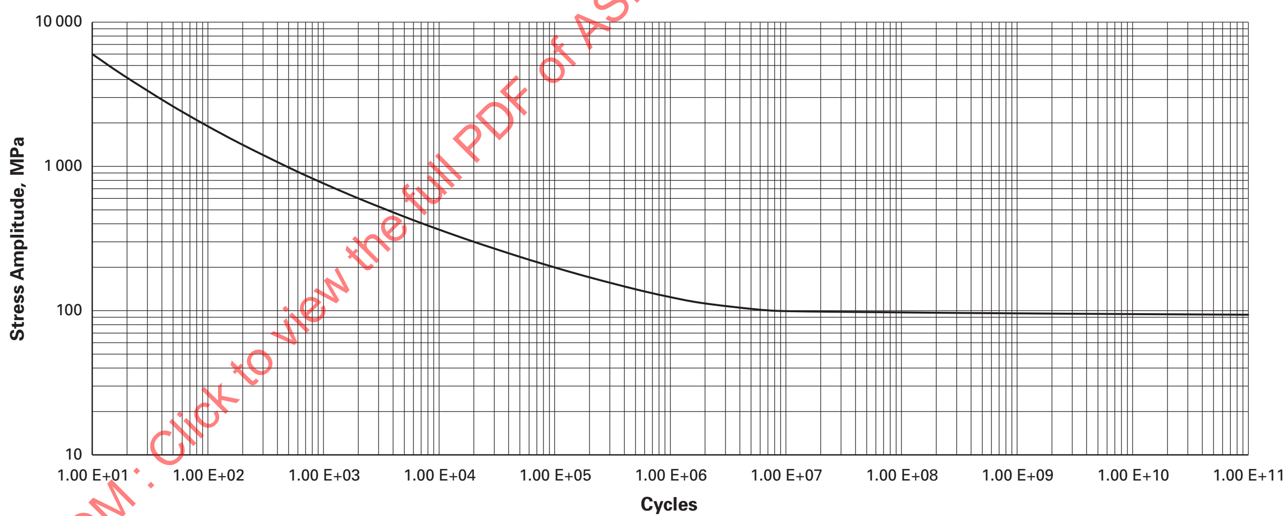


Figure 3-F.4
Fatigue Curve for Wrought 70–30 Copper–Nickel for Temperatures Not Exceeding 700°F — σ_{ys} ≤ 18 ksi

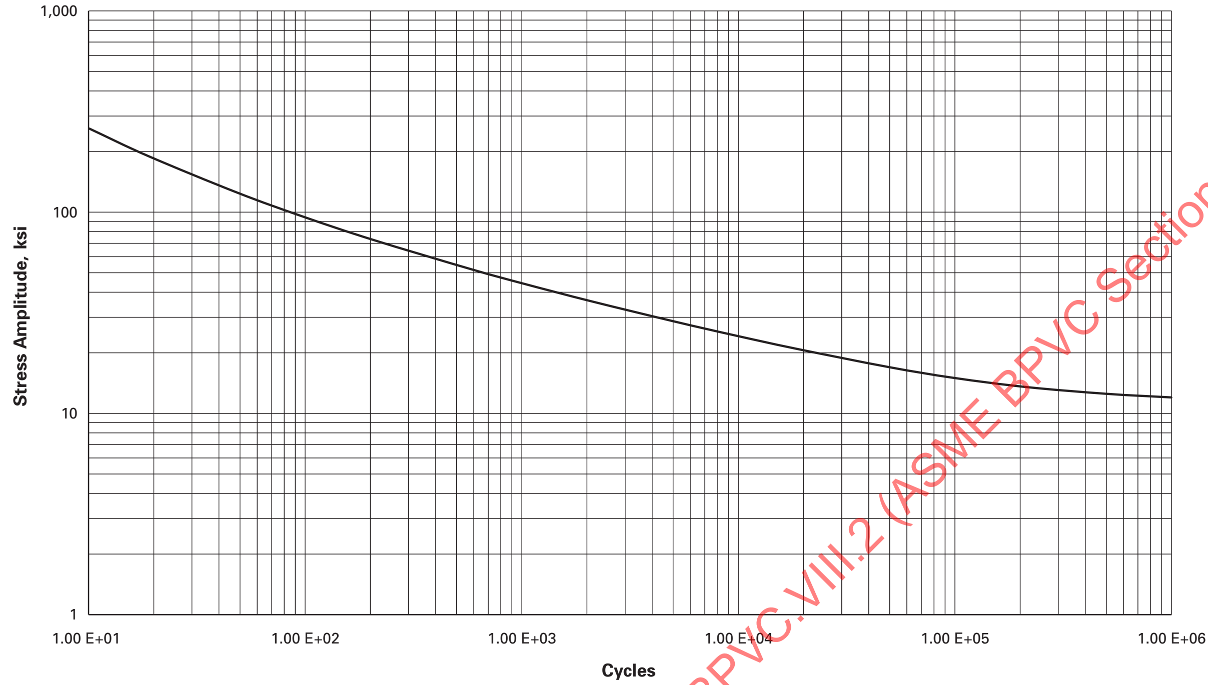


Figure 3-F.4M
Fatigue Curve for Wrought 70–30 Copper–Nickel for Temperatures Not Exceeding 371°C — σ_{ys} ≤ 134 MPa

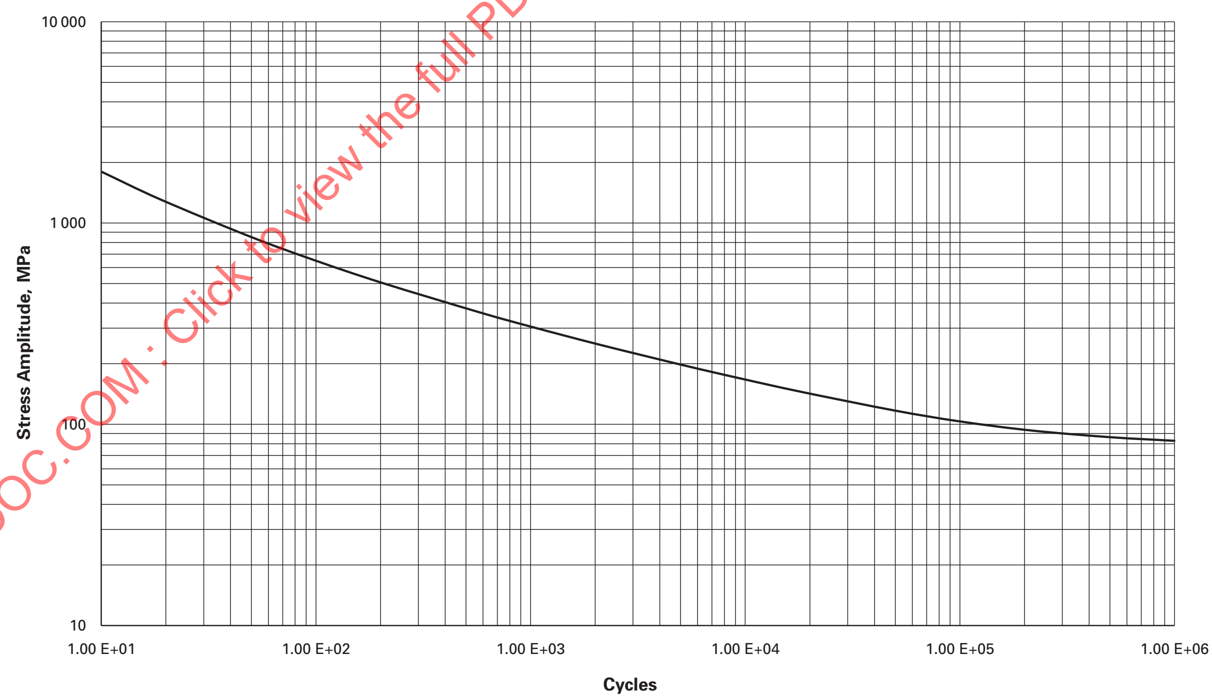


Figure 3-F.5
Fatigue Curve for Wrought 70–30 Copper–Nickel for Temperatures Not Exceeding 700°F — σ_{ys} = 30 ksi

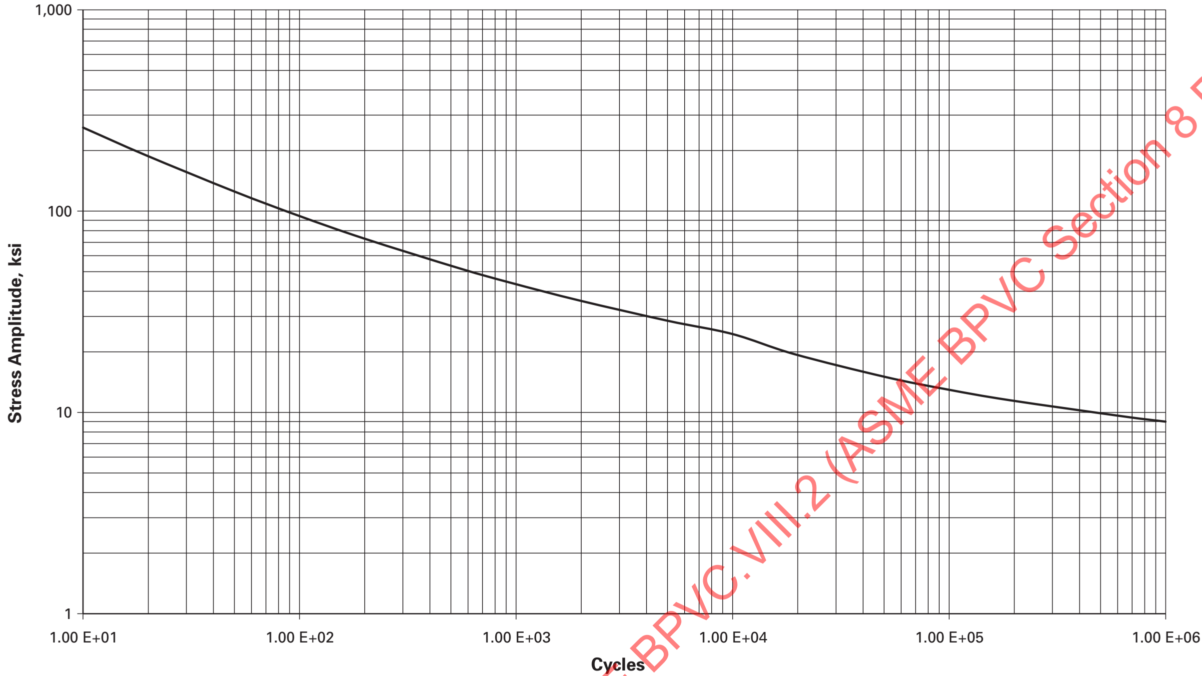


Figure 3-F.5M
Fatigue Curve for Wrought 70–30 Copper–Nickel for Temperatures Not Exceeding 371°C — σ_{ys} = 207 MPa

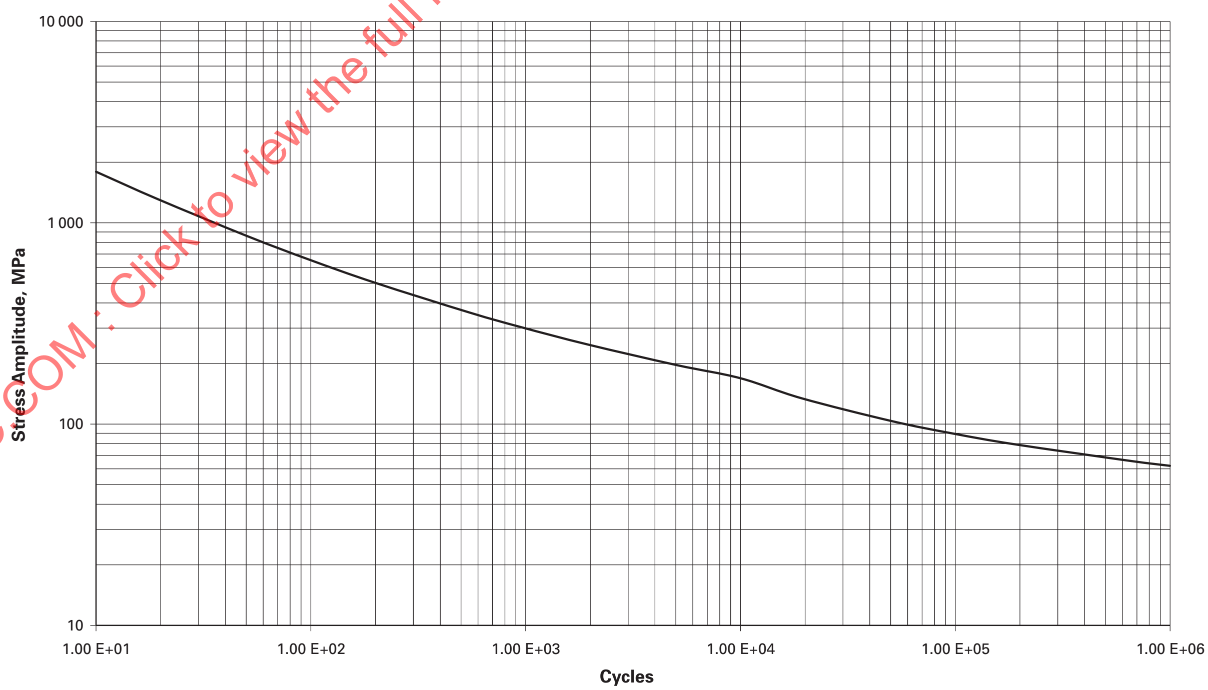


Figure 3-F.6
Fatigue Curve for Wrought 70–30 Copper–Nickel for Temperatures Not Exceeding 700°F — σ_{ys} = 45 ksi

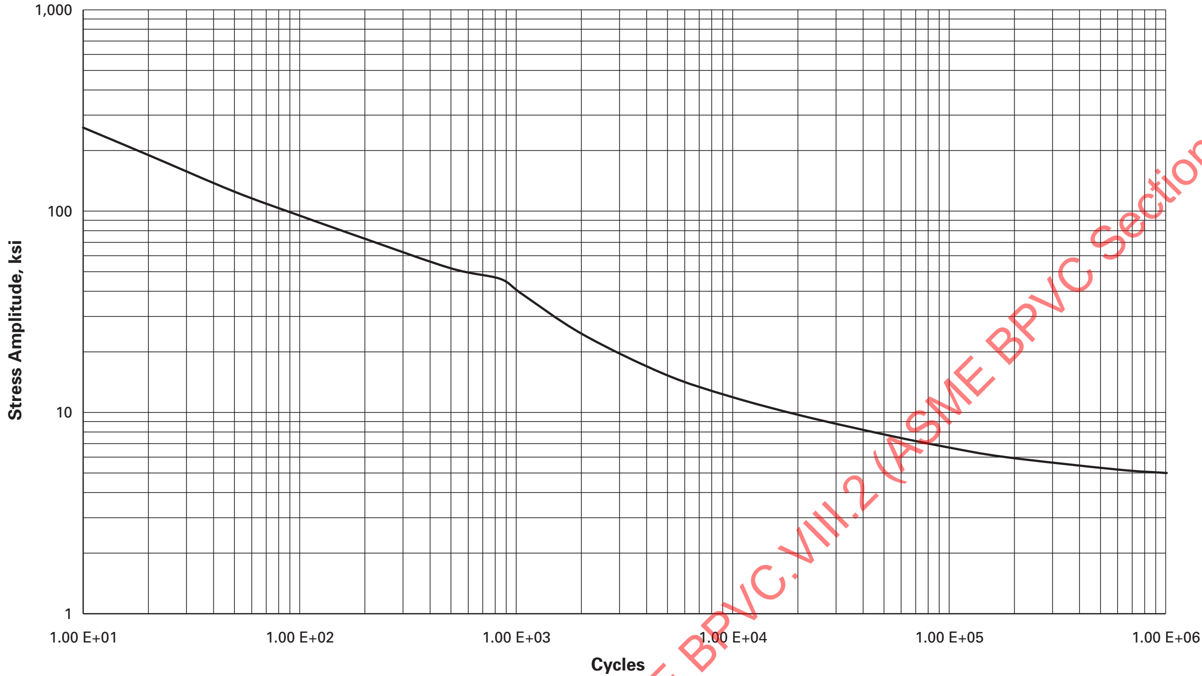


Figure 3-F.6M
Fatigue Curve for Wrought 70–30 Copper–Nickel for Temperatures Not Exceeding 371°C — σ_{ys} = 310 MPa

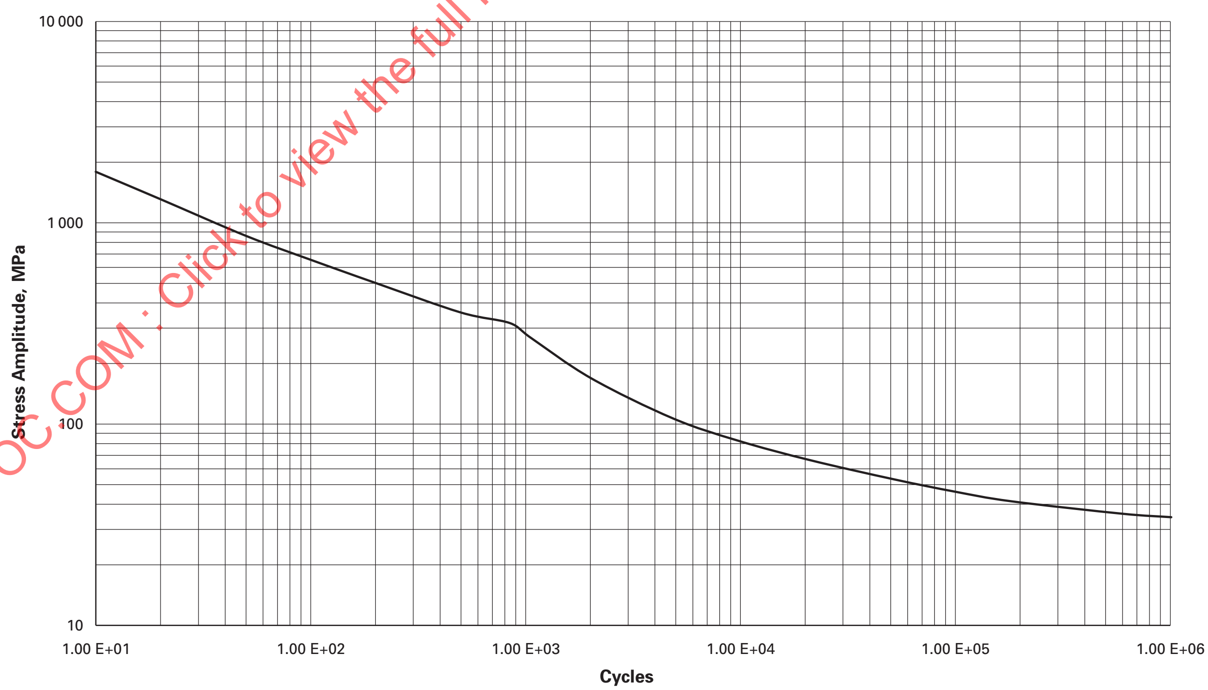


Figure 3-F.7
Fatigue Curve for Nickel–Chromium–Molybdenum–Iron, Alloys X, G, C-4, and C-276 for Temperatures Not Exceeding 800°F

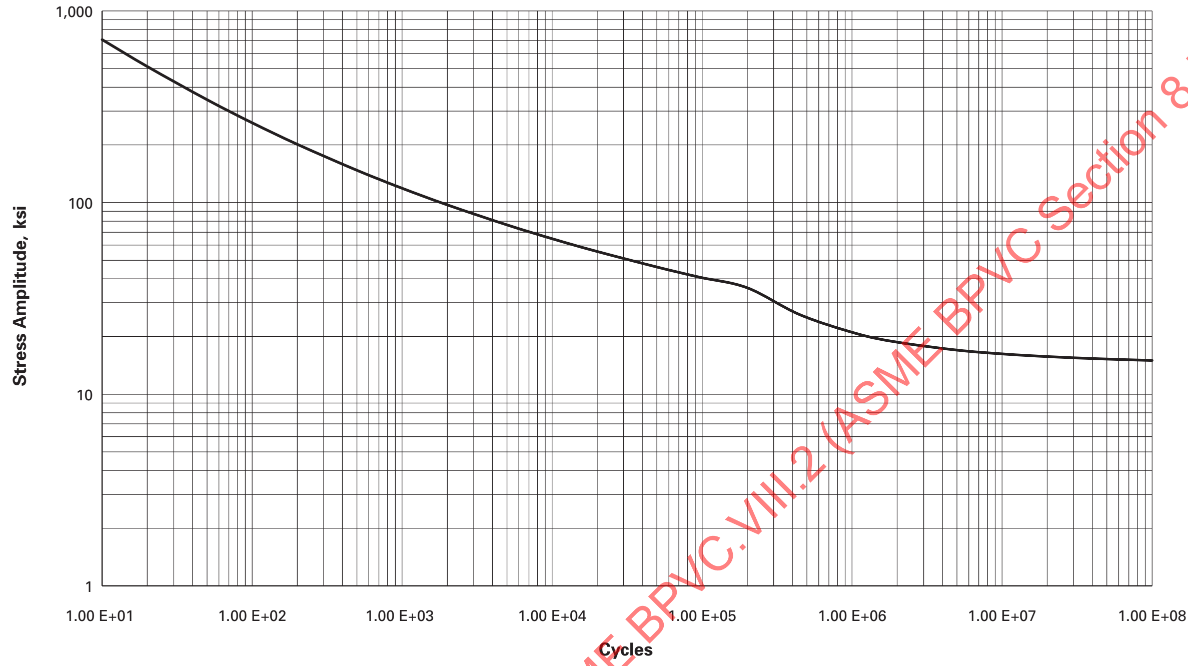


Figure 3-F.7M
Fatigue Curve for Nickel–Chromium–Molybdenum–Iron, Alloys X, G, C-4, and C-276 for Temperatures Not Exceeding 427°C

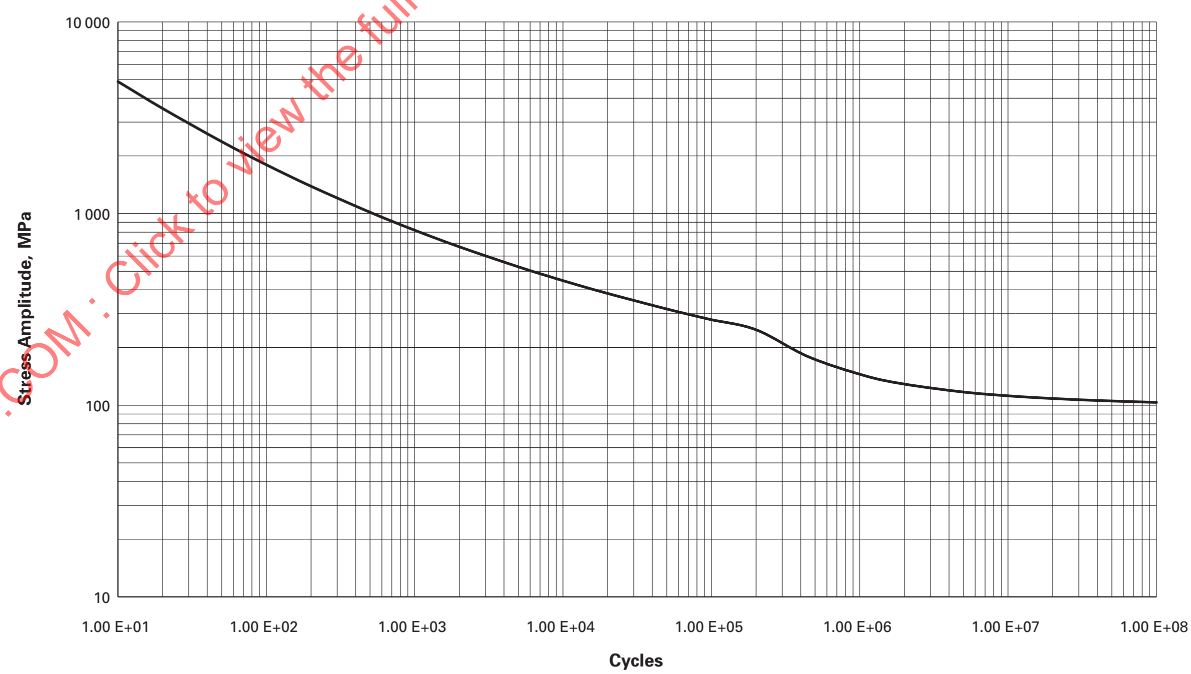


Figure 3-F.8
Fatigue Curve for High Strength Bolting for Temperatures Not Exceeding 700°F — Maximum Nominal Stress ≤ 2.7S_M

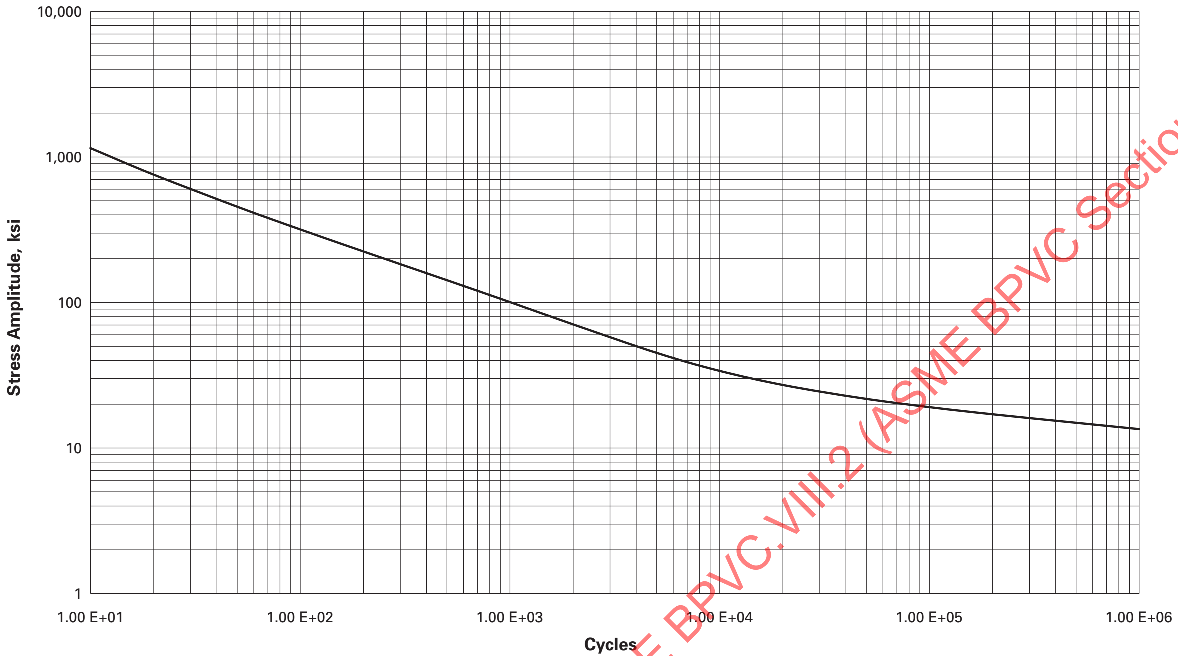


Figure 3-F.8M
Fatigue Curve for High Strength Bolting for Temperatures Not Exceeding 371°C — Maximum Nominal Stress ≤ 2.7S_M

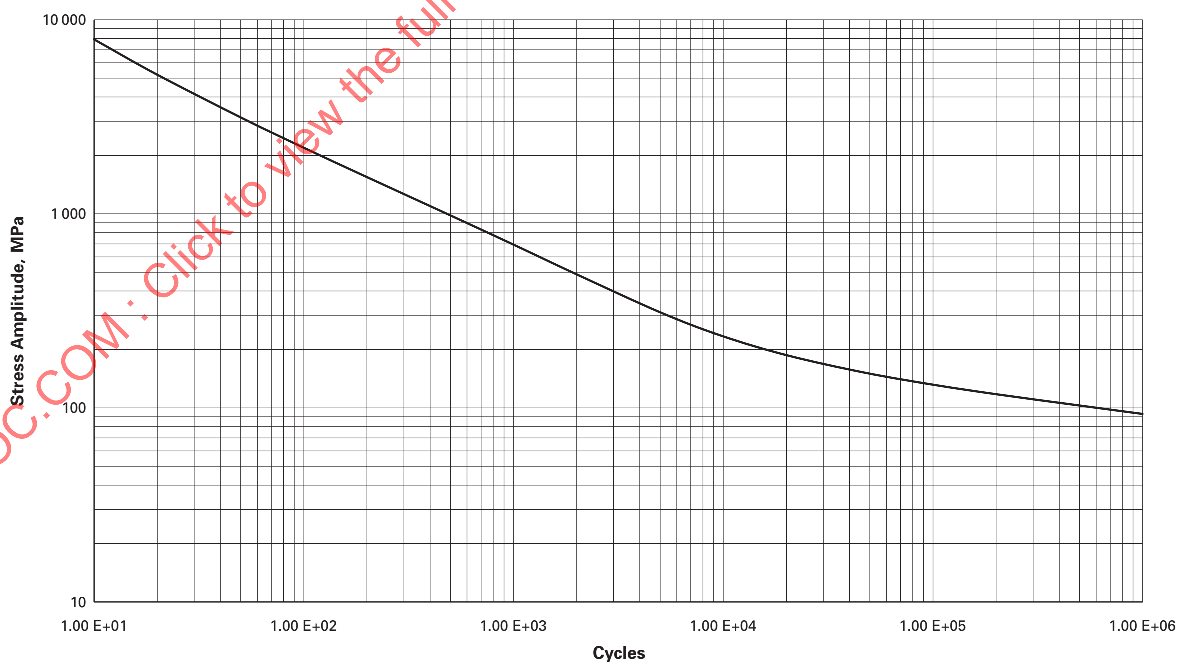


Figure 3-F.9
Fatigue Curve for High Strength Bolting for Temperatures Not Exceeding 700°F — Maximum Nominal Stress > 2.7S_M

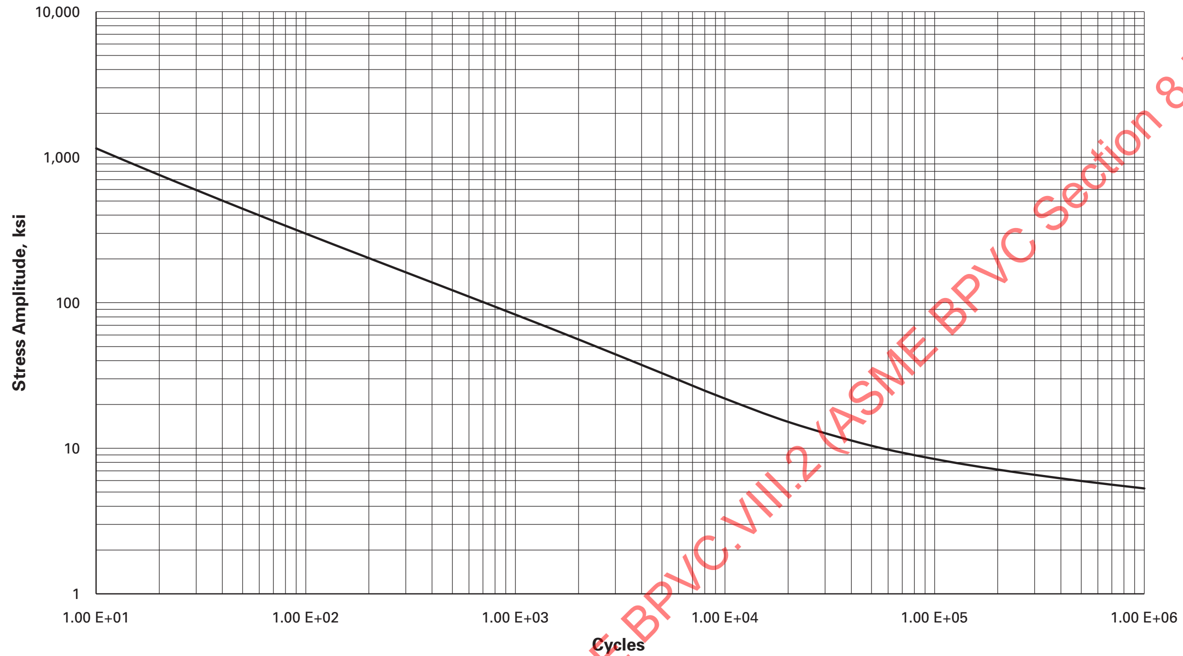
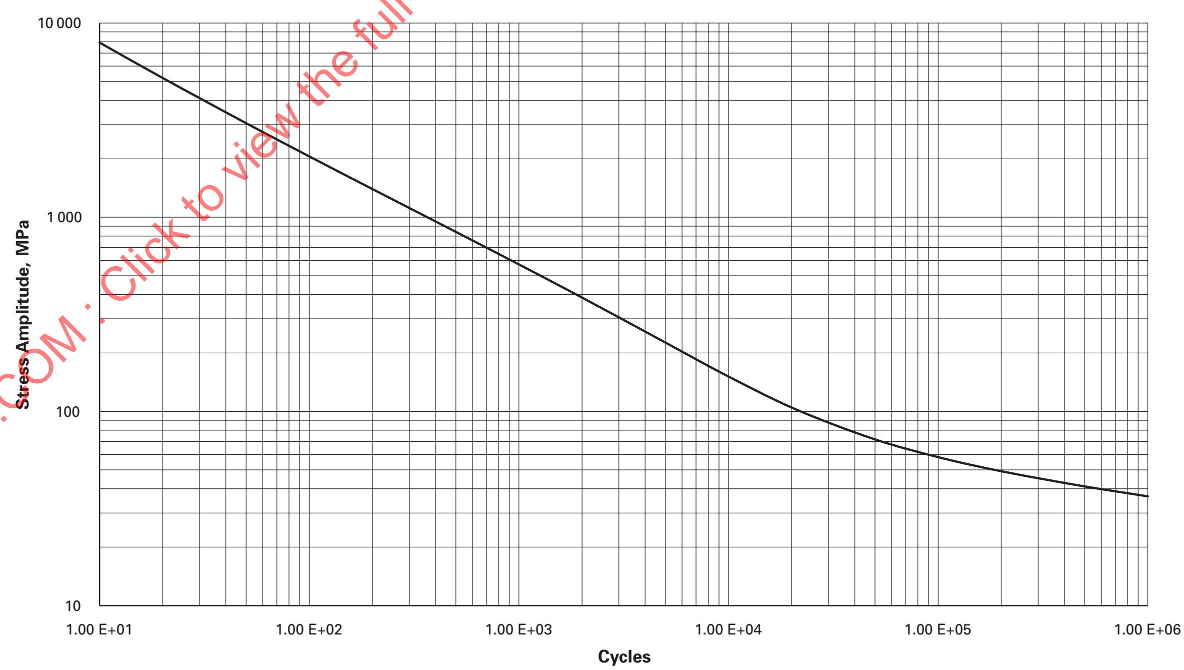


Figure 3-F.9M
Fatigue Curve for High Strength Bolting for Temperatures Not Exceeding 371°C — Maximum Nominal Stress > 2.7S_M



PART 4
DESIGN BY RULE REQUIREMENTS

4.1 GENERAL REQUIREMENTS

4.1.1 SCOPE

4.1.1.1 The basic requirements for application of the design-by-rules methods of this Division are described in 4.1. The requirements of Part 4 provide design rules for commonly used pressure vessel shapes under pressure loading and, within specified limits, rules or guidance for treatment of other loadings.

4.1.1.2 Part 4 does not provide rules to cover all loadings, geometries, and details. See Part 2 for User Responsibilities and User's Design Specification.

4.1.1.2.1 Class 1. When design rules are not provided in Part 4 for a vessel or vessel part, the Manufacturer shall either perform a stress analysis in accordance with Part 5 considering all of the loadings specified in the User's Design Specification, or, with acceptance by the Authorized Inspector, use a recognized and accepted design-by-rule method that meets the applicable design allowable stress criteria given in 4.1.6. If the design cannot be performed using Part 5 or a design-by-rule method (e.g., creep-fatigue), a design method consistent with the overall design philosophy of Class 1 and acceptable to the Authorized Inspector shall be used.

4.1.1.2.2 Class 2. When design rules are not provided for a vessel or vessel part, the Manufacturer shall perform a stress analysis in accordance with Part 5 considering all of the loadings specified in the User's Design Specification.

4.1.1.3 The design procedures in Part 4 may be used if the allowable stress at the design temperature is governed by time-independent or time-dependent properties unless otherwise noted in a specific design procedure. When the vessel is operating at a temperature where the allowable stress is governed by time dependent properties, the effects of joint alignment (see 6.1.6.1) and weld peaking (see 6.1.6.3) in shells and heads shall be considered.

4.1.1.4 A screening criterion shall be applied to all vessel parts designed in accordance with this Division to determine if a fatigue analysis is required. The fatigue screening criterion shall be performed in accordance with 5.5.2. If the results of this screening indicate that a fatigue analysis is required, then the analysis shall be performed in accordance with 5.5.2. If the allowable stress at the design temperature is governed by time-dependent properties, then a fatigue screening analysis based on experience with comparable equipment shall be satisfied (see 5.5.2.2).

4.1.1.5 See 4.1.1.5.1 and 4.1.1.5.2.

4.1.1.5.1 Class 1. Rules in Part 5 shall not be used in lieu of rules in Part 4.

4.1.1.5.2 Class 2. A design-by-analysis in accordance with Part 5 may be used to establish the design thickness and/or configuration (i.e., nozzle reinforcement configuration) in lieu of the design-by-rules in Part 4 for any geometry or loading conditions (see 4.1.5.1). Components of the same pressure vessel may be designed (thickness and configuration) using a combination of Part 4 design-by-rules or any of the three methods of Part 5 design-by-analysis in 5.2.1.1. Each component shall be evaluated for all of the applicable failure modes in 5.1.1.2 using the methodology of Part 4 or Part 5. If the failure mode is not addressed in Part 4 (e.g., ratcheting), then the analysis shall be in accordance with Part 5. Structural interactions between components shall be considered.

4.1.2 MINIMUM THICKNESS REQUIREMENTS

Except for the special provisions listed below, the minimum thickness permitted for shells and heads, after forming and regardless of product form and material, shall be 1.6 mm (0.0625 in.) exclusive of any corrosion allowance. Exceptions are:

(a) This minimum thickness does not apply to heat transfer plates of plate-type heat exchangers.

(b) This minimum thickness does not apply to the inner pipe of double pipe heat exchangers nor to pipes and tubes that are enclosed and protected by a shell, casing or ducting, where such pipes or tubes are DN 150 (NPS 6) and less. This exemption applies whether or not the outer pipe or shell is constructed to Code rules. All other pressure parts of these heat exchangers that are constructed to Code rules must meet the 1.6 mm (0.0625 in.) minimum thickness requirements.

(c) The minimum thickness of shells and heads used in compressed air service, steam service, and water service, made from carbon or low alloy steel materials shall be 2.4 mm (0.0938 in.) exclusive of any corrosion allowance.

(d) This minimum thickness does not apply to the tubes in air cooled and cooling tower heat exchangers if all of the following provisions are met:

(1) The tubes shall be protected by fins or other mechanical means.

(2) The tube outside diameter shall be a minimum of 10 mm (0.375 in.) and a maximum of 38 mm (1.5 in.).

(3) The minimum thickness used shall not be less than that calculated by the equations given in 4.3 and in no case less than 0.5 mm (0.022 in.).

4.1.3 MATERIAL THICKNESS REQUIREMENTS

4.1.3.1 Allowance for Fabrication. The selected thickness of material shall be such that the forming, heat treatment, and other fabrication processes will not reduce the thickness of the material at any point below the minimum required thickness.

4.1.3.2 Mill Undertolerance. Plate material shall be ordered not thinner than the design thickness. Vessels made of plate furnished with an undertolerance of not more than the smaller value of 0.3 mm (0.01 in.) or 6% of the ordered thickness may be used at the full maximum allowable working pressure for the thickness ordered. If the specification to which the plate is ordered allows a greater undertolerance, the ordered thickness of the materials shall be sufficiently greater than the design thickness so that the thickness of the material furnished is not more than the smaller of 0.3 mm (0.01 in.) or 6% under the design thickness.

4.1.3.3 Pipe Undertolerance. If pipe or tube is ordered by its nominal wall thickness, the manufacturing undertolerance on wall thickness shall be taken into account. After the minimum wall thickness is determined, it shall be increased by an amount sufficient to provide the manufacturing undertolerance allowed in the pipe or tube specification.

4.1.4 CORROSION ALLOWANCE IN DESIGN EQUATIONS

4.1.4.1 The dimensional symbols used in all design equations and figures throughout this Division represent dimensions in the corroded condition.

4.1.4.2 The term corrosion allowance as used in this Division is representative of loss of metal by corrosion, erosion, mechanical abrasion, or other environmental effects and shall be accounted for in the design of vessels or parts when specified in the User's Design Specification.

4.1.4.3 The user shall determine the required corrosion allowance over the life of the vessel and specify such in the User's Design Specification. The Manufacturer shall add the required allowance to all minimum required thicknesses in order to arrive at the minimum ordered material thickness. The corrosion allowance need not be the same for all parts of a vessel. If corrosion or other means of metal loss do not exist, then the user shall specify in the User's Design Specification that a corrosion allowance is not required.

4.1.5 DESIGN BASIS

4.1.5.1 Design Thickness. The design thickness of the vessel part shall be determined using the rules specified in 4.1.5.1.1 or 4.1.5.1.2 and shall not be less than the minimum thickness specified in 4.1.2 plus any corrosion allowance required by 4.1.4.

4.1.5.1.1 Class 1. The design-by-rule methods of Part 4 shall be applied using the load and load case combinations specified in 4.1.5.3 except when design rules are not provided in Part 4 (see 4.1.1.2).

4.1.5.1.2 Class 2. The design-by-rule methods of Part 4 shall be applied using the load and load case combinations specified in 4.1.5.3. Alternatively, the design thickness may be established using the design-by-analysis procedures in Part 5, even if this thickness is less than that established using Part 4 design-by-rule methods.

4.1.5.2 Definitions. The following definitions shall be used to establish the design basis of the vessel. Each of these parameters shall be specified in the User's Design Specification.

(a) Design Pressure - The pressure used in the design of a vessel component together with the coincident design metal temperature, for the purpose of determining the minimum permissible thickness or physical characteristics of the different zones of the vessel. Where applicable, static head and other static or dynamic loads shall be included in addition to the specified design pressure [2.2.3.1(d)(1)] in the determination of the minimum permissible thickness or physical characteristics of a particular zone of the vessel.

(b) Maximum Allowable Working Pressure - The maximum gage pressure permissible at the top of a completed vessel in its normal operating position at the designated coincident temperature for that pressure. This pressure is the least of the values for the internal or external pressure to be determined by the rules of this Division for any of the pressure

boundary parts, considering static head thereon, using nominal thicknesses exclusive of allowances for corrosion and considering the effects of any combination of loadings specified in the User's Design Specification at the designated coincident temperature. It is the basis for the pressure setting of the pressure-relieving devices protecting the vessel. The specified design pressure may be used in all cases in which calculations are not made to determine the value of the maximum allowable working pressure.

(c) Test Pressure - The test pressure is the pressure to be applied at the top of the vessel during the test. This pressure plus any pressure due to static head at any point under consideration is used in the applicable design equations to check the vessel under test conditions.

(d) Design Temperature and Coincident Pressure - The design temperature for any component shall not be less than the mean metal temperature expected coincidentally with the corresponding maximum pressure (internal and, if specified, external). If necessary, the mean metal temperature shall be determined by computations using accepted heat transfer procedures or by measurements from equipment in service under equivalent operating conditions. In no case shall the metal temperature anywhere within the wall thickness exceed the maximum temperature limit in (1).

(1) A design temperature greater than the maximum temperature listed for a material specification in Annex 3-A is not permitted. In addition, if the design includes external pressure (see 4.4), then the design temperature shall not exceed the temperature limits specified in Table 4.4.1.

(2) The maximum design temperature marked on the nameplate shall not be less than the expected mean metal temperature at the corresponding MAWP.

(3) When the occurrence of different mean metal temperatures and coincident pressures during operation can be accurately predicted for different zones of a vessel, the design temperature for each of these zones may be based on the predicted temperatures. These additional design metal temperatures with their corresponding MAWP, may be marked on the nameplate as required.

(e) Minimum Design Metal Temperature and Coincident Pressure - The minimum design metal temperature (MDMT) shall be the coldest expected in normal service, except when colder temperatures are permitted by 3.11. The MDMT shall be determined by the principles described in (d). Considerations shall include the coldest operating temperature, operational upsets, auto refrigeration, atmospheric temperature, and any source of cooling.

(1) The MDMT marked on the nameplate shall correspond to a coincident pressure equal to the MAWP.

(2) When there are multiple MAWP, the largest value shall be used to establish the corresponding MDMT marked on the nameplate.

(3) When the occurrence of different MDMT and coincident pressures during operation can be accurately predicted for different zones of a vessel, the MDMT for each of these zones may be based on the predicted temperatures. These additional MDMT together with their corresponding MAWP, may be marked on the nameplate as required.

(**21**) **4.1.5.3 Design Loads and Load Case Combinations.** All applicable loads and load case combinations shall be considered in the design to determine the minimum required wall thickness for a vessel part.

(a) The loads that shall be considered in the design shall include, but not be limited to, those shown in Table 4.1.1 and shall be included in the User's Design Specification.

(b) The load combinations that shall be considered shall include, but not be limited to, those shown in Table 4.1.2, except when a different recognized standard for wind loading is used. In that case, the User's Design Specification shall cite the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7. The factors for wind loading, W, in Table 4.1.2, Design Load Combinations, are based on ASCE/SEI 7 wind maps and probability of occurrence. If a different recognized standard for earthquake loading is used, the User's Design Specification shall cite the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7.

(c) When analyzing a loading combination, the value of allowable stress shall be evaluated at the coincident temperature.

(d) Combinations of loads that result in a maximum thickness shall be evaluated. In evaluating load cases involving the pressure term, P, the effects of the pressure being equal to zero shall be considered. For example, the maximum difference in pressure that may exist between the inside and outside of the vessel at any point or between two chambers of a combination unit or, the conditions of wind loading with an empty vertical vessel at zero pressure may govern the design.

(e) The applicable loads and load case combinations shall be specified in the User's Design Specification.

(f) If the vessel or part is subject to cyclic operation and a fatigue analysis is required (see 4.1.1.4), then a pressure cycle histogram and corresponding thermal cycle histogram shall be provided in the User's Design Specification.

4.1.6 DESIGN ALLOWABLE STRESS

4.1.6.1 Design Condition. The allowable stresses for all permissible materials of construction are provided in Annex 3-A. The wall thickness of a vessel computed by the rules of Part 4 for any combination of loads (see 4.1.5) that induce primary stress (see definition of *primary stress* in 5.12) and are expected to occur simultaneously during operation shall satisfy the equations shown below.

$$P_m \le S \tag{4.1.1}$$

$$P_m + P_b \le 1.5S \tag{4.1.2}$$

4.1.6.2 Test Condition. The allowable stress for the selected test pressure, P_T, shall be established by the following requirements. In the case where the stress limit is exceeded, the selected test pressure shall be reduced, but no lower than the minimum test pressure as established in Part 8.

(a) Primary Membrane Stress. A calculated P_m shall not exceed the applicable limit given below, where β_T shall be obtained from Table 4.1.3 for the appropriate test medium (hydrostatic or pneumatic) and the applicable class.

$$P_m \le \beta_T S_y \tag{4.1.3}$$

(b) Primary Membrane Stress Plus Primary Bending Stress. A calculated $P_m + P_b$ shall not exceed the applicable limit given below, where β_T and $\gamma_{\min}$ shall be obtained from Table 4.1.3 for the appropriate test medium (hydrostatic or pneumatic) and the applicable class.

(1) For $P_m \le \frac{1}{1.5} S_y$

$$P_m + P_b \le \gamma_{\min} S_y \tag{4.1.4}$$

(2) For $\frac{1}{1.5} S_y < P_m \le \beta_T S_y$

$$P_m + P_b \le \left(\frac{1 - \gamma_{\min}}{\beta_T - \frac{1}{1.5}}\right) P_m - \left[\left(\frac{1 - \gamma_{\min}}{\beta_T - \frac{1}{1.5}}\right)\beta_T - 1\right] S_y \tag{4.1.5}$$

4.1.6.3 Primary Plus Secondary Stress. The allowable primary plus secondary stress at the design temperature shall be computed as follows:

$$S_{PS} = \max.\left[3S,\, 2S_y\right] \tag{4.1.6}$$

However, S_{PS} shall be limited to $3S$ if either

(a) the room temperature ratio of the minimum specified yield strength from Annex 3-D to the ultimate tensile strength from Annex 3-D exceeds 0.70; or,

(b) the allowable stress from Annex 3-A is governed by time-dependent properties.

4.1.6.4 Shear Stress. The maximum shear stress in restricted shear, such as dowel bolts or similar construction in which the shearing member is so restricted that the section under consideration would fail without a reduction of area, shall be limited to 0.80 times the values in Section II, Part D, Subpart 1, Table 3.

4.1.6.5 Bearing Stress. Maximum bearing stress shall be limited to 1.60 times the values in Section II, Part D, Subpart 1, Table 3.

4.1.7 MATERIALS IN COMBINATION

Except as specifically prohibited by the rules of this Division, a vessel may be designed for and constructed of any combination of materials listed in Part 3. For vessels operating at temperatures other than ambient temperature, the effects of differences in coefficients of thermal expansion of dissimilar materials shall be considered.

4.1.8 COMBINATION UNITS

4.1.8.1 Combination Unit. A combination unit is a pressure vessel that consists of more than one independent or dependent pressure chamber, operating at the same or different pressures and temperatures. The parts separating each pressure chamber are the common elements. Each element, including the common elements, shall be designed for at least the most severe condition of coincident pressure and temperature expected in normal operation. Only the chambers that come within the scope of this Division need be constructed in compliance with its provisions. Additional design requirements for chambers classified as jacketed vessels are provided in 4.11.

4.1.8.2 Common Element Design. It is permitted to design each common element for a differential pressure less than the maximum of the design pressures of its adjacent chambers (differential pressure design) or a mean metal temperature less than the maximum of the design temperatures of its adjacent chambers (mean metal temperature design), or both, only when the vessel is to be installed in a system that controls the common element operating conditions.

(a) Differential Pressure Design (Dependent Pressure Chamber). When differential pressure design is permitted, the common element design pressure shall be the maximum differential design pressure expected between the adjacent chambers. The common element and its corresponding differential pressure shall be indicated in the "Remarks" section of the Manufacturer's Data Report (see 2.3.4) and marked on the vessel (see Annex 2-F). The differential pressure shall be controlled to ensure the common element design pressure is not exceeded.

(b) Mean Metal Temperature Design (Dependent Pressure Chamber). When mean metal temperature design is used, the maximum common element design temperature determined in accordance with 4.1.5.2(d) may be less than the greater of the maximum design temperatures of its adjacent chambers; however, it shall not be less than the lower of the maximum design temperatures of its adjacent chambers. The common element and its corresponding design temperature shall be indicated in the "Remarks" section of the Manufacturer's Data Report (see 2.3.4) and marked on the vessel (see Annex 2-F). The fluid temperature, flow and pressure, as required, shall be controlled to ensure the common element design temperature is not exceeded.

4.1.9 CLADDING AND WELD OVERLAY

4.1.9.1 The design calculations for integrally clad plate or overlay weld clad plate may be based on a thickness equal to the nominal thickness of the base plate plus S_C/S_B times the nominal thickness of the cladding, less any allowance provided for corrosion, provided all of the following conditions are met.

(a) The clad plate conforms to one of the specifications listed in the tables in Part 3 or is overlay weld clad plate conforming to Part 3.

(b) The joints are completed by depositing corrosion resisting weld metal over the weld in the base plate to restore the cladding.

(c) The allowable stress of the weaker material is at least 70% of the allowable stress of the stronger material.

4.1.9.2 When S_C is greater than S_B, the multiplier S_C/S_B shall be taken equal to unity.

4.1.10 INTERNAL LININGS

Corrosion resistant or abrasion resistant linings are those not integrally attached to the vessel wall, i.e., they are intermittently attached or not attached at all. In either case, such linings shall not be given any credit when calculating the thickness of the vessel wall.

4.1.11 FLANGES AND PIPE FITTINGS

(**21**) **4.1.11.1** The following standards covering flanges and pipe fittings are acceptable for use under this Division in accordance with the requirements of Part 1.

(a) ASME B16.5, Pipe Flanges and Flanged Fittings, NPS $^1/_2$ Through NPS 24 Metric/Inch Standard

(b) ASME B16.9, Factory-Made Wrought Buttwelding Fittings

(c) ASME B16.11, Forged Fittings, Socket-Welding and Threaded

(d) ASME B16.15, Cast Copper Alloy Threaded Fittings, Classes 125 and 250

(e) ASME B16.20, Metallic Gaskets for Pipe Flanges

(f) ASME B16.24, Cast Copper Alloy Pipe Flanges, Flanged Fittings, and Valves, Classes 150, 300, 600, 900, 1500, and 2500

(g) ASME B16.47, Large Diameter Steel Flanges, NPS 26 Through NPS 60 Metric/Inch Standard

4.1.11.2 Pressure–temperature ratings shall be in accordance with the applicable standard except that the pressure–temperature ratings for ASME B16.9 and ASME B16.11 fittings shall be calculated as for straight seamless pipe in accordance with the rules of this Division including the maximum allowable stress for the material.

4.1.11.3 A forged nozzle flange (i.e., long weld neck flange) may be designed using the ASME B16.5/B16.47 pressure–temperature ratings for the flange material being used, provided all of the following are met.

(a) For ASME B16.5 applications, the forged nozzle flange shall meet all dimensional requirements of a flanged fitting given in ASME B16.5 with the exception of the inside diameter. The inside diameter of the forged nozzle flange shall not exceed the inside diameter of the same size and class lap joint flange given in ASME B16.5. For ASME B16.47 applications, the inside diameter shall not exceed the weld hub diameter "A" given in the ASME B16.47 tables.

(b) For ASME B16.5 applications, the outside diameter of the forged nozzle neck shall be at least equal to the hub diameter of the same size and class ASME B16.5 lap joint flange. For ASME B16.47 applications, the outside diameter of the hub shall at least equal the "X" diameter given in the ASME B16.47 tables. Larger hub diameters shall be limited to nut stop diameter dimensions (see 4.16).

4.1.12 VESSELS IN ELEVATED TEMPERATURE SERVICE

The user and Manufacturer are cautioned that certain fabrication details allowed by this Division may result in cracking at welds and associated heat-affected zone (HAZ) for vessels designed for use at elevated temperature.

NOTE: WRC Bulletin 470, "Recommendations for Design of Vessels for Elevated Temperature Service," has information that may prove helpful to the vessel designer. WRC Bulletin 470 contains recommended design details for use at elevated temperature service, which is for the purposes of this Division, when the allowable stresses in Section II, Part D are based on time-dependent properties. The use of these details does not relieve the Manufacturer of design responsibility with regard to primary, secondary, and peak stresses associated with both steady-state conditions and transient events, such as startup, shutdown, intermittent operation, thermal cycling, etc., as defined in the User's Design Specification.

4.1.13 NOMENCLATURE

P_b = primary bending stress (see Part 5)
P_m = general primary membrane stress (see Part 5)
P_T = selected hydrostatic or pneumatic test pressure [see 8.2.1(c)]
S = allowable stress from Annex 3-A at the design temperature
S_B = allowable stress from Annex 3-A at the design temperature for the base plate at the design temperature
S_C = allowable stress from Annex 3-A at the design temperature for the cladding or, for the weld overlay, the allowable stress of the wrought material whose chemistry most closely approximates that of the cladding at the design temperature
S_{PS} = allowable primary plus secondary stress at the design temperature
S_y = yield strength at the test temperature evaluated in accordance with Annex 3-D
β = elastic–plastic load factor for Class 1 or Class 2 construction (see Table 4.1.3)
β_T = test condition load factor for hydrostatic or pneumatic test and for Class 1 or Class 2 construction (see Table 4.1.3)
γ_{min} = minimum test condition load factor for hydrostatic or pneumatic test and for Class 1 or Class 2 construction (see Table 4.1.3)
$\gamma_{St/S}$ = test condition load factor considering the ratio of the allowable stress at the test condition to the allowable stress at the design condition for hydrostatic or pneumatic test and for Class 1 or Class 2 construction (see Table 4.1.3)
Ω_P = load factor for pressure when combined with occasional load L, S_s, W, or E (see Table 4.1.1 for load parameter definitions)
= 1.0 unless otherwise specified in the User's Design Specification [see 2.2.3.1(e)]
$\Omega_P P$ = maximum anticipated operating pressure (internal or external) acting simultaneously with occasional load L, S_s, W, or E

4.1.14 TABLES

Table 4.1.1
Design Loads

Design Load Parameter	Description
P	Internal or external specified design pressure (see 4.1.5.2(a))
P_S	Static head from liquid or bulk materials (e.g., catalyst)
D	Deadweight of the vessel, contents, and appurtenances at the location of interest, including the following: • Weight of vessel including internals, supports (e.g., skirts, lugs, saddles, and legs), and appurtenances (e.g., platforms, ladders, etc.) • Weight of vessel contents under design and test conditions • Refractory linings, insulation • Static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping • Transportation loads (the static forces obtained as equivalent to the dynamic loads experienced during normal operation of a transport vessel [see 1.2.1.3(b)])
L	• Appurtenance live loading • Effects of fluid flow, steady state or transient • Loads resulting from wave action
E	Earthquake loads [see 4.1.5.3(b)]
W	Wind loads [see 4.1.5.3(b)]
S_s	Snow loads
F	Loads due to deflagration

(21)

Table 4.1.2
Design Load Combinations

Design Load Combination [Note (1)], [Note (2)]	General Primary Membrane Allowable Stress [Note (3)]
(1) $P + P_s + D$	S
(2) $P + P_s + D + L$	S
(3) $P + P_s + D + S_s$	S
(4) $\Omega_P P + P_s + D + 0.75L + 0.75S_s$	S
(5) $\Omega_P P + P_s + D + (0.6W \text{ or } 0.7E)$	S
(6) $\Omega_P P + P_s + D + 0.75(0.6W \text{ or } 0.7E) + 0.75L + 0.75S_s$	S
(7) $0.6D + (0.6W \text{ or } 0.7E)$ [Note (4)]	S
(8) $P_s + D + F$	See Annex 4-D
(9) Other load combinations as defined in the User's Design Specification	S

NOTES:
(1) The parameters used in the Design Load Combination column are defined in Table 4.1.1.
(2) See 4.1.5.3 for additional requirements.
(3) S is the allowable stress for the load case combination [see 4.1.5.3(c)].
(4) This load combination addresses an overturning condition for foundation design. It does not apply to design of anchorage (if any) to the foundation. Refer to ASCE/SEI 7, 2.4.1 Exception 2 for an additional reduction to W that may be applicable.

Table 4.1.3
Load Factor, β, and Pressure Test Factors, β_T, γ_{min}, and $\gamma_{St/S}$, for Class 1 and Class 2 Construction and Hydrostatic or Pneumatic Testing

		β_T		γ_{min}		$\gamma_{St/S}$	
Class	β	Hydrostatic	Pneumatic	Hydrostatic	Pneumatic	Hydrostatic	Pneumatic
1	3.0	0.95	0.8	$1.5\beta_T$	$1.5\beta_T$	1.25	1.15
2	2.4	0.95	0.8	$1.5\beta_T$	$1.5\beta_T$	1.25	1.15

4.2 DESIGN RULES FOR WELDED JOINTS

4.2.1 SCOPE

Design requirements for weld joints are provided in 4.2. Acceptable weld joint details are provided for most common configurations. Alternative details may be used if they can be qualified by a design procedure using Part 5. Rules for sizing welds are also provided.

4.2.2 WELD CATEGORY

The term weld category defines the location of a joint in a vessel, but not the weld joint type. The weld categories established by this paragraph are for use elsewhere in this Division in specifying special requirements regarding joint type and degree of examination for certain welded pressure joints. Since these special requirements that are based on thickness do not apply to every welded joint, only those joints to which special requirements apply are included in categories. The weld categories are defined in Table 4.2.1 and shown in Figure 4.2.1. Welded joints not defined by the category designations include but are not limited to Table 4.11.1, jacket-closure-to-shell welds; Figure 4.19.11, groove and fillet welds; and Figure 4.20.1, sketches (a), (c), and (d) corner joints. Unless limited elsewhere in this Division, 4.2.5 permissible weld joint types may be used with welded joints that are not assigned a category.

4.2.3 WELD JOINT TYPE

The weld joint type defines the type of weld between pressure and/or nonpressure parts. The definitions for the weld joint types are shown in Table 4.2.2.

4.2.4 WELD JOINT EFFICIENCY

The weld joint efficiency of a welded joint is expressed as a numerical quantity and is used in the design of a joint as a multiplier of the appropriate allowable stress value taken from Annex 3-A. The weld joint efficiency shall be determined from Table 7.2.

4.2.5 TYPES OF JOINTS PERMITTED

4.2.5.1 Definitions

(a) Butt Joint - A butt joint is a connection between the edges of two members with a full penetration weld. The weld is a double sided or single sided groove weld that extends completely through both of the parts being joined.

(b) Corner Joint - A corner joint is a connection between two members at right angles to each other in the form of an L or T that is made with a full or partial penetration weld, or fillet welds. Welds in full penetration corner joints shall be groove welds extending completely through at least one of the parts being joined and shall be completely fused to each part.

(c) Angle Joint - An angle joint is a connection between the edges of two members with a full penetration weld with one of the members consisting of a transition of diameter. The weld is a double sided or single sided groove weld that extends completely through both of the parts being joined.

(d) Spiral Weld - a weld joint having a helical seam.

(e) Fillet Weld - A fillet weld is a weld that is approximately triangular in cross section that joins two surfaces at approximately right angles to each other.

(f) Gross Structural Discontinuity - A gross structural discontinuity is a source of stress or strain intensification which affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples of gross structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

(g) Lightly Loaded Attachments - Weld stress due to mechanical loads on attached member not over 25% of allowable stress for fillet welds and temperature difference between shell and attached member not expected to exceed 14°C (25°F) shall be considered lightly loaded.

(h) Minor Attachments - Parts of small size, less than or equal to 10 mm (0.375 in.) thick or 82 cm^3 (5 $in.^3$) in volume, that carry no load or an insignificant load such that a stress calculation in designer's judgment is not required; examples include nameplates, insulation supports, and locating lugs.

(i) Major Attachments - Parts that are not minor or lightly loaded as described above.

4.2.5.2 Category A Locations

(a) All joints of Category A shall be Type No. 1 butt joints.

(b) Acceptable Category A welds are shown in Tables 4.2.4 and 4.2.5.

(c) Transition Joints Between Sections of Unequal Thickness - Unless the requirements of Part 5 are shown to be satisfied, a tapered transition shall be provided at joints between sections that differ in thickness by more than one-fourth of the thickness of the thinner section or by more than 3 mm (0.125 in.). The transition may be formed by any process that will provide a uniform taper. When Part 5 is not used, the following additional requirements shall also apply.

(1) When a taper is required on any shell section intended for butt-welded attachment, the transition geometry shall be in accordance with Table 4.2.4, Details 4, 5, and 6.

(2) When a taper is required on a hemispherical head intended for butt-welded attachment, the transition geometry shall be in accordance with Table 4.2.5, Details 2, 3, 4 and 5.

(3) A hemispherical head which has a greater thickness than a cylinder of the same inside diameter may be machined to the outside diameter of the cylinder, provided the remaining thickness is at least as great as that required for a shell of the same diameter.

(4) When the transition is formed by adding additional weld metal beyond what would otherwise be the edge of the weld, such additional weld metal buildup shall be subject to the requirements of Part 6. The butt weld may be partly or entirely in the tapered section.

(5) The requirements of this paragraph do not apply to flange hubs.

4.2.5.3 Category B Locations

(a) The joints of Category B may be any of the following types:

(1) Type No. 1 butt joints,

(2) Type No. 2 butt joints except as limited in 4.2.5.7,

(3) Type No. 3 butt joints may only be used for shells having a thickness of 16 mm (0.625 in.) or less and a diameter of 610 mm (or 24 in.) and less.

(b) Acceptable Category B welds are shown in Tables 4.2.4 and 4.2.5.

(c) Backing strips shall be removed from Type No. 2 butt joints unless access conditions prevent their removal. If a fatigue analysis of Type No. 2 butt joints with a backing strip in place is required, then a stress concentration factor of 2.0 for membrane stresses and of 2.5 for bending stress shall be applied.

(d) Transition joints between shell sections of unequal thickness shall meet the requirements of 4.2.5.2(c) and shall be in accordance with Table 4.2.4 and Table 4.2.5. An ellipsoidal head which has a greater thickness than a cylinder of the same inside diameter may be machined to the outside diameter of the cylinder, provided the remaining thickness is at least as great as that required for a shell of the same diameter.

(e) Transition joints between nozzle necks and attached piping of unequal thickness shall be in accordance with 4.2.5.9.

(f) When butt joints are required elsewhere in this Division for Category B, an angle joint connecting a transition in diameter to a cylinder shall be considered as meeting this requirement, provided the requirements of Type No. 1 butt joints are met. All requirements pertaining to the butt joint shall apply to the angle joint.

4.2.5.4 Category C Locations

(a) The joints of Category C may be any of the following types:

(1) Type No. 1 butt joints,

(2) Full penetration corner joints except as limited in 4.2.5.7.

(3) Fillet welded joints for the attachment of loose type flanges shown in Table 4.2.9, with the following limitations:

(-a) The materials of the flange and the part it is welded to are Type 1 Materials (see Table 4.2.3).

(-b) The minimum specified yield strength of both materials is less than 552 MPa (80 ksi).

(-c) The minimum elongation of both materials is 12% in 50 mm (2 in.) gauge length.

(-d) The thickness of the materials to which the flange is welded does not exceed 32 mm (1.25 in.).

(-e) The fillet weld dimensions satisfy the requirements shown in Table 4.2.9.

(-f) A fatigue-screening criterion shall be performed in accordance with 5.5.2 to determine if a fatigue analysis is required. If the results of this screening indicate that a fatigue analysis is required, then the analysis shall be performed in accordance with 5.5.2.

(-g) Loose type flanges that do not conform to ASME B16.5 are only permitted when both of the following requirements are satisfied.

(-1) The material of construction for the flange satisfies the following equation.

$$\frac{S_{yT}}{S_u} \leq 0.625 \tag{4.2.1}$$

(-2) The component is not in cyclic service, i.e., a fatigue analysis is not required in accordance with 4.1.1.4.

(b) Acceptable Category C welds are shown in the following tables.

(1) Table 4.2.4 - Some acceptable weld joints for shell seams.

(2) Table 4.2.6 - Some acceptable weld joints for unstayed flat heads, tubesheets without a bolting flange, and side plates of rectangular pressure vessels

(3) Table 4.2.7 - Some acceptable weld joints with butt weld hubs.

(4) Table 4.2.8 - Some acceptable weld joints for attachment of tubesheets with a bolting flange

(5) Table 4.2.9 - Some acceptable weld joints for flange attachments.

(c) Flat Heads, Lap Joint Stub Ends, and Tubesheets with Hubs for Butt Joints

(1) Hubs for butt welding to the adjacent shell, head, or other pressure parts such as tubesheets and flat heads as shown in Table 4.2.7 shall be forged or machined from flat plate. Forged hubs shall be forged in such a manner as to provide in the hub the full minimum tensile strength and elongation specified for the material in the direction parallel to the axis of the vessel. Proof of this shall be furnished by a tension test specimen (subsize, if necessary) taken in this direction and as close to the hub as practical. Hubs machined from flat plates should satisfy the requirements of 3.9.

(2) Flanges with hubs as shown in Table 4.2.9, Details 6, 7, and 8 shall not be machined from plate.

(d) Corner Joints - If shells, heads, or other pressure parts are welded to a forged or rolled plate to form a corner joint as shown in Table 4.2.6 and Table 4.2.8, then the welds shall meet the following requirements.

(1) On the cross section through the welded joint, the line between the weld metal and the forged or rolled plate being attached shall be projected on planes both parallel to and perpendicular to the surface of the plate being attached, in order to determine the dimensions a and b, respectively.

(2) The dimensional requirements on a and b shall meet the applicable requirements in Tables 4.2.6 and 4.2.8.

(3) Weld joint details that have a dimension through the joint that is less than the thickness of the shell, head, or other pressure part, or that provide attachment eccentric thereto are not permitted.

(4) If an integral tubesheet is located between two shells, heads, or other pressure parts, then a weld attachment detail as shown in Table 4.2.6 shall be used for each attachment.

4.2.5.5 Category D Locations

(a) The joints of Category D may be any of the following types.

(1) Type No. 1 butt joints

(2) Full penetration corner joints except as limited in 4.2.5.7

(3) Full penetration corner joints at the nozzle neck or fillet welds, or both

(4) Partial penetration corner joint at the nozzle neck

(b) Acceptable Category D welds are shown in the following tables.

(1) Table 4.2.4 - Some acceptable weld joints for shell seams

(2) Table 4.2.10 - Some acceptable full penetration welded nozzle attachments not readily radiographable

(3) Table 4.2.11 - Some acceptable pad welded nozzle attachments and other connections to shells

(4) Table 4.2.12 - Some acceptable fitting-type welded nozzle attachments and other connections to shells

(5) Table 4.2.13 - Some acceptable welded nozzle attachments that are readily radiographable

(6) Table 4.2.14 - Some acceptable partial penetration nozzle attachments

(c) Requirements for nozzle welds are shown below.

(1) Type No. 1 butt joints or full penetration joints shall be used when the opening in a shell is 64 mm (2.5 in.) or more in thickness.

(2) Nozzle Neck Abutting the Vessel Wall Without Reinforcement - Nozzle necks abutting the vessel wall without added reinforcing element shall be attached by a full penetration groove weld. Backing strips shall be used with welds deposited from only one side or when complete joint penetration cannot be verified by visual inspection. Backing strips, when used, shall be removed after welding. Permissible types of weld attachments are shown in Table 4.2.10, Details 1, 2, and 7.

(3) Insert Nozzle Necks Without Reinforcement - Nozzle necks without added reinforcing elements inserted partially into or through a hole cut in the vessel wall shall be attached by a full penetration groove weld. Backing strips, when used, shall be removed after welding. Permissible types of weld attachments are shown in Table 4.2.10, Details 3, 4, 5, 6, and 8.

(4) Insert Nozzle Necks With Reinforcement - Inserted type necks having added reinforcement in the form of one or more separate reinforcing plates shall be attached by continuous welds at the outer edge of the reinforcement plate and at the nozzle neck periphery. A fatigue-screening criterion shall be applied to nozzles with separate reinforcement and non-integral attachment designs. The welds attaching the neck to the vessel wall and to the reinforcement shall be full penetration groove welds. Permissible types of weld attachments are shown in Table 4.2.11, Details 1, 2, and 3. [Also, see (d).]

(5) Studded Pad Type Connections - Studded connections that may have externally imposed loads shall be attached using full penetration welds in accordance with Table 4.2.11, Detail 5. Studded pad type connections on which there are essentially no external loads, such as manways and handholes used only as inspection openings, thermowell connections, etc., may be attached using fillet weld in accordance with Table 4.2.11, Detail 4.

(6) Fittings With Internal Threads - Internally threaded fittings shall be limited to NPS 2 or smaller. Permissible types of weld attachments are shown in Table 4.2.12.

(7) Nozzles With Integral Reinforcement - Nozzles having integral reinforcement may be attached using butt welds of Type No.1. Nozzles or other connections with integral reinforcement that are attached with corner welds shall be attached by means of full penetration corner welds. Permissible types of weld attachments are shown in Table 4.2.13.

(8) Nozzle Attached With Partial Penetration Welds - Partial penetration welds may be used only for nozzle attachments, such as instrumentation openings, inspection openings, etc., on which there are essentially no external mechanical loadings and on which there will be no thermal stresses greater than in the vessel itself. Permissible types of weld attachments are shown in Table 4.2.14. If Table 4.2.14, Details 3 and 4 are used, then the material in the neck shall not be included in the reinforcement area calculation (see 4.5).

(d) Except for nozzles at small ends of cones reinforced in accordance with the requirements of 4.3.11, 4.3.12, 4.4.13, and 4.4.14, as applicable, added reinforcement in the form of separate reinforcing plates or pads may be used, provided the vessel and nozzles meet all of the following requirements.

(1) The materials of the nozzle, pad, and vessel wall conform to those listed in Section IX, Table QW/QB-422 for Material Types 1 and 4 shown in Table 4.2.3.

(2) The specified minimum tensile strength of the nozzle, pad, and vessel wall materials does not exceed 550 MPa (80 ksi).

(3) The minimum elongation of the nozzle, pad, and vessel wall materials is 12% in 50 mm (2 in.).

(4) The thickness of the added reinforcement does not exceed 1.5 times the vessel wall thickness.

(5) The requirements of 5.5 for pads, i.e., non-integral construction, in cyclic service are met.

4.2.5.6 Category E Locations

(a) Method of Attachment - Attachment of nonpressure parts shall be in accordance with the following requirements.

(1) Nonpressure parts, supports, lugs, brackets, and stiffeners may be attached to the inside or outside wall using butt welds, full penetration groove welds, partial penetration welds, fillet welds, or stud welds as limited in the subsequent paragraphs.

(2) Resistance-welded studs may be used for minor attachments to nonpressure parts for all materials except those included in Material Type 3 (see Table 4.2.3).

(3) Supports, lugs, brackets, stiffeners, and other attachments may be attached with stud bolts to the outside or inside of a vessel wall (see 4.15.5).

(4) All attachments shall conform to the curvature of the shell to which they are to be attached.

(5) All welds joining minor attachments [see 4.2.5.1(g)] to pressure parts may be continuous or non-continuous for Material Types 1, 2, and 4 (see Table 4.2.3).

(6) All welds joining nonpressure parts to pressure parts shall be continuous for Material Type 3 (see Table 4.2.3).

(7) Some acceptable types of attachment weld and associated minimum weld sizes are shown in Figure 4.2.2, see (e) and (f) for limitations.

(8) Some acceptable methods of attaching stiffening rings are shown in Figure 4.2.3, see (e) and (f) for limitations.

(b) Materials for Major Attachments to Pressure Parts - Attachments welded directly to pressure parts shall be of a material listed in Annex 3-A.

(1) The material and the deposited weld metal shall be compatible with that of the pressure part.

(2) For Material Type 3 (see Table 4.2.3), all permanent structural attachments other than minor attachments described in (c) below [see 3.5.3.3 and 4.2.5.1(h)] that are welded directly to pressure parts shall be made of materials whose specified minimum yield strength is within ±20% of that of the material to which they are attached. An exception to this requirement is that lightly loaded attachments of non-hardenable austenitic stainless steels conforming to SA-240, SA-312, or SA-479 are permitted to be fillet welded to pressure parts conforming to SA-353; SA-553 Types I, II, and III; or SA-645.

(c) Materials for Minor Attachments to Pressure Parts - Except as limited by (b) or for forged fabrication (see 6.7), minor attachments made from material that does not conform to a material specification permitted in this Division may be used and may be welded directly to the pressure part, provided the requirements shown below are satisfied.

(1) The material is identified and is suitable for welding in accordance with 3.2.1.3.

(2) The material is compatible insofar as welding is concerned with that to which the attachment is to be made.

(3) The welds are postweld heat treated when required in Part 6.

(d) Materials for Attachments Welded to Nonpressure Parts - Attachments welded to nonpressure parts made from material that does not conform to a material specification permitted in this Division may be used, provided the material is identified, is suitable for welding, and is compatible with the material to which attachment is made.

(e) Attachment Welds to Pressure Parts of Material Types 1 and 4 (see Table 4.2.3) - Welds attaching nonpressure parts or stiffeners to pressure parts shall be one of the following:

(1) A fillet weld not over 13 mm (0.5 in.) leg dimension and the toe of the weld not closer than $\sqrt{Rt_{sh}}$ from a gross structural discontinuity

(2) A partial penetration weld plus fillet weld; this is limited to the attachment of parts not exceeding 38 mm (1.5 in.) in thickness

(3) A full penetration groove weld plus a fillet weld on each side

(4) A full penetration butt weld; the prior deposition of weld metal to provide a boss for the butt weld is permissible, provided it is checked for soundness by suitable nondestructive examination. Heat treatment for the weld build-up region shall be considered.

(5) For attachment of support skirts or other supports involving similar attachment orientation, in addition to the weld types of (3) and (4), welds of greater effective throat dimension than 90 deg fillet welds, as obtained by increased leg dimension or angle and bevel of parts joined, may be used where the effective throat is t_a (see Figure 4.2.4); however, the limitation on thickness in (2) shall apply.

(6) Stiffening rings may be stitch welded when the material of construction satisfies Eq. (4.2.1) and the component is not in cyclic service, i.e., a fatigue analysis is not required in accordance with 4.1.1.4.

(f) Attachment Welds to Pressure Parts of Material Types 2 and 3 (see Table 4.2.3) - Welds attaching nonpressure parts or stiffeners to pressure parts shall be one of the following:

(1) Except as permitted in (2), fillet welds are permissible only for seal welds or for lightly loaded attachments with a weld size not over 10 mm (0.375 in.) leg dimension and the toe of the weld shall not be located closer than $\sqrt{Rt_{sh}}$ from a gross structural discontinuity.

(2) For materials SA-333 Grade 8, SA-334 Grade 8, SA-353, SA-522, SA-553, and SA-645, fillet welds are permissible, provided that the fillet weld leg dimension does not exceed 13 mm (0.5 in.) and the toe of the weld shall not be located closer than $\sqrt{Rt_{sh}}$ from another gross structural discontinuity.

(3) A partial penetration weld plus fillet weld; limited to the attachment of parts not exceeding 19 mm (0.75 in.) in thickness.

(4) A full penetration groove weld plus a fillet weld on each side.

(5) Full penetration butt weld (see (e)(4) for boss requirements).

(6) For attachment of support skirts or other supports involving similar attachment orientation, in addition to welds permitted by (5) above, welds of greater effective throat dimension than 90 deg fillet welds may be used where the throat is a minimum of t_a (see Figure 4.2.4). The details in this figure are limited to attachment of parts not exceeding 19 mm (0.75 in.) in thickness unless the attachment weld is double welded.

(g) Stress Values For Weld Material - Attachment weld strength shall be based on the nominal weld area and the allowable stress values in Annex 3-A for the weaker of the two materials joined, multiplied by the reduction factors, W_r, shown below.

(1) Full penetration butt or groove welds - W_r = 1.0; the nominal weld area is the depth of the weld times the length of weld.

(2) Partial penetration groove or partial penetration groove plus fillet welds - W_r = 0.75; the nominal weld area is:

(-a) Groove welds - the depth of penetration times the length of weld.

(-b) Groove welds with fillet welds - the combined throat and depth of penetration, exclusive of reinforcement, times the length of weld.

(3) Fillet welds - W_r = 0.5; the nominal weld area is the throat area.

(h) Weld Overlay and Clad Construction

(1) Attachments may be welded directly to weld overlay deposits without restriction.

(2) For clad construction where design credit is taken is taken for cladding thickness, attachment welds may be made directly to the cladding based for loadings not producing primary stress in the attachment weld not exceeding 10% of the design allowable stress value of the attachment or the cladding material, whichever is less. As an alternative, local regions of weld overlay can be located within the cladding to provide an attachment location.

(3) For applied linings, attachments should be made directly to the base metal unless an analysis, tests, or both can be performed to establish the adequacy and reliability of an attachment made directly to the lining. Note that successful experience with similar linings in comparable service may provide a basis for judgment.

(i) PWHT Requirements - For heat treatment after welding, the fabrication requirements of the vessel base metal apply.

(j) Evaluation of Need For Fatigue Analysis - In applying the fatigue screening analysis in 5.5.2, fillet welds and non-full-penetration welds shall be considered to be nonintegral attachments, except that the following welds need not be considered because of the limitations of their use:

(1) Welds covered by (c), (e)(1), (f)(1) and (f)(2)

(2) Welds covered by (e)(5) and (f)(6) may be considered integral

4.2.5.7 Special Limitations for Joints in Quenched and Tempered High Strength Steels

(a) In vessels and vessel parts constructed of quenched and tempered high strength steels (see Table 3-A.2) except as permitted in (b), all joints of Categories A, B, and C, and all other welded joints between parts of the pressure-containing enclosure that are not defined by the category designation shall be Type No. 1.

(1) If the shell plate thickness is 50 mm (2 in.) or less, then all Category D welds shall be Type No. 1 in accordance with Table 4.2.13.

(2) If the shell plate thickness is greater than 50 mm (2 in.), then the weld detail may be as permitted for nozzles in Table 4.2.10 or Table 4.2.13.

(b) For materials SA-333 Grade 8, SA-334 Grade 8, SA-353, SA-522, SA-553, and SA-645 the weld joints shall be as follows:

(1) All joints of Category A shall be Type No. 1.

(2) All joints of Category B shall be Type No. 1 or Type No. 2.

(3) All joints of Category C shall be full penetration welds extending through the entire section at the joint.

(4) All joints of Category D attaching a nozzle neck to the vessel wall and to a reinforcing pad, if used, shall be full penetration groove welds.

4.2.5.8 Tube-to-Tubesheet Welds

Requirements for tube-to-tubesheet welds are given in 4.18.

4.2.5.9 Nozzle-Neck-to-Piping Welds

In the case of nozzle necks that attach to piping of a lesser wall thickness [see 1.2.3(a)(1)], a tapered transition from the weld end of the nozzle may be provided to match the piping thickness although that thickness is less than otherwise required by the rules of this Division (see Table 4.2.15).

4.2.5.10 Special Criteria for Corner Welds in Flexible Shell Element Expansion Joints

Each corner joint shall be a full penetration corner weld with a covering fillet and no backing strip. The covering fillet shall be located on the inside of the corner and shall have a throat per Table 4.2.16. Note that a fatigue evaluation may require a larger weld. It is permitted for the corner weld to be full penetration through either element being joined.

4.2.6 NOMENCLATURE

a = geometry parameter used to determine the length requirements for a thickness transition or a required weld size, applicable

b = geometry parameter used to determine the length requirements for a thickness transition or a required weld size, applicable

c = weld size parameter
R = mean radius of the shell or head
S_{yT} = minimum specified yield strength from Annex 3-D at the design temperature
S_u = minimum specified ultimate tensile strength from Annex 3-D
t = nominal thickness of the flexible element of a flexible shell element expansion joint
t_a = thickness of the attached member
t_c = throat dimension of a corner weld
t_e = thickness of the reinforcing element
t_h = nominal thickness of the head
t_n = nominal thickness of the shell or nozzle, as applicable
t_o = nominal thickness of the outer shell element of a flexible shell element expansion joint
t_p = distance from the outside surface of a flat head, flange, or other part to either the edge or center of a weld
t_{pipe} = minimum wall thickness of the connecting pipe
t_r = required thickness of the shell in accordance with the requirements of this Division
t_{rn} = required thickness of seamless nozzle wall in accordance with the requirements of this Division
t_s = nominal thickness of the shell
t_{sh} = nominal thickness of shell, t_s, or head, t_h
t_w = depth of penetration of the weld
t_x = two times the thickness g_0 (see 4.16) when the design is calculated as an integral flange or two times the nozzle thickness of the shell nozzle wall required for internal pressure when the design is calculated as a loose flange, but in no case less than 6 mm (0.25 in.)
T = minimum thickness of a flat head, cover, flange, or tubesheet, as applicable
W_r = weld type reduction factor

4.2.7 TABLES

Table 4.2.1
Definition of Weld Categories

Weld Category	Description
A	• Longitudinal and spiral welded joints within the main shell, communicating chambers [Note (1)], transitions in diameter, or nozzles • Any welded joint within a sphere, within a formed or flat head, or within the side plates [Note (2)] of a flat-sided vessel • Any butt-welded joint within a flat tubesheet • Circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameter, to nozzles, or to communicating chambers
B	• Circumferential welded joints within the main shell, communicating chambers [Note (1)], nozzles or transitions in diameter including joints between the transition and a cylinder at either the large or small end • Circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers
C	• Welded joints connecting flanges, Van Stone laps, tubesheets or flat heads to main shell, to formed heads, to transitions in diameter, to nozzles, or to communicating chambers [Note (1)] • Any welded joint connecting one side plate [Note (2)] to another side plate of a flat-sided vessel
D	• Welded joints connecting communicating chambers [Note (1)] or nozzles to main shells, to spheres, to transitions in diameter, to heads, or to flat-sided vessels • Welded joints connecting nozzles to communicating chambers [Note (1)] (for nozzles at the small end of a transition in diameter see Category B)
E	• Welded joints attaching nonpressure parts and stiffeners

NOTES:
(1) Communicating chambers are defined as appurtenances to the vessel that intersect the shell or heads of a vessel and form an integral part of the pressure-containing enclosure, e.g., sumps.
(2) Side plates of a flat-sided vessel are defined as any of the flat plates forming an integral part of the pressure-containing enclosure.

Table 4.2.2
Definition of Weld Joint Types

Weld Joint Type	Description
1	Butt joints and angle joints where the cone half-apex angle is less than or equal to 30 deg produced by double welding or by other means which produce the same quality of deposited weld metal on both inside and outside weld surfaces. Welds using backing strips which remain in place do not qualify as Type No. 1 butt joints.
2	Butt joints produced by welding from one side with a backing strip that remains in place
3	Butt joints produced by welding from one side without a backing strip
7	Corner joints made with full penetration welds with or without cover fillet welds
8	Angle joints made with a full penetration weld where the cone half-apex angle is greater than 30 deg
9	Corner joints made with partial penetration welds with or without cover fillet welds
10	Fillet welds

Table 4.2.3
Definition of Material Types for Welding and Fabrication Requirements

Material Type	Description
1	• P-No. 1 Groups 1, 2, and 3 • P-No. 3 Group 3 except SA-302 • P-No. 4, Group 1, SA-387 Grade 12, and SA/EN 10028-2 Grade 13CrMo4–5 • P-No. 8, Groups 1 and 2 • P-No. 9A Group 1
2	Materials not included in material Types 1, 3, and 4
3	Quenched and tempered high strength steels (see Table 3-A.2) except SA-372 Grade D and Class 70 of Grades E, F, G, H, and J when used for forged bottles
4	• P-No. 21 through P-No. 25 inclusive • P-No. 31 through P-No. 35 inclusive • P-No. 41 through P-No. 45 inclusive

Table 4.2.4
Some Acceptable Weld Joints for Shell Seams

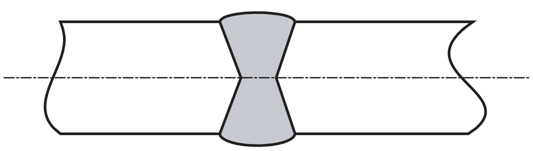
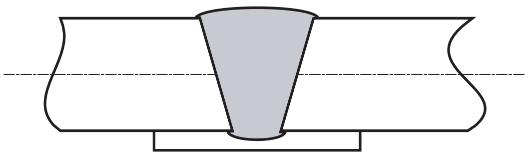
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	A, B, C, D		
2	2	B		

Table 4.2.4
Some Acceptable Weld Joints for Shell Seams (Cont'd)

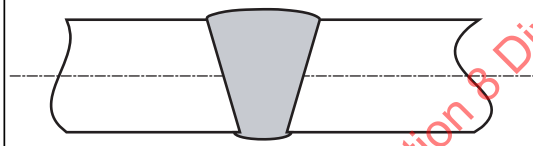
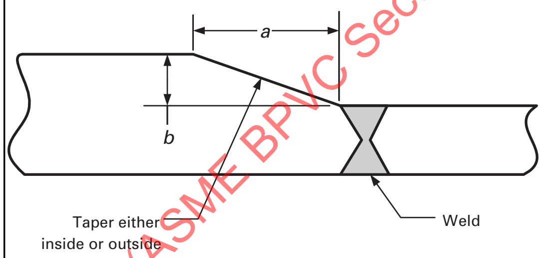
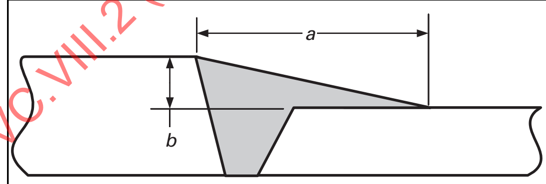
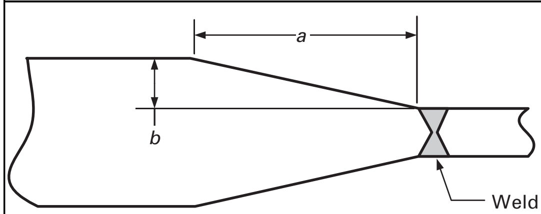
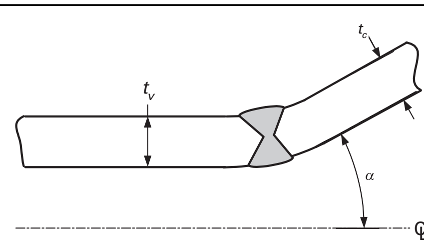
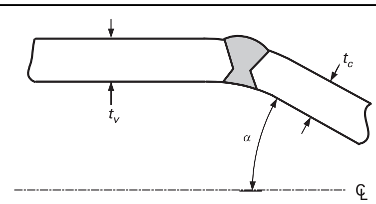
Detail	Joint Type	Joint Category	Design Notes	Figure
3	3	B		
4	1	A, B, C, D	• α ≥ 3*b* • The length of the taper, *a*, may include the weld • Joint Types 2 and 3 may be permissible, see 4.2.5.2 through 4.2.5.6 for limitations	
5	1	A, B, C, D		
6	1	A, B, C, D		
7	1	B	• α ≤ 30 deg • see 4.2.5.3(f) • Joint Types 2 and 3 may be permissible, see 4.2.5.2 through 4.2.5.6 for limitations	
8	8	B	• α > 30 deg	
9	1	B	• α ≤ 30 deg • see 4.2.5.3(f) • Joint Types 2 and 3 may be permissible, see 4.2.5.2 through 4.2.5.6 for limitations	
10	8	B	• α > 30 deg	

Table 4.2.5
Some Acceptable Weld Joints for Formed Heads

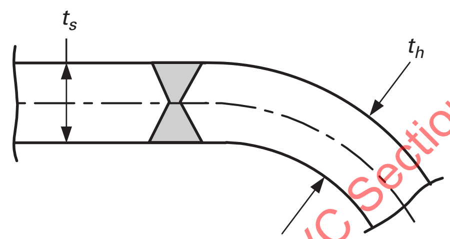
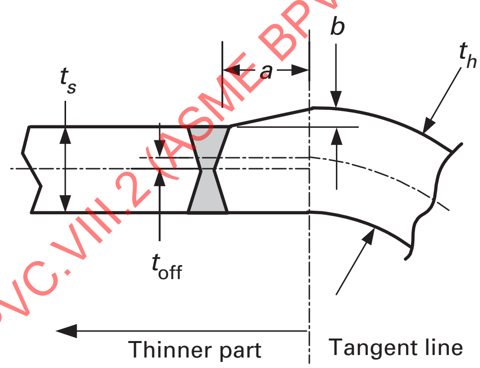
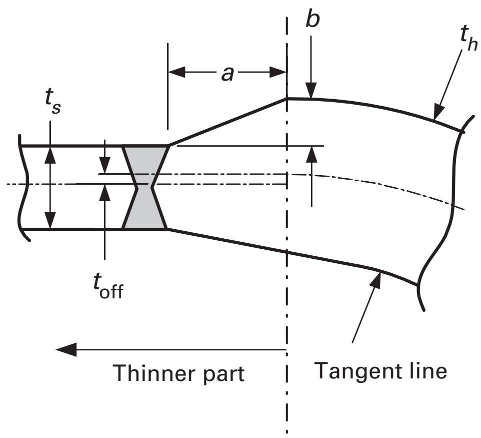
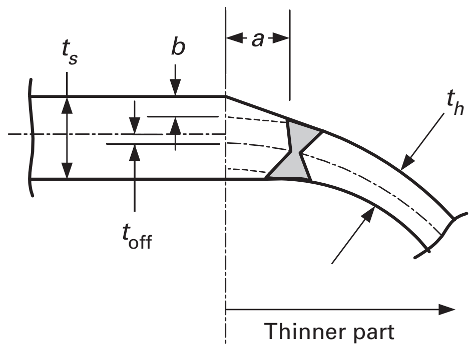
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	A, B	• Joint Types 2 and 3 may be permissible, see 4.2.5.2 through 4.2.5.6 for limitations	
2	1	A, B	• $\alpha \geq 3b$ when t_h exceeds t_s. • $t_{off} \leq 0.5\ (t_h - t_s)$ • The skirt minimum length is min[$3t_h$, 38 mm (1.5 in.)] except when necessary to provide the required taper length • If $t_h \leq 1.25t_s$, then the length of the skirt shall be sufficient for any required taper • The length of the taper a may include the width of the weld. • The shell plate center line may be on either side of the head plate center line • Joint Types 2 and 3 may be permissible, see 4.2.5.2 through 4.2.5.6 for limitations	
3	1	A, B		
4	1	A, B	• $a \geq 3b$ • $t_{off} \leq 0.5(t_s - t_h)$ • The length of the taper a may include the width of the weld. • The shell plate center line may be on either side of the head plate center line • Joint Types 2 and 3 may be permissible, see 4.2.5.2 through 4.2.5.6 for limitations	

Table 4.2.5
Some Acceptable Weld Joints for Formed Heads (Cont'd)

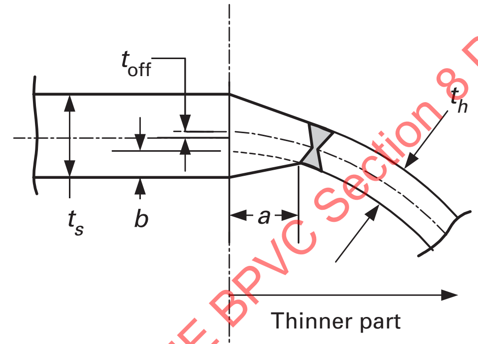
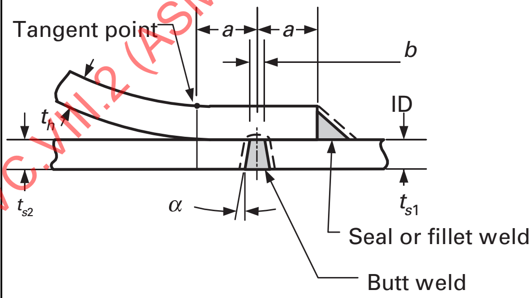
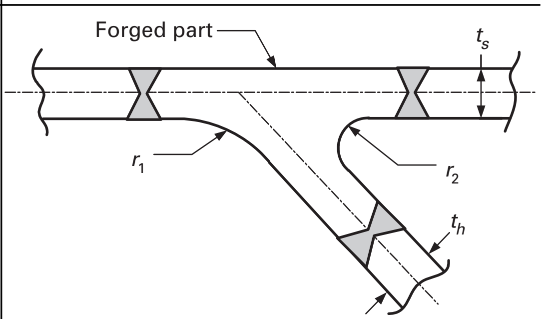
Detail	Joint Type	Joint Category	Design Notes	Figure
5	1	A, B	• See Detail 4	
6	2	B	• Butt weld and, if used, fillet weld shall be designed to take a shear load at 1.5 times the design differential pressure • $a \geq \min[2t_h, 25 \text{ mm } (1 \text{ in.})]$ • b, 13 mm (0.5 in.) minimum • The shell thicknesses t_{s1} and t_{s2} may be different • α, 15 deg ≤ α ≤ 20 deg	
7	1	A, B	• $r_1 \geq 2r_2$ • $r_2 \geq \min[t_s, t_h]$	

Table 4.2.6
Some Acceptable Weld Joints for Unstayed Flat Heads, Tubesheets Without a Bolting Flange, and Side Plates of Rectangular Pressure Vessels

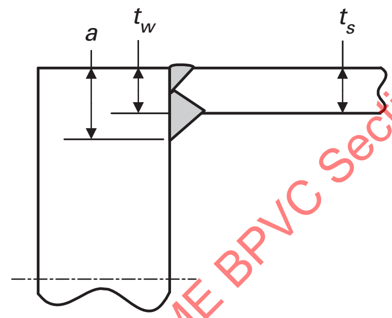
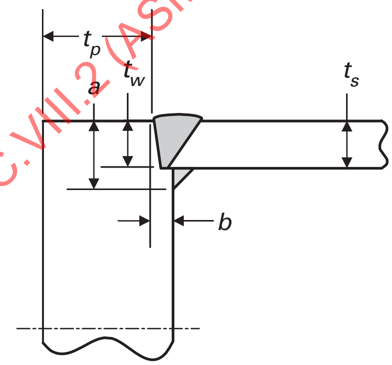
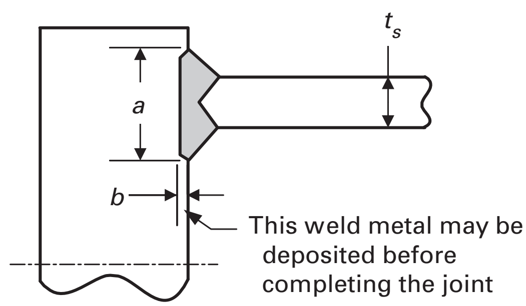
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	C	• $a \geq 2t_s$ • $t_w \geq t_s$	
2	7	C	• $a + b \geq 2t_s$ • $t_w \geq t_s$ • $t_p \geq$ min[t_s, 6 mm (0.25 in.)] • The dimension b is produced by the weld preparation and shall be verified after fit-up and before welding	
3	7	C	• $a + b \geq 2t_s$ • $b = 0$ is permissible • The dimension b is produced by the weld preparation and shall be verified after fit-up and before welding	

Table 4.2.7
Some Acceptable Weld Joints With Butt Weld Hubs

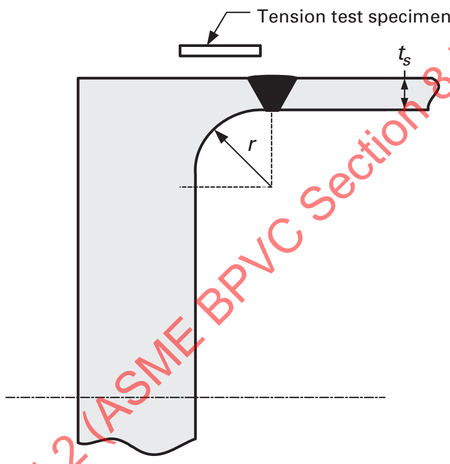
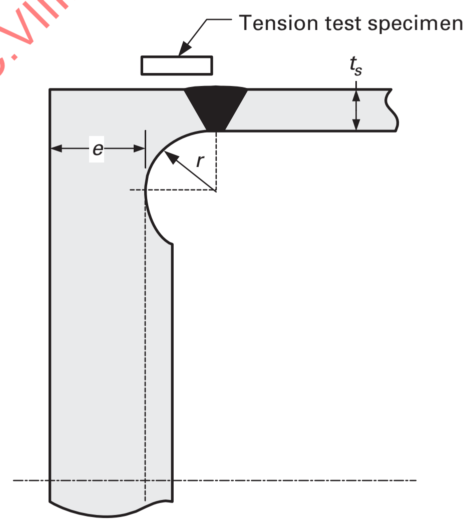
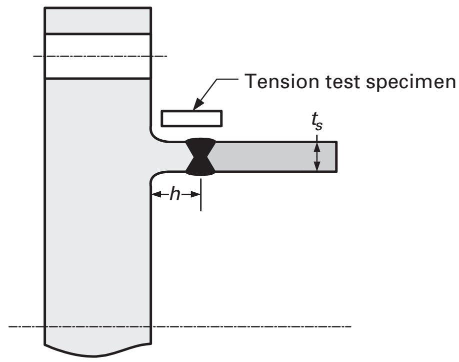
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	C	• $r \geq 10$ mm (0.357 in.) for $t_s \leq 38$ mm (1.5 in.) • $r \geq \min[0.25t_s$, 19 mm (0.75 in.)] for $t_s > 38$ mm (1.5 in.)	
2	1	C	• $r \geq 10$ mm (0.357 in.) for $t_s \leq 38$ mm (1.5 in.) • $r \geq \min[0.25t_s$, 19 mm (0.75 in.)] for $t_s > 38$ mm (1.5 in.) • $e \geq \max[t_s, T]$	
3	1	C	• $h = \max[1.5t_s$, 19 mm (0.75 in.)] but need not exceed 51 mm (2 in.)	

Table 4.2.8
Some Acceptable Weld Joints for Attachment of Tubesheets With a Bolting Flange

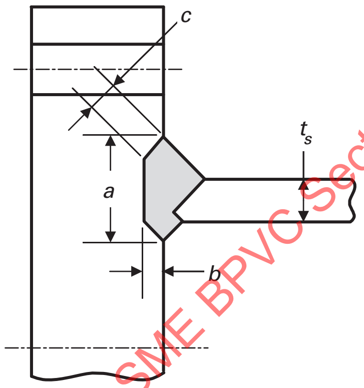
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	C	• $a + b \geq 2t_s$ • $b = 0$ is permissible • The dimension b is produced by the weld preparation and shall be verified after fit-up and before welding • $c \geq \min[0.7t_s, 1.4t_r]$	

Table 4.2.9
Some Acceptable Weld Joints for Flange Attachments

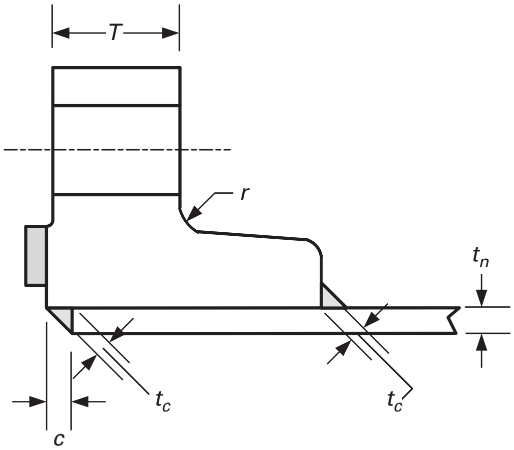
Detail	Joint Type	Joint Category	Design Notes	Figure
1	10	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq t_n + 6$ mm (0.25 in.) maximum • $r \geq \max[0.25g_1, 5$ mm (0.1875 in.)]	

Table 4.2.9
Some Acceptable Weld Joints for Flange Attachments (Cont'd)

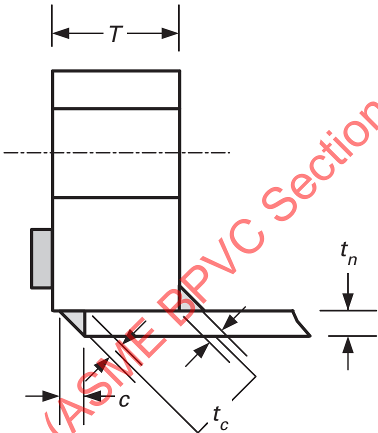
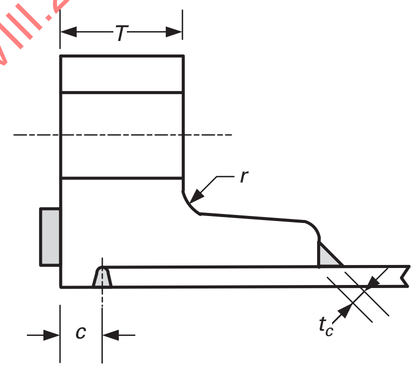
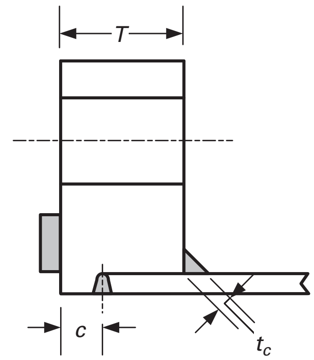
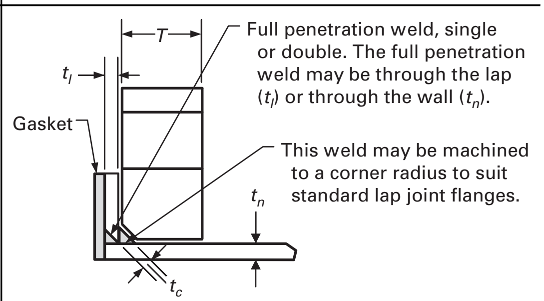
Detail	Joint Type	Joint Category	Design Notes	Figure
2	10	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq t_n$ + 6 mm (0.25 in.) maximum	
3	7	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq 0.5t$ maximum • $r \geq$ max[$0.25g_1$, 5 mm (0.1875 in.)]	
4	7	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq 0.5t$ maximum	
5	7	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $t_l \geq t_n$ + 5 mm (0.1875 in.)	

Table 4.2.9
Some Acceptable Weld Joints for Flange Attachments (Cont'd)

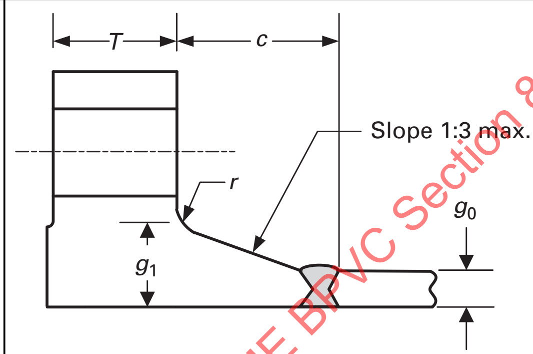
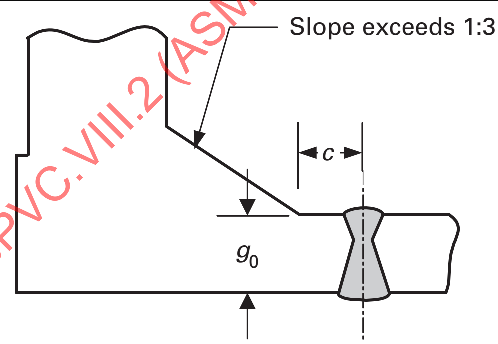
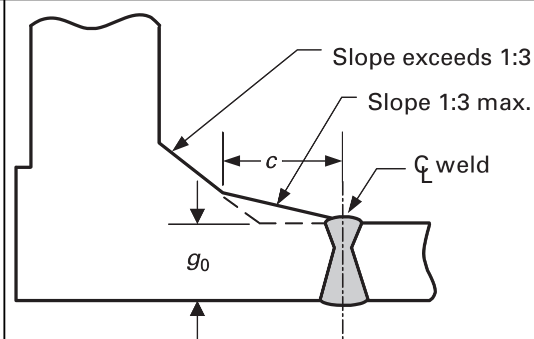
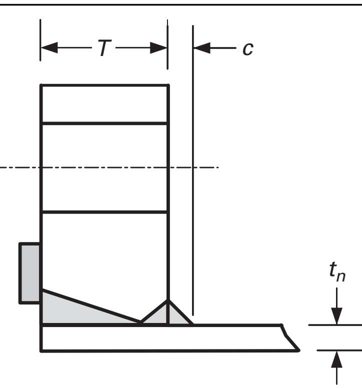
Detail	Joint Type	Joint Category	Design Notes	Figure
6	1	C	• Integral Type Flange • $c \geq 1.5g_0$ minimum • $r \geq \max[0.25g_1,\ 5\ \text{mm}\ (0.1875\ \text{in.})]$	
7	1	C	• Integral Type Flange • $c \geq 1.5g_0$ minimum	
8	1	C	• Integral Type Flange • $c \geq 1.5g_0$ minimum	
9	7	C	• Integral Type Flange • $c \geq \min[0.25g_0,\ 6\ \text{mm}\ (0.25\ \text{in.})]$	

Table 4.2.9
Some Acceptable Weld Joints for Flange Attachments (Cont'd)

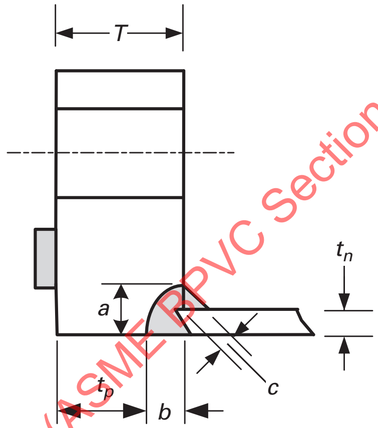
Detail	Joint Type	Joint Category	Design Notes	Figure
10	7	C	• Integral Type Flange • $a + b \geq 3t_n$ • $t_p \geq \min[t_n, 6\text{ mm }(0.25\text{ in.})]$ • $c \geq \min[t_n, 6\text{ mm }(0.25\text{ in.})]$	

Table 4.2.10
Some Acceptable Full Penetration Welded Nozzle Attachments Not Readily Radiographable

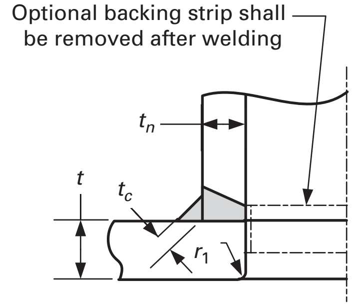
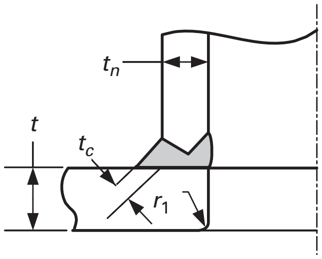
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• $t_c \geq \min[0.7t_n, 6\text{ mm }(0.25\text{ in.})]$ • $r_1 \geq \min[0.25t, 3\text{ mm }(0.125\text{ in.})]$	
2	7	D	• $t_c \geq \min[0.7t_n, 6\text{ mm }(0.25\text{ in.})]$ • $r_1 \geq \min[0.25t, 3\text{ mm }(0.125\text{ in.})]$	

Table 4.2.10
Some Acceptable Full Penetration Welded Nozzle Attachments Not Readily Radiographable (Cont'd)

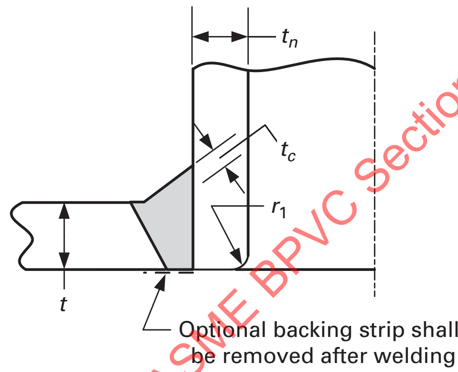
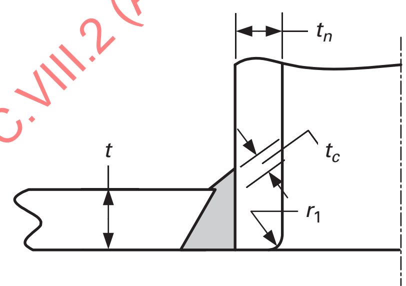
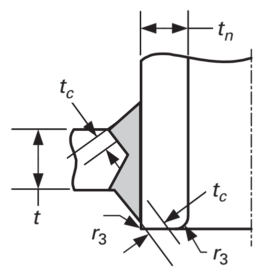
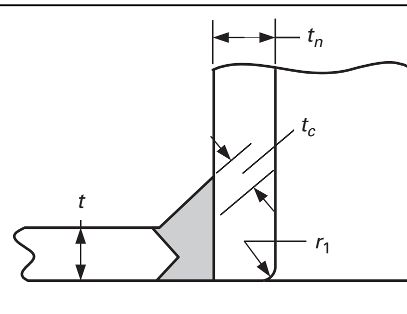
Detail	Joint Type	Joint Category	Design Notes	Figure
3	7	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)]	
4	7	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)]	
5	7	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $r_3 \geq \min[0.25t$, 3 mm (0.125 in.)] alternatively, a chamfer of $r_3 \geq \min[0.25t$, 3 mm (0.125 in.)] at 45 deg	
6	7	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)]	

Table 4.2.10
Some Acceptable Full Penetration Welded Nozzle Attachments Not Readily Radiographable (Cont'd)

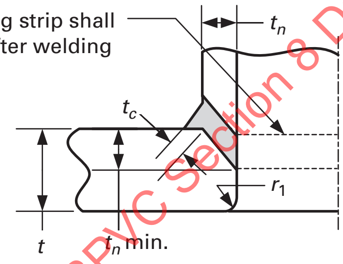
Detail	Joint Type	Joint Category	Design Notes	Figure
7	7	D	• $t_c \geq \min[0.7t_n, 6 \text{ mm } (0.25 \text{ in.})]$ • $r_1 \geq \min[0.25t, 3 \text{ mm } (0.125 \text{ in.})]$	

Table 4.2.11
Some Acceptable Pad Welded Nozzle Attachments and Other Connections to Shells

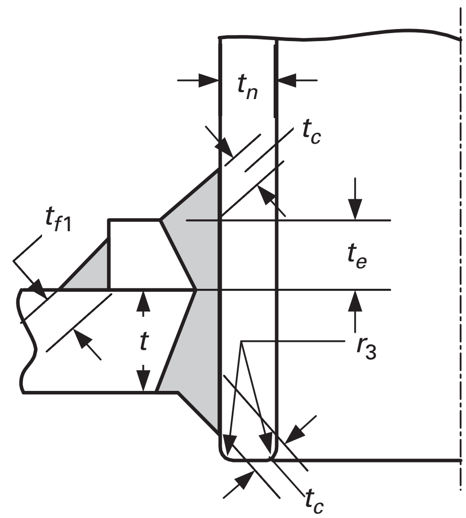
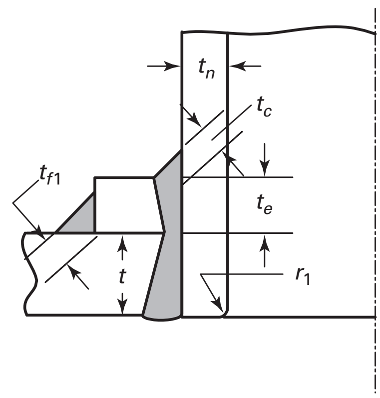
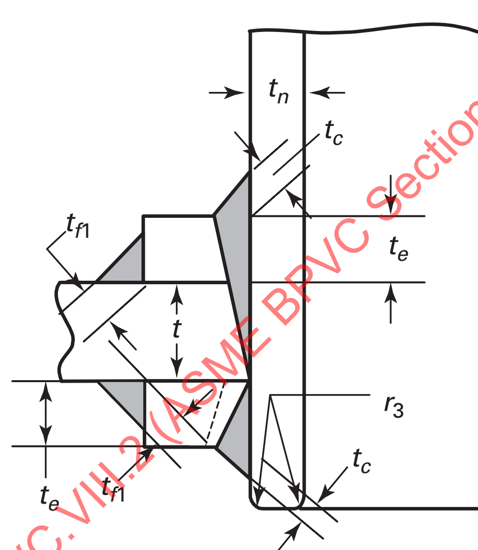
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• $t_c \geq \min[0.7t_n, 6 \text{ mm } (0.25 \text{ in.})]$ • $t_{f1} \geq \min[0.6t_e, 0.6t]$ • $r_3 \geq \min[0.25t, 3 \text{ mm } (0.125 \text{ in.})]$ alternatively, a chamfer of $r_3 \geq \min[0.25t, 3 \text{ mm } (0.125 \text{ in.})]$ at 45 deg	
2	7	D	• $t_c \geq \min[0.7t_n, 6 \text{ mm } (0.25 \text{ in.})]$ • $t_{f1} \geq \min[0.6t_e, 0.6t]$ • $r_1 \geq \min[0.25t, 3 \text{ mm } (0.125 \text{ in.})]$	

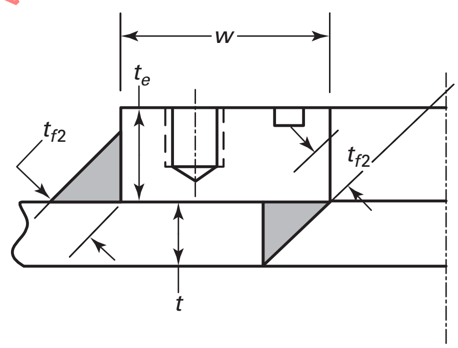
Table 4.2.11
Some Acceptable Pad Welded Nozzle Attachments and Other Connections to Shells (Cont'd)

<table>
<tr><th>Detail</th><th>Joint Type</th><th>Joint Category</th><th>Design Notes</th><th>Figure</th></tr>
<tr><td>3</td><td>7</td><td>D</td><td>

- $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)]
- $t_{f1} \geq \min[0.6t_e, 0.6t]$
- $r_3 \geq \min[0.25t$, 3 mm (0.125 in.)] alternatively, a chamfer of $r_3 \geq \min[0.25t$, 3 mm (0.125 in.)] at 45 deg

</td><td>
</td></tr>
<tr><td>4</td><td>10</td><td>D</td><td>

- $t_{f2} \geq \min[0.7t_e, 0.7t]$

</td><td>
</td></tr>
<tr><td>5</td><td>7</td><td>D</td><td>

- $t_{f2} \geq \min[0.7t_e, 0.7t]$
- $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)]

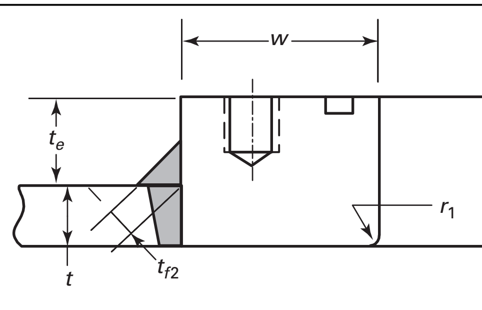
</td><td>
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Table 4.2.12
Some Acceptable Fitting-Type Welded Nozzle Attachments and Other Connections to Shells

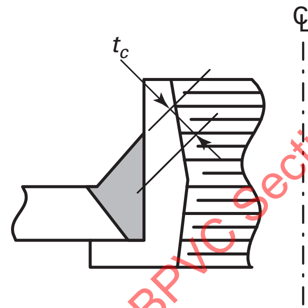
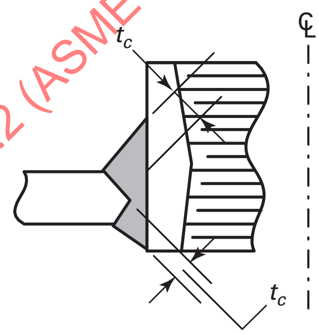
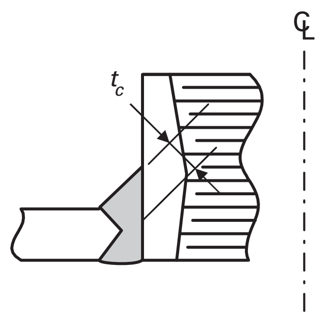
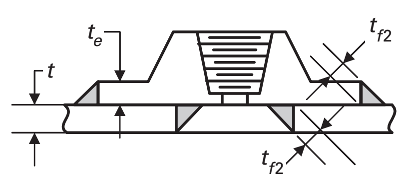
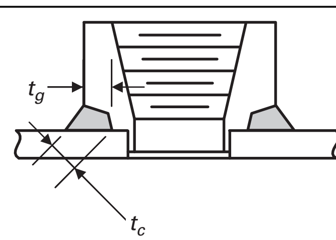
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)]	
2	7	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)]	
3	7	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)]	
4	10	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $t_{f2} \geq \min[0.7t_e, 0.7t]$	
5	9	D	• Limited to DIN 50 (NPS 2) maximum • The groove weld t_g shall not be less than the thickness of Schedule 160 • $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)]	

Table 4.2.13
Some Acceptable Welded Nozzle Attachments That Are Readily Radiographable

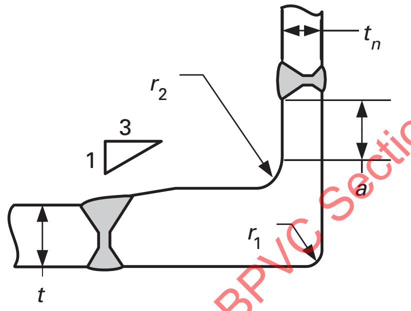
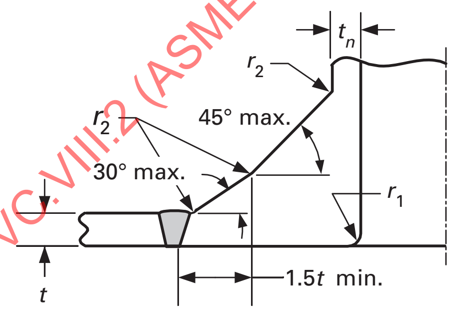
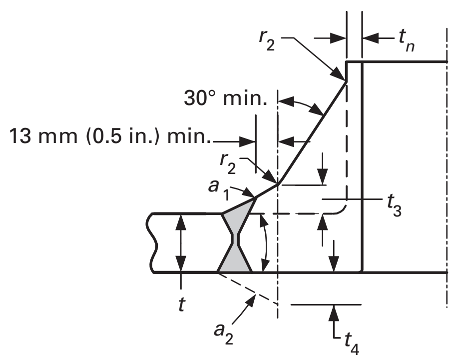
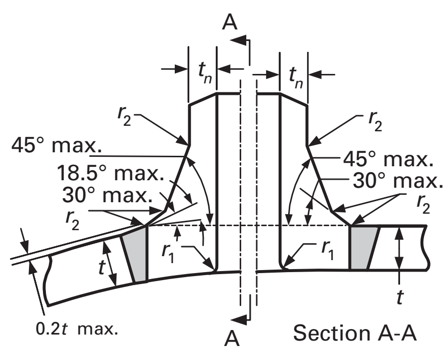
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	D	• $r_1 \geq \min[0.25t, 3\text{ mm }(0.125\text{ in.})]$ • $r_2 \geq \min[0.25t_n, 19\text{ mm }(0.75\text{ in.})]$	
2	1	D	• $r_1 \geq \min[0.25t, 3\text{ mm }(0.125\text{ in.})]$ • $r_2 \geq \min[0.25t_n, 19\text{ mm }(0.75\text{ in.})]$	
3	1	D	• $r_1 \geq \min[0.25t, 3\text{ mm }(0.125\text{ in.})]$ • $r_2 \geq \min[0.25t_n, 19\text{ mm }(0.75\text{ in.})]$ • $t_3 + t_4 \leq 0.2t$ • $a_1 + a_2 \leq 18.5$ deg	
4	1	D	• $r_1 \geq \min[0.25t, 3\text{ mm }(0.125\text{ in.})]$ • $r_2 \geq \min[0.25t_n, 19\text{ mm }(0.75\text{ in.})]$	 **Sections Perpendicular and Parallel to the Cylindrical Vessel Axis**

Table 4.2.13
Some Acceptable Welded Nozzle Attachments That Are Readily Radiographable (Cont'd)

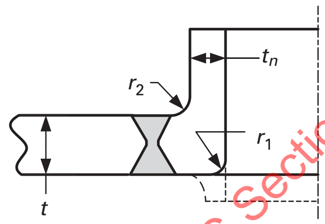
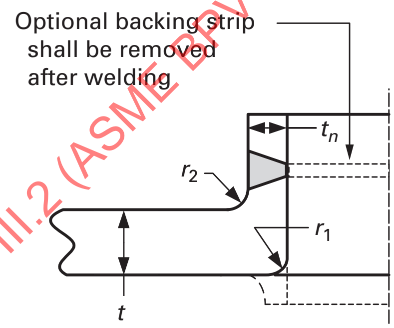
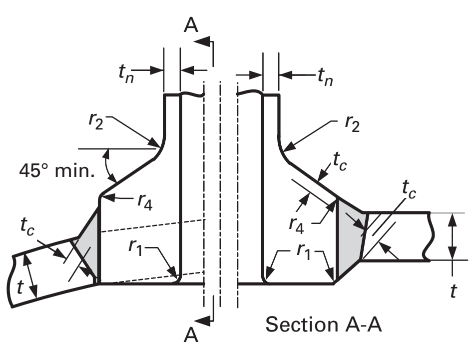
Detail	Joint Type	Joint Category	Design Notes	Figure
5	1	D	• $r_1 \geq \min[0.25t_n$, 3 mm (0.125 in.)] • $r_2 \geq \min[0.25t_n$, 19 mm (0.75 in.)]	
6	1	D	• $r_1 \geq \min[0.25t_n$, 3 mm (0.125 in.)] • $r_2 \geq \min[0.25t_n$, 19 mm (0.75 in.)]	
7	7	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)] • $r_2 \geq$ 19 mm (0.75 in.) • $r_4 \geq$ 6 mm (0.25 in.)	 **Sections Perpendicular and Parallel to the Cylindrical Vessel Axis**

Table 4.2.14
Some Acceptable Partial Penetration Nozzle Attachments

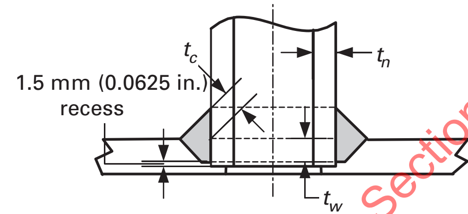
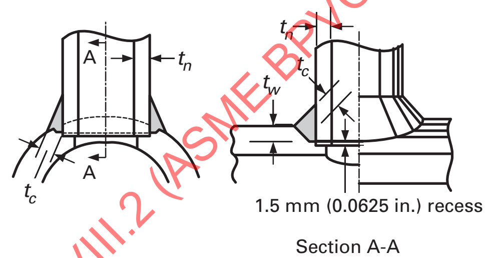
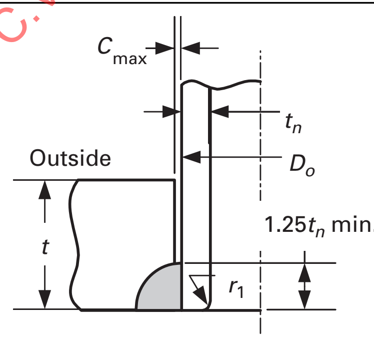
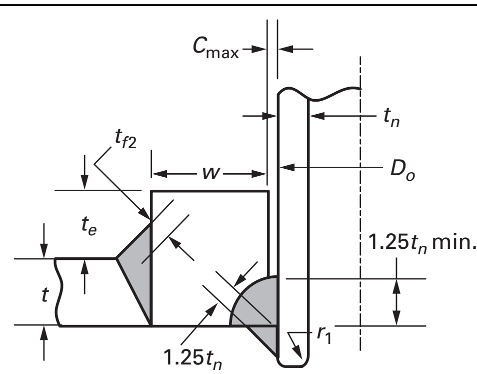
Detail	Joint Type	Joint Category	Design Notes	Figure
1	9	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $t_w \geq 1.25t_n$	
2	9	D	• $t_c \geq \min[0.7t_n$, 6 mm (0.25 in.)] • $t_w \geq 1.25t_n$	
3	9	D	• $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)] C_{max} defined as follows: 0.25 mm, $D_o \leq 25$ mm 0.51 mm, 25 mm < $D_o \leq 102$ mm • 0.76 mm, $D_o > 102$ mm 0.01 in., $D_o \leq 1$ in. 0.02 in., 1 in. < $D_o \leq 4$ in. 0.03 in., $D_o > 4$ in.	
4	9	D	• $t_{f2} \geq \min[0.7t_e, 0.7t]$ $r_1 \geq \min[0.25t$, 3 mm (0.125 in.)] C_{max} defined as follows: 0.25 mm, $D_o \leq 25$ mm 0.51 mm, 25 mm < $D_o \leq 102$ mm • 0.76 mm, $D_o > 102$ mm 0.01 in., $D_o \leq 1$ in. 0.02 in., 1 in. < $D_o \leq 4$ in. 0.03 in., $D_o > 4$ in.	

Table 4.2.15
Nozzle Necks Attached to Piping of Lesser Wall Thickness

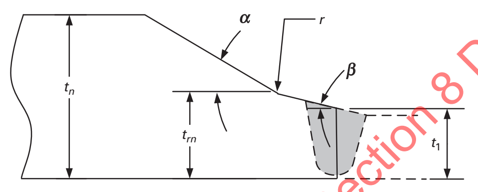
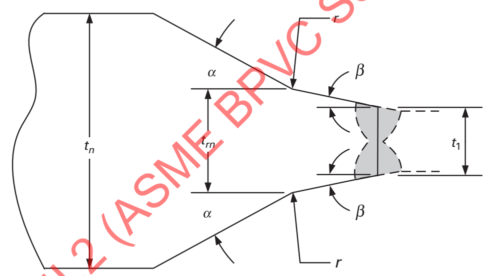
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	Not applicable	• The weld bevel is shown for illustration only • $t_1 \geq \max[0.8t_{rn}, t_{pipe}]$ • $\alpha \leq 30$ deg • β, 14 deg $\leq \beta \leq$ 18.5 deg • r, 6 mm (0.25 in.) min. radius • Joint Types 2 and 3 may be permissible; see 4.2.5.2 through 4.2.5.6 for limitations	
2	1	Not applicable	• $t_1 \geq \max[0.8t_{rn}, t_{pipe}]$ • $\alpha \leq 30$ deg • β, 14 deg $\leq \beta \leq$ 18.5 deg • r, 6 mm (0.25 in.) min. radius • Joint Types 2 and 3 may be permissible; see 4.2.5.2 through 4.2.5.6 for limitations	

Table 4.2.16
Corner Welds for Flexible Shell Element Expansion Joints

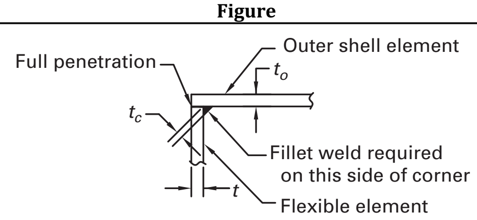
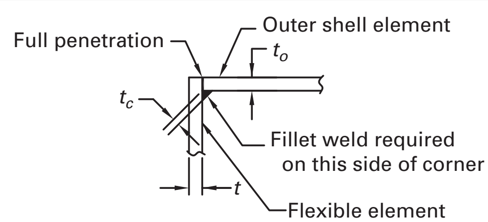
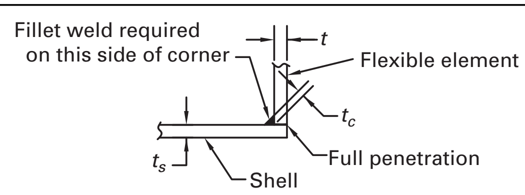
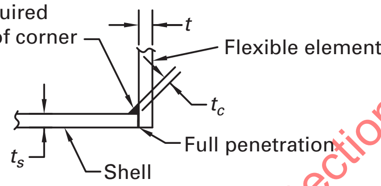
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	Not applicable	$t_c \geq \min[0.7t, 0.7t_o$, 6 mm (0.25 in.)]	
2	7	Not applicable	$t_c \geq \min[0.7t, 0.7t_o$, 6 mm (0.25 in.)]	
3	7	Not applicable	$t_c \geq \min[0.7t, 0.7t_s$, 6 mm (0.25 in.)]	

Table 4.2.16
Corner Welds for Flexible Shell Element Expansion Joints (Cont'd)

Detail	Joint Type	Joint Category	Design Notes	Figure
4	7	Not applicable	$t_c \geq \min[0.7t, 0.7t_s, 6\ \text{mm}\ (0.25\ \text{in.})]$	

4.2.8 FIGURES

Figure 4.2.1
Weld Joint Locations Typical of Categories A, B, C, D, and E

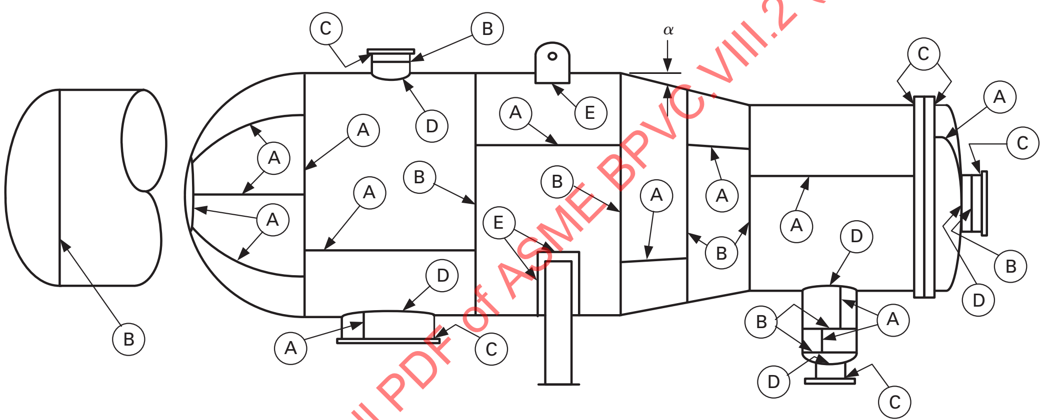
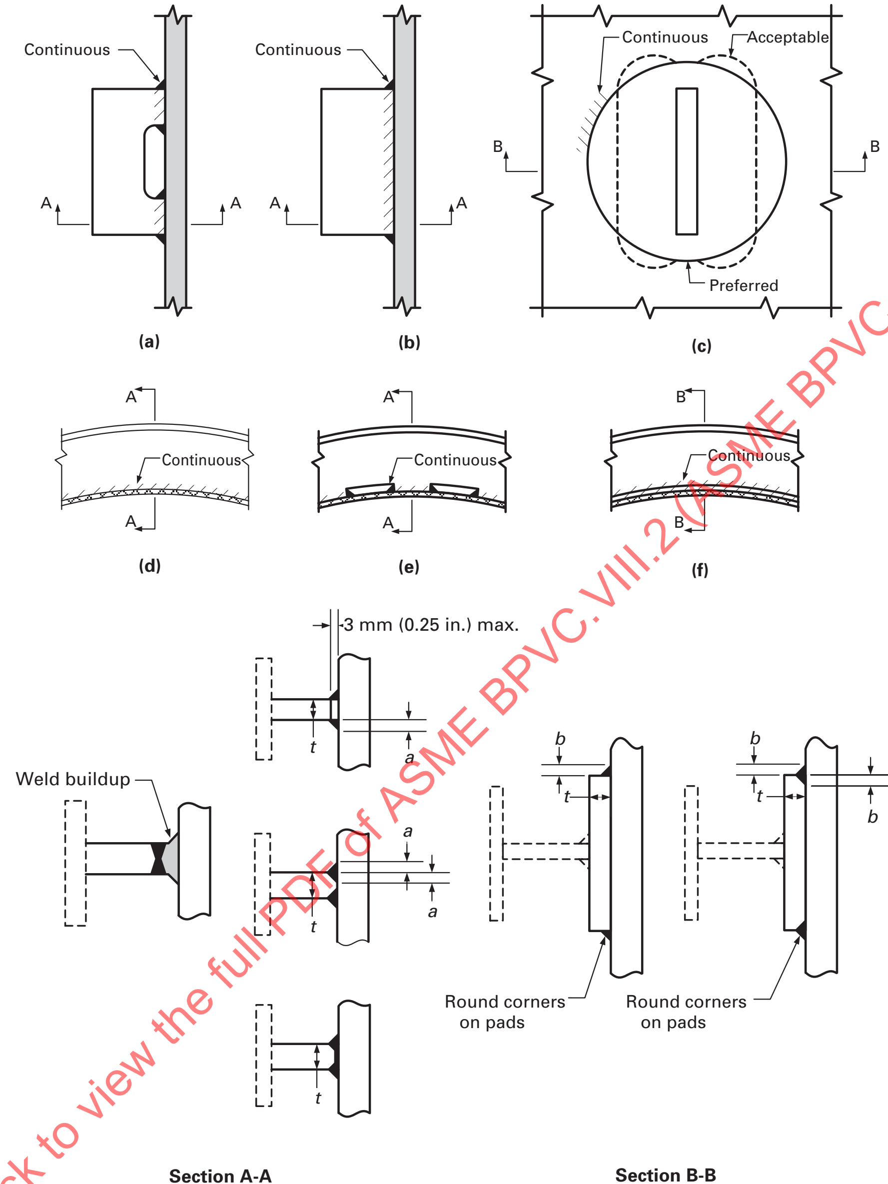
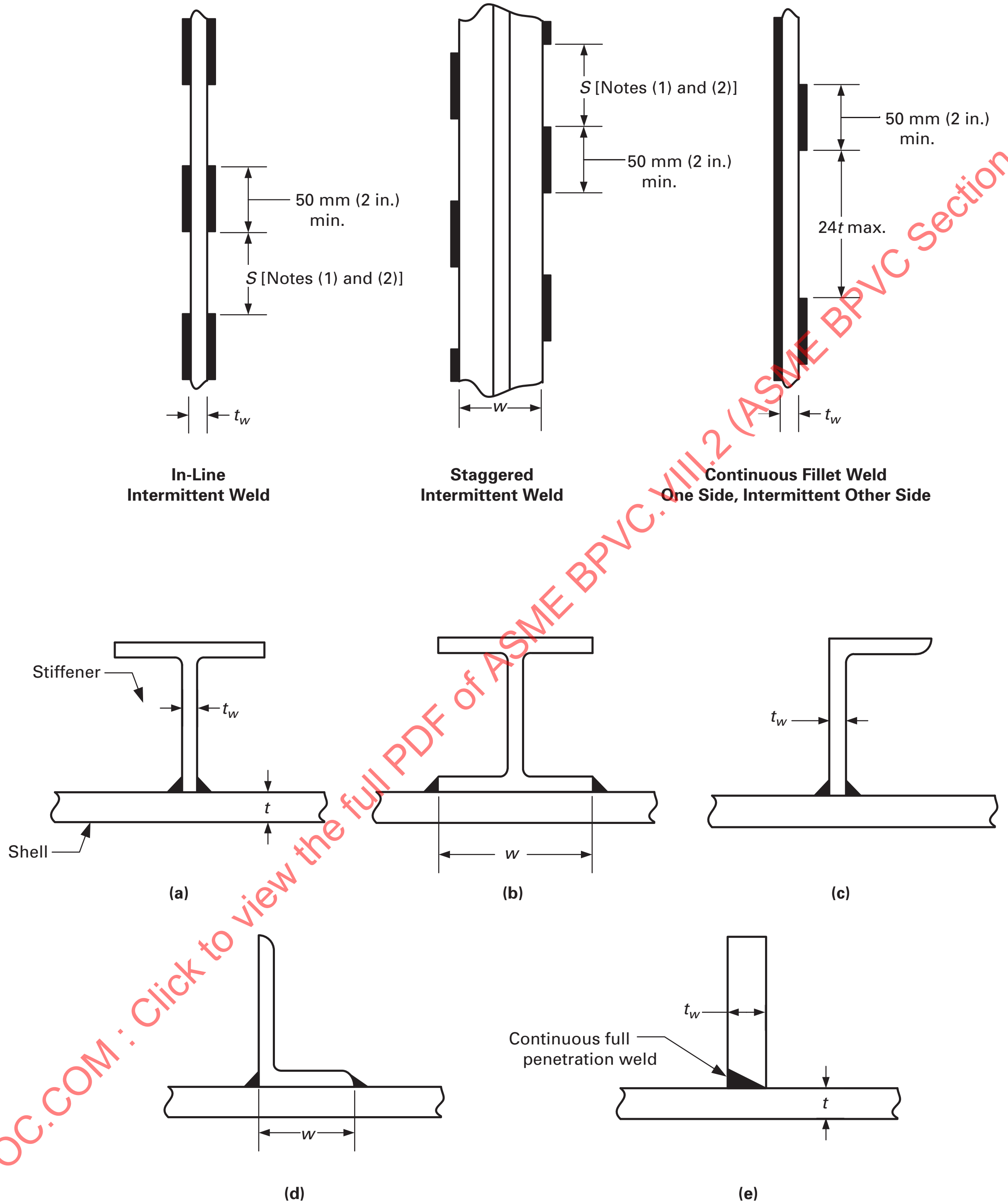


Figure 4.2.2
Some Bracket, Lug, and Stiffener Attachment Weld Details



GENERAL NOTES:
(a) Attachment weld size: $a \geq 0.25t$ and $b \geq 0.5t$.
(b) Vents holes shall be considered for continuously attached pads.
(c) For design (e) above, a minimum of 50% of the web must be welded, evenly spaced around the circumference of the shell.

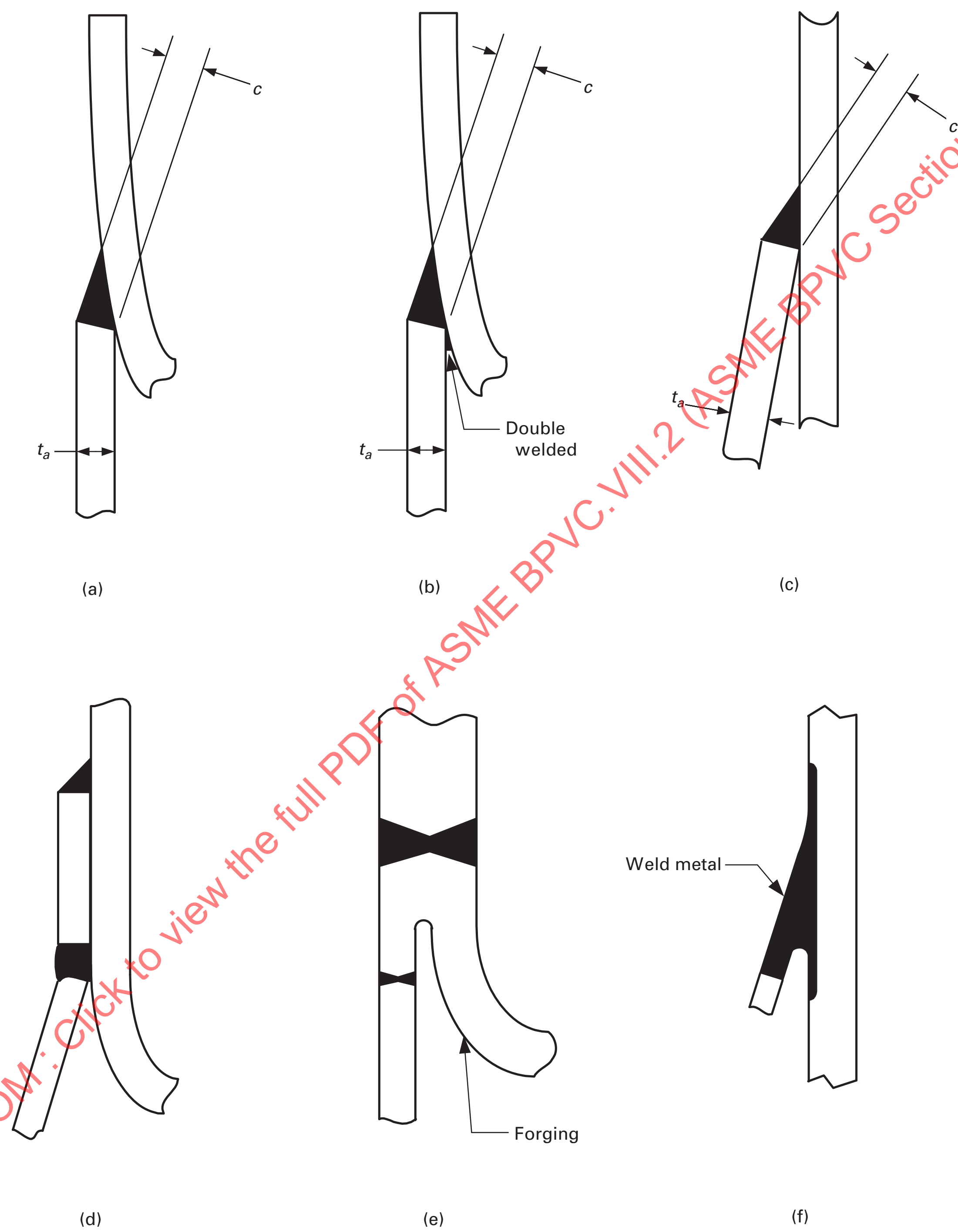
Figure 4.2.3
Some Acceptable Methods of Attaching Stiffening Rings



GENERAL NOTE: See 4.2.5.6(e) for limitations.

NOTES:
(1) For external stiffeners, $S \leq 8t$.
(2) For internal stiffeners, $S \leq 12t$.

Figure 4.2.4
Some Acceptable Skirt Weld Details



GENERAL NOTES:
(a) All welds are continuous.
(b) c is the minimum thickness of the weld metal from the root to the face of the weld.
(c) Attachment weld size: $c \geq t_a$.

4.3 DESIGN RULES FOR SHELLS UNDER INTERNAL PRESSURE

4.3.1 SCOPE

4.3.1.1 4.3 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to internal pressure. In this context, internal pressure is defined as pressure acting on the concave side of the shell.

4.3.1.2 The effects of supplemental loads are not included in design equations for shells and heads included in 4.3.3 to 4.3.7. Supplemental loads shall be defined in the User's Design Specification and their effects that result in combined loadings shall be evaluated in a separate analysis performed in accordance with the methods in 4.3.10.

4.3.1.3 Rules are provided for the design of cylindrical-to-conical shell transition junctions in 4.3.11 and 4.3.12. To facilitate the use of these rules, the shell wall thickness and stiffener configuration, as applicable, shall be designed using the rules in 4.3.3 through 4.3.7. After an initial design is determined, this design should then be checked and modified as required using the rules of 4.3.12 and 4.3.13.

4.3.2 SHELL TOLERANCES

4.3.2.1 The shell of a completed vessel shall satisfy the following requirements.

(a) The difference between the maximum and minimum inside diameters at any cross section shall not exceed 1% of the nominal diameter at the cross section under consideration. The diameters may be measured on the inside or outside of the vessel. If measured on the outside, the diameters shall be corrected for the plate thickness at the cross section under consideration.

(b) When the cross section passes through an opening or within one inside diameter of the opening measured from the center of the opening, the permissible difference in inside diameters given above may be increased by 2% of the inside diameter of the opening. When the cross section passes through any other location normal to the axis of the vessel, including head-to-shell junctions, the difference in diameters shall not exceed 1%.

4.3.2.2 Tolerances for formed heads shall satisfy the following requirements.

(a) The inner surface of torispherical, toriconical, hemispherical, or ellipsoidal heads shall not deviate outside of the specified shape by more than 1.25% of D nor inside the specified shape by more than 0.625% of D, where D is the nominal inside diameter of the vessel shell at the point of attachment. Such deviations shall be measured perpendicular to the specified shape and shall not be abrupt. The knuckle radius shall not be less than that specified

(b) Measurements for determining the deviations specified in (a) shall be taken from the surface of the base metal and not from welds.

(c) When the straight flange of any unstayed formed head is machined to make a lap joint connection to a shell, the thickness shall not be reduced to less than 90% of that required for a blank head or the thickness of the shell at the point of attachment. When so machined, the transition from the machined thickness to the original thickness of the head shall not be abrupt but shall be tapered for a distance of at least three times the difference between the thicknesses.

4.3.2.3 Shells that do not meet the tolerance requirements of this paragraph may be evaluated using 4.14.

4.3.3 CYLINDRICAL SHELLS

4.3.3.1 Required Thickness. The minimum required thickness of a cylindrical shell subjected to internal pressure shall be determined using the following equation.

$$t = \frac{D}{2}\left(\exp\left[\frac{P}{SE}\right] - 1\right) \tag{4.3.1}$$

4.3.3.2 Combined Loadings. Cylindrical shells subject to internal pressure and other loadings shall satisfy the requirements of 4.3.10.

4.3.4 CONICAL SHELLS

4.3.4.1 Required Thickness. The minimum required thickness of a conical shell (see Figure 4.3.1) subjected to internal pressure shall be determined using the following equation.

$$t = \frac{D}{2\cos[\alpha]}\left(\exp\left[\frac{P}{SE}\right] - 1\right) \tag{4.3.2}$$

4.3.4.2 Offset Transitions. The cylinders for an offset cone shall have parallel centerlines that are offset from each other by a distance no greater than the difference of their minimum radii, as shown in Figure 4.3.2. Configurations that do not satisfy this requirement shall be evaluated per Part 5. The offset cone shall be designed as a concentric cone using the angle, α, as defined in eq. (4.3.3).

$$\alpha = \max\left[\alpha_1, \alpha_2\right] \tag{4.3.3}$$

4.3.4.3 Combined Loadings. Conical shells subject to external pressure and other loadings shall satisfy the requirements of 4.3.10.

4.3.5 SPHERICAL SHELLS AND HEMISPHERICAL HEADS

4.3.5.1 The minimum required thickness of spherical shells and hemispherical heads shall be determined using the following equation:

$$t = \frac{D}{2}\left(\exp\left[\frac{0.5P}{SE}\right] - 1\right) \tag{4.3.4}$$

4.3.5.2 Combined Loadings. Spherical shells and hemispherical heads subject to internal pressure and other loadings shall satisfy the requirements of 4.3.10.

4.3.6 TORISPHERICAL HEADS

4.3.6.1 Torispherical Heads With the Same Crown and Knuckle Thicknesses. The minimum required thickness of a torispherical head (see Figure 4.3.3) subjected to internal pressure shall be calculated using the following procedure.

Step 1. Determine the inside diameter, D, and assume values for the crown radius, L, the knuckle radius, r, and the wall thickness t.

Step 2. Compute the head L/D, r/D, and L/t ratios and determine if the following equations are satisfied. If the equations are satisfied, then proceed to Step 3; otherwise, the head shall be designed in accordance with Part 5.

$$0.7 \le \frac{L}{D} \le 1.0 \tag{4.3.5}$$

$$\frac{r}{D} \ge 0.06 \tag{4.3.6}$$

$$20 \le \frac{L}{t} \le 2000 \tag{4.3.7}$$

Step 3. Calculate the following geometric constants:

$$\beta_{th} = \arccos\left[\frac{0.5D - r}{L - r}\right], \text{ radians} \tag{4.3.8}$$

$$\phi_{th} = \frac{\sqrt{Lt}}{r}, \text{ radians} \tag{4.3.9}$$

$$R_{th} = \frac{0.5D - r}{\cos\left[\beta_{th} - \phi_{th}\right]} + r \qquad \text{for} \qquad \phi_{th} < \beta_{th} \tag{4.3.10}$$

$$R_{th} = 0.5D \qquad \text{for} \qquad \phi_{th} \ge \beta_{th} \tag{4.3.11}$$

Step 4. Compute the coefficients C_1 and C_2 using the following equations.

$$C_1 = 9.31\left(\frac{r}{D}\right) - 0.086 \qquad \text{for} \qquad \frac{r}{D} \le 0.08 \tag{4.3.12}$$

$$C_1 = 0.692\left(\frac{r}{D}\right) + 0.605 \qquad \text{for} \qquad \frac{r}{D} > 0.08 \tag{4.3.13}$$

$$C_2 = 1.25 \qquad \text{for} \qquad \frac{r}{D} \le 0.08 \tag{4.3.14}$$

$$C_2 = 1.46 - 2.6\left(\frac{r}{D}\right) \qquad \text{for} \qquad \frac{r}{D} > 0.08 \tag{4.3.15}$$

Step 5. Calculate the value of internal pressure expected to produce elastic buckling of the knuckle.

$$P_{eth} = \frac{C_1 E_T t^2}{C_2 R_{th}\left(\frac{R_{th}}{2} - r\right)} \tag{4.3.16}$$

Step 6. Calculate the value of internal pressure that will result in a maximum stress in the knuckle equal to the material yield strength.

$$P_y = \frac{C_3 t}{C_2 R_{th}\left(\frac{R_{th}}{2r} - 1\right)} \tag{4.3.17}$$

If the allowable stress at the design temperature is governed by time-independent properties, then C_3 is the material yield strength at the design temperature, or $C_3 = S_y$. If the allowable stress at the design temperature is governed by time-dependent properties, then C_3 is determined as follows.

(a) If the allowable stress is established based on 90% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.1, or $C_3 = 1.1S$.

(b) If the allowable stress is established based on 67% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.5, or $C_3 = 1.5S$.

Step 7. Calculate the value of internal pressure expected to result in a buckling failure of the knuckle.

$$P_{ck} = 0.6P_{eth} \qquad \text{for} \qquad G \le 1.0 \tag{4.3.18}$$

$$P_{ck} = \left(\frac{0.77508G - 0.20354G^2 + 0.019274G^3}{1 + 0.19014G - 0.089534G^2 + 0.0093965G^3}\right)P_y \qquad \text{for} \qquad G > 1.0 \tag{4.3.19}$$

where

$$G = \frac{P_{eth}}{P_y} \tag{4.3.20}$$

Step 8. Calculate the allowable pressure based on a buckling failure of the knuckle.

$$P_{ak} = \frac{P_{ck}}{1.5} \tag{4.3.21}$$

Step 9. Calculate the allowable pressure based on rupture of the crown.

$$P_{ac} = \frac{2SE}{\frac{L}{t} + 0.5} \tag{4.3.22}$$

Step 10. Calculate the maximum allowable internal pressure.

$$P_a = \min\left[P_{ak}, P_{ac}\right] \tag{4.3.23}$$

Step 11. If the allowable internal pressure computed from Step 10 is greater than or equal to the design pressure, then the design is complete. If the allowable internal pressure computed from Step 10 is less than the design pressure, then increase the head thickness and repeat Steps 2 through 10. This process is continued until an acceptable design is achieved.

4.3.6.2 Torispherical Heads With Different Crown and Knuckle Thicknesses. A torispherical head formed from several welded components as shown in Figure 4.3.4 may have a smaller thickness in the spherical crown than in the knuckle region. The transition in thickness shall be located on the inside surface of the thicker part, and shall have a taper not exceeding 1:3.

(a) The minimum required thickness of the spherical dome of the head shall be determined in accordance with 4.3.5.

(b) The minimum required thickness of the knuckle region of the head shall be determined in accordance with 4.3.6.1, Step 2.

4.3.6.3 Combined Loadings. Torispherical heads subject to internal pressure and other loadings shall satisfy the requirements of 4.3.10. In this calculation, the torispherical head shall be approximated as an equivalent spherical shell with a radius equal to L.

4.3.7 ELLIPSOIDAL HEADS

4.3.7.1 Required Thickness. The minimum required thickness of an ellipsoidal head (see Figure 4.3.5) subjected to internal pressure shall be calculated using the equations in 4.3.6 with the following substitutions for r and L.

$$r = D\left(\frac{0.5}{k} - 0.08\right) \tag{4.3.24}$$

$$L = D\left(0.44k + 0.02\right) \tag{4.3.25}$$

where

$$k = \frac{D}{2h} \tag{4.3.26}$$

The rules in this paragraph are applicable for elliptical heads that satisfy eq. (4.3.27). Elliptical heads that do not satisfy this equation shall be designed using Part 5.

$$1.7 \le k \le 2.2 \tag{4.3.27}$$

4.3.7.2 Combined Loadings. ellipsoidal heads subject to internal pressure and other loadings shall satisfy the requirements of 4.3.10. In this calculation, the ellipsoidal head shall be approximated as an equivalent spherical shell with a radius equal to L.

4.3.8 LOCAL THIN AREAS

4.3.8.1 Local Thin Areas. Rules for the evaluation of Local Thin Areas are covered in 4.14.

4.3.8.2 Local Thin Band in Cylindrical Shells. A complete local circumferential band of reduced thickness at a weld joint in a cylindrical shell as shown in Figure 4.3.6 is permitted providing all of the following requirements are met.

(a) The design of the local reduced thickness band is evaluated by limit-load or elastic–plastic analysis in accordance with Part 5. All other applicable requirements of Part 5 for stress analysis and fatigue analysis shall be satisfied.

(b) The cylinder geometry shall satisfy $R_m/t \geq 10$.

(c) The thickness of the reduced shell region shall not be less than two-thirds of the cylinder required thickness determined in accordance with 4.3.3.

(d) The reduced thickness region shall be on the outside of the vessel shell with a minimum taper transition of 3:1 in the base metal. The transition between the base metal and weld shall be designed to minimize stress concentrations.

(e) The total longitudinal length of each local thin region shall not exceed $\sqrt{R_m t}$ (see Figure 4.3.6).

(f) The minimum longitudinal distance from the thicker edge of the taper to an adjacent structural discontinuity shall be the greater of $2.5\sqrt{R_m t}$ or the distance required to assure that overlapping of areas where the primary membrane stress intensity exceeds $1.1S$ does not occur.

4.3.9 DRILLED HOLES NOT PENETRATING THROUGH THE VESSEL WALL

4.3.9.1 Design requirements for partially drilled holes that do not penetrate completely through the vessel wall are provided in this paragraph. These rules are not applicable for studded connections or telltale holes.

4.3.9.2 Partially drilled radial holes in cylindrical and spherical shells may be used, provided the following requirements are satisfied.

(a) The drilled hole diameter is less than or equal to 50 mm (2 in.).

(b) The shell inside diameter to thickness ratio is greater than or equal to 10.

(c) The centerline distance between any two partially drilled holes or between a partially drilled hole and an unreinforced opening shall satisfy the requirements of 4.5.13.

(d) Partially drilled holes shall not be placed within the limits of reinforcement of a reinforced opening.

(e) The outside edge of the hole shall be chamfered. For flat bottom holes, the inside bottom corner of the hole shall have a minimum radius, r_{hr} of the following:

$$r_{hr} = \min\left[\frac{d}{4},\ 6\text{ mm}\left(0.25\text{ in.}\right)\right] \tag{4.3.28}$$

(f) The minimum acceptable remaining wall thickness, t_{rw} at the location of a partially drilled hole shall be determined as follows:

$$t_{rw} \geq \max\left[t_{rw1},\ 0.25t,\ 6\text{ mm}\ \left(0.25\text{ in.}\right)\right] \tag{4.3.29}$$

where,

$$t_{rw1} = t\left(-1.2261727 + 1.9842895\left(\frac{d}{D}\right) - 2.236553\left(\frac{d}{D}\right)^{0.5}\ln\left[\frac{d}{D}\right]\right) \tag{4.3.30}$$

(g) The calculated average shear stress, as determined below shall not exceed $0.8S$.

$$\tau_{pd} = \frac{Pd}{4t_{rw}} \tag{4.3.31}$$

4.3.10 COMBINED LOADINGS AND ALLOWABLE STRESSES

4.3.10.1 General. The rules of this paragraph shall be used to determine the acceptance criteria for stresses developed in cylindrical, spherical, and conical shells subjected to internal pressure plus supplemental loads of applied net section axial force, bending moment, and torsional moment, as shown in Figure 4.3.7. The rules in this paragraph are only applicable to cylindrical, spherical, and conical shells where the wall thickness is determined using the rules in 4.3.3 through 4.3.5, respectively. These rules are applicable if the requirements shown below are satisfied. If all of these requirements are not satisfied, the shell section shall be designed per Part 5.

(a) The rules are applicable for regions of shells that are $2.5\sqrt{Rt}$ from any gross structural discontinuity.

(b) These rules do not take into account the action of shear forces, since these loads generally can be disregarded.

(c) The ratio of the shell inside radius to thickness is greater than 3.0.

4.3.10.2 The following procedure shall be used to determine the acceptance criteria for stresses developed in cylindrical, spherical, and conical shells subjected to internal pressure plus supplemental loads of applied net section axial force, bending moment, and torsional moment.

Step 1. Calculate the membrane stress.

(a) For cylindrical shells:

$$\sigma_{\theta m} = \frac{PD}{E(D_o - D)} \tag{4.3.32}$$

$$\sigma_{sm} = \frac{1}{E}\left(\frac{PD^2}{D_o^2 - D^2} + \frac{4F}{\pi\left(D_o^2 - D^2\right)} \pm \frac{32MD_o\cos\left[\theta\right]}{\pi\left(D_o^4 - D^4\right)}\right) \tag{4.3.33}$$

$$\tau = \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)} \tag{4.3.34}$$

(b) For spherical shells for 0 deg < ϕ < 180 deg:

$$\sigma_{\theta m} = \frac{PD^2}{E\left(D_o^2 - D^2\right)} \tag{4.3.35}$$

$$\sigma_{sm} = \frac{1}{E}\left(\frac{PD^2}{D_o^2 - D^2} + \frac{4F}{\pi\left(D_o^2 - D^2\right)\sin^2\left[\phi\right]} \pm \frac{32MD_o\cos\left[\theta\right]}{\pi\left(D_o^4 - D^4\right)\sin^3\left[\phi\right]}\right) \tag{4.3.36}$$

$$\tau = \frac{32MD_o}{\pi\left(D_o^4 - D^4\right)}\frac{\cos\left[\phi\right]}{\sin^3\left[\phi\right]}\sin\left[\theta\right] + \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)\sin^2\left[\phi\right]} \tag{4.3.37}$$

(c) For conical shells for α ≤ 60 deg:

$$\sigma_{\theta m} = \frac{PD}{E(D_o - D)\cos[\alpha]} \tag{4.3.38}$$

$$\sigma_{sm} = \frac{1}{E}\left(\frac{PD^2}{\left(D_o^2 - D^2\right)\cos\left[\alpha\right]} + \frac{4F}{\pi\left(D_o^2 - D^2\right)\cos\left[\alpha\right]} \pm \frac{32MD_o\cos\left[\theta\right]}{\pi\left(D_o^4 - D^4\right)\cos\left[\alpha\right]}\right) \tag{4.3.39}$$

$$\tau = \frac{32MD_o}{\pi\left(D_o^4 - D^4\right)}\tan\left[\alpha\right]\sin\left[\theta\right] + \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)} \tag{4.3.40}$$

Step 2. Calculate the principal stresses.

$$\sigma_1 = 0.5\left(\sigma_{\theta m} + \sigma_{sm} + \sqrt{\left(\sigma_{\theta m} - \sigma_{sm}\right)^2 + 4\tau^2}\right) \tag{4.3.41}$$

$$\sigma_2 = 0.5\left(\sigma_{\theta m} + \sigma_{sm} - \sqrt{\left(\sigma_{\theta m} - \sigma_{sm}\right)^2 + 4\tau^2}\right) \tag{4.3.42}$$

$$\sigma_3 = \sigma_r = 0 \text{ for stress on the outside surface} \tag{4.3.43}$$

Step 3. At any point on the shell, the following limit shall be satisfied.

$$\frac{1}{\sqrt{2}}\left[\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2\right]^{0.5} \le S \tag{4.3.44}$$

Step 4. For cylindrical and conical shells, if the axial membrane stress, σ_{sm}, is compressive, then eq. (4.3.45) shall be satisfied where F_{xa} is evaluated using 4.4.12.2 with $\lambda = 0.15$. For spherical shells, the allowable compressive stress criteria in 4.4.12.4 shall be satisfied. Note that the controlling condition for this case may be the combined loadings without internal pressure.

$$\sigma_{sm} \le F_{xa} \tag{4.3.45}$$

4.3.11 CYLINDRICAL-TO-CONICAL SHELL TRANSITION JUNCTIONS WITHOUT A KNUCKLE

4.3.11.1 The following rules are applicable for the design of conical transitions or circular cross-sections that do not have a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment. Acceptable conical transition details are shown in Figure 4.3.8. Design rules for a knuckle at the large end or flare at the small end are provided in 4.3.12.

4.3.11.2 Design rules are provided for the cylinder-to-cone junction details shown in Figure 4.3.9. Details with a stiffening ring at the cylinder-to-cone junction, or other details that differ from the ones shown in this figure shall be designed in accordance with Part 5.

4.3.11.3 The length of the conical shell, measured parallel to the surface of the cone shall be equal to or greater than the following value.

$$L_c \ge 2.0\sqrt{\frac{R_L t_c}{\cos[\alpha]}} + 1.4\sqrt{\frac{R_s t_c}{\cos[\alpha]}} \tag{4.3.46}$$

4.3.11.4 The procedure that shall be used to design the large end of a cylinder-to-cone junction without a knuckle is described below.

Step 1. Compute the required thickness of the cylinder at the large end of the cone-to-cylinder junction using 4.3.3, and select the nominal thickness, t_L.

Step 2. Determine the cone half-apex angle, α; compute the required thickness of the cone at the large end of the cone-to-cylinder junction using 4.3.4; and select the nominal thickness, t_C.

Step 3. Proportion the cone geometry such that eq. (4.3.46) and the following equations are satisfied. If all of these equations are not satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5. In the calculations, if 0 deg < α ≤ 10 deg, then use α = 10 deg.

$$20 \le \frac{R_L}{t_L} \le 500 \tag{4.3.47}$$

$$1 \le \frac{t_C}{t_L} \le 2 \tag{4.3.48}$$

$$\alpha \le 60 \text{ deg} \tag{4.3.49}$$

Step 4. Determine the net section axial force, F_L, and bending moment, M_L, applied to the conical transition. The thrust load due to pressure shall not be included as part of the axial force, F_L. Determine an equivalent, X_L, using eq. (4.3.50).

$$X_L = \frac{F_L}{2\pi R_L} \pm \frac{M_L}{\pi R_L^2} \tag{4.3.50}$$

Step 5. Compute the junction transition design parameters. For calculated values of n other than those presented in Tables 4.3.3 and 4.3.4, linear interpolation of the equation coefficients, C_i, is permitted.

$$n = \frac{t_C}{t_L} \tag{4.3.51}$$

$$H = \sqrt{\frac{R_L}{t_L}} \tag{4.3.52}$$

$$B = \tan\left[\alpha\right] \tag{4.3.53}$$

Step 6. Compute the stresses in the cylinder and cone at the junction using the equations in Table 4.3.1. The allowable stress criterion for a tensile stress is provided in Table 4.3.1. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not a concern if the limits given in eqs. (4.3.54) and (4.3.55) are satisfied. F_{ha} is evaluated using 4.4.5.1, but substituting $F_{he} = 0.4E\left(\frac{t}{D_o}\right)$. F_{xa} is evaluated using 4.4.12.2(b) with $\lambda = 0.15$. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

$$\sigma_{\theta m} \le F_{ha} \tag{4.3.54}$$

$$\sigma_{sm} \le F_{xa} \tag{4.3.55}$$

Step 7. If the stress acceptance criterion in Step 6 is satisfied, then the design is complete. If the stress acceptance criterion in Step 6 is not satisfied, the cylinder thickness or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. The section of increased thickness for the cylinder and cone shall extend a minimum distance from the junction as shown in Figure 4.3.9. Proceed to Step 3 to repeat the calculation with the new wall thickness.

4.3.11.5 The procedure that shall be used to design the small end of a cylinder-to-cone junction without a flare is described below.

Step 1. Compute the required thickness of the cylinder at the small end of the cone-to-cylinder junction using 4.3.3, and select the nominal thickness, t_S.

Step 2. Determine the cone half-apex angle, α; compute the required thickness of the cone at the small end of the cone-to-cylinder junction using 4.3.4; and select the nominal thickness, t_C.

Step 3. Proportion the cone geometry such that eq. (4.3.46) and the following equations are satisfied. If all of these equations are not satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5. In the calculations, if 0 deg < α ≤ 10 deg, then use α = 10 deg.

$$20 \le \frac{R_S}{t_S} \le 500 \tag{4.3.56}$$

$$1 \le \frac{t_C}{t_S} \le 2 \tag{4.3.57}$$

$$\alpha \le 60 \text{ deg} \tag{4.3.58}$$

Step 4. Determine the net section axial force, F_S, and bending moment, M_s, applied to the conical transition. The thrust load due to pressure shall not be included as part of the axial force, F_S. Determine an equivalent, X_s, line using eq. (4.3.59).

$$X_s = \frac{F_S}{2\pi R_S} \pm \frac{M_S}{\pi R_S^2} \tag{4.3.59}$$

Step 5. Compute the junction transition design parameters. For calculated values of n other than those presented in Tables 4.3.5 and 4.3.6, linear interpolation of the equation coefficients, C_i, is permitted.

$$n = \frac{t_C}{t_S} \tag{4.3.60}$$

$$H = \sqrt{\frac{R_S}{t_S}} \tag{4.3.61}$$

$$B = \tan[\alpha] \tag{4.3.62}$$

Step 6. Compute the stresses in the cylinder and cone at the junction using the equations in Table 4.3.2. The allowable stress criterion for a tensile stress is provided in Table 4.3.2. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not of concern if the limits given in eqs. (4.3.54) and (4.3.55) are satisfied, using the procedure provided in 4.3.11.4, Step 6. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

Step 7. If the stress acceptance criterion in Step 6 is satisfied, then the design is complete. If the stress acceptance criterion in Step 6 is not satisfied, the cylinder thickness or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. The section of increased thickness for the cylinder and cone shall extend a minimum distance from the junction as shown in Figure 4.3.9. Proceed to Step 3 to repeat the calculation with the new wall thickness.

4.3.12 CYLINDRICAL-TO-CONICAL SHELL TRANSITION JUNCTIONS WITH A KNUCKLE

4.3.12.1 General. The following rules are applicable for the design of conical transitions of circular cross-section with a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment. Acceptable conical transition details are shown in Figure 4.3.10. Design rules for transition junctions without a knuckle at the large end or flare at the small end are provided in 4.3.11.

4.3.12.2 The procedure that shall be used to design the large end of a cylinder-to-cone junction with a knuckle is described below.

Step 1. Compute the required thickness of the cylinder at the large end of the cone-to-cylinder junction using 4.3.3, and select the nominal thickness, t_L.

Step 2. Determine the cone half-apex angle, α; compute the required thickness of the cone at the large end of the cone-to-cylinder junction using 4.3.4; and select the nominal thickness, t_C.

Step 3. Proportion the transition geometry by assuming a value for the knuckle radius, r_k, and knuckle thickness, t_k, such that the following equations are satisfied. If all of these equations cannot be satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5.

$$t_k \ge t_L \tag{4.3.63}$$

$$r_k > 3t_k \tag{4.3.64}$$

$$\frac{r_k}{R_L} > 0.03 \tag{4.3.65}$$

$$\alpha \leq 60 \text{ deg} \quad (4.3.66)$$

Step 4. Determine the net section axial force, F_L, and bending moment, M_L, applied to the conical transition at the location of the knuckle. The thrust load due to pressure shall not be included as part of the axial force, F_L.

Step 5. Compute the stresses in the cylinder, knuckle, and cone at the junction using the equations in Table 4.3.7. The allowable stress criterion for a tensile stress is provided in Table 4.3.7. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not a concern if the limits given in eqs. (4.3.67) and (4.3.68) are satisfied. F_{ha} is evaluated using 4.4.5.1, but substituting $F_{he} = 0.4E\left(\frac{t}{D_o}\right)$. F_{xa} is evaluated using 4.4.12.2(b) with $\lambda = 0.15$. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

$$\sigma_{\theta m} \leq F_{ha} \quad (4.3.67)$$

$$\sigma_{sm} \leq F_{xa} \quad (4.3.68)$$

Step 6. If the stress acceptance criterion in Step 5 is satisfied, then the design is complete. If the stress acceptance criterion in Step 5 is not satisfied, the knuckle thickness, cylinder thickness, or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. If the cylinder or cone thickness is increased, the section of increased thickness shall extend a length given by eqs. (4.3.69) and (4.3.70), respectively. Proceed to Step 3 to repeat the calculation with the new wall thicknesses.

$$L_{rcy} = K_m \sqrt{R_L t_L} \quad (4.3.69)$$

$$L_{rco} = K_m \sqrt{L_k t_C} \quad (4.3.70)$$

4.3.12.3 The procedure that shall be used to design the small end of a cylinder-to-cone junction with a flare is described below.

Step 1. Compute the required thickness of the cylinder at the small end of the cone-to-cylinder junction using 4.3.3, and select the nominal thickness, t_S.

Step 2. Determine the cone half-apex angle, α; compute the required thickness of the cone at the small end of the cone-to-cylinder junction using 4.3.4; and select the nominal thickness, t_C.

Step 3. Proportion the transition geometry by assuming a value for the flare radius, r_f, and flare thickness, t_f, such that the following equations are satisfied. If all of these equations cannot be satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5.

$$t_f \geq t_s \quad (4.3.71)$$

$$r_f > 3t_f \quad (4.3.72)$$

$$\frac{r_f}{R_S} > 0.03 \quad (4.3.73)$$

$$\alpha \leq 60 \text{ deg} \quad (4.3.74)$$

Step 4. Determine the net section axial force, F_S, and bending moment, M_S, applied to the conical transition at the location of the knuckle. The thrust load due to pressure shall not be included as part of the axial force, F_S.

Step 5. Compute the stresses in the cylinder, flare, and cone at the junction using the equations in Table 4.3.8. The allowable stress criterion for a tensile stress is provided in Table 4.3.8. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not of concern if the limits given in eqs. (4.3.67) and (4.3.68) are satisfied, using the procedure provided in 4.3.12.2, Step 5. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

Step 6. If the stress acceptance criterion in Step 5 is satisfied, then the design is complete. If the stress acceptance criterion in Step 5 is not satisfied, the knuckle thickness, cylinder thickness, or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. If the cylinder or cone thickness is increased, the section of increased thickness shall extend a length given by eqs. (4.3.75) and (4.3.76), respectively. Proceed to Step 3 to repeat the calculation with the new wall thicknesses.

$$L_{rcy} = K_m \sqrt{R_S t_S} \tag{4.3.75}$$

$$L_{rco} = K_m \sqrt{L_f t_C} \tag{4.3.76}$$

4.3.13 NOMENCLATURE

A_R = cross-sectional area of the stiffening ring at the junction.
α = one-half of the apex angle of a conical shell.
α_1 = cone angle in an offset transition.
α_2 = cone angle in an offset transition.
B = curve-fit geometric constant.
β_{co} = geometric factor for the cone.
β_{cy} = geometric factor for the cylinder.
β_f = angle used in the conical transition calculation when a flare is present.
β_{f1} = angle used in the conical transition calculation when a flare is present.
β_{f2} = angle used in the conical transition calculation when a flare is present.
β_k = angle used in the conical transition calculation when a knuckle is present.
β_{k1} = angle used in the conical transition calculation when a knuckle is present.
β_{k2} = angle used in the conical transition calculation when a knuckle is present.
β_{th} = angle used in the torispherical head calculation.
C_1 = angle constant used in the torispherical head calculation.
C_2 = angle constant used in the torispherical head calculation.
C_3 = strength parameter used in the torispherical head calculation.
d = diameter of a drilled hole that does not completely penetrate a shell.
D = inside diameter of a shell or head. For conical shells, the inside diameter at the point under consideration, measured perpendicular to the longitudinal axis (see Figure 4.3.1).
D_o = outside diameter of a shell or head.
E_T = modulus of elasticity at maximum design temperature.
E_{RT} = modulus of elasticity at room temperature.
E = weld joint factor (see 4.2.4), the ligament efficiency (see 4.10.2), or the casting quality factor (see Part 3), as applicable, for the weld seam being evaluated (i.e., longitudinal or circumferential).
F = net-section axial force acting at the point of consideration, a positive force produces an axial tensile stress in the cylinder.
F_L = net-section axial force acting on the large end cylindrical shell, a positive force produces an axial tensile stress in the cylinder.
F_S = net-section axial force acting on the small end cylindrical shell, a positive force produces an axial tensile stress in the cylinder.
F_{ha} = allowable compressive hoop membrane stress as given in 4.4.
F_{xa} = allowable compressive axial membrane stress as given in 4.4.
G = constant used in the torispherical head calculation.
H = curve-fit geometric constant.
h = height of the ellipsoidal head measured to the inside surface.
I_R = moment of inertia of the stiffening ring at the junction.

j_k = number of locations around the knuckle that shall be evaluated, used in the conical transition stress calculation when a non-compact knuckle is present.
j_f = number of locations around the flare that shall be evaluated, used in the conical transition stress calculation when a non-compact flare is present.
k = angle constant used in the torispherical and elliptical head calculation.
K_m = length factor used in the conical transition calculation when a flare or knuckle is present.
K_{pc} = cylinder-to-cone junction plasticity correction factor.
λ = compressive stress factor.
L = inside crown radius of a torispherical head.
L_c = length of the conical shell measured parallel to the surface of the cone.
L_f = length used in the conical transition stress calculation when a flare is present.
$L_{1\,f}$ = length used in the conical transition stress calculation when a flare is present.
$L^{j}_{1\,f}$ = length used in the conical transition stress calculation when a flare is present.
L_k = length used in the conical transition stress calculation when a knuckle is present.
$L_{1\,k}$ = length used in the conical transition stress calculation when a knuckle is present.
$L^{j}_{1\,k}$ = length used in the conical transition stress calculation when a knuckle is present.
M = net-section bending moment acting at the point of consideration.
M_{cs} = total resultant meridional moment acting on the cone.
M_{csP} = cylinder-to-cone junction resultant meridional moment acting on the cone, due to internal pressure.
M_{csX} = cylinder-to-cone junction resultant meridional moment acting on the cone, due to an equivalent line load.
M_s = total resultant meridional moment acting on the cylinder.
M_{sP} = cylinder-to-cone junction resultant meridional moment acting on the cylinder, due to internal pressure.
M_{sX} = cylinder-to-cone junction resultant meridional moment acting on the cylinder, due to an equivalent line load.
M_{sN} = normalized curve-fit resultant meridional moment acting on the cylinder.
M_L = net-section bending moment acting at the large end cylindrical shell.
M_S = net-section bending moment acting at the small end cylindrical shell.
M_t = net-section torsional moment acting on a shell section.
N_{cs} = resultant meridional membrane force acting on the cone, due to pressure plus an equivalent line load.
$N_{c\theta}$ = resultant circumferential membrane force acting on the cone, due to pressure plus an equivalent line load.
N_s = resultant meridional membrane force acting on the cylinder, due to pressure plus an equivalent line load.
N_θ = resultant circumferential membrane force acting on the cylinder, due to pressure plus an equivalent line load.
n = ratio of the thickness of the cone to the thickness of the cylinder.
P = internal design pressure.
P_a = maximum allowable internal pressure of a torispherical head.
P_{ac} = allowable internal pressure of a torispherical head based on the rupture of the crown.
P_{ak} = allowable internal pressure of a torispherical head based on a buckling failure of the knuckle.
P_{ck} = value of internal pressure expected to result in a buckling failure of the knuckle in a torispherical head.
P_e = equivalent design pressure used in the conical transition stress calculation when a knuckle or flare is present.
P_e^{j} = equivalent design pressure at locations around the knuckle or flare, used in the conical transition stress calculation when a knuckle or flare is present.
P_{eth} = value of internal pressure expected to produce elastic buckling of the knuckle in a torispherical head.
P_y = value of the internal pressure expected to result in a maximum stress equal to the material yield strength in a torispherical head.
ϕ = angle to locate a circumferential section in a spherical shell.
ϕ_f = angle used in the conical transition calculation when a flare is present.
ϕ^{j}_{f} = angle used in the conical transition calculation when a non-compact flare is present.
ϕ^{e}_{f} = angle used in the conical transition calculation when a non-compact flare is present.
ϕ^{s}_{f} = angle used in the conical transition calculation when a non-compact flare is present.
ϕ_k = angle used in the conical transition calculation when a knuckle is present.
ϕ^{j}_{k} = angle used in the conical transition calculation when a non-compact knuckle is present.
ϕ^{e}_{k} = angle used in the conical transition calculation when a non-compact knuckle is present.
ϕ^{s}_{k} = angle used in the conical transition calculation when a non-compact knuckle is present.
ϕ_{th} = angle used in the torispherical head calculation.
Q = total resultant shear force acting on the cylinder.
Q_c = total resultant shear force acting on the cone.
Q_N = normalized curve-fit resultant shear force acting on the cylinder.
Q_P = cylinder-to-cone junction resultant shear force acting on the cylinder, due to internal pressure.

Q_X = cylinder-to-cone junction resultant shear force acting on the cylinder, due to an equivalent line load.
R_C = equivalent radius of the cone.
R_f = radius to the center of curvature for the flare.
R_k = radius to the center of curvature for the knuckle.
R_L = inside radius of the large end of a conical transition.
R_m = mean radius of the cylinder.
R_S = inside radius of the small end of a conical transition.
R_{th} = radius used in the torispherical head calculation.
r = inside knuckle radius used in torispherical head calculation.
r_{hr} = minimum hole radius.
r_k = inside knuckle radius of the large end of a toriconical transition.
r_f = inside flare radius of the small end of a toriconical transition.
s_R = distance measured along the cylinder from the centroid of the stiffening ring centroid to the intersection of the cylinder and cone.
S = allowable stress value from Annex 3-A evaluated at the design temperature.
S_a = allowable stress amplitude.
S_{PS} = allowable primary plus secondary stress evaluated using 4.1.6.3 at the design temperature.
S_y = yield strength from Annex 3-D evaluated at the design temperature.
σ_r = radial stress in a shell.
σ_s = axial (longitudinal) stress in a shell.
σ_{sm} = axial (longitudinal) membrane stress in a shell.
σ_{sb} = axial (longitudinal) bending stress in a shell.
σ_θ = hoop (circumferential) stress in a shell.
$\sigma_{\theta m}$ = hoop (circumferential) membrane stress in a shell.
$\sigma_{\theta b}$ = hoop (circumferential) bending stress in a shell.
σ_1 = principal stress in the 1-direction.
σ_2 = principal stress in the 2-direction.
σ_3 = principal stress in the 3-direction.
t = minimum required thickness of a shell.
t_C = nominal thickness of the cone in a conical transition at the large end or small end as applicable.
t_L = nominal thickness of the large end cylinder in a conical transition.
t_S = nominal thickness of the small end cylinder in a conical transition.
t_j = thickness of the cylinder, knuckle, or flue, as applicable, at the junction of a toriconical transition, $t_j \geq t$ and $t_j \geq t_c$.
t_{rw} = remaining wall thickness at the location of a partially drilled hole.
t_{rw1} = limit for the remaining wall thickness at the location of a partially drilled hole.
τ = torsional shear stress in a shell.
τ_{pd} = average shear stress in a shell at the location of a partially drilled hole. location where stress is computed for shells subject to supplemental loads. A value of zero defines the location of maximum positive longitudinal stress from net-section bending moment.
v = Poisson's ratio.
X_L = equivalent line load acting on the large end cylinder, due to an axial force and bending moment.
X_S = equivalent line load acting on the small end cylinder, due to an axial force and bending moment.

4.3.14 TABLES

Table 4.3.1
Large End Junction

Cylinder	Cone
Stress Resultant Calculation	
$M_{sP} = Pt_L^2 M_{sN}$, see Table 4.3.3	$M_{csP} = M_{sP}$
$M_{sX} = X_L t_L M_{sN}$, see Table 4.3.4	$M_{csX} = M_{sX}$
$M_s = M_{sP} + M_{sX}$	$M_{cs} = M_{csP} + M_{csX}$
$Q_P = P_{tL} Q_N$, see Table 4.3.3	$Q_c = Q_{cos}[\alpha] + N_s \sin[\alpha]$ **[Note (1)]**
$Q_X = X_L Q_N$, see Table 4.3.4	$R_c = \dfrac{R_L}{\cos\left[\alpha\right]}$
$Q = Q_P + Q_x$	
$\beta_{cy} = \left[\dfrac{3\left(1-\nu^2\right)}{R_L^2 t_L^2}\right]^{0.25}$	$\beta_{co} = \left[\dfrac{3\left(1-\nu^2\right)}{R_c^2 t_c^2}\right]^{0.25}$
$N_s = \dfrac{PR_L}{2} + X_L$	$N_{cs} = N_s \cos[\alpha] - Q\sin[\alpha]$ **[Note (2)]**
$N_\theta = PR_L + 2\beta_{cy} R_L(-M_s\beta_{cy} + Q)$	$N_{c\theta} = \dfrac{PR_L}{\cos\left[\alpha\right]} + 2\beta_{co} R_c\left(-M_{cs}\beta_{co} - Q_c\right)$
$K_{pc} = 1.0$	$K_{cpc} = 1.0$
Stress Calculation	
$\sigma_{sm} = \dfrac{N_s}{t_L}$	$\sigma_{sm} = \dfrac{N_{cs}}{t_C}$
$\sigma_{sb} = \dfrac{6M_s}{t_L^2 K_{pc}}$	$\sigma_{sb} = \dfrac{6M_{cs}}{t_C^2 K_{pc}}$
$\sigma_{\theta m} = \dfrac{N_\theta}{t_L}$	$\sigma_{\theta m} = \dfrac{N_{c\theta}}{t_c}$
$\sigma_{\theta b} = \dfrac{6\nu M_s}{t_L^2 K_{pc}}$	$\sigma_{\theta b} = \dfrac{6\nu M_{cs}}{t_c^2 K_{cpc}}$
Acceptance Criteria	
$\sigma_{sm} \le 1.5S$	$\sigma_{sm} \le 1.5S$
$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$	$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$
$\sigma_{\theta m} \le 1.5S$	$\sigma_{\theta m} \le 1.5S$
$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$	$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$

NOTES:
(1) The Q and N_s values used to determine the resultant shear force in the cone, Q_c, are the same as those defined for the cylinder.
(2) The Q and N_s values used to determine the resultant meridional membrane force in the cone, N_{cs}, are the same as those defined for the cylinder.

Table 4.3.2
Small End Junction

Cylinder	Cone
Stress Resultant Calculation	
$M_{sP} = Pt_S^2 M_{sN}$, see Table 4.3.5	$M_{csP} = M_{sP}$
$M_{sX} = X_S t_S M_{sN}$, see Table 4.3.6	$M_{csX} = M_{sX}$
$M_s = M_{sP} + M_{sX}$	$M_{cs} = M_{csP} + M_{csX}$
$Q_P = Pt_S Q_N$, see Table 4.3.5	$Q_c = Q\cos[\alpha] + N_s \sin[\alpha]$ [Note (1)]
$Q_X = X_S Q_N$, see Table 4.3.6	$R_c = \dfrac{R_s}{\cos[\alpha]}$
$Q = Q_P + Q_x$	
$\beta_{cy} = \left[\dfrac{3(1-\nu^2)}{R_S^2 t_S^2}\right]^{0.25}$	$\beta_{co} = \left[\dfrac{3(1-\nu^2)}{R_c^2 t_c^2}\right]^{0.25}$
$N_s = \dfrac{PR_s}{2} + X_s$	$N_{cs} = N_s \cos[\alpha] - Q\sin[\alpha]$ [Note (2)]
$N_\theta = PR_s + 2\beta_{cy} R_s\left(-M_s \beta_{cy} - Q\right)$	$N_{c\theta} = \dfrac{PR_s}{\cos[\alpha]} + 2\beta_{co} R_C\left(-M_{cs}\beta_{co} + Q_c\right)$
$K_{pc} = 1.0$	$K_{cpc} = 1.0$
Stress Calculation	
$\sigma_{sm} = \dfrac{N_s}{t_S}$	$\sigma_{sm} = \dfrac{N_{cs}}{t_C}$
$\sigma_{sb} = \dfrac{6M_s}{t_s^2 K_{pc}}$	$\sigma_{sb} = \dfrac{6M_{cs}}{t_c^2 K_{cpc}}$
$\sigma_{\theta m} = \dfrac{N_\theta}{t_s}$	$\sigma_{\theta m} = \dfrac{N_{c\theta}}{t_c}$
$\sigma_{\theta b} = \dfrac{6\nu M_s}{t_s^2 K_{pc}}$	$\sigma_{\theta b} = \dfrac{6\nu M_{cs}}{t_c^2 K_{cpc}}$
Acceptance Criteria	
$\sigma_{sm} \le 1.5S$	$\sigma_{sm} \le 1.5S$
$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$	$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$
$\sigma_{\theta m} \le 1.5S$	$\sigma_{\theta m} \le 1.5S$
$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$	$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$

NOTES:
(1) The Q and N_s values used to determine the resultant shear force in the cone, Q_c, are the same as those defined for the cylinder.
(2) The Q and N_s values used to determine the resultant meridional membrane force in the cone, N_{cs}, are the same as those defined for the cylinder.

Table 4.3.3
Pressure Applied to Large End Junction

Equation Coefficients, C_i	n = 1	n = 1.25	n = 1.5	n = 1.75	n = 2
Junction Moment Resultant, M_{sN} [Note (1)]					
1	−3.065534	−3.113501	−3.140885	−3.129850	−3.115764
2	3.642747	3.708036	3.720338	3.674582	3.623956
3	0.810048	0.736679	0.623373	0.490738	0.360998
4	−0.221192	−0.239151	−0.241393	−0.224678	−0.209963
5	−0.081824	−0.075734	−0.056744	−0.034581	−0.013613
6	0.035052	0.083171	0.157222	0.240314	0.316184
7	0.025775	0.027432	0.027393	0.025163	0.023508
8	−0.015413	−0.015659	−0.017311	−0.019456	−0.021796
9	0.002102	0.000993	−0.004600	−0.011145	−0.017172
10	−0.005587	−0.013283	−0.025609	−0.039144	−0.050859
Junction Shear Force Resultant, Q_N [Note (1)]					
1	−1.983852	−1.911375	−1.893640	−1.852083	−1.816642
2	2.410703	2.292069	2.253430	2.184549	2.126469
3	0.626443	0.478030	0.364794	0.251818	0.152468
4	−0.119151	−0.079165	−0.075123	−0.059024	−0.048876
5	−0.115841	−0.074658	−0.047032	−0.024214	−0.007486
6	0.122993	0.219247	0.282565	0.343492	0.390839
7	0.012160	0.007250	0.007505	0.006116	0.005632
8	−0.016987	−0.021607	−0.024667	−0.027144	−0.029118
9	0.010919	−0.003818	−0.012439	−0.018971	−0.023076
10	−0.016653	−0.033814	−0.043500	−0.052435	−0.058417

NOTE:
(1) The equation to determine M_{sN} and Q_N is shown below.

$$M_{sN}, Q_N = -\exp\left[\begin{array}{c} C_1 + C_2\ln\left[H\right] + C_3\ln\left[B\right] + C_4\left(\ln\left[H\right]\right)^2 + C_5\left(\ln\left[B\right]\right)^2 + C_6\ln\left[H\right]\ln\left[B\right] \\ C_7\left(\ln\left[H\right]\right)^3 + C_8\left(\ln\left[B\right]\right)^3 + C_9\ln\left[H\right]\left(\ln\left[B\right]\right)^2 + C_{10}\left(\ln\left[H\right]\right)^2\ln\left[B\right] \end{array}\right]$$

Table 4.3.4
Equivalent Line Load Applied to Large End Junction

Equation Coefficients, C_i	n = 1	n = 1.25	n = 1.5	n = 1.75	n = 2
Junction Moment Resultant, M_{sN} [Note (1)]					
1	−5.697151	−5.727483	−5.893323	−6.159334	−6.532748
2	0.003838	0.006762	0.012440	0.019888	0.029927
3	0.476317	0.471833	0.466370	0.461308	0.454550
4	−0.213157	−0.213004	−0.211065	−0.207037	−0.200411
5	2.233703	2.258541	2.335015	2.449057	2.606550
6	0.000032	0.000010	−0.000006	−0.000008	−0.000004
7	0.002506	0.003358	0.004949	0.007005	0.009792
8	−0.001663	−0.002079	−0.003105	−0.004687	−0.007017
9	−0.212965	−0.216613	−0.224714	−0.235979	−0.251220
10	0.000138	−0.000108	−0.000721	−0.001597	−0.002797
11	−0.106203	−0.106269	−0.107142	−0.108733	−0.110901
Junction Shear Force Resultant, Q_N [Note (1)]					
1	−4.774616	−5.125169	−5.556823	−6.113380	−6.858200
2	0.000461	0.021875	0.049082	0.084130	0.131374
3	−0.002831	−0.055928	−0.127941	−0.225294	−0.361885
4	−0.197117	−0.196848	−0.196204	−0.194732	−0.193588
5	1.982132	2.156708	2.378102	2.668633	3.069269
6	0.000069	−0.000450	−0.001077	−0.001821	−0.002760
7	−0.000234	0.000188	0.000821	0.001694	0.002958
8	−0.003536	−0.005341	−0.007738	−0.010934	−0.015089
9	−0.202493	−0.223872	−0.251223	−0.287283	−0.337767
10	−0.000088	−0.002426	−0.005428	−0.009440	−0.015045
11	0.001365	0.012698	0.027686	0.047652	0.075289

NOTE:
(1) The equation to determine M_{sN} and Q_N is shown below.

$$M_{sN}, Q_N = -\exp\left[\frac{C_1 + C_3\ln\left[H^2\right] + C_5\ln\left[\alpha\right] + C_7\left(\ln\left[H^2\right]\right)^2 + C_9\left(\ln\left[\alpha\right]\right)^2 + C_{11}\ln\left[H^2\right]\ln\left[\alpha\right]}{1 + C_2\ln\left[H^2\right] + C_4\ln\left[\alpha\right] + C_6\left(\ln\left[H^2\right]\right)^2 + C_8\left(ln\left[\alpha\right]\right)^2 + C_{10}\ln\left[H^2\right]\ln\left[\alpha\right]}\right]$$

Table 4.3.5
Pressure Applied to Small End Junction

Equation Coefficients, C_i	n = 1	n = 1.25	n = 1.5	n = 1.75	n = 2
Junction Moment Resultant, M_{sN} [Note (1)]					
1	−9.615764	−10.115298	−11.531005	−14.040332	−18.457734
2	1.755095	1.858053	2.170806	2.762452	3.859890
3	3.937841	4.222547	4.872664	5.973215	7.923210
4	−0.043572	−0.053476	−0.080011	−0.131830	−0.228146
5	−1.035596	−1.100505	−1.213287	−1.388782	−1.685101
6	−0.008908	−0.033941	−0.121942	−0.288589	−0.612009
7	0.003984	0.004388	0.005287	0.006975	0.010041
8	0.115270	0.121595	0.129218	0.139465	0.154368
9	0.013712	0.015269	0.022097	0.034632	0.059879
10	−0.007031	−0.006067	−0.002848	0.003867	0.017109
Junction Shear Force Resultant, Q_N [Note (2)]					
1	0.028350	0.207327	0.376538	0.532382	0.682418
2	0.000020	0.000007	−0.000008	−0.000023	−0.000040
3	0.001668	0.003856	0.005918	0.007947	0.009881
4	0.002987	0.002885	0.002781	0.002709	0.002632
5	0.001125	−0.000330	−0.001848	−0.002664	−0.003542
6	0.000000	0.000000	0.000000	0.000000	0.000000
7	0.000001	−0.000001	−0.000003	−0.000005	−0.000006
8	−0.000122	−0.000120	−0.000118	−0.000117	−0.000116
9	−0.000181	−0.000139	−0.000106	−0.000090	−0.000079
10	0.000001	0.000001	0.000001	0.000001	0.000001
11	−0.004724	−0.004417	−0.004128	−0.003847	−0.003570

NOTES:

(1) The equation to determine M_{sN} is shown below.

$$M_{sN} = \exp\left[\begin{array}{c} C_1 + C_2\ln\left[H^2\right] + C_3\ln\left[\alpha\right] + C_4\left(\ln\left[H^2\right]\right)^2 + C_5\left(\ln\left[\alpha\right]\right)^2 + C_6\ln\left[H^2\right]\ln\left[\alpha\right] + \\ C_7\left(\ln\left[H^2\right]\right)^3 + C_8\left(\ln\left[\alpha\right]\right)^3 + C_9\ln\left[H^2\right]\left(\ln\left[\alpha\right]\right)^2 + C_{10}\left(\ln\left[H^2\right]\right)^2\ln\left[\alpha\right] \end{array}\right]$$

(2) The equation to determine Q_N is shown below.

$$Q_N = \left(\frac{C_1 + C_3H^2 + C_5\alpha + C_7H^4 + C_9\alpha^2 + C_{11}H^2\alpha}{1 + C_2H^2 + C_4\alpha + C_6H^4 + C_8\alpha^2 + C_{10}H^2\alpha}\right)$$

Table 4.3.6
Equivalent Line Load Applied to Small End Junction

Equation Coefficients, C_i	n = 1	n = 1.25	n = 1.5	n = 1.75	n = 2
Junction Moment Resultant, M_{sN} [Note (1)]					
1	−0.000792	0.000042	0.002412	0.005766	0.009868
2	−0.000627	−0.000327	−0.000033	0.000236	0.000453
3	−0.001222	−0.001188	−0.001079	−0.000951	−0.000860
4	0.142039	0.132463	0.125812	0.121877	0.120814
5	0.010704	0.009735	0.009802	0.010465	0.010928
6	0.000013	0.000006	−0.000002	−0.000009	−0.000015
7	−0.000006	−0.000001	−0.000006	−0.000008	−0.000008
8	0.009674	0.008839	0.007580	0.006261	0.005044
9	0.006254	0.005493	0.003701	0.001619	0.000381
10	−0.000046	0.000011	0.000088	0.000171	0.000230
11	0.202195	0.208304	0.205169	0.197061	0.186547
Junction Shear Force Resultant, Q_N [Note (2)]					
1	−0.460579	−0.444768	−0.428659	−0.412043	−0.396046
2	−0.002381	0.006711	0.013388	0.019509	0.026272
3	−0.400925	−0.376106	−0.353464	−0.331009	−0.309046
4	0.001550	−0.000672	−0.002169	−0.003562	−0.005266
5	−0.140077	−0.129459	−0.121074	−0.113195	−0.105461
6	0.000793	0.001950	0.002212	0.002168	0.002310
7	−0.000219	−0.000023	0.000098	0.000215	0.000374
8	−0.019081	−0.017115	−0.015814	−0.014699	−0.013625
9	0.000384	0.000618	0.000739	0.000806	0.000860
10	0.000103	0.000006	0.000038	0.000102	0.000117

NOTES:

(1) The equation to determine M_{sN} is shown below.

$$M_{sN}=\left(\frac{C_1+C_3H+C_5B+C_7H^2+C_9B^2+C_{11}HB}{1+C_2H+C_4B+C_6H^2+C_8B^2+C_{10}HB}\right)$$

(2) The equation to determine Q_N is shown below.

$$Q_N=\left(\begin{array}{c} C_1+C_2\ln\left[H\right]+C_3\ln\left[B\right]+C_4\left(\ln\left[H\right]\right)^2+C_5\left(\ln\left[B\right]\right)^2+C_6\ln\left[H\right]\ln\left[B\right]+ \\ C_7\left(\ln\left[H\right]\right)^3+C_8\left(\ln\left[B\right]\right)^3+C_9\ln\left[H\right]\left(\ln\left[B\right]\right)^2+C_{10}\left(\ln\left[H\right]\right)^2\ln\left[B\right] \end{array}\right)$$

Table 4.3.7
Stress Calculations — Knuckle — Large End Cylinder

Compact Knuckle: $\alpha r_k < 2K_m\left(\left\{R_k\left(\alpha^{-1}\cdot\tan[\alpha]\right)^{0.5}+r_k\right\}t_k\right)^{0.5}$ where $K_m = 0.7$

Stress Calculation

$$\sigma_{\theta m}=\frac{PK_m\left(R_L\sqrt{R_Lt_L}+L_k\sqrt{L_kt_C}\right)+\alpha\left(PL_{1k}r_k-0.5P_eL_{1k}^2\right)}{K_m\left(t_L\sqrt{R_Lt_L}+t_C\sqrt{L_kt_C}\right)+\alpha t_kr_k}$$

$$\sigma_{sm}=\frac{P_eL_{1k}}{2t_k}$$

$$P_e=P+\frac{F_L}{\pi L_{1k}^2\cos^2\left[\frac{\alpha}{2}\right]}\pm\frac{2M_L}{\pi L_{1k}^3\cos^3\left[\frac{\alpha}{2}\right]}$$

$$L_k=\frac{R_k}{\cos[\alpha]}+r_k$$

$$L_{1k}=R_k\left(\alpha^{-1}\cdot\tan[\alpha]\right)^{0.5}+r_k$$

Acceptance Criteria

$\sigma_{\theta m} \le S$

$\sigma_{sm} \le S$

Non-Compact Knuckle: $\alpha r_k \ge 2K_m\left(\left\{R_k\left(\alpha^{-1}\cdot\tan[\alpha]\right)^{0.5}+r_k\right\}t_k\right)^{0.5}$ *where* $K_m = 0.7$

Stress Calculation at TL-1

$$\sigma_{\theta m}=\frac{PR_LK_m\sqrt{R_Lt_L}+\beta_k\left(PL_{1k}r_k-0.5P_eL_{1k}^2\right)}{K_m\left(t_L\sqrt{R_Lt_L}+t_k\sqrt{L_{1k}t_k}\right)}$$

$$\sigma_{sm}=\frac{P_eL_{1k}}{2t_k}$$

$$P_e=P+\frac{F_L}{\pi L_{1k}^2\cos^2\left[\frac{\beta_k}{2}\right]}\pm\frac{2M_L}{\pi L_{1k}^3\cos^3\left[\frac{\beta_k}{2}\right]}$$

$$L_{1k}=R_k\left(\beta_k^{-1}\cdot\tan[\beta_k]\right)^{0.5}+r_k$$

$$\beta_k=\left(\frac{K_m}{r_k}\right)\sqrt{R_Lt_k}$$

Stress Calculation at TL-2

$$\sigma_{\theta m}=\frac{PL_kK_m\sqrt{L_kt_C}+(\alpha-\beta_k)\left(PL_{1k}r_k-0.5P_eL_{1k}^2\right)}{K_m\left(t_C\sqrt{L_kt_C}+t_k\sqrt{L_{1k}t_k}\right)}$$

$$\sigma_{sm}=\frac{P_eL_{1k}}{2t_k}$$

$$P_e=P+\frac{F_L}{\pi L_{1k}^2\cos^2[\phi_k]}\pm\frac{2M_L}{\pi L_{1k}^3\cos^3[\phi_k]}$$

$$L_k=\frac{R_k}{\cos[\alpha]}+r_k$$

$$L_{1k}=R_k\left(\left\{\tan[\alpha]-\tan[\beta_k]\right\}\left\{\alpha-\beta_k\right\}^{-1}\right)^{0.5}+r_k$$

$$\beta_k=\alpha-\left(\frac{K_m}{r_k}\right)\sqrt{L_kt_k}$$

$$\phi_k=\frac{(\alpha+\beta_k)}{2}$$

Table 4.3.7
Stress Calculations — Knuckle — Large End Cylinder (Cont'd)

Stress Calculation in the Non-Compact Knuckle Region

Note: The number of locations around the knuckle that shall be evaluated is given by the following equation:

$$j_k = 2\left(\text{int}\left[\frac{\alpha - \beta_{k1} - \beta_{k2}}{\beta_{k1} + \beta_{k2}}\right] + 1\right) + 1$$

where

$$\beta_{k1} = \frac{K_m}{r_k}\sqrt{R_L t_k} \quad \beta_{k2} = \frac{K_m}{r_k}\sqrt{L_k t_k} \quad L_k = \frac{R_k}{\cos[\alpha]} + r_k$$

For $j = 1,\ldots, j_k$, compute

$$\sigma_{\theta m}^{j} = \frac{P L_{1k}^{j}}{t_k} - \frac{P_e^{j}\left(L_{1k}^{j}\right)^2}{2 r_k t_k} \quad \sigma_{sm}^{j} = \frac{P_e^{j} L_{1k}^{j}}{2 t_k}$$

where

$$P_e^{j} = P + \frac{F_L}{\pi\left(L_{1k}^{j}\right)^2 \cos^2\left[\phi_k^{j}\right]} \pm \frac{2M_L}{\pi\left(L_{1k}^{j}\right)^3 \cos^3\left[\phi_k^{j}\right]}$$

$$L_{1k}^{j} = \frac{R_k}{\cos\left[\phi_k^{j}\right]} + r_k$$

$$\phi_k^{j} = \phi_k^{s} + \left(j - 1\right)\left(\frac{\phi_k^{e} - \phi_k^{s}}{j_k - 1}\right) \quad \phi_k^{s} = \beta_{k1} \quad \phi_k^{e} = \alpha - \beta_{k2}$$

Acceptance Criteria

$\sigma_{\theta m} \le S$

$\sigma_{sm} \le S$

$\sigma_{\theta m}^{j} \le S$

$\sigma_{sm}^{j} \le S$

Table 4.3.8
Stress Calculations — Flare — Small End Cylinder

Compact Flare: $\alpha r_f < 2K_m\left(\left\{R_f\left(\alpha^{-1}\cdot\tan[\alpha]\right)^{0.5} - r_f\right\}t_f\right)^{0.5}$ where $K_m = 0.7$

Stress Calculation

$$\sigma_{\theta m} = \frac{PK_m\left(R_s\sqrt{R_s t_s} + L_f\sqrt{L_f t_C}\right) + \alpha\left(P L_{1f} r_f - 0.5 P_e L_{1f}^2\right)}{K_m\left(t_s\sqrt{R_s t_s} + t_C\sqrt{L_f t_C}\right) + \alpha t_f r_f}$$

$$\sigma_{sm} = \frac{P_e L_{1f}}{2t_f}$$

$$P_e = P + \frac{F_s}{\pi L_{1f}^2\cos^2\left[\frac{\alpha}{2}\right]} \pm \frac{2M_s}{\pi L_{1f}^3\cos^3\left[\frac{\alpha}{2}\right]}$$

$$L_f = \frac{R_f}{\cos[\alpha]} - r_f$$

$$L_{1f} = R_f\left(\alpha^{-1}\cdot\tan[\alpha]\right)^{0.5} - r_f$$

Acceptance Criteria

$\sigma_{\theta m} \le S$

$\sigma_{sm} \le S$

Non-Compact Flare: $\alpha r_f \ge 2K_m\left(\left\{R_f\left(\alpha^{-1}\cdot\tan[\alpha]\right)^{0.5} - r_f\right\}t_f\right)^{0.5}$ where $K_m = 0.7$

Table 4.3.8
Stress Calculations — Flare — Small End Cylinder (Cont'd)

Stress Calculation at TL-3

$$\sigma_{\theta m} = \frac{PL_f K_m \sqrt{L_f t_C} + \left(\alpha - \beta_f\right)\left(PL_{1f} r_f + 0.5 P_e L_{1f}^2\right)}{K_m\left(t_C \sqrt{L_f t_C} + t_f \sqrt{L_{1f} t_C}\right)}$$

$$\sigma_{sm} = \frac{P_e L_{1f}}{2t_f}$$

$$P_e = P + \frac{F_s}{\pi L_{1f}^2 \cos^2\left[\phi_f\right]} \pm \frac{2M_s}{\pi L_{1f}^3 \cos^3\left[\phi_f\right]}$$

$$L_f = \frac{R_f}{\cos\left[\alpha\right]} - r_f$$

$$L_{1f} = R_f\left(\left\{\tan\left[\alpha\right] - \tan\left[\beta_f\right]\right\}\left\{\alpha - \beta_f\right\}^{-1}\right)^{0.5} - r_f$$

$$\beta_f = \alpha - \left(\frac{K_m}{r_f}\right)\sqrt{L_f t_f}$$

$$\phi_f = \frac{\left(\alpha + \beta_f\right)}{2}$$

Stress Calculation at TL-4

$$\sigma_{\theta m} = \frac{PR_s K_m \sqrt{R_s t_s} + \beta_f\left(PL_{1f} r_f + 0.5 P_e L_{1f}^2\right)}{K_m\left(t_s \sqrt{R_s t_s} + t_f \sqrt{L_{1f} t_f}\right)}$$

$$\sigma_{sm} = \frac{P_e L_{1f}}{2t_f}$$

$$P_e = P + \frac{F_s}{\pi L_{1f}^2 \cos^2\left[\frac{\beta_f}{2}\right]} \pm \frac{2M_s}{\pi L_{1f}^3 \cos^3\left[\frac{\beta_f}{2}\right]}$$

$$L_{1f} = R_f\left(\beta_f^{-1} \cdot \tan\left[\beta_f\right]\right)^{0.5} - r_f \quad \beta_f = \left(\frac{K_m}{r_f}\right)\sqrt{R_s t_f}$$

Stress Calculation in the Non-Compact Flare Region

Note: The number of locations around the flare that shall be evaluated is given by the following equation: $j_f = 2\left(\mathrm{int}\left[\frac{\alpha - \beta_{f1} - \beta_{f2}}{\beta_{f1} + \beta_{f2}}\right] + 1\right) + 1$

where

$$\beta_{f1} = \frac{K_m}{r_f}\sqrt{L_f t_f}$$

$$\beta_{f2} = \frac{K_m}{r_f}\sqrt{R_s t_f}$$

$$L_f = \frac{R_f}{\cos\left[\alpha\right]} - r_f$$

For $j = 1, \ldots, j_f$, compute

$$\sigma_{\theta m}^j = \frac{PL_{1f}^j}{t_f} + \frac{P_e^j\left(L_{1f}^j\right)^2}{2r_f t_f}$$

$$\sigma_{sm}^j = \frac{P_e^j L_{1f}^j}{2t_f}$$

where

$$P_e^j = P + \frac{F_S}{\pi\left(L_{1f}^j\right)^2 \cos^2\left[\phi_f^j\right]} \pm \frac{2M_S}{\pi\left(L_{1f}^j\right)^3 \cos^3\left[\phi_f^j\right]}$$

$$L_{1f}^j = \frac{R_f}{\cos\left[\phi_f^j\right]} - r_f$$

$$\phi_f^j = \phi_f^s - \left(j - 1\right)\left(\frac{\phi_f^s - \phi_f^e}{j_f - 1}\right)$$

$$\phi_f^s = \alpha - \beta_{f1}$$

$$\phi_k^e = \beta_{f2}$$

Acceptance Criteria

$$\sigma_{\theta m} \le S$$

$$\sigma_{sm} \le S$$

$$\sigma_{\theta m}^j \le S$$

$$\sigma_{sm}^j \le S$$

4.3.15 FIGURES

Figure 4.3.1
Conical Shell

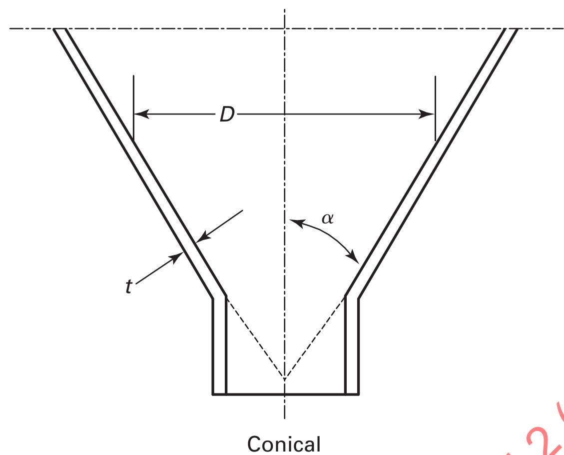


Figure 4.3.2
Offset Transition Detail

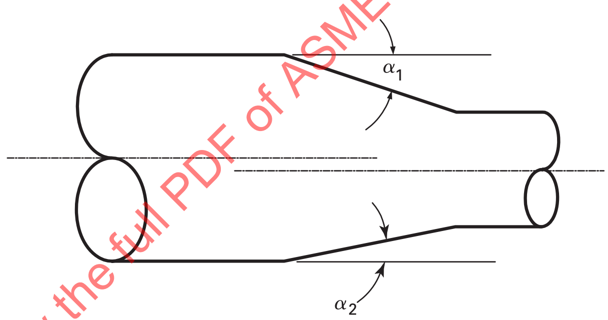


Figure 4.3.3
Torispherical Head of Uniform Thickness

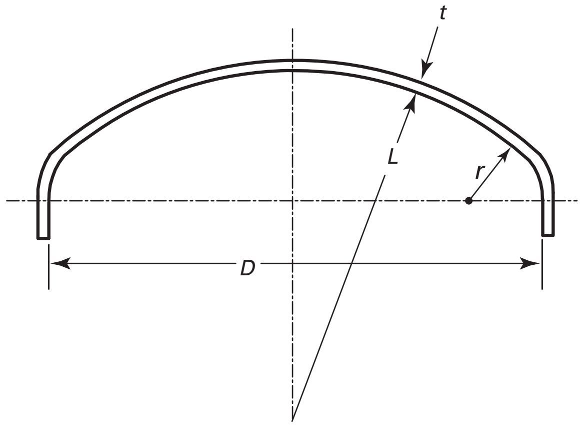


Figure 4.3.4
Torispherical Head of Different Thickness of Dome and Knuckle

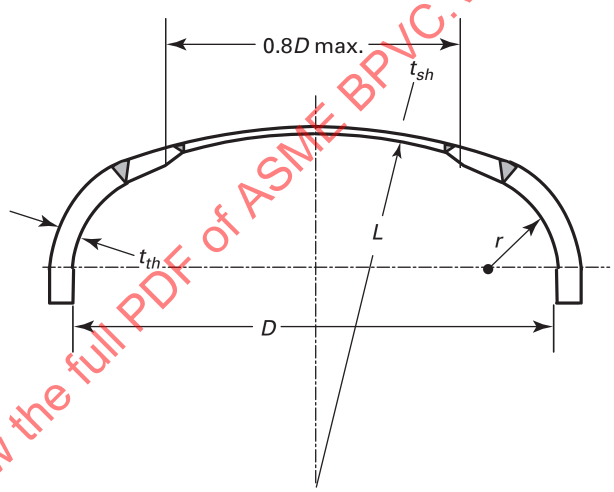


Figure 4.3.5
Ellipsoidal Head

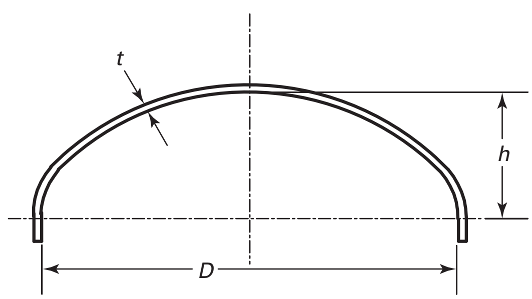


Figure 4.3.6
Local Thin Band in a Cylindrical Shell

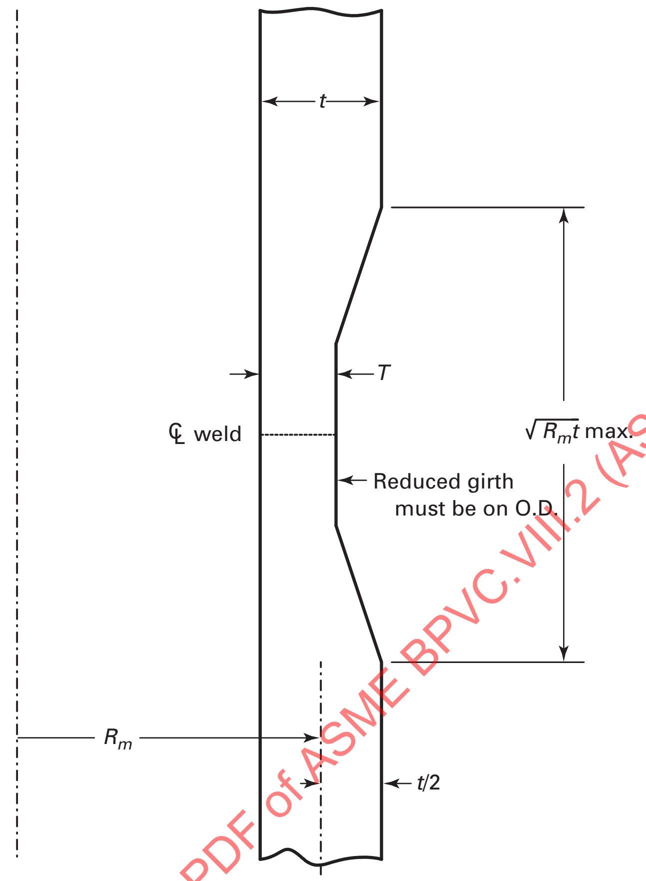
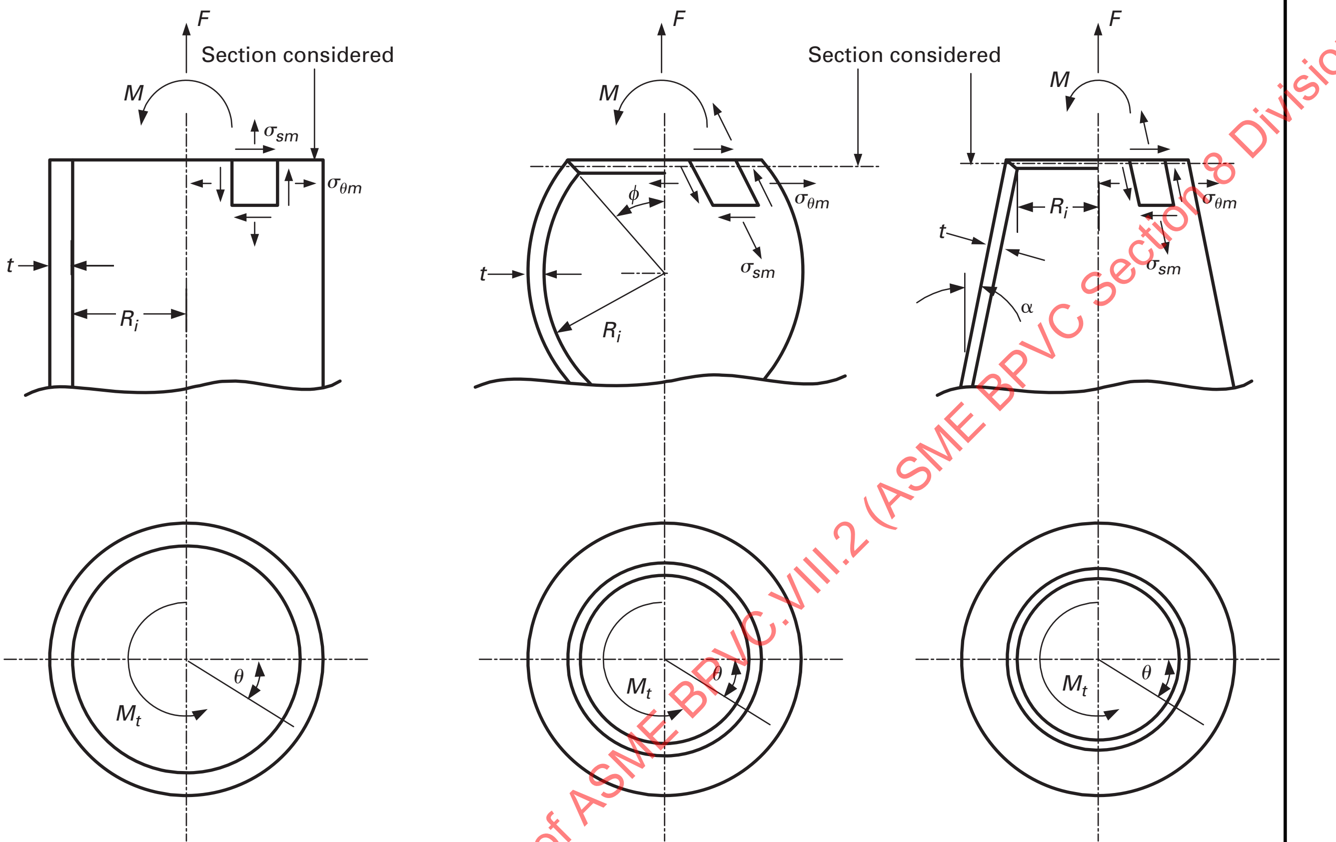


Figure 4.3.7
Shells Subjected to Supplemental Loadings

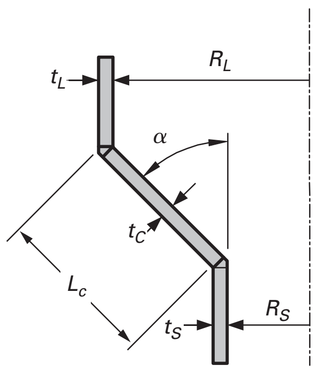


(a) Stresses Within a Cylindrical Shell

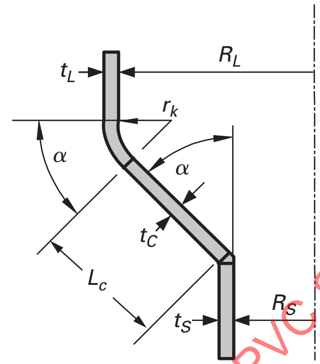
(b) Stresses Within a Spherical Shell

(c) Stresses Within a Conical Shell

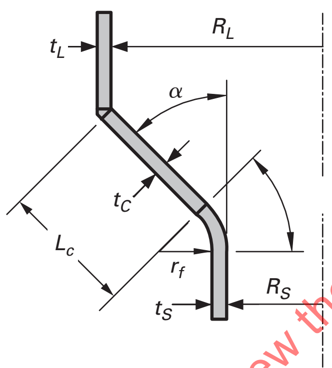
Figure 4.3.8
Conical Transition Details



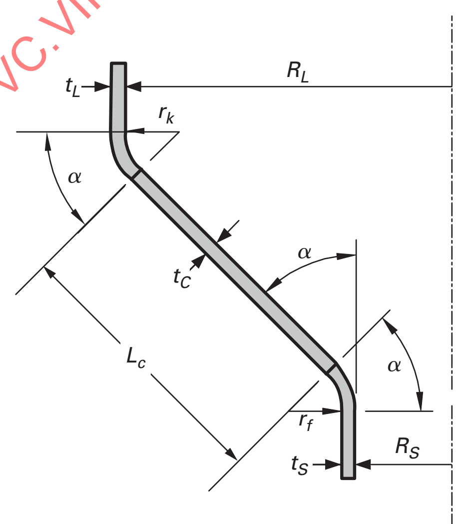
(a) Cone Without a Knuckle at Large End
Without a Flare at the Small End



(b) Cone With a Knuckle at Large End
Without a Flare at the Small End



(c) Cone Without a Knuckle at Large End
With a Flare at the Small End



(d) Cone With a Knuckle at Large End
With a Flare at the Small End

Figure 4.3.9
Reinforcement Requirements for Conical Transition Junction

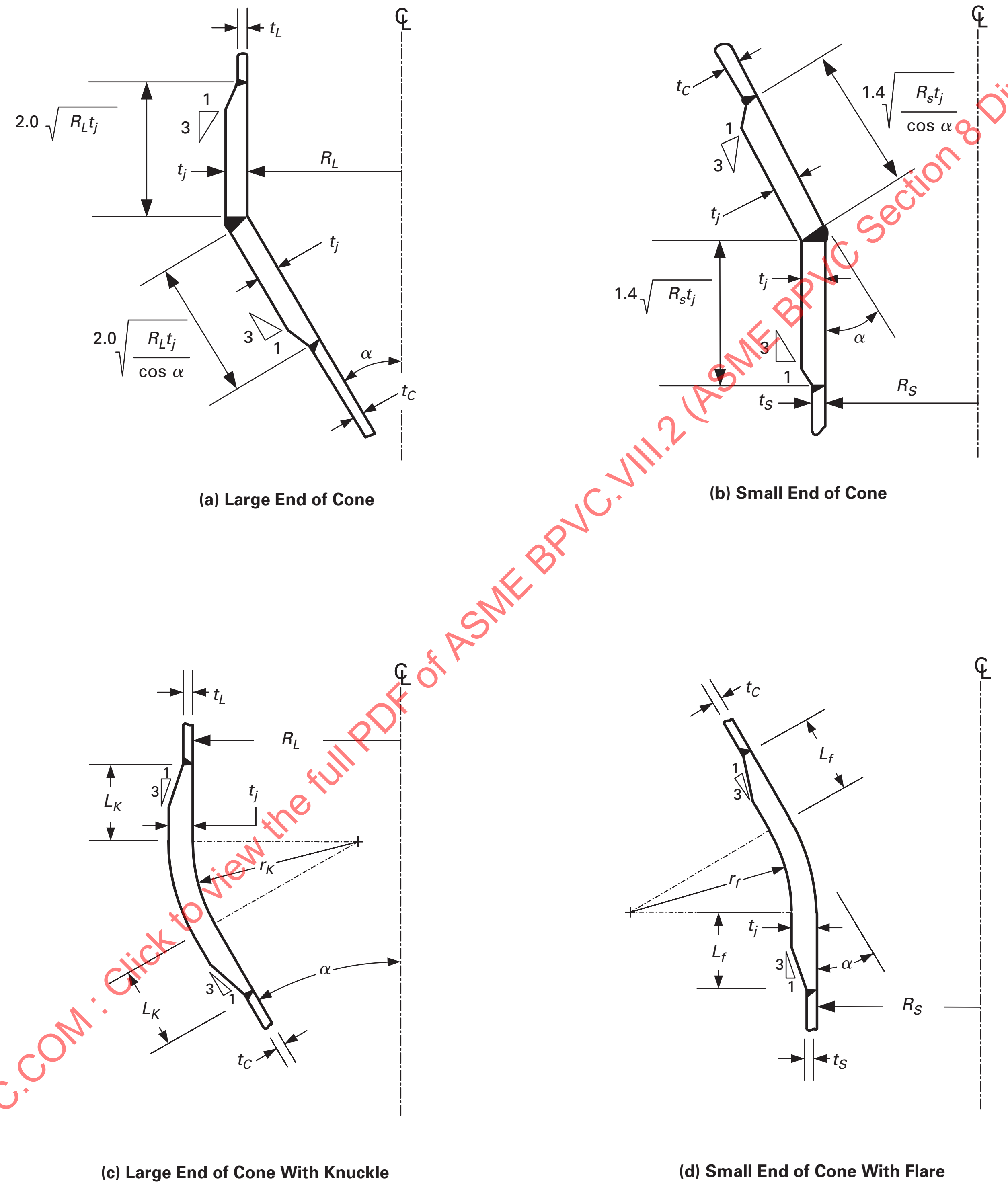
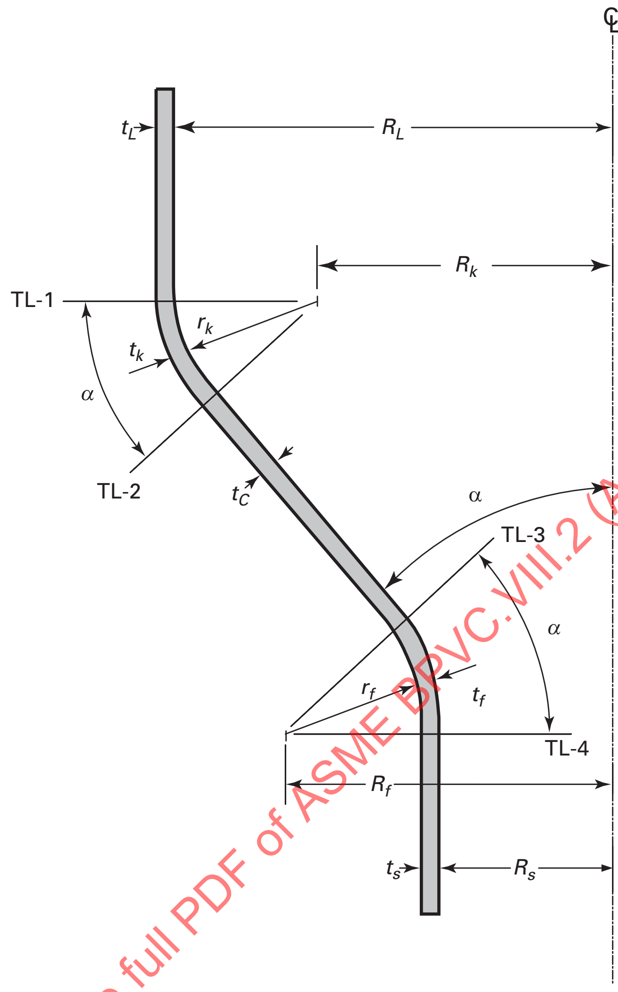


Figure 4.3.10
Parameters for Knuckle and Flare Design



4.4 DESIGN OF SHELLS UNDER EXTERNAL PRESSURE AND ALLOWABLE COMPRESSIVE STRESSES

4.4.1 SCOPE

4.4.1.1 4.4 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to external pressure. In this context, external pressure is defined as pressure acting on the convex side of the shell.

4.4.1.2 The effects of supplemental loads are not included in the design equations for shells and heads included in 4.4.5 through 4.4.9. The effects of supplemental loads that result in combined loadings shall be evaluated in a separate analysis performed in accordance with the methods in 4.4.12.

4.4.1.3 Rules are also provided for the design of cylindrical-to-conical shell transition junctions in 4.4.13 and 4.4.14. To facilitate the use of these rules, it is recommended that the shell wall thickness and stiffener configuration, as applicable, first be designed using the rules in 4.4.5 through 4.4.9. After an initial design is determined, this design should then be checked and modified as required using the rules of 4.4.13 and 4.4.14.

4.4.1.4 The equations in this paragraph are applicable for $D_o/t \leq 2\,000$. If $D_o/t > 2\,000$, then the design shall be in accordance with Part 5. In developing the equations in the paragraph, the shell section is assumed to be axisymmetric with uniform thickness for unstiffened cylinders and formed heads. Stiffened cylinders and cones are also assumed to be of uniform thickness between stiffeners. Where nozzles with reinforcing plates or locally thickened shell sections exist, the thinnest uniform thickness in the applicable unstiffened or stiffened shell section shall be used for the calculation of the allowable compressive stress.

4.4.1.5 Special consideration shall be given to ends of components (shell sections) or areas of load application where stress distribution may be in the inelastic range and localized stresses may exceed those predicted by linear theory.

4.4.1.6 When the localized stresses extend over a distance equal to one-half the length of the buckling mode (approximately $1.2\sqrt{D_o t}$), the localized stresses shall be considered as a uniform stress for the design of the shell section.

4.4.1.7 The buckling strength formulations presented in this paragraph are based upon linear structural stability theory which is modified by reduction factors which account for the effects of imperfections, boundary conditions, nonlinearity of material properties and residual stresses. The reduction factors are determined from approximate lower bound values of test data of shells with initial imperfections representative of the tolerance limits specified in this paragraph.

4.4.2 DESIGN FACTORS

The allowable stresses are determined by applying a design factor, FS, to the predicted buckling stresses. The required values of FS are 2.0 when the buckling stress is elastic and 1.667 when the predicted buckling stress equals the minimum specified yield strength at the design temperature. A linear variation shall be used between these limits. The equations for FS are given below where F_{ic} is the predicted inelastic buckling stress that is determined in 4.4.3. For combinations of design loads and earthquake loading or wind loading (see 4.1.5.3), the allowable stress for F_{bha} or F_{ba} in eqs. (4.4.85), (4.4.86), (4.4.87), (4.4.90), (4.4.91), and (4.4.92) may be increased by a factor of 1.2.

$$FS = 2.0 \qquad \text{for} \qquad F_{ic} \leq 0.55S_y \tag{4.4.1}$$

$$FS = 2.407 - 0.741\left(\frac{F_{ic}}{S_y}\right) \qquad \text{for} \qquad 0.55S_y < F_{ic} < S_y \tag{4.4.2}$$

$$FS = 1.667 \qquad \text{for} \qquad F_{ic} = S_y \tag{4.4.3}$$

4.4.3 MATERIAL PROPERTIES

4.4.3.1 The equations for the allowable compressive stress are based on the materials as given in Part 3 and are limited to the time-independent region. The maximum temperature limit permitted for these materials is defined in Table 4.4.1. If the component as designed is in the time-dependent region (i.e., creep is significant), the effects of time-dependent behavior shall be considered.

4.4.3.2 The equations for the allowable compressive stress consider both the predicted elastic buckling stress and predicted inelastic buckling stress. The predicted elastic buckling stress, F_{he}, F_{xe}, and F_{ve}, is determined based on the geometry of the component and the loading under consideration as provided in subsequent applicable paragraphs. The predicted inelastic buckling stress, F_{ic} , is determined using the following procedure:

Step 1. Calculate the predicted elastic buckling stress due to external pressure, F_{he}; axial compression, F_{xe}; and shear, F_{ve}, as applicable.

Step 2. Calculate the elastic buckling ratio factor, A_e, for the applicable loading condition.

$$A_e = \frac{F_{he}}{E}, \frac{F_{xe}}{E}, \frac{F_{ve}}{E} \tag{4.4.4}$$

Step 3. Solve for the predicted inelastic buckling stress, F_{ic}, through the determination of the material's tangent modulus, E_t, based on the stress–strain curve model at the design temperature per 3-D.5.1. The value of F_{ic} is solved for using an iterative procedure such that the following relationship is satisfied:

$$\frac{F_{ic}}{E_t} = A_e \tag{4.4.5}$$

When using the procedure of 3-D.5.1, the value of F_{ic} is substituted for σ_t.

An example of an iterative solution to determine F_{ic} is shown in Table 4.4.2.

4.4.4 SHELL TOLERANCES

4.4.4.1 Permissible Out-of-Roundness of Cylindrical and Conical Shells. The shell of a completed vessel subject to external pressure shall meet the following requirements at any cross section.

(a) The out-of-roundness requirements in 4.3.2.1 shall be satisfied.

(b) The maximum plus or minus deviation from a true circle, e, measured from a segmental circular template having the design inside or outside radius (depending on where the measurements are taken) and a chord length, L_{ec}, should not exceed the following value:

$$e = \min\left[e_c, 2t\right] \tag{4.4.6}$$

where

$$e_c = 0.0165t\left(\frac{L}{\sqrt{R_o t}} + 3.25\right)^{1.069} \quad \text{valid for } 0.1t \le e_c \le 0.0282R_o \tag{4.4.7}$$

$$L_{ec} = 2R_m \sin\left[\frac{\pi}{2n}\right] \tag{4.4.8}$$

$$n = \xi\left(\sqrt{\frac{R_m}{t}} \cdot \left(\frac{R_m}{L}\right)\right)^{\psi} \quad \text{valid for } 2 \le n \le 1.41\sqrt{\frac{R_m}{t}} \tag{4.4.9}$$

$$\xi = \min\left[2.28\left(\frac{R_m}{t}\right)^{0.054}, 2.80\right] \tag{4.4.10}$$

$$\psi = \min\left[0.38\left(\frac{R_m}{t}\right)^{0.044}, 0.485\right] \tag{4.4.11}$$

(c) The value of thickness, t, used in the above equations shall be determined as follows:

(1) For vessels with butt joints, t is the nominal plate thickness less the corrosion allowance.

(2) For vessels with lap joints, t is the nominal plate thickness and the permissible deviation is $e + t$.

(3) Where the shell at any cross section is made from plates of different thicknesses t is the nominal plate thickness of the thinnest plate less the corrosion allowance.

(d) For cones and conical sections, t shall be determined using (c) except that t shall be replaced by t_c.

(e) Measurements for out-of-tolerances shall be taken on the surface of the base metal and not on welds or other raised parts of the component.

(f) The dimensions of a completed vessel may be brought within the requirements of this paragraph by any process that will not impair the strength of the material.

(g) Sharp bends and flats spots shall not be permitted unless provision is made for them in the design or they satisfy the tolerances in 4.4.4.2 and 4.4.4.4.

(h) Vessels fabricated of pipe may have permissible variations in the outside diameter in accordance with those permitted under the specification covering its manufacture.

4.4.4.2 Cylindrical and Conical Shells Subject to Uniform Axial Compression and Axial Compression Due to a Bending Moment. The tolerance requirements in 4.3.2.1 shall be satisfied. In addition, the local inward deviation from a straight line, e, measured along a meridian over gauge length, L_x, shall not exceed the maximum permissible deviation, e_x, given below:

$$e_x = 0.002R_m \tag{4.4.12}$$

and,

$$L_x = \min\left[4\sqrt{R_m t}, L\right] \quad \text{for cylindrical shells} \tag{4.4.13}$$

$$L_x = \min\left[4\sqrt{\frac{R_m t_c}{\cos[\alpha]}}, \frac{L_c}{\cos[\alpha]}\right] \quad \text{for conical shells} \tag{4.4.14}$$

$$L_x = 25t \quad \text{across circumferential welds} \tag{4.4.15}$$

4.4.4.3 Cylindrical and Conical Shells Subject to External Pressure and Uniform Axial Compression and Axial Compression Due to a Bending Moment. All of the tolerance requirements in 4.4.4.1 and 4.4.4.2 shall be satisfied.

4.4.4.4 Spherical Shells and Formed Heads. The tolerance requirements in 4.3.2.2 shall be satisfied. In addition, the maximum local deviation from true circular form, e, for spherical shells and any spherical portion of a formed head shall not exceed the shell thickness. Measurements to determine the maximum local deviation shall be made with a template with a chord length, L_{es}, given by the following equation.

$$L_{es} = 3.72\sqrt{R_m t} \tag{4.4.16}$$

For spherical shells, the out-of-roundness requirements in 4.3.2.1 shall also be satisfied.

4.4.4.5 Shells that do not meet the tolerance requirements of this paragraph may be evaluated using 4.14.

4.4.5 CYLINDRICAL SHELL

4.4.5.1 Required Thickness. The required thickness of a cylindrical shell subjected to external pressure loading shall be determined using the following procedure.

Step 1. Assume an initial thickness, t, and unsupported length, L (see Figures 4.4.1 and 4.4.2).

Step 2. Calculate the predicted elastic buckling stress, F_{he}.

$$F_{he} = \frac{1.6 C_h E_y t}{D_o} \tag{4.4.17}$$

$$M_x = \frac{L}{\sqrt{R_o t}} \tag{4.4.18}$$

$$C_h = 0.55\left(\frac{t}{D_o}\right) \quad \text{for } M_x \geq 2\left(\frac{D_o}{t}\right)^{0.94} \tag{4.4.19}$$

$$C_h = 1.12 M_x^{-1.058} \quad \text{for } 13 < M_x < 2\left(\frac{D_o}{t}\right)^{0.94} \tag{4.4.20}$$

$$C_h = \frac{0.92}{M_x - 0.579} \quad \text{for } 1.5 < M_x \leq 13 \tag{4.4.21}$$

$$C_h = 1.0 \quad \text{for } M_x \leq 1.5 \tag{4.4.22}$$

Step 3. Calculate the predicted inelastic buckling stress, F_{ic}, per 4.4.3.

Step 4. Calculate the value of design factor, FS, per 4.4.2.

Step 5. Calculate the allowable external pressure, P_a.

$$P_a = 2F_{ha}\left(\frac{t}{D_o}\right) \tag{4.4.23}$$

where,

$$F_{ha} = \frac{F_{ic}}{FS} \tag{4.4.24}$$

Step 6. If the allowable external pressure, P_a, is less than the design external pressure, increase the shell thickness or reduce the unsupported length of the shell (i.e., by the addition of a stiffening rings) and go to Step 2. Repeat this process until the allowable external pressure is equal to or greater than the design external pressure.

4.4.5.2 Stiffening Ring Size. The following equations shall be used to determine the size of a stiffening ring.

(a) Stiffening Ring Configuration - A combination of large and small stiffening rings may be used along the length of a shell. If a single size stiffener is used, then it shall be sized as a small stiffener. Alternatively, a combination of large and small stiffeners can be used to reduce the size of the intermittent small stiffening rings. The large stiffening rings may be sized to function as end stiffeners or bulkheads with small stiffeners spaced as required between end rings based on the shell thickness selected and loading combinations considered in the design.

(b) Small Stiffening Ring - The required moment of inertia of the effective stiffening ring (i.e., actual stiffening ring plus the effective length of shell, see Figure 4.4.3) shall satisfy eq. (4.4.25). The parameter F_{he} shall be evaluated using the equations in 4.4.5.1 with $M_x = L_s / \sqrt{R_o t}$.

$$I_s^C \geq \frac{1.5 F_{he} L_s R_c^2 t}{E_y \left(n^2 - 1\right)} \tag{4.4.25}$$

where,

$$n = \sqrt{\frac{2D_o^{1.5}}{3L_B t^{0.5}}} \quad \text{where } n \text{ is an integer; for } n < 2 \text{ use } n = 2, \text{ and for } n > 10 \text{ use } n = 10 \tag{4.4.26}$$

The actual moment of inertia of the composite section comprised of the small stiffening ring and effective length of the shell about the centroidal axis shall be calculated using eq. (4.4.27):

$$I_s^C = I_s + A_s Z_s^2 \left(\frac{L_e t}{A_s + L_e t} \right) + \frac{L_e t^3}{12} \tag{4.4.27}$$

where,

$$L_e = 1.1 \sqrt{D_o t} \tag{4.4.28}$$

(c) Large Stiffening Ring or Bulkhead - The required moment of inertia of the effective stiffening ring (i.e., actual stiffening ring plus the effective length of shell) shall satisfy eq. (4.4.29). The parameter F_{hef} is the average value of the hoop buckling stress, F_{he}, over length L_F evaluated using the equations in 4.4.5.1 with $M_x = L_F / \sqrt{R_o t}$.

$$I_L^C \geq \frac{F_{hef} L_F R_c^2 t}{2E_y} \tag{4.4.29}$$

The actual moment of inertia of the composite section comprised of the large stiffening ring and effective length of the shell about the centroidal axis shall be calculated using eq. (4.4.30):

$$I_L^C = I_L + A_L Z_L^2 \left(\frac{L_e t}{A_L + L_e t} \right) + \frac{L_e t^3}{12} \tag{4.4.30}$$

where,

$$L_e = 1.1\sqrt{D_o t}\left(\frac{A_s + L_s t}{A_L + L_s t}\right) \tag{4.4.31}$$

(d) Local Stiffener Geometry Requirements for all Loading Conditions - The following equations shall be met to assure the stability of a stiffening ring.

(1) Flat bar stiffener, flange of a tee section and the outstanding leg of an angle stiffener (see Figure 4.4.3)

$$\frac{h_1}{t_1} \le 0.375\left(\frac{E_y}{S_y}\right)^{0.5} \tag{4.4.32}$$

(2) Web of a tee stiffener or leg of an angle stiffener attached to the shell (see Figure 4.4.3).

$$\frac{h_2}{t_2} \le \left(\frac{E_y}{S_y}\right)^{0.5} \tag{4.4.33}$$

(e) Stiffener Size to Increase Allowable Longitudinal Compressive Stress - ring stiffeners can be used to increase the allowable longitudinal compressive stress for cylindrical or conical shells subject to uniform axial compression and axial compression due to bending. The required size of the stiffener shall satisfy the following equations. In addition, the spacing of the stiffeners must result in a value of $M_s \le 15$ where M_s is given by eq. (4.4.37).

$$A_s \ge \left(\frac{0.334}{M_s^{0.6}} - 0.063\right)L_s t \tag{4.4.34}$$

$$A_s \ge 0.06 L_s t \tag{4.4.35}$$

$$I_s^C \ge \frac{5.33 L_s t^3}{M_s^{1.8}} \tag{4.4.36}$$

$$M_s = \frac{L_s}{\sqrt{R_o t}} \tag{4.4.37}$$

(f) Stiffener Size For Shear - The required size of the stiffener shall satisfy the following equation where C_v is evaluated using eqs. (4.4.62) through (4.4.65) with $M_x = M_s$, M_s is given by eq. (4.4.37).

$$I_s^C \ge 0.184 C_v M_s^{0.8} t^3 L_s \tag{4.4.38}$$

(g) Arrangement of Stiffening Rings

(1) As shown in Figure 4.4.4, any joints between the ends or sections of such rings, at locations (A) and (B), and any connection between adjacent portions of a stiffening ring lying inside or outside the shell, at location (C), shall be made so that the required moment of inertia of the combined ring-shell section is maintained. For a section with a strut at location (D), the required moment of inertia shall be supplied by the strut alone.

(2) As shown in Figure 4.4.4, stiffening rings placed on the inside of a vessel may be arranged as shown at locations (E) and (F), provided that the required moment of inertia of the ring at location (E) or of the combined ring-shell section at location (F) is maintained within the sections indicated. Where the gap at locations (A) or (E) does not exceed eight times the thickness of the shell plate, the combined moment of inertia of the shell and stiffener may be used.

(3) Stiffening rings shall extend completely around the vessel except as provided below. Any gap in that portion of a stiffening ring supporting the shell, as shown in Figure 4.4.4 at locations (D) and (E), shall not exceed the length of arc given in Figure 4.4.5 unless additional reinforcement is provided as shown at location (C), or unless all of the following conditions are met:

(-a) only one unsupported shell arc is permitted per ring

(-b) the length of unsupported shell arc does not exceed 90 deg

(-c) the unsupported shell arcs in adjacent stiffening rings are staggered 180 deg

(-d) the dimension L is taken as the larger of the distance between alternate stiffening rings or the distance from the head-bend line to the second stiffening ring plus one-third of the head depth

(4) When internal plane structures perpendicular to the longitudinal axis of the cylinder, such as bubble trays or baffle plates, are used in a vessel, they may also be considered to act as stiffening rings, provided they are designed to function as such.

(5) Any internal stays or supports used shall bear against the shell of the vessel through the medium of a substantially continuous ring.

(h) Attachment of Stiffening Rings - Stiffening rings shall be attached to either the outside or the inside of the vessel by continuous welding, or if the component is not in cyclic service (i.e., a fatigue analysis is not required in accordance with 4.1.1.4) intermittent welding. Where gaps occur in the stiffening ring, the attachment weld shall conform to the details in 4.2.

4.4.5.3 Combined Loadings. Cylindrical shells subject to external pressure and other loadings shall satisfy the requirements of 4.4.12.

4.4.6 CONICAL SHELL

4.4.6.1 Required Thickness. The required thickness of a conical shell subjected to external pressure loading shall be determined using the equations for a cylinder by making the following substitutions:

(a) The value of t_c is substituted for t in the equations in 4.4.5.

(b) For offset cones, the cone angle, α, shall satisfy the requirements of 4.3.4.

(c) The value of $0.5(D_L + D_S)/\cos[\alpha]$ is substituted for D_o in the equations in 4.4.5.

(d) The value of $L_{ce}/\cos[\alpha]$ is substituted for L in the equations in 4.4.5 where L_{ce} is determined as shown below.

(1) For sketches (a) and (e) in Figure 4.4.7

$$L_{ce} = L_c \tag{4.4.39}$$

(2) For sketch (b) in Figure 4.4.7

$$L_{ce} = r_k \sin[\alpha] + L_c \tag{4.4.40}$$

(3) For sketch (c) in Figure 4.4.7

$$L_{ce} = r_f \sin[\alpha] + L_c \tag{4.4.41}$$

(4) For sketch (d) in Figure 4.4.7

$$L_{ce} = \left(r_k + r_f\right)\sin[\alpha] + L_c \tag{4.4.42}$$

(e) Note that the half-apex angle of a conical transition can be computed knowing the shell geometry with the following equations. These equations were developed with the assumption that the conical transition contains a cone section, knuckle, or flare. If the transition does not contain a knuckle or flare, the radii of these components should be set to zero when computing the half-apex angle (see Figure 4.4.7).

(1) If $(R_L - r_k) \geq (R_S + r_f)$:

$$\alpha = \beta + \phi \tag{4.4.43}$$

$$\beta = \arctan\left[\frac{\left(R_L - r_k\right) - \left(R_S + r_f\right)}{L_{ce}}\right] \tag{4.4.44}$$

(2) If $(R_L - r_k) < (R_S + r_f)$:

$$\alpha = \phi - \beta \tag{4.4.45}$$

$$\beta = \arctan\left[\frac{\left(R_S + r_f\right) - \left(R_L - r_k\right)}{L_{ce}}\right] \tag{4.4.46}$$

(3) In both cases shown above, the angle ϕ is given by the following equation.

$$\phi = \arcsin\frac{\left(r_f + r_k\right)\cos\beta}{L_{ce}} \tag{4.4.47}$$

4.4.6.2 Small Stiffening Rings. Intermediate circumferential stiffening rings within the conical transition shall be sized using eq. (4.4.25) where L_s is determined from 4.4.6.1(d), and t_c is the cone thickness at the ring location.

4.4.6.3 Combined Loadings. Conical shells subject to external pressure and other loadings shall satisfy the requirements of 4.4.12.

4.4.7 SPHERICAL SHELL AND HEMISPHERICAL HEAD

4.4.7.1 Required Thickness. The required thickness of a spherical shell or hemispherical head subjected to external pressure loading shall be determined using the following procedure.

Step 1. Assume an initial thickness, t for the spherical shell.

Step 2. Calculate the predicted elastic buckling stress, F_{he}.

$$F_{he} = 0.075E_y\left(\frac{t}{R_o}\right) \tag{4.4.48}$$

Step 3. Calculate the predicted inelastic buckling stress, F_{ic}, per 4.4.3.

Step 4. Calculate the value of design margin, FS, per 4.4.2.

Step 5. Calculate the allowable external pressure, P_a

$$P_a = 2F_{ha}\left(\frac{t}{R_o}\right) \tag{4.4.49}$$

where,

$$F_{ha} = \frac{F_{ic}}{FS} \tag{4.4.50}$$

Step 6. If the allowable external pressure, P_a, is less than the design external pressure, increase the shell thickness and go to Step 2. Repeat this process until the allowable external pressure is equal to or greater than the design external pressure.

4.4.7.2 Combined Loadings. Spherical shells and hemispherical heads subject to external pressure and other loadings shall satisfy the requirements of 4.4.12.

4.4.8 TORISPHERICAL HEAD

4.4.8.1 Required Thickness. The required thickness of a torispherical head subjected to external pressure loading shall be determined using the equations for a spherical shell in 4.4.7 by substituting the outside crown radius for R_o.

4.4.8.2 Restrictions on Torispherical Head Geometry. The restriction of 4.3.6 shall apply.

4.4.8.3 Torispherical Heads With Different Dome and Knuckle Thicknesses. Heads with this configuration shall be designed in accordance with Part 5.

4.4.8.4 Combined Loadings. Torispherical heads subject to external pressure and other loadings shall satisfy the requirements of 4.4.12.

4.4.9 ELLIPSOIDAL HEAD

4.4.9.1 Required Thickness. The required thickness of an elliptical head subjected to external pressure loading shall be determined using the equations for a spherical shell in 4.4.7 by substituting K_o D_o for R_o where K_o is given by the following equation:

$$K_o = 0.25346 + 0.13995\left(\frac{D_o}{2h_o}\right) + 0.12238\left(\frac{D_o}{2h_o}\right)^2 - 0.015297\left(\frac{D_o}{2h_o}\right)^3 \tag{4.4.51}$$

4.4.9.2 Combined Loadings. Ellipsoidal heads subject to external pressure and other loadings shall satisfy the requirements of 4.4.12.

4.4.10 LOCAL THIN AREAS

Rules for the evaluation of Local Thin Areas are covered in 4.14.

4.4.11 DRILLED HOLES NOT PENETRATING THROUGH THE VESSEL WALL

Design requirements for partially drilled holes that do not penetrate completely through the vessel wall are covered in 4.3.9.

4.4.12 COMBINED LOADINGS AND ALLOWABLE COMPRESSIVE STRESSES

4.4.12.1 The rules in 4.4.2 through 4.4.11 are applicable for external pressure loading. The rules in this paragraph provide allowable compressive stresses that shall be used for the design of shells subjected to supplemental loads that result in combined loadings. The allowable stresses of this paragraph shall also be used as the acceptance criteria for shells subjected to compressive stress evaluated using Part 5.

4.4.12.2 Cylindrical Shells. The allowable compressive stresses for cylindrical shells shall be computed using the following rules that are based on loading conditions. The loading conditions are underlined for clarity in the following paragraphs. Common parameters used in each of the loading conditions are given in (k).

(a) External Pressure Acting Alone. The allowable hoop compressive membrane stress of a cylinder subject to external pressure acting alone, F_{ha}, is computed using the equations in 4.4.5.1.

(b) Axial Compressive Stress Acting Alone. The allowable axial compressive membrane stress of a cylinder subject to an axial compressive load acting alone, F_{xa}, is computed using the following equations.

(1) For $\lambda_c \le 0.15$ (Local Buckling):

Step 1. Calculate the predicted elastic buckling stress, F_{xe}.

$$F_{xe} = \frac{C_x E_y t}{D_o} \tag{4.4.52}$$

$$C_x = \min\left[\frac{409\bar{c}}{\left(389 + \frac{D_o}{t}\right)}, 0.9\right] \quad \text{for } \frac{D_o}{t} < 1247 \tag{4.4.53}$$

$$C_x = 0.25\bar{c} \quad \text{for } 1247 \le \frac{D_o}{t} \le 2000 \tag{4.4.54}$$

$$\bar{c} = 2.64 \quad \text{for } M_x \le 1.5 \tag{4.4.55}$$

$$\bar{c} = \frac{3.13}{M_x^{0.42}} \quad \text{for } 1.5 < M_x < 15 \tag{4.4.56}$$

$$\bar{c} = 1.0 \quad \text{for } M_x \ge 15 \tag{4.4.57}$$

Step 2. Calculate the predicted inelastic buckling stress, F_{ic}, per 4.4.3.

Step 3. Calculate the factor of safety, FS, per 4.4.2.

Step 4. Calculate the allowable axial compressive stress, F_{xa}, as follows:

$$F_{xa} = F_{ic}/FS \tag{4.4.58}$$

(2) For $\lambda_c > 0.15$ and $K_u L_u / r_g < 200$ (Column Buckling):

$$F_{ca} = F_{xa}\left[1 - 0.74\left(\lambda_c - 0.15\right)\right]^{0.3} \quad \text{for } 0.15 < \lambda_c < 1.2 \tag{4.4.59}$$

$$F_{ca} = \frac{0.88 F_{xa}}{\lambda_c^2} \quad \text{for } \lambda_c \ge 1.2 \tag{4.4.60}$$

(c) Compressive Bending Stress. The allowable axial compressive membrane stress of a cylindrical shell subject to a bending moment acting across the full circular cross section F_{ba}, shall be determined using the procedure in (b).

(d) Shear Stress. The allowable shear stress of a cylindrical shell, F_{va}, is computed using the following equations.

Step 1. Calculate the predicted elastic buckling stress, F_{ve}.

$$F_{ve} = \alpha_v C_v E_y \left(\frac{t}{D_o}\right) \tag{4.4.61}$$

$$C_v = 4.454 \qquad \text{for } M_x \le 1.5 \tag{4.4.62}$$

$$C_v = \left(\frac{9.64}{M_x^2}\right)\left(1 + 0.0239 M_x^3\right)^{0.5} \qquad \text{for } 1.5 < M_x < 26 \tag{4.4.63}$$

$$C_v = \frac{1.492}{M_x^{0.5}} \qquad \text{for } 26 \le M_x < 4.347\left(\frac{D_o}{t}\right) \tag{4.4.64}$$

$$C_v = 0.716\left(\frac{t}{D_o}\right)^{0.5} \qquad \text{for } M_x \ge 4.347\left(\frac{D_o}{t}\right) \tag{4.4.65}$$

$$\alpha_v = 0.8 \qquad \text{for } \frac{D_o}{t} \le 500 \tag{4.4.66}$$

$$\alpha_v = 1.389 - 0.218\log_{10}\left(\frac{D_o}{t}\right) \qquad \text{for } \frac{D_o}{t} > 500 \tag{4.4.67}$$

Step 2. Calculate the predicted inelastic buckling stress, F_{ic}, per 4.4.3.

Step 3. Calculate the factor of safety, FS, per 4.4.2.

Step 4. Calculate the allowable axial compressive stress, F_{va}, as follows:

$$F_{va} = F_{ic}/FS \tag{4.4.68}$$

(e) Axial Compressive Stress and Hoop Compression. The allowable compressive stress for the combination of uniform axial compression and hoop compression, F_{xha}, is computed using the following equations:

(1) For $\lambda_c \le 0.15$, F_{xha} is computed using the following equation with F_{ha} and F_{xa} evaluated using the equations in (a) and (b)(1), respectively.

$$F_{xha} = \left[\left(\frac{1}{F_{xa}^2}\right) - \left(\frac{C_1}{C_2 F_{xa} F_{ha}}\right) + \left(\frac{1}{C_2^2 F_{ha}^2}\right)\right]^{-0.5} \tag{4.4.69}$$

$$C_1 = \frac{F_{xa} \cdot FS + F_{ha} \cdot FS}{S_y} - 1.0 \tag{4.4.70}$$

$$C_2 = \frac{f_x}{f_h} \tag{4.4.71}$$

$$f_x = f_a + f_q \qquad \text{for } f_x \le F_{xha} \tag{4.4.72}$$

The parameters f_a and f_q are defined in (k).

The values of FS are given in 4.4.2. The values of FS are to be determined independently for axial and hoop directions.

(2) For $0.15 < \lambda_c < 1.2$, F_{xha} is computed from the following equation with $F_{ah1} = F_{xha}$ evaluated using the equations in (1) and F_{ah2} using the following procedure. The value of F_{ca} used in the calculation of F_{ah2} is evaluated using the equations in (b)(2) with $F_{xa} = F_{xha}$ as determined in (1). As noted, the load on the end of a cylinder due to external

pressure does not contribute to column buckling and therefore F_{ah1} is compared with f_a rather than f_x. The stress due to the pressure load does, however, lower the effective yield stress and the quantity in $(1 - f_q/S_y)$ accounts for this reduction.

$$F_{xha} = \min\left[F_{ah1}, F_{ah2}\right] \tag{4.4.73}$$

$$F_{ah2} = F_{ca}\left(1 - \frac{f_q}{S_y}\right) \tag{4.4.74}$$

(3) For $\lambda_c \leq 0.15$, the allowable hoop compressive membrane stress, F_{hxa}, is given by the following equation:

$$F_{hxa} = \frac{F_{xha}}{C_2} \tag{4.4.75}$$

(4) For $\lambda_c \geq 1.2$, the rules in (e) do not apply.

(f) Compressive Bending Stress and Hoop Compression. The allowable compressive stress for the combination of axial compression due to a bending moment and hoop compression, F_{bha}, is computed using the following equations.

(1) An iterative solution procedure is utilized to solve these equations for C_3 with F_{ha} and F_{ba} evaluated using the equations in (a) and (c), respectively.

$$F_{bha} = C_3 C_4 F_{ba} \tag{4.4.76}$$

$$C_4 = \left(\frac{f_b}{f_h}\right)\left(\frac{F_{ha}}{F_{ba}}\right) \tag{4.4.77}$$

$$C_3^2\left(C_4^2 + 0.6C_4\right) + C_3^{2n} - 1 = 0 \tag{4.4.78}$$

$$n = 5 - \frac{4F_{ha} \cdot FS}{S_y} \tag{4.4.79}$$

(2) The allowable hoop compressive membrane stress, F_{hba}, is given by the following equation:

$$F_{hba} = F_{bha}\left(\frac{f_h}{f_b}\right) \tag{4.4.80}$$

(g) Shear Stress and Hoop Compression. The allowable compressive stress for the combination of shear, F_{vha}, and hoop compression is computed using the following equations.

(1) The allowable shear stress is given by the following equation with F_{ha} and F_{va} evaluated using the equations in (a) and (d), respectively.

$$F_{vha} = \left[\left(\frac{F_{va}^2}{2C_5 F_{ha}}\right)^2 + F_{va}^2\right]^{0.5} - \frac{F_{va}^2}{2C_5 F_{ha}} \tag{4.4.81}$$

$$C_5 = \frac{f_v}{f_h} \tag{4.4.82}$$

(2) The allowable hoop compressive membrane stress, F_{hva}, is given by the following equation:

$$F_{hva} = \frac{F_{vha}}{C_5} \tag{4.4.83}$$

(h) Axial Compressive Stress, Compressive Bending Stress, Shear Stress, and Hoop Compression. The allowable compressive stress for the combination of uniform axial compression, axial compression due to a bending moment, and shear in the presence of hoop compression is computed using the following interaction equations.

(1) The shear coefficient is determined using the following equation with F_{va} from (d).

$$K_s = 1.0 - \left(\frac{f_v}{F_{va}}\right)^2 \tag{4.4.84}$$

(2) For $\lambda_c \leq 0.15$, the acceptability of a member subject to compressive axial and bending stresses, f_a and f_b, respectively, is determined using the following interaction equation with F_{xha} and F_{bha} evaluated using the equations in (e)(1) and (f)(1), respectively.

$$\left(\frac{f_a}{K_s F_{xha}}\right)^{1.7} + \left(\frac{f_b}{K_s F_{bha}}\right) \leq 1.0 \tag{4.4.85}$$

(3) For $0.15 < \lambda_c \leq 1.2$, the acceptability of a member subject to compressive axial and bending stresses, f_a and f_b, respectively, is determined using the following interaction equation with F_{xha} and f_{bha} evaluated using the equations in (e)(2) and (f)(1), respectively.

$$\left(\frac{f_a}{K_s F_{xha}}\right) + \left(\frac{8}{9}\frac{\Delta f_b}{K_s F_{bha}}\right) \leq 1.0 \qquad \text{for } \frac{f_a}{K_s F_{xha}} \geq 0.2 \tag{4.4.86}$$

$$\left(\frac{f_a}{2K_s F_{xha}}\right) + \left(\frac{\Delta f_b}{K_s F_{bha}}\right) \leq 1.0 \qquad \text{for } \frac{f_a}{K_s F_{xha}} < 0.2 \tag{4.4.87}$$

$$\Delta = \frac{C_m}{1 - \left(\frac{f_a \cdot FS}{F_e}\right)} \tag{4.4.88}$$

$$F_e = \frac{\pi^2 E_y}{\left(\frac{K_u L_u}{r_g}\right)^2} \tag{4.4.89}$$

(i) Axial Compressive Stress, Compressive Bending Stress, and Shear. The allowable compressive stress for the combination of uniform axial compression, axial compression due to a bending moment, and shear in the absence of hoop compression is computed using the following interaction equations:

(1) The shear coefficient is determined using the equation in (h)(1) with F_{va} from (d).

(2) For $\lambda_c \leq 0.15$, the acceptability of a member subject to compressive axial and bending stresses f_a and f_b, respectively, is determined using the following interaction equation with, F_{xa} and F_{ba} evaluated using the equations in (b)(1) and (c), respectively.

$$\left(\frac{f_a}{K_s F_{xa}}\right)^{1.7} + \left(\frac{f_b}{K_s F_{ba}}\right) \leq 1.0 \tag{4.4.90}$$

(3) For $0.15 < \lambda_c \leq 1.2$, the acceptability of a member subject to compressive axial and bending stresses, f_a and f_b, respectively, is determined using the following interaction equation with, F_{ca} and F_{ba} evaluated using the equations in (b)(2) and (c) respectively. The coefficient Δ is evaluated using the equations in (h)(3).

$$\left(\frac{f_a}{K_s F_{ca}}\right) + \left(\frac{8}{9}\frac{\Delta f_b}{K_s F_{ba}}\right) \leq 1.0 \qquad \text{for } \frac{f_a}{K_s F_{ca}} \geq 0.2 \tag{4.4.91}$$

$$\left(\frac{f_a}{2K_s F_{ca}}\right) + \left(\frac{\Delta f_b}{K_s F_{ba}}\right) \leq 1.0 \qquad \text{for } \frac{f_a}{K_s F_{ca}} < 0.2 \tag{4.4.92}$$

(j) The maximum deviation, e may exceed the value of e_x given in 4.4.4.2 if the maximum axial stress is less than F_{xa} for shells designed for axial compression only, or less than F_{xha} for shells designed for combinations of axial compression and external pressure. The change in buckling stress, F'_{xe}, is given by eq. (4.4.93). The reduced allowable buckling stress, $F_{xa(\text{reduced})}$, is determined using eq. (4.4.94) where e is the new maximum deviation, F_{xa} is determined using eq. (4.4.58), and FS_{xa} is the value of the stress reduction factor used to determine F_{xa}.

$$F'_{xe} = \left(0.944 - \left|0.286\log\left[\frac{0.0005e}{e_x}\right]\right|\right)\left(\frac{E_y t}{R}\right) \tag{4.4.93}$$

$$F_{xa(\text{reduced})} = \frac{F_{xa} \cdot FS_{xa} - F'_{xe}}{FS_{xa}} \tag{4.4.94}$$

The quantity $0.286\log[0.0005(e/e_x)]$ in eq. (4.4.93) is an absolute number (i.e., the log of a very small number is negative). For example, if $e = e_x$, then the change in the buckling stress computed using eq. (4.4.93) is $F'_{xe} = 0.086E_y(t/R)$.

(k) Section Properties, Stresses, Buckling Parameters. Equations for section properties, nominal shell stresses, and buckling parameters that are used in (a) through (i) are provided below.

$$A = \frac{\pi\left(D_o^2 - D_i^2\right)}{4} \tag{4.4.95}$$

$$S = \frac{\pi\left(D_o^4 - D_i^4\right)}{32D_o} \tag{4.4.96}$$

$$f_h = \frac{PD_o}{2t} \tag{4.4.97}$$

$$f_b = \frac{M}{S} \tag{4.4.98}$$

$$f_a = \frac{F}{A} \tag{4.4.99}$$

$$f_q = \frac{P\pi D_i^2}{4A} \tag{4.4.100}$$

$$f_v = \frac{V\sin[\phi]}{A} \tag{4.4.101}$$

$$r_g = 0.25\sqrt{D_o^2 + D_i^2} \tag{4.4.102}$$

$$M_x = \frac{L}{\sqrt{R_o t}} \tag{4.4.103}$$

$$\lambda_c = \frac{K_u L_u}{\pi r_g}\left(\frac{F_{xa} \cdot FS}{E_y}\right)^{0.5} \tag{4.4.104}$$

(21) **4.4.12.3 Conical Shells.** Unstiffened conical transitions or cone sections between stiffening rings of conical shells with a half-apex angle, α, less than 60 deg shall be evaluated as an equivalent cylinder using the equations in 4.4.12.2 with the substitutions shown below. Both the shell tolerances and stress criteria in this paragraph shall be satisfied at all cross-sections along the length of the cone.

(a) The value of t_c is substituted for t to determine the allowable compressive stress.

(b) The value of $D/\cos\alpha$ is substituted for D_o to determine the allowable compressive stress where D is the outside diameter of the cone at the point under consideration.

(c) The value of $L_c/\cos\alpha$, is substituted for L where L_c is the distance along the cone axis between stiffening rings.

4.4.12.4 Spherical Shells and Formed Heads. The allowable compressive stresses are based on the ratio of the biaxial stress state.

(a) Equal Biaxial Stresses. The allowable compressive stress for a spherical shell subject to a uniform external pressure, F_{ha}, is given by the equations in 4.4.7.

(b) Unequal Biaxial Stresses, Both Stresses Are Compressive. The allowable compressive stress for a spherical shell subject to unequal biaxial stresses, σ_1 and σ_2, where both σ_1 and σ_2 are compressive stresses resulting from the applied loads is given by the equations shown below. In these equations, F_{ha} is determined using 4.4.7. F_{1a} is the allowable compressive stress in the direction of σ_1 and is the F_{2a} allowable compressive stress in the direction of σ_2.

$$F_{1a} = \frac{F_{ha}}{0.6 + 0.4k} \tag{4.4.105}$$

$$F_{2a} = kF_{1a} \tag{4.4.106}$$

$$k = \frac{\sigma_2}{\sigma_1} \qquad \text{where } |\sigma_1| > |\sigma_2| \tag{4.4.107}$$

(c) Unequal Biaxial Stresses, One Stress Is Compressive and the Other Is Tensile. The allowable compressive stress for a spherical shell subject to unequal biaxial stresses, σ_1 and σ_2, where σ_1 is compressive and σ_2 is tensile, resulting from the applied loads is given by the equations shown below. In these equations, F_{1a} is the allowable compressive stress in the direction of σ_1, and is the value of F_{ha} determined using 4.4.7 with F_{he} computed using the following equations.

$$F_{he} = \frac{\left(C_o + C_p\right)E_y t}{R_o} \tag{4.4.108}$$

$$C_o = \frac{102.2}{195 + \frac{R_o}{t}} \qquad \text{for } \frac{R_o}{t} < 622 \tag{4.4.109}$$

$$C_o = 0.125 \qquad \text{for } 622 \le \frac{R_o}{t} \le 1\,000 \tag{4.4.110}$$

$$C_p = \frac{1.06}{3.24 + \left(\frac{E_y t}{\sigma_2 R_o}\right)} \tag{4.4.111}$$

4.4.13 CYLINDRICAL-TO-CONICAL SHELL TRANSITION JUNCTIONS WITHOUT A KNUCKLE

4.4.13.1 The design rules in 4.3.11 shall be satisfied. In these calculations, a negative value of pressure shall be used in all applicable equations.

4.4.13.2 If a stiffening ring is provided at the cone-to-cylinder junction, the design shall be made in accordance with Part 5.

4.4.14 CYLINDRICAL-TO-CONICAL SHELL TRANSITION JUNCTIONS WITH A KNUCKLE

4.4.14.1 The design rules in 4.3.12 shall be satisfied. In these calculations, a negative value of pressure shall be used in all applicable equations.

4.4.14.2 If a stiffening ring is provided within the knuckle, the design shall be made in accordance with Part 5.

4.4.15 NOMENCLATURE

A = cross-sectional area of cylinder.
A_S = cross-sectional area of a small ring stiffener.
A_L = cross-sectional area of a large ring stiffener that acts as a bulkhead.
α = one-half of the conical shell apex angle (degrees).
C_m = coefficient whose value is established as follows:

= 0.85 for compression members in frames subject to joint translation (sideway).
= $0.6 - 0.4(M_1/M_2)$for rotationally restrained members in frames braced against joint translation and not subject to transverse loading between their supports in the plane of bending; in this equation, is the ratio of the smaller to large bending moment at the ends of the portion of the member that is unbraced in the plane of bending under consideration M_1/M_2 is positive when the member is bent in reverse curvature and negative when the member is bent in single curvature).
= 0.85 for compression members in frames braced against joint translation and subject to transverse loading between support points, the member ends are restrained against rotation in the plane of bending.
= 1.0 for compression members in frames braced against joint translation and subject to transverse loading between support points, the member ends are unrestrained against rotation in the plane of bending.
= 1.0 for an unbraced skirt supported vessel.

c = distance from the neutral axis to the point under consideration.
D_c = diameter to the centroid of the composite ring section for an external ring; or the inside diameter for and internal ring (see Figure 4.4.6)
D_i = Inside diameter of cylinder (including the effects of corrosion.
D_o = outside diameter of cylinder.
D_e = outside diameter of an assumed equivalent cylinder for the design of cones or conical sections.
D_S = outside diameter of at the small end of the cone or conical section between lines of support.
D_L = outside diameter of at the large end of the cone or conical section between lines of support.
E_y = modulus of elasticity of material at the design temperature from Part 3.
E_t = tangent modulus of elasticity of material at the design temperature from Part 3.
F = applied net-section axial compression load.
f_a = axial compressive membrane stress resulting from applied axial load.
f_b = axial compressive membrane stress resulting from applied bending moment.
f_h = hoop compressive stress in the cylinder from external pressure.
f_q = axial compressive membrane stress resulting from the pressure load, Q_p, on the end of the cylinder.
f_v = shear stress from applied loads.
FS = design factor.
F_{ba} = allowable compressive membrane stress of a cylinder subject to a net-section bending moment in the absence of other loads.
F_{ca} = allowable compressive membrane stress of a cylinder due to an axial compressive load with $\lambda_c > 0.15$.
F_{bha} = allowable axial compressive membrane stress of a cylinder subject to bending in the presence of hoop compression.
F_{hba} = allowable hoop compressive membrane stress of a cylinder in the presence of longitudinal compression due to net-section bending moment.
F_{he} = elastic hoop compressive membrane failure stress of a cylinder or formed head subject to external pressure only.
F_{ha} = allowable hoop compressive membrane stress of a cylinder or formed head subject to external pressure only.
F_{hef} = average value of the hoop buckling stress, F_{he}, averaged over the length L_F where F_{he} is determined from eq. (4.4.17).
F_{hva} = allowable hoop compressive membrane stress of a cylinder in the presence of shear stress.
F_{hxa} = allowable hoop compressive membrane stress of a cylinder in the presence of axial compression.
F_{ic} = predicted inelastic buckling stress.
F_{ta} = allowable tensile stress from 3.8.
F_{va} = allowable shear stress of a cylinder subject only to shear loads.
F_{ve} = elastic shear buckling stress of a cylinder subject only to shear loads.

F_{vha} = allowable shear stress of a cylinder subject to shear stress in the presence of hoop compression.
F_{xa} = allowable compressive membrane stress of a cylinder due to an axial compressive load with $\lambda_c \le 0.15$.
F_{xe} = elastic axial compressive failure membrane stress (local buckling) of a cylinder in the absence of other loads.
F_{xha} = allowable axial compressive membrane stress of a cylinder in the presence of hoop compression for $\lambda_c \le 0.15$.
γ = Buckling parameter.
h_o = height of the elliptical head measured to the outside surface.
h_1 = length of a flat bar stiffener, or leg of an angle stiffener, or flange of a tee stiffener, as applicable.
h_2 = length of the angle leg or web of the stiffener, as applicable.
I = moment of inertia of the cylinder or cone cross section.
I_L = actual moment of inertia of the large stiffening ring.
${I^C}_L$ = actual moment of inertia of the composite section comprised of the large stiffening ring and effective length of the shell about the centroidal axis.
I_s = actual moment of inertia of the small stiffening ring.
${I_s}^C$ = actual moment of inertia of the composite section comprised of the small stiffening ring and effective length of the shell about the centroidal axis.
K_o = elliptical head factor:
K_u = coefficient based on end conditions of a member subject to axial compression:

= 2.10 for a member with one free end and the other end fixed. In this case, "member" is the unbraced cylindrical shell or cylindrical shell section as defined in the Nomenclature.
= 1.00 for a member with both ends pinned,
= 0.80 for a member with one end pinned and the other end fixed,
= 0.65 for a member with both ends fixed.

$L, L_1, L_2, \ldots$ = design lengths of the unstiffened vessel sections between lines of support (see Figure 4.4.2). A line of support is
(a) a circumferential line on a head (excluding conical heads) at one-third the depth of the head measured from the tangent line
(b) a small stiffening ring that meets the requirements of 4.4.5.2(b), or
(c) a tubesheet
$L_B, L_{B1}, L_B, \ldots$ = design lengths of the cylinder between bulkheads or large rings designated to act as bulkheads (see Figure 4.4.2).
L_c = axial length of a cone or conical section for an unstiffened cone, or the length from the cone-to-cylinder junction to the first stiffener in the cone for a stiffened cone (see Figures 4.4.6 and 4.4.7).
L_d = design length of an unsupported cylinder or conical shell (see Figure 4.4.5).
L_e = effective length of the shell.
L_{ec} = chord length of segmental circular template used to measure deviation from a true circle of a cylinder.
L_{es} = chord length of segmental circular template used to measure deviation from a true circle of a sphere.
L_F = one-half of the sum of the distances, L_B, from the center line of a large ring to the next large ring of head line of support on either side of the large ring.
L_s = one-half of the sum of the distances from the centerline of a stiffening ring to the next line of support on either side of the ring measured parallel to the axis of the cylinder. A line of support is defined in the definition of L.
L_t = overall length of the vessel.
L_u = laterally unbraced length of cylindrical member that is subject to column buckling, equal to zero when evaluating the shell of a vessel under external pressure only.
λ_c = slenderness factor for column buckling.
M = applied net-section bending moment.
M_x = shell parameter.
P = applied external pressure.
P_a = allowable external pressure in the absence of other loads.
ϕ = angle measured around the circumference from the direction of the applied shear force to the point under consideration.
r_f = inside radius of the flare.
r_g = radius of gyration.
r_k = inside radius of the knuckle.

R = inside or outside radius of cylindrical, conical, and spherical shells, as applicable
R_m = radius to the centerline of the shell.
R_c = radius to the centroid of the combined ring stiffener and effective length of the shell, $R_c = R + Z_c$ (see Figure 4.4.3)
R_L = inside radius of the cylinder at the large end of a cone to cylinder junction.
R_o = outside radius of a cylinder or spherical shell.
R_S = inside radius of the cylinder at the small end of a cone to cylinder junction.
S = section modulus of the shell.
S_y = minimum specified yield strength from Annex 3-D at specified design metal temperature.
σ_1 = principal compressive stress in the 1-direction.
σ_2 = principal compressive stress in the 2-direction.
V = net-section shear force.
t = shell thickness.
t_c = cone thickness.
t_L = shell thickness of large end cylinder at a conical transition.
t_S = shell thickness of small end cylinder at a conical transition.
t_1 = thickness of a flat bar stiffener, or leg of an angle stiffener, or flange of a tee stiffener, as applicable.
t_2 = thickness of the angle leg or web of the stiffener, as applicable.
Z_c = radial distance from the centerline of the shell to the combined section of the ring stiffener and effective length of the shell.
Z_L = radial distance from the centerline of the shell to the centroid of the large ring stiffener.
Z_s = radial distance from the centerline of the shell to the centroid of the small ring stiffener.

4.4.16 TABLES

Table 4.4.1
Maximum Metal Temperature for Compressive Stress Rules

Materials	Temperature Limit °C	Temperature Limit °F
Carbon and Low Alloy Steels — Table 3-A.1	425	800
High Alloy Steels — Table 3-A.3	425	800
Quenched and Tempered Steels — Table 3-A.2	370	700
Aluminum and Aluminum Alloys — Table 3-A.4	150	300
Copper and Copper Alloys — Table 3-A.5	65	150
Nickel and Nickel Alloys — Table 3-A.6	480	900
Titanium and Titanium Alloys — Table 3-A.7	315	600

Table 4.4.2
Algorithm for Computation of Predicted Inelastic Buckling Stress, F_{ic}

```
Read(Fe, E)
Ae = Fe / E
Ficup = MSTS
Ficlow = 0.0
TOLAdiff = 0.00001
Adiff = 1.0
Do While Adiff > TOLAdiff
    Ficg = 0.5 * (Ficup + Ficlow)
    Etg = (1/E +D1 + D2 + D3 + D4)^−1     [See Note (1)]
    Ai = Ficg / Etg
    Adiff = Ai - Ae
    IF (Adiff < 0.0)
        Ficlow = Ficg
    ELSE
        Ficup = Ficg
    END IF
        Adiff = ABS(Adiff)
End Do
Fic = Ficg
```

GENERAL NOTES:
(a) The variables are defined as follows:

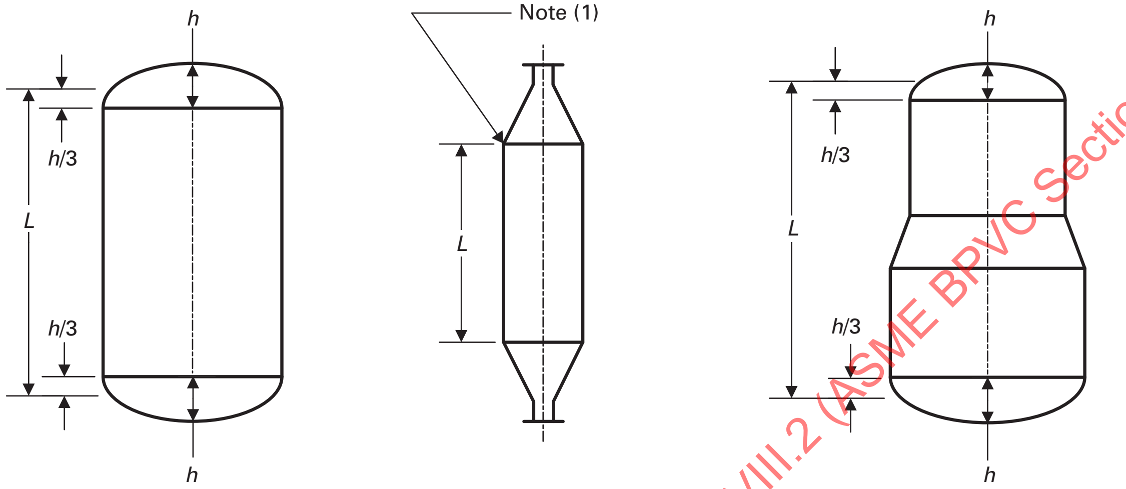
A_e = elastic buckling ratio factor
A_i = inelastic buckling ratio factor
E = modulus of elasticity of material at design temperature
E_{tg} = value of tangent modulus (see 3-D.5.1 for intermediate calculations)
F_e = predicted elastic buckling stress
F_{ic} = value of predicted inelastic buckling stress that satisfies the iterative solution
F_{icg} = value of true stress used to calculate the tangent modulus of elasticity, E_{tg}
$F_{ic,\mathrm{low}}$ = lower bound guess for value F_{icg}
$F_{ic,\mathrm{up}}$ = upper bound guess for value F_{icg}
$\mathrm{TOLA_{diff}}$ = tolerance on iterative solution

(b) This algorithm is only one possible means to determine the value of F_{ic}. Other methods that produce similar results are acceptable.

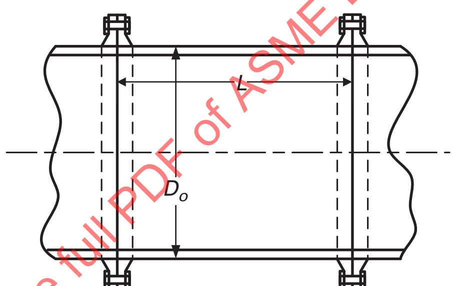
NOTE:
(1) See equations from 3-D.5.1.

4.4.17 FIGURES

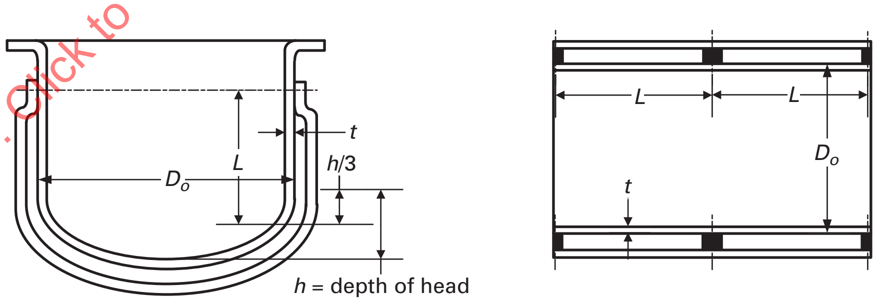
Figure 4.4.1
Lines of Support or Unsupported Length for Typical Vessel Configurations



(a) Typical Unsupported Length Dimensions on Vessel Without Stiffeners



(b) Typical Unsupported Length Between Flange Pairs



(c) Typical Unsupported Length for Jacketed Vessels

NOTE:
(1) It is assumed that both of the cone-to-cylinder junctions meet the requirements of 4.4.13.

Figure 4.4.2
Lines of Support or Unsupported Length for Unstiffened and Stiffened Cylindrical Shells

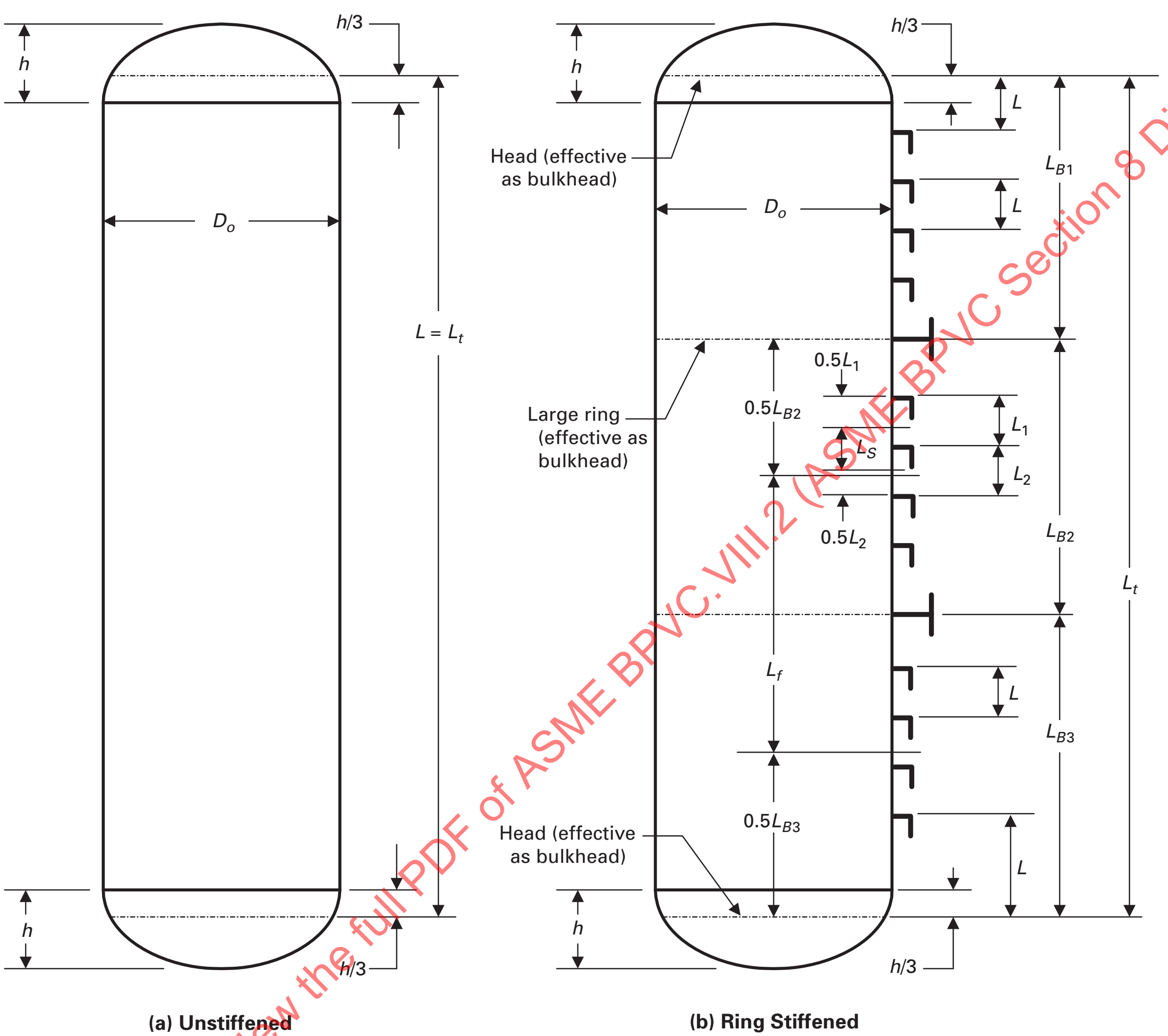
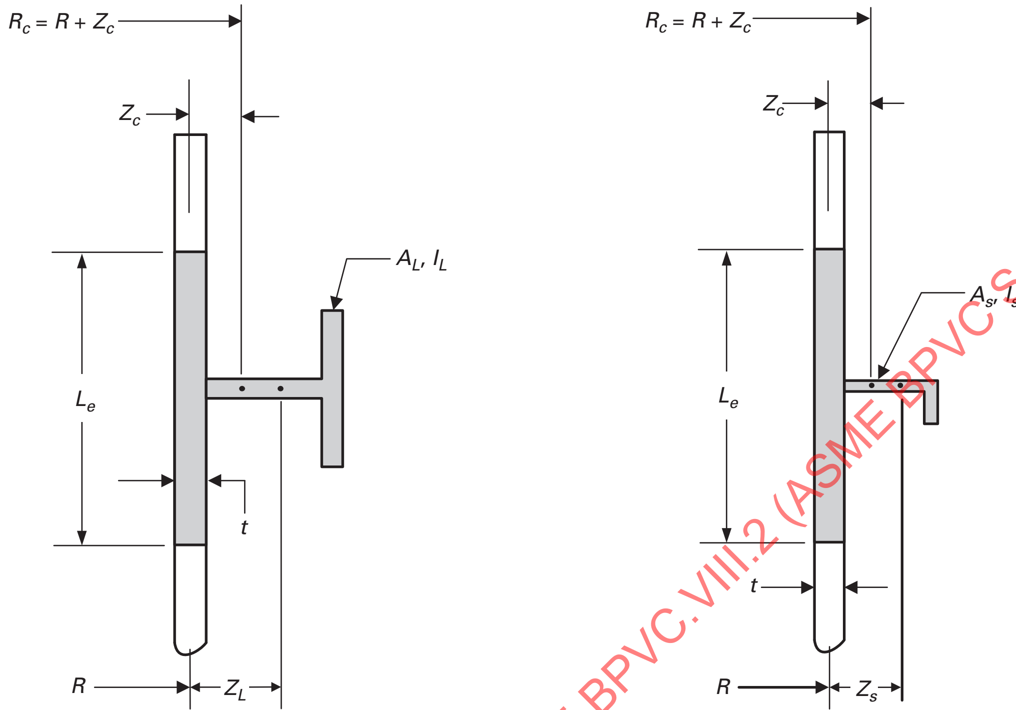


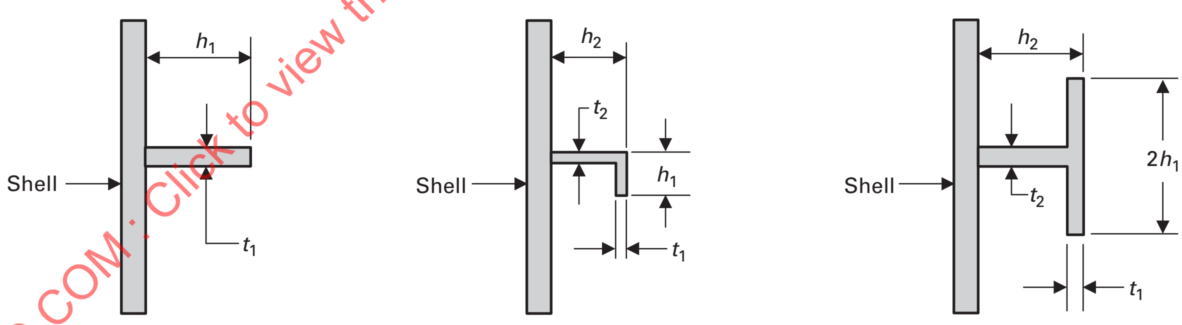
Figure 4.4.3
Stiffener Ring Parameters



(a-1) Stiffening Ring That Acts as a Bulkhead

(a-2) Small Stiffening Ring

(a) Sections Through Stiffening Rings



(b) Stiffener Variables for Local Buckling Calculation

Figure 4.4.4
Various Arrangements of Stiffening Rings for Cylindrical Vessels Subjected to External Pressure

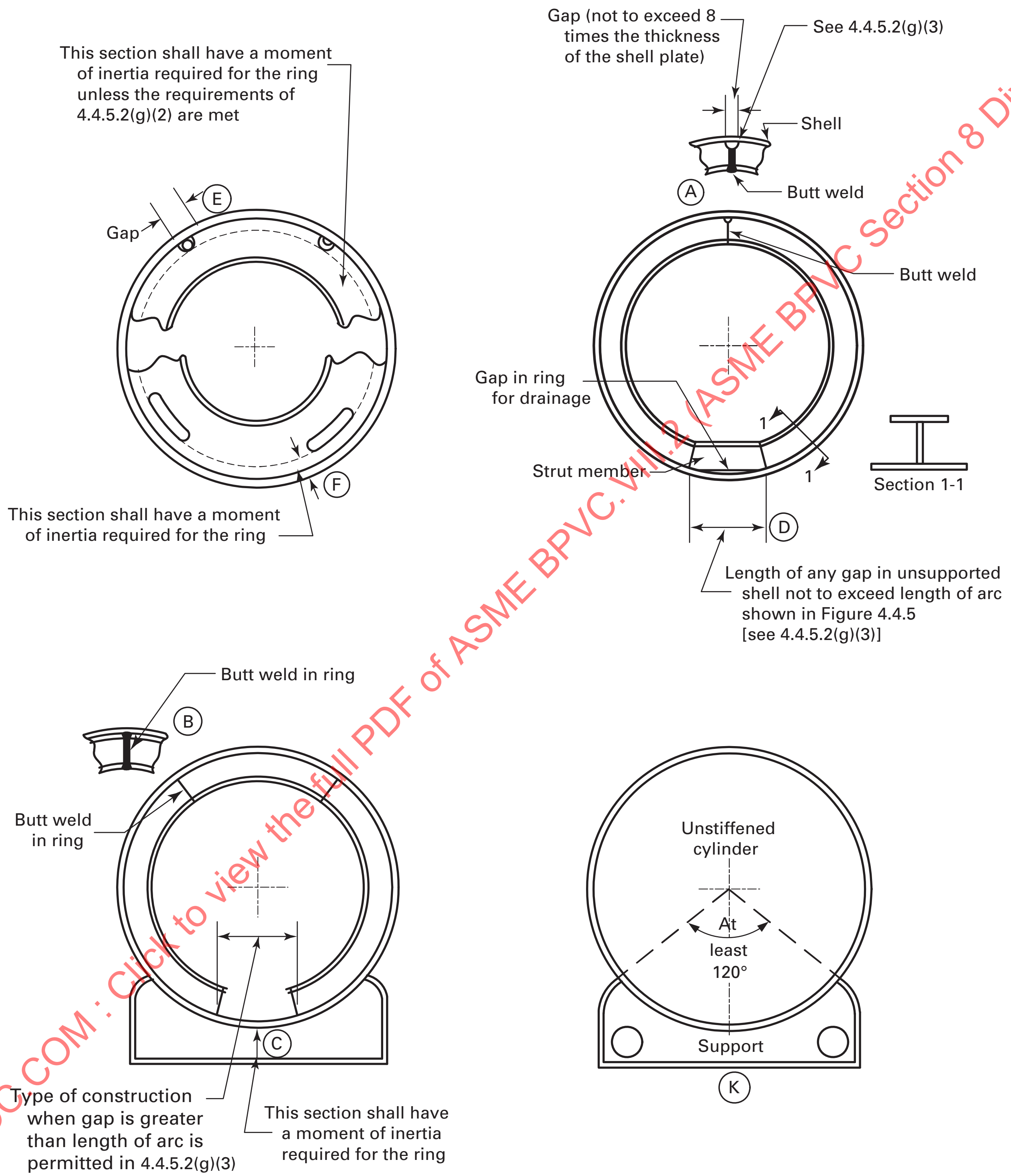
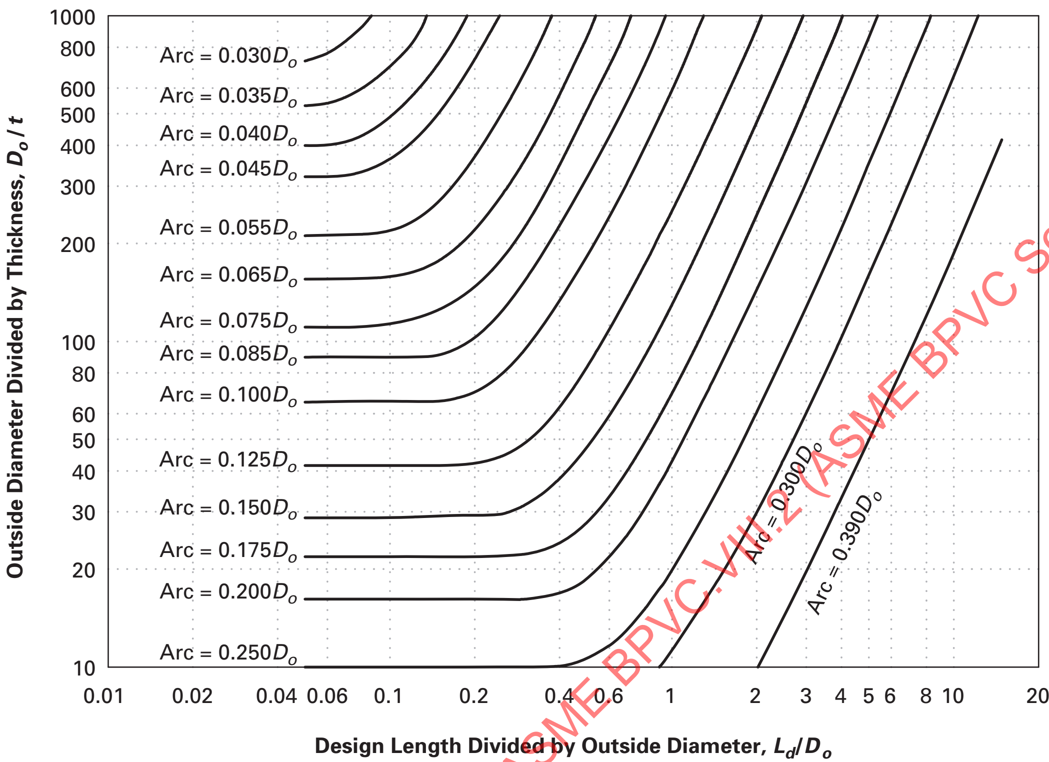


Figure 4.4.5
Maximum Arc of Shell Left Unsupported Because of a Gap in the Stiffening Ring of a Cylindrical Shell Under External Pressure



GENERAL NOTES:

(a) Cylindrical Shells - L_d is the unsupported length of the cylinder and D_o is the outside diameter.

(b) Conical Shells - L_d and D_o are established using the following equations for any cross section having a diameter D_x. In these equations D_L and D_S are the cone large end and small end outside diameters, respectively and L is the unsupported length of the conical section under evaluation.

$$L_d = \left(\frac{L}{2}\right)\left(1 + \frac{D_s}{D_L}\right)\left(\frac{D_S}{D_L}\right) \tag{4.4.133}$$

$$D_o = \frac{0.5\left(D_L + D_S\right)}{\cos\left[\alpha\right]} \tag{4.4.134}$$

Figure 4.4.6
Lines of Support or Unsupported Length for Unstiffened and Stiffened Conical Shells

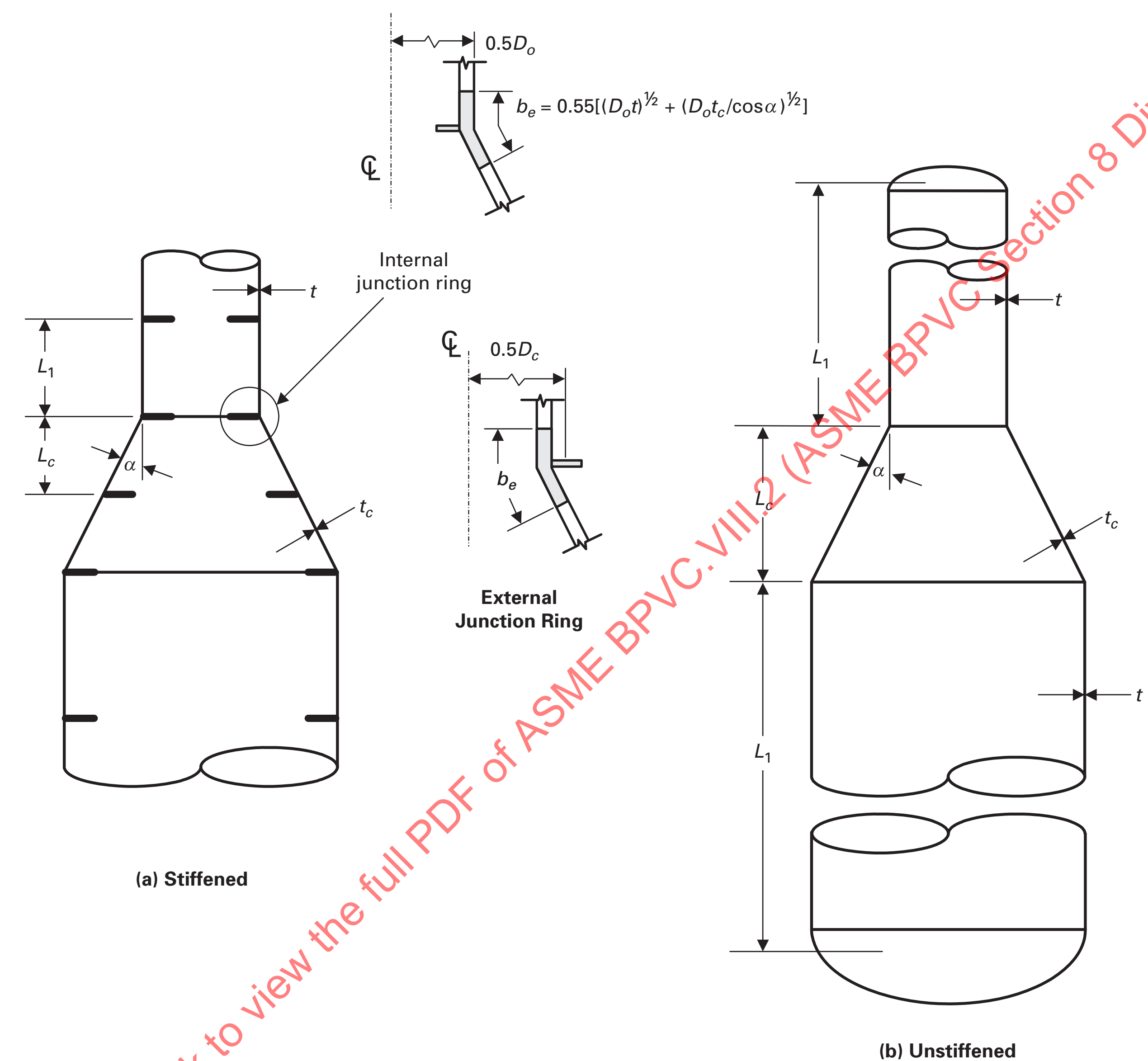
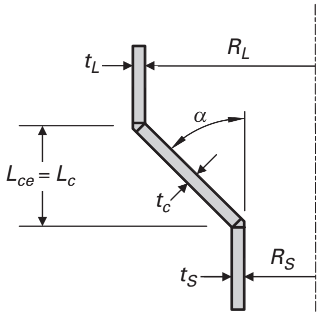
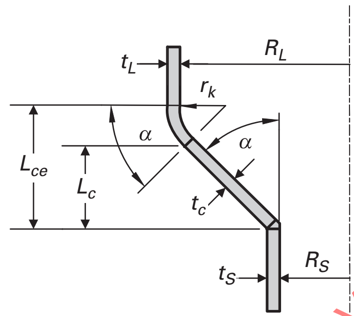


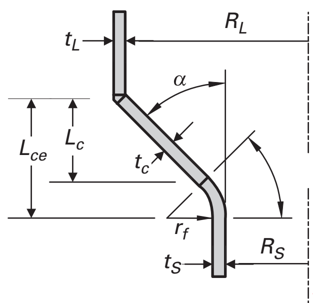
Figure 4.4.7
Lines of Support or Unsupported Length for Unstiffened and Stiffened Conical Shell Transitions With or Without a Knuckle



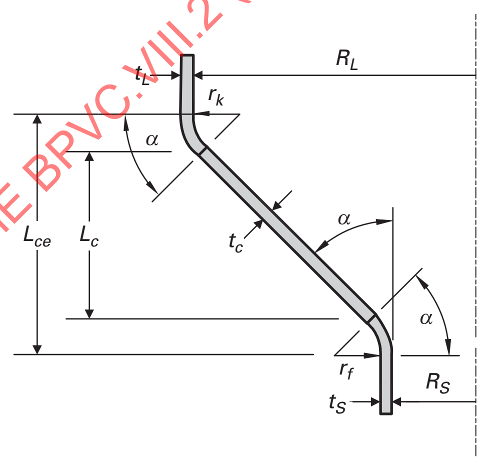
(a) Cone Without a Knuckle at Large End Without a Flare at the Small End



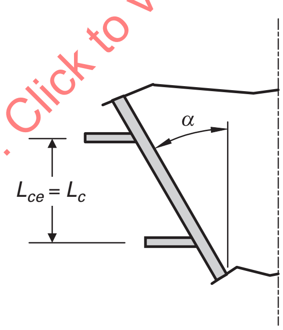
(b) Cone With a Knuckle at Large End Without a Flare at the Small End



(c) Cone Without a Knuckle at Large End With a Flare at the Small End



(d) Cone With a Knuckle at Large End With a Flare at the Small End



(e) Cone With Stiffening Rings

4.5 DESIGN RULES FOR OPENINGS IN SHELLS AND HEADS

4.5.1 SCOPE

The rules in 4.5 are applicable for the design of nozzles in shells and heads subjected to internal pressure, external pressure, and external forces and moments from supplemental loads as defined in 4.1. Configurations, including dimensions and shape, and/or loading conditions that do not satisfy the rules of this 4.5 may be designed in accordance with Part 5.

4.5.2 DIMENSIONS AND SHAPE OF NOZZLES

4.5.2.1 Nozzles shall be circular, elliptical, or of any other shape which results from the intersection of a circular or (21) elliptical cylinder with vessels of the shapes for which design equations are provided in 4.3 and 4.4. The design rules in this paragraph shall be used only if the ratio of the inside diameter of the shell and the shell thickness is less than or equal to 400, except that the rules of 4.5.10 and 4.5.11 may be used without restriction on the ratio of the inside diameter to shell thickness.

4.5.2.2 With the exception of studding outlet type flanges and the straight hubs of forged nozzle flanges (see 4.1.11.3), bolted flange material within the limits of reinforcement shall not be considered to have reinforcement value. With the exception of material within an integral hub, no material in a tubesheet or flat head shall be credited as reinforcement for an opening in an adjacent shell or head.

4.5.2.3 Nozzle openings that do not satisfy the criteria of 4.5.2.1 and other geometries shall be designed in accordance with Part 5.

4.5.3 METHOD OF NOZZLE ATTACHMENT

4.5.3.1 Nozzles may be attached to the shell or head of a vessel by the following methods.

(a) Welded Connections - Nozzles attachment by welding shall be in accordance with the requirements of 4.2.2. If other details not included in this paragraph are required, the nozzle detail shall be designed using Part 5.

(b) Studded Connections - Nozzles may be made by means of studded pad type connections. The vessel shall have a flat surface machined on the shell, or on a built-up pad, or on a properly attached plate or fitting. Drilled holes to be tapped shall not penetrate within one-fourth of the wall thickness from the inside surface of the vessel after deducting corrosion allowance, unless at least the minimum thickness required as above is maintained by adding metal to the inside surface of the vessel. Where tapped holes are provided for studs, the threads shall be full and clean and shall engage the stud for a length, L_{st}, defined by the following equations.

$$L_{st} = \min\left[L_{st1},\ 1.5d_{st}\right] \tag{4.5.1}$$

where

$$L_{st1} = \max\left[d_{st},\ 0.75d_{st}\left(\frac{S_{st}}{S_{tp}}\right)\right] \tag{4.5.2}$$

(c) Threaded Connections - Pipes, tubes, and other threaded connections that conform to the ANSI/ASME Standard for Pipe Threads, General Purpose, Inch (ASME B1.20.1) may be screwed into a threaded hole in a vessel wall, provided the connection size is less than or equal to DN 50 (NPS 2) and the pipe engages the minimum number of threads specified in Table 4.5.1 after allowance has been made for curvature of the vessel wall. The thread shall be a standard taper pipe thread except that a straight thread of at least equal strength may be used if other sealing means to prevent leakage are provided. A built-up pad or a properly attached plate or fitting may be used to provide the metal thickness and number of threads required in Table 4.5.1, or to furnish reinforcement when required.

(d) Expanded Connections - A pipe, tube, or forging may be attached to the wall of a vessel by inserting through an unreinforced opening and expanding into the shell, provided the diameter is not greater than DN 50 (NPS 2) pipe size. A pipe, tube, or forging not exceeding 150 mm (6 in.) in outside diameter may be attached to the wall of a vessel by inserting through a reinforced opening and expanding into the shell. The expanded connection shall be made using one of the following methods:

(1) Firmly rolled in and beaded

(2) Rolled in, beaded, and seal-welded around the edge of the bead

(3) Expanded and flared not less than 3 mm (0.125 in.) over the diameter of the hole

(4) Rolled, flared, and welded; or

(5) Rolled and welded without flaring or beading, provided the ends extend at least 6 mm (0.25 in.), but no more than 10 mm (0.375 in.), through the shell and the throat of the weld is at least 5 mm (0.1875 in.), but no more than 8 mm (0.3125 in.).

4.5.3.2 Additional requirements for nozzle connections are as follows.

(a) When the tube or pipe does not exceed 38 mm (1.5 in.) in outside diameter, the shell may be chamfered or recessed to a depth at least equal to the thickness of the tube or pipe and the tube or pipe may be rolled into place and welded. In no case shall the end of the tube or pipe extend more than 10 mm (0.375 in.) beyond the inside diameter of the shell.

(b) Grooving of shell openings in which tubes and pipe are to be rolled or expanded is permissible.

(c) Expanded connections shall not be used as a method of attachment to vessels used for the processing or storage of flammable and/or noxious gases and liquids unless the connections are seal-welded.

(d) Reinforcing plates and saddles attached to the outside of a vessel shall be provided with at least one vent hole [maximum diameter 11 mm ($^7/_{16}$ in.)] that may be tapped with straight or tapered threads. These vent holes may be left open or may be plugged when the vessel is in service. If the holes are plugged, the plugging material used shall not be capable of sustaining pressure between the reinforcing plate and the vessel wall. Vent holes shall not be plugged during heat treatment.

4.5.4 NOZZLE NECK MINIMUM THICKNESS REQUIREMENTS

4.5.4.1 The minimum nozzle neck thickness for nozzles excluding access openings and openings for inspection shall be determined for internal and external pressure using 4.3 and 4.4, as applicable. Corrosion allowance and the effects of external forces and moments from supplemental loads shall be considered in these calculations. The resulting nozzle neck thickness shall not be less than the smaller of the shell thickness or the thickness given in Table 4.5.2. Corrosion allowance shall be added to the minimum nozzle neck thickness.

4.5.4.2 The minimum nozzle neck thickness for access openings and openings for inspection shall be determined for internal and external pressure using 4.3 and 4.4. Corrosion allowance shall be considered in these calculations.

4.5.5 RADIAL NOZZLE IN A CYLINDRICAL SHELL

(**21**) **4.5.5.1** The procedure to design a radial nozzle in a cylindrical shell subject to pressure loading is shown below. The parameters used in this design procedure are shown in Figures 4.5.1, 4.5.2, and 4.5.3 and shall be considered in the corroded condition.

Step 1. Determine the effective radius of the shell as follows:

(a) For cylindrical shells:

$$R_{\text{eff}} = 0.5D_i \tag{4.5.3}$$

(b) For conical shells R_{eff} is the inside radius of the conical shell at the nozzle centerline to cone junction. The radius is measured normal to the longitudinal axis of the conical shell.

Step 2. Calculate the limit of reinforcement along the vessel wall.

(a) For integrally reinforced nozzles:

$$L_R = \min\left[\sqrt{R_{\text{eff}}t},\, 2R_n\right] \quad \text{for set-in nozzles} \tag{4.5.4}$$

$$L_R = \min\left[\sqrt{R_{\text{eff}}t},\, 2R_n\right] + t_n \quad \text{for set-on nozzles} \tag{4.5.5}$$

(b) For nozzles with reinforcing pads:

$$L_{R1} = \sqrt{R_{\text{eff}}t} + W \tag{4.5.6}$$

$$L_{R2} = \sqrt{\left(R_{\text{eff}} + t\right)\left(t + t_e\right)} \tag{4.5.7}$$

$$L_{R3} = 2R_n \tag{4.5.8}$$

$$L_R = \min\left[L_{R1}, L_{R2}, L_{R3}\right] \quad \text{for set-in nozzles} \tag{4.5.9}$$

$$L_R = \min\left[L_{R1}, L_{R2}, L_{R3}\right] + t_n \quad \text{for set-on nozzles} \tag{4.5.10}$$

Step 3. Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface.

$$L_{H1} = \min\left[1.5t,\ t_e\right] + \sqrt{R_n t_n} \tag{4.5.11}$$

$$L_{H2} = L_{pr1} \tag{4.5.12}$$

$$L_{H3} = 8\left(t + t_e\right) \tag{4.5.13}$$

$$L_H = \min\left[L_{H1}, L_{H2}, L_{H3}\right] + t \quad \text{for set-in nozzles} \tag{4.5.14}$$

$$L_H = \min\left[L_{H1}, L_{H2}, L_{H3}\right] \quad \text{for set-on nozzles} \tag{4.5.15}$$

Step 4. Calculate the limit of reinforcement along the nozzle wall projecting inside the vessel surface, if applicable.

$$L_{I1} = \sqrt{R_n t_n} \tag{4.5.16}$$

$$L_{I2} = L_{pr2} \tag{4.5.17}$$

$$L_{I3} = 8\left(t + t_e\right) \tag{4.5.18}$$

$$L_I = \min\left[L_{I1}, L_{I2}, L_{I3}\right] \tag{4.5.19}$$

Step 5. Determine the total available area near the nozzle opening (see Figures 4.5.1 and 4.5.2). Do not include any area that falls outside of the limits defined by L_H, L_R, and L_I. For variable thickness nozzles, see Figures 4.5.13 and 4.5.14 for metal area definitions of A_2.

$$A_T = A_1 + f_{rn}\left(A_2 + A_3\right) + A_{41} + A_{42} + A_{43} + f_{rp}A_5 \tag{4.5.20}$$

$$f_{rn} = \min\left[\frac{S_n}{S},\ 1\right] \tag{4.5.21}$$

$$f_{rp} = \min\left[\frac{S_p}{S},\ 1\right] \tag{4.5.22}$$

$$A_1 = \left(tL_R\right) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right] \tag{4.5.23}$$

$$\lambda = \min\left[\left\{\frac{2R_n + t_n}{\sqrt{\left(D_i + t_{\text{eff}}\right)t_{\text{eff}}}}\right\},\ 12.0\right] \tag{4.5.24}$$

$$t_{\text{eff}} = t + \left(\frac{A_5 f_{rp}}{L_R}\right) \quad \text{for set-in nozzles} \tag{4.5.25}$$

$$t_{\mathrm{eff}} = t + \left(\frac{A_5 f_{rp}}{L_R - t_n}\right) \quad \text{for set-on nozzles} \tag{4.5.26}$$

$$A_2 = t_n L_H \quad \text{if } t_n = t_{n2} \text{ or if } L_H \le L_{x3} \tag{4.5.27}$$

$$A_2 = A_{2a} + A_{2b} \quad \text{if } t_n > t_{n2} \text{ and } L_{x3} < L_H \le L_{x4} \tag{4.5.28}$$

$$A_2 = A_{2a} + A_{2c} \quad \text{if } t_n > t_{n2} \text{ and } L_H > L_{x4} \tag{4.5.29}$$

$$A_{2a} = t_n L_{x3} \tag{4.5.30}$$

$$A_{2b} = \left(\frac{t_n + t_{nx}}{2}\right) \cdot \min\left[0.78\sqrt{R_n\left(\frac{t_n + t_{nx}}{2}\right)}, \left(L_H - L_{x3}\right)\right] \tag{4.5.31}$$

$$A_{2c} = t_{n2} \cdot \min\left[0.78\sqrt{R_n t_{n2}}, \left(\frac{t_n + t_{n2}}{2t_{n2}}\right)\left(L_{pr4} - L_{pr3}\right) + \left(L_H - L_{x4}\right)\right] \tag{4.5.32}$$

$$L_{x3} = L_{pr3} + t \quad \text{for set-in nozzles} \tag{4.5.33}$$

$$L_{x3} = L_{pr3} \quad \text{for set-on nozzles} \tag{4.5.34}$$

$$L_{x4} = L_{pr4} + t \quad \text{for set-in nozzles} \tag{4.5.35}$$

$$L_{x4} = L_{pr4} \quad \text{for set-on nozzles} \tag{4.5.36}$$

$$t_{nx} = \left[1 + \frac{\left(t_n - t_{n2}\right)}{t_{n2}} \cdot \frac{\left(L_{x4} - L_H\right)}{\left(L_{pr4} - L_{pr3}\right)}\right] t_{n2} \tag{4.5.37}$$

$$A_3 = t_n L_I \tag{4.5.38}$$

$$A_{41} = 0.5 L_{41}^2 \tag{4.5.39}$$

$$A_{42} = 0.5 L_{42}^2 \tag{4.5.40}$$

$$A_{43} = 0.5 L_{43}^2 \tag{4.5.41}$$

$$A_{5a} = W t_e \tag{4.5.42}$$

$$A_{5b} = L_R t_e \quad \text{for set-in nozzles} \tag{4.5.43}$$

$$A_{5b} = \left(L_R - t_n\right) t_e \quad \text{for set-on nozzles} \tag{4.5.44}$$

$$A_5 = \min\left[A_{5a}, A_{5b}\right] \tag{4.5.45}$$

Step 6. Determine the applicable forces.

$$f_N = PR_{xn}L_H \qquad \text{for set-in nozzles} \tag{4.5.46}$$

$$f_N = PR_{xn}\left(L_H + t\right) \qquad \text{for set-on nozzles} \tag{4.5.47}$$

$$f_s = PR_{xs}\left(L_R + t_n\right) \qquad \text{for set-in nozzles} \tag{4.5.48}$$

$$f_s = PR_{xs}L_R \qquad \text{for set-on nozzles} \tag{4.5.49}$$

$$f_Y = PR_{xs}R_{nc} \tag{4.5.50}$$

$$R_{xn} = \frac{t_n}{\ln\left[1 + \dfrac{t_n}{R_n}\right]} \tag{4.5.51}$$

$$R_{xs} = \frac{t_{\text{eff}}}{\ln\left[1 + \dfrac{t_{\text{eff}}}{R_{\text{eff}}}\right]} \tag{4.5.52}$$

Step 7. Determine the average local primary membrane stress and the general primary membrane stress at the nozzle intersection.

$$\sigma_{\text{avg}} = \frac{\left(f_N + f_s + f_Y\right)}{A_T} \tag{4.5.53}$$

$$\sigma_{\text{circ}} = \frac{PR_{xs}}{t_{\text{eff}}} \tag{4.5.54}$$

Step 8. Determine the maximum local primary membrane stress at the nozzle intersection:

$$P_L = \max\left[\left(2\sigma_{\text{avg}} - \sigma_{\text{circ}}\right),\ \sigma_{\text{circ}}\right] \tag{4.5.55}$$

Step 9. The calculated maximum local primary membrane stress should satisfy eq. (4.5.56). If the nozzle is subjected to internal pressure, then the allowable stress, S_{allow}, is given by eq. (4.5.57). If the nozzle is subjected to external pressure, then the allowable stress is given by eq. (4.5.58).

$$P_L \leq S_{\text{allow}} \tag{4.5.56}$$

where

$$S_{\text{allow}} = 1.5SE \qquad \text{for internal pressure} \tag{4.5.57}$$

$$S_{\text{allow}} = 1.5S \qquad \text{for external pressure} \tag{4.5.58}$$

Step 10. Determine the maximum allowable working pressure at the nozzle intersection.

$$P_{\max 1} = \frac{S_{\text{allow}}}{\dfrac{2A_p}{A_T} - \dfrac{R_{xs}}{t_{\text{eff}}}} \tag{4.5.59}$$

$$P_{\max 2} = S\left(\frac{t}{R_{xs}}\right) \tag{4.5.60}$$

$$P_{\max} = \min\left[P_{\max 1}, P_{\max 2}\right] \tag{4.5.61}$$

where

$$A_p = \frac{f_N + f_S + f_Y}{P} \tag{4.5.62}$$

4.5.5.2 If the nozzle is subject to external forces and moments from supplemental loads as defined in 4.1, then the local stresses at the nozzle-to-shell intersection shall be evaluated in accordance with 4.5.15.

4.5.6 HILLSIDE NOZZLE IN A CYLINDRICAL SHELL

For a hillside nozzle in a cylindrical shell (see Figure 4.5.4), the design procedure in 4.5.5 shall be used with the following substitution.

$$R_{nc} = \max\left[\left(\frac{R_{ncl}}{2}\right),\ R_n\right] \tag{4.5.63}$$

where

$$R_{ncl} = R_{\text{eff}}\left(\theta_1 - \theta_2\right) \tag{4.5.64}$$

$$\theta_1 = \arccos\left[\frac{D_X}{R_{\text{eff}}}\right] \tag{4.5.65}$$

$$\theta_2 = \arccos\left[\frac{D_X + R_n}{R_{\text{eff}}}\right] \tag{4.5.66}$$

4.5.7 NOZZLE IN A CYLINDRICAL SHELL ORIENTED AT AN ANGLE FROM THE LONGITUDINAL AXIS

For a nozzle in a cylindrical shell oriented at an angle from the longitudinal axis, the design procedure in 4.5.5 shall be used with the following substitutions (see Figure 4.5.5):

$$R_{nc} = \frac{R_n}{\sin[\theta]} \tag{4.5.67}$$

$$f_s = PR_{xs}\left(L_R + \frac{t}{\tan[\theta]} + \frac{t_n}{\sin[\theta]}\right) \qquad \text{for set-in nozzles} \tag{4.5.68}$$

$$f_s = PR_{xs}\left(L_R + \frac{t}{\tan[\theta]}\right) \qquad \text{for set-on nozzles} \tag{4.5.69}$$

$$A_1 = t\left(L_R + \frac{t}{2\tan[\theta]}\right) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right] \tag{4.5.70}$$

4.5.8 RADIAL NOZZLE IN A CONICAL SHELL

For a radial nozzle in a conical shell (see Figure 4.5.6), the design procedure in 4.5.5 shall be used with the following substitutions.

$$f_s = \frac{P}{\cos[\alpha]}\left(R_{\text{eff}} + R_{nc}\sin[\alpha] + \frac{L_t \sin[\alpha]}{2}\right)L_t \tag{4.5.71}$$

$$f_Y = \frac{P}{\cos[\alpha]}\left(R_{\text{eff}} + \frac{R_{nc}\sin[\alpha]}{2}\right)R_{nc} \tag{4.5.72}$$

$$L_t = L_R + t_n \qquad \text{for set-in nozzles} \tag{4.5.73}$$

$$L_t = L_R \qquad \text{for set-on nozzles} \tag{4.5.74}$$

$$R_{xs} = \frac{t_{\text{eff}}}{\ln\left[1 + \dfrac{t_{\text{eff}}\cos[\alpha]}{R_{\text{eff}} + L_c\sin[\alpha]}\right]} \tag{4.5.75}$$

$$L_c = L_t + R_{nc} \tag{4.5.76}$$

$$R_{nc} = R_n \tag{4.5.77}$$

4.5.9 NOZZLE IN A CONICAL SHELL

4.5.9.1 If a nozzle in a conical shell is oriented perpendicular to the longitudinal axis (see Figure 4.5.7), then the design procedure in 4.5.8 shall be used with the following substitutions.

$$R_{nc} = \frac{R_n}{\cos[\alpha]} \tag{4.5.78}$$

$$A_1 = t\left(L_R + \frac{t\cdot\tan[\alpha]}{2}\right)\cdot\max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right] \tag{4.5.79}$$

$$L_t = L_R + \frac{t_n}{\cos[\alpha]} + t\cdot\tan[\alpha] \qquad \text{for set-in nozzles} \tag{4.5.80}$$

$$L_t = L_R + t\cdot\tan[\alpha] \qquad \text{for set-on nozzles} \tag{4.5.81}$$

4.5.9.2 If a nozzle in a conical shell is oriented parallel to the longitudinal axis (see Figure 4.5.8), then the design procedure in 4.5.8 shall be used with the following substitution.

$$R_{nc} = \frac{R_n}{\sin[\alpha]} \tag{4.5.82}$$

$$A_1 = t\left(L_R - \frac{t}{2\tan[\alpha]}\right)\cdot\max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right] \tag{4.5.83}$$

$$L_t = L_R - \frac{t}{\tan[\alpha]} + \frac{t_n}{\sin[\alpha]} \qquad \text{for set-in nozzles} \tag{4.5.84}$$

$$L_t = L_R - \frac{t}{\tan[\alpha]} \qquad \text{for set-on nozzles} \tag{4.5.85}$$

4.5.10 RADIAL NOZZLE IN A SPHERICAL SHELL OR FORMED HEAD

4.5.10.1 The procedure to design a radial nozzle in a spherical shell or formed head subject to pressure loading is shown below. The parameters used in this design procedure are shown in Figures 4.5.1, 4.5.2, and 4.5.9 and shall be considered in the corroded condition. (**21**)

Step 1. Determine the effective radius of the shell or formed head as follows.

(a) For spherical shells:

$$R_{\text{eff}} = 0.5D_i \tag{4.5.86}$$

(b) For ellipsoidal heads:

$$R_{\text{eff}} = \frac{0.9D_i}{6}\left[2 + \left(\frac{D_i}{2h}\right)^2\right] \tag{4.5.87}$$

(c) For torispherical heads:

$$R_{\text{eff}} = L \tag{4.5.88}$$

Step 2. Calculate the limit of reinforcement along the vessel wall.

(a) For integrally reinforced nozzles in spherical shells and ellipsoidal heads:

$$L_R = \min\left[\sqrt{R_{\text{eff}}t},\ 2R_n\right] \quad \text{for set-in nozzles} \tag{4.5.89}$$

$$L_R = \min\left[\sqrt{R_{\text{eff}}t},\ 2R_n\right] + t_n \quad \text{for set-on nozzles} \tag{4.5.90}$$

(b) For integrally reinforced nozzles in torispherical heads:

$$L_{R1} = \max\left[\frac{D_i}{2} - \left(D_R + \left(R_n + t_n\right)\cos\left[\theta\right]\right),\ 0.0\right] \tag{4.5.91}$$

$$\theta = \arcsin\left[\frac{D_R}{L}\right] \quad \text{when } D_R \le L\sin\left[\theta_o\right] \tag{4.5.92}$$

$$\theta = \arcsin\left[\frac{D_R - \frac{D_i}{2} + r_k}{r_k}\right] \quad \text{when } D_R > L\sin\left[\theta_o\right] \tag{4.5.93}$$

$$\theta_0 = \arcsin\left[\frac{\frac{D_i}{2} - r_k}{L - r_k}\right] \tag{4.5.94}$$

$$L_{R2} = \min\left[\sqrt{R_{\text{eff}}t},\ 2R_n\right] \tag{4.5.95}$$

$$L_R = \min\left[L_{R1}, L_{R2}\right] \tag{4.5.96}$$

(c) For pad reinforced nozzles:

$$L_{R1} = \sqrt{R_{\text{eff}}t} + W \tag{4.5.97}$$

$$L_{R2} = \sqrt{\left(R_{\text{eff}} + t\right)\left(t + t_e\right)} \tag{4.5.98}$$

$$L_{R3} = 2R_n \tag{4.5.99}$$

$$L_R = \min\left[L_{R1}, L_{R2}, L_{R3}\right] \quad \text{for set-in nozzles} \tag{4.5.100}$$

$$L_R = \min\left[L_{R1}, L_{R2}, L_{R3}\right] + t_n \quad \text{for set-on nozzles} \tag{4.5.101}$$

Step 3. Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface.

$$L_H = \min\left[t + t_e + F_p\sqrt{R_n t_n},\ L_{pr1} + t\right] \quad \text{for set-in nozzles} \tag{4.5.102}$$

$$L_H = \min\left[t_e + F_p\sqrt{R_n t_n},\ L_{pr1}\right] \quad \text{for set-on nozzles} \tag{4.5.103}$$

(a) For spherical shells and heads:

$$F_p = C_n \tag{4.5.104}$$

(b) For ellipsoidal and torispherical heads:

$$F_p = \min\left[C_n, C_p\right] \quad \text{for } X_o > 0.35D_i \tag{4.5.105}$$

$$F_p = C_n \quad \text{for } X_o \le 0.35D_i \tag{4.5.106}$$

$$X_o = \min\left[D_R + (R_n + t_n)\cos\left[\theta\right], \frac{D_i}{2}\right] \tag{4.5.107}$$

(1) For ellipsoidal heads,

$$C_p = \exp\left[\frac{0.35D_i - X_o}{16t}\right] \tag{4.5.108}$$

$$\theta = \arctan\left[\left(\frac{h}{R}\right)\cdot\left(\frac{D_R}{\sqrt{R^2 - D_R^2}}\right)\right] \tag{4.5.109}$$

(2) For torispherical heads,

$$C_p = \exp\left[\frac{0.35D_i - X_o}{8t}\right] \tag{4.5.110}$$

θ is calculated using eqs. (4.5.92) through (4.5.94).

The parameter C_n is given by eq. (4.5.111).

$$C_n = \min\left[\left(\frac{t + t_e}{t_n}\right)^{0.35},\ 1.0\right] \tag{4.5.111}$$

Step 4. Calculate the limit of reinforcement along the nozzle wall projecting inside the vessel surface, if applicable.

$$L_I = \min\left[F_p\sqrt{R_n t_n},\ L_{pr2}\right] \tag{4.5.112}$$

Step 5. Determine the total available area near the nozzle opening (see Figures 4.5.1 and 4.5.2) where f_{rn} and f_{rp} are given by eqs. (4.5.21) and (4.5.22), respectively. Do not include any area that falls outside of the limits defined by L_H, L_R, and L_I. For variable thickness nozzles, see Figures 4.5.13 and 4.5.14 for metal area definitions of A_2.

$$A_T = A_1 + f_{rn}\left(A_2 + A_3\right) + A_{41} + A_{42} + A_{43} + f_{rp}A_5 \tag{4.5.113}$$

$$A_1 = tL_R \tag{4.5.114}$$

$$A_2 = t_n L_H \quad \text{if } t_n = t_{n2} \text{ or if } L_H \le L_{x3} \tag{4.5.115}$$

$$A_2 = A_{2a} + A_{2b} \quad \text{if } t_n > t_{n2} \text{ and } L_{x3} < L_H \le L_{x4} \tag{4.5.116}$$

$$A_2 = A_{2a} + A_{2c} \quad \text{if } t_n > t_{n2} \text{ and } L_H > L_{x4} \tag{4.5.117}$$

$$A_{2a} = t_n L_{x3} \tag{4.5.118}$$

$$A_{2b} = \left(\frac{t_n + t_{nx}}{2}\right) \cdot \min\left[0.78\sqrt{R_n\left(\frac{t_n + t_{nx}}{2}\right)}, \left(L_H - L_{x3}\right)\right] \tag{4.5.119}$$

$$A_{2c} = t_{n2} \cdot \min\left[0.78\sqrt{R_n t_{n2}}, \left(\frac{t_n + t_{n2}}{2t_{n2}}\right)\left(L_{pr4} - L_{pr3}\right) + \left(L_H - L_{x4}\right)\right] \tag{4.5.120}$$

$$L_{x3} = L_{pr3} + t \qquad \text{for set-in nozzles} \tag{4.5.121}$$

$$L_{x3} = L_{pr3} \qquad \text{for set-on nozzles} \tag{4.5.122}$$

$$L_{x4} = L_{pr4} + t \qquad \text{for set-in nozzles} \tag{4.5.123}$$

$$L_{x4} = L_{pr4} \qquad \text{for set-on nozzles} \tag{4.5.124}$$

$$t_{nx} = \left[1 + \frac{\left(t_n - t_{n2}\right)}{t_{n2}} \cdot \frac{\left(L_{x4} - L_H\right)}{\left(L_{pr4} - L_{pr3}\right)}\right] t_{n2} \tag{4.5.125}$$

$$A_3 = t_n L_I \tag{4.5.126}$$

$$A_{41} = 0.5L_{41}^2 \tag{4.5.127}$$

$$A_{42} = 0.5L_{42}^2 \tag{4.5.128}$$

$$A_{43} = 0.5L_{43}^2 \tag{4.5.129}$$

$$A_{5a} = Wt_e \tag{4.5.130}$$

$$A_{5b} = L_R t_e \qquad \text{for set-in nozzles} \tag{4.5.131}$$

$$A_{5b} = \left(L_R - t_n\right) t_e \qquad \text{for set-on nozzles} \tag{4.5.132}$$

$$A_5 = \min\left[A_{5a}, A_{5b}\right] \tag{4.5.133}$$

Step 6. Determine the applicable forces.

$$f_N = PR_{xn} L_H \qquad \text{for set-in nozzles} \tag{4.5.134}$$

$$f_N = PR_{xn}\left(L_H + t\right) \qquad \text{for set-on nozzles} \tag{4.5.135}$$

$$f_S = \frac{PR_{xs}\left(L_R + t_n\right)}{2} \qquad \text{for set-in nozzles} \tag{4.5.136}$$

$$f_S = \frac{PR_{xs} L_R}{2} \qquad \text{for set-on nozzles} \tag{4.5.137}$$

$$f_Y = \frac{PR_{xs}R_{nc}}{2} \tag{4.5.138}$$

$$R_{xn} = \frac{t_n}{\ln\left[1 + \frac{t_n}{R_n}\right]} \tag{4.5.139}$$

$$R_{xs} = \frac{t_{\text{eff}}}{\ln\left[1 + \frac{t_{\text{eff}}}{R_{\text{eff}}}\right]} \tag{4.5.140}$$

$$t_{\text{eff}} = t + \left(\frac{A_5 f_{rp}}{L_R}\right) \qquad \text{for set-in nozzles} \tag{4.5.141}$$

$$t_{\text{eff}} = t + \left(\frac{A_5 f_{rp}}{L_R - t_n}\right) \qquad \text{for set-on nozzles} \tag{4.5.142}$$

Step 7. Determine the average local primary membrane stress and the general primary membrane stress in the vessel.

$$\sigma_{\text{avg}} = \frac{\left(f_N + f_S + f_Y\right)}{A_T} \tag{4.5.143}$$

$$\sigma_{\text{circ}} = \frac{PR_{xs}}{2t_{\text{eff}}} \tag{4.5.144}$$

Step 8. Determine the maximum local primary membrane stress at the nozzle intersection.

$$P_L = \max\left[\left\{2\sigma_{\text{avg}} - \sigma_{\text{circ}}\right\}, \sigma_{\text{circ}}\right] \tag{4.5.145}$$

Step 9. The calculated maximum local primary membrane stress should satisfy eq. (4.5.146). If the nozzle is subjected to internal pressure, then the allowable stress, S_{allow}, is given by eq. (4.5.57). If the nozzle is subjected to external pressure, then the allowable stress is given by eq. (4.5.58).

$$P_L \le S_{\text{allow}} \tag{4.5.146}$$

Step 10. Determine the maximum allowable working pressure of the nozzle.

$$P_{\max 1} = \frac{S_{\text{allow}}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{2t_{\text{eff}}}} \tag{4.5.147}$$

$$P_{\max 2} = 2S\left(\frac{t}{R_{xs}}\right) \tag{4.5.148}$$

$$P_{\max} = \min\left[P_{\max 1}, P_{\max 2}\right] \tag{4.5.149}$$

where

$$A_p = \frac{\left(f_N + f_S + f_Y\right)}{P} \tag{4.5.150}$$

4.5.10.2 If the nozzle is subject to external forces and moments from supplemental loads as defined in 4.1, then the local stresses at the nozzle-to-shell intersection shall be evaluated in accordance with 4.5.15.

4.5.11 HILLSIDE OR PERPENDICULAR NOZZLE IN A SPHERICAL SHELL OR FORMED HEAD

4.5.11.1 If a hillside nozzle is located in a spherical shell or hemispherical head [see Figure 4.5.10, sketch (a)], the design procedure in 4.5.10 shall be used with $R_{nc} = R_{ncl}$, which is calculated using eqs. (4.5.64) through (4.5.66) by substituting D_R for D_X.

4.5.11.2 If a hillside or perpendicular nozzle is located in an ellipsoidal head or torispherical head [see Figure 4.5.10, sketch (b)], the design procedure in 4.5.10 shall be used with the following substitutions.

$$R_{nc} = \frac{R_n}{\cos\left[\theta\right]} \qquad \text{for hillside nozzles} \tag{4.5.151}$$

$$R_{nc} = \frac{R_n}{\sin\left[\theta\right]} \qquad \text{for perpendicular nozzles} \tag{4.5.152}$$

$$\theta = \arctan\left[\left(\frac{h}{R}\right) \cdot \left(\frac{D_R}{\sqrt{R^2 - D_R^2}}\right)\right] \qquad \text{for ellipsoidal heads} \tag{4.5.153}$$

For torispherical heads, θ is calculated using eqs. (4.5.92) through (4.5.94).

4.5.12 CIRCULAR NOZZLES IN A FLAT HEAD

(21) **4.5.12.1** The procedure to design a nozzle in a flat head subject to pressure loading is shown below. The parameters used in this design procedure are shown in Figures 4.5.1 and 4.5.2. As an alternative, a central nozzle in an integral flat head may be designed using the procedure in 4.6.4.

Step 1. Calculate the maximum unit moment at the nozzle intersection.

$$M_o = \frac{S t_f^4}{6\left(t + C_e t_e\right)^2} \tag{4.5.154}$$

$$C_e = \min\left[\left\{\frac{\left(W + 0.5L_{42}\right)t_e}{R_n t}\right\},\ 0.6\right] \tag{4.5.155}$$

Step 2. Calculate the nozzle parameters.

$$\lambda_n = \frac{1.285}{\sqrt{R_{nm} t_n}} \tag{4.5.156}$$

$$C_1 = \sinh^2\left[C_L\right] + \sin^2\left[C_L\right] \tag{4.5.157}$$

$$C_2 = \sinh^2\left[C_L\right] - \sin^2\left[C_L\right] \tag{4.5.158}$$

$$C_L = \min\left[\left\{\lambda_n\left(L_{pr1} + t + L_{pr2}\right)\right\},\ 3.0\right] \tag{4.5.159}$$

$$C_3 = \frac{L_{pr1} + t}{L_{pr1} + t + \min\left[\left(\lambda_n\right)^{-1},\ L_{pr2}\right]} \tag{4.5.160}$$

$$R_{nm} = R_n + 0.5t_n \tag{4.5.161}$$

$$R_{xn} = \frac{t_n}{\ln\left[1 + \frac{t_n}{R_n}\right]} \tag{4.5.162}$$

$$x_t = 0.5\lambda_n\left(t + t_e + L_{41} + L_{43}\right) \quad \text{for set-in nozzles} \tag{4.5.163}$$

$$x_t = 0.5\lambda_n\left(t_e + L_{41}\right) \quad \text{for other set-on nozzles} \tag{4.5.164}$$

$$x_t = 0.5\lambda_n\left(t + t_e + L_{41}\right) \quad \text{for set-on nozzles with full penetration welds} \tag{4.5.164a}$$

$$C_t = \exp\left[-x_t\right] \tag{4.5.165}$$

Step 3. Determine the maximum local primary membrane stress in the nozzle at the intersection.

$$P_L = \frac{2M_o\lambda_n^2R_{nm}C_tC_1C_3}{t_nC_2} + \frac{PR_{xn}}{t_n} \tag{4.5.166}$$

Step 4. The maximum local primary membrane stress at the nozzle intersection shall satisfy eq. (4.5.167). The allowable stress, S_{allow}, is given by eq. (4.5.57).

$$P_L \le S_{allow} \tag{4.5.167}$$

4.5.12.2 If the nozzle is subject to external forces and moments from supplemental loads as defined in 4.1, then the local stresses at the nozzle-to-shell intersection shall be evaluated in accordance with 4.5.15.

4.5.13 SPACING REQUIREMENTS FOR NOZZLES

4.5.13.1 The limit of reinforcement, L_R (see Figures 4.5.1 and 4.5.2), for a nozzle shall not overlap with a gross structural discontinuity (see 4.2.5.1). The limit of reinforcement, L_S (see Figure 4.5.11), may be reduced from the maximum permitted by other rules to allow closer placement of nozzles so long as all opening reinforcement requirements are satisfied.

4.5.13.2 If the limits of reinforcement determined in accordance with 4.5.5 for nozzles in cylindrical or conical shells or 4.5.10 for nozzles in spherical or formed heads, do not overlap, no additional analysis is required. If the limits of reinforcement overlap, the following procedure shall be used or the design shall be evaluated in accordance with the design by analysis rules of Part 5.

4.5.13.3 The maximum local primary membrane stress and the nozzle maximum allowable working pressure shall be determined following 4.5.5 or 4.5.10, for each individual nozzle with the value of L_R determined as follows.

(a) For two openings with overlapping limits of reinforcement (see Figure 4.5.11):

$$L_R = L_S\left(\frac{R_{nA}}{R_{nA} + R_{nB}}\right) \quad \text{for nozzle A} \tag{4.5.168}$$

$$L_R = L_S\left(\frac{R_{nB}}{R_{nA} + R_{nB}}\right) \quad \text{for nozzle B} \tag{4.5.169}$$

(b) For three openings with overlapping limits of reinforcement (see Figure 4.5.12):

$$L_R = \min\left[L_{S1}\left(\frac{R_{nA}}{R_{nA} + R_{nB}}\right),\ L_{S2}\left(\frac{R_{nA}}{R_{nA} + R_{nC}}\right)\right] \quad \text{for nozzle A} \tag{4.5.170}$$

$$L_R = \min\left[L_{S1}\left(\frac{R_{nB}}{R_{nA} + R_{nB}}\right),\ L_{S3}\left(\frac{R_{nB}}{R_{nB} + R_{nC}}\right)\right] \quad \text{for nozzle B} \tag{4.5.171}$$

$$L_R = \min\left[L_{S2}\left(\frac{R_{nC}}{R_{nA} + R_{nC}}\right),\ L_{S3}\left(\frac{R_{nC}}{R_{nB} + R_{nC}}\right)\right] \quad \text{for nozzle C} \tag{4.5.172}$$

(c) For more than three openings with overlapping limits of reinforcement, repeat the above procedure for each pair of adjacent nozzles.

4.5.14 STRENGTH OF NOZZLE ATTACHMENT WELDS

4.5.14.1 The strength of nozzle attachment welds shall be sufficient to resist the discontinuity force imposed by pressure for nozzles attached to a cylindrical, conical, or spherical shell or formed head as determined in 4.5.14.2. Nozzles attached to flat heads shall have their strength of attachment welds evaluated as determined in 4.5.14.3. The effects of external forces and moments from supplemental loads shall be considered.

4.5.14.2 The procedure to evaluate attachment welds of nozzles in a cylindrical, conical, or spherical shell or formed head subject to pressure loading is shown below.

Step 1. Determine the discontinuity force factor

(a) For set-on nozzles:

$$k_y = 1.0 \tag{4.5.173}$$

(b) For set-in nozzles:

$$k_y = \frac{R_{nc} + t_n}{R_{nc}} \tag{4.5.174}$$

Step 2. Calculate Weld Length Resisting Discontinuity Force

(a) Weld length of nozzle to shell weld

$$L_\tau = \frac{\pi}{2}\left(R_n + t_n\right) \quad \text{for radial nozzles} \tag{4.5.175}$$

$$L_\tau = \frac{\pi}{2}\sqrt{\frac{\left(R_{nc} + t_n\right)^2 + \left(R_n + t_n\right)^2}{2}} \quad \text{for nonradial nozzles} \tag{4.5.176}$$

(b) Weld length of pad to shell weld

$$L_{\tau p} = \frac{\pi}{2}\left(R_n + t_n + W\right) \quad \text{for radial nozzles} \tag{4.5.177}$$

$$L_{\tau p} = \frac{\pi}{2}\sqrt{\frac{\left(R_{nc} + t_n + W\right)^2 + \left(R_n + t_n + W\right)^2}{2}} \quad \text{for nonradial nozzles} \tag{4.5.178}$$

Step 3. Compute the weld throat dimensions, as applicable.

$$L_{41T} = 0.7071 L_{41} \tag{4.5.179}$$

$$L_{42T} = 0.7071 L_{42} \tag{4.5.180}$$

$$L_{43T} = 0.7071 L_{43} \tag{4.5.181}$$

Step 4. Determine if the weld sizes are acceptable.

(a) If the nozzle is integrally reinforced, and the computed shear stress in the weld given by eq. (4.5.182) satisfies eq. (4.5.183), then the design is complete. If the shear stress in the weld does not satisfy eq. (4.5.183), increase the weld size and return to Step 3. For nozzles on heads, A_2 and A_3 are to be calculated using $F_p = 1.0$, when computing f_{welds} using eq. (4.5.184).

$$\tau = \frac{f_{\text{welds}}}{L_\tau\left(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T}\right)} \tag{4.5.182}$$

$$\tau \le S \tag{4.5.183}$$

where

$$f_{\text{welds}} = \min\left[f_Y k_y, 1.5S_n\left(A_2 + A_3\right), \frac{\pi}{4}PR_n^2 k_y^2\right] \tag{4.5.184}$$

(b) If the nozzle is pad reinforced, and the computed shear stresses in the welds given by eqs. (4.5.185) through (4.5.187) satisfy eq. (4.5.188), then the design is complete. If the shear stress in the weld does not satisfy eq. (4.5.188), increase the weld size and return to Step 3.

$$\tau_1 = \frac{f_{ws}}{L_\tau\left(0.6t_{w1} + 0.49L_{43T}\right)} \tag{4.5.185}$$

$$\tau_2 = \frac{f_{wp}}{L_\tau\left(0.6t_{w2} + 0.49L_{41T}\right)} \tag{4.5.186}$$

$$\tau_3 = \frac{f_{wp}}{L_{\tau p}\left(0.49L_{42T}\right)} \tag{4.5.187}$$

$$\max\left[\tau_1, \tau_2, \tau_3\right] \le S \tag{4.5.188}$$

where

$$f_{ws} = \frac{f_{\text{welds}} t \cdot S}{t \cdot S + t_e S_p} \tag{4.5.189}$$

$$f_{wp} = \frac{f_{\text{welds}} t_e S_p}{t \cdot S + t_e S_p} \tag{4.5.190}$$

4.5.14.3 The procedure to evaluate attachment welds of a nozzle in a flat head subject to pressure loading is shown below.

Step 1. Compute the weld throat dimensions, as applicable.

$$L_{41T} = 0.7071L_{41} \tag{4.5.191}$$

$$L_{42T} = 0.7071L_{42} \tag{4.5.192}$$

$$L_{43T} = 0.7071L_{43} \tag{4.5.193}$$

Step 2. Determine if the weld sizes are acceptable.

(a) If the nozzle is integrally reinforced and set-in the flat head, and the computed shear stress in the welds given by eqs. (4.5.194) through (4.5.196) satisfy eq. (4.5.197), then the design is complete. If the shear stress in the welds does not satisfy eq. (4.5.197), increase the weld size and return to Step 1.

$$\tau_1 = \frac{V_s}{0.6t_{x1} + 0.49L_{43T}} \tag{4.5.194}$$

$$\tau_2 = \frac{V_s}{0.6t_{x2} + 0.49L_{41T}} \tag{4.5.195}$$

$$\tau_3 = \frac{P\left(R_n + t_n\right)}{2\left(0.49L_{41T} + 0.6t_{w1} + 0.49L_{43T}\right)} \tag{4.5.196}$$

$$\max\left[\tau_1, \tau_2, \tau_3\right] \le S \tag{4.5.197}$$

where

$$V_s = \frac{0.3 S t_{rf}^4}{t^3} \tag{4.5.198}$$

$$t_{x1} = \min\left[t_{w1},\ 0.5t\right] \tag{4.5.199}$$

$$t_{x2} = \min\left[\max\left[\left(t_{w1} - 0.5t\right),\ 0\right],\ 0.5t\right] \tag{4.5.200}$$

(b) If the nozzle is pad reinforced and set-in the flat head, and the computed shear stress in the welds given by eqs. (4.5.201) through (4.5.204) satisfy eq. (4.5.205), then the design is complete. If the shear stress in the welds does not satisfy eq. (4.5.205), increase the weld size and return to Step 1.

$$\tau_1 = \frac{V_s}{0.6 t_{w1} + 0.49 L_{43T}} \tag{4.5.201}$$

$$\tau_2 = \frac{V_s}{0.6 t_{w2} + 0.49 L_{41T}} \tag{4.5.202}$$

$$\tau_3 = \frac{V_s\left(R_n + t_n\right)}{0.49 L_{42T}\left(R_n + t_n + W\right)} \tag{4.5.203}$$

$$\tau_4 = \frac{P\left(R_n + t_n\right)}{2\left(0.49 L_{41T} + 0.6 t_{w1} + 0.6 t_{w2} + 0.49 L_{43T}\right)} \tag{4.5.204}$$

$$\max\left[\tau_1, \tau_2, \tau_3, \tau_4\right] \le S \tag{4.5.205}$$

The parameter V_s is given by eq. (4.5.198).

(c) If the nozzle is integrally reinforced and set-on the flat head, and the computed shear stress in the weld given by eqs. (4.5.206) through (4.5.207) satisfies eq. (4.5.208), then the design is complete. If the shear stress in the weld does not satisfy eq. (4.5.208), increase the weld size and return to Step 1.

$$\tau_1 = \frac{2M_o}{t\left(0.6 t_{w1} + 0.49 L_{41T}\right)} \tag{4.5.206}$$

$$\tau_2 = \frac{P R_n}{2\left(0.6 t_{w1} + 0.49 L_{41T}\right)} \tag{4.5.207}$$

$$\max\left[\tau_1, \tau_2\right] \le S \tag{4.5.208}$$

(21) 4.5.15 LOCAL STRESSES IN SHELLS, FORMED HEADS, AND NOZZLES FROM EXTERNAL LOADS ON NOZZLES

Localized stresses in shells, formed heads, and nozzle necks at nozzle locations shall be evaluated using one of the methods shown below. Some of the methods calculate the stresses in the shell or formed head only, while others calculate the stresses in the shell or formed head and the attached nozzle. Regardless of the method chosen, the stresses in the shell or formed head and the nozzle shall be evaluated. For each method, the acceptance criteria shall be in accordance with Part 5 including the requirements for the nozzle neck in 5.6.

(a) Nozzles in cylindrical shells – stress calculations shall be in accordance with WRC 537 (supersedes WRC 107), WRC 297, or ASME STP-PT-074.

(b) Nozzles in formed heads – stress calculations shall be in accordance with WRC 537 (supersedes WRC 107) or ASME STP-PT-074.

(c) For all configurations, and as an alternative to (a) and (b), the stress calculations may be performed using a numerical analysis such as the finite element method in accordance with Part 5.

4.5.16 INSPECTION OPENINGS

4.5.16.1 All pressure vessels for use with compressed air and those subject to internal corrosion or having parts subject to erosion or mechanical abrasion (see 4.1.4), except as permitted otherwise in this paragraph, shall be provided with a suitable manhole, handhole, or other inspection opening(s) for examination and cleaning. Compressed air as used in this paragraph is not intended to include air which has had moisture removed to provide an atmospheric dew point of −46°C (−50°F) or less.

4.5.16.2 Inspection openings may be omitted in heat exchangers where the construction does not permit access to the shell side, such as fixed tubesheet heat exchangers or U-tube and floating tubesheet heat exchangers with Configuration a, b, or c as shown in Figure 4.18.4 or Figure 4.18.11. When inspection openings are not provided, the Manufacturer's Data Report shall include one of the following notations under "Remarks":

(a) "4.5.16.2" when inspection openings are omitted in fixed tubesheet heat exchangers, or U-tube and floating tubesheet heat exchangers with Configuration a, b, or c as shown in Figure 4.18.4 or Figure 4.18.11;

(b) "4.5.16.3", "4.5.16.4", "4.5.16.5" when provision for inspection is made in accordance with one of these paragraphs;

(c) the statement "for noncorrosive service."

4.5.16.3 Vessels over 300 mm (12 in.) inside diameter under air pressure which also contain, as an inherent requirement of their operation, other substances which will prevent corrosion need not have openings for inspection only, provided the vessel contains suitable openings through which inspection can be made conveniently, and provided such openings are equivalent in size and number to the requirements for inspection openings in 4.5.16.6.

4.5.16.4 For vessels 300 mm (12 in.) or less in inside diameter, openings for inspection only may be omitted if there are at least two removable pipe connections not less than DN 20 (NPS $^{3}/_{4}$).

4.5.16.5 Vessels less than 400 mm (16 in.) and over 300 mm (12 in.) inside diameter shall have at least two handholes or two threaded pipe plug inspection openings of not less than DN 40 (NPS $1^{1}/_{2}$) except as permitted by the following: when vessels less than 400 mm (16 in.) and over 300 mm (12 in.) inside diameter are to be installed so that inspection cannot be made without removing the vessel from the assembly, openings for inspection only may be omitted, provided there are at least two removable pipe connections of not less than DN 40 (NPS $1^{1}/_{2}$).

4.5.16.6 Vessels that require access or inspection openings shall be equipped as follows:

(a) All vessels less than 450 mm (18 in.) and over 300 mm (12 in.) inside diameter shall have at least two handholes or two plugged, threaded inspection openings of not less than DN 40 (NPS $1^{1}/_{2}$);

(b) All vessels 450 mm (18 in.) to 900 mm (36 in.), inclusive, inside diameter shall have a manhole or at least two handholes or two plugged, threaded inspection openings of not less than DN 50 (NPS 2);

(c) All vessels over 900 mm (36 in.) inside diameter shall have a manhole, except that those whose shape or use makes one impracticable shall have at least two handholes 100 mm × 150 mm (4 in. × 6 in.) or two equal openings of equivalent area;

(d) When handholes or pipe plug openings are permitted for inspection openings in place of a manhole, one handhole or one pipe plug opening shall be in each head or in the shell near each head service;

(e) Openings with removable heads or cover plates intended for other purposes may be used in place of the required inspection openings, provided they are equal at least to the size of the required inspection openings;

(f) A single opening with removable head or cover plate may be used in place of all the smaller inspection openings, provided it is of such size and location as to afford at least an equal view of the interior;

(g) Flanged and/or threaded connections from which piping, instruments, or similar attachments can be removed may be used in place of the required inspection openings, provided that:

(1) The connections are at least equal to the size of the required openings; and

(2) The connections are sized and located to afford at least an equal view of the interior as the required inspection openings.

4.5.16.7 When inspection or access openings are required, they shall comply at least with the following requirements.

(a) An elliptical or obround manhole shall be not less than 300 mm × 400 mm (12 in. × 16 in.). A circular manhole shall be not less than 400 mm (16 in.) inside diameter.

(b) A handhole opening shall be not less than 50 mm × 75 mm (2 in. × 3 in.), but should be as large as is consistent with the size of the vessel and the location of the opening.

4.5.16.8 All access and inspection openings in a shell or unstayed head shall be designed in accordance with the rules of this Part for openings.

4.5.16.9 When a threaded opening is to be used for inspection or cleaning purposes, the closing plug or cap shall be of a material suitable for the pressure and no material shall be used at a temperature exceeding the maximum temperature allowed in Part 3 for that material. The thread shall be a standard taper pipe thread except that a straight thread of at least equal strength may be used if other sealing means to prevent leakage are provided.

4.5.16.10 Manholes of the type in which the internal pressure forces the cover plate against a flat gasket shall have a minimum gasket bearing width of 17 mm (0.6875 in.).

(21) 4.5.17 OPENINGS SUBJECT TO AXIAL COMPRESSION, EXTERNAL PRESSURE, AND THE COMBINATION THEREOF

4.5.17.1 The reinforcement for openings in cylindrical and conical vessels subject to axial compression, external pressure, and the combination thereof that do not exceed 25% of the cylinder diameter or 80% of the ring spacing into which the opening is placed may be designed in accordance with the following rules. Openings in cylindrical and conical vessels that exceed these limitations shall be designed in accordance with Part 5.

4.5.17.2 Reinforcement for nozzle openings in cylindrical and conical vessels designed for external pressure alone shall be in accordance with the requirements of 4.5.5 through 4.5.9, as applicable. The required thickness shall be determined in accordance with 4.5.4. The external pressure shall be used as P.

4.5.17.3 For cylindrical and conical vessels designed for axial compression (which includes axial load and/or bending moment) without external pressure, the reinforcement of openings shall be in accordance with the following:

$$A_r = 0 \qquad \text{for} \qquad d \le 0.4\sqrt{Rt} \tag{4.5.209}$$

$$A_r = 0.5dt_r \qquad \text{for} \qquad d > 0.4\sqrt{Rt} \qquad \text{and} \qquad \gamma_n \le \left(\frac{(R/t)}{291} + 0.22\right)^2 \tag{4.5.210}$$

$$A_r = dt_r \qquad \text{for} \qquad d > 0.4\sqrt{Rt} \qquad \text{and} \qquad \gamma_n > \left(\frac{(R/t)}{291} + 0.22\right)^2 \tag{4.5.211}$$

where

$$\gamma_n = \left(\frac{d}{2\sqrt{Rt}}\right) \tag{4.5.212}$$

The reinforcement shall be placed within a distance of $0.75\sqrt{Rt}$ from the edge of the opening. Reinforcement available from the nozzle neck shall be limited to a thickness not exceeding the shell plate thickness at the nozzle attachment, and be placed within a limit measured normal to the outside surface of the vessel shell of $0.5\sqrt{(d/2)t_n}$, but not exceeding $2.5t_n$.

4.5.17.4 For cylindrical and conical vessels designed for axial compression in combination with external pressure, the reinforcement shall be the larger of that required for external pressure alone, 4.5.17.2, or axial compression alone, 4.5.17.3.

(21) 4.5.18 NOMENCLATURE

- A_1 = area contributed by the vessel wall.
- A_2 = area contributed by the nozzle outside the vessel wall.
- A_{2a} = portion of area A_2 for variable nozzle wall thickness, contributed by the nozzle wall within L_{pr3} (see Figures 4.5.13 and 4.5.14).
- A_{2b} = portion of area A_2 for variable nozzle wall thickness, contributed by the nozzle wall outside of L_{pr3} when $L_H \le L_{x4}$ (see Figures 4.5.13 and 4.5.14).
- A_{2c} = portion of area A_2 for variable nozzle wall thickness, contributed by the nozzle wall outside of L_{pr3} when $L_H > L_{x4}$ (see Figures 4.5.13 and 4.5.14).
- A_3 = area contributed by the nozzle inside the vessel wall.
- A_{41} = area contributed by the outside nozzle fillet weld.

A_{42} = area contributed by the pad to vessel fillet weld.
A_{43} = area contributed by the inside nozzle fillet weld.
A_5 = area contributed by the reinforcing pad.
A_p = area resisting pressure, used to determine the nozzle opening discontinuity force.
A_r = area of reinforcement required.
A_T = total area within the assumed limits of reinforcement.
α = one-half of the apex angle of a conical shell.
C_1 = geometry-dependent coefficient of a flat head.
C_2 = geometry-dependent coefficient of a flat head.
C_3 = geometry-dependent coefficient of a flat head.
C_4 = geometry-dependent coefficient of a flat head.
C_7 = geometry-dependent coefficient of a flat head.
C_8 = geometry-dependent coefficient of a flat head.
C_{10} = geometry-dependent coefficient of a flat head.
C_e = pad thickness credit factor of a flat head.
C_L = dimensionless scale factor of a flat head.
C_{md} = thickness modification factor of a flat head.
C_n = finite element analysis derived factor to modify the effective nozzle length L_H.
C_p = finite element analysis derived factor to modify the effective nozzle length L_H.
C_t = geometry-dependent coefficient of a flat head.
D_i = inside diameter of a shell or head.
D_R = distance from the head center line to the nozzle center line.
D_X = distance from the cylinder center line to the nozzle center line.
d = inside diameter of the opening.
d_{st} = nominal diameter of the stud.
E = weld joint factor (see 4.2);
= 1.0 if the nozzle does not intersect a weld seam.
F_p = nozzle attachment factor.
F_{ha} = minimum value of the allowable compressive stress of the shell and nozzle material from 4.4, evaluated at the design temperature.
f_N = force from internal pressure in the nozzle outside of the vessel.
f_{rn} = nozzle material factor.
f_{rp} = pad material factor.
f_S = force from internal pressure in the shell.
f_{welds} = overall discontinuity induced by existence of a nozzle.
f_{wp} = discontinuity force carried by welds t_{w2} and L_{43}.
f_Y = discontinuity force from internal pressure.
F_p = nozzle attachment factor.
h = height of the ellipsoidal head measured to the inside surface.
k_y = discontinuity force factor that adjusts the discontinuity force to the nozzle outer diameter.
L = inside crown radius of a torispherical head.
L_{41} = weld leg length of the outside nozzle fillet weld.
L_{42} = weld leg length of the pad to vessel fillet weld.
L_{43} = weld leg length of the inside nozzle fillet weld.
L_{41T} = throat dimension of the outside nozzle fillet weld.
L_{42T} = throat dimension for the pad to vessel fillet weld.
L_{43T} = throat dimension for inside nozzle fillet weld.
L_c = effective length of the vessel wall from the central axis on the nozzle (see Figures 4.5.6 through 4.5.8).
L_H = effective length of nozzle wall outside the vessel.
L_I = effective length of nozzle wall inside the vessel.
L_{pr1} = nozzle projection from the outside of the vessel wall.
L_{pr2} = nozzle projection from the inside of the vessel wall.
L_{pr3} = nozzle projection from the outside of the vessel wall for variable thickness nozzles within constant thickness t_n (see Figures 4.5.13 and 4.5.14.)
L_{pr4} = nozzle projection from the outside of the vessel wall for variable thickness nozzles to nozzle thickness t_{n2} (see Figures 4.5.13 and 4.5.14.)
L_R = effective length of the vessel wall.

L_S = shortest distance between the outer surface of the two adjacent nozzle walls.
L_{s1} = shortest distance between the outer surface of nozzle A and nozzle B.
L_{s2} = shortest distance between the outer surface of nozzle A and nozzle C.
L_{s3} = shortest distance between the outer surface of nozzle B and nozzle C.
L_{st} = thread engagement length.
L_t = effective length of the vessel wall from the inside corner of the nozzle-vessel intersection (see Figures 4.5.5 through 4.5.8).
L_τ = weld length of the nozzle to shell weld.
$L_{\tau p}$ = weld length of the pad to shell weld.
L_{x3} = nozzle projection from the nozzle end for variable thickness nozzles within constant thickness, t_n (see Figures 4.5.13 and 4.5.14).
L_{x4} = nozzle projection from nozzle end for variable thickness nozzles to nozzle thickness, t_{n2} (see Figures 4.5.13 and 4.5.14).
λ = non-linearity parameter applied to the metal area A_1.
λ_n = nozzle scale factor of a flat head.
M_o = maximum bending moment per unit length at the nozzle intersection.
P = internal or external design pressure.
$P_{\max}$ = maximum allowable pressure at the nozzle-shell intersection.
$P_{\max 1}$ = maximum allowable pressure in the nozzle.
$P_{\max 2}$ = maximum allowable pressure in the shell.
P_L = maximum local primary membrane stress at the nozzle intersection.
R = vessel inside radius.
R_{eff} = effective pressure radius.
R_n = nozzle inside radius.
R_{nA} = internal radius of nozzle A.
R_{nB} = internal radius of nozzle B.
R_{nC} = internal radius of nozzle C.
R_{nc} = radius of the nozzle opening in the vessel along the long chord, for radial nozzles $R_{nc} = R_n$
R_{ncl} = radius of the nozzle opening in the vessel along the long chord for hillside nozzle (see Figure 4.5.4).
R_{nm} = nozzle mean radius.
R_{xn} = nozzle radius for force calculation.
R_{xs} = shell radius for force calculation.
r_k = knuckle radius at the junction for torispherical heads.
S = allowable stress from Annex 3-A for the vessel (shell, head, or cone, as applicable) at the design temperature.
S_{allow} = local allowable membrane stress at the nozzle intersection.
S_n = allowable stress from Annex 3-A for the nozzle at the design temperature.
S_p = allowable stress from Annex 3-A for the pad at the design temperature.
S_{st} = allowable stress from Annex 3-A of the stud material at the design temperature.
S_{tp} = allowable stress from Annex 3-A of the tapped material at the design temperature.
σ_{avg} = average primary membrane stress.
σ_{circ} = general primary membrane stress.
θ = angle between the nozzle center line and the vessel center line.
θ_1 = angle between the vessel horizontal axis and the hillside nozzle center line (see Figure 4.5.4).
θ_2 = angle between the vessel horizontal axis and the hillside nozzle inside radius at the nozzle to vessel intersection (see Figure 4.5.4).
t = nominal thickness of the vessel wall.
t_e = thickness of the reinforcing pad.
t_{eff} = effective thickness used in the calculation of pressure stress near the nozzle opening.
t_n = nominal thickness of the nozzle wall.
t_{n2} = nominal wall thickness of the thinner portion of a variable thickness nozzle.
t_{nx} = wall thickness at the variable thickness portion of the nozzle, which is a function of position.
t_r = thickness of shell required for axial compression loads without external pressure.
t_{rf} = minimum required flat head thickness, exclusive of corrosion allowance, as required by 4.6.
t_{w1} = nozzle to shell groove weld depth.
t_{w2} = nozzle to reinforcing pad groove weld depth.
τ = average "effective" shear stress in welds due to pressure (includes joint efficiency).
τ_1 = shear stress through load path 1.

τ_2 = shear stress through load path 2.
τ_3 = shear stress through load path 3.
τ_4 = shear stress through load path 4.
V_s = shear load per unit length.
W = width of the reinforcing pad.
X_o = distance from the nozzle outside diameter to the head center.
x_t = dimensions scale factor of a flat head.

4.5.19 TABLES

Table 4.5.1
Minimum Number of Pipe Threads for Connections

Size of Pipe	Threads Engaged	Minimum Plate Thickness Required
DIN 15, 20 (NPS 0.5, 0.75 in.)	6	11 mm (0.43 in.)
DIN 25, 32, 40 (NPS 1.0, 1.25, 1.5 in.)	7	16 mm (0.61 in.)
DIN 50 (NPS 2 in.)	8	18 mm (0.70 in.)

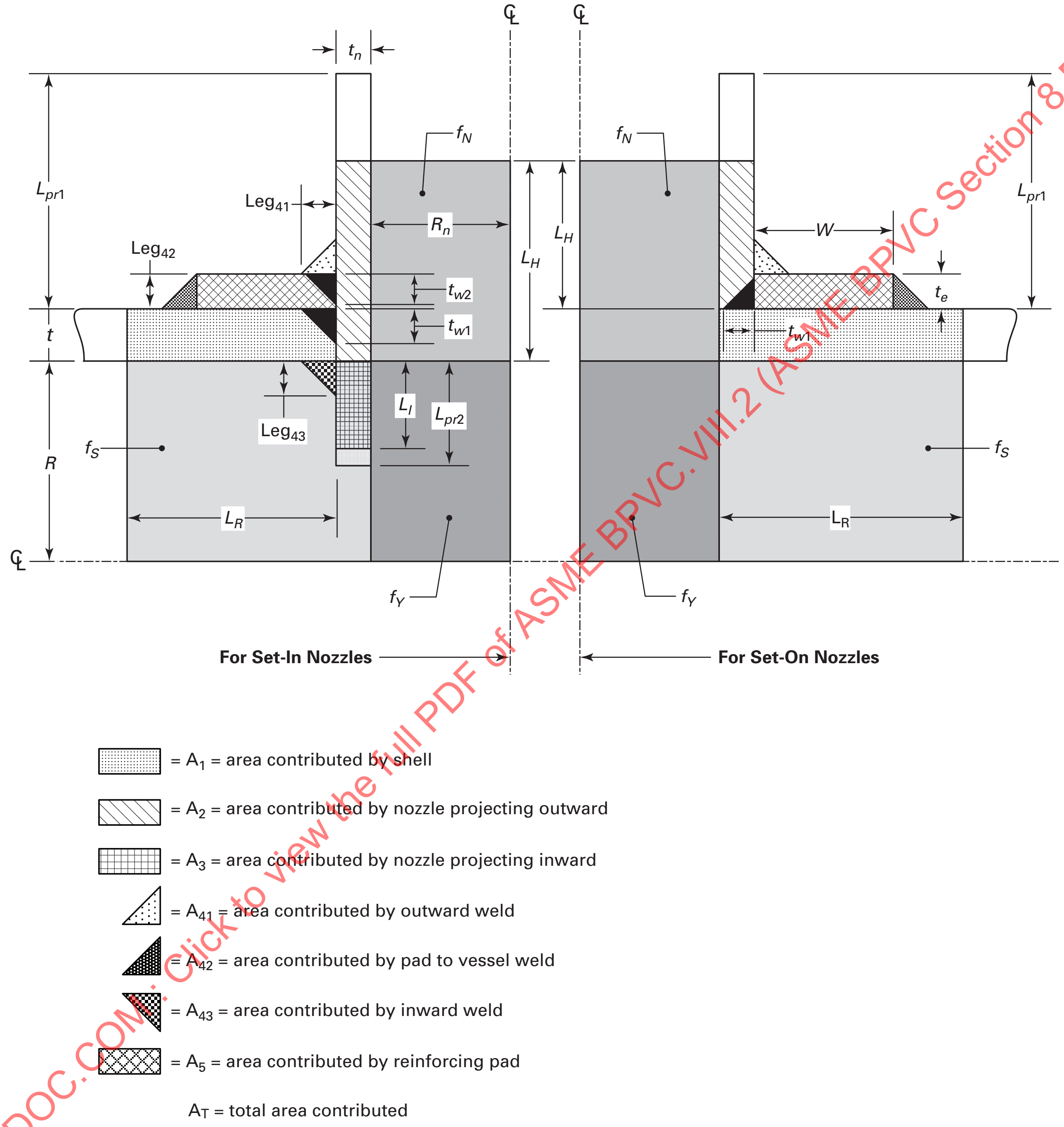
Table 4.5.2
Nozzle Minimum Thickness Requirements

	Minimum Thickness	
Nominal Size	mm	in.
DN 6 (NPS $^1/_8$)	1.51	0.060
DN 8 (NPS $^1/_4$)	1.96	0.077
DN 10 (NPS $^3/_8$)	2.02	0.080
DN 15 (NPS $^1/_2$)	2.42	0.095
DN 20 (NPS $^3/_4$)	2.51	0.099
DN 25 (NPS 1)	2.96	0.116
DN 32 (NPS $1^1/_4$)	3.12	0.123
DN 40 (NPS $1^1/_2$)	3.22	0.127
DN 50 (NPS 2)	3.42	0.135
DN 65 (NPS $2^1/_2$)	4.52	0.178
DN 80 (NPS 3)	4.80	0.189
DN 90 (NPS $3^1/_2$)	5.02	0.198
DN 100 (NPS 4)	5.27	0.207
DN 125 (NPS 5)	5.73	0.226
DN 150 (NPS 6)	6.22	0.245
DN 200 (NPS 8)	7.16	0.282
DN 250 (NPS 10)	8.11	0.319
≥ DN 300 (NPS 12)	8.34	0.328

GENERAL NOTE: For nozzles having a specified outside diameter not equal to the outside diameter of an equivalent standard DN (NPS) size, the DN (NPS) chosen from the table shall be one having an equivalent outside diameter larger than the actual nozzle outside diameter.

4.5.20 FIGURES

Figure 4.5.1
Nomenclature for Reinforced Openings



(21)

Figure 4.5.2
Nomenclature for Variable Thickness Openings

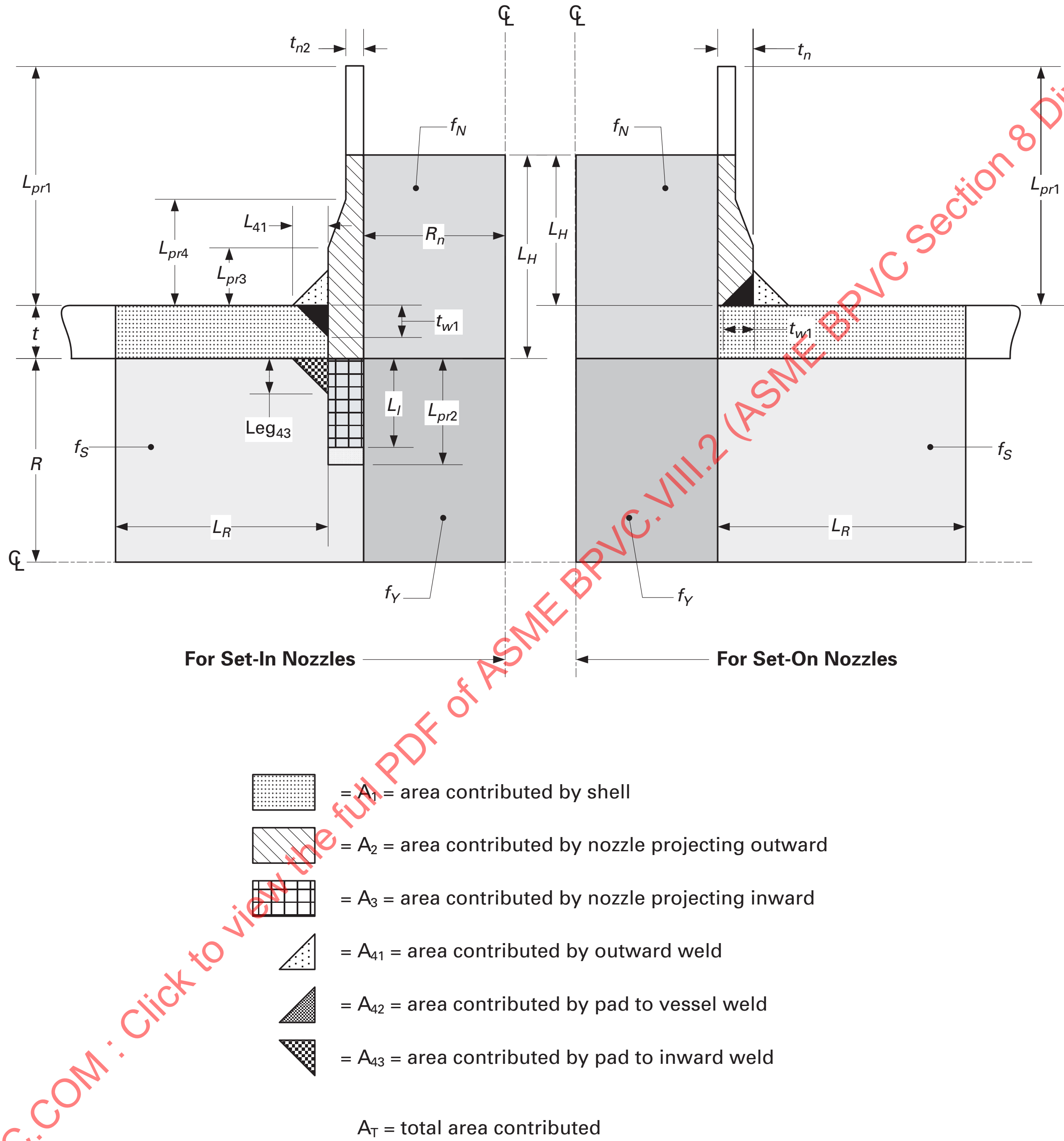


Figure 4.5.3
Radial Nozzle in a Cylindrical Shell

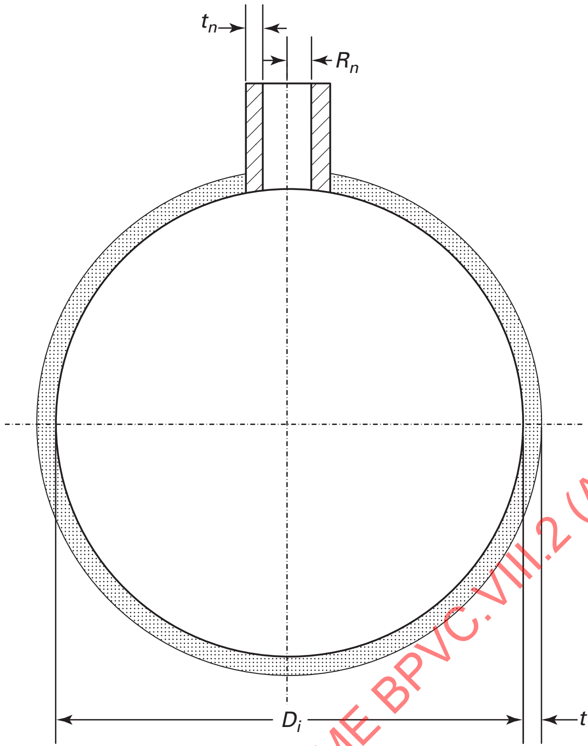


Figure 4.5.4
Hillside Nozzle in a Cylindrical Shell

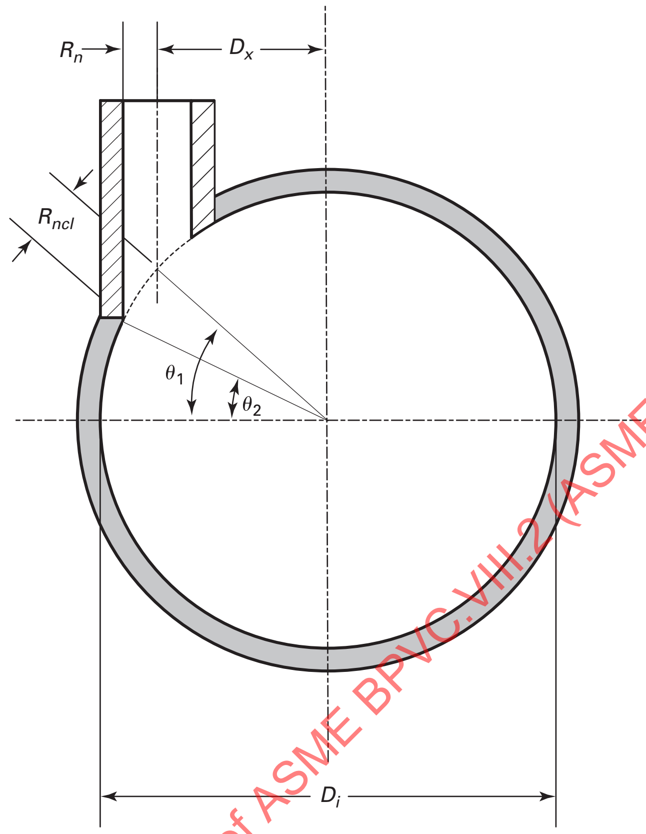


Figure 4.5.5
Nozzle in a Cylindrical Shell Oriented at an Angle From the Longitudinal Axis

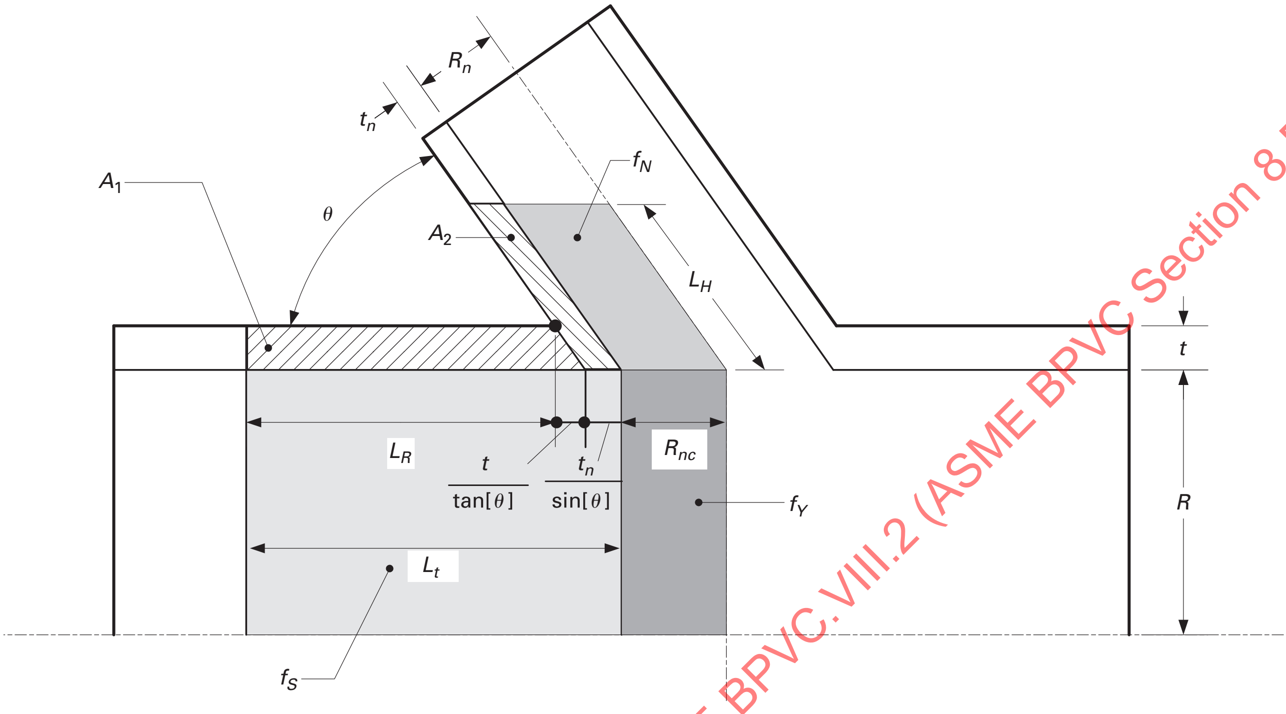


Figure 4.5.6
Radial Nozzle in a Conical Shell

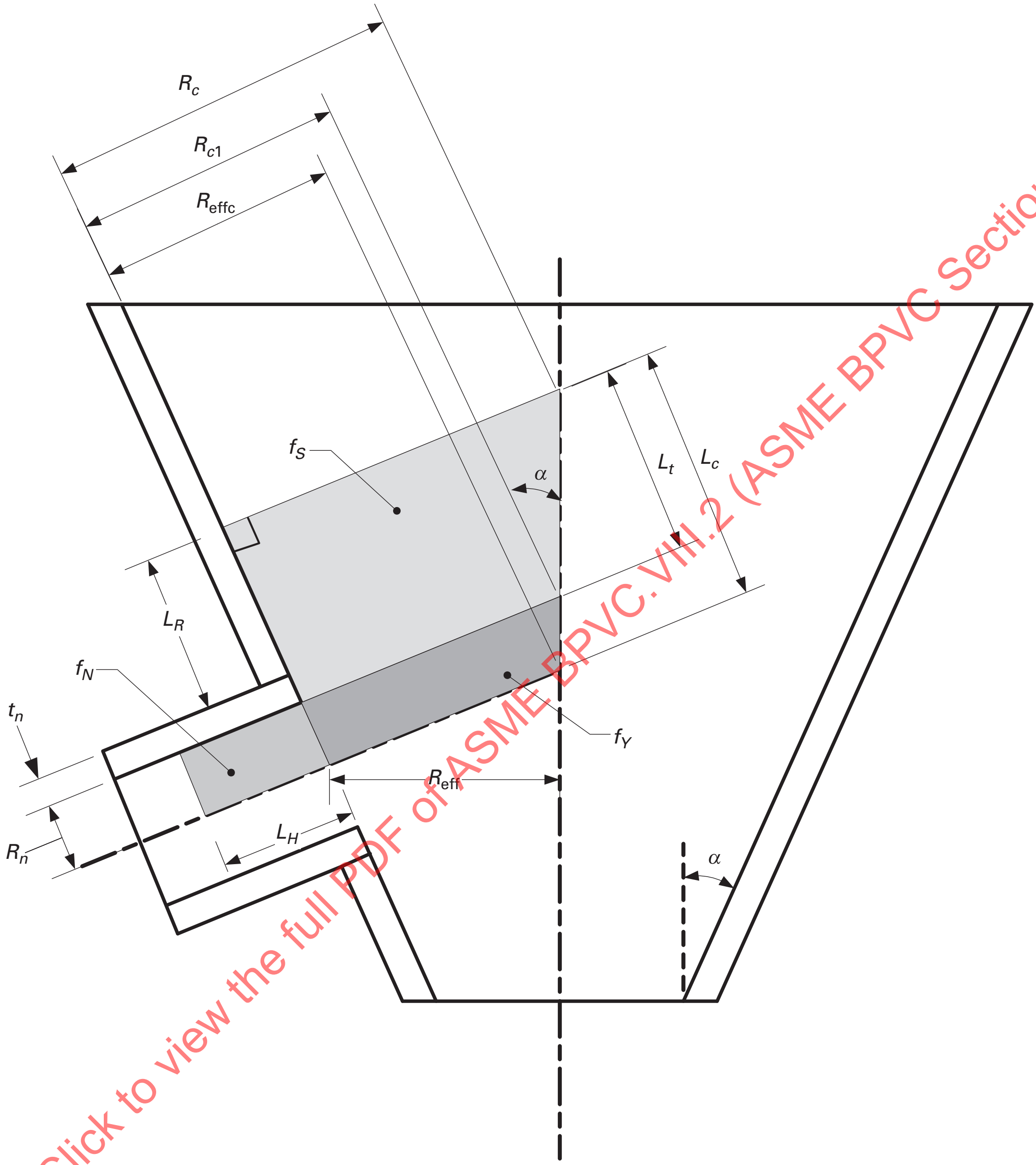


Figure 4.5.7
Nozzle in a Conical Shell Oriented Perpendicular to the Longitudinal Axis

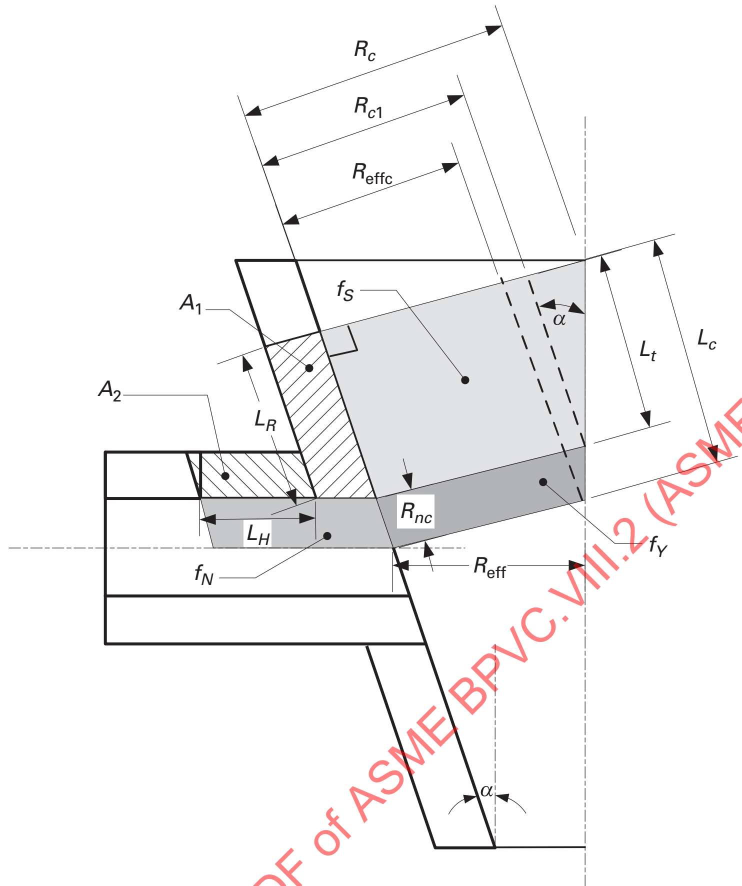


Figure 4.5.8
Nozzle in a Conical Shell Oriented Parallel to the Longitudinal Axis

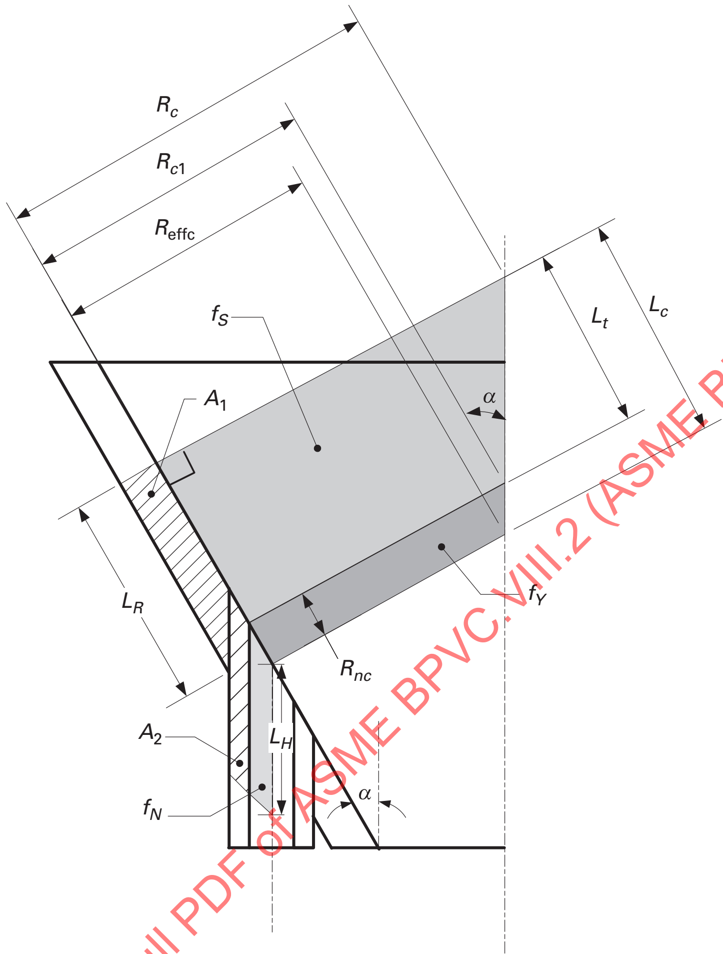


Figure 4.5.9
Radial Nozzle in a Formed Head

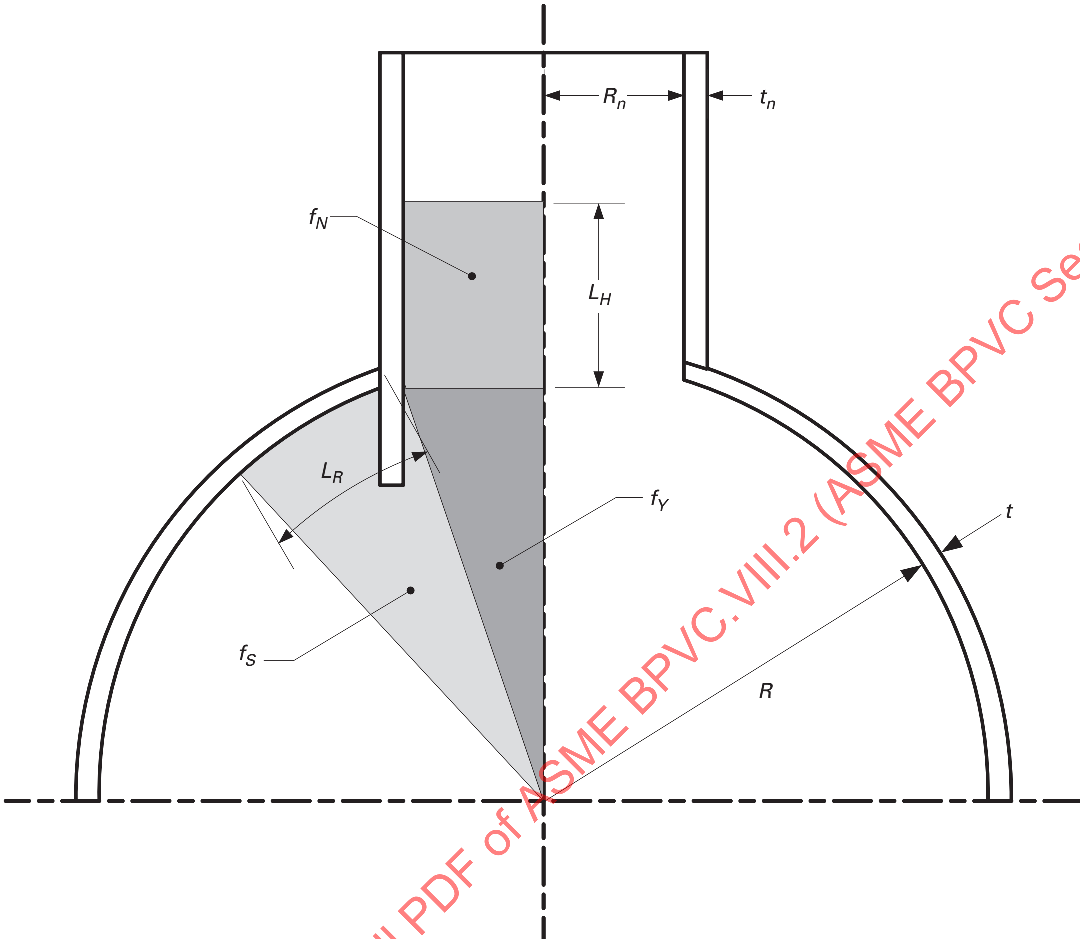
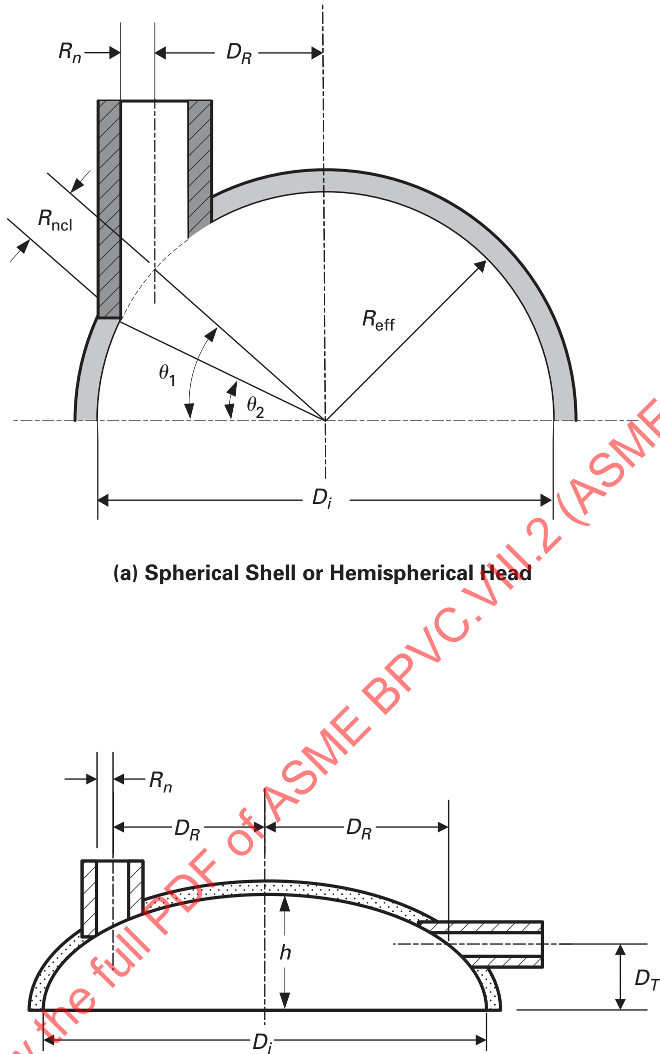


Figure 4.5.10
Hillside or Perpendicular Nozzle in a Spherical Shell or Formed Head



(a) Spherical Shell or Hemispherical Head

(b) Formed Heads Other Than Hemispherical Head

Figure 4.5.11
Example of Two Adjacent Nozzle Openings

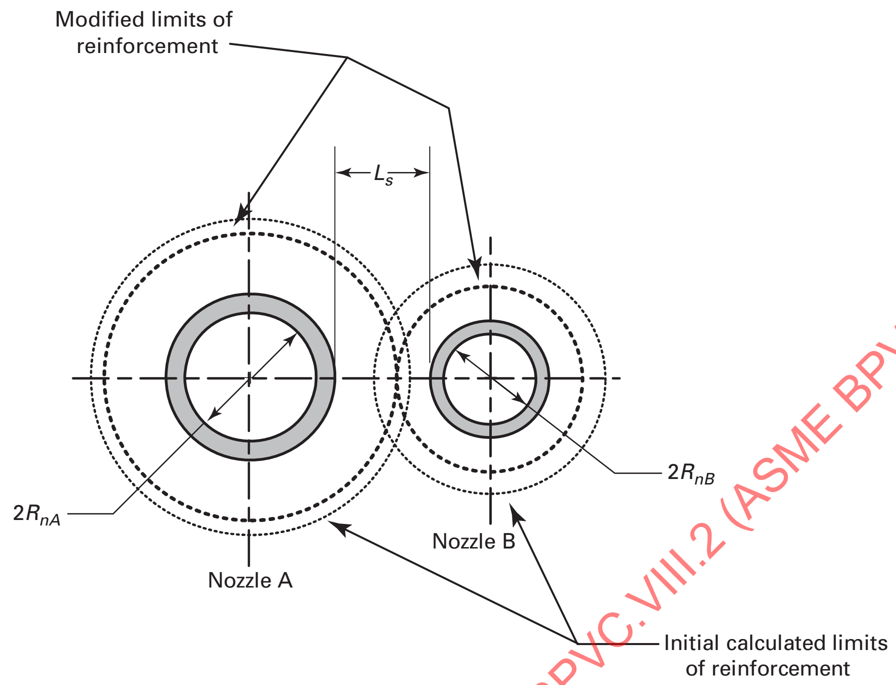


Figure 4.5.12
Example of Three Adjacent Nozzle Openings

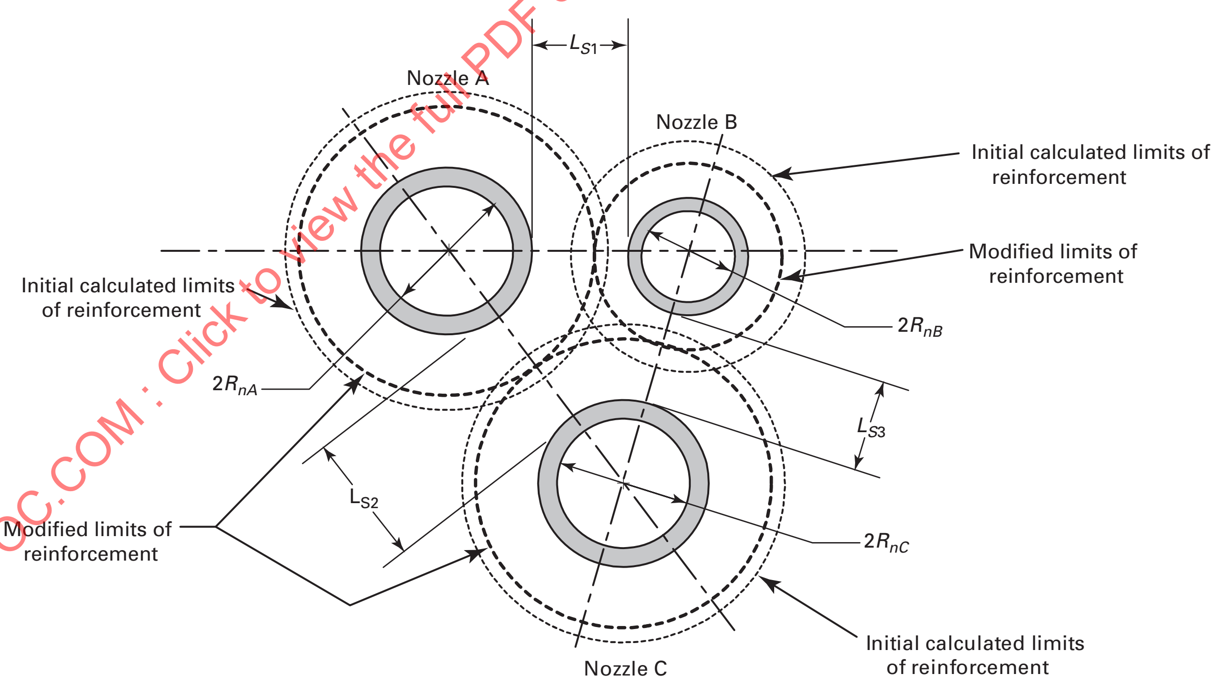


Figure 4.5.13
Metal Area Definition for A_2 With Variable Thickness of Set-in Nozzles

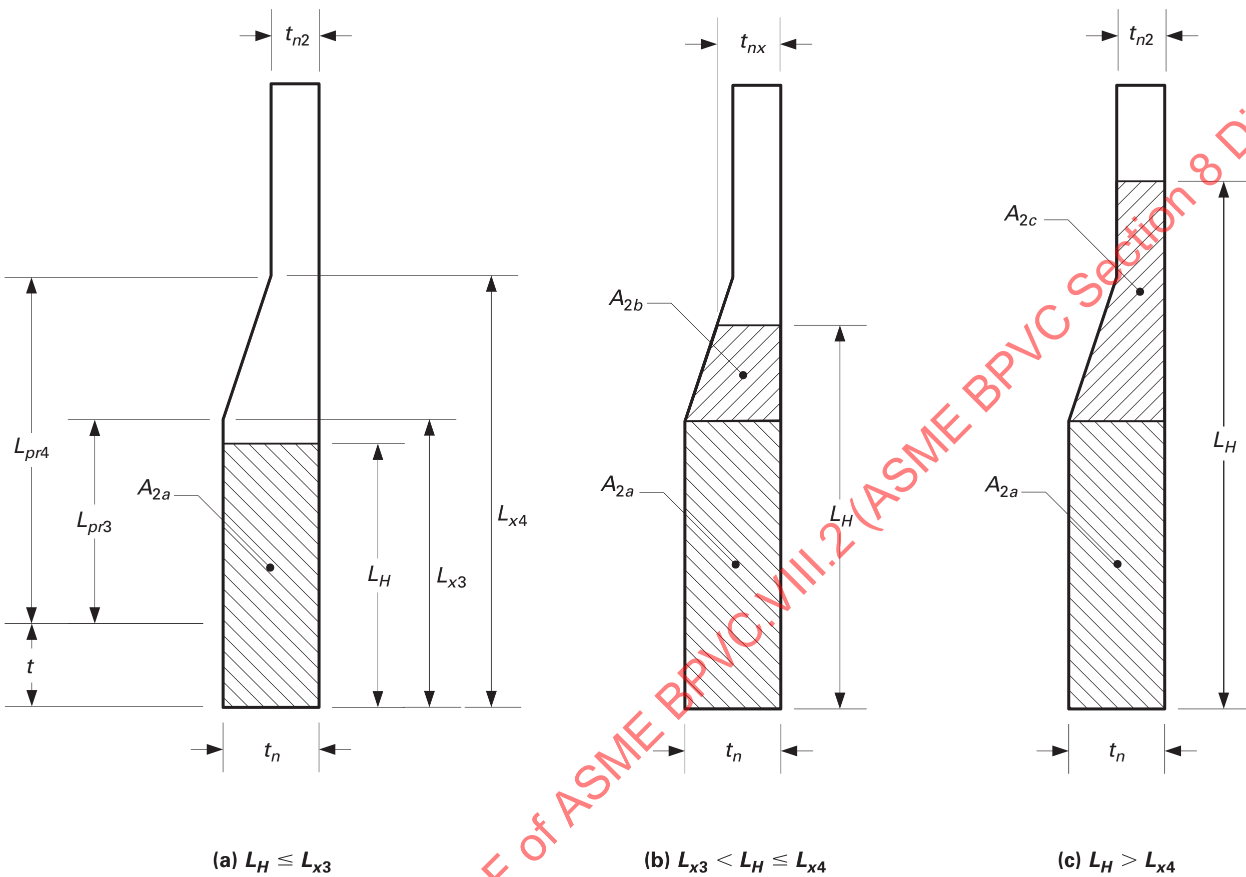
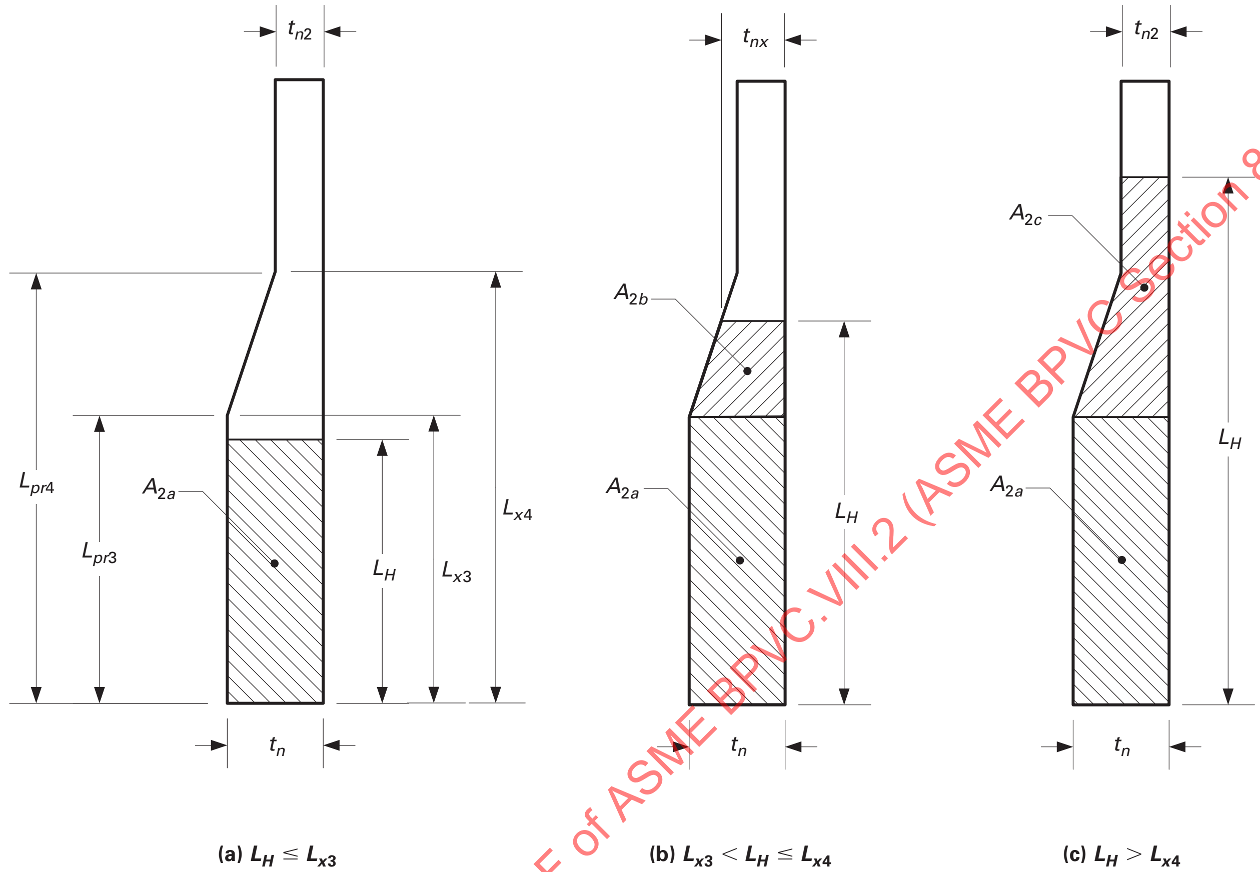


Figure 4.5.14
Metal Area Definition for A_2 With Variable Thickness of Set-on Nozzles



4.6 DESIGN RULES FOR FLAT HEADS

4.6.1 SCOPE

4.6.1.1 The minimum thickness of unstayed flat heads, cover plates and blind flanges shall conform to the requirements given in 4.6. These requirements apply to both circular and noncircular heads and covers. Some acceptable types of flat heads and covers are shown in Table 4.6.1. In this table, the dimensions of the component parts and the dimensions of the welds are exclusive of extra metal required for corrosion allowance.

4.6.1.2 The design methods in this paragraph provide adequate strength for the design pressure. A greater thickness may be necessary if a deflection criterion is required for operation (e.g., leakage at threaded or gasketed joints).

4.6.1.3 For flat head types with a bolted flange connection where the gasket is located inside the bolt circle, calculations shall be made for two design conditions, gasket seating and operating conditions. Details regarding computation of design bolt loads for these two conditions are provided in 4.16.

4.6.2 FLAT UNSTAYED CIRCULAR HEADS

4.6.2.1 Circular blind flanges conforming to any of the flange standards listed in Part 1 and the requirements of 4.1.11 are acceptable for the diameters and pressure–temperature ratings in the respective standard when the blind flange is of the types shown in Table 4.6.1, Detail 7.

4.6.2.2 The minimum required thickness of a flat unstayed circular head or cover that is not attached with bolting that results in an edge moment shall be calculated by the following equation.

$$t = d\sqrt{\frac{CP}{S_{ho}E}} \tag{4.6.1}$$

4.6.2.3 The minimum required thickness of a flat unstayed circular head, cover, or blind flange that is attached with bolting that results in an edge moment (see Table 4.6.1, Detail 7) shall be calculated by the equations shown below. The operating and gasket seating bolt loads, W_o and W_g, and the moment arm of this load, h_G, in these equations shall be computed based on the flange geometry and gasket material as described in 4.16.

$$t = \max\left[t_o, t_g\right] \tag{4.6.2}$$

where

$$t_o = d\sqrt{\frac{CP}{S_{ho}E} + \frac{1.9W_o h_G}{S_{ho}Ed^3}} \tag{4.6.3}$$

$$t_g = d\sqrt{\frac{1.9W_g h_G}{S_{hg}Ed^3}} \tag{4.6.4}$$

4.6.3 FLAT UNSTAYED NONCIRCULAR HEADS

4.6.3.1 The minimum required thickness of a flat unstayed noncircular head or cover that is not attached with bolting that results in an edge moment shall be calculated by the following equation.

$$t = d\sqrt{\frac{ZCP}{S_{ho}E}} \tag{4.6.5}$$

where

$$Z = \min\left[2.5, \left(3.4 - \frac{2.4d}{D}\right)\right] \tag{4.6.6}$$

4.6.3.2 The minimum required thickness of a flat unstayed noncircular head, cover, or blind flange that is attached with bolting that results in an edge moment (see Table 4.6.1, Detail 7) shall be calculated by the equations shown below. The operating and gasket seating bolt loads, W_o and W_g, and the moment arm of this load, h_G, in these equations shall be computed based on the flange geometry and gasket material as described in 4.16.

$$t = \max\left[t_o, t_g\right] \tag{4.6.7}$$

where

$$t_o = d\sqrt{\frac{ZCP}{S_{ho}E} + \frac{6W_o h_G}{S_{ho}ELd^2}} \tag{4.6.8}$$

$$t_g = d\sqrt{\frac{6W_g h_G}{S_{hg}ELd^2}} \tag{4.6.9}$$

The parameter Z is given by eq. (4.6.6).

4.6.4 INTEGRAL FLAT HEAD WITH A CENTRALLY LOCATED OPENING

4.6.4.1 Flat heads which have a single, circular, centrally located opening that exceeds one-half of the head diameter shall be designed in accordance with the rules which follow. A general arrangement of an integral flat head with or without a nozzle attached at the central opening is shown in Figure 4.6.1.

(a) The shell-to-flat head juncture shall be integral, as shown in Table 4.6.1, Details 1, 2, 3, and 4. Alternatively, a butt weld, or a full penetration corner weld similar to the joints shown in Table 4.6.1 Details 5 and 6 may be used.

(b) The central opening in the flat head may have a nozzle that is integral or integrally attached by a full penetration weld, or a nozzle attached by non-integral welds (i.e.,: a double fillet or partial penetration weld, or may have an opening without an attached nozzle or hub. In the case of a nozzle attached by non-integral welds, the head is designed as a head without an attached nozzle or hub.

4.6.4.2 The head thickness does not have to meet the rules in 4.6.2 or 4.6.3. The flat head thickness and other geometry parameters need only satisfy the allowable stress limits in Table 4.6.3.

4.6.4.3 A procedure that can be used to design an integral flat head with a single, circular centrally located opening is shown below.

Step 1. Determine the design pressure and temperature of the flat head opening.

Step 2. Determine the geometry of the flat head opening (see Figure 4.6.1).

Step 3. Calculate the operating moment, M_o, using the following equation.

$$M_o = 0.785 B_n^2 P\left(R + \frac{g_{1n}}{2}\right) + 0.785\left(B_s^2 - B_n^2\right)P\left(\frac{R + g_{1n}}{2}\right) \tag{4.6.10}$$

where

$$R = \frac{B_s - B_n}{2} - g_{1n} \tag{4.6.11}$$

Step 4. Calculate F, V, and f based on B_n, g_{1n}, g_{0n} and h_n using the equations in Table 4.16.4 and Table 4.16.5, designate the resulting values as F_n, V_n, and f_n.

Step 5. Calculate F, V, and f based on B_s, g_{1s}, g_{0s} and h_s using the equations in Table 4.16.4 and Table 4.16.5, designate the resulting values as F_s, V_s, and f_s.

Step 6. Calculate Y, T, U, Z, L, e, and d based on $K = A/B_n$ using the equations in Table 4.16.4.

Step 7. Calculate the quantity $(E\theta)^*$ using one of the following equations:

For an opening with an integrally attached nozzle:

$$(E\theta)^* = \frac{0.91\left(\frac{g_{1n}}{g_{on}}\right)^2 \left(B_n + g_{0n}\right)V_n}{f_n\sqrt{B_n g_{0n}}} S_H \tag{4.6.12}$$

Where S_H is evaluated using the equation in Table 4.6.2.

For an opening without an attached nozzle or with a nozzle or hub attached with non-integral welds:

$$(E\theta)^* = \frac{B_n S_T}{t} \tag{4.6.13}$$

Where S_T is evaluated using the equation in Table 4.6.2.

Step 8. Calculate the quantity M_H using the following equation:

$$M_H = \frac{(E\theta)^*}{\dfrac{1.74 V_s\sqrt{B_s g_{0s}}}{g_{0s}^3\left(B_s + g_{0s}\right)} + \dfrac{(E\theta)^*}{M_o}\left(1 + \dfrac{F_s t}{\sqrt{B_s g_{0s}}}\right)} \tag{4.6.14}$$

Step 9. Calculate the quantity X_1 using the following equation:

$$X_1 = \frac{M_o - M_H\left(1 + \dfrac{F_s t}{\sqrt{B_s g_{0s}}}\right)}{M_o} \tag{4.6.15}$$

Step 10. Calculate the stresses at the shell-to-flat head junction and opening-to-flat-head junction using Table 4.6.2.

Step 11. Check the flange stress acceptance criteria in Table 4.6.3. If the stress criteria are satisfied, then the design is complete. If the stress criteria are not satisfied, then re-proportion the flat head and/or opening dimensions and go to Step 3.

4.6.5 NOMENCLATURE

A = shell outside diameter.
B_s = inside diameter of the shell.
B_n = inside diameter of the opening.
C = factor depending upon the method of attachment of head, shell dimensions, and other items as described in Table 4.6.1.
D = is the long span of noncircular heads or covers measured perpendicular to short span.
d = diameter, or short span, measured as indicated in figure shown in Table 4.6.1.
E = joint factor.
e = flange stress factor.
f_n = hub stress correction factor for the nozzle opening-to-flat head junction.
f_s = hub stress correction factor the shell-to-flat head junction
E = weld joint factor (see 4.2).
F_n = flange stress factor for the nozzle opening-to-flat head junction.
F_s = flange stress factor for the shell-to-flat head junction
g_{1s} = hub thickness at the large end of the shell-to-flat head junction.
g_{0n} = hub thickness at the small end of the nozzle opening-to-flat head junction.
g_{0s} = hub thickness at the small end of the shell-to-flat head junction.
g_{1n} = hub thickness at the large end of the nozzle opening-to-flat head junction.
g_{1s} = hub thickness at the large end of the shell-to-flat head junction.
h_G = gasket moment arm (see Table 4.16.6).
h_n = hub length at the large end of the nozzle opening-to-flat head junction.
h_s = hub length at the large end of the shell-to-flat head junction.
L = perimeter of a noncircular bolted head measured along the centers of the bolt holes, or the flange stress factor, as applicable.
M_o = operating moment.
M_H = moment acting at the shell-to-flat head junction.
m = thickness ratio t_r/t_s.
P = internal design pressure.
r = inside corner radius on a head formed by flanging or forging.
S_{ho} = allowable stress from Annex 3-A for the head evaluated at the design temperature.
S_{hg} = allowable stress from Annex 3-A for the head evaluated at the gasket seating condition.
T = flange stress factor.
t = minimum required thickness of the flat head or cover.
t_g = required thickness of the flat head or cover for the gasket seating condition.
t_o = required thickness of the flat head or cover for the design operating condition.
t_f = nominal thickness of the flange on a forged head at the large end.
t_h = nominal thickness of the flat head or cover.
t_r = required thickness of a seamless shell.
t_s = nominal thickness of the shell.
t_1 = throat dimension of the closure weld
U = flange stress factor.
V_n = flange stress factor for the nozzle opening-to-flat head junction.
V_s = flange stress factor for the shell-to-flat head junction
W_o = operating bolt load at the design operating condition.
W_g = gasket seating bolt load at the design gasket seating condition.
Y = length of the flange of a flanged head, measured from the tangent line of knuckle, or the flange stress factor, as applicable.
Z = factor for noncircular heads and covers that depends on the ratio of short span to long span, or the flange stress factor, as applicable.
Z_1 = integral flat head stress parameter.

$(E\theta)^*$ = slope of head with central opening or nozzle times the modulus of elasticity, disregarding the interaction of the integral shell at the outside diameter of the head.

4.6.6 TABLES

Table 4.6.1
C Parameter for Flat Head Designs

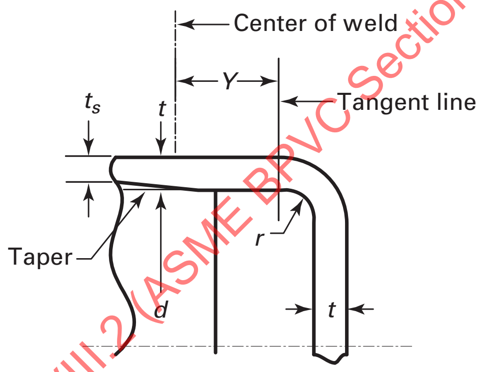
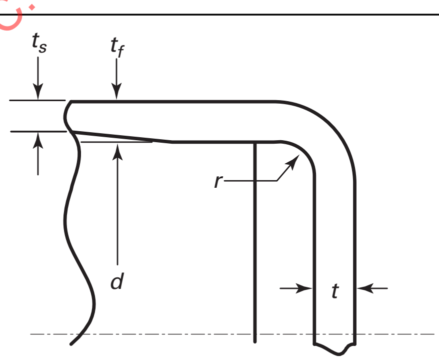
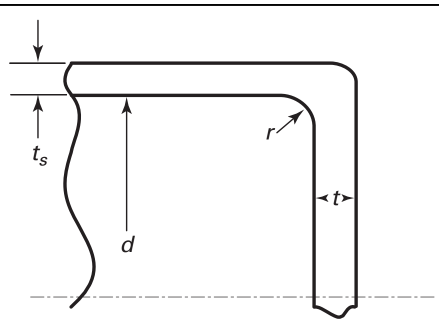
Detail	Requirements	Figure
1	• C = 0.17 for flanged circular and noncircular heads forged integral with or butt welded to the vessel with an inside corner radius not less than three times the required head thickness, with no special requirement with regard to length of flange. • C = 0.10 for circular heads, when the flange length for heads of the above design is not less than: $$Y = \left(1.1 - 0.8\left(\frac{t_s}{t_h}\right)^2\right)\sqrt{dt_h}$$ • C = 0.10 for circular heads, when the flange length Y less than the requirements in the above equation but the shell thickness is not less than: $t_s = 1.12t_h\sqrt{1.1 - Y/\sqrt{dt_h}}$ for a length of at least $2\sqrt{dt_s}$. When C = 0.10 is used, the taper shall be at least 1:3. • $r = 3t$ minimum shall be used	
2	• C = 0.17 for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than two times the shell thickness, the corner radius on the inside is not less than three times the flange thickness. • $r = 3t_f$ minimum shall be used	
3	• C = max[0.33m, 0.20] for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than the shell thickness, the corner radius on the inside is not less than the following: $$r = 10\ mm\ (0.375\ in.)\quad for\ t_s \le 38\ mm\ (1.5\ in.)$$ $$r = \min\left[0.25t_s,\ 19\ mm\ (0.75\ in.)\right]\quad for\ t_s > 38\ mm\ (1.5\ in.)$$	

Table 4.6.1
C Parameter for Flat Head Designs (Cont'd)

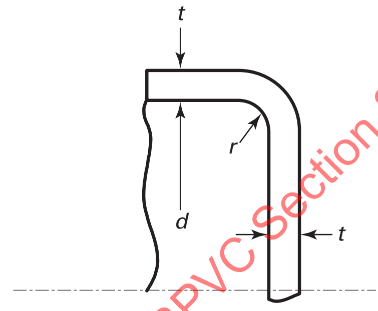
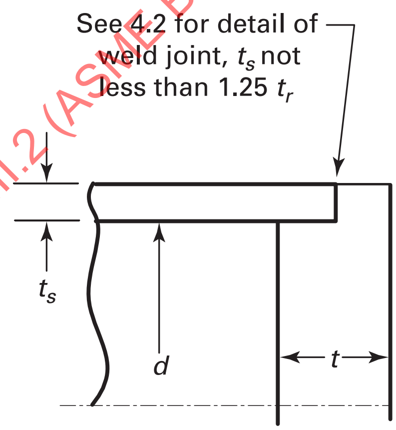
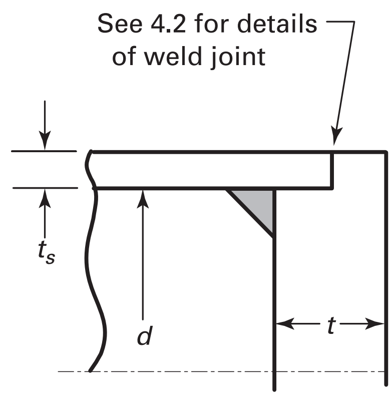
Detail	Requirements	Figure
4	• C = 0.13 for integral flat circular heads when: – the dimension d does not exceed 610 mm (24 in.) – the ratio of thickness of the head to the dimension d is not less than 0.05 or greater than 0.25 – the head thickness t_h is not less than the shell thickness t_s – the inside corner radius is not less than 0.25t – the construction is obtained by special techniques of upsetting and spinning the end of the shell, such as employed in closing header ends. • r = 3t minimum shall be used	
5	C = 0.33 for circular plates welded to the end of the shell when t_s is at least 1.25t_r and the weld details conform to the requirements of 4.2.	
6	C = max[0.33m, 0.20] for circular plates if an inside fillet weld with minimum throat thickness of 0.7t_s is used and the details of the outside weld conform to the requirements of 4.2.	

Table 4.6.1
C Parameter for Flat Head Designs (Cont'd)

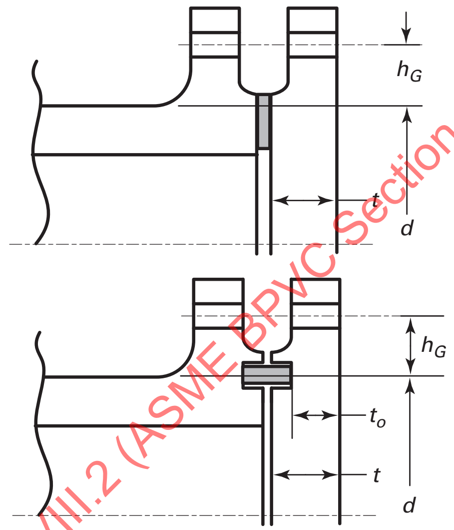
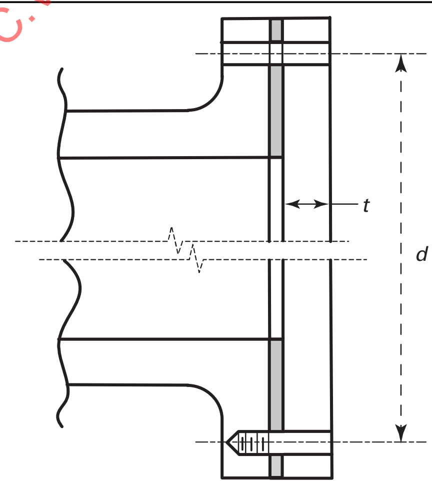
Detail	Requirements	Figure
7	• C = 0.3 for circular and noncircular heads and covers bolted to the vessel as indicated in the figures. • When the cover plate is grooved for a peripheral gasket, the net cover plate thickness under the groove or between the groove and the outer edge of the cover plate shall be not less than the following thickness. For circular heads and covers: $t_o = d\sqrt{\dfrac{1.9W_o h_G}{S_{ho} d^3}}$ For noncircular heads and covers: $t_o = d\sqrt{\dfrac{6W_o h_G}{S_{ho} L d^2}}$	
8	C = 0.25 for circular covers bolted with a full-face gasket to shells and flanges.	

Table 4.6.1
C Parameter for Flat Head Designs (Cont'd)

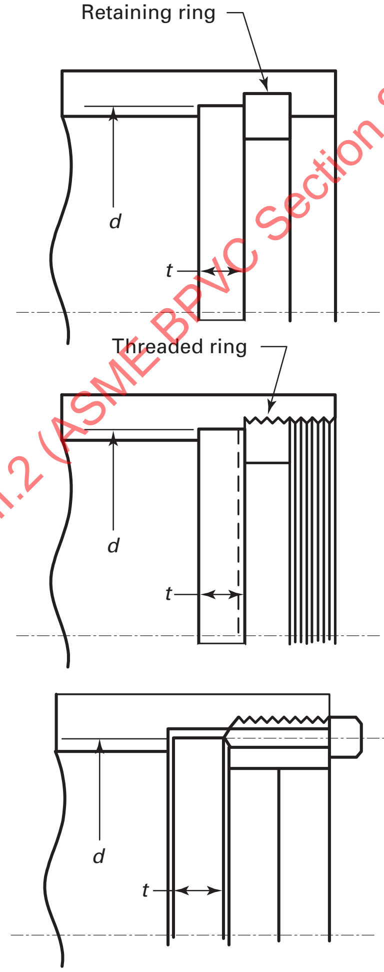
Detail	Requirements	Figure
9	C = 0.3 for a circular plate inserted into the end of a vessel and held in place by a positive mechanical locking arrangement when all possible means of failure (either by shear, tension, compression, or radial deformation, including flaring, resulting from pressure and differential thermal expansion) are resisted with a design factor of at least four. Seal welding may be used, if desired.	

Table 4.6.2
Junction Stress Equations for an Integral Flat Head With Opening

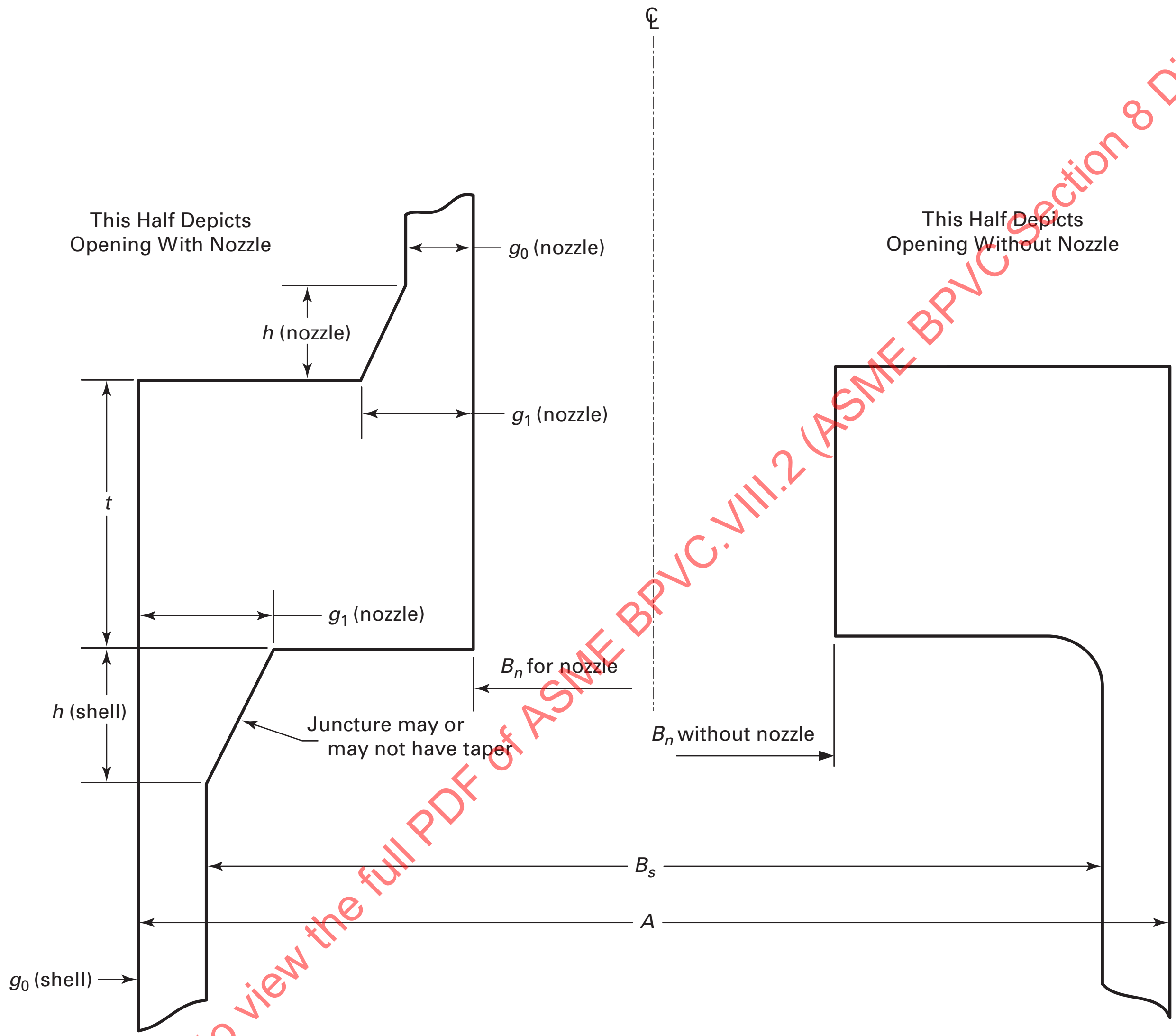
Head/Shell Junction Stresses	Opening/Head Junction Stresses
$S_{HS} = \dfrac{1.1 f_s X_1 (E\theta)^* \sqrt{B_s g_{0s}}}{\left(\dfrac{g_{1s}}{g_{0s}}\right)^2 B_s V_s}$ $S_{RS} = \dfrac{1.91 M_H \left(1 + \dfrac{F_s t}{\sqrt{B_s g_{0s}}}\right)}{B_s t^2} + \dfrac{0.64 F_s M_H}{B_s \sqrt{B_s g_{0s}}\, t}$ $S_{TS} = \dfrac{X_1 (E\theta)^* t}{B_s} - \dfrac{0.57 M_H \left(1 + \dfrac{F_s t}{\sqrt{B_s g_{0s}}}\right)}{B_s t^2} + \dfrac{0.64 Z F_s M_H}{B_s \sqrt{B_s g_{0s}}\, t}$	$S_{HO} = X_1 S_H$ $S_{RO} = X_1 S_R$ $S_{TO} = X_1 S_T + \dfrac{0.64 Z_1 F_s M_H}{B_s \sqrt{B_s g_{0s}}\, t}$ where $S_H = \dfrac{f_n M_o}{L g_{1n}^2 B_n}$ $S_R = \dfrac{(1.33te + 1) M_o}{L t^2 B_n}$ $S_T = \dfrac{Y M_o}{t^2 B_n} - Z S_R$ $Z_1 = \dfrac{2K^2}{K^2 - 1}$ Note: $S_R = S_H = 0.0$ for the case of an opening without a nozzle

Table 4.6.3
Stress Acceptance Criteria for an Integral Flat Head With Opening

Head/Shell Junction Stresses	Opening/Head Junction Stresses
$S_{HS} \le 1.5 S_{ho}$	$S_{HO} \le 1.5 S_{ho}$
$S_{RS} \le S_{ho}$	$S_{RO} \le S_{ho}$
$S_{TS} \le S_{ho}$	$S_{TO} \le S_{ho}$
$\dfrac{(S_{HS} + S_{RS})}{2} \le S_{ho}$	$\dfrac{(S_{HO} + S_{RO})}{2} \le S_{ho}$
$\dfrac{(S_{HS} + S_{TS})}{2} \le S_{ho}$	$\dfrac{(S_{HO} + S_{TO})}{2} \le S_{ho}$

4.6.7 FIGURES

Figure 4.6.1
Integral Flat Head With a Large Central Opening



4.7 DESIGN RULES FOR SPHERICALLY DISHED BOLTED COVERS

4.7.1 SCOPE

4.7.1.1 Design rules for four configurations of circular spherically dished heads with bolting flanges are provided in 4.7. The four head types are shown in Figures 4.7.1, 4.7.2, 4.7.3, and 4.7.4. The design rules cover both internal and external pressure, pressure that is concave and convex to the spherical head, respectively. The maximum value of the pressure differential shall be used in all of the equations.

4.7.1.2 For head types with a bolted flange connection where the gasket is located inside the bolt circle, calculations shall be made for two design conditions, gasket seating and operating conditions. Details regarding computation of design bolt loads and flange moments for these two conditions are provided in 4.16. If a flange moment is computed as a negative number, the absolute value of this moment shall be used in all of the equations.

4.7.1.3 Calculations shall be performed using dimensions in the corroded condition and the uncorroded condition, and the more severe case shall control.

(**21**) **4.7.1.4** Flanges designed to Figures 4.7.2 through 4.7.5 do not need to comply with the flange rigidity criterion in Table 4.16.10.

4.7.2 TYPE A HEAD THICKNESS REQUIREMENTS

4.7.2.1 The thickness of the head and skirt for a Type A Head Configuration (see Figure 4.7.1) shall be determined in accordance with the rules in 4.3 for internal pressure (pressure on the concave side), and 4.4 for external pressure (for pressure on the convex side). The skirt thickness shall be determined using the appropriate equation for cylindrical shells. The head radius, L, and knuckle radius, r, shall comply with the limitations given in these paragraphs.

4.7.2.2 The flange thickness of the head for a Type A Head Configuration shall be determined in accordance with the rules of 4.16. When a slip-on flange conforming to the standards listed in Table 1.1 is used, design calculations per 4.16 need not be done, provided the design pressure–temperature is within the pressure–temperature rating permitted in the flange standard.

4.7.2.3 Detail (a) in Figure 4.7.1 is permitted if both of the following requirements are satisfied.

(a) The material of construction satisfies the following equation.

$$\frac{S_{yT}}{S_u} \leq 0.625 \tag{4.7.1}$$

(b) The component is not in cyclic service, i.e., a fatigue analysis is not required in accordance with 4.1.1.4.

4.7.3 TYPE B HEAD THICKNESS REQUIREMENTS

4.7.3.1 The thickness of the head for a Type B Head Configuration (see Figure 4.7.2) shall be determined by the following equations.

(a) Internal pressure (pressure on the concave side)

$$t = \frac{5PL}{6S} \tag{4.7.2}$$

(b) External pressure (pressure on the convex side) - the head thickness shall be determined in accordance with the rules in 4.4.

4.7.3.2 The flange thickness of the head for a Type B Head Configuration shall be determined by the following equations where the flange moments M_o and M_g for the operating and gasket seating conditions, respectively, are determined from 4.16.

(a) Flange thickness for a ring gasket

$$T = \max\left[T_g, T_o\right] \tag{4.7.3}$$

where

$$T_g = \sqrt{\frac{M_g}{S_{fg}B}\left(\frac{A+B}{A-B}\right)} \tag{4.7.4}$$

$$T_o = \sqrt{\frac{M_o}{S_{fo}B}\left(\frac{A+B}{A-B}\right)} \tag{4.7.5}$$

(b) Flange thickness for a full face gasket

$$T = 0.6\sqrt{\frac{|P|}{S_{fo}}\left(\frac{B(A+B)(C-B)}{A-B}\right)} \tag{4.7.6}$$

4.7.3.3 A Type B head may only be used if both of the requirements in 4.7.2.3 are satisfied.

4.7.4 TYPE C HEAD THICKNESS REQUIREMENTS

4.7.4.1 The thickness of the head for a Type C Head Configuration (see Figure 4.7.3) shall be determined by the following equations.

(a) Internal pressure (pressure on the concave side) - the head thickness shall be determined using eq. (4.7.2).

(b) External pressure (pressure on the convex side) - the head thickness shall be determined in accordance with the rules in 4.4.

4.7.4.2 The flange thickness of the head for a Type C Head Configuration shall be determined by the following equations where the flange moments M_o and M_g for the operating and gasket seating conditions, respectively, are determined from 4.16.

(a) Flange thickness for a ring gasket for heads with round bolting holes

$$T = \max\left[T_g, T_o\right] \tag{4.7.7}$$

where

$$T_g = \sqrt{\frac{1.875M_g}{S_{fg}B}\left(\frac{C+B}{7C-5B}\right)} \tag{4.7.8}$$

$$T_o = Q + \sqrt{\frac{1.875M_o}{S_{fo}B}\left(\frac{C+B}{7C-5B}\right)} \tag{4.7.9}$$

$$Q = \frac{|P|L}{4S_{fo}}\left(\frac{C+B}{7C-5B}\right) \tag{4.7.10}$$

(b) Flange thickness for ring gasket for heads with bolting holes slotted through the edge of the head

$$T = \max\left[T_g, T_o\right] \tag{4.7.11}$$

where

$$T_g = \sqrt{\frac{1.875M_g}{S_{fg}B}\left(\frac{C+B}{3C-B}\right)} \tag{4.7.12}$$

$$T_o = Q + \sqrt{\frac{1.875M_o}{S_{fo}B}\left(\frac{C+B}{3C-B}\right)} \tag{4.7.13}$$

$$Q = \frac{|P|L}{4S_{fo}}\left(\frac{C+B}{3C-B}\right) \tag{4.7.14}$$

(c) Flange thickness for full-face gasket for heads with round bolting holes

$$T = Q + \sqrt{Q^2 + \frac{3BQ(C-B)}{L}} \tag{4.7.15}$$

The parameter Q is given by eq. (4.7.10).

(d) Flange thickness for full-face gasket for heads with bolting holes slotted through the edge of the head

$$T = Q + \sqrt{Q^2 + \frac{3BQ(C-B)}{L}} \tag{4.7.16}$$

The parameter Q is given by eq. (4.7.14).

4.7.5 TYPE D HEAD THICKNESS REQUIREMENTS

4.7.5.1 The thickness of the head for a Type D Head Configuration (see Figure 4.7.4) shall be determined by the following equations.

(a) Internal pressure (pressure on the concave side) - the head thickness shall be determined using eq. (4.7.2).

(b) External pressure (pressure on the convex side) - the head thickness shall be determined in accordance with the rules in 4.4.

4.7.5.2 The flange thickness of the head for a Type D Head Configuration shall be determined by the following equations.

$$T = \max\left[T_g, T_o\right] \tag{4.7.17}$$

where

$$T_g = \sqrt{\frac{M_g}{S_{fg}B} + \left(\frac{A+B}{A-B}\right)} \tag{4.7.18}$$

$$T_o = Q + \sqrt{Q^2 + \frac{M_o}{S_{fo}B} + \left(\frac{A+B}{A-B}\right)} \tag{4.7.19}$$

$$Q = \frac{|P|B\sqrt{4L^2 - B^2}}{8S_{fo}(A-B)} \tag{4.7.20}$$

When determining the flange design moment for the design condition, M_o, using 4.16, the following modifications shall be made. The moment arm, h_D, shall be computed using eq. (4.7.21). An additional moment term, M_r, computed using eq. (4.7.22) shall be added to M_o as defined 4.16. The term M_{oe} in the equation for M_o as defined 4.16 shall be set to zero in this calculation. Note that this term may be positive or negative depending on the orientation of t_v, R, A_R.

$$h_D = 0.5(C-B) \tag{4.7.21}$$

$$M_r = \left(0.785B^2P\cot[\beta]\right)h_r \tag{4.7.22}$$

where

$$\beta = \arcsin\left[\frac{B}{2L + t}\right] \tag{4.7.23}$$

4.7.5.3 As an alternative to the rules in 4.7.5.1 and 4.7.5.2, the following procedure can be used to determine the required head and flange thickness of a Type D head. This procedure accounts for the continuity between the flange ring and the head, and represents a more accurate method of analysis. (**21**)

Step 1. Determine the design pressure and temperature of the flange joint. If the pressure is negative, a negative value must be used for P in all of the equations of this procedure, and

$$P_e = 0.0 \qquad \text{for internal pressure} \tag{4.7.24}$$

$$P_e = P \qquad \text{for external pressure} \tag{4.7.25}$$

Step 2. Determine an initial Type D head configuration geometry (see Figure 4.7.5). The following geometry parameters are required:

(a) The flange bore, B
(b) The bolt circle diameter, C
(c) The outside diameter of the flange, A
(d) Flange thickness, T
(e) Mean head radius, R
(f) Head thickness, t
(g) Inside depth of flange to the base of the head, q

Step 3. Select a gasket configuration and determine the location of the gasket reaction, G, and the design bolt loads for the gasket seating, W_g, and operating conditions, W_o, using the rules of 4.16.

Step 4. Determine the geometry parameters.

$$h_1 = \frac{(C - G)}{2} \tag{4.7.26}$$

$$h_2 = \frac{(G - B)}{2} \tag{4.7.27}$$

$$d = \frac{(A - B)}{2} \tag{4.7.28}$$

$$n = \frac{T}{t} \tag{4.7.29}$$

$$K = \frac{A}{B} \tag{4.7.30}$$

$$\phi = \arcsin\left[\frac{B}{2R}\right] \tag{4.7.31}$$

$$e = q - \frac{1}{2}\left(T - \frac{t}{\cos[\phi]}\right) \tag{4.7.32}$$

$$k_1 = 1 - \left(\frac{1 - 2\nu}{2\lambda}\right)\cot[\phi] \tag{4.7.33}$$

$$k_2 = 1 - \left(\frac{1 + 2\nu}{2\lambda}\right)\cot[\phi] \tag{4.7.34}$$

$$\lambda = \left[3\left(1 - \nu^2\right)\left(\frac{R}{t}\right)^2\right]^{0.25} \tag{4.7.35}$$

Step 5. Determine the shell discontinuity geometry factors.

$$C_1 = \frac{0.275n^3 t \cdot \ln[K]}{k_1} - e \tag{4.7.36}$$

$$C_2 = \frac{1.1\lambda n^3 t \cdot \ln[K]}{Bk_1} + 1 \tag{4.7.37}$$

$$C_4 = \frac{\lambda \sin[\phi]}{2}\left(k_2 + \frac{1}{k_1}\right) + \frac{B}{4nd} + \frac{1.65e}{tk_1} \tag{4.7.38}$$

$$C_5 = \frac{1.65}{tk_1}\left(1 + \frac{4\lambda e}{B}\right) \tag{4.7.39}$$

Step 6. Determine the shell discontinuity load factors for the operating and gasket conditions.

$$C_{3o} = \frac{\pi B^2 P}{4}\left(e \cdot \cot[\phi] + \frac{2q(T - q)}{B} - h_2\right) - W_o h_1 \tag{4.7.40}$$

$$C_{6o} = \frac{\pi B^2 P}{4}\left(\frac{4q - B \cdot \cot[\phi]}{4nd} - \frac{0.35}{\sin[\phi]}\right) \tag{4.7.41}$$

$$C_{3g} = -W_g h_1 \tag{4.7.42}$$

$$C_{6g} = 0.0 \tag{4.7.43}$$

Step 7. Determine the shell discontinuity force and moment for the operating and gasket conditions.

$$V_{do} = \frac{C_2 C_{6o} - C_{3o} C_5}{C_2 C_4 - C_1 C_5} \tag{4.7.44}$$

$$M_{do} = \frac{C_1 C_{6o} - C_{3o} C_4}{C_2 C_4 - C_1 C_5} \tag{4.7.45}$$

$$V_{dg} = \frac{C_2 C_{6g} - C_{3g} C_5}{C_2 C_4 - C_1 C_5} \tag{4.7.46}$$

$$M_{dg} = \frac{C_1 C_{6g} - C_{3g} C_4}{C_2 C_4 - C_1 C_5} \tag{4.7.47}$$

Step 8. Calculate the stresses in the head and at the head-to-flange junction using Table 4.7.1 and check the stress acceptance criteria. If the stress criteria are satisfied, then the design is complete. If the stress criteria are not satisfied, then re-proportion the bolted head dimensions and go to Step 3.

4.7.6 NOMENCLATURE

A = flange outside diameter.
B = flange inside diameter.
β = angle formed by the tangent to the center line of the dished cover thickness at its point of intersection with the flange ring, and a line perpendicular to the axis of the dished cover
C = bolt circle diameter.
C_1 = shell discontinuity geometry parameter for the Type D head alternative design procedure.

C_2 = shell discontinuity geometry parameter for the Type D head alternative design procedure.
C_{3g} = shell discontinuity load factor for the gasket seating condition for the Type D head alternative design procedure.
C_{3o} = shell discontinuity load factor for the design operating condition for the Type D head alternative design procedure.
C_4 = shell discontinuity geometry parameter for the Type D head alternative design procedure.
C_5 = shell discontinuity geometry parameter for the Type D head alternative design procedure.
C_{6g} = shell discontinuity load factor for the gasket seating condition for the Type D head alternative design procedure.
C_{6o} = shell discontinuity load factor for the design operating condition for the Type D head alternative design procedure.
e = geometry parameter for the Type D head alternative design procedure.
h_r = moment arm of the head reaction force.
h_1 = geometry parameter for the Type D head alternative design procedure.
h_2 = geometry parameter for the Type D head alternative design procedure.
k_1 = geometry parameter for the Type D head alternative design procedure.
k_2 = geometry parameter for the Type D head alternative design procedure.
K = geometry parameter for the Type D head alternative design procedure.
L = inside crown radius.
λ = geometry parameter for the Type D head alternative design procedure.
M_{dg} = shell discontinuity moment for the gasket seating condition.
M_{do} = shell discontinuity moment for design operating condition.
M_g = flange design moment for the gasket seating condition determined using 4.16.
M_o = flange design moment for the design condition determined using 4.16 (see 4.7.5.2 for exception)
M_r = moment from the head reaction force.
n = geometry parameter for the Type D head alternative design procedure.
ν = Poisson's ratio.
P = design pressure.
P_e = pressure factor to adjust the design rules for external pressure.
ϕ = one-half central angle of the head for the Type D head alternative design procedure.
q = inside depth of the flange to the base of the head.
R = mean radius of a Type D head.
r = inside knuckle radius. S_{fg} allowable stress from Annex 3-A for the flange evaluated at the gasket seating condition.
S_{fm} = membrane stress in the flange.
S_{fmbi} = membrane plus bending stress on the inside surface of the flange.
S_{fmbo} = membrane plus bending stress on the outside surface of the flange.
S_{fo} = allowable stress from Annex 3-A for the flange evaluated at the design temperature.
S_{hb} = bending stress at the head-to-flange junction.
S_{hg} = allowable stress from Annex 3-A for the head evaluated at the gasket seating condition.
S_{hm} = head membrane stress.
S_{hl} = local membrane stress at the head-to-flange junction.
S_{hlbi} = local membrane plus bending stress at the head-to-flange junction on the inside surface of the head.
S_{hlbo} = local membrane plus bending stress at the head-to-flange junction on the outside surface of the head.
S_{ho} = allowable stress from Annex 3-A for the head evaluated at the design temperature.
S_{yT} = yield strength from Annex 3-D evaluated at the design temperature.
S_u = minimum specified ultimate tensile strength from Annex 3-D.
T = flange thickness.
T^* = flange thickness for a Type C Head.
T_g = required flange thickness for the gasket seating condition.
T_o = required flange thickness for design operating condition.
t = required head thickness.
V_{dg} = shell discontinuity shear force for the gasket seating condition.
V_{do} = shell discontinuity shear force for design operating condition.
W_g = bolt load for the gasket seating condition.
W_o = bolt load for design operating condition.

4.7.7 TABLES

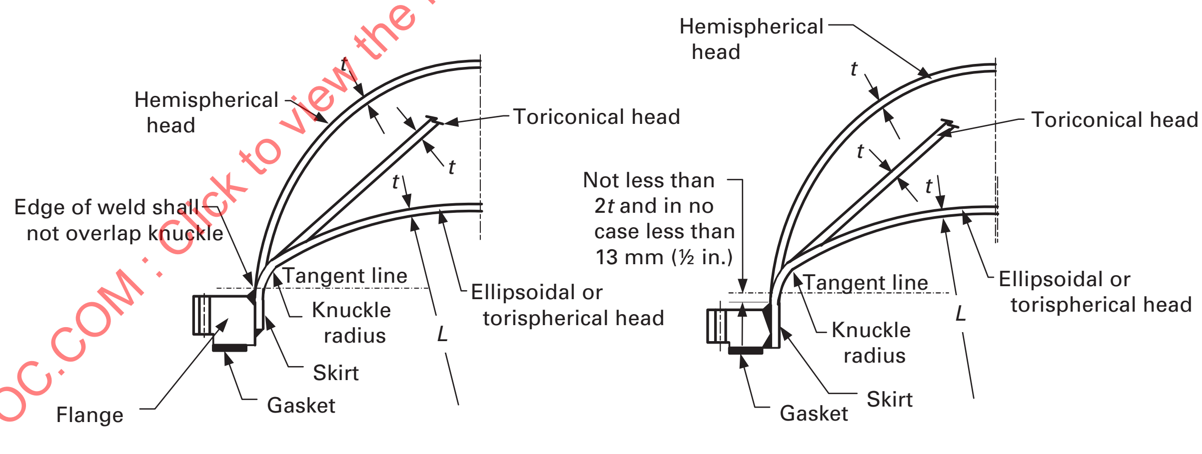
(21)

Table 4.7.1
Junction Stress Equations and Acceptance Criteria for a Type D Head

Operating Conditions	Gasket Seating Conditions
$S_{hm} = \dfrac{PR}{2t} + P_e$	$S_{hm} = 0.0$
$S_{hl} = \dfrac{PR}{2t} + \dfrac{V_{do}\cos[\phi]}{\pi Bt} + P_e$	$S_{hl} = \dfrac{V_{dg}\cos[\phi]}{\pi Bt}$
$S_{hb} = \dfrac{6M_{do}}{\pi Bt^2}$	$S_{hb} = \dfrac{6M_{dg}}{\pi Bt^2}$
$S_{hlbi} = S_{hl} - S_{hb}$	$S_{hlbi} = S_{hl} - S_{hb}$
$S_{hlbo} = S_{hl} + S_{hb}$	$S_{hlbo} = S_{hl} + S_{hb}$
$S_{fm} = \dfrac{1}{\pi BT}\left(\dfrac{\pi B^2 P}{4}\left(\dfrac{4q}{B} - \cot[\phi]\right) - V_{do}\right)\left(\dfrac{K^2+1}{K^2-1}\right) + P_e$	$S_{fm} = \dfrac{1}{\pi BT}\left(-V_{dg}\right)\left(\dfrac{K^2+1}{K^2-1}\right)$
$S_{fb} = \dfrac{0.525n}{Btk_1}\left(V_{do} - \dfrac{4M_{do}\lambda}{B}\right)$	$S_{fb} = \dfrac{0.525n}{Btk_1}\left(V_{dg} - \dfrac{4M_{dg}\lambda}{B}\right)$
$S_{fmbo} = S_{fm} - S_{fb}$	$S_{fmbo} = S_{fm} - S_{fb}$
$S_{fmbi} = S_{fm} + S_{fb}$	$S_{fmbi} = S_{fm} + S_{fb}$
Acceptance Criteria	
$S_{hm} \le S_{ho}$	$S_{hm} \le S_{hg}$
$S_{hl} \le 1.5S_{ho}$	$S_{hl} \le 1.5S_{hg}$
$S_{hlbi} \le 1.5S_{ho}$	$S_{hlbi} \le 1.5S_{hg}$
$S_{hlbo} \le 1.5S_{ho}$	$S_{hlbo} \le 1.5S_{hg}$
$S_{fm} \le S_{fo}$	$S_{fm} \le S_{fg}$
$S_{fmbo} \le 1.5S_{fo}$	$S_{fmbo} \le 1.5S_{fg}$
$S_{fmbi} \le 1.5S_{fo}$	$S_{fmbi} \le 1.5S_{fg}$

4.7.8 FIGURES

Figure 4.7.1
Type A Dished Cover With a Bolting Flange



(a) Loose Flange Type **(b) Integral Flange Type**

GENERAL NOTE: See Table 4.2.5, Details 2 and 3 for transition requirements for a head and skirt with different thicknesses.

Figure 4.7.2
Type B Spherically Dished Cover With a Bolting Flange

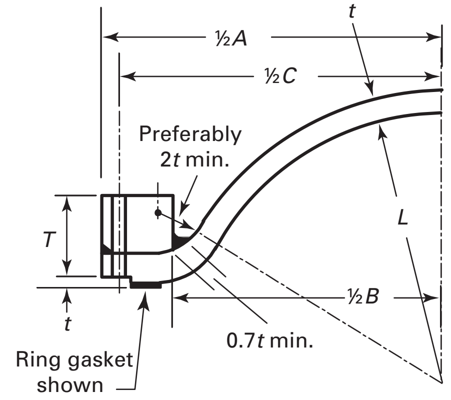
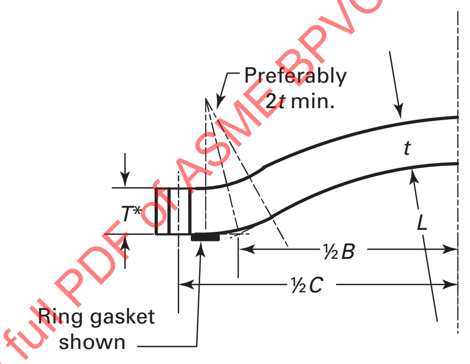


Figure 4.7.3
Type C Spherically Dished Cover With a Bolting Flange



$T^* = T \geq t$

Figure 4.7.4
Type D Spherically Dished Cover With a Bolting Flange

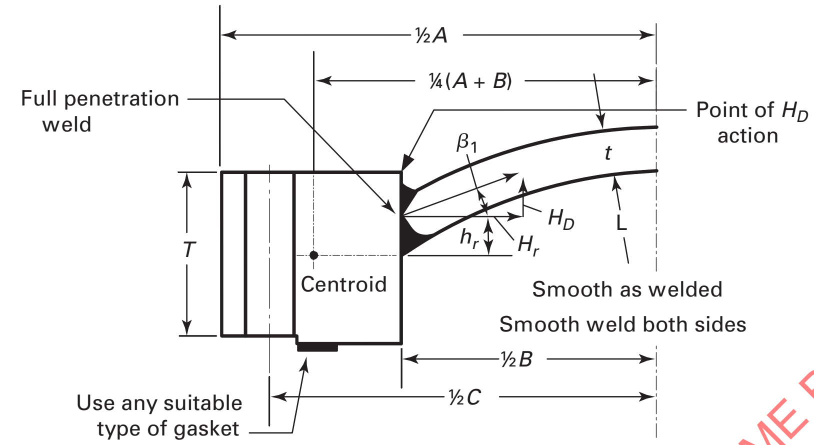
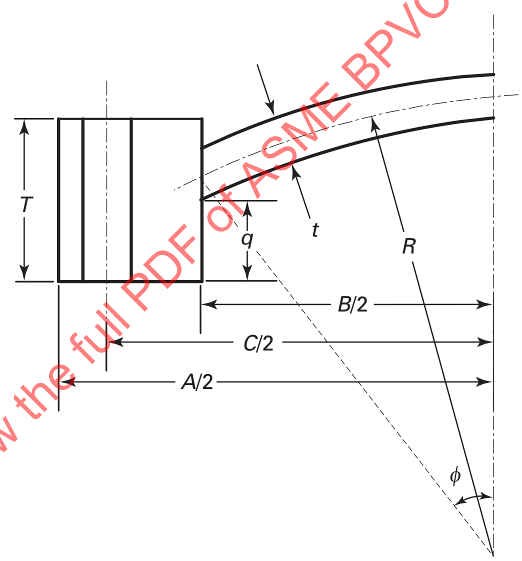


Figure 4.7.5
Type D Head Geometry for Alternative Design Procedure



4.8 DESIGN RULES FOR QUICK-ACTUATING (QUICK-OPENING) CLOSURES

4.8.1 SCOPE

4.8.1.1 Design requirements for quick-actuating or quick-opening closures are provided in 4.8. Specific calculation methods are not provided. However, the rules of Part 4 and Part 5 can be used to qualify the design of a quick-actuating or quick-opening closure.

4.8.2 DEFINITIONS

4.8.2.1 Quick-actuating or quick-opening closures are those that permit substantially faster access to the contents space of a pressure vessel than would be expected with a standard bolted flange connection (bolting through one or both flanges). Closures with swing bolts are not considered quick actuating (quick opening).

4.8.2.2 Holding elements are structural members of the closure used to attach or hold the cover to the vessel, and/or to provide the load required to seal the closure. Hinge pins or bolts can be holding elements.

4.8.2.3 Locking components are parts of the closure that prevent a reduction in the load on a holding element that provides the force required to seal the closure, or prevent the release of a holding element. Locking components may also be used as holding elements.

4.8.2.4 The locking mechanism or locking device consists of a combination of locking components.

4.8.2.5 The use of a multi-link component, such as a chain, as a holding element is not permitted.

4.8.3 GENERAL DESIGN REQUIREMENTS

4.8.3.1 Quick-actuating closures shall be designed such that the locking elements will be engaged prior to or upon application of the pressure and will not disengage until the pressure is released.

4.8.3.2 Quick-actuating closures shall be designed such that the failure of a single locking component while the vessel is pressurized (or contains a static head of liquid acting at the closure) will not:

(a) Cause or allow the closure to be opened or leak; or

(b) Result in the failure of any other locking component or holding element; or

(c) Increase the stress in any other locking component or holding element by more than 50% above the allowable stress of the component.

4.8.3.3 Quick-actuating closures shall be designed and installed such that it may be determined by visual external observation that the holding elements are in satisfactory condition.

4.8.3.4 Quick-actuating closures shall also be designed so that all locking components can be verified to be fully engaged by visual observation or other means prior to the application of pressure to the vessel.

4.8.3.5 When installed, all vessels having quick-actuating closures shall be provided with a pressure-indicating device visible from the operating area and suitable to detect pressure at the closure.

4.8.4 SPECIFIC DESIGN REQUIREMENTS

4.8.4.1 Quick-actuating closures that are held in position by positive locking devices and that are fully released by partial rotation or limited movement of the closure itself or the locking mechanism and any closure that is other than manually operated shall be so designed that when the vessel is installed the following conditions are met:

(a) The closure and its holding elements are fully engaged in their intended operating position before pressure can be applied in the vessel.

(b) Pressure tending to force the closure open or discharge the contents clear of the vessel shall be released before the closure can be fully opened for access.

(c) In the event that compliance with (a) and (b) above is not inherent in the design of the closure and its holding elements, provisions shall be made so that devices to accomplish this can be added when the vessel is installed.

4.8.4.2 The design rules of 4.16 of this code may not be applicable to design Quick-Actuating or Quick-Opening Closures, see 4.16.1.4.

4.8.4.3 The designer shall consider the effects of cyclic loading, other loadings (see 4.1.5.3) and mechanical wear on the holding and locking components.

4.8.4.4 It is recognized that it is impractical to write requirements to cover the multiplicity of devices used for quick access, or to prevent negligent operation or the circumventing of safety devices. Any device or devices that will provide the safeguards broadly described in 4.8.4.1(a) through 4.8.4.1(c) above will meet the intent of this Division.

4.8.5 ALTERNATIVE DESIGNS FOR MANUALLY OPERATED CLOSURES

4.8.5.1 Quick-actuating closures that are held in position by a locking mechanism designed for manual operation shall be designed such that if an attempt is made to open the closure when the vessel is under pressure, the closure will leak prior to full disengagement of the locking components and release of the closure. The design of the closure and vessel shall be such that any leakage shall be directed away from the normal position of the operator.

4.8.5.2 Manually operated closures need not satisfy 4.8.4.1(a) through 4.8.4.1(c) but such closures shall be equipped with an audible or visible warning device that will warn the operator if pressure is applied to the vessel before the holding elements and locking components are fully engaged in their intended position or if an attempt is made to disengage the locking mechanism before the pressure within the vessel is released.

4.8.6 SUPPLEMENTARY REQUIREMENTS FOR QUICK-ACTUATING (QUICK-OPENING) CLOSURES

Annex 4-B provides additional design information for the Manufacturer and provides installation, operational, and maintenance requirements for the Owner.

4.9 DESIGN RULES FOR BRACED AND STAYED SURFACES

4.9.1 SCOPE

4.9.1.1 Design requirements for braced and stayed surfaces are provided in this paragraph. Requirements for the plate thickness and requirements for the staybolt or stay geometry including size, pitch, and attachment details are provided.

4.9.2 REQUIRED THICKNESS OF BRACED AND STAYED SURFACES

4.9.2.1 The minimum thickness for braced and stayed flat plates and those parts that, by these rules, require staying as flat plates with braces or staybolts of uniform diameter symmetrically spaced, shall be calculated by the following equation.

$$t = p_s\sqrt{\frac{P}{SC}} \tag{4.9.1}$$

4.9.2.2 When stays are used to connect two plates, and only one of these plates requires staying, the value of C shall be governed by the thickness of the plate requiring staying.

4.9.3 REQUIRED DIMENSIONS AND LAYOUT OF STAYBOLTS AND STAYS

4.9.3.1 The required area of a staybolt or stay at its minimum cross section, usually located at the root of the thread, exclusive of any corrosion allowance, shall be obtained by dividing the load on the staybolt computed in accordance with 4.9.3.2 by the allowable tensile stress value for the staybolt material, multiplying the result by 1.10.

4.9.3.2 The area supported by a staybolt or stay shall be computed on the basis of the full pitch dimensions, with a deduction for the area occupied by the stay. The load carried by a stay is the product of the area supported by the stay and the maximum allowable working pressure. When a staybolt or stay for a shell is unsymmetrical because of interference with other construction details, the area supported by the staybolt or stay shall be computed by taking the distance from the center of the spacing on one side of the staybolt or stay to the center of the spacing on the other side.

4.9.3.3 When the edge of a flat stayed plate is flanged, the distance from the center of the outermost stays to the inside of the supporting flange shall not be greater than the pitch of the stays plus the inside radius of the flange.

4.9.4 REQUIREMENTS FOR WELDED-IN STAYBOLTS AND WELDED STAYS

4.9.4.1 Welded-in staybolts may be used, provided the following requirements are satisfied.

(a) The configuration is in accordance with the typical arrangements shown in Figure 4.9.1.

(b) The required thickness of the plate shall not exceed 38 mm (1.5 in.).

(c) The maximum pitch shall not exceed 15 times the diameter of the staybolt; however, if the required plate thickness is greater than 19 mm (0.75 in.), the staybolt pitch shall not exceed 508 mm (20 in.).

(d) The size of the attachment welds is not less than that shown in Figure 4.9.1.

(e) The allowable load on the welds shall not exceed the product of the weld area (based on the weld dimension parallel to the staybolt), the allowable tensile stress of the material being welded, and a weld joint factor of 60%.

4.9.4.2 Welded stays may be used, provided the following requirements are satisfied.

(a) The configuration is in accordance with the typical arrangements shown in Figure 4.9.1.

(b) The pressure does not exceed 2 MPa (300 psi).

(c) The required thickness of the plate does not exceed 13 mm (0.5 in.).

(d) The size of the fillet welds is not less than the plate thickness requiring stay.

(e) The inside welds are visually examined before the closing plates are attached.

(f) The allowable load on the fillet welds shall not exceed the product of the weld area (based on the minimum leg dimension), the allowable tensile stress of the material being welded, and a weld joint factor of 55%.

(g) The maximum diameter or width of the hole in the plate shall not exceed 32 mm (1.25 in.).

(h) The maximum pitch, p_s, is determined by eq. (4.9.1) with C= 2.1 if either plate thickness is less than or equal to 11 mm (0.4375 in.) thick, and C= 2.2 for all other plate thicknesses.

4.9.5 NOMENCLATURE

C = stress factor for braced and stayed surfaces (see Table 4.9.1).

P = design pressure.

P_s = maximum pitch. The maximum pitch is the greatest distance between any set of parallel straight lines passing through the centers of staybolts in adjacent rows. Each of the three parallel sets running in the horizontal, the vertical, and the inclined planes shall be considered.

S = allowable stress from Annex 3-A evaluated at the design temperature.

t = minimum required plate thickness.

t_s = nominal thickness of the thinner stayed plate (see Figure 4.9.1).

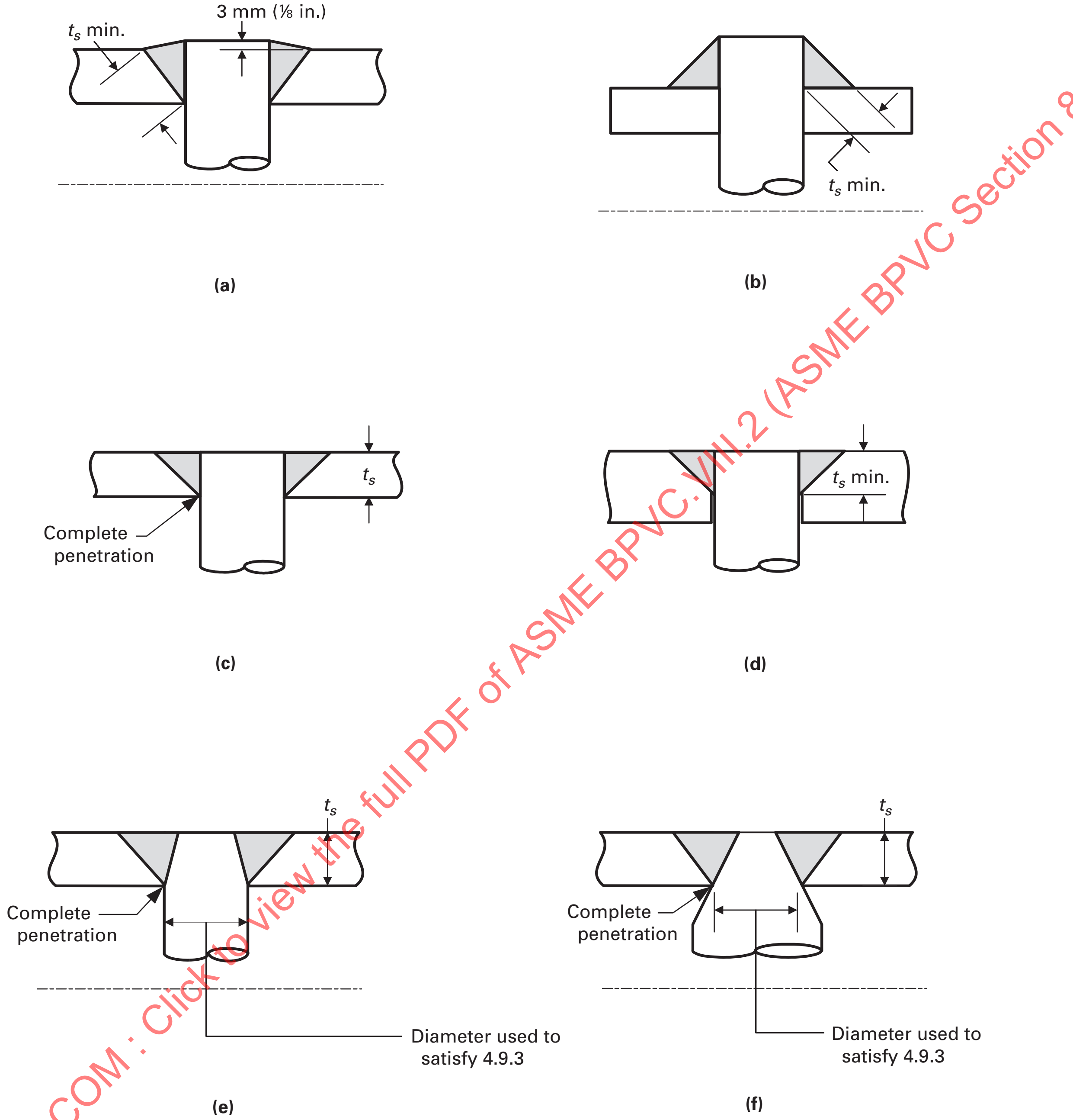
4.9.6 TABLES

Table 4.9.1
Stress Factor for Braced and Stayed Surfaces

Braced and Stayed Surface Construction	C
Welded stays through plates not over 11 mm (0.4375 in.) in thickness	2.1
Welded stays through plates over 11 mm (0.4375 in.) in thickness	2.2

4.9.7 FIGURES

Figure 4.9.1
Typical Forms of Welded Staybolts



GENERAL NOTE: t_s is the nominal thickness of the thinner stayed plate.

4.10 DESIGN RULES FOR LIGAMENTS

4.10.1 SCOPE

4.10.1.1 Rules for determining the ligament efficiency for hole patterns in cylindrical shells are covered in this paragraph. The ligament efficiency or weld joint factor (see 4.10.3) is used in conjunction with the design equations for shells in 4.3.

4.10.2 LIGAMENT EFFICIENCY

4.10.2.1 When a cylindrical shell is drilled for tubes in a line parallel to the axis of the shell for substantially the full length of the shell as shown in Figures 4.10.1 through 4.10.3, the efficiency of the ligaments between the tube holes shall be determined as follows.

(a) When the pitch of the tube holes on every row is equal (see Figure 4.10.1), the ligament efficiency is:

$$E = \frac{p - d}{p} \tag{4.10.1}$$

(b) When the pitch of tube holes on any one row is unequal (as in Figures 4.10.2 and 4.10.3), the ligament efficiency is:

$$E = \frac{p_1 - nd}{p_1} \tag{4.10.2}$$

(c) When the adjacent longitudinal rows are drilled as described in (b), diagonal and circumferential ligaments shall also be examined. The least equivalent longitudinal efficiency shall be used to determine the minimum required thickness and the maximum allowable working pressure.

(d) When a cylindrical shell is drilled for holes so as to form diagonal ligaments, as shown in Figure 4.10.4, the efficiency of these ligaments shall be determined by Figures 4.10.5 and 4.10.6. Figure 4.10.5 is used to determine the efficiency of longitudinal and diagonal ligaments with limiting boundaries where the condition of equal efficiency of diagonal and longitudinal ligaments form one boundary and the condition of equal efficiency of diagonal and circumferential ligaments form the other boundary. Figure 4.10.6 is used for determining the equivalent longitudinal efficiency of diagonal ligaments. This efficiency is used in the equations for setting the minimum required thickness.

(1) Figure 4.10.5 is used when either or both longitudinal and circumferential ligaments exist with diagonal ligaments. To use Figure 4.10.5, compute the value of p^*/p_1 and also the efficiency of the longitudinal ligament. Next find in the diagram, the vertical line corresponding to the longitudinal efficiency of the ligament and follow this line vertically to the point where it intersects the diagonal line representing the ratio of p^*/p_1. Then project this point horizontally to the left, and read the diagonal efficiency of the ligament on the scale at the edge of the diagram. The minimum shell thickness and the maximum allowable working pressure shall be based on the ligament that has the lower efficiency.

(2) Figure 4.10.6 is used for holes that are not in-line, or holes that are placed longitudinally along a cylindrical shell. The diagram may be used for pairs of holes for all planes between the longitudinal plane and the circumferential plane. To use Figure 4.10.6, determine the angle θ between the longitudinal shell axis and the line between the centers of the openings and compute the value of p^*/d. Find in the diagram, the vertical line corresponding to the value of θ and follow this line vertically to the line representing the value of p^*/d. Then project this point horizontally to the left, and read the equivalent longitudinal efficiency of the diagonal ligament. This equivalent longitudinal efficiency is used to determine the minimum required thickness and the maximum allowable working pressure.

(e) When tube holes in a cylindrical shell are arranged in symmetrical groups which extend a distance greater than the inside diameter of the shell along lines parallel to the axis and the same spacing is used for each group, the efficiency for one of the groups shall be not less than the efficiency on which the maximum allowable working pressure is based.

(f) The average ligament efficiency in a cylindrical shell, in which the tube holes are arranged along lines parallel to the axis with either uniform or non-uniform spacing, shall be computed by the following rules and shall satisfy the requirements of both. These rules only apply to ligaments between tube holes and not to single openings. They may give lower efficiencies in some cases than those for symmetrical groups which extend a distance greater than the inside diameter of the shell as covered in (e). When this occurs, the efficiencies computed by the rules under (b) shall govern.

(1) For a length equal to the inside diameter of the shell for the position which gives the minimum efficiency, the efficiency shall be not less than that on which the maximum allowable working pressure is based. When the inside diameter of the shell exceeds 1 520 mm (60 in.), the length shall be taken as 1 520 mm (60 in.), in applying this rule.

(2) For a length equal to the inside radius of the shell for the position which gives the minimum efficiency, the efficiency shall be not less than 80% of that on which the maximum allowable working pressure is based. When the inside radius of the shell exceeds 760 mm (30 in.), the length shall be taken as 760 mm (30 in.), in applying this rule.

4.10.3 LIGAMENT EFFICIENCY AND THE WELD JOINT FACTOR

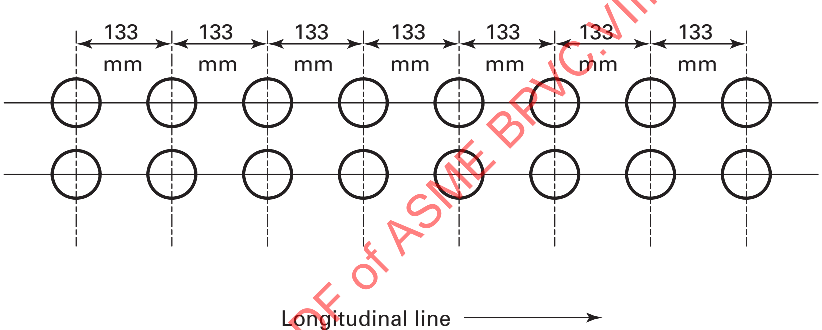
When ligaments occur in cylindrical shells made from welded pipe or tubes and their calculated efficiency is less than 85% (longitudinal) or 50% (circumferential), the efficiency to be used in 4.3 to determine the minimum required thickness is the calculated ligament efficiency. In this case, the appropriate stress value in tension may be multiplied by the factor 1.18.

4.10.4 NOMENCLATURE

d = diameter of tube holes
E = longitudinal ligament efficiency
E_{long} = longitudinal ligament efficiency in percent
p = longitudinal pitch of tube holes
p_1 = unit length of ligament
p^* = diagonal pitch of tube holes
θ = angle of the diagonal pitch with respect to the longitudinal line
s = longitudinal dimension of diagonal pitch, $p^* {\cdot} \cos\theta$
n = number of tube holes in length p_1

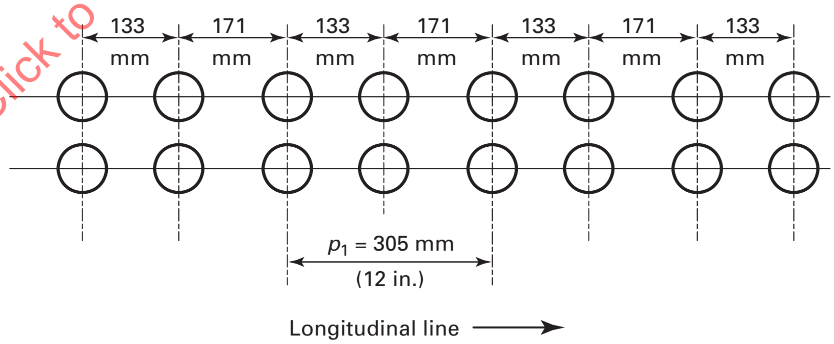
4.10.5 FIGURES

Figure 4.10.1
Example of Tube Spacing With the Pitch of Holes Equal in Every Row



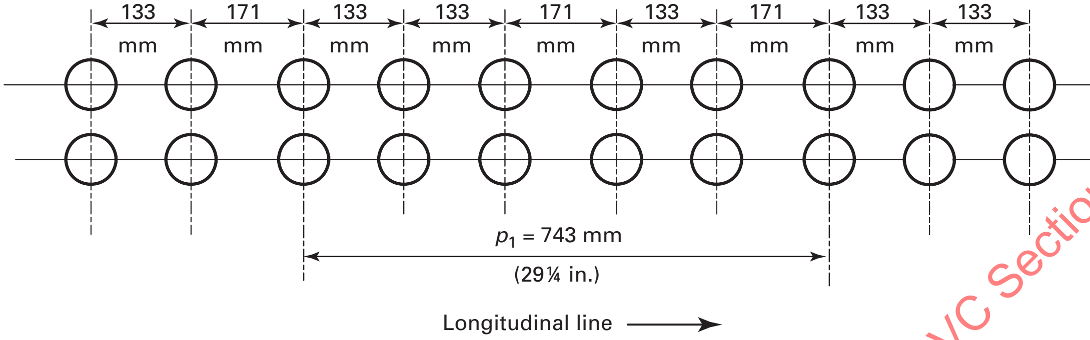
GENERAL NOTE: 133 mm = $5\frac{1}{4}$ in.

Figure 4.10.2
Example of Tube Spacing With the Pitch of Holes Unequal in Every Second Row



GENERAL NOTES:
(a) 133 mm = $5\frac{1}{4}$ in.
(b) 171 mm = $6\frac{3}{4}$ in.

Figure 4.10.3
Example of Tube Spacing With the Pitch of Holes Varying in Every Second and Third Row



GENERAL NOTES:
(a) 133 mm = $5^1/_4$ in.
(b) 171 mm = $6^3/_4$ in.

Figure 4.10.4
Example of Tube Spacing With the Tube Holes on Diagonal Lines

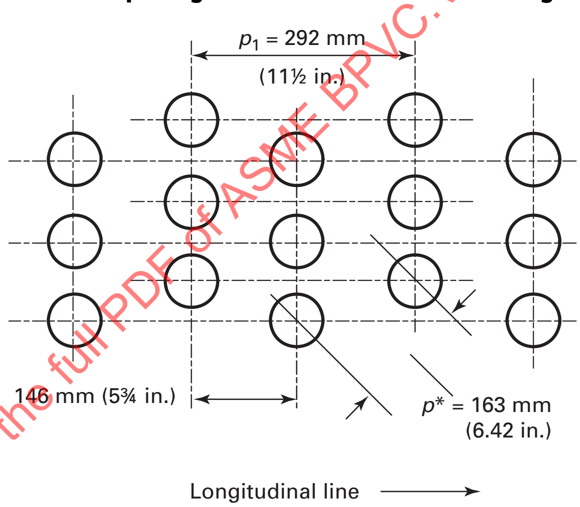
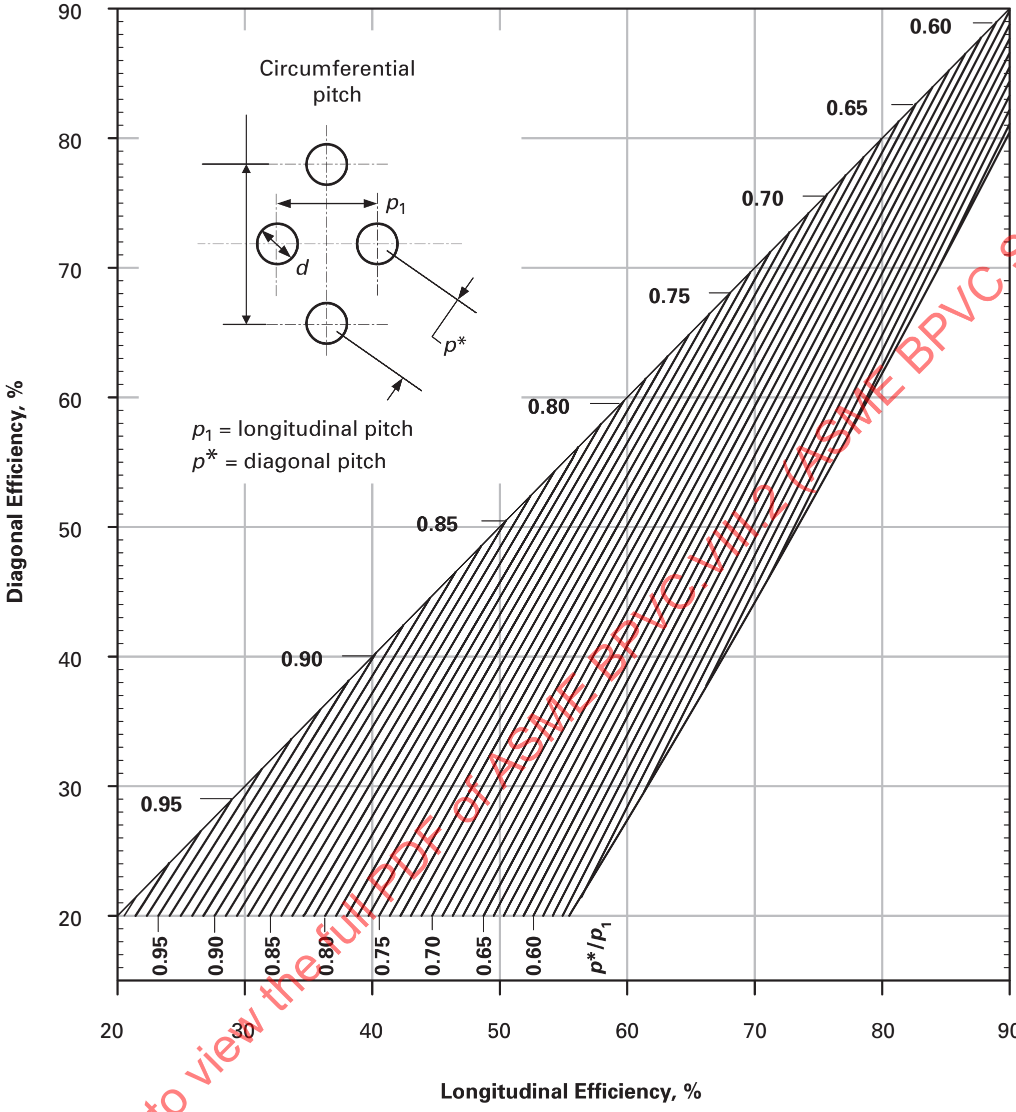


Figure 4.10.5
Diagram for Determining the Efficiency of Longitudinal and Diagonal Ligaments Between Openings in Cylindrical Shells



GENERAL NOTES:

(a) Equations are provided for the curve shown above, use of these equations is permitted for values beyond the values shown in this curve.

(b)

$$\text{Diagonal efficiency: } \% = \frac{J + 0.25 - \left(1 - 0.01 \cdot E_{long}\right)\sqrt{0.75 + J}}{0.00375 + 0.005 \cdot J}, \qquad \text{where } J = \left(\frac{p^*}{p_1}\right)^2$$

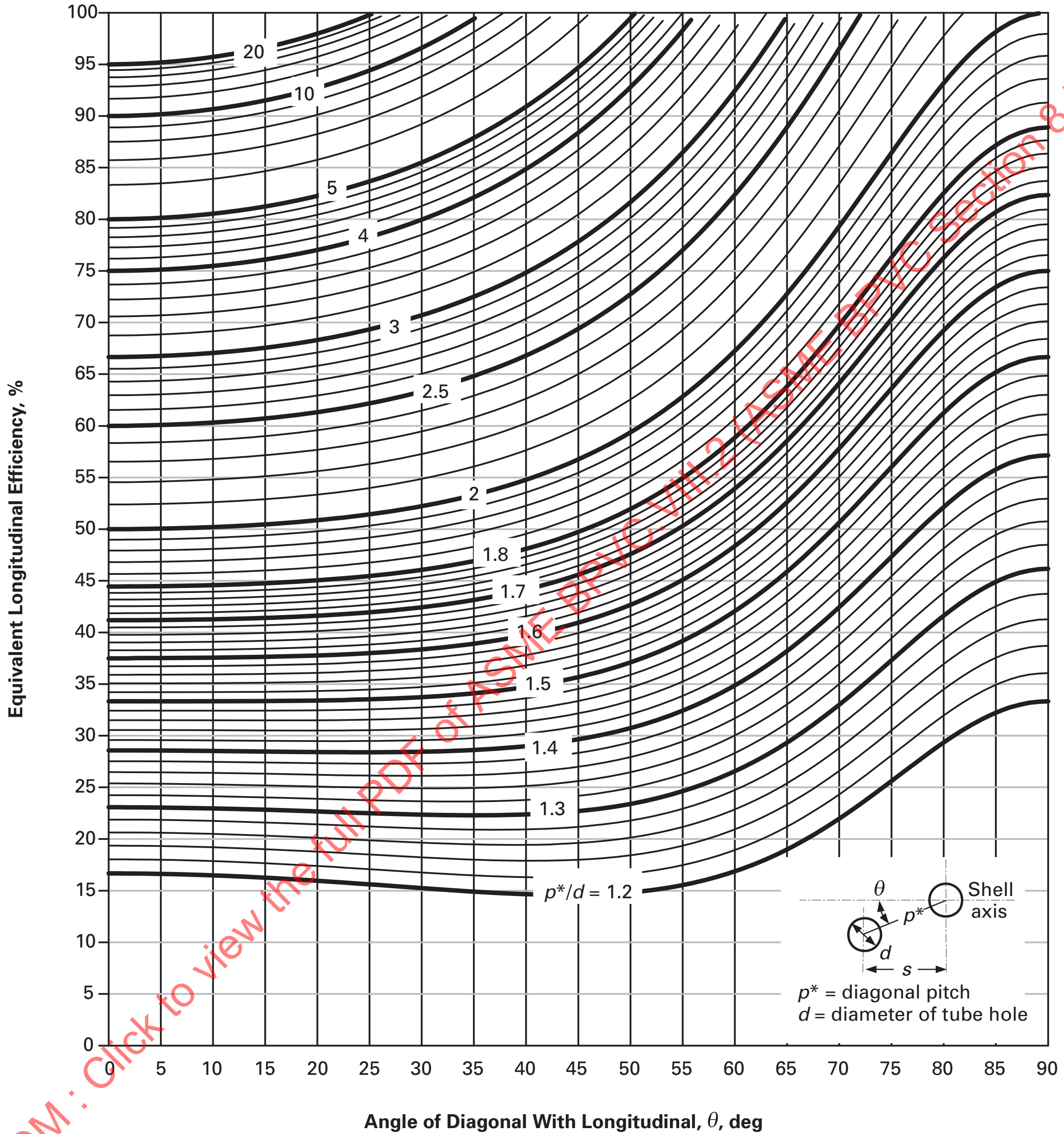
(c) The curve of condition of equal efficiency of diagonal and circumferential ligaments is given by:

$$\% = \frac{200M + 100 - 2\left(100 - E_{long}\right)\sqrt{1 + M}}{1 + M}, \qquad \text{where } M = \left(\frac{100 - E_{long}}{200 - 0.5E_{long}}\right)^2$$

(d)

$$E_{long} = 100\left(\frac{p_1 - d}{p_1}\right)$$

Figure 4.10.6
Diagram for Determining the Equivalent Efficiency of Diagonal Ligaments Between Openings in Cylindrical Shells



GENERAL NOTES:

(a) An equation is provided for the curve shown above, the use of this equation is permitted for values beyond the values shown in this curve.

(b) Equivalent longitudinal efficiency: $\% = \dfrac{\sec^2\theta + 1 - \left(\dfrac{\sec\theta}{p^*/d}\right)\sqrt{3 + \sec^2\theta}}{0.015 + 0.005\sec^2\theta}$

4.11 DESIGN RULES FOR JACKETED VESSELS

4.11.1 SCOPE

4.11.1.1 The minimum requirements for the design of the jacketed portion of a pressure vessel shall conform to the requirements given in 4.11. The jacketed portion of the vessel is defined as the inner and outer walls, the closure devices and all other penetration or parts within the jacket that are subjected to pressure stress. Parts such as nozzle closure members and stay rings are included in this definition. For the purposes of this section, jackets are assumed to be integral pressure chambers, attached to a vessel for one or more purposes, such as:

(a) To heat the vessel and its contents,

(b) To cool the vessel and its contents, or

(c) To provide a sealed insulation chamber for the vessel.

4.11.1.2 4.11 applies only to jacketed vessels having jackets over the shell or heads as illustrated in Figure 4.11.1, partial jackets as illustrated in Figure 4.11.2, and half-pipe jackets as illustrated in Figure 4.11.3.

(**21**) **4.11.1.3** The jacketed vessels shown in Figure 4.11.1 are categorized as five types shown below. For these types of vessels, the jackets shall be continuous circumferentially for Type 1, 2, 4, or 5 and shall be circular in cross section for Type 3. The use of any combination of the types shown is permitted on a single vessel, provided the individual requirements for each are met. When nozzles or other openings in Type 1, 2, 3, 4, or 5 jackets also penetrate the vessel shell or head, design of the opening in the inner vessel wall shall be in accordance with 4.5. 4.11 does not cover dimpled or embossed jackets.

(a) Type 1 - Jacket of any length confined entirely to the cylindrical shell

(b) Type 2 - Jacket covering a portion of the cylindrical shell and one head

(c) Type 3 - Jacket covering a portion of one head

(d) Type 4 - Jacket with addition of stay or equalizer rings to the cylindrical shell portion to reduce the effective length

(e) Type 5 - Jacket covering the cylindrical shell and any portion of either head.

4.11.1.4 4.11 does not contain rules to cover all details of design and construction. Jacket types defined in 4.11.1.3 subject to general loading conditions (i.e., thermal gradients) or jacket types of different configurations subject to general loading conditions shall be designed using Part 5.

4.11.1.5 If the internal pressure is 100 kPa (15 psi) or less, and any combination of pressures and vacuum in the vessel and jacket will produce a total pressure greater than 100 kPa (15 psi) on the inner vessel wall, then the entire jacket is within the scope of 4.11.

4.11.2 DESIGN OF JACKETED SHELLS AND JACKETED HEADS

4.11.2.1 Shell and head thickness shall be determined using 4.3 and 4.4 as applicable. In consideration of the loadings given in 4.1, particular attention shall be given to the effects of local internal and external pressure loads and differential thermal expansion (see 4.11.1.4). Where vessel supports are attached to the jacket, consideration shall be given to the transfer of the supported load of the inner vessel and contents.

4.11.2.2 The requirements for inspection openings in jackets shall be in accordance with 4.5.16 except that the maximum size of inspection openings in the jacketed portion of the vessel need not exceed DN 50 (NPS 2) pipe for all diameter vessels.

4.11.2.3 The use of impingement plates or baffles at the jacket inlet connection to reduce erosion of the inner wall shall be considered for media where vapors are condensed (i.e., steam).

4.11.2.4 Flat plate regions of jacketed vessels may be designed as braced and stayed surfaces using the rules of 4.9.

4.11.3 DESIGN OF CLOSURE MEMBER OF JACKET TO VESSEL

4.11.3.1 The design of jacket closure members shall be in accordance with Table 4.11.1 and the additional requirements of 4.11.3. Alternative geometries to those illustrated may be used in accordance with 4.11.1.4.

4.11.3.2 Any radial welds in closure members shall be butt-welded joints penetrating through the full thickness of the member and shall be ground flush where attachment welds are to be made.

4.11.3.3 Partial penetration and fillet welds are permitted when both of the following requirements are satisfied.

(a) The material of construction satisfies the following equation.

$$\frac{S_{yT}}{S_u} \leq 0.625 \tag{4.11.1}$$

(b) The component is not in cyclic service, i.e., a fatigue analysis is not required in accordance with 4.1.1.4.

4.11.3.4 Closures for any type of stay-bolted jacket may be designed in accordance with the requirements of Type 1 jackets shown in Figure 4.11.1, provided the entire jacket is stay-bolted to compensate for pressure end forces.

4.11.4 DESIGN OF PENETRATIONS THROUGH JACKETS (21)

4.11.4.1 Jacket penetrations other than those shown in Table 4.11.2 shall conform to the following requirements:

(a) Design of openings that only penetrate the jacket shall be in accordance with the rules given in 4.5.

(b) Design of openings through the jacket space that also penetrate the vessel shell or head shall be in accordance with Part 5.

4.11.4.2 For jacket penetrations shown in Table 4.11.2, Detail 1, design of openings shall be in accordance with the rules given in 4.5. However, when applying these rules, the limits of reinforcement for the vessel opening and jacket opening shall not overlap. For all other jacket penetrations shown in Table 4.11.2, design of openings through the jacket space shall be in accordance with Part 5.

4.11.4.3 Jacket penetration closure member designs shown in Table 4.11.2 shall conform to the following requirements stipulated in this table and the following provisions. Alternative geometries to those illustrated may be used if the design is based on Part 5.

(a) The jacket penetration closure member minimum thickness considers only pressure membrane loading. Axial pressure loadings and secondary loadings given in 4.1 shall be considered in the design.

(b) The design Details 2, 3, 4, 5 and 6 shown in Table 4.11.2 provide some flexibility. Only pressure membrane loading is considered in establishing the minimum thickness of the penetration closure member. If the localized stresses at the penetration detail need to be established, the methodology in Part 5 shall be used.

(c) All radial welds in opening sealer membranes shall be butt-welded joints that penetrate through the full thickness of the member.

(d) Closure member welds shall be circular, elliptical, or obround in shape where possible. Rectangular member welds are permissible, provided that corners are rounded to a suitable radius.

(e) The requirements of 4.11.3.3 shall be satisfied.

4.11.5 DESIGN OF PARTIAL JACKETS

4.11.5.1 Partial jackets include jackets that encompass less than the full circumference of the vessel. Some variations are shown in Figure 4.11.2.

4.11.5.2 The rules for construction of jacketed vessels in the preceding paragraphs also apply to partial jackets, with the following exceptions.

(a) Stayed partial jackets shall be designed and constructed in accordance with 4.9 with closures designed in accordance with 4.11.3.

(b) Partial jackets that, by virtue of their service or configuration, do not lend themselves to staybolt construction may be fabricated by other means, provided they are designed using Part 5.

4.11.6 DESIGN OF HALF-PIPE JACKETS

4.11.6.1 The rules in this section are applicable for the design of half-pipe jackets constructed of NPS 2, 3 or 4 pipes and subjected to internal pressure loading (see Figure 4.11.3). Configurations that do not satisfy the rules in 4.11.6.1 may be designed in accordance with Part 5.

4.11.6.2 The fillet weld attaching the half-pipe jacket to the vessel shall have a throat thickness not less than the smaller of the jacket or shell thickness. Consideration should be given to the selection of the half-pipe jacket pitch needed to provide welder access. In addition, the requirements of 4.11.3.3 shall be satisfied.

4.11.6.3 The minimum required thickness of a half pipe jacket is given by the following equation. For a design to be acceptable, the additional condition that $P_j \le P_{jpm}$ where P_{jpm} is given by eq. (4.11.3) must also be satisfied.

$$t_{rp} = \frac{P_j r_p}{0.85S_j - 0.6P_j} \tag{4.11.2}$$

(21) **4.11.6.4** The maximum permissible pressure in the half-pipe jacket, P_{jpm}, shall be determine using the following equation.

$$P_{jpm} = \frac{F_p}{K_p} \tag{4.11.3}$$

where

$$F_p = \min\left[\left(1.5S - S^*\right), 1.5S\right] \tag{4.11.4}$$

$$K_p = C_1 + C_2\left(\frac{D}{C_{ul}}\right)^{0.5} + C_3\frac{D}{C_{ul}} + C_4\left(\frac{D}{C_{ul}}\right)^{1.5} + C_5\left(\frac{D}{C_{ul}}\right)^{2} + C_6\left(\frac{D}{C_{ul}}\right)^{2.5} + C_7\left(\frac{D}{C_{ul}}\right)^{3} + C_8\left(\frac{D}{C_{ul}}\right)^{3.5} + C_9\left(\frac{D}{C_{ul}}\right)^{4} + C_{10}\left(\frac{D}{C_{ul}}\right)^{4.5} \tag{4.11.5}$$

The coefficients for eq. (4.11.5) are provided in Table 4.11.3.

(21) 4.11.7 NOMENCLATURE

c_{ul} = conversion factor for length. c_{ul} = 1.0 in. for U.S. Customary Units, and c_{ul} = 25.4 mm for SI Units.
D = inside diameter of the inner vessel.
D_{pj} = nominal pipe size of the half-pipe jacket.
K_p = half-pipe jacket rating factor.
P_j = design pressure in the jacket chamber.
P_{jpm} = permissible jacket pressure based on the jacket and shell geometry.
j = jacket space defined as the inside radius of the jacket minus the outside radius of the inner vessel.
L = length of the jacket.
t_c = nominal thickness of the closure member.
t_j = nominal thickness of the outer jacket wall.
t_n = nominal thickness of the nozzle.
t_s = nominal thickness of the shell inner wall.
t_{rj} = required minimum thickness of the outer jacket wall.
t_{rc} = required minimum thickness of the closure member.
t_{rp} = required minimum thickness of the half-pipe jacket.
R_j = inside radius of the jacket.
R_p = radius of the opening in the jacket at the jacket penetration
R_s = outside radius of the inner vessel.
r = corner radius of torus closures.
r_p = inside radius of the half-pipe jacket.
S = allowable stress of the inner shell from Annex 3-A at the design temperature.
S_c = allowable stress of the jacket closure from Annex 3-A at the design temperature.
S_j = allowable stress of the jacket from Annex 3-A at the design temperature.
S_{yT} = yield strength from Annex 3-D at the design temperature.
S_u = minimum specified ultimate tensile strength from Annex 3-D.
S^* = actual longitudinal tensile stress in the head or shell due to internal pressure and other axial forces; when axial forces are negligible, $S^* = PD/4t_s$. If the combination of axial forces and pressure results in a negative value of S^*, then $S^* = 0$.

4.11.8 TABLES

Table 4.11.1
Design of Closure Member of Jacket to Shell

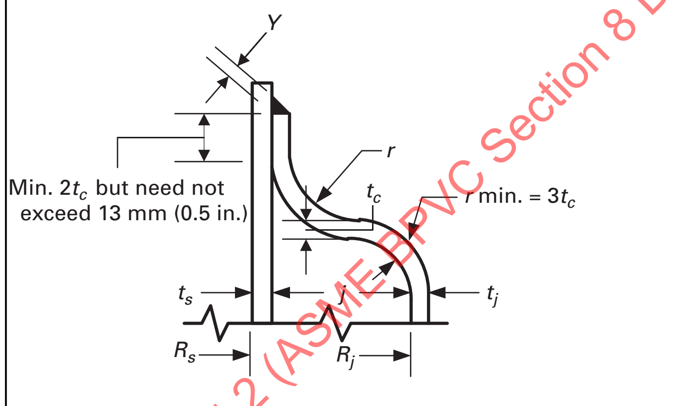
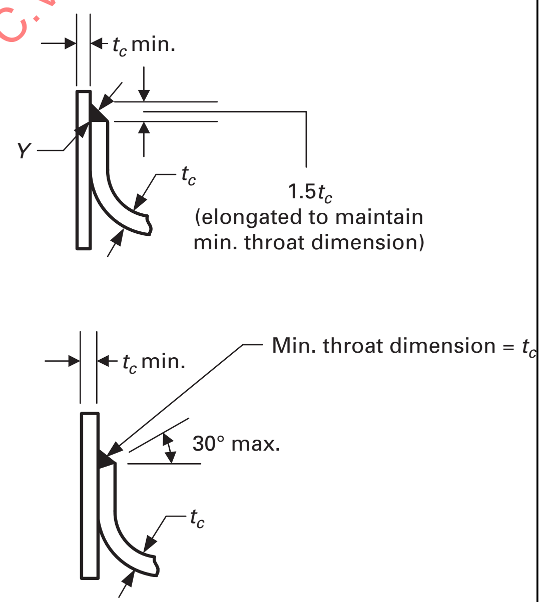
Detail	Requirements	Figure
1	Closure details (a) and (b) shall only be used when the requirements of 4.11.3.3 are satisfied. These closures may be used on Types 1, 2, and 4 jacketed vessels as shown in Figure 4.11.1 and shall have t_{rc} of at least equal to t_{rj} and corner radius r shall not be less than $3t_c$. These closure designs are limited to a maximum thickness t_{rc} of 16 mm (0.625 in.) When this construction is used on Type 1 jacketed vessels, the weld dimension Y shall be not less than $0.7t_c$; and when used on Type 2 and 4 jacketed vessels, the weld dimension Y shall be not less than $0.83t_c$.	 (a) Type 1 Jackets  (b) Types 2 and 4 Jackets
2	These closures shall have t_{rc} at least equal to t_{rj}. In addition for Detail (c), t_{rc} shall be not less than the following: $$t_{rc} = 0.707j\sqrt{\frac{P_j}{S_c}}$$ A groove weld attaching the closure to the inner vessel and fully penetrating the closure thickness t_c may be used with any of the types of jacketed vessels shown in Figure 4.11.1. However, a fillet weld having a minimum throat dimension of $0.7t_c$ may also be used to join the closure of the inner vessel on Type 1 jacketed vessels of Figure 4.11.1.	

Table 4.11.1
Design of Closure Member of Jacket to Shell (Cont'd)

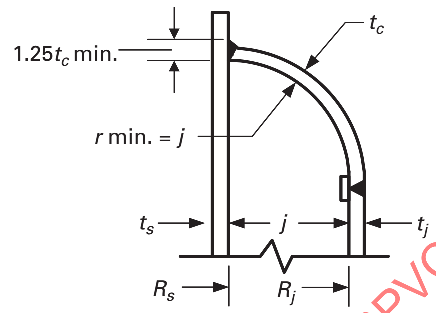
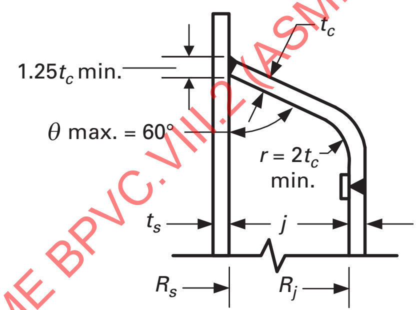
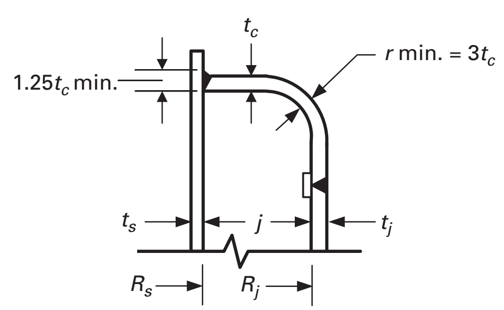
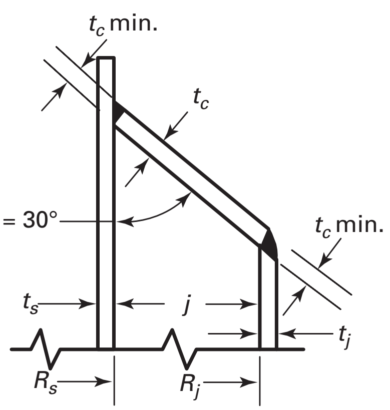
Detail	Requirements	Figure
	The closure and jacket shell may be one-piece construction or welded using a full penetration butt weld. A backing strip may be used.	 (a)  (b)  (c)
3	This closure shall be used only on Type 1 jacketed vessels shown in Figure 4.11.1. The closure thickness t_{rc} shall be computed using the Equation for a conical shell in 4.3, but shall be not less than t_{rj}. The angle θ shall be limited to 30 deg maximum.	

Table 4.11.1
Design of Closure Member of Jacket to Shell (Cont'd)

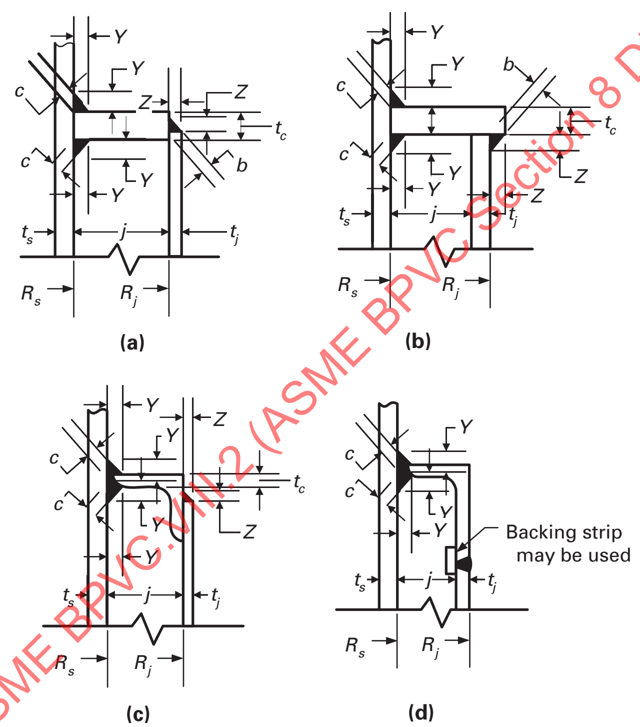
Detail	Requirements	Figure
4	Closure details (a), (b), and (c) shall only be used when the requirements of 4.11.3.3 are satisfied. These closures shall be used only on Type 1 jacketed vessels as shown in Figure 4.11.1 and with the further limitation that t_{rj} does not exceed 16 mm (0.625 in.). The required minimum thickness for the closure bar shall be equal to: $t_{rc} = \max\left[2t_{rj}, \left(0.707j\sqrt{\frac{P_j}{S_c}}\right)\right]$ Fillet weld sizes shall be as follows: $Y \geq \min[0.75t_c, 0.75t_s]$ and $c = 0.7Y$ min $Z \geq t_j$ and $b = 0.7Z$ min	
5	Closure details (a), (b), and (c) shall only be used when the requirements of 4.11.3.3 are satisfied. These closures may be used on any of the types of jacketed vessels shown in Figure 4.11.1. For Type 1 jacketed vessels, the required minimum closure bar thickness shall be determined from the equations in Table 4.11.1, Detail 4. For all other types of jacketed vessels, the required minimum closure bar thickness and the maximum allowable width of the jacket space shall be determined from the following equations: $t_{rc} = 1.414\sqrt{\left(\frac{P_j R_s j}{S_c}\right)}$ $j = \frac{2S_c t_s^2}{P_j R_j} - \frac{(t_s + t_j)}{2}$ Weld sizes connecting the closure bar to the inner vessel shall be as follows: $Y \geq \min[1.5t_c, 1.5t_s]$, and shall be measured as the sum of dimensions a and b as shown. In addition, $a, b \geq \min[6 \text{ mm } (^1/_4 \text{ in.}), t_c, t_s]$ Z is equal to the minimum fillet size necessary when used in conjunction with a groove weld or another fillet weld to maintain the minimum required Y dimension.	

Table 4.11.1
Design of Closure Member of Jacket to Shell (Cont'd)

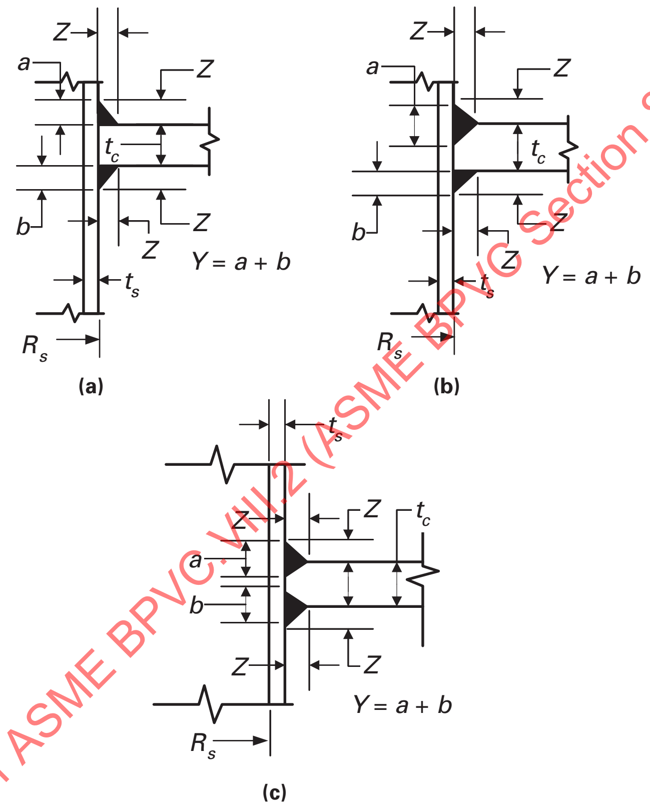
Detail	Requirements	Figure
		
6	Closure details (a), (b), and (c) shall only be used when the requirements of 4.11.3.3 are satisfied. The jacket to closure bar attachment welds shown in Details (a), (b) and (c) may be used on any of the types of jacketed vessels shown in Figure 4.11.1. Attachment welds shown in Details (d), (e) and (f) may be used on any of the types of jacketed vessels shown in Figure 4.11.1 where t_{rj} does not exceed 16 mm (0.625 in.). The required minimum closure bar thickness and the maximum allowable width of the jacket space shall be determined from the following equations: $t_{rc} = 1.414\sqrt{\frac{P_j R_s j}{S_c}}$ $j = \frac{2 S_c t_s^2}{P_j R_j} - \frac{\left(t_s + t_j\right)}{2}$	

Table 4.11.1
Design of Closure Member of Jacket to Shell (Cont'd)

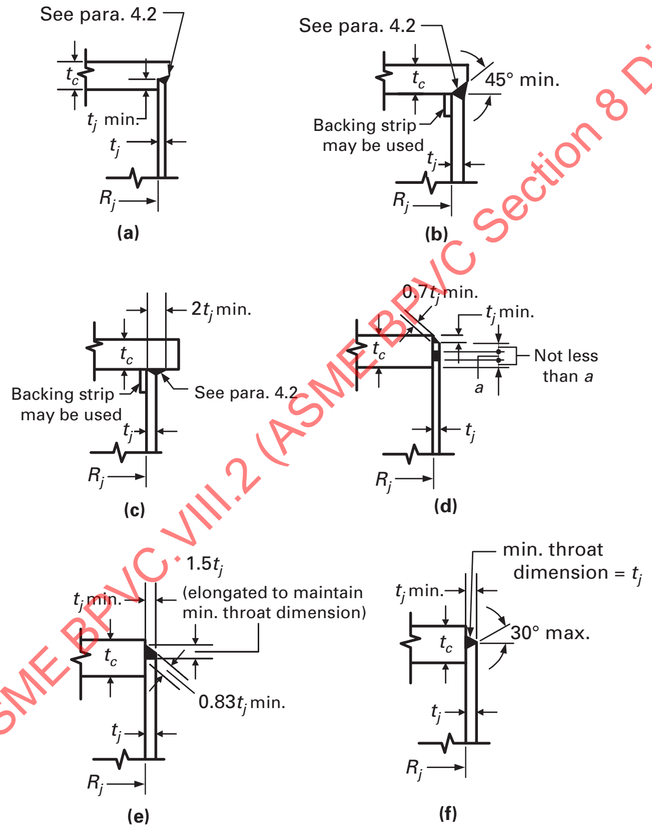
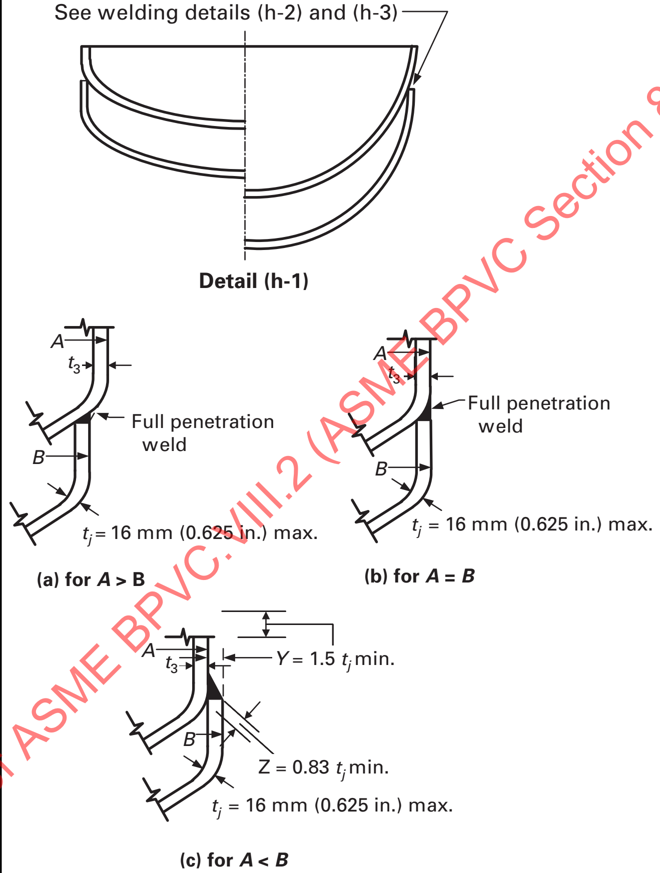
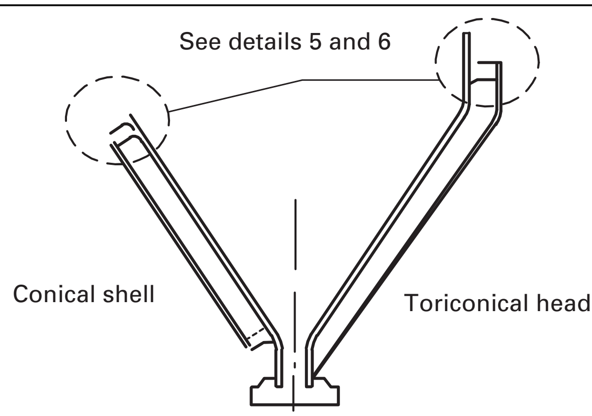
Detail	Requirements	Figure
		
7	Closure details (a), (b), and (c) shall only be used when the requirements of 4.11.3.3 are satisfied. These closures may be used on Type 3 jacketed vessels shown in Figure 4.11.1 shall have attachment welds in accordance with Details (a), (b) or (c). This construction is limited to jackets where t_{rj} does not exceed 16 mm (0.625 in.). For torispherical, ellipsoidal, and hemispherical heads, the outside diameter of jacket head shall not be greater than the outside diameter of the vessel head, or the inside diameter of the jacket head shall be nominally equal to the outside diameter of vessel head.	

Table 4.11.1
Design of Closure Member of Jacket to Shell (Cont'd)

Detail	Requirements	Figure
		
8	Closures for conical or toriconical jackets shall comply with the requirements for Type 2 jacketed vessels shown in Figure 4.11.1.	

(21)

Table 4.11.2
Design of Jacket Penetration Details

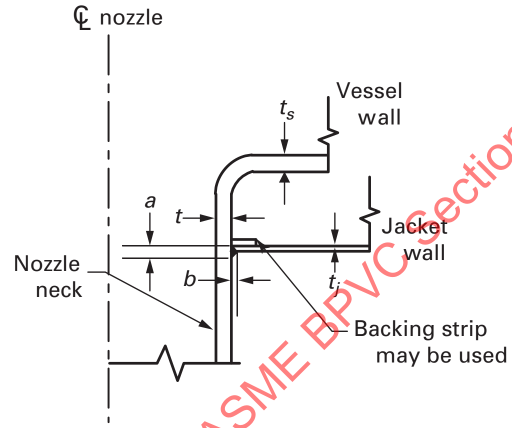
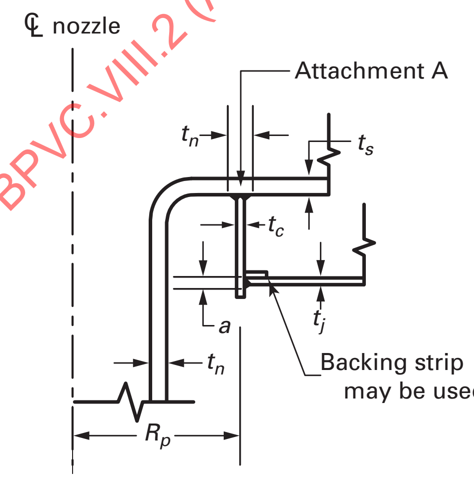
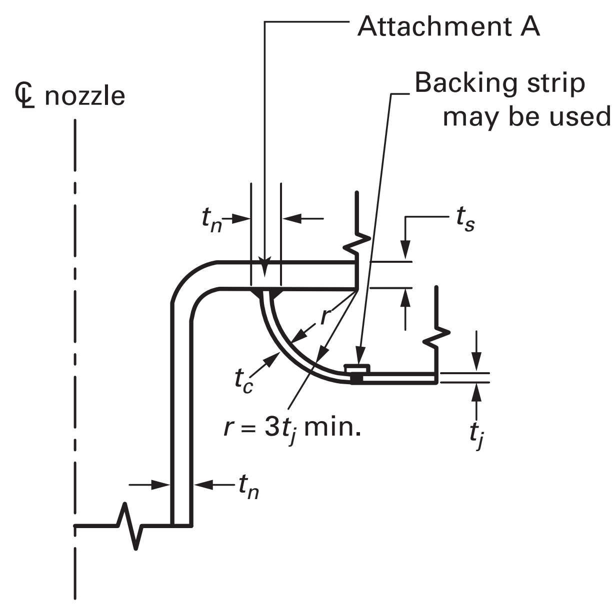
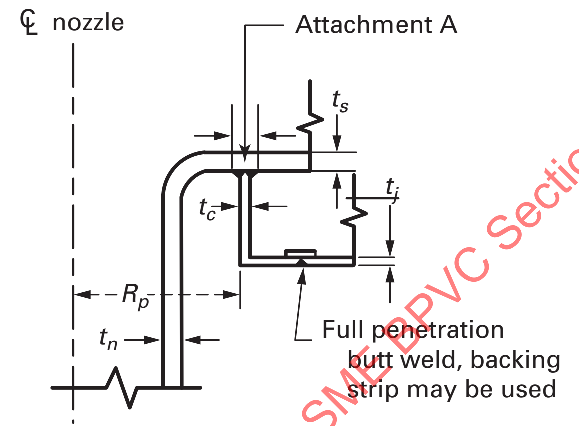
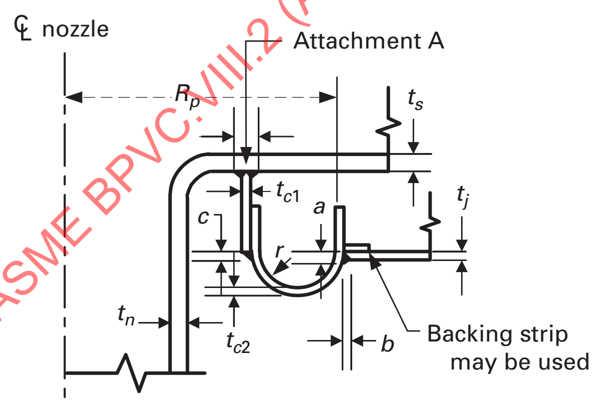
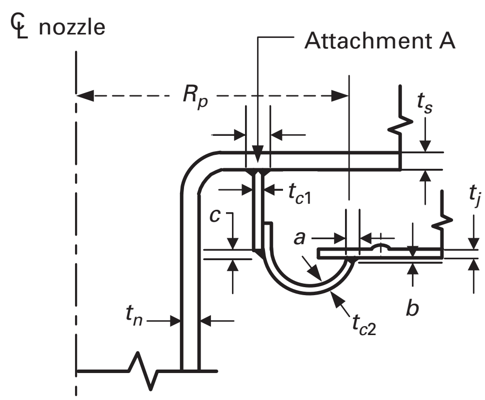
Detail	Requirements	Figure
1	This closure details shall only be used when the requirements of 4.11.3.3 are satisfied. The nozzle wall may be used as the closure member where the jacket is welded to the nozzle. The nozzle wall shall comply with the requirements of 4.4. The external design pressure, P, (see 4.4) shall be taken as the design pressure of the jacket chamber, P_j, (see 4.11) plus the external design pressure of the main vessel, if applicable. The unsupported length, L, (see 4.4) shall be taken as the jacket space, j, (see 4.11). $a = 2t_j$ min and $b = t_j$ min	
2	This closure details shall only be used when the requirements of 4.11.3.3 are satisfied. The minimum required thickness, t_{rc}, for the geometries shall be calculated as a shell under external pressure in accordance with 4.4. $a = 2t_j$ min and $b = t_j$ min Attachment A shall be made using details in Table 4.2.6.	
3	This closure details shall only be used when the requirements of 4.11.3.3 are satisfied. The minimum required thickness, t_{rc}, shall be equal to t_{rj}. Attachment A shall be made using details in Table 4.2.6.	

Table 4.11.2
Design of Jacket Penetration Details (Cont'd)

<table>
<tr><th>Detail</th><th>Requirements</th><th>Figure</th></tr>
<tr><td>4</td><td>This closure details shall only be used when the requirements of 4.11.3.3 are satisfied.
The minimum required thickness, t_{rc}, shall be calculated as a shell under external pressure in accordance with 4.4.
Attachment A shall be made using details in Table 4.2.6.</td><td>
</td></tr>
<tr><td>5</td><td>This closure details shall only be used when the requirements 4.11.3.3 are satisfied.
The thickness required of the closure member attached to the inner vessel, t_{rc1}, shall be calculated as a shell under external pressure in accordance with 4.4. The required thickness of the flexible member, t_{rc2}, shall be determined as follows:
When a tubular section does not exist between jacket and torus:
$$t_{rc2} = \frac{Pr}{S_c E - 0.6P_j}$$
When a tubular section does exist between jacket and torus:
$$t_{rc2} = \frac{P_j R_p}{S_c E - 0.6P_j}$$
$a = 2t_j$, $b = t_j$, and $c = 1.25t_{c1}$
Attachment A shall be made using details in Table 4.2.6.</td><td>

(a)


(b)</td></tr>
</table>

Table 4.11.2
Design of Jacket Penetration Details (Cont'd)

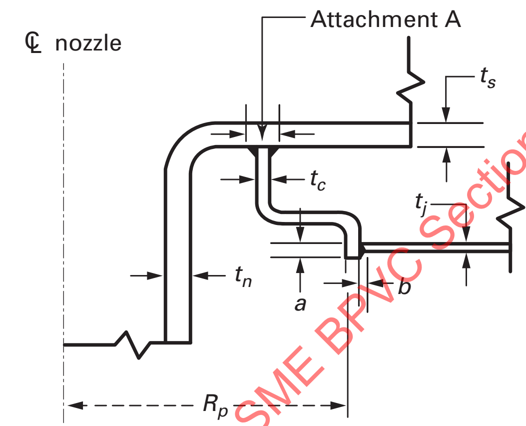
Detail	Requirements	Figure
6	This closure detail shall only be used when the requirements of 4.11.3.3 are satisfied. The minimum thickness, t_{rc}, shall be calculated as a shell of radius R_p under external pressure in accordance with 4.4. $a = 2t_j$ and $b = t_j$ Attachment A shall be made using details in Table 4.2.6.	

Table 4.11.3
Coefficients for Eq. (4.11.5)

D_{pj}	Coefficients	Shell Thickness 5 mm ($^3/_{16}$ in.)	6 mm ($^1/_4$ in.)	10 mm ($^3/_8$ in.)	13 mm ($^1/_2$ in.)
DN50 (NPS 2)	C_1	−3.6674510 E+01	−1.8874043 E+04	4.0083779 E+02	−2.6447784 E+02
	C_2	1.2306994 E+01	1.7869518 E+04	−5.7029108 E+02	1.8066952 E+02
	C_3	3.5701684 E+00	−7.2846419 E+03	3.1989698 E+02	−4.9294965 E+01
	C_4	−7.9516583 E−01	1.6723763 E+03	−9.4286208 E+01	7.1522422 E+00
	C_5	5.8791041 E−02	−2.3648930 E+02	1.6391764 E+01	−5.7900069 E−01
	C_6	−1.5365397 E−03	2.1101742 E+01	−1.7431218 E+00	2.4758486 E−02
	C_7	0.0000000 E+00	−1.1608890 E+00	1.1160179 E−01	−4.3667599 E−04
	C_8	0.0000000 E+00	3.6022711 E−02	−3.9549592 E−03	0.0000000 E+00
	C_9	0.0000000 E+00	−4.8303253 E−04	5.9644209 E−05	0.0000000 E+00
	C_{10}	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00
DN80 (NPS 3)	C_1	−3.7588705 E+03	−1.2551406 E+04	−3.8104460 E+04	−1.4263782 E+04
	C_2	2.9919870 E+03	1.2149900 E+04	4.0491537 E+04	1.6228077 E+04
	C_3	−9.4177823 E+02	−5.0657776 E+03	−1.8844078 E+04	−8.0227888 E+03
	C_4	1.5278500 E+02	1.1910361 E+03	5.0415301 E+03	2.2676555 E+03
	C_5	−1.3452359 E+01	−1.7255075 E+02	−8.5435371 E+02	−4.0440980 E+02
	C_6	6.1167422 E−01	1.5770136 E+01	9.5115501 E+01	4.7257835 E+01
	C_7	−1.1235632 E−02	−8.8782173 E−01	−6.9588768 E+00	−3.6233229 E+00
	C_8	−2.1465752 E−06	2.8148933 E−02	3.2277515 E−01	1.7597455 E−01
	C_9	0.0000000 E+00	−3.8488963 E−04	−8.6172557 E−03	−4.9179021 E−03
	C_{10}	0.0000000 E+00	0.0000000 E+00	1.0094910 E−04	6.0315412 E−05
DN100 (NPS 4)	C_1	−2.1336346 E+04	7.3995872 E+03	8.3115447 E+02	−4.0097574 E+02
	C_2	1.5982068 E+04	−6.7592710 E+03	−7.6253222 E+02	4.2602525 E+02
	C_3	−4.9936486 E+03	2.6131811 E+03	2.9500674 E+02	−1.7446665 E+02
	C_4	8.4914220 E+02	−5.4873257 E+02	−6.1135935 E+01	3.7753845 E+01
	C_5	−8.4931392 E+01	6.7571708 E+01	7.4233181 E+00	−4.6748939 E+00
	C_6	5.0044853 E+00	−4.8769663 E+00	−5.2938127 E−01	3.3376011 E−01
	C_7	−1.6105634 E−01	1.9112909 E−01	2.0558271 E−02	−1.2795569 E−02
	C_8	2.1857714 E−03	−3.1412698 E−03	−3.3593696 E−04	2.0405896 E−04
	C_9	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00
	C_{10}	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00

Table 4.11.3
Coefficients for Eq. (4.11.5) (Cont'd)

		Shell Thickness			
D_{pj}	Coefficients	19 mm (3/4 in.)	25 mm (1 in.)	50 mm (2 in.)	
DN50 (NPS 2)	C_1	−4.0085121 E+02	3.6782666 E+00	1.0000000 E+00	...
	C_2	3.5652906 E+02	−1.2669560 E+00	0.0000000 E+00	...
	C_3	−1.3171601 E+02	4.5491492 E−01	0.0000000 E+00	...
	C_4	2.6480374 E+01	−6.2883969 E−02	0.0000000 E+00	...
	C_5	−3.1258388 E+00	3.9401350 E−03	0.0000000 E+00	...
	C_6	2.1680455 E−01	−9.3433360 E−05	0.0000000 E+00	...
	C_7	−8.1908188 E−03	0.0000000 E+00	0.0000000 E+00	...
	C_8	1.3019970 E−04	0.0000000 E+00	0.0000000 E+00	...
	C_9	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	...
	C_{10}	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	...
DN80 (NPS 3)	C_1	−1.5045135 E+03	8.1206324 E+00	−3.2789303 E+03	...
	C_2	1.4487653 E+03	−8.3943593 E+00	3.4419302 E+03	...
	C_3	−5.9846696 E+02	3.7870074 E+00	−1.5852932 E+03	...
	C_4	1.3910417 E+02	−7.0886182 E−01	4.2063167 E+02	...
	C_5	−1.9888205 E+01	6.6972430 E−02	−7.0855807 E+01	...
	C_6	1.7922925 E+00	−3.1488859 E−03	7.8593168 E+00	...
	C_7	−9.9521276 E−02	5.8511141 E−05	−5.7415834 E−01	...
	C_8	3.1164737 E−03	0.0000000 E+00	2.6647325 E−02	...
	C_9	−4.2181627 E−05	0.0000000 E+00	−7.1319265 E−04	...
	C_{10}	0.0000000 E+00	0.0000000 E+00	8.3899940 E−06	...
DN100 (NPS 4)	C_1	−3.5172282 E+00	−2.5016604 E+02	−5.3121462 E+00	...
	C_2	4.3499616 E+00	1.7178270 E+02	3.4090615 E+00	...
	C_3	−2.7157682 E−01	−4.6844914 E+01	−5.5605535 E−01	...
	C_4	1.1186450 E−02	6.6874346 E+00	4.2156128 E−02	...
	C_5	−7.1328067 E−04	−5.2507555 E−01	−1.2921987 E−03	...
	C_6	2.2962890 E−05	2.1526948 E−02	6.6740230 E−06	...
	C_7	0.0000000 E+00	−3.6091550 E−04	0.0000000 E+00	...
	C_8	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	...
	C_9	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	...
	C_{10}	0.0000000 E+00	0.0000000 E+00	0.0000000 E+00	...

4.11.9 FIGURES

Figure 4.11.1
Types of Jacketed Vessels

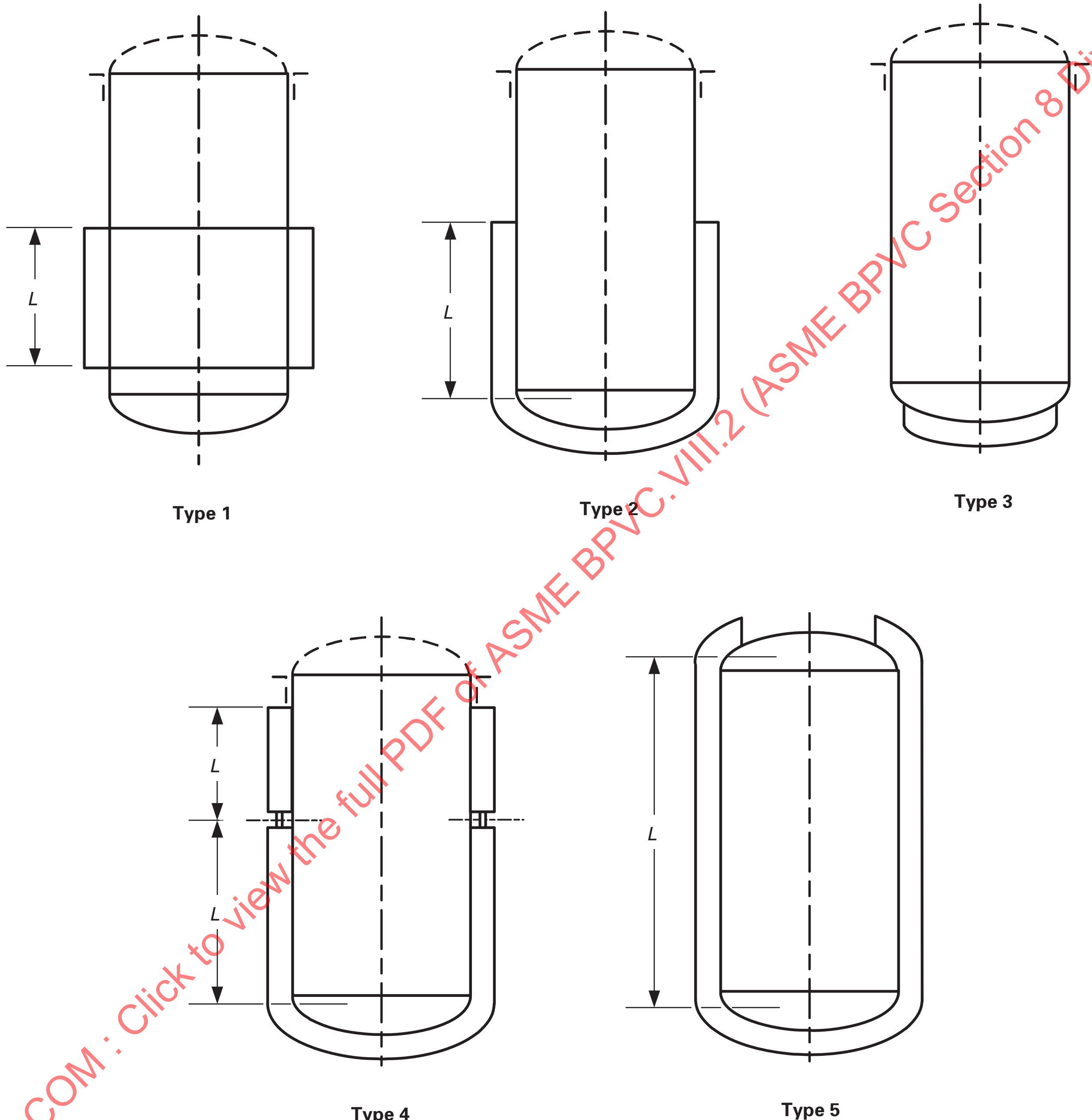
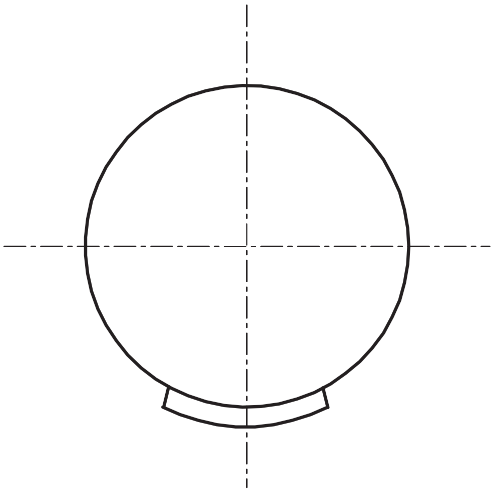
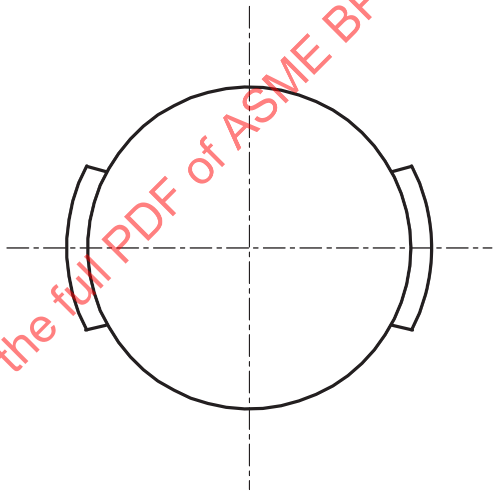


Figure 4.11.2
Types of Partial Jackets

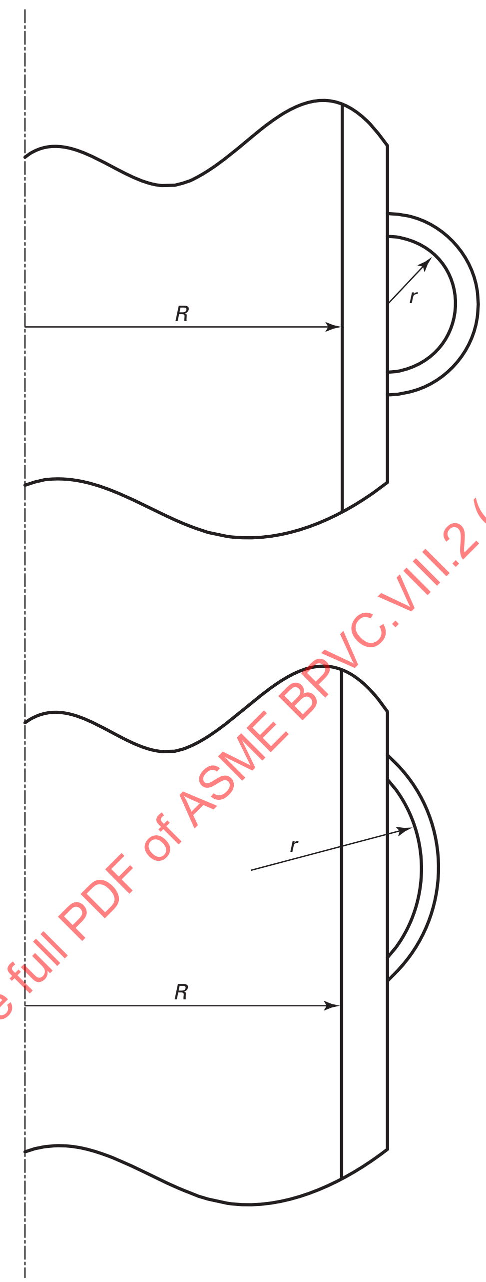


Continuous Partial Jacket



Multiple or Pod-Type Jacket

Figure 4.11.3
Half Pipe Jackets



4.12 DESIGN RULES FOR NONCIRCULAR VESSELS

4.12.1 SCOPE

4.12.1.1 The procedures in 4.12 cover the design requirements for single wall vessels having a rectangular or obround cross section. The design rules cover the walls and parts of the vessels subject to pressure stresses including stiffening, reinforcing and staying members. All other types of loadings shall be evaluated in accordance with the design-by-analysis rules of Part 5.

4.12.1.2 The design rules in this paragraph cover noncircular vessels of the types shown in Table 4.12.1. Vessel configurations other than Types 1 to 12, illustrated in Figures 4.12.1 through 4.12.13, may be used. However, in this case, the design-by-analysis rules of Part 5 shall be used.

4.12.2 GENERAL DESIGN REQUIREMENTS

4.12.2.1 In the noncircular vessel configurations covered in this paragraph, the walls of the vessel can have different thicknesses. Therefore, the design of a noncircular vessel requires an iterative approach where the vessel configuration and wall thickness are initially set and the stresses at locations on the cross section are computed and compared to allowable values. If the allowable values are exceeded, the configuration and/or wall thickness are changed, and the stresses are revaluated. This process is continued until a final configuration including wall thickness is obtained where all allowable stress requirements are satisfied.

4.12.2.2 In the design rules of this paragraph, both membrane and bending stresses shall be computed at locations on the cross section. The membrane stress is added algebraically to the bending stress at both the outermost surface of the shell plate or reinforcement (when used) and the innermost surface of the shell plate to obtain two values of total stress. The total stresses at the section shall be compared to the allowable stress.

4.12.2.3 The total stresses (membrane plus bending) at the cross section of a vessel with and without reinforcement shall be calculated as follows.

(a) For a vessel without reinforcement, the total stresses shall be determined at the inside and outside surfaces of the cross section of the shell plate.

(b) For a vessel with reinforcement, when the reinforcing member has the same allowable stress as the vessel, the total stress shall be determined at the inside and outside surfaces of the composite cross section. The appropriate value of c (the location from the neutral axis) for the composite section properties shall be used in the bending equations. The total stresses at the inside and outside surfaces shall be compared to the allowable stress.

(c) For a vessel with reinforcement, when the reinforcing member does not have the same allowable stress as the vessel, the total stresses shall be determined at the inside and outside surfaces of each component of the composite cross section. The appropriate value of c (the location from the neutral axis) for the composite section properties shall be used in the bending equations considering location of desired stress with respect to the composite section neutral axis. The total stresses at the inside and outside surfaces shall be compared to the allowable stress.

(1) For locations of stress below the neutral axis, the bending equation used to compute the stress shall be that considered acting on the inside surface.

(2) For locations of stress above the neutral axis, the bending equation used to compute the stress shall be that considered acting on the outside surface.

4.12.2.4 Particular attention shall be given to the effects of local internal and external loads and expansion differentials at design temperature, including reactions at supporting lugs, piping, and other types of attachments (see 4.12.1.1).

4.12.2.5 Except as otherwise specified in 4.12.8, vessel parts of noncircular cross section subject to external pressure shall be designed in accordance with Part 5.

4.12.2.6 The end closures for noncircular vessels covered in this paragraph shall be designed in accordance with the provisions of Part 5 except in cases where the ends are flat plates. For this case, the design rules of 4.6 shall be used except that 0.20 shall be used for the value of the C factor in all of the calculations.

4.12.2.7 The design equations in this paragraph are based on vessels in which the ratio of the long side to short-side length is greater than four. These equations are conservatively applicable to vessels of aspect ratio less than four. Vessel side plates with aspect ratios less than four are strengthened by the interaction of the end closures and may be designed in accordance with the provisions of Part 5. Short unreinforced or unstayed vessels of rectangular cross section having an aspect ratio smaller than two may be designed in accordance with 4.12.5.

4.12.2.8 Bolted full side or end plates and flanges may be provided for vessels of rectangular cross section. Many acceptable configurations are possible. Therefore, rules for specific designs are not provided, and these parts shall be designed in accordance with Part 5. The analysis of the components shall consider thermal loads, gasket reactions, bolting forces, and resulting moments, as well as pressure and other mechanical loading.

4.12.2.9 Openings may be provided in vessels of noncircular cross section as follows:

(a) Openings in noncircular vessels do not require reinforcement other than that inherent in the construction, provided they meet the conditions given in 4.5.2.

(b) Compensation for openings in noncircular vessels must account for the bending strength as well as the membrane strength of the side with the opening. In addition, openings may significantly affect the stresses in adjacent sides. Because many acceptable configurations are possible, rules for specific designs are not provided and the design shall be in accordance with Part 5.

4.12.3 REQUIREMENTS FOR VESSELS WITH REINFORCEMENT

4.12.3.1 Design rules are provided for Types 4, 5, and 6 (see Table 4.12.1) where the welded on reinforcement members are in a plane perpendicular to the long axis of the vessel; however, the spacing between reinforcing members need not be uniform. All reinforcement members attached to two opposite plates shall have the same moment of inertia. The design for any other type of reinforced rectangular cross section vessel shall be in accordance with Part 5.

4.12.3.2 For a Type 4 vessel, when the side plate thicknesses are equal, the plates may be formed to a radius at the corners. The analysis is, however, carried out in the same manner as if the corners were not rounded. For corners that are cold formed, the provisions of Part 6 shall apply. For the special case where $L_1 = 0.0$, the analysis methodology for a Type 11 vessel shall be used.

4.12.3.3 A Type 5 vessel has rounded corners and non-continuous reinforcement. If continuous reinforcement is provided that follows the contour of the vessel, the design requirements for a Type 4 vessel shall be used.

4.12.3.4 For a Type 6 vessel, the corner region consists of a flat, chamfered segment joined to the adjacent sides by curved segments with constant radii. The chamfered segments shall be perpendicular to diagonal lines drawn through the points where the sides would intersect if they were extended.

4.12.3.5 Reinforcing members shall be placed on the outside of the vessel and shall be attached to the plates of the vessel by welding on each side of the reinforcing member. For continuous reinforcement, welding may be either continuous or intermittent. The total length of intermittent welding on each side of the reinforcing member shall be not less than one-half the length being reinforced on the shell. Welds on opposite sides of the reinforcing member may be either staggered or in-line and the distance between intermittent welds shall be no more than eight times the plate thickness of the plate being reinforced. For assuring the composite section properties, for non-continuous reinforcement, the welds must be capable of developing the necessary shear (see Manual of Steel Construction, AISC, American Institute of Steel Construction).

4.12.3.6 The maximum distance between reinforcing members is computed as follows.

(a) The maximum distance between any reinforcing member centerlines is given by eq. (4.12.1). In the equations for calculating stresses for reinforced noncircular vessels, the value of p shall be taken as the sum of one-half the distances to the next reinforcing member on each side.

$$p = \min\left[p_1,\ p_2\right] \tag{4.12.1}$$

where

$$p_1 = t_1\sqrt{\frac{SJ_1}{P}} \qquad \text{for } H \geq p \tag{4.12.2}$$

$$p_1 = \frac{t_1}{\beta_1}\sqrt{\frac{SJ_1}{P}} \qquad \text{for } H < p \tag{4.12.3}$$

$$\beta_1 = \frac{H}{p_{b1}} \qquad \text{for rectangular vessels} \tag{4.12.4}$$

$$\beta_1 = \frac{2R}{p_{b1}} \quad \text{for obround vessels} \tag{4.12.5}$$

$$p_{b1} = t_1\sqrt{\frac{2.1S}{P}} \tag{4.12.6}$$

$$J_1 = -0.26667 + \frac{24.222}{\left(\beta_{1\max}\right)} - \frac{99.478}{\left(\beta_{1\max}\right)^2} + \frac{194.59}{\left(\beta_{1\max}\right)^3} - \frac{169.99}{\left(\beta_{1\max}\right)^4} + \frac{55.822}{\left(\beta_{1\max}\right)^5} \tag{4.12.7}$$

$$\beta_{1\max} = \min\left[\max\left[\beta_1, \frac{1}{\beta_1}\right], 4.0\right] \tag{4.12.8}$$

$$p_2 = t_2\sqrt{\frac{SJ_2}{P}} \quad \text{for } h \geq p \tag{4.12.9}$$

$$p_2 = \frac{t_2}{\beta_2}\sqrt{\frac{SJ_2}{P}} \quad \text{for } h < p \tag{4.12.10}$$

$$\beta_2 = \frac{h}{p_{b2}} \quad \text{for rectangular vessels} \tag{4.12.11}$$

$$\beta_2 = \frac{2L_2}{p_{b2}} \quad \text{for obround vessels} \tag{4.12.12}$$

$$J_2 = -0.26667 + \frac{24.222}{\left(\beta_{2\max}\right)} - \frac{99.478}{\left(\beta_{2\max}\right)^2} + \frac{194.59}{\left(\beta_{2\max}\right)^3} - \frac{169.99}{\left(\beta_{2\max}\right)^4} + \frac{55.822}{\left(\beta_{2\max}\right)^5} \tag{4.12.13}$$

$$\beta_{2\max} = \min\left[\max\left[\beta_2, \frac{1}{\beta_2}\right], 4.0\right] \tag{4.12.14}$$

$$p_{b2} = t_2\sqrt{\frac{2.1S}{P}} \tag{4.12.15}$$

(b) The allowable effective widths of the shell plate, w_1 and w_2, shall not be greater than the value given by eq. (4.12.16) or eq. (4.12.17) nor greater than the actual value of p if this value is less than that computed in (a). One-half of w shall be considered to be effective on each side of the reinforcing member centerline, but the effective widths shall not overlap. The effective width shall not be greater than the actual width available.

$$w_1 = \min\left[w_{\max}, p_1\right] \tag{4.12.16}$$

$$w_2 = \min\left[w_{\max}, p_2\right] \tag{4.12.17}$$

where

$$w_{\max} = \frac{t\Delta}{\sqrt{S_y}}\left(\frac{E_y}{E_{ya}}\right) \tag{4.12.18}$$

(c) At locations, other than in the corner regions where the shell plate is in tension, the effective moments of inertia I_{11} and I_{21} of the composite section (reinforcement and shell plate acting together) shall be computed based on the values of w_1 and w_2 computed in (b). The equations given in (b) do not include the effects of high-localized stresses.

In the corner regions of some Type 4 configurations, the localized stresses may significantly exceed the calculated stress. Only a very small width of the shell plate may be effective in acting with the composite section in the corner region. The localized stresses in this region shall be evaluated using the principles of Part 5.

4.12.4 REQUIREMENTS FOR VESSELS WITH STAYS

4.12.4.1 Three types of stayed construction are considered, Types 7, 8, and 11. In these types of construction the staying members may be plates welded to the side plates for the entire length of the vessel. In this case, the stay plates shall not be constructed so as to create pressure-containing partitions. Alternatively, the stays may be bars of circular cross section fastened to the side plates on a uniform pitch designed in accordance with 4.9.

4.12.4.2 The Type 12 noncircular vessel is comprised of a cylindrical shell with a single-stay plate that divides the cylinder into two compartments. The design rules ensure that the various vessel members will not be overstressed when there is full pressure in both vessel compartments or when there is full pressure in one compartment and zero pressure in the other compartment. Stresses may be computed only at the shell-plate junction since this is the location of maximum stress.

4.12.5 REQUIREMENTS FOR RECTANGULAR VESSELS WITH SMALL ASPECT RATIOS

4.12.5.1 Type 1 and Type 2 noncircular vessels with aspect ratios of L_v/H or L_v/h between 1.0 and 2.0, and with flat heads welded to the sides may be designed using the procedure in 4.12.7 except that the following plate parameters shall be utilized in the calculations.

$$J_{2s} = J_2\left(x = \frac{L_v}{H}\right) \tag{4.12.19}$$

$$J_{2l} = J_2\left(x = \frac{L_v}{h}\right) \tag{4.12.20}$$

$$J_{3s} = J_3\left(x = \frac{L_v}{H}\right) \tag{4.12.21}$$

$$J_{3l} = J_3\left(x = \frac{L_v}{h}\right) \tag{4.12.22}$$

where

$$J_2(x) = -0.65979 + 1.0052x + 0.86072x^2 - 0.82362x^3 + 0.17483x^4 \tag{4.12.23}$$

$$J_3(x) = -0.37508 + 0.66706x + 0.99709x^2 - 0.84305x^3 + 0.17483x^4 \tag{4.12.24}$$

Note in the above nomenclature, $J_{2s} = J_2\left(x = \frac{L_v}{H}\right)$ is defined as computing J_{2s} using the function $J_2(x)$ evaluated at $x = \frac{L_v}{H}$.

4.12.5.2 For vessels with aspect ratios of L_v/H or L_v/h less than 1.0, the axis of the vessel shall be rotated so that the largest dimension becomes the length L_v, and the new ratios L_v/H or L_v/h are greater than or equal to 1.0. All stresses shall be recalculated using the new orientation.

4.12.6 WELD JOINT FACTORS AND LIGAMENT EFFICIENCY

4.12.6.1 The stress calculations for the noncircular vessel shall include a weld joint factor for weld locations and ligament efficiency for those locations containing holes. In the stress calculations two factors E_m and E_b are used to account for the weld joint factor and ligament efficiency that is to be applied to the membrane and bending stresses, respectively. The weld joint factor shall be determined from 4.2 and the ligament efficiency shall be determined from 4.12.6.3. The correct combination of weld joint factor and ligament efficiencies to be used in the design is shown below.

(a) If there is not a weld or a hole pattern at the stress calculation location, then:

$$E_m = 1.0 \tag{4.12.25}$$

$$E_b = 1.0 \tag{4.12.26}$$

(b) If there is a weld, and there is not a hole pattern at the stress calculation location, then:

$$E_m = E \tag{4.12.27}$$

$$E_b = E \tag{4.12.28}$$

(c) If there is not a weld, and there is a hole pattern at the stress calculation location, then:

$$E_m = e_m \tag{4.12.29}$$

$$E_b = e_b \tag{4.12.30}$$

(d) If there is a weld and a hole pattern at the stress calculation location, then:

(1) If e_m and e_b are less than the joint efficiency, E, which would be used if there were no ligaments in the plate, then use eqs. (4.12.29) and (4.12.30).

(2) If e_m and e_b are greater than the weld joint factor, E, which would be used if there were no ligaments in the plate, then use eqs. (4.12.27) and (4.12.28).

4.12.6.2 Cases may arise where application of a weld joint factor, E, at nonwelded locations results in unnecessarily increased plate thicknesses. If the butt weld occurs at one of the locations for which equations are provided in this paragraph, then no relief can be provided. However, if the weld occurs at some intermediate location, it is permissible to calculate the bending stress at the weld location and compare it to the allowable stress considering the weld joint factor in the calculation. An alternate location for computing stresses is provided for some of the noncircular geometry types, and is identified as "Maximum Membrane and Bending Stresses - Defined Locations" in the stress calculation tables. The value X of Y or to be used in the equations is the distance from the midpoint of the side to the location of the weld joint.

4.12.6.3 The ligament efficiency factors e_m and e_b, for membrane and bending stresses, respectively, shall only be applied to the calculated stresses for the plates containing the ligaments.

(a) For the case of uniform diameter holes, the ligament efficiency factors e_m and e_b shall be the same and computed in accordance with 4.10.

(b) For the case of multi-diameter holes, the neutral axis of the ligament may no longer be at mid-thickness of the plate; the bending stress is higher at one of the plate surfaces than at the other surface. Therefore, for multi-diameter holes, the ligament efficiency factor shall be computed using the following equations.

(1) The ligament efficiency of plate with multi-diameter holes subject to membrane stress is computed as follows.

$$e_m = \frac{\left(p_h - D_E\right)}{P_h} \tag{4.12.31}$$

where

$$D_E = \frac{1}{t}\left(d_0 T_0 + d_1 T_1 + d_2 T_2 + ... + d_n T_n\right) \tag{4.12.32}$$

(2) The ligament efficiency and location from the neutral axis of a plate with multi-diameter holes (see Figure 4.12.14) subject to bending stress is computed as follows.

$$e_b = \frac{\left(p_h - D_E\right)}{p_h} \tag{4.12.33}$$

where

$$D_E = p_h - \frac{6I_E}{t^2 c_E} \tag{4.12.34}$$

$$\begin{aligned} I_E = {} & \frac{1}{12}\left(b_0 T_0^3 + b_1 T_1^3 + b_2 T_2^3 + ... + b_n T_n^3\right) + \\ & b_0 T_0\left(\frac{T_0}{2} + T_1 + T_2 + ... + T_n - \overline{X}\right)^2 + b_1 T_1\left(\frac{T_1}{2} + T_2 + ... + T_n - \overline{X}\right)^2 + \\ & b_2 T_2\left(\frac{T_2}{2} + ... + T_n - \overline{X}\right)^2 + ... + b_n T_n\left(\overline{X} - \frac{T_n}{2}\right)^2 \end{aligned} \tag{4.12.35}$$

$$\overline{X} = \begin{bmatrix} b_0 T_0\left(\frac{T_0}{2} + T_1 + T_2 + ... + T_n\right) + \\ b_1 T_1\left(\frac{T_1}{2} + T_2 + ... + T_n\right) + \\ b_2 T_2\left(\frac{T_2}{2} + ... + T_n\right) + ... + b_n T_n\left(\frac{T_n}{2}\right) \end{bmatrix} \cdot \left[b_0 T_0 + b_1 T_1 + b_2 T_2 + ... + b_n T_n\right]^{-1} \tag{4.12.36}$$

where

$$b_0 = p_h - d_0 \tag{4.12.37}$$

$$b_1 = p_h - d_1 \tag{4.12.38}$$

$$b_2 = p_h - d_2 \tag{4.12.39}$$

$$b_n = p_h - d_n \tag{4.12.40}$$

$$c_E = \max\left[\bar{X}, \left(t - \bar{X}\right)\right] \tag{4.12.41}$$

If T_o is measured from the inside surface, then

$$c_i = \bar{X} \tag{4.12.42}$$

$$c_o = t - \bar{X} \tag{4.12.43}$$

If T_o is measured from the outside surface, then

$$c_i = t - \bar{X} \tag{4.12.44}$$

$$c_o = \bar{X} \tag{4.12.45}$$

(c) Rows of holes may be located in regions of relatively low bending moments to keep the required plate thickness to a minimum. Therefore, it is permissible to calculate the stresses at the centerline of each row of holes closest to the locations where the highest bending moments occurs (i.e., at the midpoint of the sides and at the corners). If the diameter of all the holes is not the same, the stresses must be calculated for each set of e_m and e_b values.

(d) The applied gross area stresses may be calculated using the same procedure as for calculating the stresses at a weld joint (see 4.12.3.2). The value of X or Y to be used in the equations is the distance from the midpoint of the side to the plane containing the centerlines of the holes.

4.12.7 DESIGN PROCEDURE

4.12.7.1 A procedure that can be used to design a noncircular vessel subject to internal pressure is shown below.

Step 1. Determine the design pressure and temperature.

Step 2. Determine the configuration of the noncircular vessel by choosing a Type from Table 4.12.1.

Step 3. Determine the initial configuration (i.e., width, height, length, etc.) and wall thicknesses of the pressure-containing plates.

(a) If the vessel has stiffeners, then determine the spacing (see 4.12.3) and size of the stiffeners.

(b) If the vessel has stays, then determine the stay type and configuration (see 4.12.4), and check the stay plate welds using 4.2.

(c) If the vessel aspect ratio is less than two, then determine the plate parameters in 4.12.5.

Step 4. Determine the location of the neutral axis from the inside and outside surfaces.

(a) If the section under evaluation has stiffeners, then c_i and c_o are determined from the cross section of the combined plate and stiffener section using strength of materials concepts.

(b) If the section under evaluation has multi-diameter holes, then c_i and c_o are determined from 4.12.6.3.

(c) If the section under evaluation does not have a stiffener, does not have holes, or has uniform diameter holes, then $c_i = c_o = t/2$ where t is the thickness of the plate.

Step 5. Determine the weld joint factor and ligaments efficiencies, as applicable (see 4.12.6), and determine the factors E_m or E_b.

Step 6. Complete the stress calculation for the selected noncircular vessel Type (see Table 4.12.1), and check the acceptance criteria. If the criteria are satisfied, then the design is complete. If the criteria are not satisfied, then modify the plate thickness and/or stiffener size and go to Step 3 and repeat the calculation. Continue this process until a design is achieved that satisfies the acceptance criteria.

4.12.7.2 If the vessel is subject to external pressure, the additional requirements of 4.12.8 shall be satisfied.

4.12.8 NONCIRCULAR VESSELS SUBJECT TO EXTERNAL PRESSURE

4.12.8.1 Rectangular vessel Types 1 and 2 subject to external pressure shall meet the following requirements.

(a) The stresses shall be calculated in accordance with Tables 4.12.2 and 4.12.3 except that the design external pressure shall be substituted for P. These computed stresses shall meet the acceptance criteria defined in these tables.

(b) The four side plates and the two end plates shall be checked for stability in accordance with eq. (4.12.46). The required calculations for S_{mA}, S_{mB}, S^*_{crA}, S^{**}_{crA}, S^*_{crB} and S^{**}_{crB} are shown in Table 4.12.15. In the equations, the subscript A is used to identify stress or load acting in a direction parallel to the long dimension of the panel being considered and the subscript B is used to identify stress or load acting in a direction parallel to the short dimension of the panel being considered. In the calculations, the plate thickness t shall be adjusted if the plate is perforated. This can be accomplished by multiplying t by e_m in the equations for S_{mA} and S_{mB}. It is not necessary to make this adjustment in the equations for S_{crA} and S_{crB}.

$$\frac{2S_{mA}}{S_{crA}} + \frac{2S_{mB}}{S_{crB}} \leq 1.0 \tag{4.12.46}$$

where

$$S_{crA} = S^*_{crA} \qquad \text{when } S^*_{crA} \leq \frac{S_y}{2} \tag{4.12.47}$$

$$S_{crA} = S^{**}_{crA} \qquad \text{when } S^*_{crA} > \frac{S_y}{2} \tag{4.12.48}$$

$$S_{crB} = S^{*}_{crB} \qquad \text{when } S^{*}_{crB} \leq \frac{S_y}{2} \tag{4.12.49}$$

$$S_{crB} = S^{**}_{crB} \qquad \text{when } S^{*}_{crB} > \frac{S_y}{2} \tag{4.12.50}$$

(c) In addition to checking each of the four side plates and the two end plates for stability, the cross section shall be checked for column stability using the following equations. eq. (4.12.52) applies to vessels where the long plate thicknesses are equal. If the thicknesses are not equal, replace $2t_2$ with $(t_2 + t_{22})$ in the equation.

$$\frac{S_a}{F_a} + \frac{S_b}{S\left(1 - \frac{S_a}{F^{*}_e}\right)} \leq 1.0 \tag{4.12.51}$$

where

$$S_a = \frac{P_e(h + 2t_1)(H + 2t_2)}{2t_1(H + 2t_2) + 2t_2(h + 2t_1)} \tag{4.12.52}$$

$$S_b = \frac{\left[P_e(h + 2t_1)(H + 2t_2)\bar{y}\right] c_e}{I_e} \tag{4.12.53}$$

$$C_c = \sqrt{\frac{2\pi^2 E_y}{S_y}} \tag{4.12.54}$$

$$F_a = \frac{\left[1 - \frac{1}{2C_c^2}\left(\frac{2L_v}{R_{ge}}\right)^2\right] S_y}{\frac{5}{3} + \frac{3}{8C_c}\left(\frac{2L_v}{R_{ge}}\right) - \frac{1}{8C_c^3}\left(\frac{2L_v}{R_{ge}}\right)^3} \qquad \text{when } \frac{2L_v}{R_{ge}} \leq C_c \tag{4.12.55}$$

$$F_a = \frac{12\pi^2 E_y}{23\left(\frac{2L_v}{R_{ge}}\right)^2} \qquad \text{when } \frac{2L_v}{R_{ge}} > C_c \tag{4.12.56}$$

$$F^{*}_e = \frac{12\pi^2 E_y}{23\left(\frac{2L_v}{R_{ge}}\right)^2} \tag{4.12.57}$$

4.12.9 RECTANGULAR VESSELS WITH TWO OR MORE COMPARTMENTS OF UNEQUAL SIZE

Typical rectangular cross section vessels having unequal compartments are shown in Figure 4.12.15. These types of vessels shall be qualified using either of the two methods shown below.

(a) A design can be qualified by selecting the compartment having the maximum dimensions and analyzing the vessel as a Type 7 for the case of a two-compartment vessel or Type 8 for the case of a vessel with more than two compartments. For example, if the vessel has two unequal compartments, use the geometry for a Type 7 with each compartment having the maximum dimension of the actual vessel. For a vessel with more than two compartments, use the geometry for a Type 8 with three compartments having the maximum dimensions of the actual vessel. Thus a five or six compartment vessel would be analyzed as if it had only three compartments.

(b) The vessel can be designed in accordance with Part 5.

4.12.10 FABRICATION

4.12.10.1 Provided the requirements of the applicable Parts of this Division are satisfied, fabrication methods other than welding are permitted.

4.12.10.2 Category A joints may be of Type 3 when the thickness does not exceed 16 mm (0.625 in.).

4.12.11 NOMENCLATURE

4.12.11.1 The nomenclature used in this paragraph is defined below except for computed stresses. The nomenclature for computed stress is defined in 4.12.11.2.

A_1 = cross-sectional area of the reinforcing member associated with t_1.
A_2 = cross-sectional area of the reinforcing member associated with t_2.
b = unit width per cross section. In the equations for the areas, moments of inertia, and bending moments for all vessel configurations without external reinforcements are given for cross sections with a unit width.
C = stress factor for braced and stayed surfaces (see Table 4.9.1).
c_e = location from the neutral axis to the outer most surface of a composite section associated t_v, R, A_R.
c_i = distance from the neutral axis to the inside surface of the shell or reinforcing member on the short side, long side, curved element, or stay plate as applicable (e.g., for a plate with uniform holes without a stiffener, $c = t/2$ where t is the thickness of the plate); the sign of this parameter is always positive (the sign for the bending stress is included in the applicable equation).
c_o = distance from the neutral axis to the outside surface of the shell or reinforcing member on the short side, long side, curved element, or stay plate as applicable (e.g., for a plate with uniform holes without a stiffener, $c = t/2$ where t is the thickness of the plate); the sign of this parameter is always positive (the sign for the bending stress is included in the applicable equation).
Δ = effective width coefficient (see Table 4.12.14)
d_j = hole diameter jth location.
e_b = bending stress ligament efficiency of a hole pattern.
e_m = membrane stress ligament efficiency of a hole pattern.
E = weld joint factor.
E_b = factor applied to the bending stress to account for a ligament or weld joint factor.
E_m = factor applied to the membrane stress to account for a ligament or weld joint factor.
E_y = Young's Modulus from Annex 3-E at design temperature.
E_{ya} = Young's Modulus from Annex 3-E at ambient temperature.
H = inside length of the short side of a rectangular vessel. For Types 5 and 6, $H = 2(L_1 + L_{11})$ and for Type 10 $H = 2R$.
H_1 = centroidal length of the reinforcing member on the short side of a rectangular vessel.
h = inside length of the long side of a rectangular vessel. For Types 5 and 6, $h = 2(L_2 + L_{21})$ and for Type 10 $H = 2L_2$.
h_1 = centroidal length of the reinforcing member on the long side of a rectangular vessel.
I_e = least moment of inertia of noncircular cross-section vessel.
I_1 = moment of inertia of strip thickness t_1.
I_2 = moment of inertia of strip thickness t_2.
I_{22} = moment of inertia of strip thickness t_{22}.
I_3 = moment of inertia of strip thickness t_3.
I_{11} = moment of inertia of combined reinforcing member and effective with of plate w of thickness t_1.
I_{21} = moment of inertia of combined reinforcing member and effective with of plate w of thickness t_2.
L_1 = half-length of the short side of a rounded vessel without reinforcement or the half-length of reinforcement on the short side for a reinforced rectangular vessel.
L_2 = half-length of the long side of a rounded vessel without reinforcement or the half-length of reinforcement on the long side for a reinforced rectangular vessel.
L_3 = half-length dimension of the short side of Type 5 and Type 6 rectangular vessel.
L_4 = half-length dimension of the long side of Type 5 and Type 6 rectangular vessel.
L_{11} = length measured from the edge of the reinforcement to the end of the straight side of the short side of a Type 5 and Type 6 rectangular vessel.
L_{21} = length measured from the edge of the reinforcement to the end of the straight side of the long side of a Type 5 and Type 6 rectangular vessel.
L_v = length of the vessel.
M_A = bending moment at the mid-side of the long side, a positive sign indicates a compressive stress on the outside surface of the plate.

N = rectangular vessel parameter.
P = internal design pressure.
P_1 = internal design pressure of a two compartment vessel where $P_1 \geq P_2$.
P_2 = internal design pressure of a two compartment vessel where $P_1 \geq P_2$.
p = distance between reinforcing members; plate width between edges of reinforcing members
p_h = pitch distance between holes.
R = inside radius.
R_{ge} = least radius of gyration of a noncircular cross-section vessel.
r = radius to centroidal axis of reinforcement member on obround vessel.
S = allowable stress from Annex 3-A at the design temperature.
S_y = yield stress at the design temperature evaluated in accordance Annex 3-D.
t = plate thickness.
t_1 = thickness of the short-side plate.
t_2 = thickness of the long-side plate.
t_3 = thickness or diameter of staying member.
t_4 = thickness or diameter of staying member.
t_5 = thickness of end closure plate or head of vessel.
t_{22} = thickness of the thicker long-side plate.
T_j = hole depth jth location.
v = Poisson's ratio.
w = width of plate included in the moment of inertia calculation of the reinforced section.
$\overline{y}$ = distance from geometric center of end plate to centroid of cross-sectional area of a rectangular vessel.

4.12.11.2 The nomenclature for all computed stress quantities is shown in the following tables and figures.

(a) For Types 1, 4, 7, and 8 noncircular vessels see Tables 4.12.2, 4.12.5, 4.12.8, and 4.12.9 and Figures 4.12.1, 4.12.4, 4.12.8, and 4.12.9

(b) For the Type 2 noncircular vessel, see Table 4.12.3 and Figure 4.12.2

(c) For the Type 3 noncircular vessel, see Table 4.12.4 and Figure 4.12.3

(d) For the Type 5 noncircular vessel, see Table 4.12.6 and Figure 4.12.5

(e) For the Type 6 noncircular vessel, see Table 4.12.7 and Figures 4.12.6 and 4.12.7

(f) For the Types 9, 10, and 11 noncircular vessels, see Tables 4.12.10, 4.12.11. and 4.12.12 and Figures 4.12.10, 4.12.11, and 4.12.12

(g) For the Type 12 noncircular vessels, see Table 4.12.13 and Figure 4.12.13

(h) For the compressive stress calculations for Type 1 and 2 see Table 4.12.15

4.12.12 TABLES

Table 4.12.1
Noncircular Vessel Configurations and Types

Configuration	Type	Figure Number	Table Containing Design Rules
Rectangular cross-section in which the opposite sides have the same wall thickness. Two opposite sides may have a wall thickness different than that of the other two opposite sides.	1	4.12.1	4.12.2
Rectangular cross-section in which two opposite members have the same thickness and the other two members have two different thicknesses.	2	4.12.2	4.12.3
Rectangular cross section having uniform wall thickness and corners bent to a radius. For corners which are cold formed, the provisions Part 6 shall apply	3	4.12.3	4.12.4
Rectangular cross-section similar to Type 1 but reinforced by stiffeners welded to the sides.	4	4.12.4	4.12.5
Rectangular cross-section similar to Type 3 but externally reinforced by stiffeners welded to the flat surfaces of the vessel.	5	4.12.5	4.12.6

Table 4.12.1
Noncircular Vessel Configurations and Types (Cont'd)

Configuration	Type	Figure Number	Table Containing Design Rules
Rectangular cross section with chamfered corner segments (octagonal cross-section) joined to the adjacent sides by small curved segments with constant radii and reinforced by stiffeners welded to the flat surfaces of the vessel.	6	4.12.6, 4.12.7	4.12.7
Rectangular cross section similar to Type 1 but having two opposite sides stayed at mid-length.	7	4.12.8	4.12.8
Rectangular cross section similar to Type 1 but having two opposite sides stayed at the third points.	8	4.12.9	4.12.9
Obround cross-section in which the opposite sides have the same wall thickness. The flat sidewalls may have a different thickness than the semicylindrical parts.	9	4.12.10	4.12.10
Obround cross-section similar to Type 9 but reinforced by stiffeners welded to the curved and flat surfaces of the vessel.	10	4.12.11	4.12.11
Obround cross-section similar to Type 9 but having the flat side plates stayed at mid-length.	11	4.12.12	4.12.12
Circular section with a single-stay plate	12	4.12.13	4.12.13

Table 4.12.2
Stress Calculations and Acceptance Criteria for Type 1 Noncircular Vessels (Rectangular Cross Section)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{Ph}{2t_1E_m}$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{PbJ_{2s}c_i}{12I_1E_b}\left[-1.5H^2 + h^2\left(\frac{1+\alpha^2K}{1+K}\right)\right]$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2J_{3s}c_i}{12I_1E_b}\left[\frac{1+\alpha^2K}{1+K}\right]$$

$$S_m^l = \frac{PH}{2t_2E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2J_{2l}c_i}{12I_2E_b}\left[-1.5 + \left(\frac{1+\alpha^2K}{1+K}\right)\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2J_{3l}c_i}{12I_2E_b}\left[\frac{1+\alpha^2K}{1+K}\right]$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^{sX} = -S_{bo}^{sX}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{12I_1E_b}\left[-1.5H^2 + h^2\left(\frac{1+\alpha^2K}{1+K}\right) + 6X^2\right]$$

$$S_{bi}^{lY} = -S_{bo}^{lY}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{12I_2E_b}\left[-1.5h^2 + h^2\left(\frac{1+\alpha^2K}{1+K}\right) + 6Y^2\right]$$

Table 4.12.2
Stress Calculations and Acceptance Criteria for Type 1 Noncircular Vessels (Rectangular Cross Section) (Cont'd)

Equation Constants

$I_1 = \frac{bt_1^3}{12}$	$J_{2s} = 1.0$ (see 4.12.5 for exception)
$I_2 = \frac{bt_2^3}{12}$	$J_{3s} = 1.0$ (see 4.12.5 for exception)
$K = \frac{I_2}{I_1}\alpha$	$J_{2l} = 1.0$ (see 4.12.5 for exception)
$\alpha = \frac{H}{h}$	$J_{3l} = 1.0$ (see 4.12.5 for exception)

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \leq S$	$S_m^l \leq S$
$S_m^s + S_{bi}^{sC} \leq 1.5S$	$S_m^l + S_{bi}^{lA} \leq 1.5S$
$S_m^s + S_{bo}^{sC} \leq 1.5S$	$S_m^l + S_{bo}^{lA} \leq 1.5S$
$S_m^s + S_{bi}^{sB} \leq 1.5S$	$S_m^l + S_{bi}^{lB} \leq 1.5S$
$S_m^s + S_{bo}^{sB} \leq 1.5S$	$S_m^l + S_{bo}^{lB} \leq 1.5S$

Acceptance Criteria — Defined Locations for Stress Calculation

$S_m^s + S_{bi}^{sX} \leq 1.5S$	$S_m^l + S_{bi}^{lY} \leq 1.5S$
$S_m^s + S_{bo}^{sX} \leq 1.5S$	$S_m^l + S_{bo}^{lY} \leq 1.5S$

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB} = bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC} = bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX} = bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l = membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st} = membrane stress in the stay bar or plate, as applicable.

Table 4.12.3
Stress Calculations and Acceptance Criteria for Type 2 Noncircular Vessels (Rectangular Cross Section With Unequal Side Plate Thicknesses)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{Ph}{2t_1 E_m}$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3s} c_i}{4NI_1 E_b}\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right]$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3s} c_i}{4NI_1 E_b}\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right]$$

$$S_m^{l2} = \frac{P}{8NHt_2 E_m}\left\{4NH^2 - 2h^2\left[\left(K_2 + k_2\right) - k_1\left(K_1 + k_2\right) + \alpha^2 k_2\left(K_2 - K_1\right)\right]\right\}$$

$$S_{bi}^{lD} = -S_{bo}^{lD}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{2l} c_i}{8NI_2 E_b}\left\{2\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right] - N\right\}$$

$$S_{bi}^{lC} = -S_{bo}^{lC}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3l} c_i}{4NI_2 E_b}\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right]$$

$$S_m^{l22} = \frac{P}{8NHt_{22} E_m}\left\{4NH^2 - 2h^2\left[-\left(K_2 + k_2\right) + k_1\left(K_1 + k_2\right) - \alpha^2 k_2\left(K_2 - K_1\right)\right]\right\}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{2l} c_i}{8NI_{22} E_b}\left\{2\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right] - N\right\}$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3l} c_i}{4NI_{22} E_b}\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right]$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^{lY2} = -S_{bo}^{lY2}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{2I_2 E_b}\left\{\frac{h^2}{2N}\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right] - \frac{h^2}{4} + Y_2^2\right\}$$

$$S_{bi}^{lY22} = -S_{bi}^{lY22}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{2I_{22} E_b}\left\{h^2\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right] - \frac{h^2}{4} + Y_{22}^2\right\}$$

Equation Constants

$$I_1 = \frac{bt_1^3}{12}$$

$$I_{22} = \frac{bt_{22}^3}{12}$$

$$k_1 = \frac{I_{22}}{I_2}$$

$$k_2 = \frac{I_{22}\alpha}{I_1}$$

$$K_1 = 2k_2 + 3$$

$$K_2 = 3k_1 + 2k_2$$

$$N = K_1 K_2 - k_2^2$$

$J_{2s} = 1.0$ (see 4.12.5 for exception)
$J_{3s} = 1.0$ (see 4.12.5 for exception)
$J_{2l} = 1.0$ (see 4.12.5 for exception)
$J_{3l} = 1.0$ (see 4.12.5 for exception)

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^{l2} \le S$	$S_m^{l22} \le S$
$S_m^s + S_{bi}^{sB} \le 1.5S$	$S_m^{l2} + S_{bi}^{lC} \le 1.5S$	$S_m^{l22} + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$	$S_m^{l2} + S_{bo}^{lC} \le 1.5S$	$S_m^{l22} + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^{l2} + S_{bi}^{lD} \le 1.5S$	$S_m^{l22} + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^{l2} + S_{bo}^{lD} \le 1.5S$	$S_m^{l22} + S_{bo}^{lB} \le 1.5S$

Table 4.12.3 Stress Calculations and Acceptance Criteria for Type 2 Noncircular Vessels (Rectangular Cross Section With Unequal Side Plate Thicknesses) (Cont'd)

Acceptance Criteria — Defined Locations for Stress Calculation

Not Applicable	$S_m^{l2} + S_{bi}^{lY_2} \le 1.5S$ $S_m^{l2} + S_{bo}^{lY_2} \le 1.5S$	$S_m^{l22} + S_{bi}^{lY_{22}} \le 1.5S$ $S_m^{l22} + S_{bo}^{lY_{22}} \le 1.5S$

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.

S_{bi}^{sB} , S_{bo}^{sB} = bending stress in the short side at point B on the inside and outside surfaces, respectively.

S_{bi}^{sC} , S_{bo}^{sC} = bending stress in the short side at point C on the inside and outside surfaces, respectively.

S_m^{l2} = membrane stress in the long side with thickness t_2.

S_{bi}^{lD} , S_{bo}^{lD} = bending stress in the long side at point D on the inside and outside surfaces, respectively.

S_{bi}^{lC} , S_{bo}^{lC} = bending stress in the long side at point C on the inside and outside surfaces, respectively.

$S_{bi}^{lY_2}$, $S_{bo}^{lY_2}$ = bending stress in the long side at a point defined by Y_2 on the inside and outside surfaces, respectively.

S_m^{l22} = membrane stress in the long side with thickness t_{22}.

S_{bi}^{lD} , S_{bo}^{lD} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lC} , S_{bo}^{lC} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

$S_{bi}^{lY_{22}}$, $S_{bo}^{lY_{22}}$ = bending stress in the long side at a point defined by Y_{22} on the inside and outside surfaces, respectively.

Table 4.12.4 Stress Calculations and Acceptance Criteria for Type 3 Noncircular Vessels (Chamfered Rectangular Cross Section)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{P(R + L_2)}{t_1 E_m}$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{2I_1 E_b}\left[2M_A + P\left(2RL_2 - 2RL_1 + L_2^2\right)\right]$$

$$S_{bi}^{sD} = -S_{bo}^{sD}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{2I_1 E_b}\left[2M_A + P\left(L_2^2 + 2RL_2 - 2RL_1 - L_1^2\right)\right]$$

$$S_m^l = \frac{P(R + L_1)}{t_1 E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{la}\left(\frac{c_i}{c_o}\right) = \frac{bM_A c_i}{I_1 E_b}$$

$$S_{bi}^{lB} = -S_{bo}^{lb}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{2I_1 E_b}\left[2M_A + PL_2^2\right]$$

$$S_m^c = \frac{P}{t_1 E_m}\left(R + \sqrt{L_2^2 + L_1^2}\right)$$

$$S_{bi}^{cBC} = -S_{bo}^{cBC}\left(\frac{c_i}{c_o}\right) = \frac{bM_r c_i}{I_1 E_b}$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^{sX} = -S_{bi}^{sX}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{I_1 E_b}\left[M_A + \frac{P}{2}\left(L_2^2 + 2RL_2 - 2RL_1 - L_1^2 + X^2\right)\right]$$

$$S_{bi}^{lY} = -S_{bi}^{lY}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{I_1 E_b}\left[M_A + \frac{PY^2}{2}\right]$$

Table 4.12.4
Stress Calculations and Acceptance Criteria for Type 3 Noncircular Vessels (Chamfered Rectangular Cross Section) (Cont'd)

Equation Constants

$$I_1 = \frac{bt_1^3}{12}$$

$$M_A = \frac{-PL_1^2\left(6\phi^2\alpha_3 - 3\pi\phi^2 + 6\phi^2 + \alpha_3^3 + 3\alpha_3^2 - 6\phi - 2 + 1.5\pi\phi\alpha_3^2 + 6\phi\alpha_3\right)}{3\left(2\alpha_3 + \pi\phi + 2\right)}$$

$$\phi = \frac{R}{L_1}$$

$$\alpha_3 = \frac{L_2}{L_1}$$

$$M_r = M_A + P\left(R\left\{L_2\cos\theta - L_1\left(1 - \sin\theta\right)\right\} + \frac{L_2^2}{2}\right)$$

M_r is a maximum at $\theta = \arctan\left(\frac{L_1}{L_2}\right)$

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^l \le S$	$S_m^c \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$	$S_m^c + S_{bi}^{cBC} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$	$S_m^c + S_{bo}^{cBC} \le 1.5S$
$S_m^s + S_{bi}^{sD} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$	
$S_m^s + S_{bo}^{sD} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$	

Acceptance Criteria — Defined Locations for Stress Calculation

$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lX} \le 1.5S$	Not Applicable
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lX} \le 1.5S$	

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.
S_{bi}^{sC}, S_{bo}^{sC} = bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sD}, S_{bo}^{sD} = bending stress in the short side at point D on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX} = bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l = membrane stress in the long side.
S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^c = membrane stress in the circular arc between B and C.
S_{bi}^{cBC}, S_{bo}^{cBC} = bending stress in the circular arc between B and C on the inside and outside surfaces, respectively.

Table 4.12.5
Stress Calculations and Acceptance Criteria for Type 4 Noncircular Vessels (Reinforced Rectangular Cross Section)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{Php}{2(A_1 + t_1 p)E_m}$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{Ppc_i}{24I_{11}E_b}\left[-3H^2 + 2h^2\left(\frac{1+\alpha_1^2 k}{1+k}\right)\right]$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Ph^2pc_i}{12I_{11}E_b}\left[\frac{1+\alpha_1^2 k}{1+k}\right]$$

$$S_m^l = \frac{PHp}{2(A_2 + t_2 p)E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Ph^2pc_i}{24I_{21}E_b}\left[-3 + 2\left(\frac{1+\alpha_1^2 k}{1+k}\right)\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Ph^2pc_i}{12I_{21}E_b}\left[\frac{1+\alpha_1^2 k}{1+k}\right]$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^{sX} = -S_{bo}^{sX}\left(\frac{c_i}{c_o}\right) = \frac{Ppc_i}{24I_{11}E_b}\left[-3H^2 + 2h^2\left(\frac{1+\alpha_1^2 k}{1+k}\right) + 12X^2\right]$$

$$S_{bi}^{lY} = -S_{bo}^{lY}\left(\frac{c_i}{c_o}\right) = \frac{Ph^2pc_i}{24I_{21}E_b}\left[-3 + 2\left(\frac{1+\alpha_1^2 k}{1+k}\right) + \frac{12Y^2}{h^2}\right]$$

Equation Constants

$k = \frac{I_{21}}{I_{11}}\alpha_1$	$\alpha_1 = \frac{H_1}{h_1}$

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$

Acceptance Criteria — Defined Locations for Stress Calculation

$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lY} \le 1.5S$
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lY} \le 1.5S$

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB} = bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC} = bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX} = bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l = membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st} = membrane stress in the stay bar or plate, as applicable.

Table 4.12.6
Stress Calculations and Acceptance Criteria for Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{P\left(R + L_2 + L_{21}\right)}{t_1 E_m}$$

$$S_{bi}^{sE} = -S_{bo}^{sE}\left(\frac{c_i}{c_o}\right) = \frac{C_i}{I_1 E_b}\left[M_A + Pp\left\{\frac{\left(L_2 + L_{21}\right)^2}{2} + 2R\left(L_2 + L_{21} - L_1 - L_{11}\right)\right\}\right]$$

$$S_{bi}^{sF} = -S_{bo}^{sF}\left(\frac{c_i}{c_o}\right) = \frac{C_i}{I_1 E_b}\left[M_A + \frac{Pp}{2}\left\{\begin{array}{c} L_2^2 + 2L_2L_{21} + L_{21}^2 - 2L_1L_{11} - L_{11}^2 + \\ 2R\left(L_2 + L_{21} - L_1 - L_{11}\right)\end{array}\right\}\right]$$

$$S_{bi}^{sG} = -S_{bo}^{sG}\left(\frac{c_i}{c_o}\right) = \frac{C_i}{I_{11} E_b}\left[M_A + \frac{Pp}{2}\left\{\left(L_2 + L_{21}\right)^2 + 2R\left(L_2 + L_{21} - L_1 - L_{11}\right) - \left(L_1 + L_{11}\right)^2\right\}\right]$$

$$S_m^l = \frac{P\left(R + L_1 + L_{11}\right)}{t_2 E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{M_A c_i}{I_{21} E_b}$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_2 E_b}\left[M_A + \frac{PpL_2^2}{2}\right]$$

$$S_{bi}^{lC} = -S_{bo}^{lC}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_2 E_b}\left[M_A + \frac{Pp\left(L_2 + L_{21}\right)^2}{2}\right]$$

$$S_m^c = \frac{P}{t_1 E_m}\left(R + \sqrt{\left(L_2 + L_{21}\right)^2 + \left(L_2 + L_{11}\right)^2}\right)$$

$$S_{bi}^{cD} = -S_{bo}^{cD}\left(\frac{c_i}{c_o}\right) = \frac{M_r c_i}{I_1 E_b}$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^{sX} = -S_{bo}^{sX}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{11} E_b}\left[M_A + \frac{Pp}{2}\left\{\begin{array}{c}\left(L_2 + L_{21}\right)^2 + 2R\left(L_2 + L_{21} - L_1 - L_{11}\right) - \\ \left(L_1 + L_{11}\right)^2 + X^2\end{array}\right\}\right] \qquad X \le L_1$$

$$S_{bi}^{sX} = -S_{bo}^{sX}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[M_A + \frac{Pp}{2}\left\{\begin{array}{c} L_2^2 + 2L_2L_{21} + L_{21}^2 - 2L_1L_{11} - L_{11}^2 + \\ 2R\left(L_2 + L_{21} - L_1 - L_{11}\right) + X^2\end{array}\right\}\right] \qquad X > L_1$$

$$S_{bi}^{lY} = -S_{bo}^{lY}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{21} E_b}\left[M_A + \frac{PpY^2}{2}\right] \qquad Y \le L_2$$

$$S_{bi}^{lY} = -S_{bo}^{lY}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_2 E_b}\left[M_A + \frac{PpY^2}{2}\right] \qquad Y > L_2$$

Table 4.12.6
Stress Calculations and Acceptance Criteria for Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners) (Cont'd)

Equation Constants

$$I_1 = \frac{p t_1^3}{12}$$

$$I_2 = \frac{p t_2^3}{12}$$

$$M_A = Pp \frac{\left[\begin{array}{c} -3RL_2\left(4R + \pi L_2\right) - L_{21}\left(12R^2 + 3\pi R L_{21} + 2L_{21}^2\right) + 12RL_{11}^2 - \\ 6L_2L_{21}\left(L_2 + L_{21} + \pi R + 2L_{11}\right) - 6L_2L_{11}\left(2R + L_2\right) - 6L_{21}L_{11}\left(2R + L_{21}\right) + \\ 6L_1L_{11}\left(2R + L_{11}\right) + 6R^2\left(\pi - 2\right)\left(L_1 + L_{11}\right) + 4L_{11}^3 - 2L_2^3\left(\frac{I_1}{I_{21}}\right) - \\ 2\left(\frac{I_1}{I_{11}}\right)\left(\begin{array}{c} 6L_2L_{21}L_1 + 3L_2^2L_1 + 3L_{21}^2L_1 - 6L_1^2L_{11} - 3L_1L_{11}^2 - 2L_1^3 - \\ 6R\left\{L_1^2 - L_2L_1 - L_{21}L_1 + L_1L_{11}\right\} \end{array}\right) \end{array}\right]}{\left[6\left\{2L_{21} + 2L_{11} + \pi R + 2L_1\left(\frac{I_1}{I_{11}}\right) + 2L_2\left(\frac{I_1}{I_{21}}\right)\right\}\right]}$$

$$M_r = M_A + Pp\left\{\left(L_2 + L_{21}\right)\left(\frac{L_2 + L_{21}}{2} + R\cos\theta\right) + \left(1 - \sin\theta\right)\left[R^2 - R\left(L_1 + L_{11} + R\right)\right]\right\}$$

S_{bi}^{cD} is a maximum when $M_r = M_D$ when $\theta = \arctan\left(\frac{L_1 + L_{11}}{L_2 + L_{21}}\right)$

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^l \le S$	$S_m^c \le S$
$S_m^s + S_{bi}^{sE} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$	$S_m^c + S_{bi}^{cD} \le 1.5S$
$S_m^s + S_{bo}^{sE} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$	$S_m^c + S_{bo}^{cD} \le 1.5S$
$S_m^s + S_{bi}^{sF} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$	
$S_m^s + S_{bo}^{sF} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$	
$S_m^s + S_{bi}^{sG} \le 1.5S$	$S_m^l + S_{bi}^{lC} \le 1.5S$	
$S_m^s + S_{bo}^{sG} \le 1.5S$	$S_m^l + S_{bo}^{lC} \le 1.5S$	

Acceptance Criteria — Defined Locations for Stress Calculation

$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lY} \le 1.5S$	Not Applicable
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lY} \le 1.5S$	

Table 4.12.6
Stress Calculations and Acceptance Criteria for Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners) (Cont'd)

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.

S_{bi}^{sE}, S_{bo}^{sE} = bending stress in the short side at point E on the inside and outside surfaces, respectively.

S_{bi}^{sF}, S_{bo}^{sF} = bending stress in the short side at point F on the inside and outside surfaces, respectively.

S_{bi}^{sG}, S_{bo}^{sG} = bending stress in the short side at point G on the inside and outside surfaces, respectively.

S_{bi}^{sX}, S_{bo}^{sX} = bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.

S_m^l = membrane stress in the long side.

S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

S_{bi}^{lC}, S_{bo}^{lC} = bending stress in the long side at point C on the inside and outside surfaces, respectively.

S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.

S_m^c = membrane stress in the circular arc between B and E.

S_{bi}^{cD}, S_{bo}^{cD} = bending stress in the circular arc at point B on the inside and outside surfaces, respectively.

Table 4.12.7
Stress Calculations and Acceptance Criteria for Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^A = \frac{PpL_3}{A_cE_m}$$

$$S_{bi}^A = -S_{bo}^A\left(\frac{c_i}{c_o}\right) = \frac{M_Ac_i}{I_{21}E_b}$$

$$S_m^B = \frac{PpL_3}{A_cE_m}$$

$$S_{bi}^B = -S_{bo}^B\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[M_A - V_A\bar{Y}_2 + WL_2^2\right]$$

$$S_m^C = \frac{PpL_3}{A_cE_m}$$

$$S_{bi}^C = -S_{bo}^C\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[M_A + WK_5^2 + 2L_3W\bar{Y}_2\right]$$

$$S_m^M = \frac{Pp}{A_cE_m}\left(C_M^2 + \left(L_3 - E_M\right)^2\right)^{0.5}\cos\left(\theta_M - \beta_M\right)$$

$$S_{bi}^M = -S_{bo}^M\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[M_A + W\left(C_M^2 + C_MV_M + E_M^2 - E_MW_M - L_3\left\{2E_M + t_1 - W_M + 2\bar{Y}_2\right\}\right)\right]$$

$$S_m^D = \frac{PpO_{DE}}{A_cE_m}$$

$$S_{bi}^D = -S_{bi}^D\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[M_A + W\left(C_3^2 + C_3V_1 + E_{\theta 1}^2 - E_{\theta 1}W_1 - L_3\left\{2E_{\theta 1} + t_1 - W_M + 2\bar{Y}_2\right\}\right)\right]$$

$$S_m^U = \frac{PpO_{DE}}{A_cE_m}$$

$$S_{bi}^U = -S_{bi}^U\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[M_A + W\left(\begin{array}{c}\left\{C_3 + U_{2Y}\right\}^2 + \left\{C_3 + U_{2Y}\right\}V_1 + \left\{E_{\theta 1} + U_{2X}\right\}^2 - \\ \left\{E_{\theta 1} + U_{2X}\right\}W_1 - 2L_3\left\{\bar{Y}_2 + \dfrac{t_1\left(1 - \cos\theta_1\right)}{2} + E_{\theta 1} + U_{2X}\right\}\end{array}\right)\right]$$

Table 4.12.7
Stress Calculations and Acceptance Criteria for Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners) (Cont'd)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^E = \frac{PpO_{DE}}{A_c E_m}$$

$$S_{bi}^E = -S_{bo}^E\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[M_A + W\left(\begin{array}{c} C_{E1}^2 + C_{E1}V_1 + C_{E2}^2 - C_{E2}W_1 - \\ 2L_3\left\{\bar{Y}_2 + \frac{t_1(1-\cos\theta_1)}{2} + C_{E2}\right\}\end{array}\right)\right]$$

$$S_m^N = \frac{Pp}{A_c E_m}\left(C_M^2 + O_K^2\right)^{0.5}\cos\left(\theta_N - \beta_N\right)$$

$$S_{bi}^N = -S_{bo}^N\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[M_A + W\left(\begin{array}{c} \left\{L_4 - F_N\right\}^2 + \left\{L_4 - F_N\right\}V_N + \left\{M_1 + G_N\right\}^2 - \\ \left\{M_1 - G_N\right\}W_N - L_3\left\{2\bar{Y}_2 + t_1 + 2M_1 - 2G_N - W_N\right\}\end{array}\right)\right]$$

$$S_m^F = \frac{PpL_4}{A_c E_m}$$

$$S_{bi}^F = -S_{bo}^F\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[M_A + W\left(L_4^2 + L_4 t_1 + M_1^2 - 2L_3 J_2\right)\right]$$

$$S_m^G = \frac{P_p L_4}{A_c E_m}$$

$$S_{bi}^G = -S_{bo}^G\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[M_A + W\left(L_4^2 + L_4 t_1 + \left\{M_1 + L_{11}\right\}^2 - 2L_3\left\{J_2 + L_{11}\right\}\right)\right]$$

$$S_m^H = \frac{PpL_4}{A_c E_m}$$

$$S_{bi}^H = -S_{bo}^H\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{11} E_b}\left[M_A + W\left(L_4^2 + L_4 t_1 + 2L_4\bar{Y}_1 - L_3^2 - 2L_3\left\{\bar{Y}_2 + \frac{t_1}{2}\right\}\right)\right]$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^Y = -S_{bo}^Y\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{21} E_b}\left[M_A + PpY^2\right] \qquad 0 \le Y \le L_2$$

$$S_{bi}^Y = -S_{bo}^Y\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[M_A + PpY^2\right] \qquad L_2 \le Y \le \left(L_2 + L_{21}\right)$$

$$S_{bi}^X = -S_{bo}^X\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{11} E_b}\left[\begin{array}{c} M_A + \frac{PpX^2}{2} + \\ W\left\{L_4^2 + L_4 t_1 + 2L_4\bar{Y}_1 - L_3^2 - 2L_3\left(\bar{Y}_2 + \frac{t_1}{2}\right)\right\}\end{array}\right] \qquad 0 \le X \le L_1$$

$$S_{bi}^X = -S_{bo}^X\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1 E_b}\left[\begin{array}{c} M_A + \frac{PpX^2}{2} + \\ W\left\{L_4^2 + L_4 t_1 + 2L_4\bar{Y}_1 - L_3^2 - 2L_3\left(\bar{Y}_2 + \frac{t_1}{2}\right)\right\}\end{array}\right] \qquad L_1 \le X \le \left(L_1 + L_{11}\right)$$

Table 4.12.7
Stress Calculations and Acceptance Criteria for Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners) (Cont'd)

Equation Constants

$$I_1 = \frac{pt_1^3}{12}$$

$$M_A = Pp\frac{\left[K_{AB} + K_{BC} + K_{CD} + K_{DE} + K_{EF} + K_{FG} + K_{GH}\right]}{\left[-6\left(\left\{\frac{I_1}{I_{21}}\right\}L_2 + L_{21} + \frac{\pi R}{2} + U_1 + L_{11} + \left\{\frac{I_1}{I_{11}}\right\}L_1\right)\right]}$$

$$K_{AB} = \left(\frac{I_1}{I_{21}}\right)\left(L_2^3 - D_2L_2\right)$$

$$K_{BC} = 3L_2L_{11}K_5 + L_{21}^3 - D_2L_{11}$$

$$K_{CD} = 3R\theta_1\left[K_5^2 + 2R^2 + Rt_1 - L_3\left(S_1 + 2\bar{Y}_2\right)\right] + 3\left[K_5E_{\theta 1}S_1 + H_{\theta 1}S_1\left(L_3 - R\right)\right]$$

$$K_{DE} = \begin{array}{c} 3U_1\left[C_3^2 + C_3V_1 + E_{\theta 1}^2 - E_{\theta 1}W_1\right] - 6L_3U_1\left[\bar{Y}_2 + \frac{t_1}{2}\left(1 - \cos\theta_1\right) + E_{\theta 1}\right] + \\ 3U_1^2\left[C_3\cos\theta_1 + \sin\theta_1\left(E_{\theta 1} - L_3\right)\right] + U_1^3 \end{array}$$

$$K_{EF} = 3R\alpha_{ab}\left[D_3^2 + M_1^2 - 2L_3J_2 + R^2 + Rt_1\right] + 3G_1D_3S_1 + 3F_1S_1\left(L_3 - M_1\right)$$

$$K_{FG} = 3L_{11}\left[L_4^2 + L_4t_1 + M_1^2 - 2L_3J_2\right] + 3\left[M_1 - L_3\right]L_{11}^2 + L_{11}^3$$

$$K_{GH} = \left(\frac{I_1}{I_{11}}\right)\left[3L_1\left(L_4^2 + 2L_4\bar{Y} + L_4t_1 + \left\{M_1 + L_{11}\right\}^2 - 2L_3\left\{J_2 + L_{11}\right\}\right) - 2L_1^3\right]$$

$$A_c = t_1p$$

$$A_{DE} = \left[L_4 - \left(L_2 + L_{21} + R\tan\left(\frac{\theta_1}{2}\right)\right)\right]\sin\theta_1$$

$$C_3 = L_2 + L_{21} + R\sin\theta_1$$

$$C_{E1} = C_3 + N_1 - R$$

$$C_{E2} = E_{\theta 1} + M_1 - R$$

$$C_M = L_2 + L_{21} + R\sin\theta_M$$

$$C_N = L_4 - R + R\sin\beta_N$$

$$D_2 = 6L_4\bar{Y}_2$$

$$D_3 = L_4 - R$$

$$D_4 = L_1 + L_{11} + R\cos\theta_1$$

$$E_{\theta 1} = R\left(1 - \cos\theta_1\right)$$

$$E_M = R\left(1 - \cos\theta_M\right)$$

$$F_1 = R\left(1 - \sin\theta_1\right)$$

$$F_N = R\left(1 - \sin\beta_N\right)$$

$$G_1 = R\cos\theta_1$$

$$G_N = R\cos\beta_N$$

$$H_{\theta 1} = R\sin\theta_1$$

$$J_2 = \bar{Y}_2 + \frac{t_2}{2} + M_1$$

$$K_5 = L_2 + L_{21}$$

$$M_1 = L_3 - \left(L_1 + L_{11}\right)$$

$$N_1 = L_4 - \left(L_2 + L_{21}\right)$$

$$O_{DE} = \sqrt{L_3^2 + L_4^2} - A_{DE}$$

$$O_K = L_1 + L_{11} + R\cos\beta_N$$

$$S_1 = 2R + t_1$$

$$U_1 = \sqrt{\left(M_1 - R\right)^2 + \left(N_1 - R\right)^2}$$

$$U_2 = 0.5U_1$$

$$U_{2X} = U_2\sin\theta_1$$

$$U_{2Y} = U_2\cos\theta_1$$

$$V_1 = t_1\sin\theta_1$$

$$V_A = PpL_3$$

$$V_M = t_1\sin\theta_M$$

$$V_N = t_1\sin\beta_N$$

$$W = 0.5Pp$$

$$W_1 = t_1\cos\theta_1$$

$$W_M = t_1\cos\theta_M$$

$$W_N = t_1\cos\theta_N$$

$$\alpha_{ab} = \arctan\left[\frac{L_3}{L_4}\right]$$

$$\beta_M = \arctan\left[\frac{C_M}{L_3 - E_{\theta 1}}\right]$$

$$\beta_N = \arctan\left[\frac{L_4 - R}{L_1 + L_{11}}\right]$$

$$\theta_1 = \arctan\left[\frac{L_4}{L_3}\right]$$

$$\theta_M = \arctan\left[\frac{-K_5S_1}{2R^2 - RS_1 - L_3t_1}\right]$$

$$\theta_N = \arctan\left[\frac{C_N}{O_K}\right]$$

Table 4.12.7
Stress Calculations and Acceptance Criteria for Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners) (Cont'd)

Acceptance Criteria — Critical Locations of Maximum Stress		
$S_m^A \le S$	$S_m^M \le S$	$S_m^F \le S$
$S_m^A + S_{bi}^A \le 1.5S$	$S_m^M + S_{bi}^M \le 1.5S$	$S_m^F + S_{bi}^F \le 1.5S$
$S_m^A + S_{bo}^A \le 1.5S$	$S_m^M + S_{bo}^M \le 1.5S$	$S_m^F + S_{bo}^F \le 1.5S$
$S_m^B \le S$	$S_m^D \le S$	$S_m^G \le S$
$S_m^B + S_{bi}^B \le 1.5S$	$S_m^D + S_{bi}^D \le 1.5S$	$S_m^G + S_{bi}^G \le 1.5S$
$S_m^B + S_{bo}^B \le 1.5S$	$S_m^D + S_{bo}^D \le 1.5S$	$S_m^G + S_{bo}^G \le 1.5S$
$S_m^C \le S$	$S_m^U \le S$	$S_m^H \le S$
$S_m^C + S_{bi}^C \le 1.5S$	$S_m^U + S_{bi}^U \le 1.5S$	$S_m^H + S_{bi}^H \le 1.5S$
$S_m^C + S_{bo}^C \le 1.5S$	$S_m^U + S_{bo}^U \le 1.5S$	$S_m^H + S_{bo}^H \le 1.5S$
	$S_m^E \le S$	
	$S_m^E + S_{bi}^E \le 1.5S$	
	$S_m^E + S_{bo}^E \le 1.5S$	
	$S_m^N \le S$	
	$S_m^N + S_{bi}^N \le 1.5S$	
	$S_m^N + S_{bo}^N \le 1.5S$	
Acceptance Criteria — Defined Locations for Stress Calculation		
$S_m^Y + S_{bi}^Y \le 1.5S$	Not Applicable	$S_m^X + S_{bi}^X \le 1.5S$
$S_m^Y + S_{bo}^Y \le 1.5S$		$S_m^X + S_{bo}^X \le 1.5S$

Table 4.12.7
Stress Calculations and Acceptance Criteria for Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners) (Cont'd)

Nomenclature for Stress Results

S_m^A = membrane stress at point A.
S_{bi}^A, S_{bo}^A = bending stress at point A on the inside and outside surfaces, respectively.
S_m^B = membrane stress at point B.
S_{bi}^B, S_{bo}^B = bending stress at point B on the inside and outside surfaces, respectively.
S_m^C = membrane stress at point C.
S_{bi}^C, S_{bo}^C = bending stress at point C on the inside and outside surfaces, respectively.
S_m^M = membrane stress at point M.
S_{bi}^M, S_{bo}^M = bending stress at point M on the inside and outside surfaces, respectively.
S_m^D = membrane stress at point D.
S_{bi}^D, S_{bo}^D = bending stress at point D on the inside and outside surfaces, respectively.
S_m^U = membrane stress at point U.
S_{bi}^U, S_{bo}^U = bending stress at point U on the inside and outside surfaces, respectively.
S_m^E = membrane stress at point E.
S_{bi}^E, S_{bo}^E = bending stress at point E on the inside and outside surfaces, respectively.
S_m^N = membrane stress at point N.
S_{bi}^N, S_{bo}^N = bending stress at point N on the inside and outside surfaces, respectively.
S_m^F = membrane stress at point F.
S_{bi}^F, S_{bo}^F = bending stress at point F on the inside and outside surfaces, respectively.
S_m^H = membrane stress at point G.
S_{bi}^G, S_{bo}^G = bending stress at point G on the inside and outside surfaces, respectively.
S_m^H = membrane stress at point H.
S_{bi}^H, S_{bo}^H = bending stress at point H on the inside and outside surfaces, respectively.
S_{bi}^X, S_{bo}^X = bending stress at a point defined by X on the inside and outside surfaces, respectively.
S_{bi}^Y, S_{bo}^Y = bending stress at a point defined by Y on the inside and outside surfaces, respectively.

Table 4.12.8
Stress Calculations and Acceptance Criteria for Type 7 Noncircular Vessels (Rectangular Cross Section With Single-Stay Plate or Multiple Bars)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{Ph}{4t_1E_m}\left[4 - \left(\frac{2 + K(5 - \alpha^2)}{1 + 2K}\right)\right]$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{24I_1E_b}\left[-3H^2 + 2h^2\left(\frac{1 + 2\alpha^2K}{1 + 2K}\right)\right]$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_1E_b}\left[\frac{1 + 2\alpha^2K}{1 + 2K}\right]$$

$$S_m^l = \frac{PH}{2t_2E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_2E_b}\left[\frac{1 + K(3 - \alpha^2)}{1 + 2K}\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_2E_b}\left[\frac{1 + 2\alpha^2K}{1 + 2K}\right]$$

$$S_m^{st} = \frac{Ph}{2t_3E_{st}}\left[\frac{2 + K(5 - \alpha^2)}{1 + 2K}\right] \quad \text{for a stay plate}$$

$$S_m^{st} = \frac{2Php}{\pi t_3^2E_{st}}\left[\frac{2 + K(5 - \alpha^2)}{1 + 2K}\right] \quad \text{for stay bars}$$

Equation Constants

$$I_1 = \frac{bt_1^3}{12} \qquad K = \frac{I_2}{I_1}\alpha$$

$$I_2 = \frac{bt_2^3}{12} \qquad \alpha = \frac{H}{h}$$

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^{st} \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$		$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$		$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$		$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$		$S_m^l + S_{bo}^{lB} \le 1.5S$

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.

S_{bi}^{sB}, S_{bo}^{sB} = bending stress in the short side at point B on the inside and outside surfaces, respectively.

S_{bi}^{sC}, S_{bo}^{sC} = bending stress in the short side at point C on the inside and outside surfaces, respectively.

S_{bi}^{sX}, S_{bo}^{sX} = bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.

S_m^l = membrane stress in the long side.

S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.

S_m^{st} = membrane stress in the stay bar or plate, as applicable.

Table 4.12.9
Stress Calculations and Acceptance Criteria for Type 8 Noncircular Vessels (Rectangular Cross Section With Double-Stay Plate or Multiple Bars)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^s = \frac{Ph}{2t_1E_m}\left[3 - \left(\frac{6 + K\left(11 - \alpha^2\right)}{3 + 5K}\right)\right]$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{24I_1E_b}\left[-3H^2 + 2h^2\left(\frac{3 + 5\alpha^2K}{3 + 5K}\right)\right]$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_1E_b}\left[\frac{3 + 5\alpha^2K}{3 + 5K}\right]$$

$$S_m^l = \frac{PH}{2t_2E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_2E_b}\left[\frac{3 + K\left(6 - \alpha^2\right)}{3 + 5K}\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_2E_b}\left[\frac{3 + 5\alpha^2K)}{3 + 5K}\right]$$

$$S_m^{st} = \frac{Ph}{2t_4E_{st}}\left[\frac{6 + K\left(11 - \alpha^2\right)}{3 + 5K}\right] \qquad \text{for a stay plate}$$

$$S_m^{st} = \frac{2Php}{\pi t_4^2E_{st}}\left[\frac{6 + K\left(11 - \alpha^2\right)}{3 + 5K}\right] \qquad \text{for stay bars}$$

Equation Constants

$$I_1 = \frac{bt_1^3}{12} \qquad K = \frac{I_2}{I_1}\alpha$$

$$I_2 = \frac{bt_2^3}{12} \qquad \alpha = \frac{H}{h}$$

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^{st} \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$		$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$		$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$		$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$		$S_m^l + S_{bo}^{lB} \le 1.5S$

Nomenclature for Stress Results

S_m^s = membrane stress in the short side.

S_{bi}^{sB}, S_{bo}^{sB} = bending stress in the short side at point B on the inside and outside surfaces, respectively.

S_{bi}^{sC}, S_{bo}^{sC} = bending stress in the short side at point C on the inside and outside surfaces, respectively.

S_{bi}^{sX}, S_{bo}^{sX} = bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.

S_m^l = membrane stress in the long side.

S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.

S_m^{st} = membrane stress in the stay bar or plate, as applicable.

Table 4.12.10
Stress Calculations and Acceptance Criteria for Type 9 Noncircular Vessels (Obround Cross Section)

Membrane and Bending Stresses — Critical Locations of Maximum Stress	
$S_m^{cB} = \frac{PR}{t_1 E_m}$	(91) [Note (1)]
$S_{bi}^{cB} = -S_{bo}^{cB}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{6I_1 E_b}\left[3L_2 - \frac{C_1}{A}\right]$	(94) [Note (1)]
$S_m^{cC} = \frac{P(R + L_2)}{t_1 E_m}$	(92) [Note (1)]
$S_{bi}^{cC} = -S_{bo}^{cC}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{6I_1 E_b}\left[3(L_2 + 2R) - \frac{C_1}{A}\right]$	(95) [Note (1)]
$S_m^{l} = \frac{PR}{t_2 E_m}$	(93) [Note (1)]
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 C_1 c_i}{6AI_2 E_b}$	(96) [Note (1)]
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{6I_2 E_b}\left[3L_2 - \frac{C_1}{A}\right]$	(97) [Note (1)]
Membrane and Bending Stresses — Defined Locations for Stress Calculation	
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\frac{c_i}{c_o}\right) = \frac{Pc_i}{I_2 E_b}\left[\frac{-L_2 C_1}{6A} + \frac{Y^2}{2}\right]$	
Equation Constants	
$I_1 = \frac{bt_1^3}{12}$	
$I_2 = \frac{bt_2^3}{12}$	
$A = R\left[2\left(\frac{L_2}{R}\right) + \pi\left(\frac{I_2}{I_1}\right)\right]$	
$C_1 = R^2\left[2\left(\frac{L_2}{R}\right)^2 + 3\pi\left(\frac{L_2}{R}\right)\left(\frac{I_2}{I_1}\right) + 12\left(\frac{I_2}{I_1}\right)\right]$	
Acceptance Criteria — Critical Locations of Maximum Stress	
$S_m^{cB} \le S$	$S_m^{l} \le S$
$S_m^{cB} + S_{bi}^{cB} \le 1.5S$	$S_m^{l} + S_{bi}^{lA} \le 1.5S$
$S_m^{cB} + S_{bo}^{cB} \le 1.5S$	$S_m^{l} + S_{bo}^{lA} \le 1.5S$
$S_m^{cC} \le S$	$S_m^{l} + S_{bi}^{lB} \le 1.5S$
$S_m^{cC} + S_{bi}^{cC} \le 1.5S$	$S_m^{l} + S_{bo}^{lB} \le 1.5S$
$S_m^{cC} + S_{bo}^{cC} \le 1.5S$	
Acceptance Criteria — Defined Locations for Stress Calculation	
Not Applicable	$S_m^{l} + S_{bi}^{lY} \le 1.5S$ $S_m^{l} + S_{bo}^{lY} \le 1.5S$

Table 4.12.10
Stress Calculations and Acceptance Criteria for Type 9 Noncircular Vessels (Obround Cross Section) (Cont'd)

Nomenclature for Stress Results

S_m^{cB} = membrane stress in the circular arc at point B.

S_{bi}^{cB}, S_{bo}^{cB} = bending stress in the circular arc at point B on the inside and outside surfaces, respectively.

S_m^{cC} = membrane stress in the circular arc at point C.

S_{bi}^{cC}, S_{bo}^{cC} = bending stress in the circular arc at point C on the inside and outside surfaces, respectively.

S_m^{l} = membrane stress in the long side.

S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.

S_m^{st} = membrane stress in stay bar or plate, as applicable.

GENERAL NOTE: The variable b is the nominal width of the vessel flat section, usually corresponding to the vessel or header length. Its value will cancel out in the above equation so the actual value selected is not critical. It is sometimes convenient to choose the pitch of multiple holes in a header application.

NOTE:
(1) Equation numbers correspond to those in "Pressure Vessels of Noncircular Cross Section (Commentary on New Rules for ASME Code)," J.P. Faupel, *Journal of Pressure Vessel Technology*, August 1979, vol. 101.

Table 4.12.11
Stress Calculations and Acceptance Criteria for Type 10 Noncircular Vessels (Reinforced Obround Cross Section)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

$$S_m^{cB} = \frac{PRp}{(A_1 + pt_1)E_m}$$

$$S_{bi}^{cB} = -S_{bo}^{cB}\left(\frac{c_i^c}{c_o^c}\right) = \frac{PpL_2c_i^c}{6I_{11}E_b}\left[3L_2 - \frac{C_2}{A_3}\right]$$

$$S_m^{cC} = \frac{P(R + L_2)p}{(A_1 + pt_1)E_m}$$

$$S_{bi}^{cC} = -S_{bo}^{cC}\left(\frac{c_i^c}{c_o^c}\right) = \frac{PpL_2c_i^c}{6I_{11}E_b}\left[3(L_2 + 2r) - \frac{C_2}{A_3}\right]$$

$$S_m^{l} = \frac{PRp}{(A_1 + pt_1)E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i^l}{c_o^l}\right) = \frac{PL_2pc_i^l}{6I_{11}E_b}\left[\frac{-C_2}{A_3}\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i^l}{c_o^l}\right) = \frac{PL_2pc_i^l}{6I_{11}E_b}\left[3L_2 - \frac{C_2}{A_3}\right]$$

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_{bi}^{lY} = -S_{bo}^{ly}\left(\frac{c_i^l}{c_o^l}\right) = \frac{Ppc_i^l}{I_{11}E_b}\left[\frac{-L_2C_2}{6A_3} + \frac{Y^2}{2}\right]$$

Equation Constants

$$A_3 = r\left[2\left(\frac{L_2}{r}\right) + \pi\right] \qquad C_2 = r^2\left[2\left(\frac{L_2}{r}\right)^2 + 3\pi\left(\frac{L_2}{r}\right) + 12\right]$$

Table 4.12.11
Stress Calculations and Acceptance Criteria for Type 10 Noncircular Vessels (Reinforced Obround Cross Section) (Cont'd)

Acceptance Criteria — Critical Locations of Maximum Stress	
$S_m^{cB} \le S$	$S_m^{l} \le S$
$S_m^{cB} + S_{bi}^{cB} \le 1.5S$	$S_m^{l} + S_{bi}^{lA} \le 1.5S$
$S_m^{cB} + S_{bo}^{cB} \le 1.5S$	$S_m^{l} + S_{bo}^{lA} \le 1.5S$
$S_m^{cC} \le S$	$S_m^{l} + S_{bi}^{lB} \le 1.5S$
$S_m^{cC} + S_{bi}^{cC} \le 1.5S$	$S_m^{l} + S_{bo}^{lB} \le 1.5S$
$S_m^{cC} + S_{bo}^{cC} \le 1.5S$	
Acceptance Criteria — Defined Locations for Stress Calculation	
Not Applicable	$S_m^{l} + S_{bi}^{lY} \le 1.5S$
	$S_m^{l} + S_{bo}^{lY} \le 1.5S$

Nomenclature for Stress Results

S_m^{cB} = membrane stress in the circular arc at point B.

S_{bi}^{cB}, S_{bo}^{cB} = bending stress in the circular arc at point B on the inside and outside surfaces, respectively.

S_m^{cC} = membrane stress in the circular arc at point C.

S_{bi}^{cC}, S_{bo}^{cC} = bending stress in the circular arc at point C on the inside and outside surfaces, respectively.

S_m^{l} = membrane stress in the long side.

S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.

S_m^{st} = membrane stress in stay bar or plate, as applicable.

Table 4.12.12
Stress Calculations and Acceptance Criteria for Type 11 Noncircular Vessels (Obround Cross Section With Single-Stay Plate or Multiple Bars)

Membrane and Bending Stresses — Defined Locations for Stress Calculation

$$S_m^{cB} = \frac{PR}{t_1 E_m}$$

$$S_{bi}^{cB} = S_{bo}^{cB}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{2I_1 AE_b}\left[F\left(B - AL_2\right) - \frac{C_1}{3} + AL_2\right]$$

$$S_m^{cC} = \frac{P}{2t_1 E_m}\left[2\left(R + L_2\right) - L_2 F\right]$$

$$S_{bi}^{cC} = S_{bo}^{cC}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{2I_1 AE_b}\left[F\left(B - AL_2 - AR\right) - \frac{C_1}{3} + A\left(L_2 + 2R\right)\right]$$

$$S_m^{l} = \frac{PR}{t_2 E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{2I_2 AE_b}\left[BF - \frac{C_1}{3}\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{PbL_2 c_i}{2I_2 AE_b}\left[F\left(B - AL_2\right) - \frac{C_1}{3} + AL_2\right]$$

$$S_m^{st} = \frac{PL_2 F}{t_3 E_{st}} \quad \text{for a stay plate}$$

$$S_m^{st} = \frac{4PL_2 Fp}{\pi t_3^2 E_{st}} \quad \text{for stay bars}$$

Equation Constants

$$I_1 = \frac{bt_1^3}{12}$$

$$I_2 = \frac{bt_2^3}{12}$$

$$A = R\left[2\left(\frac{L_2}{R}\right) + \pi\left(\frac{I_2}{I_1}\right)\right]$$

$$B = R^2\left[\left(\frac{L_2}{R}\right)^2 + \pi\left(\frac{L_2}{R}\right)\left(\frac{I_2}{I_1}\right) + 2\left(\frac{I_2}{I_1}\right)\right]$$

$$C_1 = R^2\left[2\left(\frac{L_2}{R}\right)^2 + 3\pi\left(\frac{L_2}{R}\right)\left(\frac{I_2}{I_1}\right) + 12\left(\frac{I_2}{I_1}\right)\right]$$

$$F = \frac{\left(3AD_1 - 2BC_1\right)}{\left(AE_1 - 6B^2\right)}$$

$$D_1 = R^3\left[\begin{array}{c}\left(\frac{L_2}{R}\right)^3 + 2\pi\left(\frac{L_2}{R}\right)^2\left(\frac{I_2}{I_1}\right) + \\ 12\left(\frac{L_2}{R}\right)\left(\frac{I_2}{I_1}\right) + 2\pi\left(\frac{I_2}{I_1}\right)\end{array}\right]$$

$$E_1 = R^3\left[\begin{array}{c}4\left(\frac{L_2}{R}\right)^3 + 6\pi\left(\frac{L_2}{R}\right)^2\left(\frac{I_2}{I_1}\right) + \\ 24\left(\frac{L_2}{R}\right)\left(\frac{I_2}{I_1}\right) + 3\pi\left(\frac{I_2}{I_1}\right)\end{array}\right]$$

Acceptance Criteria — Critical Locations of Maximum Stress

$S_m^{cB} \le S$	$S_m^{st} \le S$	$S_m^{l} \le S$
$S_m^{cB} + S_{bi}^{cB} \le 1.5S$		$S_m^{l} + S_{bi}^{lA} \le 1.5S$
$S_m^{cB} + S_{bo}^{cB} \le 1.5S$		$S_m^{l} + S_{bo}^{lA} \le 1.5S$
$S_m^{cC} \le S$		$S_m^{l} + S_{bi}^{lB} \le 1.5S$
$S_m^{cC} + S_{bi}^{cC} \le 1.5S$		$S_m^{l} + S_{bo}^{lB} \le 1.5S$
$S_m^{cC} + S_{bo}^{cC} \le 1.5S$		

Table 4.12.12
Stress Calculations and Acceptance Criteria for Type 11 Noncircular Vessels (Obround Cross Section With Single-Stay Plate or Multiple Bars) (Cont'd)

Nomenclature for Stress Results

S_m^{cB} = membrane stress in the circular arc at point B.

S_{bi}^{cB}, S_{bo}^{cB} = bending stress in the circular arc at point B on the inside and outside surfaces, respectively.

S_m^{cC} = membrane stress in the circular arc at point C.

S_{bi}^{cC}, S_{bo}^{cC} = bending stress in the circular arc at point C on the inside and outside surfaces, respectively.

S_m^{l} = membrane stress in the long side.

S_{bi}^{lB}, S_{bo}^{lB} = bending stress in the long side at point B on the inside and outside surfaces, respectively.

S_{bi}^{lA}, S_{bo}^{lA} = bending stress in the long side at point A on the inside and outside surfaces, respectively.

S_{bi}^{lY}, S_{bo}^{lY} = bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.

S_m^{st} = membrane stress in stay bar or plate, as applicable.

Table 4.12.13
Stress Calculations and Acceptance Criteria for Type 12 Noncircular Vessels (Circular Cross Section With Single-Stay Plate)

Membrane and Bending Stresses — Critical Locations of Maximum Stress

Equal Pressure	Unequal Pressure
$S_m = \dfrac{P_1 R}{t_1 E_m}$	$S_m = \dfrac{PR}{t_1 E_m}$
$S_{bi} = -S_{bo}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{I_1 E_b}\left[\dfrac{2P_1 t_1^2}{3\left(\pi^2 - 8\right)}\right]$	$S_{bi} = -S_{bo}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{3I_1 E_b}\left[P_1\left(\dfrac{2t_1^2}{\pi^2 - 8}\right) + \dfrac{3R^2\left(P_1 - P_2\right)}{6 + \left(\dfrac{t_3}{t_1}\right)^3}\right]$
$S_m^{st} = \dfrac{2\pi P_1 t_1^2}{3Rt_3\left(\pi^2 - 8\right)E_m}$	$S_m^{st} = \dfrac{\pi t_1^2\left(P_1 + P_2\right)}{3Rt_3\left(\pi^2 - 8\right)E_m}$
	$S_{bi}^{st} = -S_{bo}^{st}\left(\dfrac{c_i}{c_o}\right) = \dfrac{J_1\left(P_1 - P_2\right)L_v^2 bc_i}{I_3 E_b}$ for $L_1 \le 2R$
	$S_{bi}^{st} = -S_{bo}^{st}\left(\dfrac{c_i}{c_o}\right) = \dfrac{J_1\left(P_1 - P_2\right)4R^2 bc_i}{I_3 E_b}$ for $L_1 > 2R$

Equation Constants

$$I_1 = \frac{bt_1^3}{12}$$

$$I_3 = \frac{bt_3^3}{12}$$

$$J_1 = \min\left[\left(\begin{array}{c} -0.055314 + 0.14237\left(R_h\right) - 0.041311\left(R_h\right)^2 + \\ 0.0050644\left(R_h\right)^3 - 0.00021145\left(R_h\right)^4 \end{array}\right), 5.0\right]$$

$$R_h = \max\left[\frac{L_v}{2R}, \frac{2R}{L_v}\right]$$

Table 4.12.13
Stress Calculations and Acceptance Criteria for Type 12 Noncircular Vessels (Circular Cross Section With Single-Stay Plate) (Cont'd)

Acceptance Criteria — Critical Locations of Maximum Stress	
Equal Pressure	**Unequal Pressure**
$S_m \le S$	$S_m \le S$
$S_m + S_{bi} \le 1.5S$	$S_m + S_{bi} \le 1.5S$
$S_m + S_{bo} \le 1.5S$	$S_m + S_{bo} \le 1.5S$
$S_m^{st} \le S$	$S_m^{st} \le S$
	$S_m^{st} + S_{bi}^{st} \le 1.5S$
	$S_m^{st} + S_{bo}^{st} \le 1.5S$

Nomenclature for Stress Results

S_m = membrane stress in the pipe.

S_{bi} , S_{bo} = bending stress in the pipe.

S_m^{st} = membrane stress in stay plate, as applicable.

S_{bi}^{st}, S_{bo}^{st} = bending stress in the stay plate on the inside and outside surfaces, respectively.

Table 4.12.14
Effective Width Coefficient

Material	Effective Width Coefficient, Δ	
	$\sqrt{psi}$	$\sqrt{kPa}$
Carbon Steel	6,000	15 754
Austenitic Stainless Steel	5,840	15 334
Ni–Cr–Fe	6,180	16 229
Ni–Fe–Cr	6,030	15 834
Aluminum	3,560	9 348
Nickel Copper	5,720	15 021
Unalloyed Titanium	4,490	11 789

Table 4.12.15
Compressive Stress Calculations

Short-Side Plates	Long-Side Plates	End Plates
$S_{mA} = \frac{P_e h H}{2\left(t_1 H + t_2 h\right)}$	$S_{mA} = \frac{P_e h H}{2\left(t_1 H + t_2 h\right)}$	$S_{mA} = \frac{P_e H L_v}{2\left(t_2 L_v + t_5 H\right)}$
$S_{mB} = \frac{P_e h}{2t_1}$	$S_{mB} = \frac{P_e H}{2t_2}$	$S_{mB} = \frac{P_e h L_v}{2\left(t_1 L_v + t_5 H\right)}$
$S_{crA}^{*} = \frac{\pi^2 E_y}{12\left(1 - \nu^2\right)}\left(\frac{t_1}{H}\right)^2 K_A$	$S_{crA}^{*} = \frac{\pi^2 E_y}{12\left(1 - \nu^2\right)}\left(\frac{t_2}{h}\right)^2 K_A$	$S_{crA}^{*} = \frac{\pi^2 E_y}{12\left(1 - \nu^2\right)}\left(\frac{t_5}{H}\right)^2 K_A$
$S_{crA}^{**} = S_y - \frac{S_y^2}{4S_{crA}^{*}}$	$S_{crA}^{**} = S_y - \frac{S_y^2}{4S_{crA}^{*}}$	$S_{crA}^{**} = S_y - \frac{S_y^2}{4S_{crA}^{*}}$
$S_{crB}^{*} = \frac{\pi^2 E_y}{12\left(1 - \nu^2\right)}\left(\frac{t_1}{L_v}\right)^2 K_B$	$S_{crB}^{*} = \frac{\pi^2 E_y}{12\left(1 - \nu^2\right)}\left(\frac{t_2}{L_v}\right)^2 K_B$	$S_{crB}^{*} = \frac{\pi^2 E_y}{12\left(1 - \nu^2\right)}\left(\frac{t_5}{h}\right)^2 K_B$
$S_{crB}^{**} = S_y - \frac{S_y^2}{4S_{crB}^{*}}$	$S_{crB}^{**} = S_y - \frac{S_y^2}{4S_{crB}^{*}}$	$S_{crB}^{**} = S_y - \frac{S_y^2}{4S_{crB}^{*}}$
$K_A = K_A\left(x = \frac{L_v}{H}\right)$	$K_A = K_A\left(x = \frac{L_v}{h}\right)$	$K_A = K_A\left(x = \frac{h}{H}\right)$
$K_B = K_B\left(x = \frac{H}{L_v}\right)$	$K_B = K_B\left(x = \frac{h}{L_v}\right)$	$K_B = K_B\left(x = \frac{H}{h}\right)$

Nomenclature for Stress Results

S_{mA} = compressive stress applied to the short edge of the side panels due to external pressure on the end plates.
S_{mB} = compressive stress applied to the long edge of the side panels due to external pressure on the end plates.
S_{crA} = plate buckling stress when the panel is subjected to stress on the short edge.
S_{crB} = plate buckling stress when the panel is subjected to stress on the long edge.

GENERAL NOTES:
(a) The equations for K_A and K_B are:

$$K_A(x) = \max\left[\left(5.2184 + \frac{1.4597}{x} + \frac{0.30384}{x^2}\right), 5.50\right] \tag{4.12.58}$$

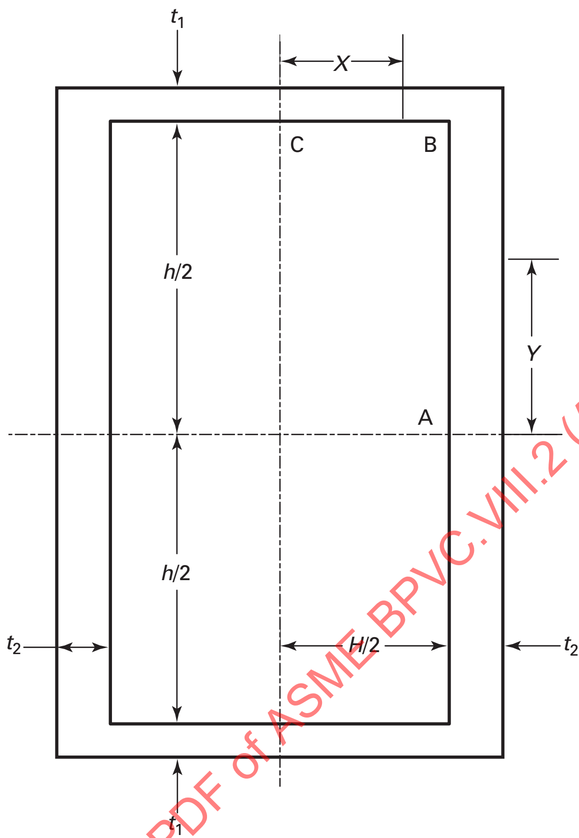
$$K_B(x) = 5.2184 + \frac{1.4597}{x} + \frac{0.30384}{x^2} \qquad \text{for } x \geq 0.258 \tag{4.12.59}$$

$$K_B(x) = 1.0 \qquad \text{for } x < 0.258 \tag{4.12.60}$$

(b) The membrane equations for S_{mA} in this table apply to vessels where the long plate thicknesses are equal. If the thicknesses are not equal, replace $2t_2$ with $\left(t_2 + t_{22}\right)$ in the calculations.
(c) The membrane equation S_{mB} in this table for the long-side plate applies to vessels where the long plate thicknesses are equal. If the thicknesses are not equal, the membrane stress for the long-side plates shall be determined in accordance with Table 4.12.3.
(d) Note in the above nomenclature, $K_A = K_A\left(x = \frac{H}{L_v}\right)$ is defined as computing K_A using the function $K_A(x)$ evaluated at $x = \frac{H}{L_v}$.

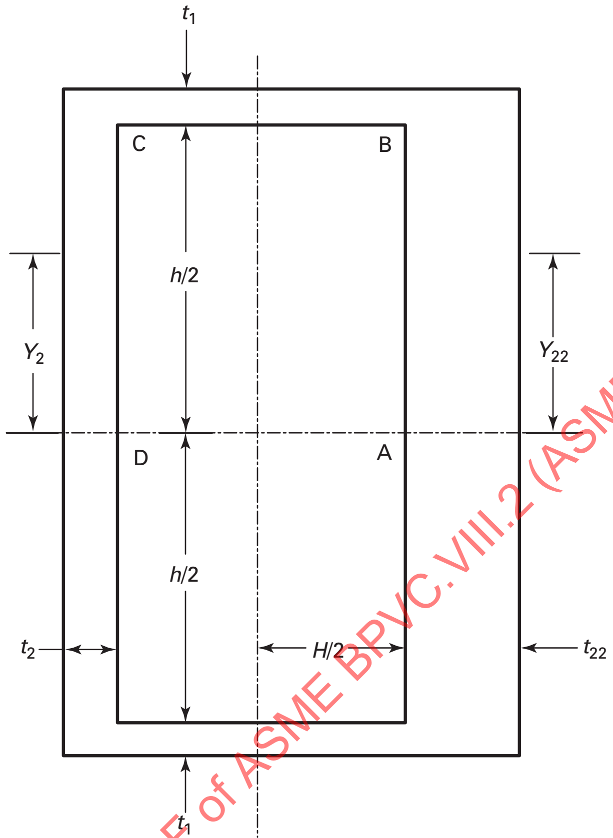
4.12.13 FIGURES

Figure 4.12.1
Type 1 Noncircular Vessels
(Rectangular Cross Section)



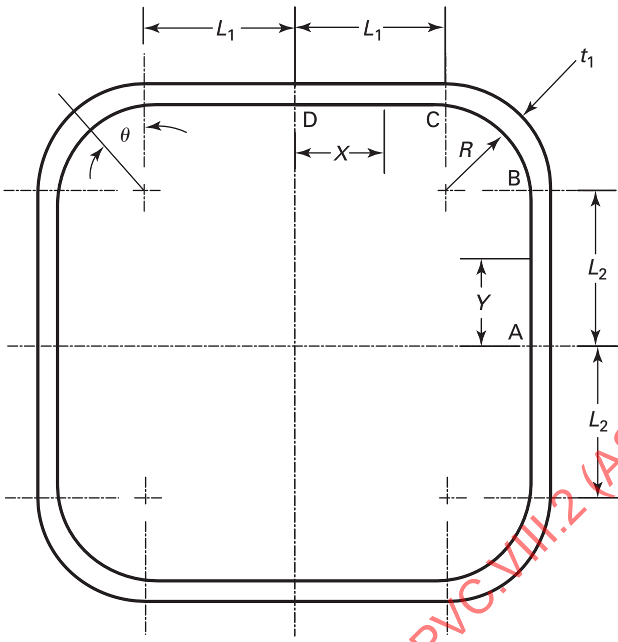
GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, and C.
(b) Defined Locations for Stress Calculations are determined using variables X and Y.

Figure 4.12.2
Type 2 Noncircular Vessels
(Rectangular Cross Section With Unequal Side Plate Thicknesses)



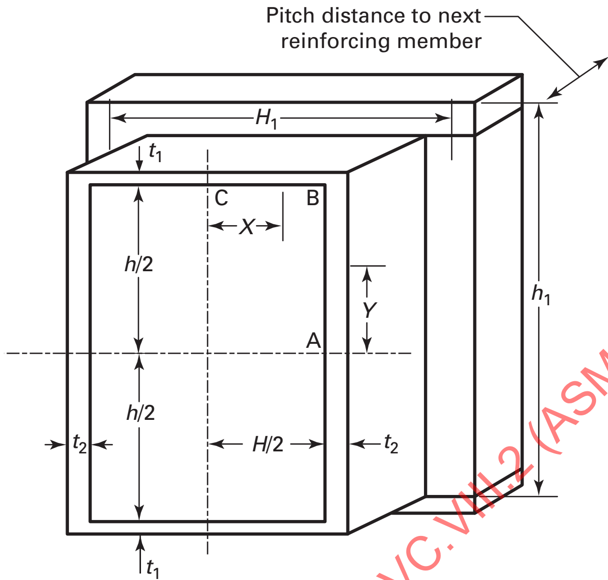
GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, C, and D.
(b) Defined Locations for Stress Calculations are determined using variables Y_2 and Y_{22}.

Figure 4.12.3
Type 3 Noncircular Vessels
(Chamfered Rectangular Cross Section)



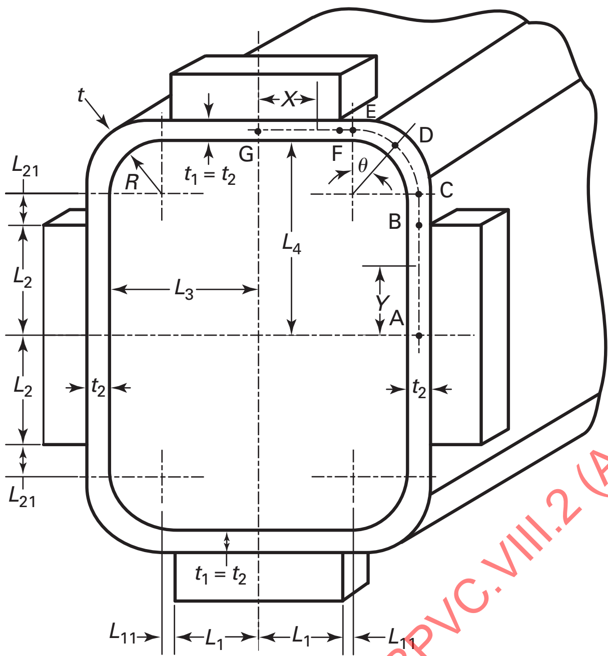
GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, C, and D.
(b) Defined Locations for Stress Calculations are determined using variables X and Y.

Figure 4.12.4
Type 4 Noncircular Vessels
(Reinforced Rectangular Cross Section)



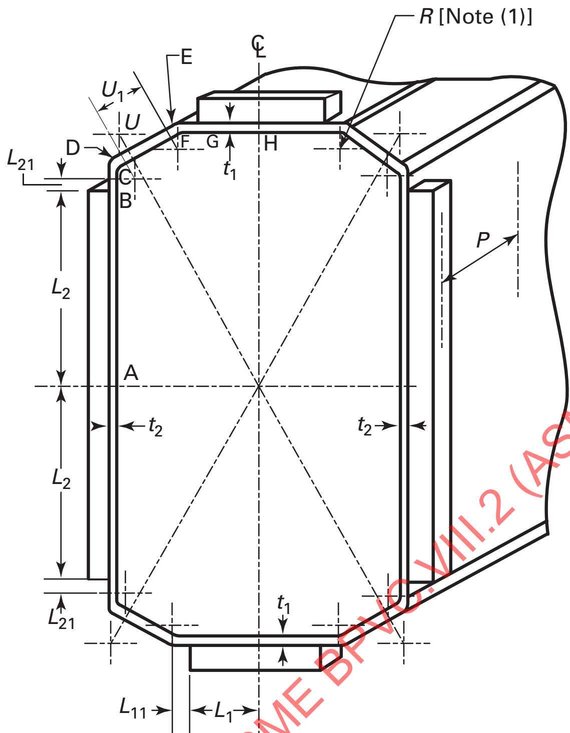
GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B and C.
(b) Defined Locations for Stress Calculations are determined using variables X and Y.

Figure 4.12.5
Type 5 Noncircular Vessels
(Reinforced Rectangular Cross Section With Chamfered Corners)



GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, C, E, F, and G.
(b) Defined Locations for Stress Calculations are determined using variables X and Y.

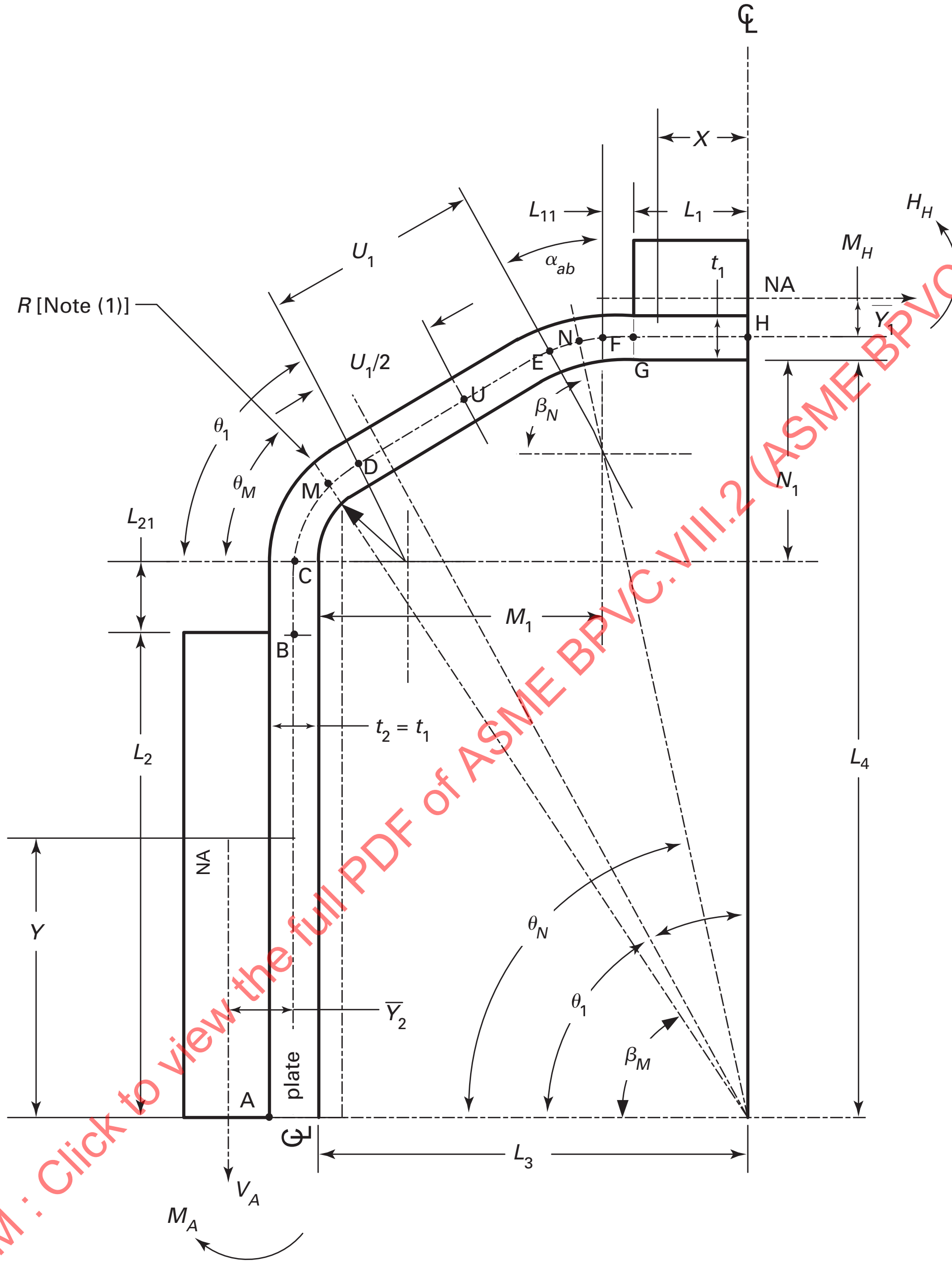
Figure 4.12.6
Type 6 Noncircular Vessels
(Reinforced Octagonal Cross Section With Chamfered Corners)



GENERAL NOTE: Critical Locations of Maximum Stress are defined at points A, B, C, E, F, G, and H.

NOTE:
(1) The radius must be the same at all eight corners.

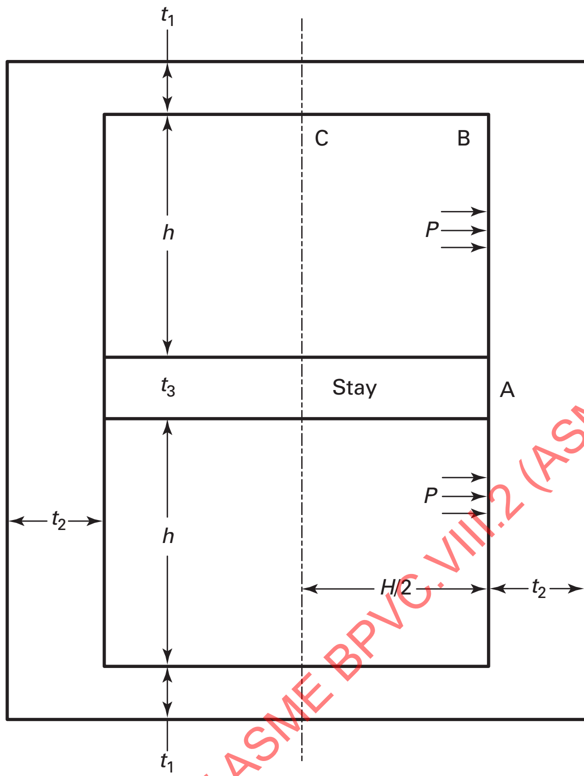
Figure 4.12.7
Type 6 Noncircular Vessels
(Reinforced Octagonal Cross Section With Chamfered Corners — Details)



GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, C, E, F, G, H, M, N, and U.
(b) Defined Locations for Stress Calculations are determined using variables X and Y.

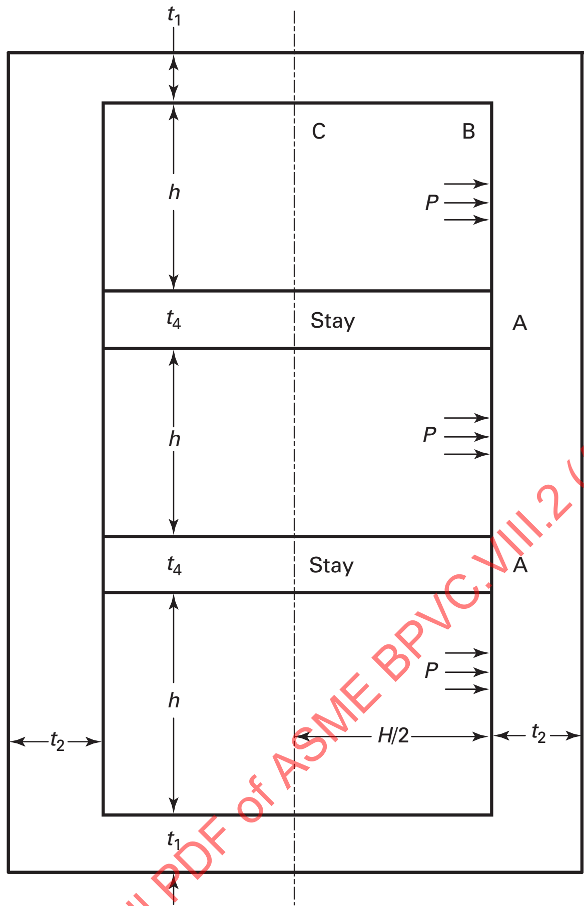
NOTE:
(1) The radius must be the same at all eight corners.

Figure 4.12.8
Type 7 Noncircular Vessels
(Rectangular Cross Section With Single-Stay Plate or Multiple Bars)



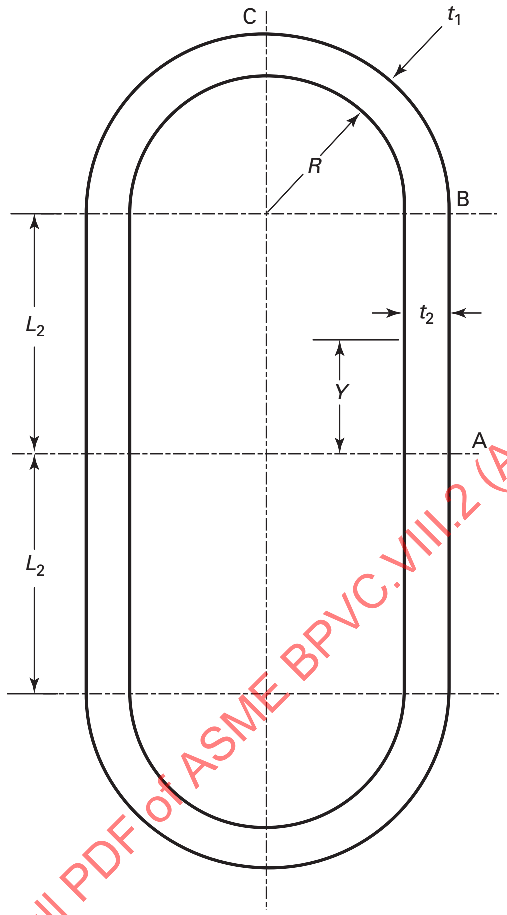
GENERAL NOTE: Critical Locations of Maximum Stress are defined at points A, B, and C.

Figure 4.12.9
Type 8 Noncircular Vessels
(Rectangular Cross Section With Double-Stay Plate or Multiple Bars)



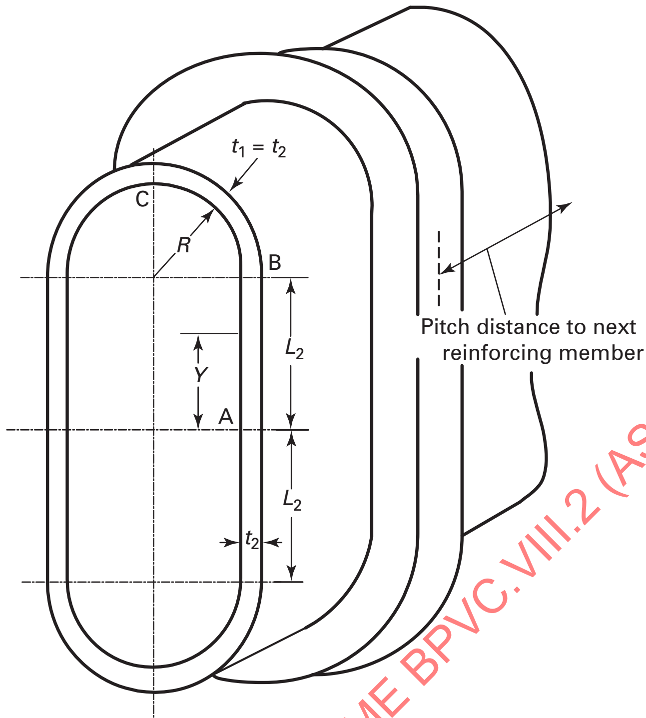
GENERAL NOTE: Critical Locations of Maximum Stress are defined at points A, B, and C.

Figure 4.12.10
Type 9 Noncircular Vessels
(Obround Cross Section)



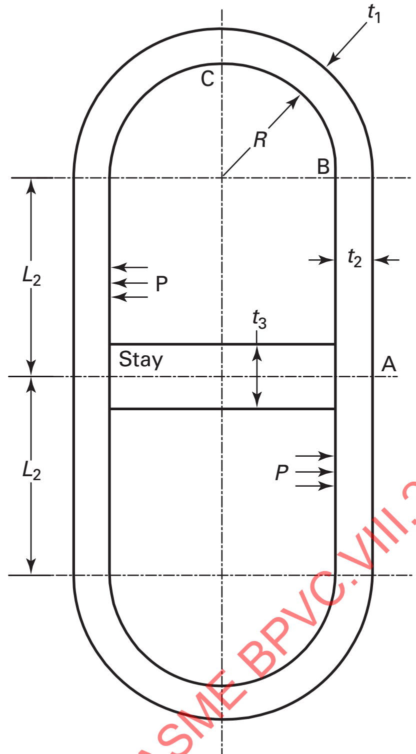
GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, and C.
(b) Defined Locations for Stress Calculations are determined using variable Y.

Figure 4.12.11
Type 10 Noncircular Vessels
(Reinforced Obround Cross Section)



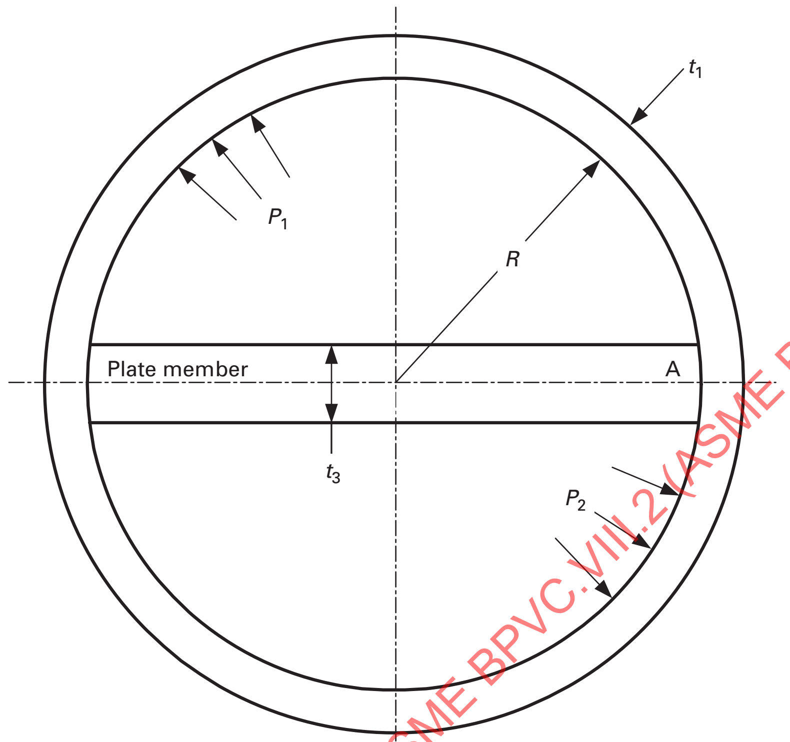
GENERAL NOTES:
(a) Critical Locations of Maximum Stress are defined at points A, B, and C.
(b) Defined Locations for Stress Calculations are determined using variable Y.

Figure 4.12.12
Type 11 Noncircular Vessels
(Obround Cross Section With Single-Stay Plate or Multiple Bars)



GENERAL NOTE: Critical Locations of Maximum Stress are defined at points A, B, and C.

Figure 4.12.13
Type 12 Noncircular Vessels
(Circular Cross Section With Single-Stay Plate)



GENERAL NOTE: Critical Locations of Maximum Stress are defined at point A.

Figure 4.12.14
Multi-Diameter Holes

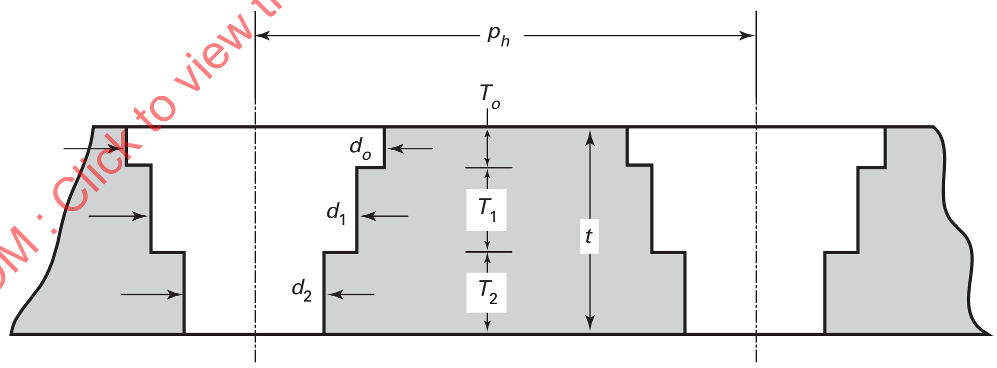
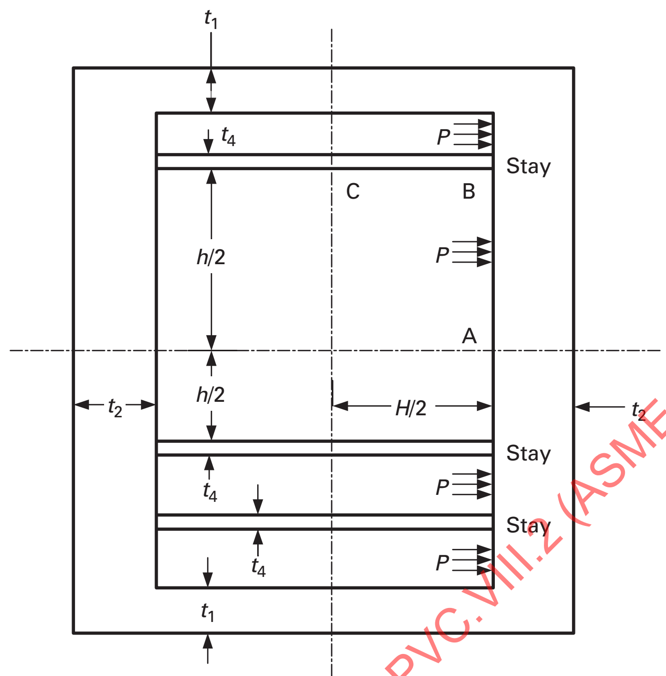
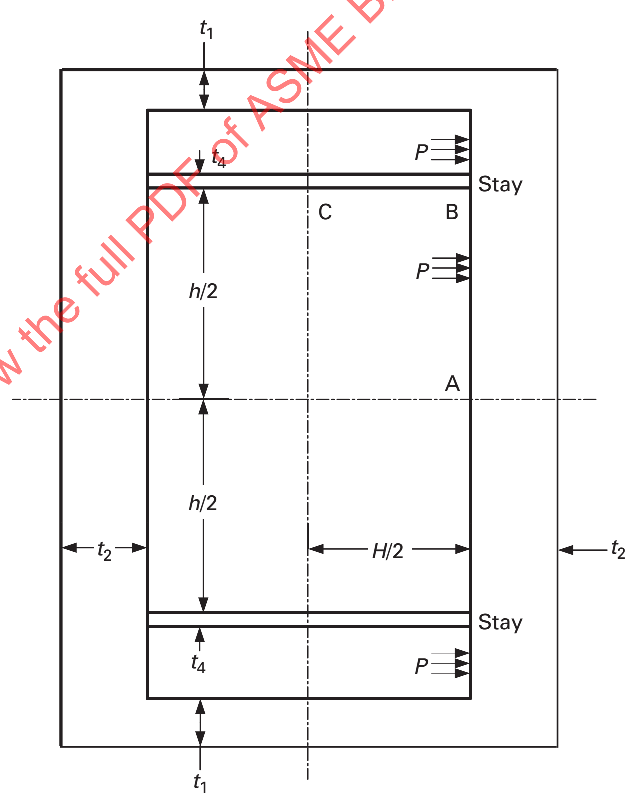


Figure 4.12.15
Rectangular Vessels With Multiple Compartments





GENERAL NOTE: Critical Locations of Maximum Stress are defined at points A, B, and C.

4.13 DESIGN RULES FOR LAYERED VESSELS

4.13.1 SCOPE

Design rules for layered vessels are covered in 4.13. There are several manufacturing techniques used to fabricate layered vessels, and these rules have been developed to cover most techniques used today for which there is extensive documented construction and operational data. Examples of acceptable layered shell and head types are shown in Figures 4.13.1 and 4.13.2.

4.13.2 DEFINITIONS

The following terms are used in this paragraph to define components of a layered vessel.

(a) Layered Vessel - a vessel having a shell and/or heads made up of two or more separate layers.

(b) Inner Shell - the inner cylinder that forms the pressure tight membrane.

(c) Inner Head - the inner head that forms the pressure tight membrane.

(d) Shell Layer - layers may be cylinders formed from plate, sheet, forgings, or the equivalent formed by coiling. This does not include wire winding.

(e) Head Layer - anyone of the head layers of a layered vessel except the inner head.

(f) Overwraps - layers added to the basic shell or head thickness for the purpose of building up the thickness of a layered vessel for reinforcing shell or head openings, or making a transition to thicker sections of the layered vessel.

(g) Dummy Layer - a layer used as a filler between the inner shell (or inner head) and other layers, and not considered as part of the required total thickness.

4.13.3 GENERAL

4.13.3.1 The design for layered pressure vessels shall conform to the general design requirements given in 4.1.

4.13.3.2 A fatigue analysis in accordance with Part 5 shall be performed in all cases unless the fatigue analysis screening based on experience with comparable equipment in accordance with 5.5.2.2 is satisfied.

4.13.3.3 The Manufacturer's Quality Control System shall include the construction procedure that will outline the sequence and method of application of layers and measurement of layer gaps.

4.13.4 DESIGN FOR INTERNAL PRESSURE

4.13.4.1 The total thickness of layered shells of revolution under internal pressure shall not be less than that computed by the equations in 4.3.

4.13.4.2 An inner shell or inner head material that has a lower allowable design stress than the layer materials may only be included as credit for part of the total wall thickness if S_i is not less than $0.5S_L$ by considering its effective thickness to be:

$$t_{\text{eff}} = t_{\text{act}}\left(\frac{S_i}{S_L}\right) \tag{4.13.1}$$

4.13.4.3 Layers in which the stress intensity value of the materials is within 20% of the other layers may be used by prorating the allowable stress from Annex 3-A evaluated at the design temperature of the layers in the thickness equation, provided the materials are compatible in modulus of elasticity and coefficient of thermal expansion (see Part 3).

4.13.4.4 The minimum thickness of any layer shall not be less than 3.2 mm (0.125 in.).

4.13.5 DESIGN FOR EXTERNAL PRESSURE

4.13.5.1 When layered shells are used for external pressure, the requirements of 4.4 shall be applied with the following additional requirements.

(a) The thickness used for establishing external pressure applied to the outer layer shall be the thickness of the total layers, except as given in (b). The design of the vent holes shall be such that the external pressure is not transmitted through the vent holes in the outer layer.

(b) The thickness used for establishing vacuum pressure shall be only the thickness of the inner shell or inner head.

4.13.5.2 Layered shells under axial compression shall be calculated in accordance with 4.4, utilizing the total layered shell thickness.

4.13.6 DESIGN OF WELDED JOINTS

4.13.6.1 The design of welded joints shall conform to the requirements given in 4.2 except as modified herein.

4.13.6.2 Category A and B joints of inner shells and inner heads of layered sections shall be as follows.

(a) Category A joints shall be Type No. 1 (see 4.2).

(b) Category B joints shall be Type No. 1 or Type No. 2 (see 4.2).

4.13.6.3 Category A joints of layered sections shall be as follows.

(a) Category A joints of layers over 22 mm (0.875 in.) in thickness shall be Type No. 1 (see 4.2).

(b) Category A joints of layers 22 mm (0.875 in.) or less in thickness shall be of Type 1 or 2 (see 4.2), except the final outside weld joint of spiral wrapped layered shells may be a single lap weld.

4.13.6.4 Category B joints of layered shell sections to layered shell sections, or layered shell sections to solid shell sections, shall be of Type 1 or 2 (see 4.2).

(a) Category B joints of layered sections to layered sections of unequal thickness shall have transitions as shown in Figure 4.13.3, sketch (a) or (b).

(b) Category B joints of layered sections to solid sections of unequal thickness shall have transitions as shown in Figure 4.13.3, sketch (c), (d), (e), or (f).

(c) Category B joints of layered sections to layered sections of equal thickness shall be as shown in Figure 4.13.4, sketch (b), (c), (d), (f), or (g).

(d) Category B joints of layered sections to solid sections of equal thickness shall be as shown in Figure 4.13.4, sketch (a) or (e).

4.13.6.5 Category A joints of solid hemispherical heads to layered shell sections shall be of Type 1 or 2 (see 4.2).

(a) Transitions shall be as shown in Figure 4.13.5, sketch (a), (b-1), (b-2), or (b-3) when the hemispherical head thickness is less than the thickness of the layered shell section and the transition is made in the layered shell section.

(b) Transitions shall be as shown in Figure 4.13.5, sketch (c), (d-1), or (e) when the hemispherical head thickness is greater than the thickness of the layered shell section and the transition is made in the layered shell section.

(c) Transition shall be as shown in Figure 4.13.5, sketch (f) when the hemispherical head thickness is less than the thickness of the layered shell section and the transition is made in the hemispherical head section.

4.13.6.6 Category B joints of solid elliptical, torispherical, or conical heads to layered shell sections shall be of Type 1 or 2 (see 4.2). Transitions shall be as shown in Figure 4.13.5, sketch (c), (d-1), (d-2), (e), or (f).

4.13.6.7 Category C joints of solid flat heads and tube-sheets to layered shell sections shall be of Type 1 or 2 (see 4.2) as indicated in Figure 4.13.6. Transitions, if applicable, shall be used as shown in Figure 4.13.3, sketch (c), (d), (e), or (f).

4.13.6.8 Category C joints attaching solid flanges to layered shell sections and layered flanges to layered shell sections shall be of Type 1 or 2 (see 4.2) as indicated in Figure 4.13.7.

4.13.6.9 Category A joints of layered hemispherical heads to layered shell sections shall be of Type 1 or 2 (see 4.2) with transition as shown in Figure 4.13.8, sketch (a-1) or (a-2).

4.13.6.10 Category B joints of layered conical heads to layered shell sections shall be of Type 1 or 2 (see 4.2) with transitions as shown in Figure 4.13.8, sketch (b).

4.13.6.11 Category B joints of layered shell sections to layered shell sections or layered shells to solid heads or shells may be butt joints as shown in Figure 4.13.4, sketches (c), (d), and (e), or step welds as shown in Figure 4.13.4, sketches (a), (b), (f), and (g).

4.13.6.12 Category D joints of solid nozzles, manholes, and other connections to layered shell or layered head sections shall be full penetration welds as shown in Figure 4.13.9 except as permitted in sketch (i), (j), (k), or (l). Category D joints between layered nozzles and shells or heads are not permitted.

4.13.6.13 When layers of Category A joints as shown in Figure 4.13.5, sketches (a), (b-1), (b-2), and (b-3) and Figure 4.13.8, sketches (a-1) and (a-2) are welded with fillet welds having a taper less than 3:1, an analysis of the head-to-shell junction shall be done in accordance with Part 5. Resistance due to friction shall not be considered in the analysis. The longitudinal load resisted by the weld shall consider the load transferred from the remaining outer layers.

4.13.7 NOZZLES AND NOZZLE REINFORCEMENT

4.13.7.1 All openings, except as provided in 4.13.7.2 shall meet the requirements for reinforcing per 4.5. All reinforcements required for openings shall be integral with the nozzle or provided in the layered section or both. Additional layers may be included for required reinforcement.

4.13.7.2 Openings, DN 50 (NPS 2) and smaller, need not be reinforced when installed in layered construction but may be welded on the inside as shown in Figure 4.13.9, sketch (j). The nozzle nominal wall thickness shall not be less than Schedule 80 pipe as fabricated in addition to meeting the requirements of 4.5.5.

4.13.7.3 Some acceptable nozzle geometries and attachments are shown in Figure 4.13.9.

4.13.7.4 Openings up to and including 6 in. nominal pipe size (DN 150) may be constructed as shown in Figure 4.13.9, sketches (k) and (I). Such partial penetration weld attachments may only be used for instrumentation openings, inspection openings, etc. on which there are no external mechanical loadings, provided the following requirements are met.

(a) The requirements for reinforcing specified in 4.13.7 apply except that the diameter of the finished openings in the wall shall be d^* as specified in Figure 4.13.9, sketches (k) and (l), and the thickness t_n is the required thickness of the layered shells computed by the design requirements.

(b) Additional reinforcement, attached to the inside surface of the inner shell, may be included after the corrosion allowance is deducted from all exposed surfaces. The attachment welds shall comply with 4.2 and Figure 4.13.9, sketch (k) or (I).

(c) Metal in the nozzle neck available for reinforcement shall be limited by the boundaries specified in 4.5 except that the inner layer shall be considered the shell.

4.13.7.5 Openings greater than 51 mm (2 in.) may be constructed as shown in Figure 4.13.9, sketch (i). The requirements for reinforcing specified in 4.13.7.4(a) apply except that:

(a) The diameter of the finished openings in the walls shall be d' as specified in Figure 4.13.9, sketch (i), and the thickness t_n is the required thickness of the layered shells computed by the design requirements;

(b) Additional reinforcement may be included in the solid hub section as shown in Figure 4.13.9, sketch (i);

(c) Metal in the nozzle neck available for reinforcement shall be limited by the boundaries specified in 4.5, except that the inner layer shall be considered the shell.

4.13.8 FLAT HEADS

4.13.8.1 Design criteria shall meet the requirements of 4.6.

4.13.8.2 The design of welded joints shall be in accordance with 4.13.6.

4.13.9 BOLTED AND STUDDED CONNECTIONS

4.13.9.1 Design criteria shall meet the requirements of 4.16.

4.13.9.2 The design of welded joints shall be in accordance with 4.13.6.

4.13.10 ATTACHMENTS AND SUPPORTS

4.13.10.1 Supports for layered pressure vessels may be designed in accordance with 4.15. Examples of some acceptable supports are shown in Figure 4.13.10.

4.13.10.2 When attaching supports or other connections to the outside or inside of layered pressure vessels, only the immediate layer shall be used in the calculation, except where provisions are made to transfer the load to other layers.

4.13.10.3 When jacketed closures are used, provisions shall be made for extending layer vents through the jacket (see 4.13.11.1). Partial jackets covering only a portion of the circumference are not permitted on layered shells.

4.13.11 VENT HOLES

4.13.11.1 Vent holes shall be provided to detect leakage of the inner shell and to prevent buildup of pressure within the layers as follows.

4.13.11.2 In each shell course or head segment, a layer may be made up of one or more plates. Each layer plate shall have at least two vent holes 6 mm (0.25 in.) minimum diameter. Holes may be drilled radially through the multiple layers or may be staggered in individual layer plates.

4.13.11.3 For continuous coil wrapped layers, each layered section shall have at least four vent holes 6 mm (0.25 in.) minimum diameter. Two of these vent holes shall be located near each end of the section and spaced approximately 180 deg apart.

4.13.11.4 The minimum requirement for spirally wound strip layered construction shall be 6 mm (0.25 in.) minimum diameter vent holes drilled near both edges of the strip. They shall be spaced for the full length of the strip and shall be located a distance of approximately $\pi R/\tan\theta$ from each other (where R is the mean radius of the shell and θ is the acute angle of spiral wrap measured from the longitudinal centerline, deg).

4.13.11.5 If a strip weld covers a vent hole, partially or totally, an additional vent hole shall be drilled on each side of the obstructed hole.

4.13.11.6 In addition to the above, holes may be drilled radially through the multiple layers.

4.13.11.7 Vent holes shall not be obstructed. If a monitoring system is used, it shall be designed to prevent buildup of pressure within the layers.

4.13.12 SHELL TOLERANCES

4.13.12.1 Contact Between Layers. The following requirements shall be satisfied.

(a) Category A weld joints shall be ground to ensure contact between the weld area and the succeeding layer, before application of the layer.

(b) Category A weld joints of layered shell sections shall be in an offset pattern so that the centers of the welded longitudinal joints of adjacent layers are separated circumferentially by a distance of at least five times the layer thickness.

(c) Category A weld joints in layered heads may be in an offset pattern; if offset, the joints of adjacent layers shall be separated by a distance of at least five times the layer thickness.

(d) After weld preparation and before welding circumferential seams, the height of the radial gaps between any two adjacent layers shall be measured at the ends of the layered shell section or layered head section at right angles to the vessel axis, and also the length of the relevant radial gap in inches shall be measured (neglecting radial gaps of less than 0.25 mm (0.010 in.) as non relevant). The gap area, A_g, shall not exceed the thickness of a layer expressed in square inches. An approximation of the area of the gap shall be calculated using eq. (4.13.2). The maximum length of any gap shall not exceed the inside diameter of the vessel. Where more than one gap exists between any two adjacent layers, the sum of the gap lengths shall not exceed the inside diameter of the vessel. The maximum height of any gap shall not exceed 4.8 mm (0.1875 in.). It is recognized that there may be vessels of dimensions wherein it would be desirable to calculate a maximum permissible gap height and length, and also when cyclical service conditions require it. This procedure is provided in 4.13.12.3 and may be used in lieu of the maximum gap area given above, (see Figure 4.13.12.3).

$$A_g = \frac{2}{3} bh \tag{4.13.2}$$

(e) In the case of layered spheres or layered heads, if the gaps cannot be measured as required in (d), measurement of gap heights shall be taken through vent holes in each layer course to assure that the height of layer gaps between any two layers does not exceed the gap permitted in (d). The spacing of the vent holes shall be such that gap lengths can be determined. In the event an excessive gap height is measured through a vent hole, additional vent holes shall be drilled as required to determine the gap length. There shall be at least one vent hole per layer segment.

4.13.12.2 Alternative to Measuring Contact Between Layers During Construction. As an alternative to 4.13.12.1(d), the following measurements shall be taken at the time of the hydrostatic test to check on the contact between successive layers, and the effect of gaps which may or may not be present between layers.

(a) The circumference shall be measured at the midpoint between adjacent circumferential joints, or between a circumferential joint and any nozzle in a shell course. Measurements shall be taken at zero pressure and, following application of hydrostatic test pressure, at the design pressure. The difference in measurements shall be averaged for each course in the vessel and the results recorded as average middle circumferential expansion, e_m.

(b) The theoretical circumferential expansion of a solid vessel of the same dimensions and materials as the layered vessel shall be calculated from eq. (4.13.3). The acceptance criterion for circumferential expansion at the design pressure is: $e_m \geq 0.5 e_{th}$.

$$e_{th} = \frac{1.7\pi P\left(2R_m - t_s\right)^2\left(2R_m + t_s\right)}{8E_y R_m t_s} \tag{4.13.3}$$

4.13.12.3 Rules for Calculating Maximum Permissible Gaps. The maximum number and size of gaps permitted in any cross section of a layered vessel shall be limited by (a) and (b).

(a) The maximum gap between any layers shall be evaluated as follows:

(1) The circumferential stress of the shell and the bending stress due to the gap can be calculated as

$$S = \frac{R_0^2 + R_1^2}{R_0^2 - R_1^2} P \tag{4.13.4}$$

$$S_b = \frac{1.812 E_y h}{R_g} \tag{4.13.5}$$

(2) When $S_b \geq 0.71S$

$$\Delta S_n = 0.5S + S_b + P \quad (4.13.6)$$

(3) When $S_b < 0.71S$

$$\Delta S_n = S + 0.3S_b + P \quad (4.13.7)$$

(4) The stress amplitude for fatigue analysis at the gap is

$$S_{ag} = \frac{K_e \Delta S_n}{2} \quad (4.13.8)$$

where

$$K_e = 1.0 \quad \text{for } \Delta S_n \leq 3S_m$$

$$K_e = 1.0 + \frac{(1 - n)}{n(m - 1)}\left(\frac{\Delta S_n}{3S_m} - 1\right) \quad \text{for } 3S_m < \Delta S_n < 3mS_m$$

$$K_e = \frac{1}{n} \quad \text{for } \Delta S_n \geq 3mS_m$$

(5) The maximum gap shall be determined when the calculated number of fatigue cycles using S_{ag} in eq. (4.13.8) is equal to or greater than the specified number of fatigue cycles.

(b) Maximum permissible number of gaps and their corresponding arc lengths at any cross section of a layered vessel shall be calculated as follows.

(1) Measure each gap and its corresponding length throughout the cross section.

(2) Calculate the value of F for each of the gaps using the following equation:

$$F = 0.109\left(\frac{bh}{R_g^2}\right) \quad (4.13.9)$$

(3) The total sum of the calculated F values shall not exceed the quantity

$$F_T = \frac{1 - \nu^2}{E_y}\left(2S_a/K_e - \frac{2PR_o^2}{R_o^2 - R_i^2}\right) \quad (4.13.10)$$

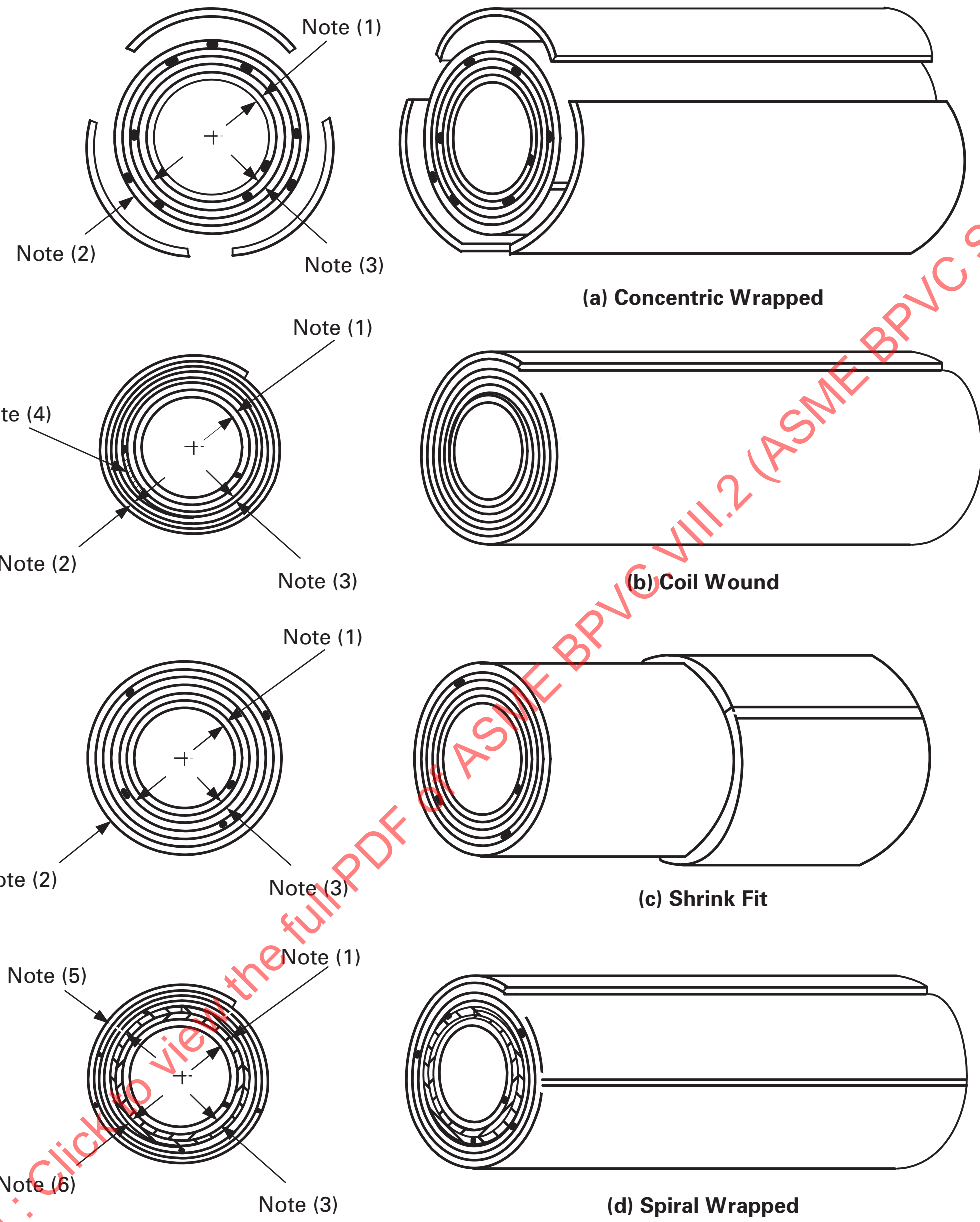
4.13.13 NOMENCLATURE

A_g = gap area.
b = length of the gap between any two layers.
C = equal to 3 mm (0.125 in.) radial clearance between the nozzle neck and vessel opening
$d*$ = finished opening in the wall
e_{th} = theoretical circumferential expansion.
e_m = average middle recorded circumferential expansion.
E_y = Modulus of Elasticity for the layer material from Part 3.
F = gap value.
F_T = total permissible gap value.
h = gap height between any two layers.
K_e = fatigue penalty factor.
m, n = material constants for the fatigue penalty factor used in the simplified elastic–plastic analysis (see Table 5.13).
P = design pressure of the vessel.
R_g = outside radius of the layer above where the gap is located.
R_i = inside radius of the vessel.
R_m = mean radius of the vessel.
R_o = outside radius of the vessel.
S = shell circumferential stress.

S_a = stress amplitude from the applicable fatigue curve for the layer material from Annex 3-F.
S_{ag} = calculated stress amplitude at the gap.
S_b = bending stress due to gap between layered shell.
S_i = allowable stress for the inner layer from Annex 3-A at the design temperature.
S_L = allowable stress for the layers from Annex 3-A at the design temperature.
S_m = allowable stress for the layer material from Annex 3-A at the design temperature.
r_1 = equal to min[$0.25t_n$, 3 mm (0.125 in.)]
r_2 = equal to 6 mm (0.25 in.) minimum
r_3 = equal to min[$0.25t_n$, 19 mm (0.75 in.)]
t = actual thickness of the head or tubesheet or for nozzle details equal to min[t_n, 19 mm (0.75 in.)] , as applicable.
t_{act} = actual thickness of inner shell or inner head.
t_c = equal to the larger of 6 mm (0.25 in.) or 0.7 min[t_n, 19 mm (0.75 in.)]
t_{eff} = effective thickness of inner shell or inner head.
t_H = thickness of the head at the head-to-cylinder joint.
t_L = thickness of the layer.
t_n = nominal thickness of the nozzle wall less corrosion allowance
t_S = total wall thickness of the layered vessel.
Y = offset.
ΔS_n = total stress range.
v = Poisson's ratio.

4.13.14 FIGURES

Figure 4.13.1
Some Acceptable Layered Shell Types



(a) Concentric Wrapped

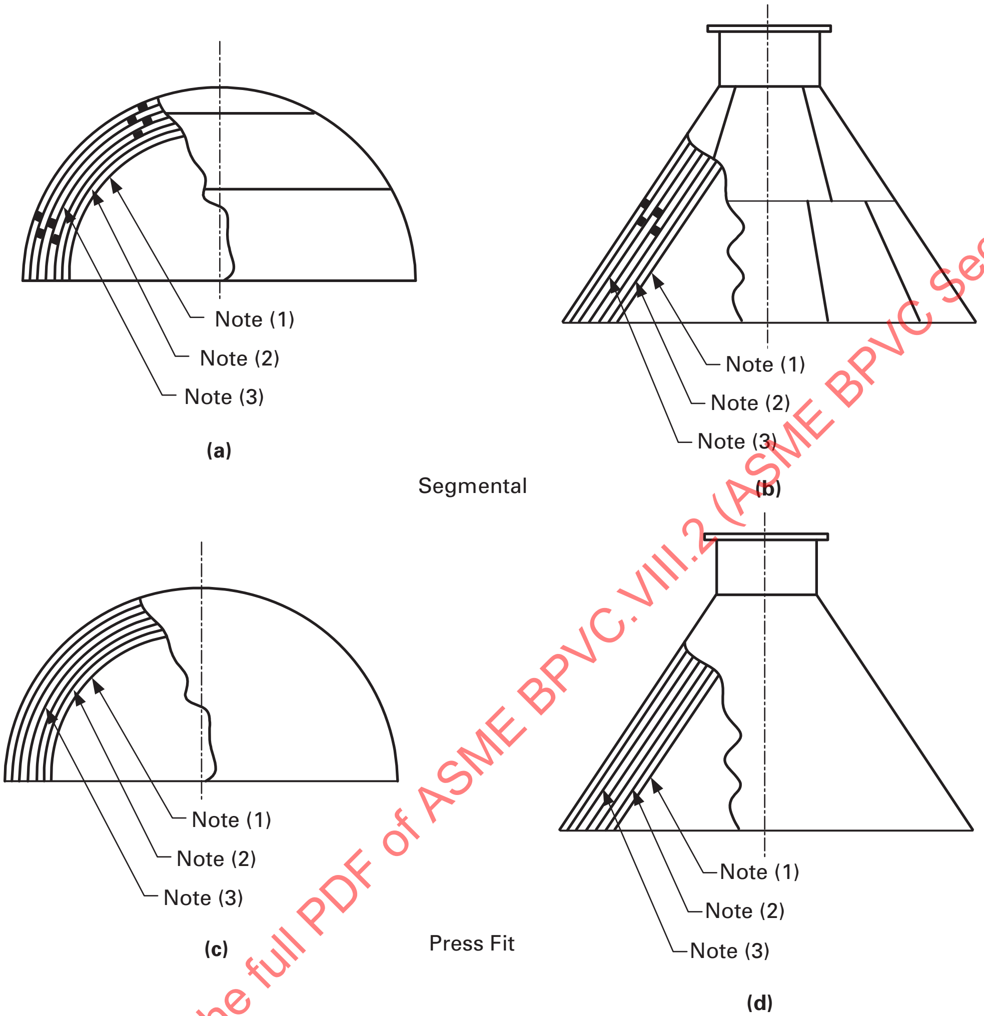
(b) Coil Wound

(c) Shrink Fit

(d) Spiral Wrapped

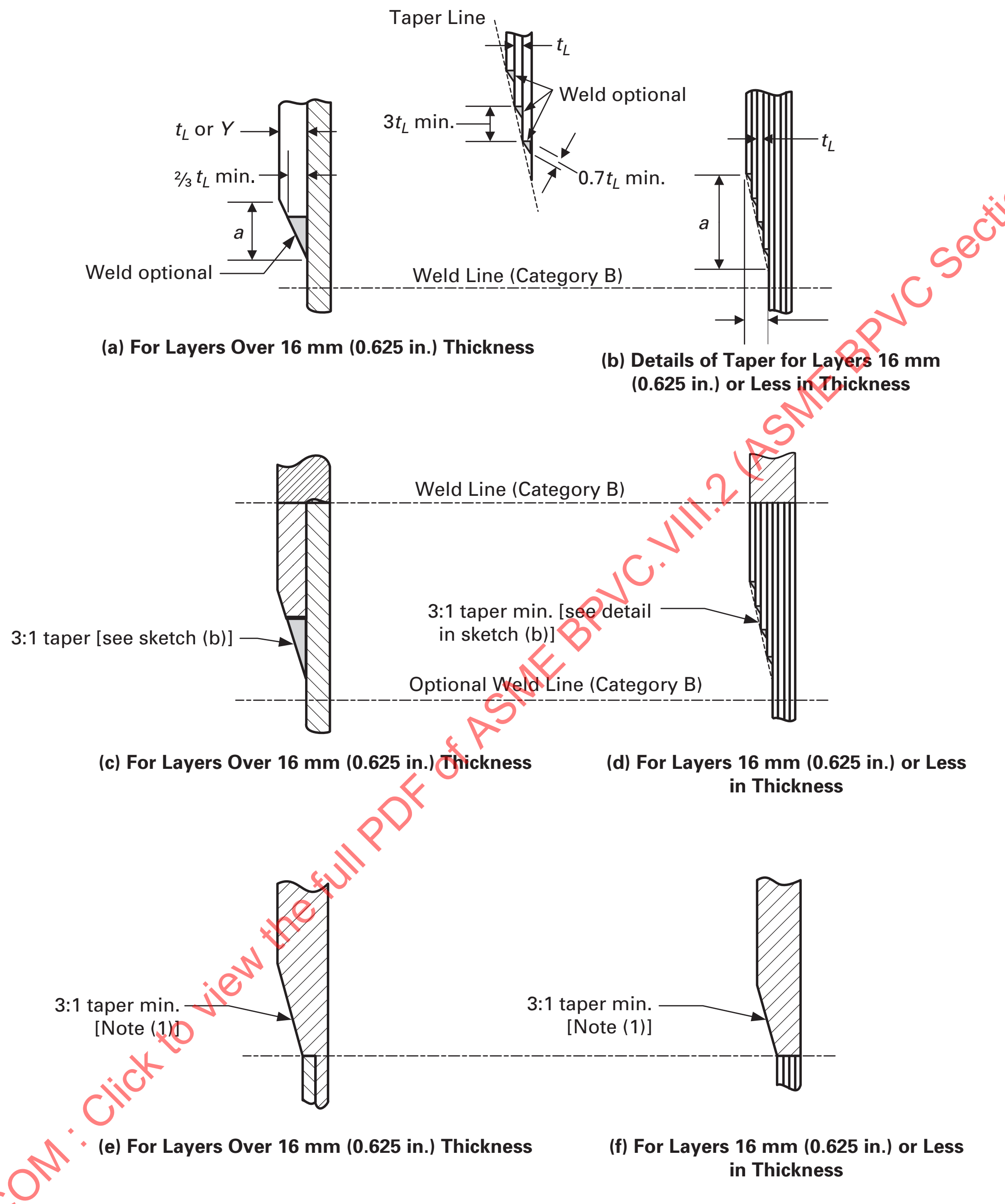
NOTES:
(1) Inner shell
(2) Layers
(3) Dummy layer (if used)
(4) Gap
(5) Balance of layers
(6) Shell layer (tapered)

Figure 4.13.2
Some Acceptable Layered Head Types



NOTES:
(1) Inner head
(2) Dummy layer (if used)
(3) Head layers

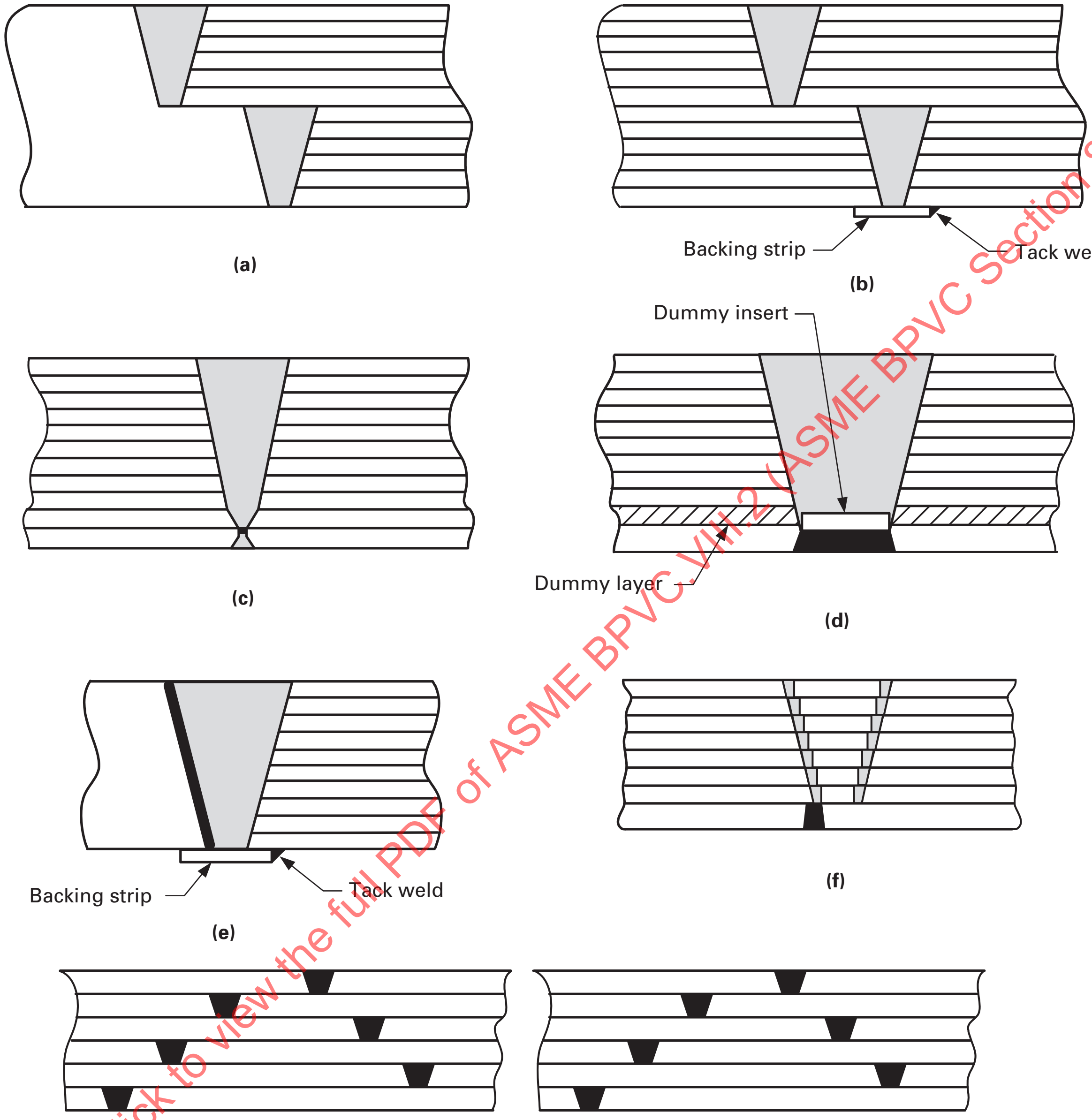
Figure 4.13.3
Transitions of Layered Shell Sections



GENERAL NOTES:
(a) $a \geq 3Y$ where a is the required length of the taper and Y is the offset.
(b) The length of the required taper may include the width of the weld.
(c) The transition may be on either or both sides.

NOTE:
(1) Taper may be inside or outside or both.

Figure 4.13.4
Some Acceptable Welded Joints of Layered-to-Layered and Layered-to-Solid Sections



(g) Butt Girth Welds

Figure 4.13.5
Some Acceptable Solid Head Attachments to Layered Shell Sections

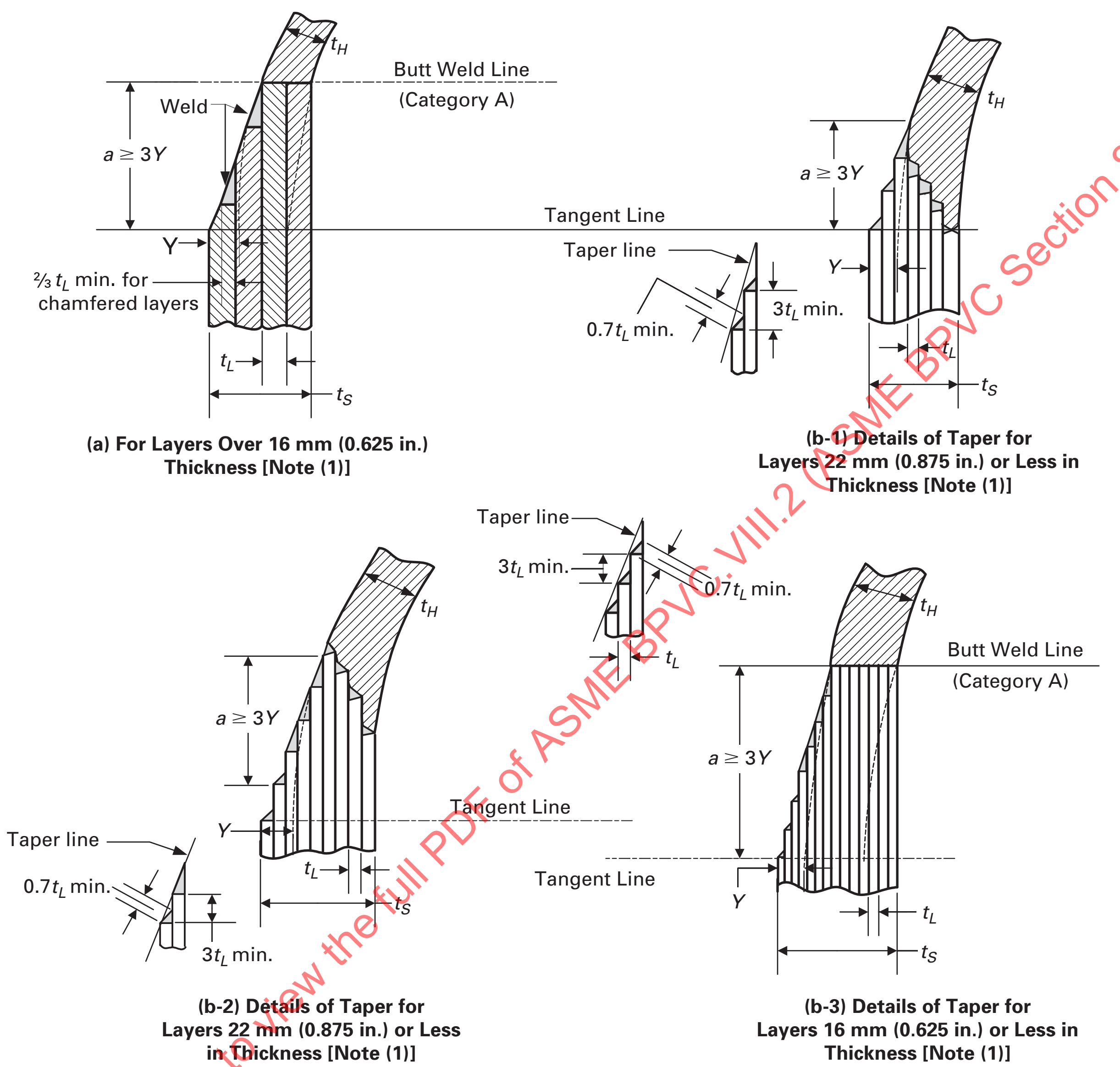
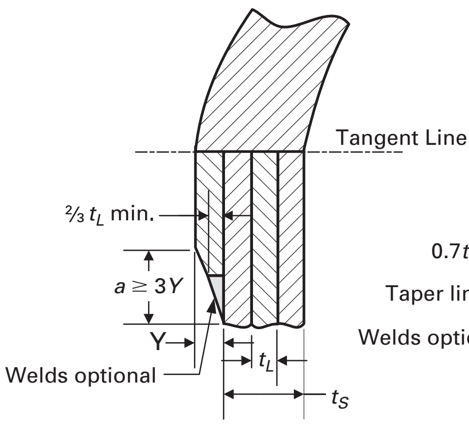
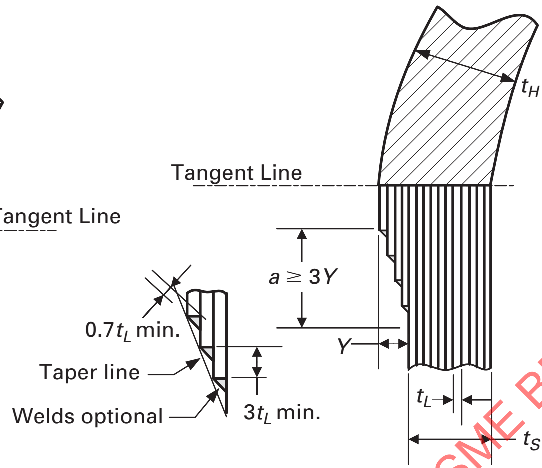


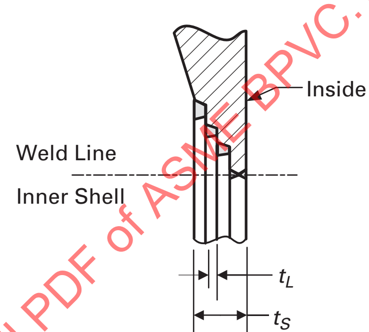
Figure 4.13.5
Some Acceptable Solid Head Attachments to Layered Shell Sections (Cont'd)



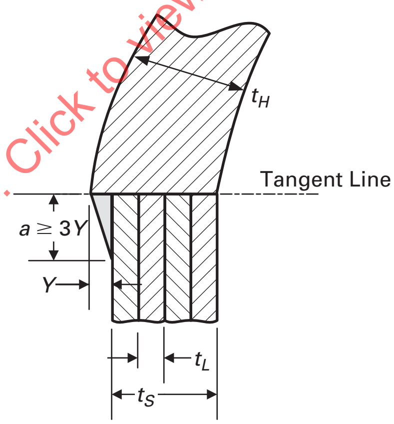
(c) For Layers Over 16 mm (0.625 in.) Thickness [Note (2)]



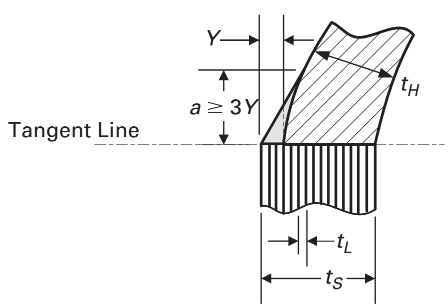
(d-1) Details of Taper for Layers Over 16 mm (0.625 in.) Thickness [Note (2)]



(d-2) Permissible for Layers 22 mm (0.875 in.) or Less in Thickness



(e) For Layers 16 mm (0.625 in.) or Less in Thickness [Note (2)]



(f) For Layers of Any Thickness [Note (2)]

Figure 4.13.5
Some Acceptable Solid Head Attachments to Layered Shell Sections (Cont'd)

GENERAL NOTES:
(a) In sketch (e), $Y \le t_L$ shall be satisfied; in sketch (f), $Y \le 0.5t_s$ shall be satisfied.
(b) In all cases, $a \ge 3Y$ shall be satisfied. The shell centerline may be on either side of the head centerline by a maximum distance of $0.5(t_s - t_H)$. The length of the required taper may include the width of the weld.
(c) The actual thickness shall not be less than the theoretical head thickness.

NOTES:
(1) Sketches (a) through (b-3) apply to hemispherical heads only.
(2) Butt weld line may be at or below tangent line depending on Code requirement for type of head and weld.

Figure 4.13.6
Some Acceptable Flat Heads and Tubesheets With Hubs Joining Layered Shell Sections

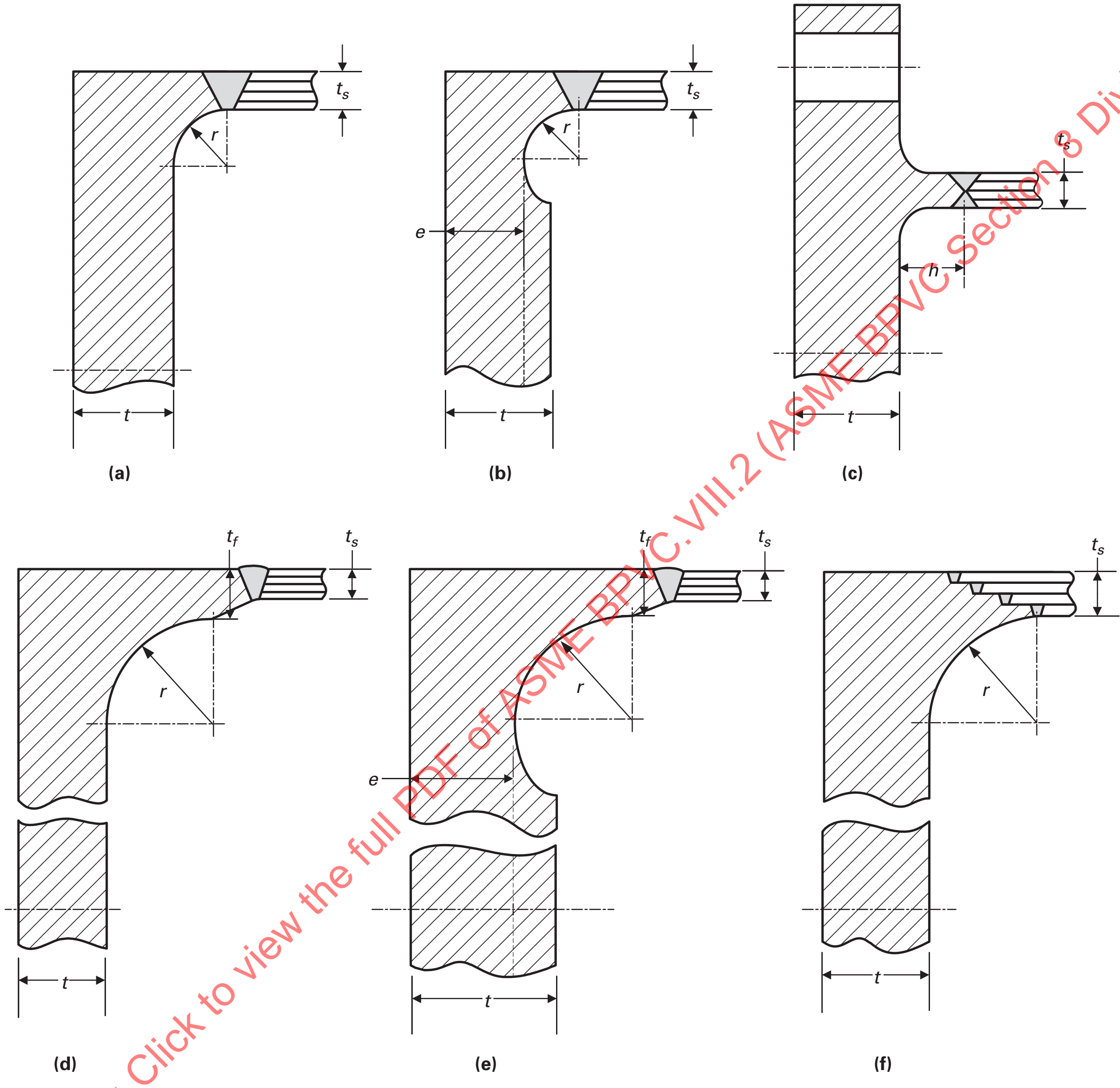
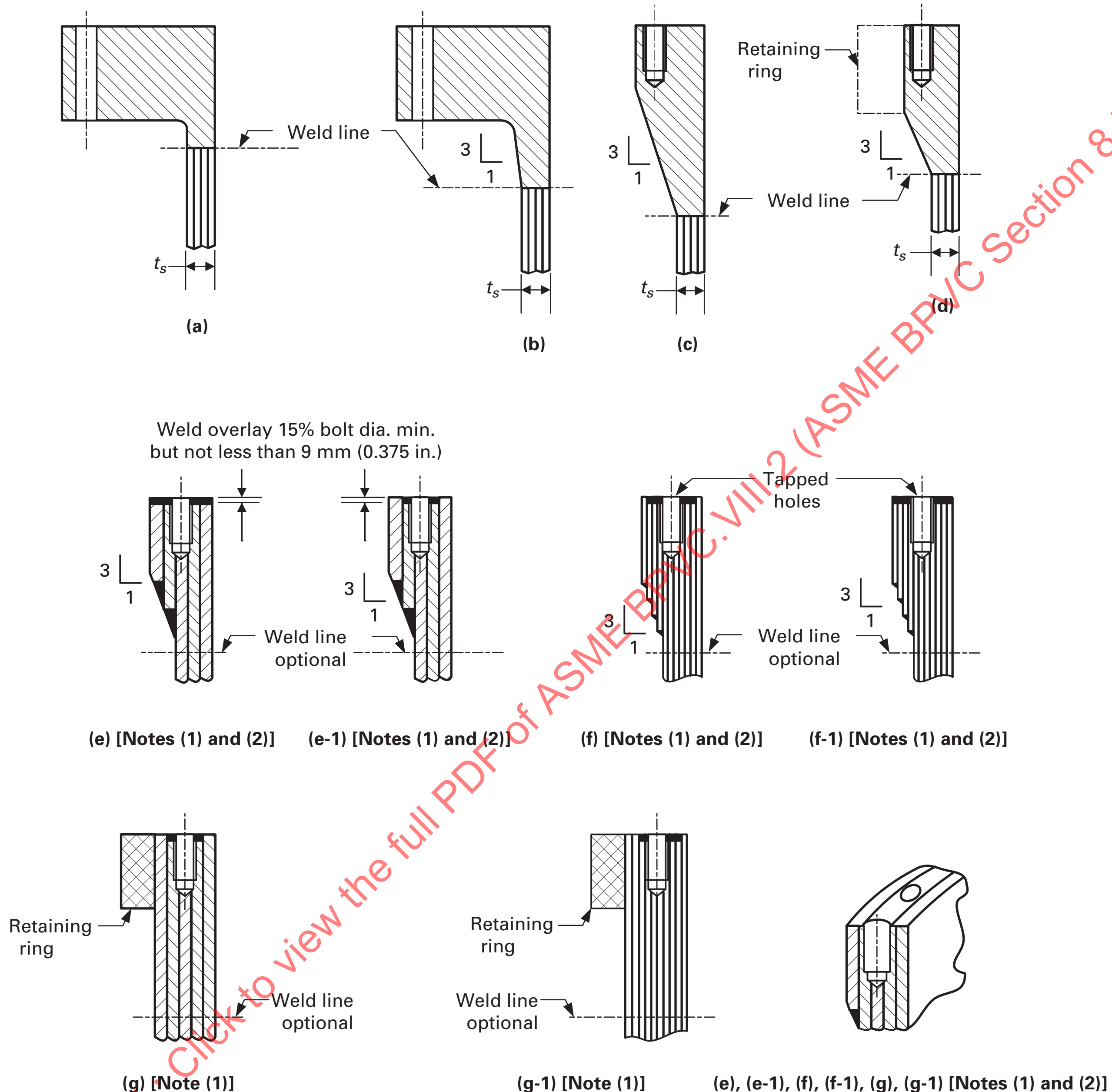


Figure 4.13.7
Some Acceptable Flanges for Layered Shells



NOTES:
(1) The weld overlay shall tie the overlay, the overwraps, and layers together, and the bolt circle shall not exceed the outside diameter of the shell.
(2) The angle of the transition and size of the fillet welds are optional, and the bolt circle shall not exceed the outside diameter of the shell.

Figure 4.13.8
Some Acceptable Layered Head Attachments to Layered Shells

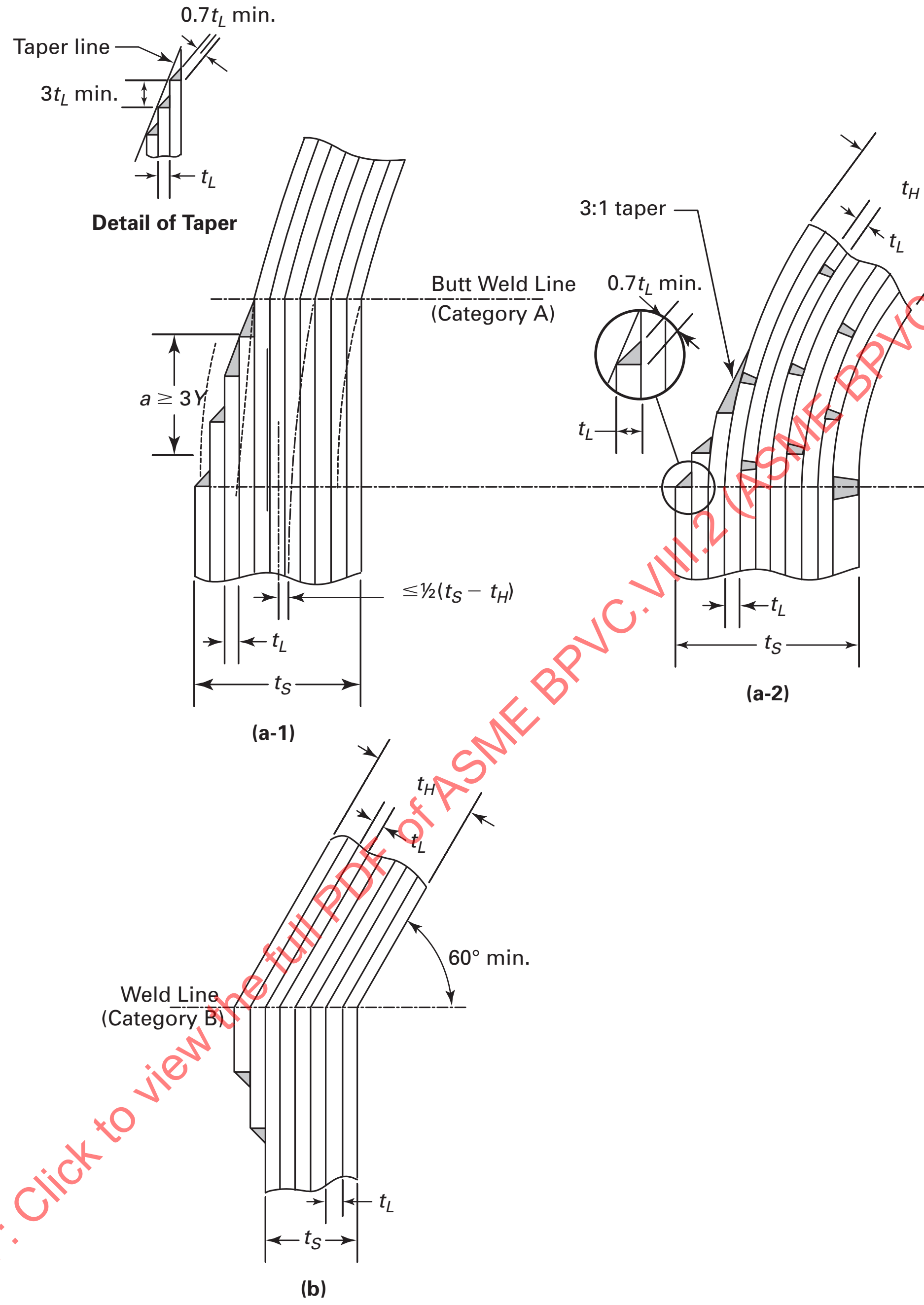


Figure 4.13.9
Some Acceptable Nozzle Attachments to Layered Shell Sections

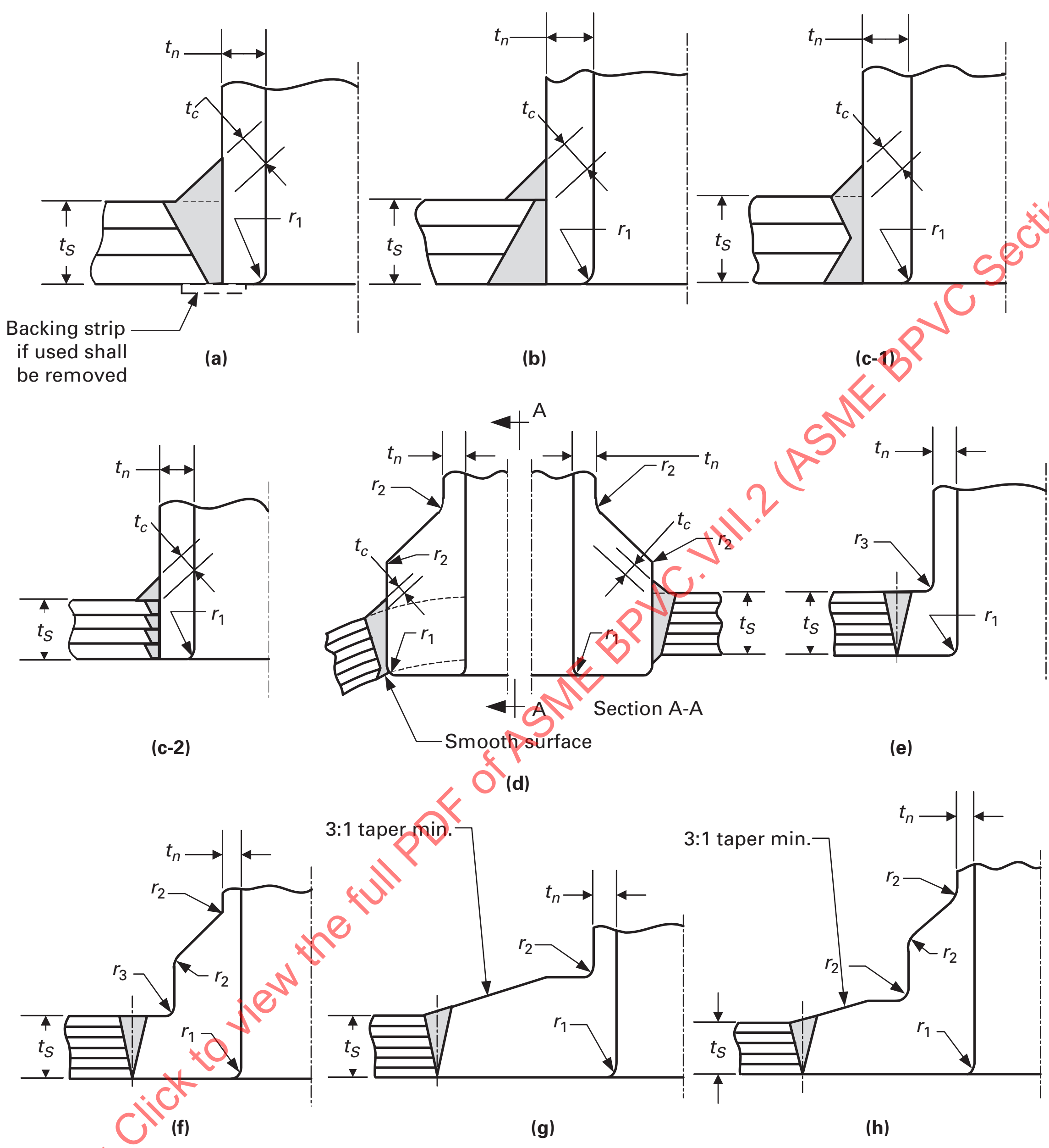
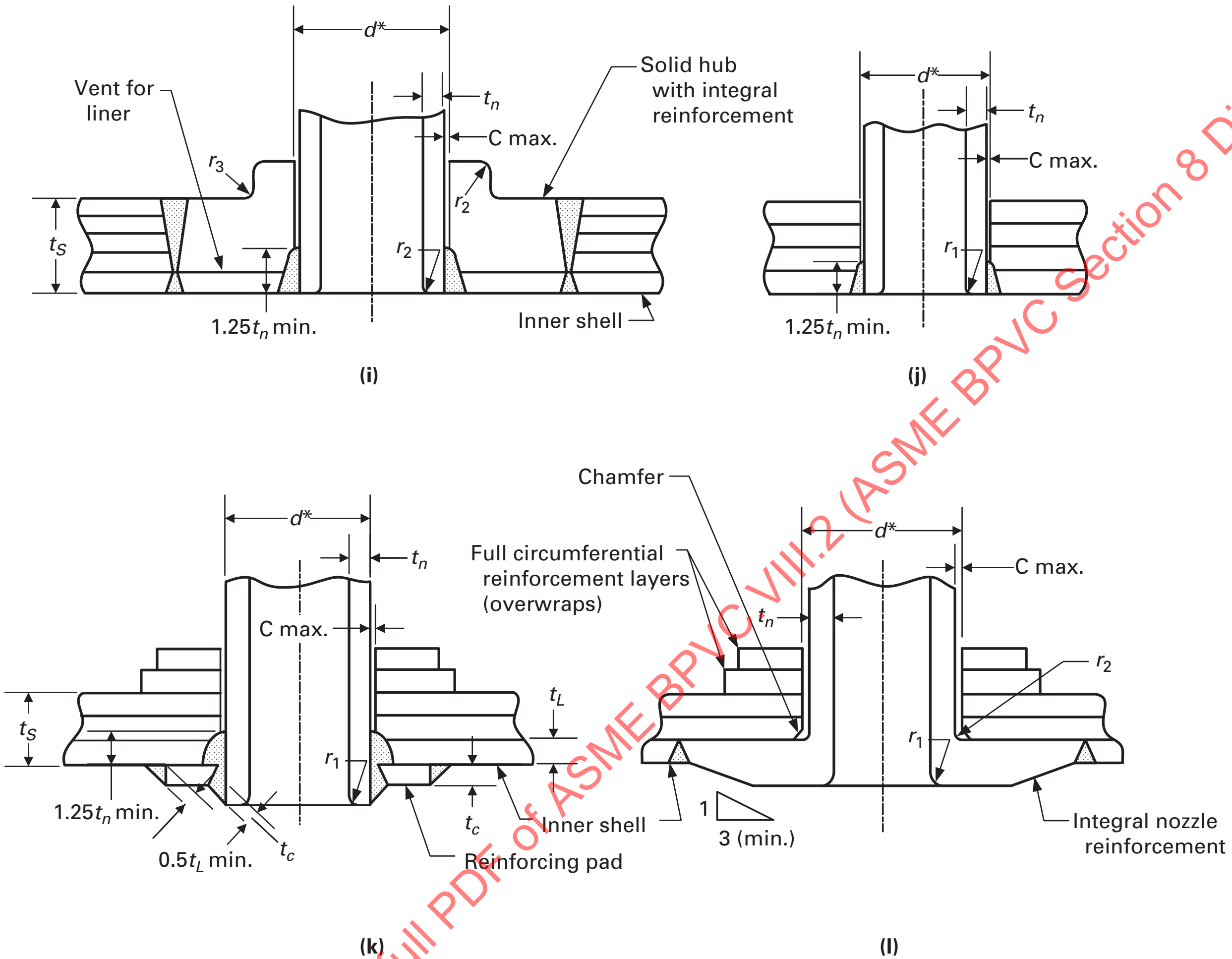


Figure 4.13.9
Some Acceptable Nozzle Attachments to Layered Shell Sections (Cont'd)



GENERAL NOTE: Provide a means, other than by seal welding, to prevent entry of external foreign matter into the annulus between the layers and the nozzle neck outside diameter for sketches (i), (j), (k), and (l).

Figure 4.13.10
Some Acceptable Supports for Layered Vessels

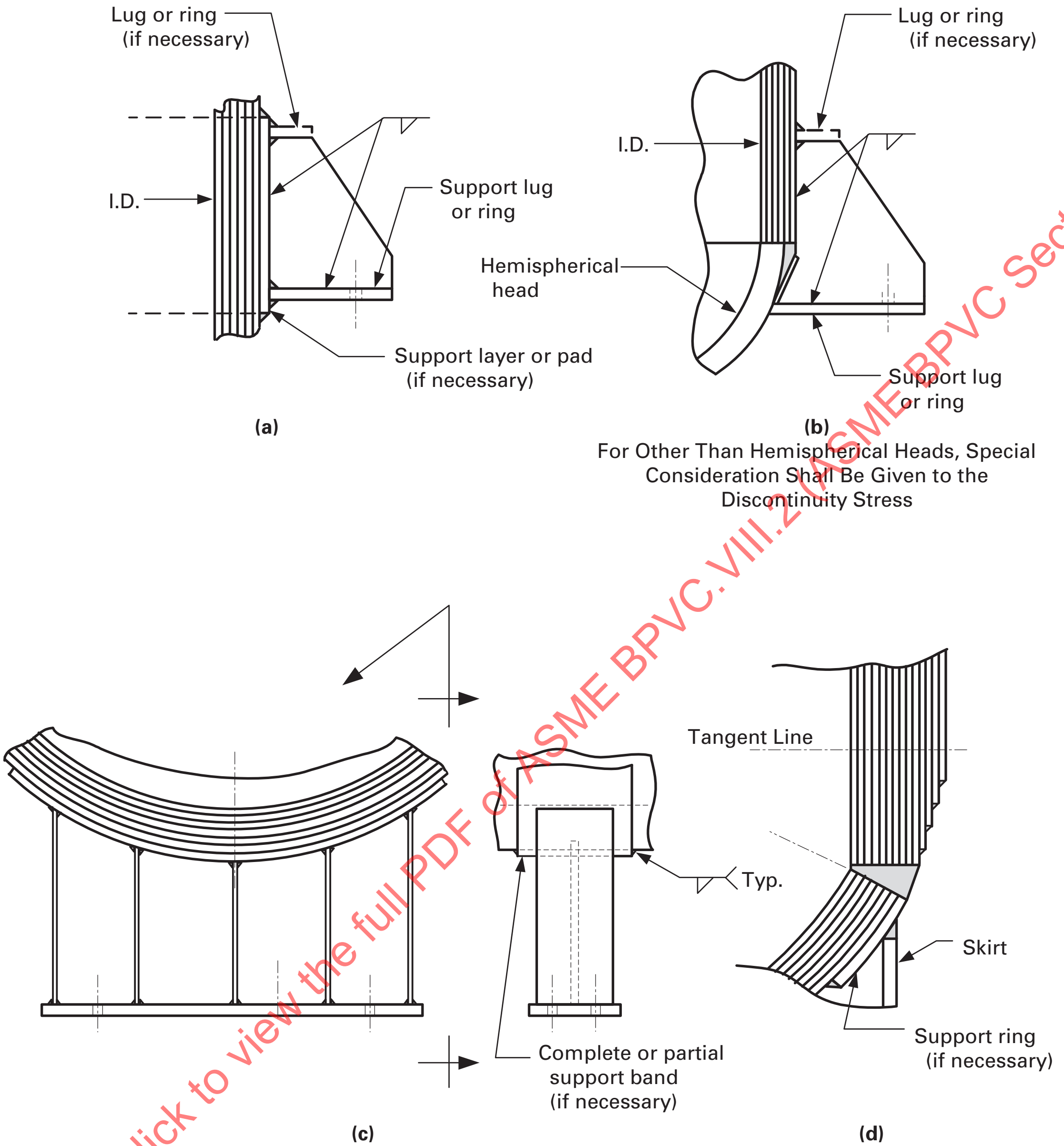
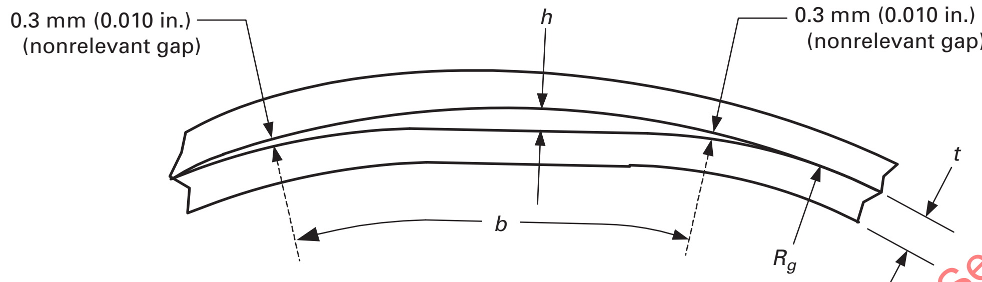


Figure 4.13.11
Gap Between Vessel Layers



4.14 EVALUATION OF VESSELS OUTSIDE OF TOLERANCE

4.14.1 SHELL TOLERANCES

If agreed to by the user, the assessment procedures in Part 5 or in API 579-1/ASME FFS-1 may be used to qualify the design of components that have shell tolerances that do not satisfy the fabrication tolerances in 4.3.2 and 4.4.4. If API 579-1/ASME FFS-1 is used in the assessment, a Remaining Strength Factor of 0.95 shall be used in the calculations unless another value is agreed to by the user. However, the Remaining Strength Factor shall not be less than 0.90. In addition, a fatigue analysis shall be performed in accordance with API 579-1/ASME FFS-1 as applicable.

4.14.2 LOCAL THIN AREAS

4.14.2.1 If agreed to by the user, the assessment procedures in Part 5 or in API 579-1/ASME FFS-1 may be used to qualify the design of components that have a local thin area. A local thin area (LTA) is a region of metal loss on the surface of the component that has a thickness that is less than required by 4.3 and 4.4, as applicable. If API 579-1/ASME FFS-1 is used in the assessment, a Remaining Strength Factor of 0.98 shall be used in the calculations unless another value is agreed to by the user. However, the Remaining Strength Factor shall not be less than 0.90. In addition, a fatigue analysis shall be performed in accordance with API 579-1/ASME FFS-1 as applicable.

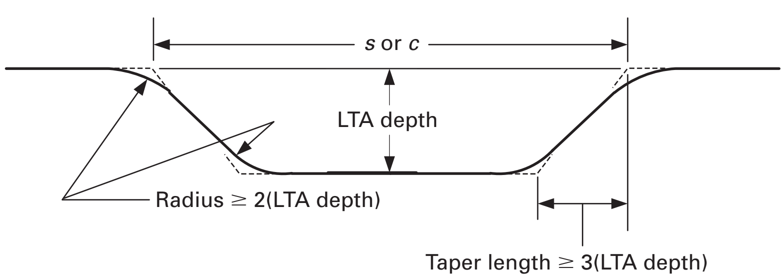
4.14.2.2 The transition between the LTA and the thicker surface shall be made with a taper length not less than three times the LTA depth. The minimum bottom blend radius shall be equal to or greater than two times the LTA depth (see Figure 4.14.1)

4.14.3 MARKING AND REPORTS

The Manufacturer shall maintain records of all calculations including the location and extent of the fabrication tolerances outside the prescribed limits and/or LTAs that are evaluated using 4.14. This information shall be provided to the user if requested and shall be included in the Manufacturer's Design Report.

4.14.4 FIGURES

Figure 4.14.1
LTA Blend Radius Requirements



Blend Grind Detail

4.15 DESIGN RULES FOR SUPPORTS AND ATTACHMENTS

4.15.1 SCOPE

The rules in 4.15 cover requirements for the design of structural support system(s) for vessels. The structural support system may be, but not limited to, saddles for a horizontal vessel, a skirt for a vertical vessel, or lug and leg type supports for either of these vessel configurations.

4.15.2 DESIGN OF SUPPORTS

4.15.2.1 Vessels shall be supported for all specified design conditions. The design conditions including load and load case combinations defined in 4.1.5.3 shall be considered in the design of all vessel supports.

4.15.2.2 Unless otherwise defined in this paragraph, if a stress analysis of the vessel and support attachment configuration is performed, the stress results in the vessel and in the support within the scope of this Division shall satisfy the acceptance criteria in Part 5.

4.15.2.3 The vessel support attachment shall be subject to the fatigue screening criteria of 5.5.2. In this evaluation, supports welded to the vessel may be considered as integral attachments.

4.15.2.4 All supports shall be designed to prevent excessive localized stresses due to deformations produced by the internal pressure or to thermal gradients in the vessel and support system.

4.15.2.5 Vessel support systems composed of structural steel shapes shall be designed in accordance with a recognized code or standard that cover structural design (e.g., Specification for Structural Steel Buildings published by the American Institute of Steel Construction). If the support is at a temperature above ambient due to vessel operation and the recognized code or standard does not provide allowable stresses at temperatures above ambient conditions, then the allowable stress, yield strength, and ultimate tensile strength, as applicable, shall be determined from Annex 3-A and Annex 3-D using a material with a similar minimum specified yield strength and ultimate tensile strength.

4.15.2.6 Attachment welds for structural supports shall be in accordance with 4.2.

4.15.2.7 Reinforcing plates and saddles attached to the outside of a vessel shall be provided with at least one vent hole that may be tapped for a preliminary compressed air and soap solution (or equivalent) test for tightness of welds that seal the edge of the reinforcing plates and saddles. These vent holes may be left open or may be plugged when the vessel is in service. If the holes are plugged, the plugging material used shall not be capable of sustaining pressure between the reinforcing plate and the vessel wall. Vent holes shall not be plugged during heat treatment.

4.15.2.8 If nonpressure parts such as support lugs, brackets, leg supports and saddles extend over pressure-retaining welds, then these welds shall be ground flush for the portion of weld that is covered, or the nonpressure parts shall be notched or coped to clear these welds.

4.15.3 SADDLE SUPPORTS FOR HORIZONTAL VESSELS

(21) **4.15.3.1 Application of Rules.**

(a) Design Method - The design method in this paragraph is based on an analysis of the longitudinal stresses exerted within the cylindrical shell by the overall bending of the vessel, considered as a beam on two single supports, the shear stresses generated by the transmission of the loads on the supports, and the circumferential stresses within the cylindrical shell, the head shear and additional tensile stress in the head, and the possible stiffening rings of this shell, by this transmission of the loads on the supports. The stress calculation method is based on linear elastic mechanics and covers modes of failure by excessive deformation and elastic instability. Alternatively, saddle supports may be designed in accordance with Part 5.

(b) Geometry - A typical horizontal vessel geometry is shown in Figure 4.15.1. Saddle supports for horizontal vessels shall be configured to provide continuous support for at least one-third of the shell circumference, or $\theta \geq 120$ deg.

(c) Reinforcing Plates A reinforcing plate may be included in the design to reduce the circumferential stresses in the cylindrical shell at the saddle support [see 4.15.3.5(c)]. A typical reinforcing plate arrangement is shown in Figure 4.15.2.

(d) Stiffening Rings - Stiffening rings may be used at the saddle support location, on either the inside or outside of the cylindrical shell. The stiffening rings may be mounted in the plane of the saddle (see Figure 4.15.3) or two stiffening rings may be mounted on each side of the saddle support equidistant from the saddle support (see Figure 4.15.4). In the later case, the spacing between the two stiffening rings, h, as shown in Figure 4.15.4 shall not be greater than R_m. If $h \leq 1.56\sqrt{R_m t}$ as shown in Figure 4.15.3, sketch (c), then both of the stiffening rings shall be considered as a single stiffening ring situated in the plane of the saddle in the stress calculations.

4.15.3.2 Moment and Shear Force. (21)

(a) If the vessel is composed of a cylindrical shell with a formed head (i.e., torispherical, elliptical, or hemispherical) at each end that is supported by two saddle supports equally spaced and with $a \le 0.25L$, then the moment at the saddle, M_1, the moment at the center of the vessel, M_2, and the shear force at the saddle, T, may be computed using the following equations.

$$M_1 = -Qa\left(1 - \frac{1 - \frac{a}{L} + \frac{R_m^2 - h_m^2}{2aL}}{1 + \frac{4h_m}{3L}}\right) \quad (4.15.1)$$

$$M_2 = \frac{QL}{4}\left(\frac{1 + \frac{2\left(R_m^2 - h_m^2\right)}{L^2}}{1 + \frac{4h_m}{3L}} - \frac{4a}{L}\right) \quad (4.15.2)$$

$$T = \frac{Q\left(L - 2a\right)}{L + \frac{4h_m}{3}} \quad (4.15.3)$$

(b) If the vessel supports are not symmetric, or more than two supports are provided, then the highest moment in the vessel, and the moment and shear force at each saddle location shall be evaluated. The moments and shear force may be determined using strength of materials (i.e., beam analysis with a shear and moment diagram). If the vessel is supported by more than two supports, then differential settlement should be considered in the design.

4.15.3.3 Longitudinal Stress. (21)

(a) The longitudinal membrane plus bending stresses in the cylindrical shell between the supports are given by the following equations.

$$\sigma_1 = \frac{PR_m}{2t} - \frac{M_2}{\pi R_m^2 t} \qquad \text{(top of shell)} \quad (4.15.4)$$

$$\sigma_2 = \frac{PR_m}{2t} + \frac{M_2}{\pi R_m^2 t} \qquad \text{(bottom of shell)} \quad (4.15.5)$$

(b) The longitudinal stresses in the cylindrical shell at the support location are given by the following equations. The values of these stresses depend on the rigidity of the shell at the saddle support. The cylindrical shell may be considered as suitably stiffened if it incorporates stiffening rings at, or on both sides of the saddle support, or if the support is sufficiently close defined as $a \le 0.5R_m$, to a torispherical or elliptical head (a hemispherical head is not considered a stiffening element), a flat cover, or tubesheet.

(1) Stiffened Shell - The maximum values of longitudinal membrane plus bending stresses at the saddle support are given by the following equations.

$$\sigma_3 = \frac{PR_m}{2t} - \frac{M_1}{\pi R_m^2 t} \qquad \text{(top of shell)} \quad (4.15.6)$$

$$\sigma_4 = \frac{PR_m}{2t} + \frac{M_1}{\pi R_m^2 t} \qquad \text{(bottom of shell)} \quad (4.15.7)$$

(2) Unstiffened Shell - The maximum values of longitudinal membrane plus bending stresses at the saddle support are given by the following equations. The coefficients K_1 and K^*_1 are given in Table 4.15.1.

$$\sigma_3^* = \frac{PR_m}{2t} - \frac{M_1}{K_1\,\pi R_m^2 t} \qquad \text{(points A and B in Figure 4.15.5)} \quad (4.15.8)$$

$$\sigma_4^* = \frac{PR_m}{2t} + \frac{M_1}{k_1^*\,\pi R_m^2 t} \qquad \text{(bottom of shell)} \quad (4.15.9)$$

(c) Acceptance Criteria

(1) The absolute value of σ_1, σ_2, and σ_3, σ_4 or σ^*_3, σ^*_4, as applicable shall not exceed SE.

(2) If the any of the stresses in (a) or (b) above are negative, the absolute value of the stress shall not exceed S_c that is given by eq. (4.15.10) where K = 1.0 for normal operating conditions and K = 1.35 for exceptional operating or hydrotest condition.

$$S_c = \frac{KtE_y}{16R_m} \tag{4.15.10}$$

(21) **4.15.3.4 Shear Stresses.**

(a) The shear stress in the cylindrical shell with a stiffening ring in the plane of the saddle support is a maximum at Points C and D of Figure 4.15.5, sketch (b) and shall be computed using eq. (4.15.11).

$$\tau_1 = \frac{T}{\pi R_m t} \tag{4.15.11}$$

(b) The shear stress in the cylindrical shell with stiffening rings on both sides of the saddle support is a maximum at Points E and F of Figure 4.15.5, sketch (c) and shall be computed using eq. (4.15.12). The coefficient K_2 is given in Table 4.15.1.

$$\tau_2 = \frac{K_2 T}{R_m t} \tag{4.15.12}$$

(c) The shear stress in a cylindrical shell without stiffening ring(s) that is not stiffened by a formed head, flat cover, or tubesheet, ($a > 0.5R_m$) is also at Points E and F of Figure 4.15.5, sketch (c) and shall be computed using eq. (4.15.12).

(d) The shear stress in the cylindrical shell without stiffening ring(s) and stiffened by a torispherical or elliptical head, flat cover, or tubesheet, ($a \le 0.5R_m$) is a maximum at Points E and F of Figure 4.15.5, sketch (c) and shall be computed using the equations shown below. In addition to the shear stress, the membrane stress in the formed head, if applicable, shall also be computed using the equations shown below.

(1) Shear stress, the coefficient K_3 is given in Table 4.15.1.

$$\tau_3 = \frac{K_3 Q}{R_m t} \quad \text{(in the cylindrical shell)} \tag{4.15.13}$$

$$\tau_3^* = \frac{K_3 Q}{R_m t_h} \quad \text{(in the formed head)} \tag{4.15.14}$$

(2) Membrane stress in a torispherical or elliptical head acting as a stiffener, the coefficient K_4 is given in Table 4.15.1.

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{PR_{mh}}{2t_h} \quad \text{(torispherical head)} \tag{4.15.15}$$

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{PR_{mh}}{2t_h}\left(\frac{R_{mh}}{h_m}\right) \quad \text{(elliptical head)} \tag{4.15.16}$$

$$\sigma_5 = 0 \quad \text{(flat cover)} \tag{4.15.17}$$

(e) Acceptance Criteria

(1) The absolute value of τ_1, τ_2, and τ_3, as applicable, shall not exceed min(0.8S, 0.533S_y).

(2) The absolute value of $\tau^*{}_3$ shall not exceed min(0.8S_h, 0.533S_{hy}).

(3) The absolute value of σ_5 shall not exceed 1.25S_h.

(21) **4.15.3.5 Circumferential Stress.**

(a) Maximum circumferential bending moment - the distribution of the circumferential bending moment at the saddle support is dependent on the use of stiffeners at the saddle location.

(1) Cylindrical shell without a stiffening ring or with a stiffening ring in the plane of the saddle - the maximum circumferential bending moment is shown in Figure 4.15.6, sketch (a) and shall be computed using eq. (4.15.18). The coefficient K_7 is given in Table 4.15.1.

$$M_\beta = K_7 Q R_m \tag{4.15.18}$$

(2) Cylindrical shell with stiffening rings on both side of the saddle - the maximum circumferential bending moment is shown in Figure 4.15.6, sketch (b) and shall be computed using eq. (4.15.19). The coefficient K_{10} is given in Table 4.15.1.

$$M_\beta = K_{10} Q R_m \tag{4.15.19}$$

(b) Width of cylindrical shell - the width of the cylindrical shell that contributes to the strength of the cylindrical shell at the saddle location shall be determined using eq. (4.15.20). If the width x_1 extends beyond the limits in Figures 4.15.2, 4.15.3 or 4.15.4, as applicable, then the width x_1 shall be reduced such as not to exceed this limit.

$$x_1, x_2 \leq 0.78\sqrt{R_m t} \tag{4.15.20}$$

(c) Circumferential stresses in the cylindrical shell without stiffening ring(s)

(1) The maximum compressive circumferential membrane stress in the cylindrical shell at the base of the saddle support shall be computed using eq. (4.15.21). The coefficient K_5 is given in Table 4.15.1.

$$\sigma_6 = \frac{-K_5 Q k}{t\left(b + x_1 + x_2\right)} \tag{4.15.21}$$

(2) The circumferential compressive membrane plus bending stress at Points G and H of Figure 4.15.6, sketch (a) is determined as follows. The coefficient K_7 is given in Table 4.15.1.

(-a) If $L \geq 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using eq. (4.15.22).

$$\sigma_7 = \frac{-Q}{4t\left(b + x_1 + x_2\right)} - \frac{3K_7 Q}{2t^2} \tag{4.15.22}$$

(-b) If $L < 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using eq. (4.15.23).

$$\sigma_7^* = \frac{-Q}{4t\left(b + x_1 + x_2\right)} - \frac{12K_7 Q R_m}{L t^2} \tag{4.15.23}$$

(3) The stresses σ_6, σ_7, and σ^*_7 may be reduced by adding a reinforcement or wear plate at the saddle location that is welded to the cylindrical shell.

(-a) If the width of the reinforcement plate, b_1, satisfies eq. (4.15.24), the stress σ_6 can be computed as shown in eq. (4.15.25).

$$b_1 = \min\left[\left(b + 1.56\sqrt{R_m t}\right), 2a\right] \tag{4.15.24}$$

$$\sigma_{6,r} = \frac{-K_5 Q k}{b_1\left(t + \eta t_r\right)} \tag{4.15.25}$$

where

$$\eta = \min\left[\frac{S_r}{S}, 1.0\right] \tag{4.15.26}$$

(-b) If the reinforcement plate provides a supporting arc length, θ_1 , that satisfies eq. (4.15.27), the stresses σ_7 and σ^*_7 can be computed as shown in eq. (4.15.28) and eq. (4.15.29), respectively.

$$\theta_1 = \theta + \frac{\theta}{12} \tag{4.15.27}$$

$$\sigma_{7,r} = \frac{-Q}{4\left(t + \eta t_r\right) b_1} - \frac{3K_7 Q}{2\left(t + \eta t_r\right)^2} \tag{4.15.28}$$

$$\sigma_{7,r}^* = \frac{-Q}{4\left(t + \eta t_r\right) b_1} - \frac{12K_7 Q R_m}{L\left(t + \eta t_r\right)^2} \tag{4.15.29}$$

(4) If $t_r > 2t$, then the compressive membrane plus bending stress at the ends of the reinforcing plate [points G1 and H1 in Figure 4.15.2, sketch (b)] shall be computed using the equations shown below. In these equations, coefficient $K_{7,1}$ is computed using the equation for K_7 in Table 4.15.1 evaluated at the angle θ_1, see eq. (4.15.27).

(-a) If $L \geq 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using eq. (4.15.30)

$$\sigma_{7,1} = \frac{-Q}{4t\left(b + x_1 + x_2\right)} - \frac{3K_{7,1}Q}{2t^2} \tag{4.15.30}$$

(-b) If $L < 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using eq. (4.15.31).

$$\sigma^*_{7,1} = \frac{-Q}{4t\left(b + x_1 + x_2\right)} - \frac{12K_{7,1}QR_m}{Lt^2} \tag{4.15.31}$$

(d) Circumferential stresses in the cylindrical shell with a stiffening ring along the plane of the saddle support.

(1) The maximum compressive circumferential membrane stress in the cylindrical shell shall be computed using eq. (4.15.32). The coefficient K_5 is given in Table 4.15.1.

$$\sigma^*_6 = \frac{-K_5Qk}{A} \tag{4.15.32}$$

(2) The circumferential compressive membrane plus bending stress at Points G and H of Figure 4.15.6, sketch (a) for stiffening rings located on the inside of the shell are determined as follows. The coefficients K_8 and K_6 are given in Table 4.15.1.

$$\sigma_8 = \frac{-K_8Q}{A} - \frac{K_6QR_mC_1}{I} \qquad \text{(stress in the shell)} \tag{4.15.33}$$

$$\sigma_9 = \frac{-K_8Q}{A} + \frac{K_6QR_mc_2}{I} \qquad \text{(stress in the stiffening ring)} \tag{4.15.34}$$

(3) The circumferential compressive membrane plus bending stress at Points G and H of Figure 4.15.6, sketch (a) for stiffening rings located on the outside of the shell are determined as follows. The coefficients K_8 and K_6 are given in Table 4.15.1.

$$\sigma^*_8 = \frac{-K_8Q}{A} + \frac{K_6QR_mC_1}{I} \qquad \text{(stress in the shell)} \tag{4.15.35}$$

$$\sigma^*_9 = \frac{-K_8Q}{A} - \frac{K_6QR_mC_2}{I} \qquad \text{(stress in the stiffening ring)} \tag{4.15.36}$$

(e) Circumferential stresses in the cylindrical shell with stiffening rings on both sides of the saddle support

(1) The maximum compressive circumferential membrane stress in the cylindrical shell shall be computed using eq. (4.15.37). The coefficient K_5 is given in Table 4.15.1.

$$\sigma_6 = \frac{-K_5Q_k}{t\left(b + 2x_2\right)} \tag{4.15.37}$$

(2) The circumferential compressive membrane plus bending stress at Points I and J of Figure 4.15.6, sketch (b) for stiffening rings located on the inside of the shell are determined as follows. The coefficients K_9 and K_{10} are given in Table 4.15.1.

$$\sigma_{10} = \frac{-K_9Q}{A} + \frac{K_{10}QR_mc_1}{I} \qquad \text{(stress in the shell)} \tag{4.15.38}$$

$$\sigma_{11} = \frac{-K_9Q}{A} - \frac{K_{10}QR_mc_2}{I} \qquad \text{(stress in the stiffening ring)} \tag{4.15.39}$$

(3) The circumferential compressive membrane plus bending stress at Points I and J of Figure 4.15.6, sketch (b) for stiffening rings located on the outside of the shell are determined as follows. The coefficients K_9 and K_{10} are given in Table 4.15.1.

$$\sigma_{10}^{*} = \frac{-K_9 Q}{A} - \frac{K_{10} Q R_m c_1}{I} \quad \text{(stress in the shell)} \tag{4.15.40}$$

$$\sigma_{11}^{*} = \frac{-K_9 Q}{A} + \frac{K_{10} Q R_m c_2}{I} \quad \text{(stress in the stiffening ring)} \tag{4.15.41}$$

(f) Acceptance Criteria

(1) The absolute value of σ_6 or $\sigma_{6,r}$, as applicable, shall not exceed S.

(2) The absolute value of σ^*_6, as applicable, shall not exceed min[S, S_r].

(3) The absolute value of σ_7, σ^*_7, $\sigma_{7,r}$, $\sigma^*_{7,r}$, $\sigma_{7,1}$, $\sigma^*_{7,1}$, σ_8, σ^*_8, σ_{10}, and σ^*_{10}, as applicable, shall not exceed $1.25S$.

(4) The absolute value of σ_9, σ^*_9, σ_{11}, and σ^*_{11}, as applicable, shall not exceed $1.25S_s$.

4.15.3.6 Saddle Support. The horizontal force at the minimum section at the low point of the saddle is given by eq. (4.15.42). The saddle shall be designed to resist this force.

$$F_h = Q\left(\frac{1 + \cos\beta - 0.5\sin^2\beta}{\pi - \beta + \sin\beta\,\cos\beta}\right) \tag{4.15.42}$$

4.15.4 SKIRT SUPPORTS FOR VERTICAL VESSELS

4.15.4.1 The following shall be considered in the design of vertical vessels supported on skirts.

(a) The skirt reaction

(1) The weight of vessel and contents transmitted in compression to the skirt by the shell above the level of the skirt attachment;

(2) The weight of vessel and contents transmitted to the skirt by the weight in the shell below the level of skirt attachment;

(3) The load due to externally applied moments and forces when these are a factor, e.g., wind, earthquake, or piping loads.

(b) Localized Stresses at The Skirt Attachment Location - High localized stresses may exist in the shell and skirt in the vicinity of the skirt attachment if the skirt reaction is not in line with the vessel wall. When the skirt is attached below the head tangent line, localized stresses are introduced in proportion to the component of the skirt reaction which is normal to the head surface at the point of attachment. When the mean diameter of the skirt and shell approximately coincide (see Figure 4.15.7) and a minimum knuckle radius in accordance with 4.3 is used, the localized stresses are minimized. In other cases an investigation of local effects may be warranted depending on the magnitude of the loading, location of skirt attachment, etc., and an additional thickness of vessel wall or compression rings may be necessary. Localized stresses at the skirt attachment location may be evaluated by the design by analysis methods in Part 5.

(c) Thermal Gradients - Thermal gradients may produce high localized stresses in the vicinity of the vessel to skirt attachment. A "hot-box" detail (see Figure 4.15.8) shall be considered to minimize thermal gradients and localized stresses at the skirt attachment to the vessel wall. If a hot-box is used, the thermal analysis shall consider convection and thermal radiation in the hot-box cavity.

4.15.4.2 The rules of 4.3.10 shall be used to determine the thickness requirements for the skirt support. Alternatively, skirt supports may be designed using the design by analysis methods in Part 5.

4.15.5 LUG AND LEG SUPPORTS

4.15.5.1 Lug supports may be used on horizontal or vertical vessels.

(21) **4.15.5.2** The localized stresses at the lug support locations on the shell may be evaluated using one of the following methods. If an acceptance criterion is not provided, the results from this analysis shall be evaluated in accordance with Part 5.

(a) Part 5 of this Division.

(b) Welding Research Council Bulletin Number 537, Local Stresses in Spherical and Cylindrical Shells Due to External Loadings.

(c) Welding Research Council Bulletin 198, Part 1, Secondary Stress Indices for Integral Structural Attachments to Straight Pipes; Part 2, Stress Indices at Lug Supports on Piping Systems.

(d) Welding Research Council Bulletin 353, Position Paper on Nuclear Plant Pipe Supports.

(e) Welding Research Council Bulletin 448, Evaluation of Welded Attachments on Pipe and Elbows.

(f) Other analytical methods contained in recognized codes and standards for pressure vessel construction (i.e., British Standard PD-5500, Specification for Fusion Welded Pressure Vessels (Advanced Design and Construction) for Use in the Chemical, Petroleum, and Allied Industries).

4.15.5.3 If vessels are supported by lugs, legs, or brackets attached to the shell, then the supporting members under these bearing attachments should be as close to the shell as possible to minimize local bending stresses in the shell.

4.15.5.4 Supports, lugs, brackets, stiffeners, and other attachments may be attached with stud bolts to the outside or inside of a vessel wall.

4.15.5.5 Lug and column supports should be located away from structural discontinuities (i.e., cone-to-cylinder junctions) and Category A or B weld seams. If these supports are located within $1.8\sqrt{Dt}$ of these locations, then a stress analysis shall be performed and the results from this analysis shall be evaluated in accordance with 4.15.5.2.

(21) 4.15.6 NOMENCLATURE

A = cross-sectional area of the stiffening ring(s) and the associated shell width used in the stress calculation.
a = distance from the axis of the saddle support to the tangent line on the curve for a dished head or to the inner face of a flat cover or tubesheet.
b = width of contact surface of the cylindrical shell and saddle support.
b_1 = width of the reinforcing plate welded to the cylindrical shell at the saddle location
c_1, c_2 = distance to the extreme axes of the cylinder-stiffener cross section to the neutral axis of the cylinder-stiffener cross-section
E_y = modulus of elasticity.
E = weld joint efficiency (see 4.2.4) for the circumferential weld seam being evaluated.
η = shell to reinforcing plate strength reduction factor.
F_h = saddle horizontal force.
h = spacing between two mounted stiffening rings placed on each side of the saddle support.
h_m = mean depth of formed head.
I = moment of inertia of cross-sectional area A in relation to its neutral axis that is parallel to the axis of the cylindrical shell.
k = factor to account for the vessel support condition; k = 1 is the vessel is resting on the support and k = 0.1 is the vessel is welded to the support.
K = factor to set the allowable compressive stress for the cylindrical shell material.
L = length of the cylindrical shell measured from tangent line to tangent line for a vessel with dished heads or from the inner face to inner face for vessels with flat covers or tubesheets.
M_1 = net-section maximum longitudinal bending moment at the saddle support; this moment is negative when it results in a tensile stress on the top of the shell.
M_2 = net-section maximum longitudinal bending moment between the saddle supports; this moment is positive when it results in a compressive stress on the top of the shell.
P = design pressure, positive for internal pressure and negative for external pressure.
Q = maximum value of the reaction at the saddle support from weight and other loads as applicable.
R_{mh} = mean radius of the spherical dome or a torispherical head.
R_m = mean radius of the cylindrical shell.
S = allowable stress from Annex 3-A for the cylindrical shell material at the design temperature.
S_c = allowable compressive stress for the cylindrical shell material at the design temperature.
S_h = allowable stress from Annex 3-A for the head material at the design temperature.
S_{hy} = yield strength from Annex 3-A for the head material at the design temperature.
S_r = allowable stress from Annex 3-A for the reinforcing plate material at the design temperature.

S_s = allowable stress from Annex 3-A for the stiffener material at the design temperature.
S_y = yield strength from Annex 3-A for the cylindrical shell material at the design temperature.
t = cylindrical shell or shell thickness, as applicable.
t_h = head thickness.
t_r = reinforcing plate thickness.
T = maximum shear force at the saddle.
θ = opening of the supported cylindrical shell arc.
θ_1 = opening of the cylindrical shell arc engaged by a welded reinforcing plate.
x_1,x_2 = width of cylindrical shell used in the circumferential normal stress strength calculation.

4.15.7 TABLES

Table 4.15.1
Stress Coefficients for Horizontal Vessels on Saddle Supports

Stress Coefficient

$$K_1 = \frac{\Delta + \sin\Delta \cdot \cos\Delta - \dfrac{2\sin^2\Delta}{\Delta}}{\pi\left(\dfrac{\sin\Delta}{\Delta} - \cos\Delta\right)}$$

$$K_1^* = \frac{\Delta + \sin\Delta \cdot \cos\Delta - \dfrac{2\sin^2\Delta}{\Delta}}{\pi\left(1 - \dfrac{\sin\Delta}{\Delta}\right)}$$

$$K_2 = \frac{\sin\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}$$

$$K_3 = \left(\frac{\sin\alpha}{\pi}\right)\left(\frac{\alpha - \sin\alpha \cdot \cos\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}\right)$$

$$K_4 = \frac{3}{8}\left(\frac{\sin^2\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}\right)$$

$$K_5 = \frac{1 + \cos\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}$$

$$K_6 = \frac{\dfrac{3\cos\beta}{4}\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{5\sin\beta\cos^2\beta}{4\beta} + \dfrac{\cos^3\beta}{2} - \dfrac{\sin\beta}{4\beta} + \dfrac{\cos\beta}{4} - \beta\sin\beta\left[\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{1}{2} - \dfrac{\sin 2\beta}{4\beta}\right]}{2\pi\left[\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{1}{2} - \dfrac{\sin 2\beta}{4\beta}\right]}$$

$$K_7 = \frac{K_6}{4} \qquad \text{when } \frac{a}{R_m} \le 0.5$$

$$K_7 = \frac{3}{2}K_6\left(\frac{a}{R_m}\right) - \frac{1}{2}K_6 \qquad \text{when } 0.5 < \frac{a}{R_m} < 1$$

$$K_7 = K_6 \qquad \text{when } \frac{a}{R_m} \ge 1$$

$$K_8 = \frac{\cos\beta\left[1 - \dfrac{\cos 2\beta}{4} + \dfrac{9\sin\beta\cos\beta}{4\beta} - 3\left(\dfrac{\sin\beta}{\beta}\right)^2\right]}{2\pi\left[\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{1}{2} - \dfrac{\sin 2\beta}{4\beta}\right]} + \frac{\beta\sin\beta}{2\pi}$$

$$K_9 = \frac{1}{2\pi}\left\{\left[-\frac{1}{2} + (\pi - \beta)\cot\beta\right]\cos\rho + \rho\sin\rho\right\}$$

$$K_{10} = \frac{1}{2\pi}\left\{\rho\sin\rho + \cos\rho\left[\frac{3}{2} + (\pi - \beta)\cot\beta\right] - \frac{(\pi - \beta)}{\sin\beta}\right\}$$

NOTES:

(1) $\Delta = \dfrac{\pi}{6} + \dfrac{5\theta}{12}$

(2) $\alpha = 0.95\left(\pi - \dfrac{\theta}{2}\right)$

(3) $\beta = \pi - \dfrac{\theta}{2}$

(4) The relationship between ρ and θ is given by

$\rho = \tan\rho\,[0.5 + (\pi - \beta)\cot\beta]$

Values for ρ for a specified θ are shown in the table below.

Relationship Between ρ and θ

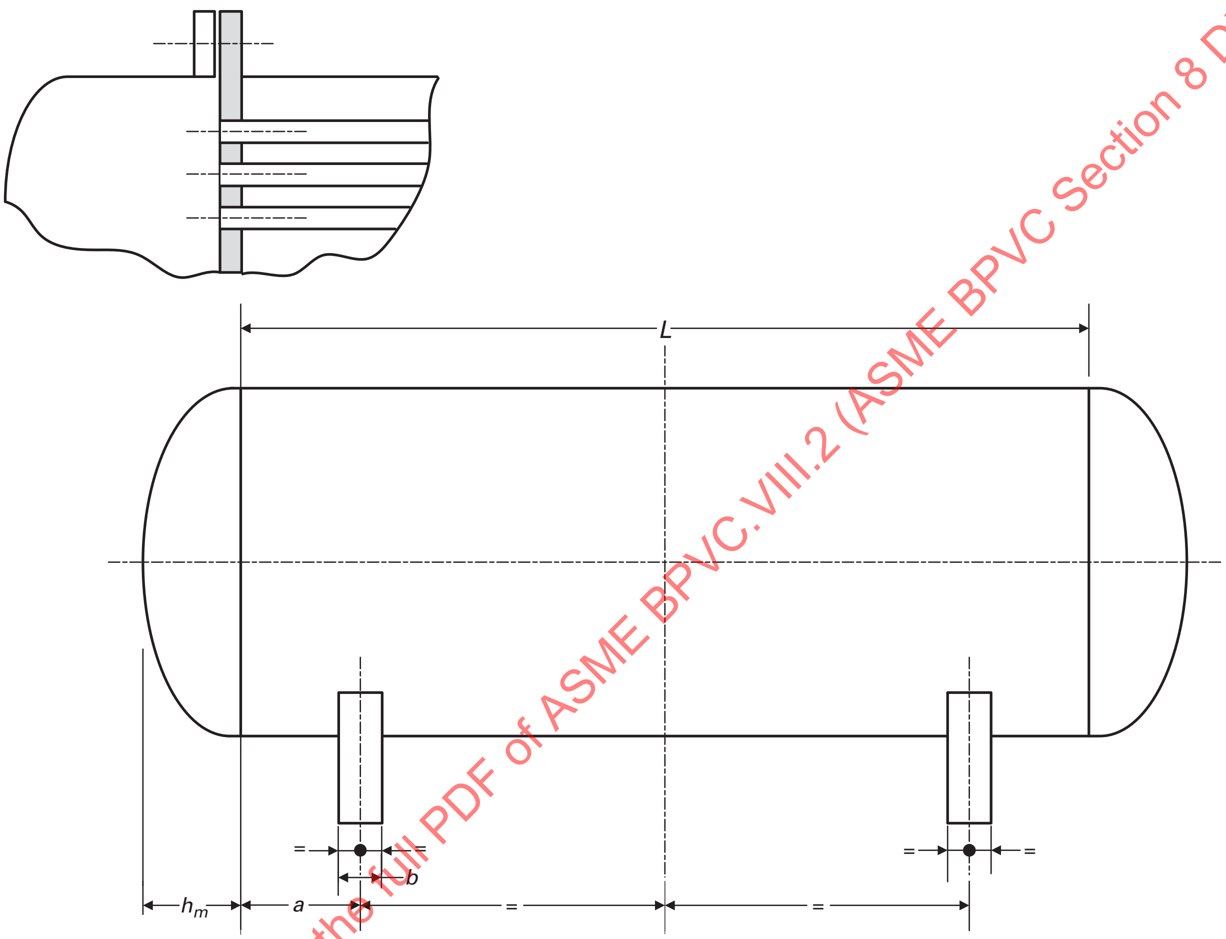
θ	120°	130°	140°	150°	160°	170°	180°
ρ	93.667°	91.133°	87.833°	84.167°	79.667°	74°	66.933°

GENERAL NOTE:
$\rho = -158.58 + 7.86668\theta - 8.8037(10)^{-2}\theta^2 + 4.3011(10)^{-4}\theta^3 - 8.0644(10)^{-7}\theta^4$ for all values of θ that satisfy $120° \le \theta \le 180°$. This curve fit provides ρ in degrees.

(5) The angles Δ, θ, β, and ρ are in radians in the calculations.

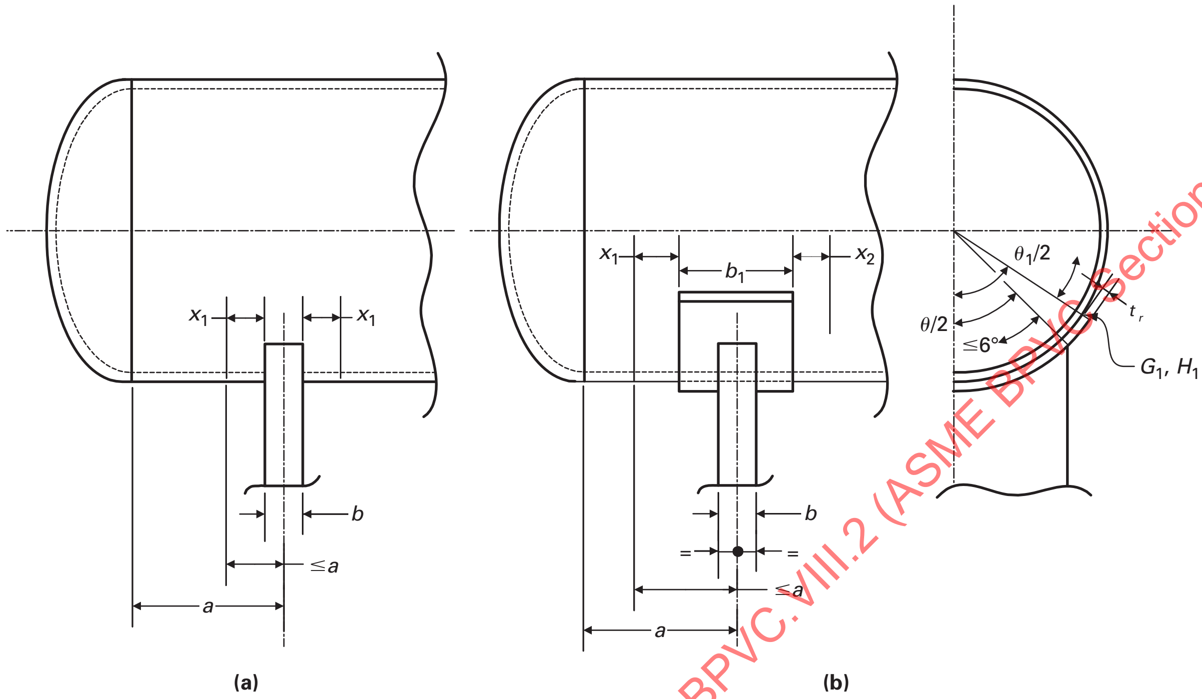
4.15.8 FIGURES

Figure 4.15.1
Horizontal Vessel on Saddle Supports (21)



(21)

Figure 4.15.2
Cylindrical Shell Without Stiffening Rings



(21)

Figure 4.15.3
Cylindrical Shell With Stiffening Rings in the Plane of the Saddle

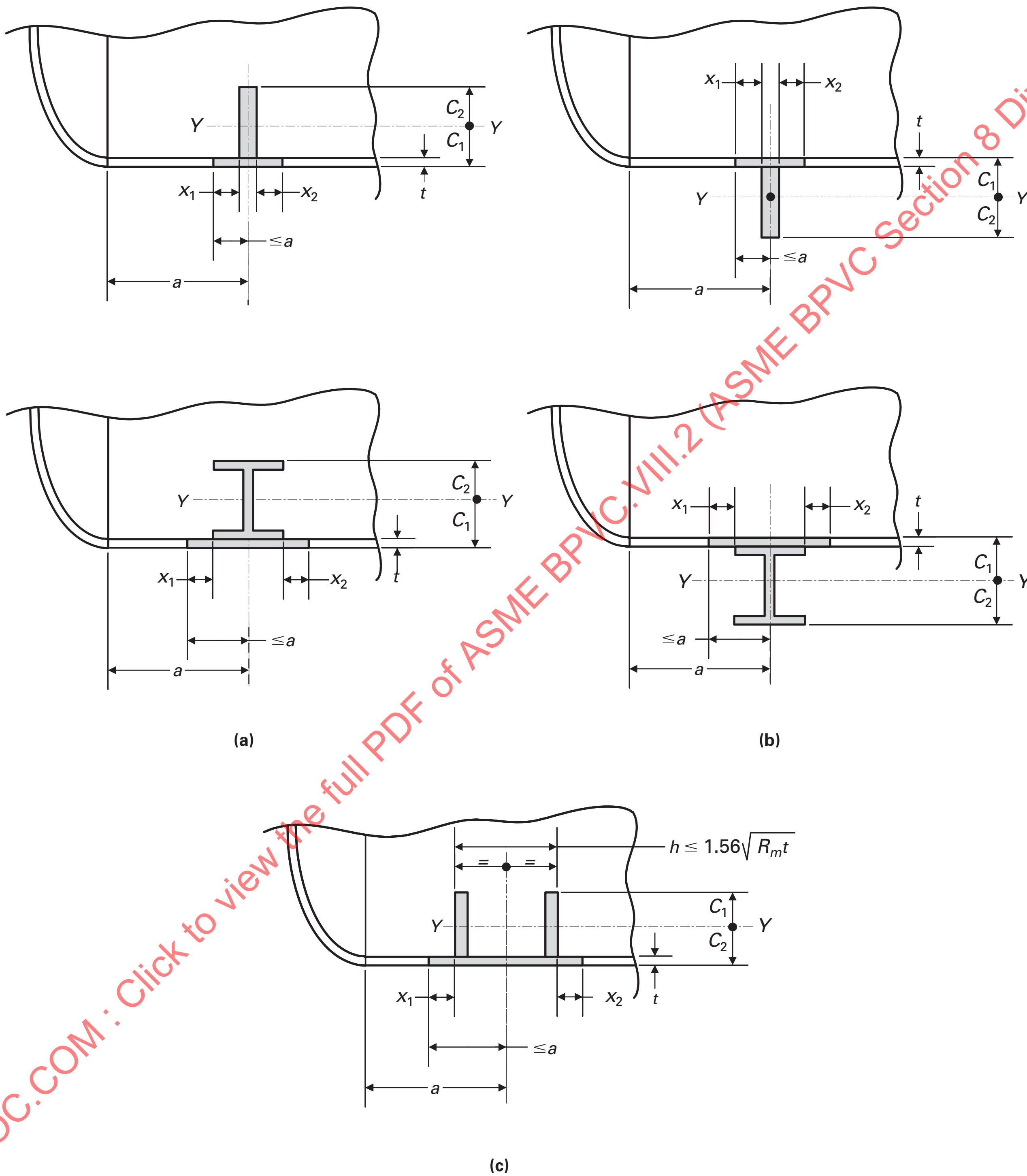


Figure 4.15.4
Cylindrical Shell With Stiffening Rings on Both Sides of the Saddle

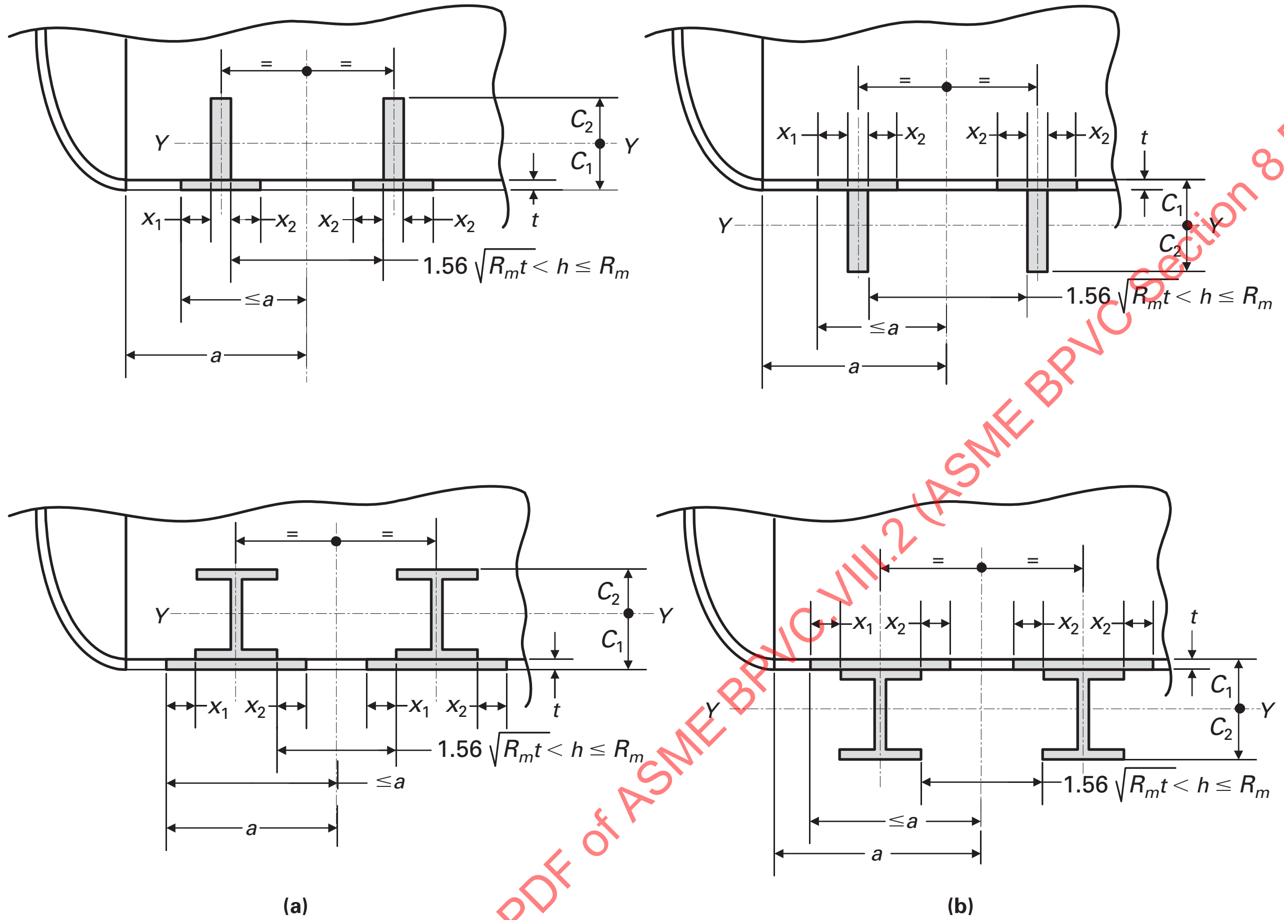


Figure 4.15.5
Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Cylinder

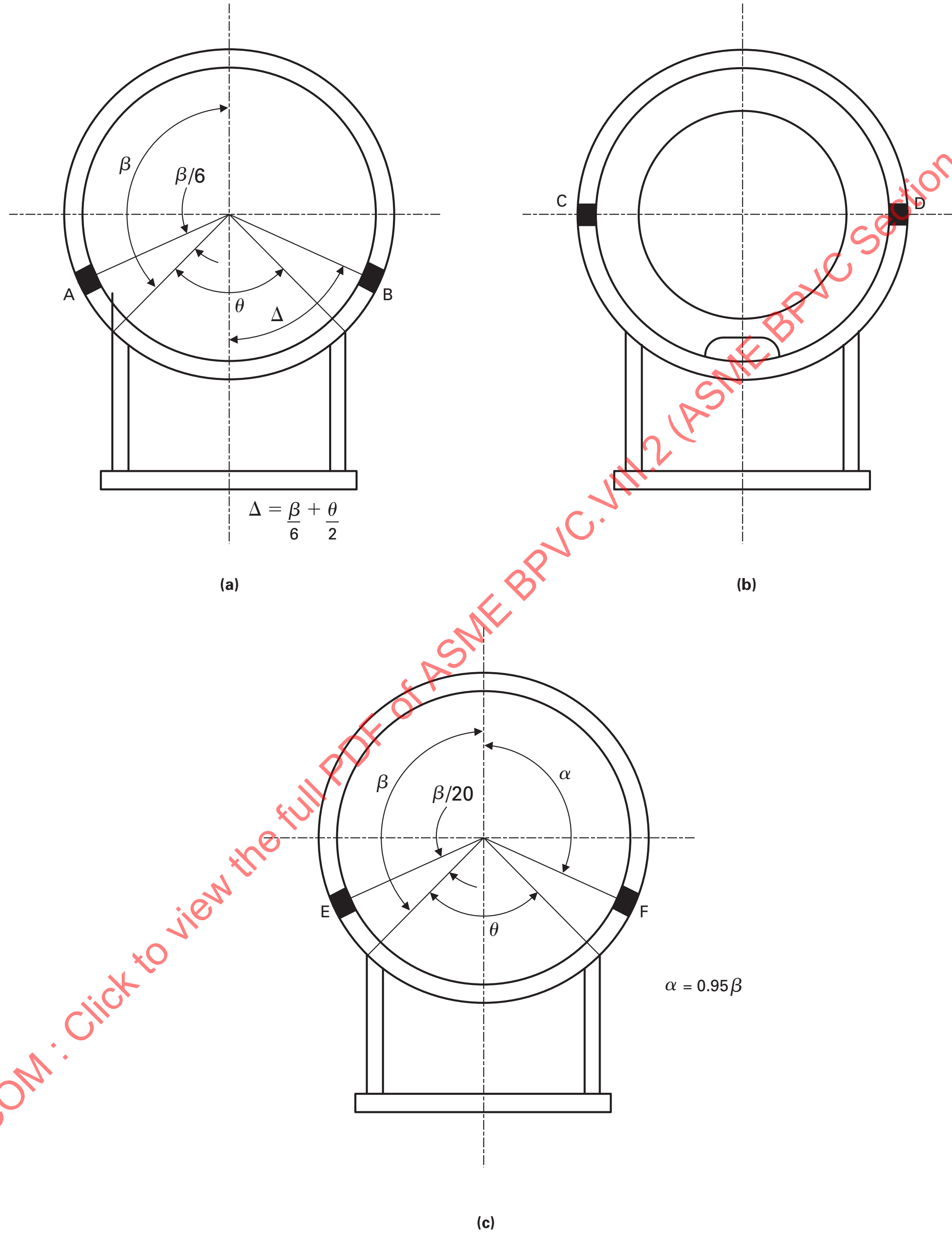
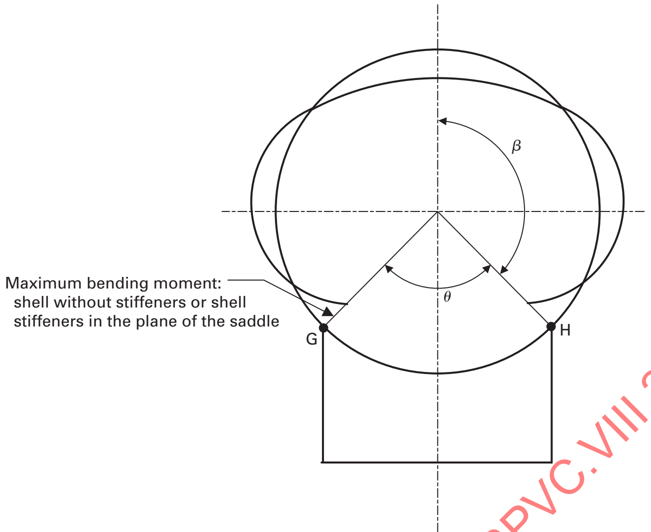
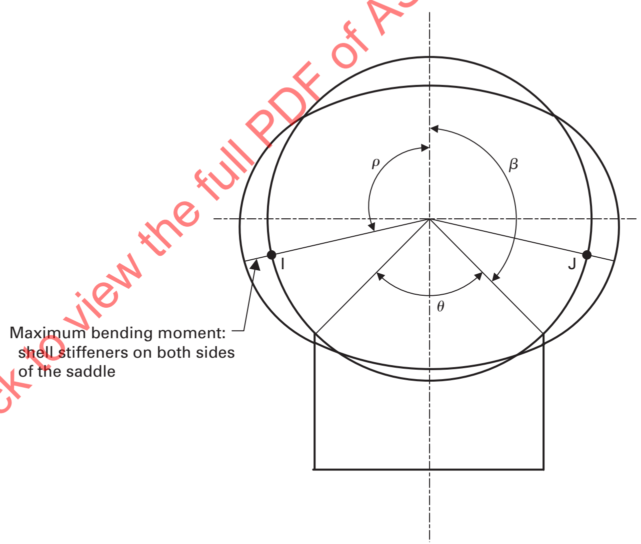


Figure 4.15.6
Locations of Maximum Circumferential Normal Stresses in the Cylinder



(a)



(b)

Figure 4.15.7
Skirt Attachment Location on Vertical Vessels

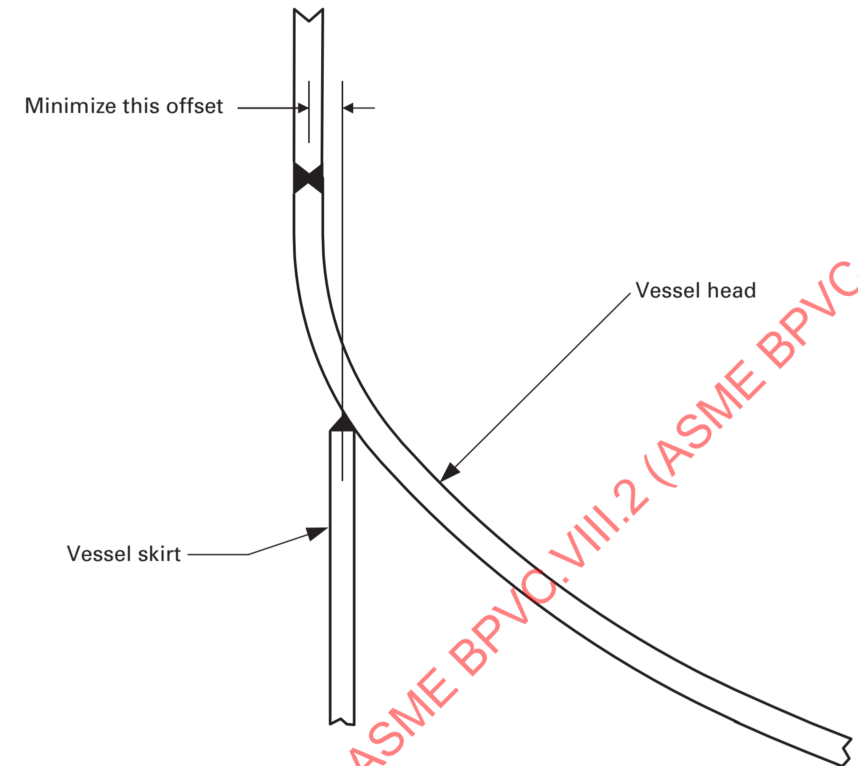
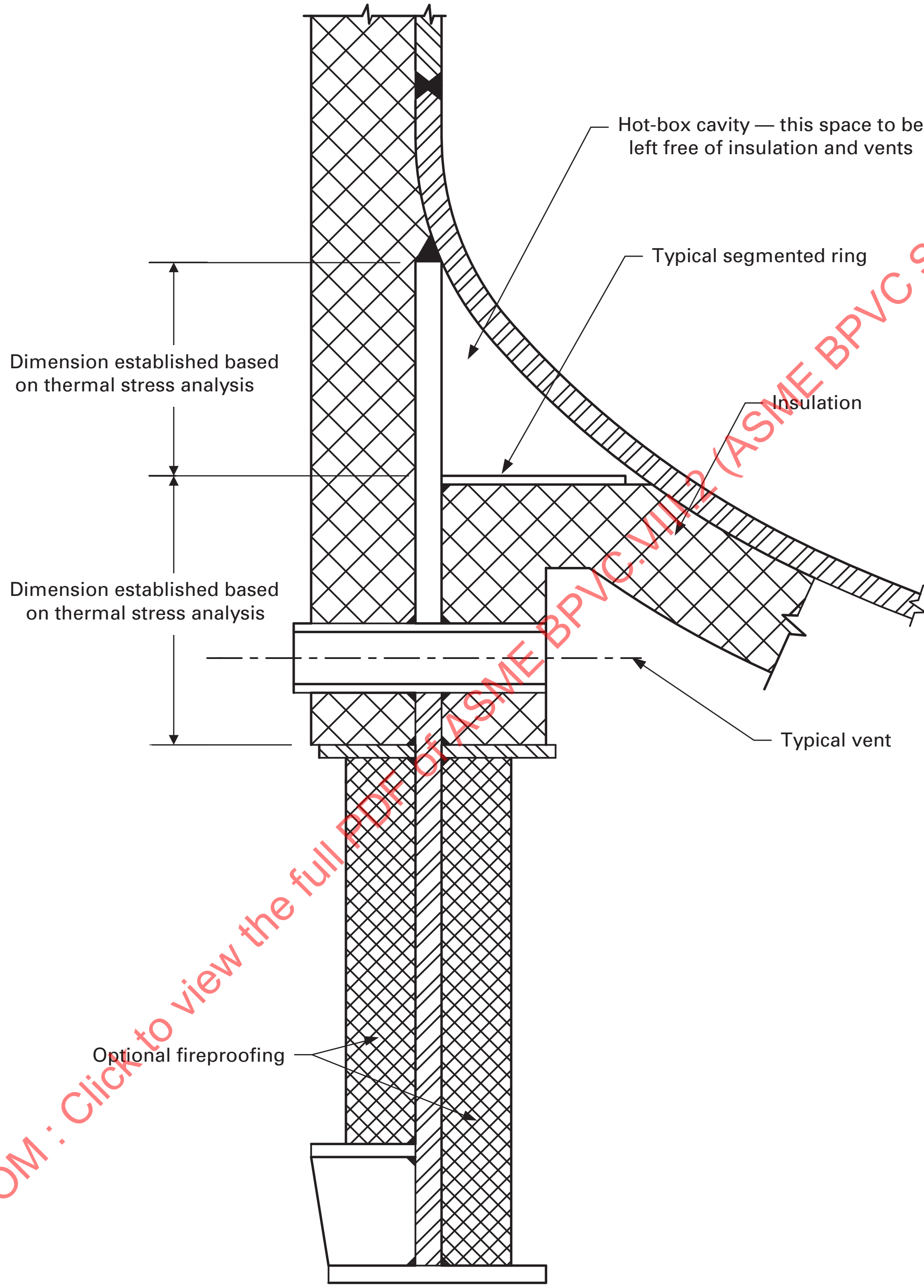


Figure 4.15.8
A Typical Hot-Box Arrangement for Skirt Supported Vertical Vessels



4.16 DESIGN RULES FOR FLANGED JOINTS

4.16.1 SCOPE

4.16.1.1 The rules in 4.16 shall be used to design circular flanges subject to internal and/or external pressure. These rules provide for hydrostatic end loads, gasket seating, and externally applied axial force and net-section bending moment.

4.16.1.2 The rules in 4.16 apply to the design of bolted flange connections with gaskets that are entirely located within the circle enclosed by the bolt holes. The rules do not cover the case where the gasket extends beyond the bolt hole circle or where metal-metal contact is made outside of the bolt circle.

4.16.1.3 It is recommended that bolted flange connections conforming to the standards listed in 4.1.11 be used for connections to external piping. These standards may be used for other bolted flange connections and dished covers within the limits of size in the standard and pressure–temperature ratings permitted in 4.1.11. The ratings in these standards are based on the hub dimensions given or on the minimum specified thickness of flanged fittings of integral construction. Flanges fabricated from rings may be used in place of the hub flanges in these standards, provided that their strength and rigidity, calculated by the rules in this paragraph, are not less than that calculated for the corresponding size of hub flange.

4.16.1.4 The rules of this paragraph should not be construed to prohibit the use of other types of flanged connections, provided they are designed in accordance with Part 5.

4.16.2 DESIGN CONSIDERATIONS

4.16.2.1 The design of a flange involves the selection of the flange type, gasket material, flange facing, bolting, hub proportions, flange width, and flange thickness. The flange dimensions shall be selected such that the stresses in the flange and the flange rigidity satisfy the acceptability criteria of this paragraph.

4.16.2.2 In the design of a bolted flange connection, calculations shall be made for the following two design conditions, and the most severe condition shall govern the design of the flanged joint.

(a) Operating Conditions. The conditions required to resist the hydrostatic end force of the design pressure and any applied external forces and moments tending to part the joint at the design temperature.

(b) Gasket Seating Condition. The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts during assembly of the joint, at atmospheric temperature and pressure.

4.16.2.3 Calculations shall be performed using dimensions of the flange in the corroded and uncorroded conditions.

4.16.2.4 In the design of flange pairs, each flange is designed for its particular design loads of pressure and gasket reactions. The bolt load used to design each flange, however, is that load common to the flange pair and equal to the larger of the bolt loads calculated for each flange individually. No additional rules are required for design of flange pairs. After the loads for the most severe condition are determined, calculations shall be made for each flange following the rules of this paragraph.

4.16.2.5 In the design of flange pairs where pass partitions with gaskets are used, the gasket loads from the partition(s) shall be included in the calculation of bolt loads. Partition gaskets may have different gasket constants than the ring gasket inside the bolt circle. In the design of flanges with noncircular gaskets or with partitions of any shape, gasket reactions from all surfaces with gaskets shall be included in calculating bolt loads.

4.16.3 FLANGE TYPES

4.16.3.1 For the purpose of computation, there are two major categories of flanges:

(a) Integral Type Flanges - This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel or pipe wall, butt welded thereto, or attached by other forms of welding such that the flange and nozzle neck, vessel or pipe wall are structurally equivalent to integral construction. Integral flanges shall be designed considering structural interaction between the flange and the nozzle neck, vessel, or pipe wall, which the rules account for by considering the neck or wall to act as a hub. Integral type flanges are referenced below. The design flange and bolt loads are shown in Figures 4.16.1 and 4.16.2.

(1) Integral type flanges - Figure 4.16.1, sketch (a) and Table 4.2.9, Details 9 and 10

(2) Integral type flanges where $g_1 = g_o$ - Figure 4.16.1, sketch (b)

(3) Integral type flanges with a hub - Figure 4.16.2 and Table 4.2.9, Details 6, 7, and 8

(4) Integral type flanges with nut stops - Figure 4.16.3 and Figure 4.16.4

(b) Loose Type Flanges - This type covers those designs in which the flange has no substantial integral connection to the nozzle neck, vessel, or pipe wall, and includes welded flange connections where the welds are not considered to give the mechanical strength equivalent of an integral attachment. Loose type flanges are referenced below. The design flange and bolt loads are shown in Figures 4.16.5 and 4.16.6.

(1) Loose type flanges - Figure 4.16.5 and Table 4.2.9, Details 1,2,3 and 4

(2) Loose type lap joint flanges - Figure 4.16.6 and Table 4.2.9, Detail 5

4.16.3.2 The integral and loose type flanges described above can also be applied to reverse flange configurations. Integral and loose type reverse flanges are shown in Figure 4.16.7.

4.16.4 FLANGE MATERIALS

4.16.4.1 Materials used in the construction of bolted flange connections, excluding gasket materials, shall comply with the requirements given in Part 3.

4.16.4.2 Flanges made from ferritic steel shall be given a normalizing or full-annealing heat treatment when the thickness of the flange, t (see Figures 4.16.1 through 4.16.7), exceeds 75 mm (3 in.).

(**21**) **4.16.4.3** Flanges with hubs that are machined from plate, bar stock, or billet shall be in accordance with the following:

(a) Flanges with hubs shall not be machined from plate or bar (except as permitted in 3.2.5.2) material unless the material has been formed into a ring and the following additional conditions are met:

(1) In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange.

(2) The joints in the ring are welded butt joints that conform to the requirements of Part 6. The thickness to be used to determine postweld heat treatment and radiographic requirements shall be $\min[t, (A - B)/2]$.

(3) The back of the flange and outer surface of the hub shall be examined by either the magnetic particle method or the liquid penetrant method in accordance with Part 7.

4.16.4.4 Bolts, studs, nuts, and washers shall comply with the requirements of Part 3 and referenced standards. It is recommended that bolts and studs have a nominal diameter of not less than 12 mm (0.5 in.). If bolts or studs smaller than 12 mm (0.5 in.) are used, then ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid overstressing small-diameter bolts. When washers are used, they shall be through hardened to minimize the potential for galling.

4.16.5 GASKET MATERIALS

4.16.5.1 The gasket constants for the design of the bolt load (m and y), are provided in Table 4.16.1. Other values for the gasket constants may be used if based on actual testing or data in the literature, as agreed upon between designer and the user.

4.16.5.2 The minimum width of sheet and composite gaskets, N, is recommended to be no less than that given in Table 4.16.2.

NOTE: Gasket materials should be selected that are suitable for the design conditions. Corrosion, chemical attack, creep and thermal degradation of gasket materials over time should be considered.

4.16.6 DESIGN BOLT LOADS

4.16.6.1 The procedure to determine the bolt loads for the operating and gasket seating conditions is shown below.

Step 1. Determine the design pressure and temperature of the flange joint.

Step 2. Select a gasket and determine the gasket factors m and y from Table 4.16.1, or other sources. The selected gasket width should comply with the guidelines detailed in Table 4.16.2.

Step 3. Determine the width of the gasket, N, basic gasket seating width, b_0, the effective gasket seating width, b, and the location of the gasket reaction, G, based on the flange and gasket geometry, the information in Table 4.16.3 and Figure 4.16.8, and the equations shown below. Note that for lap joint flanges, G is equal to the midpoint of contact between the flange and the lap, see Figure 4.16.6 and Figure 4.16.8.

(a) For $b_0 \le 6$ mm (0.25 in.), G is the mean diameter of the gasket contact face and

$$b = b_0 \tag{4.16.1}$$

(b) For $b_0 > 6$ mm (0.25 in.)

$$b = 0.5C_{ul}\sqrt{\frac{b_0}{C_{ul}}} \tag{4.16.2}$$

$$G = G_c - 2b \tag{4.16.3}$$

Step 4. Determine the design bolt load for the operating condition.

$$W_o = 0.785G^2P + 2b\pi GmP \qquad \text{for non-self-energized gaskets} \tag{4.16.4}$$

$$W_o = 0.785G^2P \qquad \text{for self-energized gaskets} \tag{4.16.5}$$

Step 5. Determine the design bolt load for the gasket seating condition.

$$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg} \tag{4.16.6}$$

The parameter A_b is the actual total cross-sectional area of the bolts that is selected such that $A_b \geq A_m$, where:

$$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right] \tag{4.16.7}$$

$$W_{gs} = \pi bGy \qquad \text{for non-self-energized gaskets} \tag{4.16.8}$$

$$W_{gs} = 0.0 \qquad \text{for self-energized gaskets} \tag{4.16.9}$$

NOTE: Where significant axial force is required to compress the gasket during assembly of a joint containing a self-energizing gasket, the value of W_{gs} shall be taken as equal to that axial force. In addition, some self-energizing gaskets generate axial load due to their wedging action and this load shall be considered in setting the value of W_{gs}.

4.16.7 FLANGE DESIGN PROCEDURE

4.16.7.1 The procedure in this paragraph can be used to design circular integral, loose or reverse flanges, subject to internal or external pressure, and external loadings. The procedure incorporates both a strength check and a rigidity check for flange rotation.

4.16.7.2 The procedure to design a flange is shown below.

Step 1. Determine the design pressure and temperature of the flange joint, and the external net-section axial force, F_A, and bending moment, M_E. If the pressure is negative, the absolute value of the pressure should be used in this procedure.

Step 2. Determine the design bolt loads for operating condition, W_o, and the gasket seating condition, W_g, and corresponding actual bolt area, A_b, from 4.16.6.

Step 3. Determine an initial flange geometry, in addition to the information required to determine the bolt load, the following geometric parameters are required:

(a) The flange bore, B
(b) The bolt circle diameter, C
(c) The outside diameter of the flange, A
(d) The flange thickness, t
(e) The thickness of the hub at the large end, g_1
(f) The thickness of the hub at the small end, g_0
(g) The hub length, h

Step 4. Determine the flange stress factors using the equations in Tables 4.16.4 and 4.16.5.

Step 5. Determine the flange forces.

$$H_D = 0.785B^2P \tag{4.16.10}$$

$$H = 0.785G^2P \tag{4.16.11}$$

$$H_T = H - H_D \tag{4.16.12}$$

$$H_G = W_o - H \tag{4.16.13}$$

Step 6. Determine the flange moment for the operating condition using Equation (4.16.14) or Equation (4.16.15), as applicable. When specified by the user or his designated agent, the maximum bolt spacing (B_{smax}) and the bolt spacing correction factor (B_{sc}) shall be applied in calculating the flange moment for internal pressure using the equations in Table 4.16.11. The flange moment M_o for the operating condition and flange moment M_g for the gasket seating condition without correction for bolt spacing B_{sc} = 1 is used for the calculation of the rigidity index in Step 10. In these equations, h_D, h_T, and h_G are determined from Table 4.16.6. For integral and loose type flanges, the moment M_{oe} is calculated using Equation (4.16.16) where I and I_p in this equation are determined from Table 4.16.7. For reverse type flanges, the procedure to determine M_{oe} shall be agreed upon between the Designer and the Owner.

$$M_o = abs\left[\left(\left(H_D h_D + H_T h_T + H_G h_G\right)B_{sc} + M_{oe}\right)F_s\right] \quad \text{for internal pressure} \tag{4.16.14}$$

$$M_o = abs\left[\left(H_D\left(h_D - h_G\right) + H_T\left(h_T - h_G\right) + M_{oe}\right)F_s\right] \quad \text{for external pressure} \tag{4.16.15}$$

$$M_{oe} = 4M_E\left[\frac{I}{0.3846 I_p + I}\right]\left[\frac{h_D}{\left(C - 2h_D\right)}\right] + F_A h_D \tag{4.16.16}$$

Step 7. Determine the flange moment for the gasket seating condition using eq. (4.16.17) or (4.16.18), as applicable.

$$M_g = \frac{W_g\left(C - G\right)B_{sc}F_s}{2} \quad \text{for internal pressure} \tag{4.16.17}$$

$$M_g = W_g h_G F_s \quad \text{for external pressure} \tag{4.16.18}$$

Step 8. Determine the flange stresses for the operating and gasket seating conditions using the equations in Table 4.16.8.

Step 9. Check the flange stress acceptance criteria. The two criteria shown below shall be evaluated. If the stress criteria are satisfied, go to Step 10. If the stress criteria are not satisfied, re-proportion the flange dimensions and go to Step 4.

(a) Allowable Normal Stress - The criteria to evaluate the normal stresses for the operating and gasket seating conditions are shown in Table 4.16.9.

(b) Allowable Shear Stresses - In the case of loose type flanges with lap, as shown in Figure 4.16.6 where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed $0.8S_{no}$ or $0.8S_{ng}$, as applicable, for the material of the lap. In the case of welded flanges where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed $0.8S_{no}$ or $0.8S_{ng}$, as applicable. The shearing stress shall be calculated for both the operating and gasket seating load cases. Similar situations where flange parts are subjected to shearing stresses shall be checked using the same requirement.

Step 10. Check the flange rigidity criterion in Table 4.16.10. If the flange rigidity criterion is satisfied, then the design is complete. If the flange rigidity criterion is not satisfied, then re-proportion the flange dimensions and go to Step 3. The flange moment M_o for the operating condition (Step 6) and flange moment M_g for the gasket seating condition (Step 7) without correction for bolt spacing B_{sc} = 1 is used for the calculation of the rigidity index.

4.16.8 SPLIT LOOSE TYPE FLANGES

Loose flanges split across a diameter and designed under the rules given in this paragraph may be used under the following provisions.

(a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment, F_s = 2.0.

(b) When the flange consists of two split rings, each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment, F_s = 0.75. The pair of rings shall be assembled so that the splits in one ring are 90 deg from the splits in the other ring.

(c) The flange split locations should preferably be midway between bolt holes.

(d) It is not a requirement that the flange rigidity rules of 4.6.10 be applied to split loose flanges.

4.16.9 NONCIRCULAR SHAPED FLANGES WITH A CIRCULAR BORE

The outside diameter, A, for a noncircular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. The bolt loads, flange moments, and stresses shall be calculated in the same manner as for a circular flange using a bolt circle whose size is established by drawing a circle through the centers of the outermost bolts.

4.16.10 FLANGES WITH NUT STOPS

When flanges are designed per 4.16, or are fabricated to the dimensions of ASME B16.5 or other acceptable standards, except that the dimension $0.5(C - B) - g_1$ is decreased to provide a nut-stop, the fillet radius shall be as shown in Figures 4.16.3 and 4.16.4 except that:

(a) For flanges designed to this paragraph, the minimum thickness of the hub at the large end, g_1, shall be the smaller of $2t_n$ or $4r_u$, but not less than 12 mm (0.5 in.).

(b) For ASME B16.5 or other standard flanges, the thickness of the hub at the small end, g_0, shall be increased as necessary to provide a nut-stop.

4.16.11 JOINT ASSEMBLY PROCEDURES

Bolted joints should be assembled and bolted-up in accordance with a written procedure that has been demonstrated to be acceptable for similar joint configurations in similar services. Further guidance can be found in ASME PCC-1, Guidelines for Pressure Boundary Bolted Flange Joint Assembly.

4.16.12 EVALUATION OF EXTERNAL FORCES AND MOMENTS FOR FLANGED JOINTS WITH STANDARD FLANGES

External loads (forces and bending moments) may be evaluated for flanged joints with welding neck flanges chosen in accordance with 4.1.11.1(a), 4.1.11.1(g), and 4.1.11.3, using the following requirements:

(a) The vessel design pressure (corrected for the static head from liquid or bulk material acting on the flange) at the design temperature cannot exceed the pressure–temperature rating of the flange.

(b) The actual assembly bolt load (see 4.16.11) shall comply with ASME PCC-1, Nonmandatory Appendix O.

(c) The bolt material shall have an allowable stress equal to or greater than SA-193 B8 Cl. 2 at the specified bolt size and temperature.

(d) The combination of vessel design pressure, P (corrected for the static head from liquid or bulk material acting on the flange), with external moment and external axial force shall satisfy eq. (4.16.19). (The units of the variables in this equation shall be consistent with the pressure rating.)

$$16M_E + 4F_A G \leq \pi G^3\left[\left(P_R - P\right) + F_M P_R\right] \tag{4.16.19}$$

4.16.13 NOMENCLATURE (21)

A = outside diameter of the flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots.
A_b = cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion.
A_m = total minimum required cross-sectional area of the bolts.
a = nominal bolt diameter
B = inside diameter of the flange. When $B < 20g_1$, B_1 may be used for B in the equation for the longitudinal stress.
B_1 = $B + g_1$ for loose type flanges and for integral type flanges that have a value of f less than 1.0, (although a minimum value of $f = 1.0$ is permitted). B_1 is equal to $B + g_0$ for integral type flanges when $f \geq 1.0$.
B^* = inside diameter of the reverse flange.
B_s = bolt spacing, The bolt spacing may be taken as the bolt circle circumference divided by the number of bolts or as the chord length between adjacent bolt locations.
B_{smax} = maximum bolt spacing
B_{sc} = bolt spacing correction factor
b = effective gasket contact width.
b_0 = basic gasket seating width.
C = bolt circle diameter.
C_{ul} = conversion factor for length, $C_{ul} = 1.0$ for U.S. Customary Units and $C_{ul} = 25.4$ for Metric Units.

d = flange stress factor.

d_r = flange stress factor d for a reverse type flange.

E_{yg} = Modulus of Elasticity at the gasket seating load case temperature.

E_{yo} = Modulus of Elasticity at the operating load case temperature.

e = flange stress factor.

e_r = flange stress factor e for a reverse type flange.

F = flange stress factor for integral type flanges.

F_A = value of the external tensile net-section axial force. Compressive net-section forces are to be neglected and for that case, FA should be taken as equal to zero.

F_L = flange stress factor for loose type flanges.

F_M = moment factor, in accordance with Table 4.16.12.

F_s = moment factor used to design split rings (see 4.16.8), F_s = 1.0 for non-split rings.

f = hub stress correction factor for integral flanges.

G = diameter at the location of the gasket load reaction (see Figure 4.16.8).

G_{avg} = average of the hub thicknesses g_1 and g_0.

G_c = outside diameter of the gasket contact area (see Figure 4.16.8).

g_0 = thickness of the hub at the small end.

(a) for integral type flanges per Figure 4.16.1(a), $g_0 = t_n$

(b) for other integral type flanges, g_0 = the minimum of t_n or the thickness of the hub at the small end

g_1 = thickness of the hub at the large end.

H = total hydrostatic end force.

H_D = total hydrostatic end force on the area inside of the flange.

H_G = gasket load for the operating condition.

H_T = difference between the total hydrostatic end force and hydrostatic end force on the area inside the flange.

h = hub length.

h_n = the axial length of the cylinder to which the flange ring is attached. This is measured from the back face of flange to the other end of the cylinder. This is used when $g_0 = g_1$.

h_o = hub length parameter.

h_{or} = hub length parameter for a reverse flange.

h_p = effective hub length used to determine I_p.

h_D = moment arm for load H_D.

h_G = moment arm for load H_G.

h_T = moment arm for load H_T.

I = bending moment of inertia of the flange cross-section.

I_p = polar moment of inertia of the flange cross-section.

J = flange rigidity index.

K = ratio of the flange outside diameter to the flange inside diameter.

K_R = rigidity index factor.

L = flange stress factor.

L_r = flange stress factor L for a reverse type flange.

M_E = absolute value of the external net-section bending moment.

M_g = flange design moment for the gasket seating condition.

M_o = flange design moment for the operating condition.

M_{oe} = component of the flange design moment resulting from a net section bending moment and/or axial force.

m = factor for the gasket operating condition.

N = gasket contact width, N = 0.0 for self-energizing gaskets.

P = design pressure.

P_R = flange pressure rating at design temperature.

r_1 = radius to be at least $0.25g_1$ but not less than 5 mm (0.1875 in.).

r_u = radius of the undercut on a flange with nut stops.

S_{bg} = allowable stress from Annex 3-A for the bolt evaluated at the gasket seating temperature.

S_{bo} = allowable stress from Annex 3-A for the bolt evaluated at the design temperature.

S_{fg} = allowable stress from Annex 3-A for the flange evaluated at the gasket seating temperature.

S_{fo} = allowable stress from Annex 3-A for the flange evaluated at the design temperature.

S_{ng} = allowable stress from Annex 3-A for the nozzle neck, vessel, or pipe evaluated at the gasket seating temperature.

S_{no} = allowable stress from Annex 3-A for the nozzle neck, vessel, or pipe evaluated at the design temperature.

S_H = flange hub stress.
S_R = flange radial stress.
S_T = flange tangential stress.
S_{T1} = flange tangential stress at the outside diameter of a reverse flange.
S_{T2} = flange tangential stress at the inside diameter of a reverse flange.
T = flange stress factor.
T_r = flange stress factor T for a reverse flange.
t = flange thickness, including the facing thickness or the groove depth if either do not exceed 2 mm (0.0625 in.); otherwise, the facing thickness or groove depth is not included in the overall flange thickness.
t_n = nominal thickness of the shell, pipe, or nozzle to which the flange is attached.
t_x = is $2g_o$ when the design is calculated as an integral flange, or two times the minimum required thickness of the shell or nozzle wall when the design is based on a loose flange, but not less than 6 mm (0.25 in.).
U = flange stress factor.
U_r = flange stress factor U for a reverse type flange.
V = flange stress factor for integral type flanges.
V_L = flange stress factor for loose type flanges.
W_g = design bolt load for the gasket seating condition.
W_o = design bolt load for the operating condition.
w = width of the nubbin.
y = factor for the gasket seating condition
Y = flange stress factor.
Y_r = flange stress factor Y for a reverse type flange.
Z = flange stress factor.

4.16.14 TABLES

Table 4.16.1
Gasket Factors for Determining the Bolt Loads

Gasket Material	Gasket Factor, m	Min. Design Seating Stress, y, MPa (psi)	Column in Table 4.16.3	Facing Sketch in Table 4.16.3
Self-energizing types (O rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0 in. 0	...	...
Elastomers without fabric or high percent of mineral fiber:				
below 75 A Shore Durometer	0.50	0		
75 A or higher Shore Durometer	1.00	0 1.4 (200)	II	(1a), (1b), (1c), (1d), (4), (5)
Mineral fiber with suitable binder for operating conditions:				
3.2 mm ($^1/_8$ in.) thick	2.00	11 (1,600)		
1.6 mm ($^1/_{16}$ in.) thick	2.75	26 (3,700)		
0.8 mm ($^1/_{32}$ in.) thick	3.50	45 (6,500)	II	(1), (1b), (1c), (1d), (4), (5)
Elastomers with cotton fabric insertion	1.25	2.8 (400)	II	(1a), (1b), (1c), (1d), (4), (5)
Elastomers with mineral fiber insertion (with or without wire reinforcement):				
3-ply	2.25	15 (2,200)		
2-ply	2.50	20 (2,900)		
1-ply	2.75	26 (3,700)	II	(1), (1b), (1c), (1d), (5)
Vegetable fiber	1.75	7.6 (1,100)	II	(1a), (1b), (1c), (1d), (4), (5)
Spiral-wound metal, mineral fiber filler:				
Carbon steel	2.50	69 (10,000)		
Stainless steel, Monel, and nickel-base alloy	3.00	69 (10,000)	II	(1a), (1b)

Table 4.16.1
Gasket Factors for Determining the Bolt Loads (Cont'd)

Gasket Material	Gasket Factor, m	Min. Design Seating Stress, y, MPa (psi)	Column in Table 4.16.3	Facing Sketch in Table 4.16.3
Corrugated metal, mineral fiber inserted, or corrugated metal, jacketed mineral fiber filled:				
Soft aluminum	2.50	20 (2,900)		
Soft copper or brass	2.75	26 (3,700)		
Iron or soft steel	3.00	31 (4,500)		
Monel or 4%–6% chrome	3.25	38 (5,500)		
Stainless steels and nickel-base alloys	3.50	45 (6,500)	II	(1a), (1b)
Corrugated metal:				
Soft aluminum	2.75	26 (3,700)		
Soft copper or brass	3.00	31 (4,500)		
Iron or soft steel	3.25	38 (5,500)		
Monel or 4%–6% chrome	3.50	45 (6,500)		(1a), (1b), (1c),
Stainless steels and nickel-base alloys	3.75	52 (7,600)	II	(1d)
Flat metal, jacketed mineral fiber filled:				
Soft aluminum	3.25	38 (5,500)		
Soft copper or brass	3.50	45 (6,500)		
Iron or soft steel	3.75	52 (7,600)		
Monel	3.50	55 (8,000)		
4%–6% chrome	3.75	62 (9,000)		(1a), (1b), (1c),
Stainless steels and nickel-base alloys	3.75	62 (9,000)	II	(1d), (2)
Grooved metal:				
Soft aluminum	3.25	38 (5,500)		
Soft copper or brass	3.50	45 (6,500)		
Iron or soft steel	3.75	52 (7,600)		
Monel or 4%–6% chrome	3.75	62 (9,000)		(1a), (1b), (1c),
Stainless steels and nickel-base alloys	4.25	70 (10,100)	II	(1d), (2), (3)
Sold flat metal:				
Soft aluminum	4.00	61 (8,800)		
Soft copper or brass	4.75	90 (13,000)		
Iron or soft steel	5.50	124 (18,000)		(1a), (1b), (1c),
Monel or 4%–6% chrome	6.00	150 (21,800)		(1d), (2), (3),
Stainless steels and nickel-base alloys	6.50	180 (26,000)	I	(4), (5)
Ring joint:				
Iron or soft steel	5.50	124 (18,000)		
Monel or 4%–6% chrome	6.00	150 (21,800)		
Stainless steel and nickel-base alloys	6.50	180 (26,000)	I	(6)

GENERAL NOTE: This table gives a list of commonly used gasket materials and contact facings with suggested values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b. The design values and other details given in this table are suggested only and are not mandatory.

Table 4.16.2
Recommended Minimum Gasket Contact Width

Gasket Type	Gasket Contact Width, N — Gasket Outside Diameter: <150 mm (6 in.)	<300 mm (12 in.)	<600 mm (24 in.)	<900 mm (36 in.)	900 mm (36 in.) and Over
Sheet gaskets including laminated sheets gaskets with or without a metal core	9 mm (0.375 in.)	12 mm (0.5 in.)	16 mm (0.625 in.)	16 mm (0.625 in.)	19 mm (0.75 in.)
Preformed composite gaskets including spiral wound, jacketed, and solid flat metal gaskets	6 mm (0.25 in.)	9 mm (0.375 in.)	12 mm (0.5 in.)	16 mm (0.625 in.)	16 mm (0.625 in.)

Table 4.16.3
Effective Gasket Width for Determining the Bolt Loads

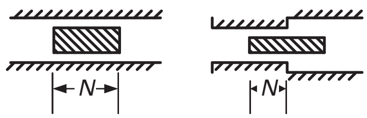
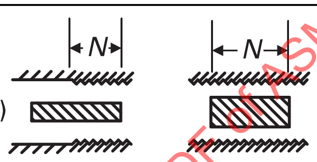
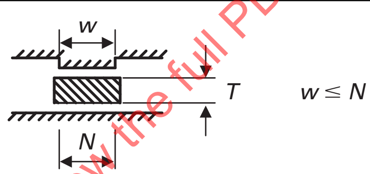
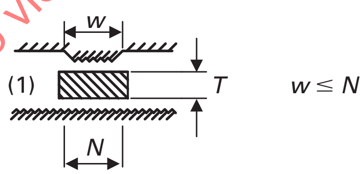
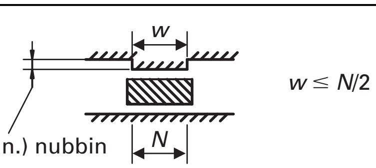
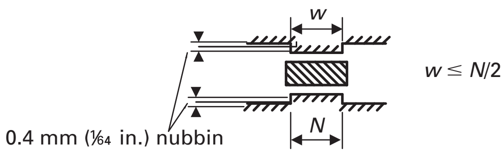
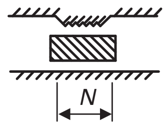
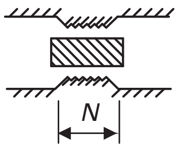
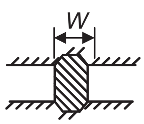
Facing Sketch	Facing Sketch Detail (Exaggerated)	Basic Gasket Seating Width, b_o — Column I	Column II
1a		$\frac{N}{2}$	$\frac{N}{2}$
1b	Note (1) 		
1c		$\min\left[\frac{w+T}{2}, \frac{w+N}{4}\right]$	$\min\left[\frac{w+T}{2}, \frac{w+N}{4}\right]$
1d	Note (1) 		
2		$\frac{w+N}{4}$	$\frac{w+3N}{8}$

Table 4.16.3
Effective Gasket Width for Determining the Bolt Loads (Cont'd)

Facing Sketch	Facing Sketch Detail (Exaggerated)	Basic Gasket Seating Width, b_o	
		Column I	Column II
3		$\frac{N}{4}$	$\frac{3N}{8}$
4	Note (1) 	$\frac{3N}{8}$	$\frac{7N}{16}$
5	Note (1) 	$\frac{N}{4}$	$\frac{3N}{8}$
6		$\frac{w}{8}$	– – –

NOTES:
(1) Where serrations do not exceed 0.4 mm (0.0156 in.) depth and 0.8 mm (0.0313 in.) width spacing, sketches (1b) and (1d) shall be used.
(2) The gasket factors listed in this table only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

Table 4.16.4
Flange Stress Factors Equations Involving Diameter

(21)

Flange Type	Stress Factors Involving Diameter
Integral-type flange and loose-type flange with a hub	$K = \frac{A}{B}$ $Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2 \log_{10} K}{K^2 - 1}\right)\right]$ $T = \frac{K^2\left(1 + 8.55246 \log_{10} K\right) - 1}{\left(1.04720 + 1.9448K^2\right)(K-1)}$ $U = \frac{K^2\left(1 + 8.55246 \log_{10} K\right) - 1}{1.36136\left(K^2 - 1\right)(K-1)}$ $Z = \frac{\left(K^2 + 1\right)}{\left(K^2 - 1\right)}$ $L = \frac{te + 1}{T} + \frac{t^3}{d}$ $e = \frac{F}{h_o}$ for integral-type flanges $e = \frac{F_L}{h_o}$ for loose-type flanges with a hub $d = \frac{U g_o^2 h_o}{V}$ for integral-type flanges $d = \frac{U g_o^2 h_o}{V_L}$ for loose-type flanges with a hub $h_o = \sqrt{B g_0}$ $X_g = \frac{g_1}{g_0}$ $X_h = \frac{h}{h_o}$ for $g_1 > g_0$ $X_h = 2.0$ for $g_1 = g_0$

Table 4.16.4
Flange Stress Factors Equations Involving Diameter (Cont'd)

Flange Type	Stress Factors Involving Diameter
Reverse integral-type flange and reverse loose-type flanges with a hub	The parameters K, T, U, Y, and Z are determined using the equations for integral- and loose-type flanges with: $$K = \frac{A}{B^*}$$ Then, the reverse flange parameters are computed as follows: $$Y_r = \alpha_r Y$$ $$T_r = \frac{(Z + 0.3)}{(Z - 0.3)}\alpha_r T$$ $$U_r = \alpha_r U$$ $$L_r = \frac{(te_r + 1)}{T_r} + \frac{t^3}{d_r}$$ $$\alpha_r = \frac{1}{K^2}\left[1 + \frac{0.668(K+1)}{Y}\right]$$ $e_r = \frac{F}{h_{or}}$ for integral-type flanges $e_r = \frac{F_L}{h_{or}}$ for loose-type flanges with a hub $d_r = \frac{U_r g_o^2 h_{or}}{V}$ for integral-type flanges $d_r = \frac{U_r g_o^2 h_{or}}{V_L}$ for loose-type flanges with a hub $$h_{or} = \sqrt{A g_0}$$ $$X_g = \frac{g_1}{g_0}$$ $$X_h = \frac{h}{h_{or}} \text{ for } g_1 > g_0$$ $$X_h = 2.0 \text{ for } g_1 = g_0$$

Table 4.16.5
Flange Stress Factor Equations

Flange Type	Stress Factors
Integral-type flange, reverse integral-type flange	$F = \left(\begin{array}{c} 0.897697 - 0.297012 lnX_g + 9.5257\left(10^{-3}\right) lnX_h + \\ 0.123586\left(lnX_g\right)^2 + 0.0358580\left(lnX_h\right)^2 - 0.194422\left(lnX_g\right)\left(lnX_h\right) - \\ 0.0181259\left(lnX_g\right)^3 + 0.0129360\left(lnX_h\right)^3 - \\ 0.0377693\left(lnX_g\right)\left(lnX_h\right)^2 + 0.0273791\left(lnX_g\right)^2\left(lnX_h\right) \end{array} \right)$ For $0.1 \le X_h \le 0.5$ $V = \left(\begin{array}{c} 0.500244 + \frac{0.227914}{X_g} - 1.87071X_h - \frac{0.344410}{X_g^2} + 2.49189X_h^2 + \\ 0.873446\left(\frac{X_h}{X_g}\right) + \frac{0.189953}{X_g^3} - 1.06082X_h^3 - 1.49970\left(\frac{X_h^2}{X_g}\right) + \\ 0.719413\left(\frac{X_h}{X_g^2}\right) \end{array} \right)$ For $0.5 < X_h \le 2.0$ $V = \left(\begin{array}{c} 0.0144868 - \frac{0.135977}{X_g} - \frac{0.0461919}{X_h} + \frac{0.560718}{X_g^2} + \frac{0.0529829}{X_h^2} + \\ \frac{0.244313}{X_gX_h} + \frac{0.113929}{X_g^3} - \frac{0.00928265}{X_h^3} - \frac{0.0266293}{X_gX_h^2} - \frac{0.217008}{X_g^2X_h} \end{array} \right)$ $f = \max\left[1.0, \left(\frac{\begin{array}{c} 0.0927779 - 0.0336633X_g + 0.964176X_g^2 + \\ 0.0566286X_h + 0.347074X_h^2 - 4.18699X_h^3 \end{array}}{\begin{array}{c} 1 - 5.96093\left(10^{-3}\right)X_g + 1.62904X_h + \\ 3.49329X_h^2 + 1.39052X_h^3 \end{array}} \right)\right]$

Table 4.16.5
Flange Stress Factor Equations (Cont'd)

Flange Type	Stress Factors
Loose-type flange with a hub, reverse loose-type flange with a hub	$F_L = \left(\dfrac{0.941074 + 0.176139\left(\ln X_g\right) - 0.188556\left(\ln X_h\right) + 0.0689847\left(\ln X_g\right)^2 + 0.523798\left(\ln X_h\right)^2 - 0.513894\left(\ln X_g\right)\left(\ln X_h\right)}{1 + 0.379392\left(\ln X_g\right) + 0.184520\left(\ln X_h\right) - 0.00605208\left(\ln X_g\right)^2 - 0.00358934\left(\ln X_h\right)^2 + 0.110179\left(\ln X_g\right)\left(\ln X_h\right)} \right)$ For $0.1 \le X_h \le 0.25$ $\ln\left[V_L\right] = \left(6.57683 - 0.115516 X_g + 1.39499\sqrt{X_g}\left(\ln X_g\right) + 0.307340\left(\ln X_g\right)^2 - 8.30849\sqrt{X_g} + 2.62307\left(\ln X_g\right) + 0.239498 X_h\left(\ln X_h\right) - 2.96125\left(\ln X_h\right) + \dfrac{7.035052\left(10^{-4}\right)}{X_h} \right)$ For $0.25 < X_h \le 0.50$ $V_L = \left(1.56323 - 1.80696\left(\ln X_g\right) - \dfrac{1.33458}{X_h} + 0.276415\left(\ln X_g\right)^2 + \dfrac{0.417135}{X_h^2} + \dfrac{1.39511\left(\ln X_g\right)}{X_h} + 0.0137129\left(\ln X_g\right)^3 + \dfrac{0.0943597}{X_h^3} - \dfrac{0.402096\left(\ln X_g\right)}{X_h^2} - \dfrac{0.101619\left(\ln X_g\right)^2}{X_h} \right)$ For $0.50 < X_h \le 1.0$ $V_L = \left(-0.0213643 - \dfrac{0.0763597}{X_g} + \dfrac{0.1029900}{X_h} + \dfrac{0.725776}{X_g^2} - \dfrac{0.160603}{X_h^2} - \dfrac{0.0918061}{X_g \cdot X_h} + \dfrac{0.472277}{X_g^3} + \dfrac{0.0873530}{X_h^3} + \dfrac{0.527487}{X_g \cdot X_h^2} - \dfrac{0.980209}{X_g^2 \cdot X_h} \right)$ For $1.0 < X_h \le 2.0$ $V_L = \left(7.96687\left(10^{-3}\right) - \dfrac{0.220518}{X_g} + \dfrac{0.0602652}{X_h} + \dfrac{0.619818}{X_g^2} - \dfrac{0.223212}{X_h^2} + \dfrac{0.421920}{X_g \cdot X_h} + \dfrac{0.0950195}{X_g^3} + \dfrac{0.209813}{X_h^3} - \dfrac{0.158821}{X_g \cdot X_h^2} - \dfrac{0.242056}{X_g^2 \cdot X_h} \right)$ $f = 1.0$

Table 4.16.6
Moment Arms for Flange Loads for the Operating Condition

Flange Type	h_D	h_T	h_G
Integral-type flanges	$\frac{C - B - g_1}{2}$	$\frac{1}{2}\left[\frac{C - B}{2} + h_G\right]$	$\frac{C - G}{2}$
Loose-type flanges, except lap joint flanges	$\frac{C - B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{C - G}{2}$
Loose-type lap joint flanges	$\frac{C - B}{2}$	$\frac{C - G}{2}$	$\frac{C - G}{2}$
Reverse integral-type flanges	$\frac{C + g_1 - 2g_o - B}{2}$	$\frac{1}{2}\left(C - \frac{B + G}{2}\right)$	$\frac{C - G}{2}$
Reverse loose-type flanges	$\frac{C - B}{2}$	$\frac{1}{2}\left(C - \frac{B + G}{2}\right)$	$\frac{C - G}{2}$

Table 4.16.7
Flange Moments of Inertia (21)

Flange Type	I	I_p
Integral-type flange	$I = \frac{0.0874 L g_o^2 h_o B}{V}$	$I_p = \max\left\{K_{AB} + K_{CD}, A_R t^3\left[\frac{1}{3} - 0.21\left(\frac{t}{A_r}\right)\left(1 - \frac{1}{12}\left\{\frac{t}{A_R}\right\}^4\right)\right]\right\}$ $K_{AB} = \left(A_A B_B^3\right)\left[\frac{1}{3} - 0.21\left(\frac{B_B}{A_A}\right)\left(1 - \frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$ $K_{CD} = \left(C_C D_{DG}^3\right)\left[\frac{1}{3} - 0.105\left(\frac{D_{DG}}{C_C}\right)\left(1 - \frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$ $A_R = 0.5(A - B)$ $G_{avg} = 0.5(g_0 + g_1)$ If $g_1 > g_0$: $h_p = h$ If $g_1 = g_0$: $h_p = \max\left\{0.35 g_0, \min\left[3 g_0, h_n, 0.78(0.5 B g_0)^{0.5}\right]\right\}$ If $t \geq G_{avg}$: $A_A = A_R, B_B = t, C_C = h_p, D_{DG} = G_{avg}$ If $t < G_{avg}$: $A_A = h_p + t, B_B = G_{avg}, C_C = A_R - G_{avg}, D_{DG} = t$
Loose-type flange with a hub	$I = \frac{0.0874 L g_o^2 h_o B}{V_L}$	

Table 4.16.7
Flange Moments of Inertia (Cont'd)

Flange Type	I	I_p
Loose-type flange without a hub	$I = \dfrac{Bt^3 \ln K}{24}$	$I_p = A_R t^3 \left[\dfrac{1}{3} - 0.21\left(\dfrac{t}{A_R}\right)\left(1 - \dfrac{1}{12}\left\{\dfrac{t}{A_R}\right\}^4\right)\right]$ $A_R = 0.5(A - B)$

Table 4.16.8
Flange Stress Equations

Flange Type	Stress Equations	
	Operating Condition	Gasket Seating Conditions
Integral-type flange or loose-type flange with a hub	$S_H = \dfrac{fM_o}{Lg_1^2 B}$ $S_R = \dfrac{(1.33te + 1)M_o}{Lt^2 B}$ $S_T = \dfrac{YM_o}{t^2 B} - ZS_R$	$S_H = \dfrac{fM_g}{Lg_1^2 B}$ $S_R = \dfrac{(1.33te + 1)M_g}{Lt^2 B}$ $S_T = \dfrac{YM_g}{t^2 B} - ZS_R$
Loose-type flange without a hub	$S_T = \dfrac{YM_o}{t^2 B}$	$S_T = \dfrac{YM_g}{t^2 B}$
Reverse integral-type flange or reverse loose-type flange with a hub	$S_H = \dfrac{fM_o}{L_r g_1^2 B^*}$ $S_R = \dfrac{(1.33te_r + 1)M_o}{L_r t^2 B^*}$ $S_{T1} = \dfrac{Y_r M_o}{t^2 B^*} - \dfrac{ZS_R(0.67te_r + 1)}{(1.33te_r + 1)}$ $S_{T2} = \left[Y - \dfrac{2K^2(0.67te_r + 1)}{(K^2 - 1)L_r}\right]\dfrac{M_o}{t^2 B^*}$	$S_H = \dfrac{fM_g}{L_r g_1^2 B^*}$ $S_R = \dfrac{(1.33te_r + 1)M_g}{L_r t^2 B^*}$ $S_{T1} = \dfrac{Y_r M_g}{t^2 B^*} - \dfrac{ZS_R(0.67te_r + 1)}{(1.33te_r + 1)}$ $S_{T2} = \left[Y - \dfrac{2K^2(0.67te_r + 1)}{(K^2 - 1)L_r}\right]\dfrac{M_g}{t^2 B^*}$
Reverse loose-type flange without a hub	$S_T = \dfrac{YM_o}{t^2 B^*}$	$S_T = \dfrac{YM_g}{t^2 B^*}$

Table 4.16.9
Flange Stress Acceptance Criteria

Flange Type	Stress Acceptance Criteria	
	Operating Condition	Gasket Seating Conditions
Integral-type flange or loose-type flange with a hub	$S_H \le \min\left[1.5S_{fo}, 2.5S_{no}\right]$ [Note (1)] $S_H \le 1.5S_{fo}$ [Note (2)] $S_R \le S_{fo}$ $S_T \le S_{fo}$ $\dfrac{(S_H + S_R)}{2} \le S_{fo}$ $\dfrac{(S_H + S_T)}{2} \le S_{fo}$	$S_H \le \min\left[1.5S_{fg}, 2.5S_{ng}\right]$ [Note (1)] $S_H \le 1.5S_{fg}$ [Note (2)] $S_R \le S_{fg}$ $S_T \le S_{fg}$ $\dfrac{(S_H + S_R)}{2} \le S_{fg}$ $\dfrac{(S_H + S_T)}{2} \le S_{fg}$

Table 4.16.9
Flange Stress Acceptance Criteria (Cont'd)

Flange Type	Stress Acceptance Criteria: Operating Condition	Stress Acceptance Criteria: Gasket Seating Conditions
Loose-type flange without a hub	$S_T \le S_{fo}$	$S_T \le S_{fg}$
Reverse integral-type flange or reverse loose-type flange with a hub	$S_H \le 1.5S_{fo}$ $S_R \le S_{fo}$ $S_{T1} \le S_{fo}$ $\frac{(S_H + S_R)}{2} \le S_{fo}$ $\frac{(S_H + S_{T1})}{2} \le S_{fo}$ $S_{T2} \le S_{fo}$	$S_H \le 1.5S_{fg}$ $S_R \le S_{fg}$ $S_{T1} \le S_{fg}$ $\frac{(S_H + S_R)}{2} \le S_{fg}$ $\frac{(S_H + S_{T1})}{2} \le S_{fg}$ $S_{T2} \le S_{fg}$
Reverse loose-type flanges	$S_T \le S_{fo}$	$S_T \le S_{fg}$

NOTES:
(1) For integral flanges with hubs welded to a nozzle neck, pipe, or vessel shell.
(2) For loose type flanges with a hub.
(3) Flanges made of non-ductile material, such as cast iron, are not addressed by this section.

Table 4.16.10
Flange Rigidity Criterion

Flange Type	Rigidity Criterion: Operating Condition	Rigidity Criterion: Gasket Seating Conditions
Integral-type flange	$J = \frac{52.14VM_o}{LE_{yo}g_0^2K_Rh_o} \le 1.0$	$J = \frac{52.14VM_g}{LE_{yg}g_0^2K_Rh_o} \le 1.0$
Loose-type flange with a hub	$J = \frac{52.14V_LM_o}{LE_{yo}g_0^2K_Rh_o} \le 1.0$	$J = \frac{52.14V_LM_g}{LE_{yg}g_0^2K_Rh_o} \le 1.0$
Reverse integral-type flange	$J = \frac{52.14VM_o}{L_rE_{yo}g_0^2K_Rh_o} \le 1.0$	$J = \frac{52.14VM_g}{L_rE_{yg}g_0^2K_Rh_o} \le 1.0$
Reverse loose-type flange with a hub	$J = \frac{52.14V_LM_o}{L_rE_{yo}g_0^2K_Rh_o} \le 1.0$	$J = \frac{52.14V_LM_g}{L_rE_{yg}g_0^2K_Rh_o} \le 1.0$
Loose-type and reverse loose-type flange without a hub	$J = \frac{109.4M_o}{E_{yo}t^3K_R(\ln K)} \le 1.0$	$J = \frac{109.4M_g}{E_{yg}t^3K_R(\ln K)} \le 1.0$

GENERAL NOTES:
(a) For an integral type flange, K_R = 0.3 unless other values are specified by the user.
(b) For a loose type flange with or without a hub, K_R = 0.2 unless other values are specified by the user.

Table 4.16.11
Bolt Spacing Equations

Flange Type	Bolt Spacing Factors
All	$B_{s\max} = 2a + \frac{6t}{m + 0.5}$ $B_{sc} = \max\left(1, \sqrt{\frac{B_s}{2a + t}}\right)$

Table 4.16.12
Moment Factor, F_M

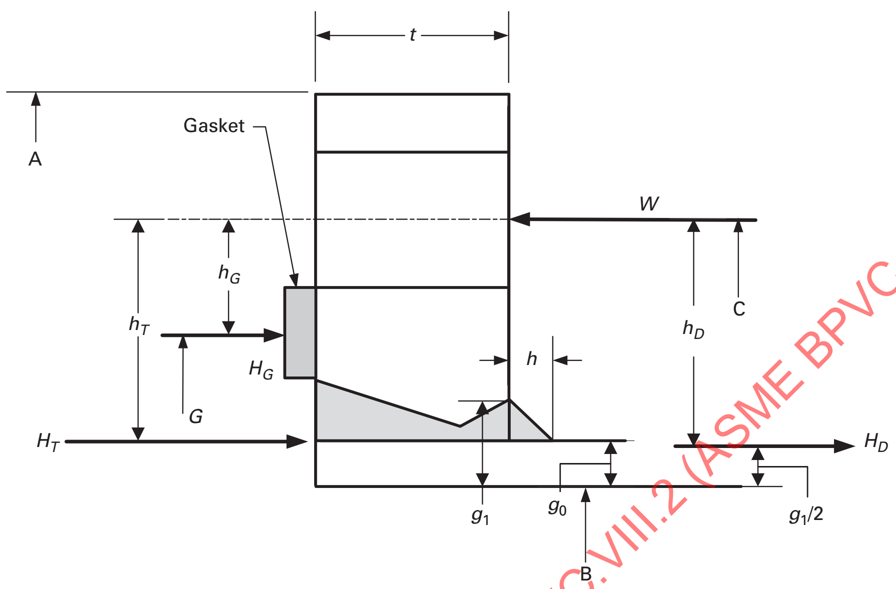
Standard	Size Range	Flange Pressure Rating Class 150	300	600	900	1500	2500
ASME B16.5	≤NPS 12	1.2	0.5	0.5	0.5	0.5	0.5
	>NPS 12 and ≤NPS 24	1.2	0.5	0.5	0.3	0.3	...
ASME B16.47							
Series A	All	0.6	0.1	0.1	0.1	...	...
Series B	<NPS 48	[Note (1)]	[Note (1)]	0.13	0.13	...	...
	≥NPS 48	0.1	[Note (2)]	...	...	...	...

GENERAL NOTES:
(a) The combinations of size ranges and flange pressure classes for which this Table gives no moment factor value are outside the scope of this Table.
(b) The designer should consider reducing the moment factor if the loading is primarily sustained in nature and the bolted flange joint operates at a temperature where gasket creep/relaxation will be significant.

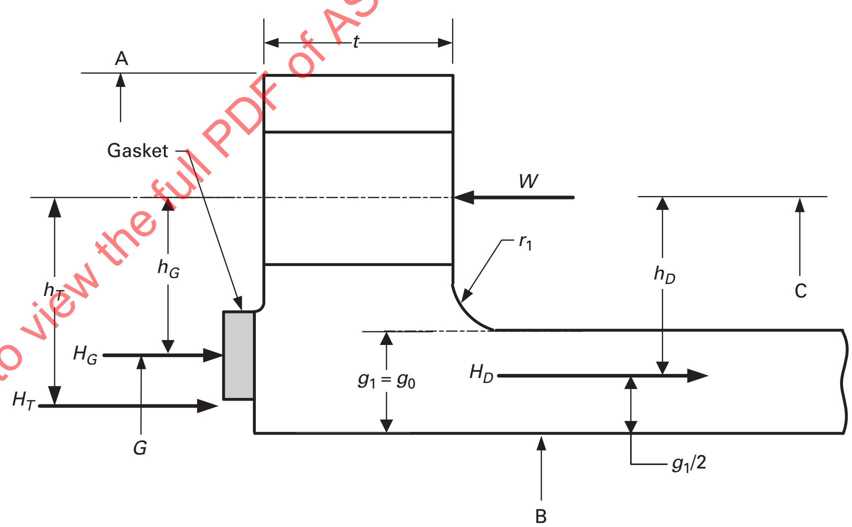
NOTES:
(1) $F_M = 0.1 + (48 - \text{NPS})/56$.
(2) $F_M = 0.1$ except NPS 60, Class 300, in which case $F_M = 0.03$.

4.16.15 FIGURES

Figure 4.16.1
Integral Type Flanges



(a) Integral Flange Without a Hub



(b) Integral Flange With $g_0 = g_1$

Figure 4.16.2
Integral Type Flanges With a Hub

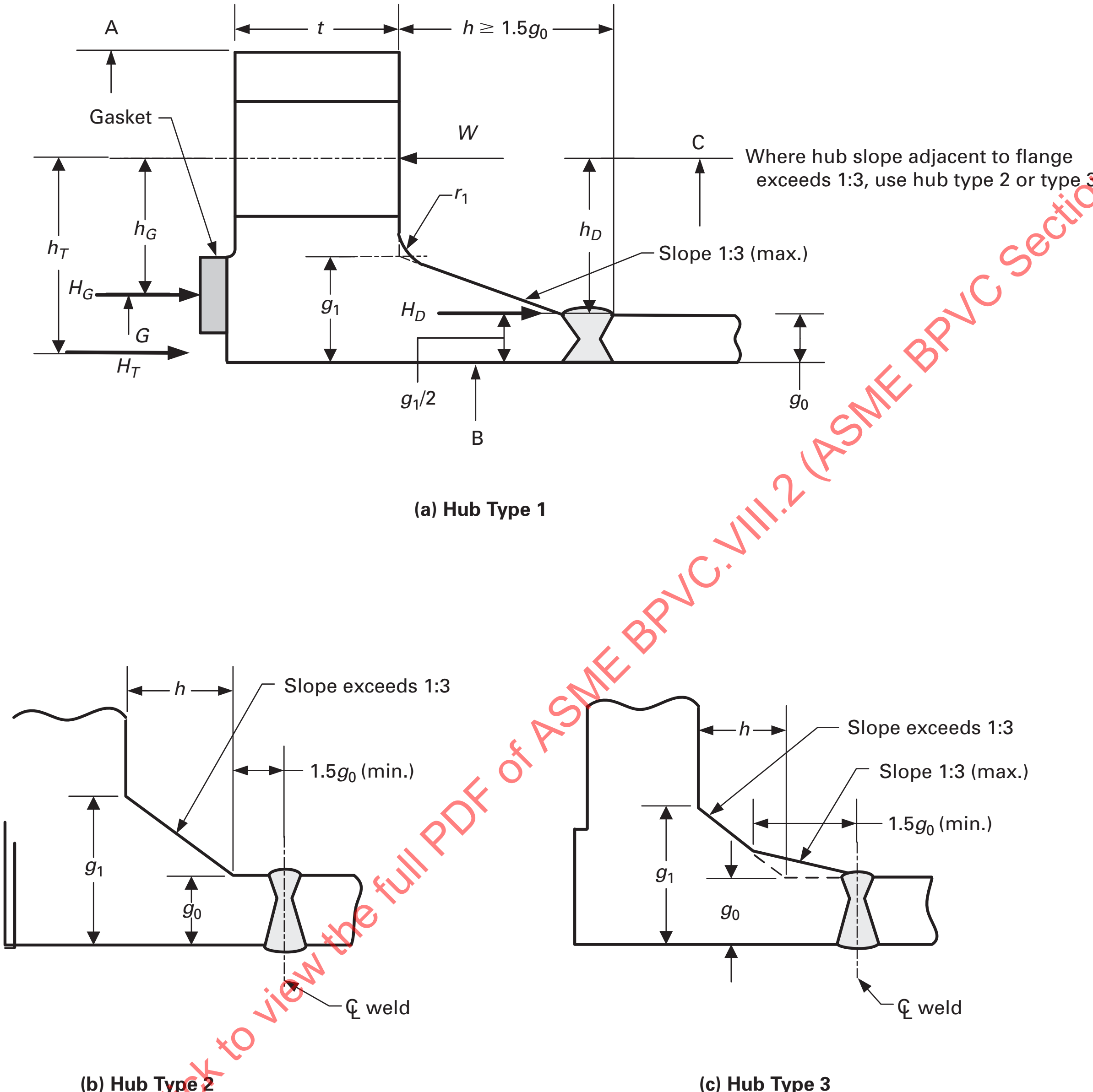


Figure 4.16.3
Integral Type Flanges With Nut Stops — Diameter Less Than or Equal to 450 mm (18 in.)

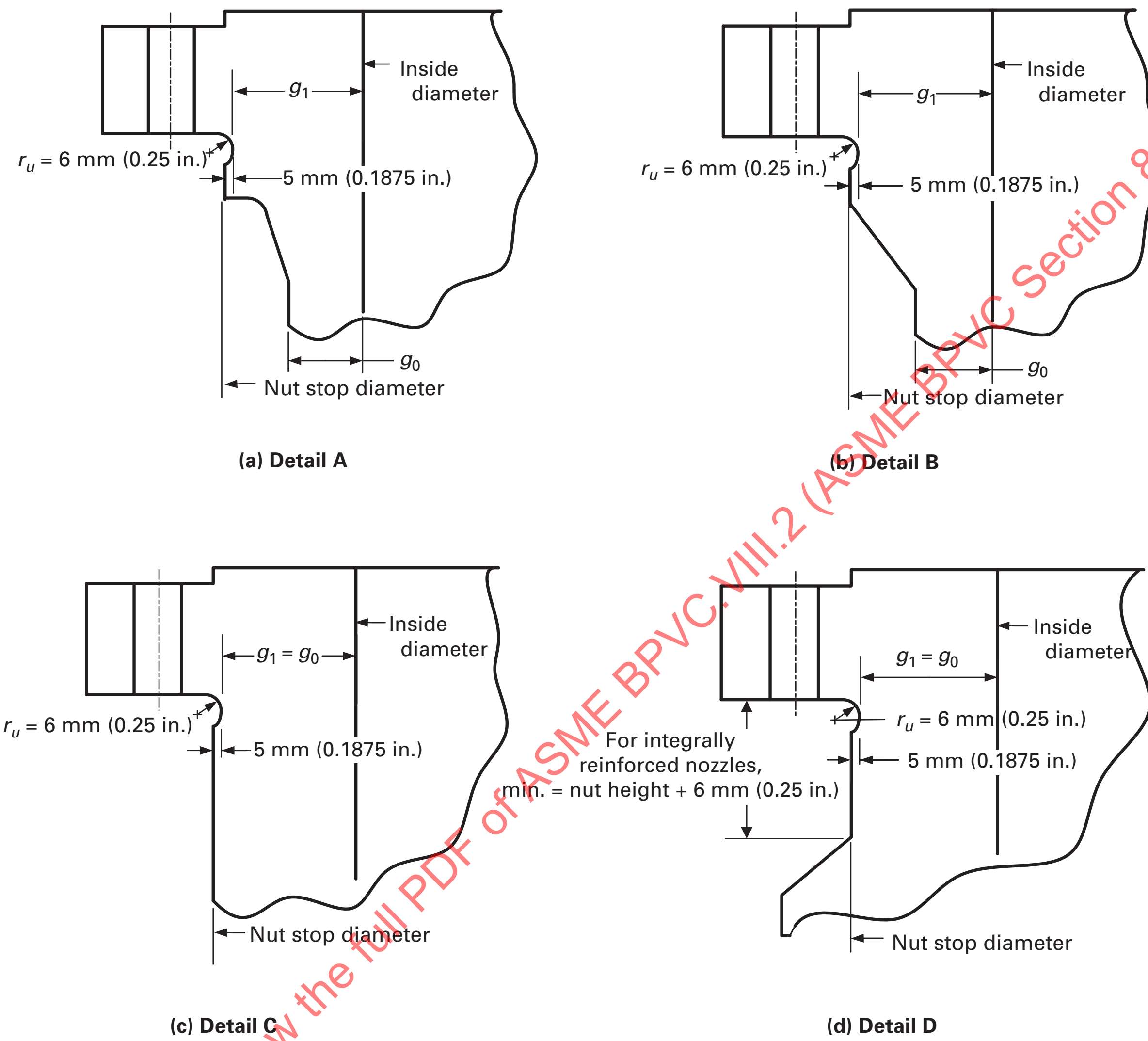
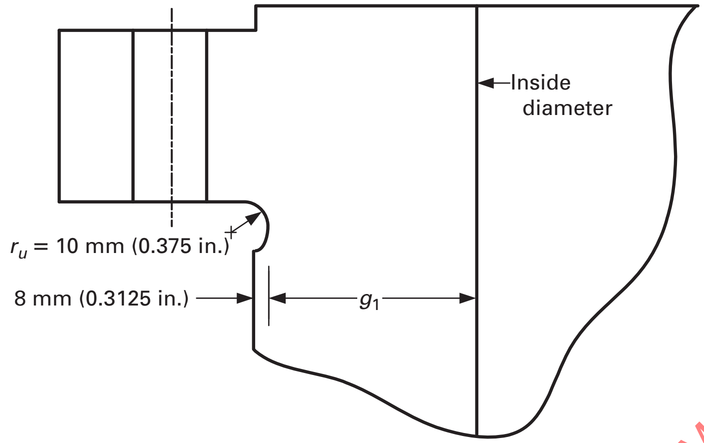
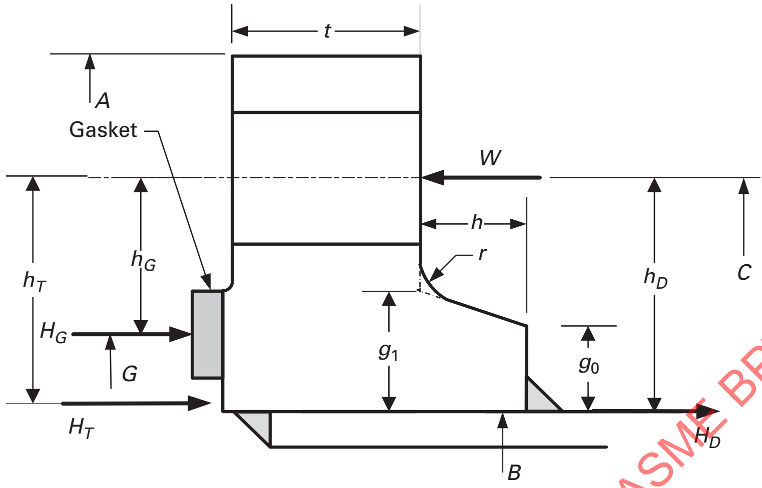


Figure 4.16.4
Integral Type Flanges With Nut Stops — Diameter Greater Than 450 mm (18 in.)

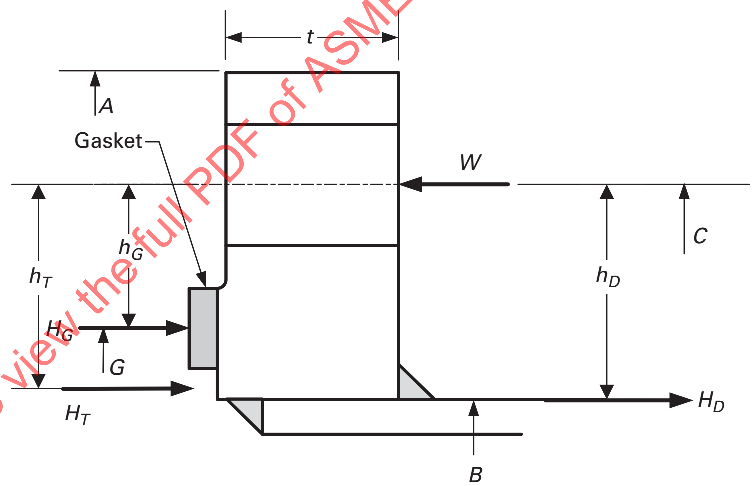


GENERAL NOTE: All other details per Figure 4.16.3.

Figure 4.16.5
Loose Type Flanges



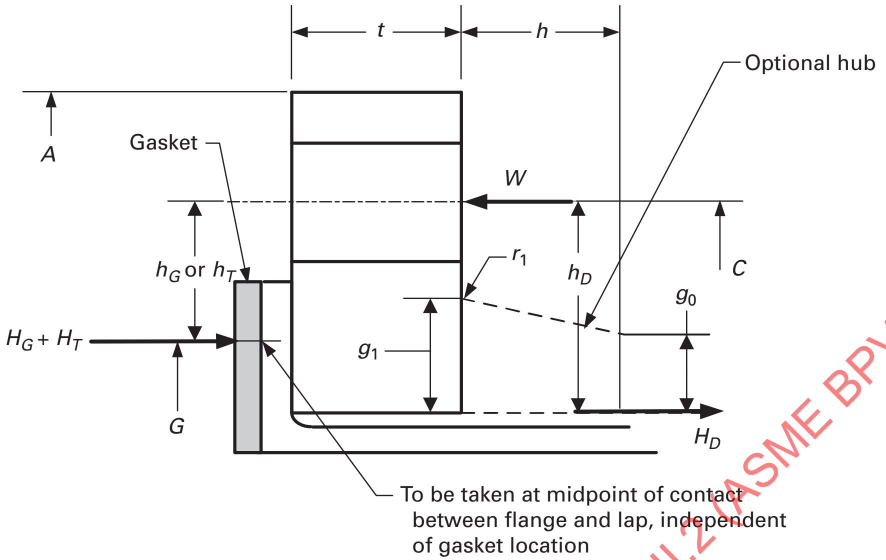
(a) Loose Flange With a Hub [Note (1)]



(b) Loose Flange Without a Hub

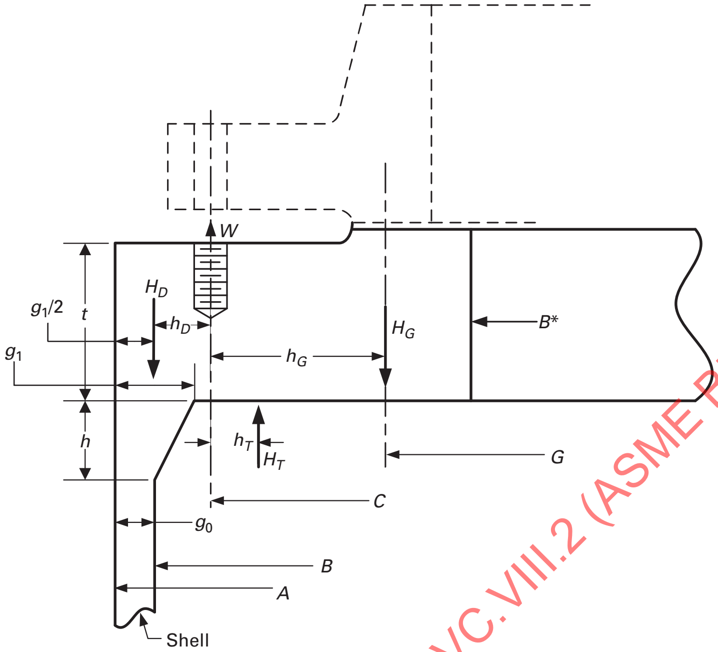
NOTE:
(1) For hub tapers 6 deg or less, use $g_0 = g_1$.

Figure 4.16.6
Loose-Type Lap Joint Type Flanges

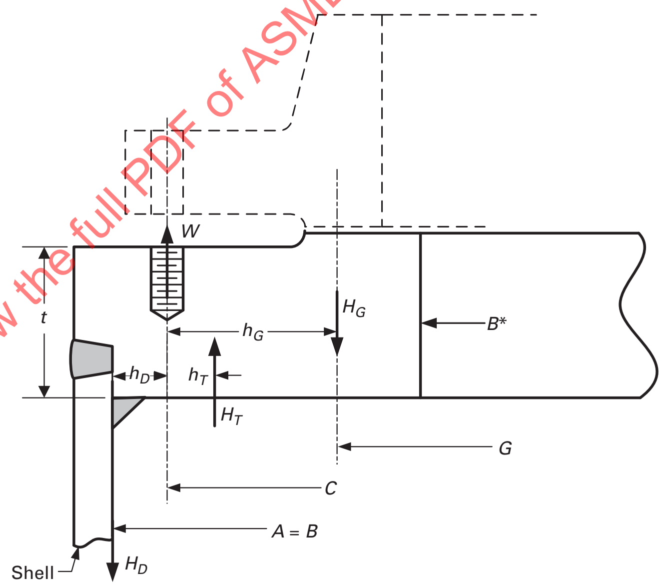


GENERAL NOTE: For hub tapers 6 deg or less, use $g_0 = g_1$ (loose-type flanges).

Figure 4.16.7
Reverse Flanges

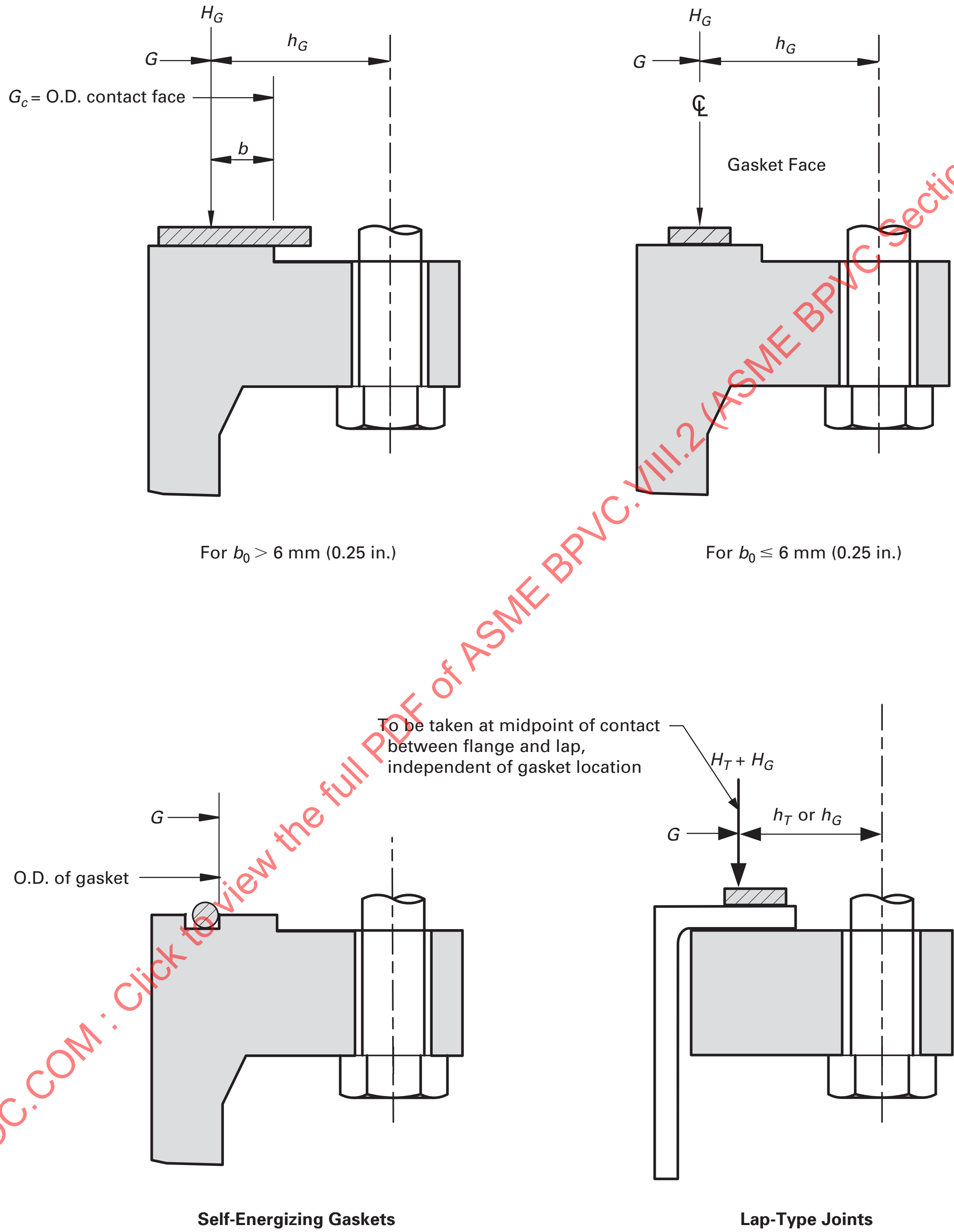


(a) Integral-Type Reverse Flange



(b) Loose-Type Reverse Flange

Figure 4.16.8
Location of Gasket Reaction Load Diameter



4.17 DESIGN RULES FOR CLAMPED CONNECTIONS

4.17.1 SCOPE

The rules in 4.17 apply specifically to the design of clamp connections for pressure vessels and vessel parts. These rules shall not be used for the determination of thickness of tubesheets integral with a hub nor for the determination of thickness of covers. These rules provide only for hydrostatic end loads, assembly, and gasket seating. Consideration shall be given to loads other than pressure, such as piping loads that may act upon the clamp connection (see 2.2.3.1).

4.17.2 DESIGN CONSIDERATIONS

4.17.2.1 The design of a clamp connection involves the selection of the gasket, bolting, hub, and clamp geometry. Connection dimensions shall be such that the calculated stresses in the clamp and the hub do not exceed the acceptability criteria of this paragraph.

4.17.2.2 In the design of a bolted flange connection, calculations shall be made for the following two design conditions, and the most severe condition shall govern the design of the flanged joint.

(a) Operating Conditions - The conditions required to resist the hydrostatic end force of the design pressure and any applied external forces and moments tending to part the joint, and to maintain on the gasket or joint-contact surface sufficient compression to assure a joint that meets the required tightness, all at the design temperature.

(b) Gasket Seating and Assembly Condition - The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts when assembling the joint, at atmospheric temperature and pressure.

4.17.2.3 Calculations shall be performed using dimensions of the flange in the corroded condition and the uncorroded condition, and the more severe case shall control.

4.17.2.4 It is recommended that either a pressure energized and/or low seating load gasket be used to compensate for possible non-uniformity in the gasket seating force distribution. Hub faces shall be designed such as to have metal-to-metal contact outside the gasket seal diameter. This may be provided by recessing the hub faces or by use of a metal spacer (see Figure 4.17.1). The contact area shall be sufficient to prevent yielding of either the hub face or spacer under both operating and assembly loads.

4.17.2.5 It is recognized that there are clamp designs that do not utilize wedging action during assembly since clamping surfaces are parallel to the hub faces. Such designs are acceptable and shall satisfy the bolting and corresponding clamp and hub requirements of a clamp connection designed for a total included clamping angle of 10 degrees.

4.17.2.6 The design method used in this paragraph to calculate stresses, loads, and moments may also be used for designing clamp connections of shapes differing from those shown in Figures 4.17.1 and 4.17.2, and for clamps consisting of more than two circumferential segments. The design equations in this paragraph may be modified when designing clamp connections of shapes differing from those shown in Figures 4.17.1 and 4.17.2, provided that the basis for the modifications is in accordance with 1.1.1.2. The clamp connections designed in this manner shall be provided with a bolt retainer. The retainer shall be designed such that in case of failure of the primary bolting, the retainer shall hold the clamps together independently under the operating loads. Multiple bolting (two or more bolts per lug) is an acceptable alternative for meeting this redundancy requirement. See 4.8.3.2 for additional requirements for bolt retainers and redundant bolting. No credit shall be taken for clamp hub friction toward satisfying the redundancy requirement.

4.17.3 FLANGE MATERIALS

4.17.3.1 Materials used in the construction of clamp connections shall comply with the requirements given in Part 3.

4.17.3.2 Hubs made from ferritic steel and designed in accordance with the rules herein shall be given a normalizing or full-annealing heat treatment when the thickness of the hub neck section exceeds 76 mm (3 in.).

4.17.3.3 Cast steel hubs and clamps shall be examined and repaired, if required, in accordance with Part 3.

4.17.3.4 Hubs and clamps shall not be machined from plate.

4.17.3.5 Bolts, studs, nuts and washers shall comply with Part 3. The minimum bolt diameter shall be 12 mm (0.5 in.). (**21**)

4.17.4 DESIGN BOLT LOADS

4.17.4.1 During assembly of the clamp connection, the design bolt load W is resolved into an effective clamp preload W_e, which is a function of the clamp-to-hub taper angle ϕ and the friction angle μ. An appropriate friction angle shall be established by the manufacturer, based on test results for both assembly and operating conditions.

4.17.4.2 The procedure to determine the bolt loads for the operating, and gasket seating and assembly conditions are shown below.

Step 1. Determine the design pressure and temperature of the flange joint.

Step 2. Select a gasket and determine the gasket factors m and y from Table 4.16.1.

Step 3. Determine the width of the gasket, basic gasket seating width, the location of the gasket reaction, G, based on the flange and gasket geometry and the information in Table 4.16.3.

(a) When $b_0 \leq 6$ mm (0.25 in.), then G is the mean diameter of the gasket or joint contact face

(b) When $b_0 > 6$ mm (0.25 in.), then G is the outside diameter of the gasket contact face minus $2b$.

Step 4. Determine the flange forces for the bolt load calculation.

$$H = 0.785G^2P \tag{4.17.1}$$

$$H_p = 2b\pi GmP \quad \text{for non-self-energized gaskets} \tag{4.17.2}$$

$$H_m = \pi bGy \quad \text{for non-self-energized gaskets} \tag{4.17.3}$$

$$H_p = 0.0 \quad \text{for self-energized gaskets} \tag{4.17.4}$$

$$H_m = 0.0 \quad \text{for self-energized gaskets} \tag{4.17.5}$$

Note that where a significant axial force is required to compress the gasket during assembly of a joint containing a self-energizing gasket, the value of H_m shall be taken as equal to that axial force.

Step 5. Determine the design bolt load for the operating condition.

$$W_o = \frac{2}{\pi}\left(H + H_p\right)\tan[\phi - \mu] \tag{4.17.6}$$

Step 6. Determine the minimum required total bolt load for gasket seating and assembly conditions.

$$W_{g1} = \frac{2}{\pi}H_m\tan[\phi + \mu] \tag{4.17.7}$$

$$W_{g2} = \frac{2}{\pi}\left(H + H_p\right)\tan[\phi + \mu] \tag{4.17.8}$$

Step 7. Determine the design bolt load for the gasket seating and assembly condition.

$$W_g = \left(A_m + A_b\right) S_{bg} \tag{4.17.9}$$

Alternatively, if controlled bolting (e.g., torque-control or bold tensioning) techniques are used to assemble the clamp, the assembly design bolt load shall be taken as

$$W_g = 2A_m S_{bg} \tag{4.17.10}$$

NOTE: In addition, the Manufacturer shall provide to the User a copy of the bolting instructions that were used. It is recommended that the Manufacturer refer to ASME PCC-1, *Guidelines for Pressure Boundary Bolted Flange Joint Assembly*. It is cautioned that bolt loads in excess of those calculated using eq. (4.17.10) can overstress the clamp.

The parameter A_b is the actual total cross-sectional area of the bolts that is selected such that $A_b \geq A_m$, where

$$A_m = \max\left[\frac{W_o}{2S_{bo}}, \frac{W_{g1}}{2S_{bg}}, \frac{W_{g2}}{2S_{bg}}\right] \tag{4.17.11}$$

4.17.4.3 In eq. (4.17.6), credit for friction is allowed based on clamp connection geometry and experience, but the bolt load shall not be less than that determined using a value of $(\phi - \mu)$ equal to 5 deg. Friction is also considered in determining bolt loads by eqs. (4.17.7) and (4.17.8), but the μ factor used shall not be less than 5 deg.

4.17.5 FLANGE AND CLAMP DESIGN PROCEDURE

The procedure to design a clamp connection is shown below.

Step 1. Determine the design pressure and temperature of the flange joint.

Step 2. Determine an initial flange and clamp geometry (see Figures 4.17.1 and 4.17.2).

Step 3. Determine the design bolt loads for operating condition, W_o, and the gasket seating and assembly condition, W_g, from 4.17.4.2.

Step 4. Determine the flange forces, H, H_p, and H_m from 4.17.4.2, Step 4, and

$$H_D = 0.785B^2P \tag{4.17.12}$$

$$H_G = \frac{1.571W_o}{\tan\left[\phi + \mu\right]} - \left(H + H_p\right) \tag{4.17.13}$$

$$H_T = H - H_D \tag{4.17.14}$$

Step 5. Determine the flange moment for the operating condition.

$$M_o = M_D + M_G + M_T + M_F + M_P + M_R \tag{4.17.15}$$

where

$$M_D = H_D\left[\frac{C - \left(B + g_1\right)}{2}\right] \tag{4.17.16}$$

$$M_G = H_G h_G \tag{4.17.17}$$

$$M_T = H_T\left[\frac{C}{2} - \frac{\left(B + G\right)}{4}\right] \tag{4.17.18}$$

$$M_F = H_D\left(\frac{g_1 - g_0}{2}\right) \tag{4.17.19}$$

$$M_P = PBT\pi\left(\frac{T}{2} - \bar{h}\right) \tag{4.17.20}$$

$$M_R = 1.571W_o\left(\bar{h} - T + \frac{\left(C - N\right)\tan\left(\phi\right)}{2}\right) \tag{4.17.21}$$

$$C = \frac{\left(A + C_i\right)}{2} \tag{4.17.22}$$

$$\bar{h} = \frac{T^2 g_1 + h_2^2 g_2}{2\left(T g_1 + h_2 g_2\right)} \tag{4.17.23}$$

$$h_2 = T - \frac{g_2 \tan\left[\phi\right]}{2} \tag{4.17.24}$$

Step 6. Determine the flange moment for the gasket seating condition

$$M_g = \frac{0.785W_g\left(C - G\right)}{\tan\left[\phi + \mu\right]} \tag{4.17.25}$$

Step 7. Determine the hub factors

$$F_H = 1 + \frac{1.818}{\sqrt{Bg_1}}\left[T - \bar{h} + \frac{3.305 I_h}{g_1^2\left(0.5B + \bar{g}\right)}\right] \tag{4.17.26}$$

$$I_h = \frac{g_1 T^3}{3} + \frac{g_2 h_2^3}{3} - \left(g_2 h_2 + g_1 T\right)\bar{h}^2 \tag{4.17.27}$$

$$\bar{g} = \frac{T g_1^2 + h_2 g_2\left(2g_1 + g_2\right)}{2\left(T g_1 + h_2 g_2\right)} \tag{4.17.28}$$

Step 8. Determine the reaction shear force at the hub neck for the operating condition.

$$Q_o = \frac{1.818 M_o}{F_H\sqrt{B g_1}} \tag{4.17.29}$$

Step 9. Determine the reaction shear force at the hub neck for the gasket seating condition.

$$Q_g = \frac{1.818 M_g}{F_H\sqrt{B g_1}} \tag{4.17.30}$$

Step 10. Determine the clamp factors.

$$e_b = B_c - \frac{C_i}{2} - l_c - X \tag{4.17.31}$$

$$X = \frac{\left(\frac{C_w}{2} - \frac{C_t}{3}\right)C_t^2 - 0.5\left(C_w - C_g\right)l_c^2}{A_c} \tag{4.17.32}$$

$$A_c = A_1 + A_2 + A_3 \tag{4.17.33}$$

$$I_c = \left(\frac{A_1}{3} + \frac{A_2}{4}\right)C_t^2 + \frac{A_3 l_c^2}{3} - A_c X^2 \tag{4.17.34}$$

$$A_1 = \left(C_w - 2C_t\right)C_t \tag{4.17.35}$$

$$A_2 = 1.571 C_t^2 \tag{4.17.36}$$

$$A_3 = \left(C_w - C_g\right)l_c \tag{4.17.37}$$

Step 11. Determine the hub stress correction factor, f, based on g_1, g_0, h, and B using the equations in Table 4.16.5 and l_m using the following equation.

$$l_m = l_c - 0.5\left(C - C_i\right) \tag{4.17.38}$$

Step 12. Determine the flange and clamp stresses for the operating and gasket seating conditions using the equations in Table 4.17.1.

Step 13. Check the flange stress acceptance criteria for the operating and gasket seating conditions are shown in Table 4.17.2. If the stress criteria are satisfied, then the design is complete. If the stress criteria are not satisfied, then re-proportion the flange dimensions and go to Step 2.

4.17.6 NOMENCLATURE

A = outside diameter of the hub.
A_b = total cross-sectional area of the bolts per lug based on the root diameter or the least diameter of the unthreaded portion, if less.
A_c = effective clamp cross-sectional area.
A_m = total minimum required cross-sectional area of the bolts per lug.
A_1 = partial clamp area.
A_2 = partial clamp area.
A_3 = partial clamp area
α = hub transition angle, maximum 45 deg.
B = inside diameter of the hub.
B_c = radial distance from connection center line to the center of the bolts.
b = effective gasket contact width.
b_0 = basic gasket seating width.
C = diameter of effective clamp-hub reaction circle.
C_i = inside diameter of clamp.
C_g = effective clamp gap determined at diameter C.
C_t = effective clamp thickness subject to the following condition $C_t \geq r$.
C_w = clamp width.
e_b = radial distance from center of the bolts to the centroid of the clamp cross section
f = hub stress correction factor.
F_H = factor relating total rotational moment to the reaction moment at the hub neck
g_0 = thickness of hub neck at small end.
g_1 = thickness of hub neck at intersection with hub shoulder.
g_2 = height of hub shoulder.
$\bar{g}$ = radial distance from the hub inside diameter B to the hub shoulder ring centroid.
G = location of the gasket reaction.
h = taper hub length.
h_G = radial distance from effective clamp-hub reaction circle to the circle on which H_G acts.
h_2 = average thickness of hub shoulder
$\bar{h}$ = axial distance from the hub face to the hub shoulder ring centroid
H = total hydrostatic end force.
H_D = hydrostatic end force on bore area.
H_G = difference between total effective axial clamping preload and the sum of total hydrostatic end force and total joint contact surface compression.
H_m = total axial gasket seating requirements for makeup.
H_p = total joint contact surface compression load.
H_T = difference between total hydrostatic end force and hydrostatic end force on bore area.
I_c = moment of inertia of clamp relative to neutral axis of entire section.
I_h = moment of inertia of hub shoulder relative to its neutral axis
L_a = distance from W to the point where the clamp lug joins the clamp body.
L_h = clamp lug height.
L_w = clamp lug width.
l_c = effective clamp lip length.
l_m = effective clamp lip moment arm.
M_D = moment due to H_D.
M_F = offset moment.
M_G = moment due to H_G.
M_P = pressure moment.
M_R = radial clamp equilibriating moment.
M_T = moment due to H_T.
M_o = flange design moment for the operating condition.
M_g = flange design moment for the gasket seating condition.
μ = friction angle.
m = gasket factor.
N = outside diameter of hub neck.

m = gasket factor.
P = design pressure.
ϕ = clamp shoulder angle, maximum 40 deg.
Q_g = reaction shear force at the hub neck for the gasket seating condition.
Q_o = reaction shear force at the hub neck for the design operating condition.
r = clamp or hub cross section corner radius.
S_{bg} = allowable stress from Annex 3-A for the bolt evaluated at the gasket seating temperature.
S_{bo} = allowable stress from Annex 3-A 3 for the bolt evaluated at the design temperature.
S_{cg} = allowable stress from Annex 3-A for the clamp evaluated at the gasket seating temperature.
S_{co} = allowable stress from Annex 3-A for the clamp evaluated at the design temperature.
S_{hg} = allowable stress from Annex 3-A for the hub evaluated at the gasket seating temperature.
S_{ho} = allowable stress from Annex 3-A for the hub evaluated at the design temperature.
S_{1g} = hub longitudinal stress on outside at hub neck for the design gasket seating condition.
S_{1o} = hub longitudinal stress on outside at hub neck for the design operating condition.
S_{2g} = maximum Lame hoop stress at bore of hub for the design gasket seating condition.
S_{2o} = maximum Lame hoop stress at bore of hub for the design operating condition.
S_{3g} = maximum hub shear stress at shoulder for the design gasket seating condition.
S_{3o} = maximum hub shear stress at shoulder for the design operating condition.
S_{4g} = maximum radial hub shear stress in neck for the design gasket seating condition.
S_{4o} = maximum radial hub shear stress in neck for the design operating condition.
S_{5g} = clamp longitudinal stress at clamp body inner diameter for the design gasket seating condition.
S_{5o} = clamp longitudinal stress at clamp body inner diameter for the design operating condition.
S_{6g} = clamp tangential stress at clamp body outer diameter for the design gasket seating condition.
S_{6o} = clamp tangential stress at clamp body outer diameter for the design operating condition.
S_{7g} = maximum shear stress in clamp lips for the design gasket seating condition.
S_{7o} = maximum shear stress in clamp lips for the design operating condition.
S_{8g} = clamp lug bending stress for the design gasket seating condition.
S_{8o} = clamp lug bending stress for the design operating condition.
S_{9g} = effective bearing stress between clamp and hub for the design gasket seating condition.
S_{9o} = effective bearing stress between clamp and hub for the design operating condition.
T = thickness of hub shoulder.
W_g = total clamp connection design bolt load on both lugs for the gasket seating and assembly condition.
W_{g1} = total clamp connection design bolt load on both lugs for the gasket seating condition.
W_{g2} = total clamp connection design bolt load on both lugs for the assembly condition.
W_o = total clamp connection design bolt load on both lugs for the operating condition.
X = clamp dimension to neutral axis.
y = gasket seating stress.
Z = clamp-hub taper angle.

4.17.7 TABLES

Table 4.17.1
Flange Stress Equations

(21)

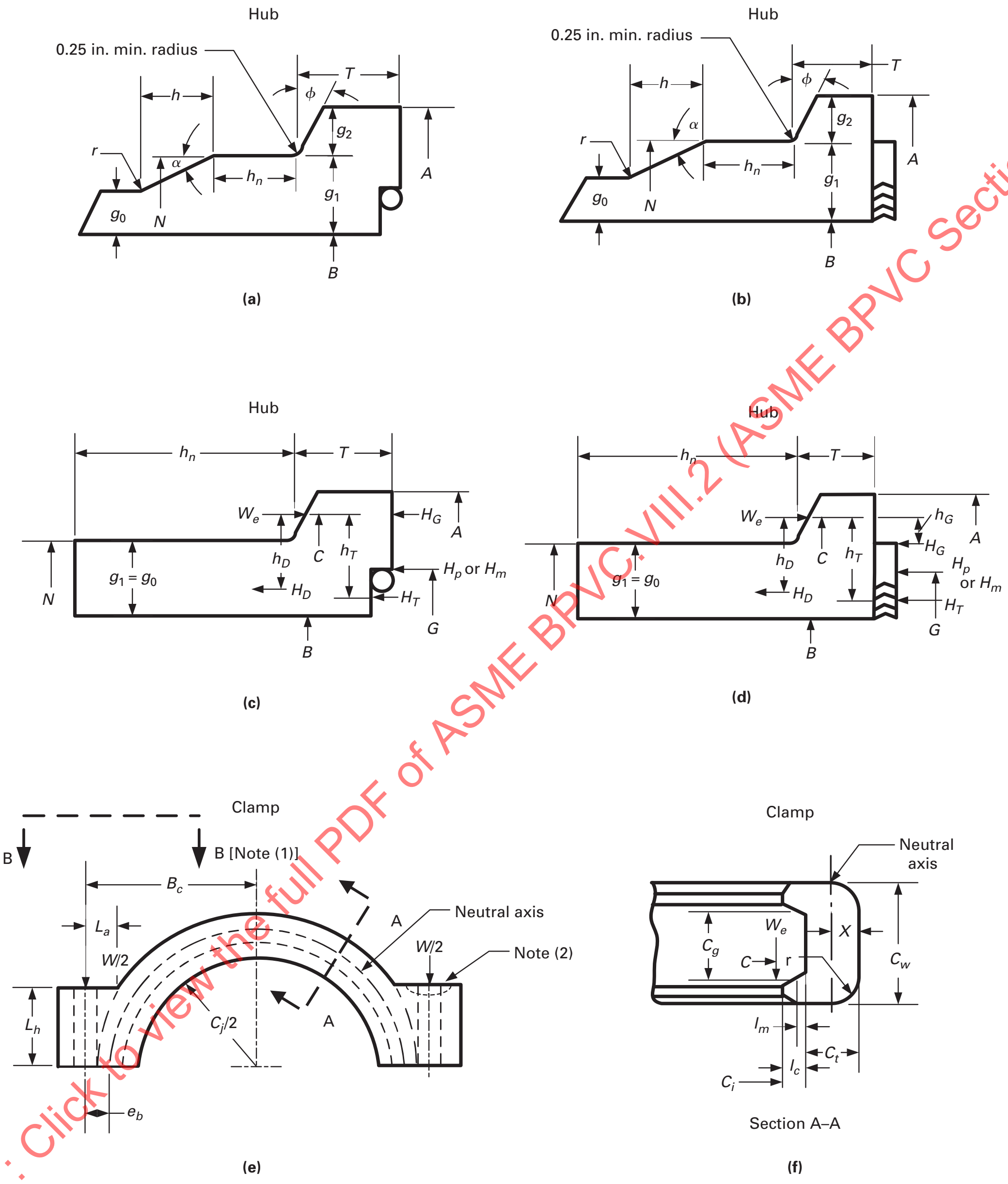
Location	Stress Equations: Operating Condition	Stress Equations: Gasket Seating/Assembly Conditions
Flange	$S_{1o} = f\left[\dfrac{PB^2}{4g_1(B+g_1)} + \dfrac{1.91M_o}{g_1^2(B+g_1)F_H}\right]$	$S_{1g} = f\left[\dfrac{1.91M_g}{g_1^2(B+g_1)F_H}\right]$
	$S_{2o} = P\left(\dfrac{N^2+B^2}{N^2-B^2}\right)$	$S_{2g} = 0.0$
	$S_{3o} = \dfrac{0.75W_o}{T(B+2g_1)\tan(\phi-\mu)}$	$S_{3g} = \dfrac{0.75W_g}{T(B+2g_1)\tan(\phi+\mu)}$
	$S_{4o} = \dfrac{0.477Q_o}{g_1(B+g_1)}$	$S_{4g} = \dfrac{0.477Q_g}{g_1(B+g_1)}$
Clamp	$S_{5o} = \dfrac{W_o}{2C\tan(\phi-\mu)}\left[\dfrac{1}{C_t} + \dfrac{3(C_t+2l_m)}{C_t^2}\right]$	$S_{5g} = \dfrac{W_g}{2C\tan(\phi+\mu)}\left[\dfrac{1}{C_t} + \dfrac{3(C_t+2l_m)}{C_t^2}\right]$
	$S_{6o} = \dfrac{W_o}{2}\left[\dfrac{1}{A_c} + \dfrac{\lvert e_b\rvert\cdot(C_t-X)}{I_c}\right]$	$S_{6g} = \dfrac{W_g}{2}\left[\dfrac{1}{A_c} + \dfrac{\lvert e_b\rvert\cdot(C_t-X)}{I_c}\right]$
	$S_{7o} = \dfrac{1.5W_o}{(C_w-C_g)C\tan(\phi-\mu)}$	$S_{7g} = \dfrac{1.5W_g}{(C_w-C_g)C\tan(\phi+\mu)}$
	$S_{8o} = \dfrac{3W_oL_a}{L_wL_h^2}$	$S_{8g} = \dfrac{3W_gL_a}{L_wL_h^2}$
	$S_{9o} = \dfrac{W_o}{(A-C_i)C\tan(\phi-\mu)}$	$S_{9g} = \dfrac{W_g}{(A-C_i)C\tan(\phi+\mu)}$

Table 4.17.2
Flange Stress Acceptance Criteria

Location	Stress Acceptance Criteria	
	Operating Condition	Gasket Seating/Assembly Conditions
Flange	$S_{1o} \leq 1.5S_{ho}$	$S_{1g} \leq 1.5S_{hg}$
	$S_{2o} \leq S_{ho}$	$S_{2g} \leq S_{hg}$
	$S_{3o} \leq 0.8S_{ho}$	$S_{3g} \leq 0.8S_{hg}$
	$S_{4o} \leq 0.8S_{ho}$	$S_{4g} \leq 0.8S_{hg}$
Clamp	$S_{5o} \leq 1.5S_{co}$	$S_{5g} \leq 1.5S_{cg}$
	$S_{6o} \leq 1.5S_{co}$	$S_{6g} \leq 1.5S_{cg}$
	$S_{7o} \leq 0.8S_{co}$	$S_{7g} \leq 0.8S_{cg}$
	$S_{8o} \leq S_{co}$	$S_{8g} \leq S_{cg}$
	$S_{9o} \leq 1.6\min[S_{ho}, S_{co}]$	$S_{9g} \leq 1.6\min[S_{hg}, S_{cg}]$

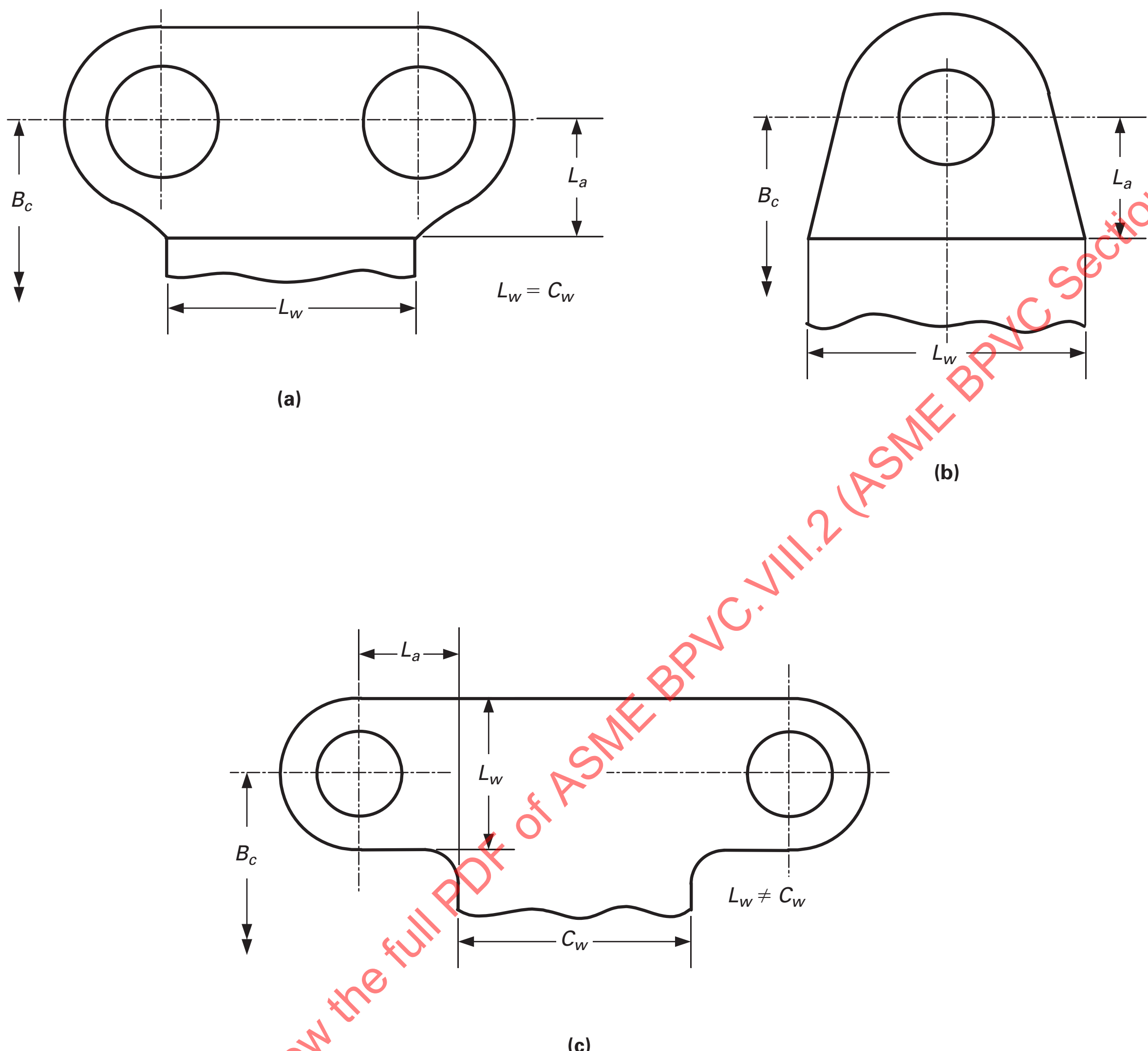
4.17.8 FIGURES

Figure 4.17.1
Typical Hub and Clamp Configuration



NOTES:
(1) See Figure 4.17.2 for section B–B.
(2) Clamp may have spherical depressions at bolt holes to facilitate the use of spherical nuts.

Figure 4.17.2
Typical Clamp Lugs Configurations



4.18 DESIGN RULES FOR SHELL-AND-TUBE HEAT EXCHANGERS

4.18.1 SCOPE

(a) The rules in 4.18 cover the minimum requirements for design, fabrication, and inspection of shell-and-tube heat exchangers.

(b) The rules in 4.18 cover the common types of shell-and-tube heat exchangers and their elements but are not intended to limit the configurations or details to those illustrated or otherwise described herein. Designs that differ from those covered in 4.18 shall be in accordance with 4.1.1.2.

4.18.2 TERMINOLOGY

(a) U-Tube Heat Exchanger - A heat exchanger with one stationary tubesheet attached to the shell and channel. The heat exchanger contains a bundle of U-tubes attached to the tubesheet [see Figure 4.18.1, sketch (a)].

(b) Fixed Tubesheet Heat Exchanger - A heat exchanger with two stationary tubesheets, each attached to the shell and channel. The heat exchanger contains a bundle of straight tubes connecting both tubesheets [see Figure 4.18.1, sketch (b)].

(c) Floating Tubesheet Heat Exchanger - A heat exchanger with one stationary tubesheet attached to the shell and channel, and one floating tubesheet that can move axially. The heat exchanger contains a bundle of straight tubes connecting both tubesheets [see Figure 4.18.1, sketch (c)].

4.18.3 GENERAL DESIGN CONSIDERATIONS

(a) The design of all components shall be in accordance with the applicable rules of all Parts of this Division.

(b) The design of flanges shall consider the effects of pass partition gasketing in determining the minimum required bolt loads, W_o and W_g, of 4.16. When the tubesheet is gasketed between the shell and channel flanges, the shell and channel flange bolt loads are identical and shall be treated as flange pairs in accordance with 4.16.

(c) Rules for U-tube heat exchangers are covered in 4.18.7.

(d) Rules for fixed tubesheet heat exchangers are covered in 4.18.8.

(e) Rules for floating tubesheet heat exchangers are covered in 4.18.9.

(f) Distribution and vapor belts shall be designed in accordance with the following:

(1) Where the shell is not continuous across the belt, the design shall be in accordance with 4.18.12.

(2) Where the shell is continuous across the belt, the design shall be in accordance with 4.11 for Type 1. The longitudinal stress in the shell section with openings (for flow into the shell) shall be based on the net area of the shell (the shell area less that removed by the openings) and shall not exceed the applicable allowable stress criteria. For U-tube and floating head exchangers, the allowable axial stress is S (see 4.1.12) for the shell material, and for fixed tubesheet exchangers, the allowable stress is as defined in 4.18.8.4, Step 10.

(g) Requirements for tubes shall be as follows:

(1) The allowable axial tube stresses in fixed and floating tubesheet heat exchangers shall be in accordance with 4.18.8 and 4.18.9.

(2) The thickness of U-tubes after forming shall not be less than the design thickness.

(h) Except as limited in (1) and (2), nozzles in cylindrical shells or cylindrical channels adjacent to integral tubesheets (see Figure 4.18.19) may be located at any distance from the tubesheet (refer to 4.5.18 for nomenclature not defined in this paragraph). These requirements do not apply to nozzles in shells or channels having tubesheets that are calculated as simply supported (see 4.18.7.5, 4.18.8.8, and 4.18.9.7).

(1) For a circular nozzle with d greater than 30% of D, no part of d may be located within $1.8(Dt)^{1/2}$ of the adjacent tubesheet face (see Figure 4.18.19).

(2) For a noncircular nozzle, d_{max} (major axis) is defined as the maximum diameter of d, and d_{min} is defined as the minimum diameter of d.

(-a) For a noncircular nozzle having its major axis not parallel to the tubesheet face and $d_{max}/D > 30\%$, d is limited to the distance specified in (1).

(-b) For a noncircular nozzle having its major axis parallel to the tubesheet face and $d_{max}/D > 30\%$, no part of d may be within $1.8(Dt)^{1/2} + (d_{max} - d_{min})/2$ of the adjacent tubesheet face.

(3) Nozzles subject to the limitations in (1) or (2) may have their required reinforcement located within $1.8(Dt)^{1/2}$ of the adjacent tubesheet face.

NOTE: Tubesheet deflection, especially when the tubesheet thickness is less than the tube diameter, may contribute to tube-to-tubesheet joint leakage; likewise, deflection of a tubesheet or flat bolted cover may result in fluid leakage across a gasketed pass partition plate. Such leakages can be detrimental to the thermal performance of the heat exchanger and deflection may need to be considered by the designer.

4.18.4 GENERAL CONDITIONS OF APPLICABILITY FOR TUBESHEETS

(a) The tubesheet shall be flat and circular.

(b) The tubesheet shall be of uniform thickness, except that the thickness of a tubesheet extension as determined in 4.18.5 may differ from the center thickness as determined in 4.18.7, 4.18.8, and 4.18.9. The outside diameter A used for the tubesheet calculations shall not exceed the diameter at which the thickness of the tubesheet extension is less than the minimum of $0.75h$ or h – 10 mm (h – 0.375 in.).

(c) The tubesheet shall be uniformly perforated over a nominally circular area, in either equilateral triangular or square patterns. However, untubed lanes for pass partitions are permitted.

(d) The channel component integral with the tubesheet (Configurations a, e, f and A for U-Tube, Fixed, and Floating Tubesheets) shall be either a cylinder or a hemispherical head (see Figure 4.18.15). The hemispherical head rules shall be used when the head is attached directly to the tubesheet and there are no cylindrical straight sections between the head and the tubesheet. If a hemispherical head is attached to the hub of a tubesheet, the hub may be considered part of the hemispherical head and not require an intervening cylinder, provided the hub complies with one of the following:

(1) It is shaped as a continuation of the head in accordance with Figure 4.18.15, sketch (b).

(2) It meets the requirements of Figure 4.18.15, sketch (c).

For both cases, the tangent line of the head is coincident with the adjacent face of the tubesheet.

(e) The tube side and shell side pressures are assumed to be uniform. These rules do not cover weight loadings or pressure drop.

(f) For the design pressure-only conditions (design loading cases), the design pressure shall be used. For the operating thermal-pressure conditions (operating loading cases), either the operating pressure or design pressure shall be used.

(g) The design rules in 4.18.7, 4.18.8, and 4.18.9 are based on a fully assembled heat exchanger. If pressure is to be applied to a partially assembled heat exchanger having a Configuration "d" tubesheet that is extended for bolting, special consideration, in addition to the rules in 4.18.5, 4.18.7, 4.18.8, and 4.18.9, shall be given to ensure that the tubesheet is not overstressed for the condition considered.

4.18.5 TUBESHEET FLANGED EXTENSION

4.18.5.1 Scope.

(a) Tubesheet extensions, if present, may be extended as a flange (flanged) or not extended as a flange (unflanged).

(1) Configuration a tubesheets may have no extension or an unflanged extension.

(2) Configurations b, e, and B tubesheets have flanged extensions.

(3) Configurations c, f, and C tubesheets have unflanged extensions.

(4) Configuration d may have a flanged or unflanged tubesheet extension.

(5) Configurations A and D do not have tubesheet extensions.

(b) These rules cover the design of tubesheet extensions that have loads applied to them.

(c) The required thickness of the tubesheet extension may differ from that required for the interior of the tubesheet as calculated in 4.18.7, 4.18.8, and 4.18.9.

4.18.5.2 Conditions of Applicability.

(a) The general conditions of applicability given in 4.18.4 apply.

(b) These rules do not apply to Configurations a, A, and D.

(c) These rules apply to flanged extensions that have bolt loads applied to them (Configurations b, e, and B). This includes Configuration d, if the extension is flanged and there are bolt loads applied to the extension.

(d) These rules apply to unflanged extensions (Configurations c, d, f, and C) and flanged extensions that have no bolt loads applied to them (Configuration d), if the thickness of the extension is less than the tubesheet thickness, h. If the tubesheet extension is equal to or greater than the tubesheet thickness, h, no analysis is required.

4.18.5.3 Design Considerations.

(a) The designer should take appropriate consideration of the stresses resulting from the pressure test required in 4.1.6.2 and Part 8. Special consideration shall be required for tubesheets that are gasketed on both sides when the pressure test in each chamber is conducted independently, and the bolt loading is only applied to the flanged extension during the pressure test.

(b) If the tubesheet is grooved for a peripheral gasket, the thinnest section of the flanged extension shall not be less than h_r. Figure 4.18.16 depicts h_r for some representative configurations

4.18.5.4 Calculation Procedure.

(a) For flanged extensions that have bolt loads applied to them [Configurations b, d (extended for bolting), e, and B], the procedure for calculating the minimum required thickness of the extension, h_r, is as follows:

$$h_r = \max\left(\sqrt{\frac{1.9W_g h_G}{S_a G}}\right), \left(\sqrt{\frac{1.9W_o h_G}{S_{fe} G}}\right) \tag{4.18.1a}$$

(b) For unflanged Configurations c and f, the minimum required thickness of the extension, h_r, shall be calculated in accordance with 4.16.7.2, Step 9(b), for loose type flanges with laps.

(c) For unflanged Configurations d and C and for flanged Configuration d having no bolt loads applied to the extension, the minimum required thickness of the extension, h_r, shall be the maximum of the values determined for each design loading case as follows:

$$h_r = \left(\frac{D_E}{3.2 S_{fe} G}\right) \left| P_s - P_t \right| \tag{4.18.1b}$$

4.18.6 TUBESHEET CHARACTERISTICS

4.18.6.1 Scope. These rules cover the determination of the ligament efficiencies, effective depth of the tube side pass partition groove, and effective elastic constants to be used in the calculation of U-tube, fixed, and floating tubesheets.

4.18.6.2 Conditions of Applicability. The general conditions of applicability given in 4.18.4 apply.

4.18.6.3 Design Considerations.

(a) Elastic moduli and allowable stresses shall be taken at the design temperatures. However, for cases involving thermal loading, it is permitted to use the operating metal temperatures instead of the design temperatures.

(b) When the values calculated in this section are to be used for fixed tubesheets, they shall be determined in both the corroded and uncorroded conditions.

(c) The tube expansion depth ratio given by eq. (4.18.4) may be either calculated or chosen as a constant.

4.18.6.4 Calculation Procedure.

(a) Determination of Effective Dimensions and Ligament Efficiencies - From the geometry (see Figures 4.18.2 and 4.18.3) and material properties of the exchanger, calculate the required parameters in accordance with (1) or (2).

(1) For geometries where the tubes extend through the tubesheet [see Figure 4.18.2, sketch (b)], calculate the following parameters.

$$D_o = 2r_o + d_t \tag{4.18.2}$$

$$\mu = \frac{p - d_t}{p} \tag{4.18.3}$$

$$\rho = \frac{l_{tx}}{h} \qquad \text{where } 0 \le \rho \le 1 \tag{4.18.4}$$

$$d^* = \max\left[\left\{d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right\},\ \left(d_t - 2t_t\right)\right] \tag{4.18.5}$$

$$p^* = p\left(1 - \frac{4 \cdot \min\left[A_L,\ \left(4D_o p\right)\right]}{\pi D_o^{\,2}}\right)^{-0.5} \tag{4.18.6}$$

$$\mu^* = \frac{p^* - d^*}{p^*} \tag{4.18.7}$$

$$h'_g = \max\left[\left(h_g - c_t\right),\ 0.0\right] \tag{4.18.8}$$

(2) For tubes welded to the backside of the tubesheet [see Figure 4.18.2, sketch (d)], calculate the following parameters.

$$D_o = 2r_o + d \tag{4.18.9}$$

$$\mu = \frac{p - d}{p} \tag{4.18.10}$$

$$p^* = p\left(1 - \frac{4 \cdot \min\left[A_L,\ \left(4D_0 p\right)\right]}{\pi D_o^{\,2}}\right)^{-0.5} \tag{4.18.11}$$

$$\mu^* = \frac{p^* - d}{p^*} \tag{4.18.12}$$

$$h'_g = \max\left[\left(h_g - c_t\right),\ 0.0\right] \tag{4.18.13}$$

(b) Determination of Effective Elastic Properties - Determine the values for E^*/E and v^* based on μ^* and h/p using Table 4.18.1 (equilateral triangular pattern) or Table 4.18.2 (square pattern).

4.18.7 RULES FOR THE DESIGN OF U-TUBE TUBESHEETS

4.18.7.1 Scope. These rules cover the design of tubesheets for U-tube heat exchangers. The tubesheet may have one of the six configurations shown in Figure 4.18.4.

(a) Configuration a - tubesheet integral with shell and channel.

(b) Configuration b - tubesheet integral with shell and gasketed with channel, extended as a flange.

(c) Configuration c - tubesheet integral with shell and gasketed with channel, not extended as a flange.

(d) Configuration d - tubesheet gasketed with shell and channel, extended or not extended as a flange.

(e) Configuration e - tubesheet gasketed with shell and integral with channel, extended as a flange.

(f) Configuration f - tubesheet gasketed with shell and integral with channel, not extended as a flange.

4.18.7.2 Conditions of Applicability. The general conditions of applicability given in 4.18.4 apply.

(21) **4.18.7.3 Design Considerations.**

(a) The various loading conditions to be considered shall include, but not be limited to, normal operating, start-up, shutdown, cleaning, and upset conditions, which may govern the design of the tubesheet.

(1) For each of these conditions, the loading cases in Table 4.18.7 shall be considered.

(2) When differential design pressure is specified by the user, the design shall be based only on Loading Case 3. For the design of common elements, P_t and P_s shall be determined as follows:

(-a) If the tube side is the higher-pressure side, P_t shall be the tube-side design pressure and P_s shall be P_t less the differential design pressure.

(-b) If the shell side is the higher-pressure side, P_s shall be the shell-side design pressure and P_t shall be P_s less the differential design pressure.

(3) The designer should take appropriate consideration of the stresses resulting from the pressure test required by 4.1.6.2 and Part 8.

(b) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, shell, and channel are within the maximum permissible stress limits.

(c) The designer may consider the tubesheet as simply supported in accordance with 4.18.7.5.

(21) **4.18.7.4 Calculation Procedure.**

Step 1. Determine D_o, μ, μ^*, and h'_g from 4.18.6.4(a).

Step 2. Calculate the diameter ratios ρ_s and ρ_c using the following equations.

$$\rho_s = \frac{D_s}{D_o} \qquad \text{Configurations a, b, c} \tag{4.18.14}$$

$$\rho_s = \frac{G_s}{D_o} \qquad \text{Configurations d, e, f} \tag{4.18.15}$$

$$\rho_c = \frac{D_c}{D_o} \qquad \text{Configurations a, e, f} \tag{4.18.16}$$

$$\rho_c = \frac{G_c}{D_o} \qquad \text{Configurations b, c, d} \tag{4.18.17}$$

For each loading case, calculate moment M_{TS} due to pressures P_s and P_t acting on the unperforated tubesheet rim.

$$M_{TS} = \frac{D_o{}^2\left[\left(\rho_s - 1\right)\left(\rho_s^2 + 1\right)P_s - \left(\rho_c - 1\right)\left(\rho_c^2 + 1\right)P_t\right]}{16} \tag{4.18.18}$$

Step 3. Calculate h/p. Determine E^*/E and v^* using 4.18.6.4(b). For Configurations a, b, c, e, and f, proceed to Step 4. For Configuration d, proceed to Step 5.

Step 4. Calculate the shell coefficients.

(a) Configurations a, b, and c:

$$\beta_s = \frac{\left[12\left(1 - \nu_s^2\right)\right]^{0.25}}{\left[\left(D_s + t_s\right)t_s\right]^{0.5}} \tag{4.18.19}$$

$$k_s = \frac{\beta_s E_s t_s^3}{6\left(1 - \nu_s^2\right)} \tag{4.18.20}$$

$$\lambda_s = \frac{6\ D_s k_s}{h^3}\left(1 + h\beta_s + \frac{h^2\beta_s^2}{2}\right) \tag{4.18.21}$$

$$\delta_s = \frac{D_s^2}{4E_s\ t_s}\left(1 - \frac{\nu_s}{2}\right) \tag{4.18.22}$$

$$\omega_s = \rho_s k_s \beta_s \delta_s \left(1 + h\beta_s\right) \tag{4.18.23}$$

(b) Configurations a, e, and f:

$$\beta_c = \frac{\left[12\left(1 - \nu_c^2\right)\right]^{0.25}}{\left[\left(D_c + t_c\right)t_c\right]^{0.5}} \tag{4.18.24}$$

$$k_c = \frac{\beta_c E_c t_c^3}{6\left(1 - \nu_c^2\right)} \tag{4.18.25}$$

$$\lambda_c = \frac{6D_c k_c}{h^3}\left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right) \tag{4.18.26}$$

$$\delta_c = \frac{D_c^2}{4E_c\, t_c}\left(1 - \frac{\nu_c}{2}\right) \qquad \text{for a cylinder} \tag{4.18.27}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(\frac{1 - \nu_c}{2}\right) \qquad \text{for a hemispherical head} \tag{4.18.28}$$

$$\omega_c = \rho_c k_c \beta_c \delta_c\ \left(1 + h\,\beta_c\right) \tag{4.18.29}$$

Step 5. Calculate the diameter ratio, K, and the coefficient F.

$$K = \frac{A}{D_o} \tag{4.18.30}$$

$$F = \frac{(1 - \nu^*)\left(\lambda_s + \lambda_c + E\ln K\right)}{E^*} \qquad \text{Configuration a} \tag{4.18.31}$$

$$F = \frac{(1 - \nu^*)(\lambda_s + E\ln K)}{E^*} \qquad \text{Configurations b, c} \tag{4.18.32}$$

$$F = \frac{\left(1 - \nu^*\right)\left(E\ln K\right)}{E^*} \qquad \text{Configuration d} \tag{4.18.33}$$

$$F = \frac{\left(1 - v^*\right)\left(\lambda_c + E \ln K\right)}{E^*} \qquad \text{Configurations e, f} \tag{4.18.34}$$

Step 6. For each loading case, calculate the moment M^* acting on the unperforated tubesheet rim.

$$M^* = M_{TS} + \omega_c P_t - \omega_s P_s \qquad \text{Configuration a} \tag{4.18.35}$$

$$M^* = M_{TS} - \omega_s P_s - \frac{\left(C - G_c\right) W^*}{2\pi D_o} \qquad \text{Configuration b} \tag{4.18.36}$$

$$M^* = M_{TS} - \omega_s P_s - \frac{\left(G_1 - G_c\right) W^*}{2\pi D_o} \qquad \text{Configuration c} \tag{4.18.37}$$

$$M^* = M_{TS} + \frac{\left(G_c - G_s\right) W^*}{2\pi D_o} \qquad \text{Configuration d} \tag{4.18.38}$$

$$M^* = M_{TS} + \omega_c P_t + \frac{\left(C - G_s\right) W^*}{2\pi D_o} \qquad \text{Configuration e} \tag{4.18.40}$$

$$M^* = M_{TS} + \omega_c P_t + \frac{\left(G_1 - G_s\right) W^*}{2\pi D_o} \qquad \text{Configuration f} \tag{4.18.41}$$

Step 7. For each loading case, calculate the maximum bending moments acting on the tubesheet at the periphery, M_P, and at the center, M_o.

$$M_P = \frac{M^* - \dfrac{D_o^2 F\left(P_s - P_t\right)}{32}}{1 + F} \tag{4.18.42}$$

$$M_o = M_p + \frac{D_o^2\left(3 + v^*\right)\left(P_s - P_t\right)}{64} \tag{4.18.43}$$

Determine the maximum bending moment M acting on the tubesheet.

$$M = \max\left[\left|M_o\right|, \left|M_p\right|\right] \tag{4.18.44}$$

Step 8. For each loading case, check the tubesheet bending stress.

(a) Calculate the bending stress.

$$\sigma = \frac{6M}{\mu^*\left(h - h'_g\right)^2} \tag{4.18.45}$$

(b) Acceptance Criteria

If $\sigma \leq 2S$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to Step 1.

Step 9. For each loading case, check the average shear stress in the tubesheet at the outer edge of the perforated region, if required.

(a) Calculate the average shear stress.

If $\left|P_s - P_t\right| \leq \dfrac{4\mu h}{D_o}\min\left[0.8S,\ 0.533S_y\right]$, the shear stress is not required to be calculated; proceed to (c) below.

Otherwise:

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right)\left|P_s - P_t\right| \tag{4.18.47}$$

(b) Acceptance Criteria

If $\tau \le \min(0.8S, 0.533S_y)$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness h and return to Step 1.

(c) For Configurations a, b, c, e, and f, proceed to Step 10. For Configuration d, the calculation procedure is complete.

Step 10. For each loading case, check the stresses in the shell and/or channel integral with the tubesheet.

(a) Configurations a, b, and c - The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{s,m}$, the bending stress, $\sigma_{s,b}$, and total axial stress, σ_s, in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{D_s^2 P_s}{4t_s\left(D_s + t_s\right)} \tag{4.18.48}$$

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s \delta_s P_s + \frac{6\left(1-\nu^*\right)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_s}{2}\right)\left(M_p + \frac{D_o^2}{32}\left(P_s - P_t\right)\right)\right] \tag{4.18.49}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right| \tag{4.18.50}$$

(b) Configurations a, e, and f - A cylindrical channel shall have a uniform thickness of t_c or a minimum length of $1.8\sqrt{D_c t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{c,m}$, the bending stress, $\sigma_{c,b}$, and total axial stress, σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c\left(D_c + t_c\right)} \tag{4.18.51}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c \delta_c P_t - \frac{6\left(1-\nu^*\right)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}\left(P_s - P_t\right)\right)\right] \tag{4.18.52}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right| \tag{4.18.53}$$

(c) Acceptance Criteria

(1) Configuration a - If $\sigma_s \le 1.5S_s$ and $\sigma_c \le 1.5S_c$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to Step 11.

(2) Configurations b and c - If $\sigma_s \le 1.5S_s$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to Step 11.

(3) Configurations e and f - If $\sigma_c \le 1.5S_c$, the channel design is acceptable and the calculation procedure is complete. Otherwise, proceed to Step 11.

Step 11. The design shall be reconsidered. One or a combination of the following three options may be used.

(a) Option 1 - Increase the assumed tubesheet thickness h and return to Step 1.

(b) Option 2 - Increase the integral shell and/or channel thickness as follows, and return to Step 2.

(1) Configurations a, b, and c - If $\sigma_s > 1.5S_s$, increase the shell thickness t_s

(2) Configurations a, e, and f - If $\sigma_c > 1.5S_c$, increase the channel thickness t_c.

(c) Option 3 -Perform a simplified elastic–plastic calculation for each applicable loading case by using a reduced effective modulus for the integral shell and/or channel to reflect the anticipated load shift resulting from plastic action at the integral shell and/or channel-to-tubesheet junction. This may result in a higher tubesheet bending stress, σ. This option shall not be used at temperatures where the time-dependent properties govern the allowable stress.

(1) Configuration a - This option may only be used when $\sigma_s \le S_{PS,\ s}$ and $\sigma_c \le S_{PS,\ c}$. In Step 4, if $\sigma_s > 1.5S_s$, replace E_s with $E_s^* = E_s\sqrt{1.5S_s/\sigma_s}$ and recalculate k_s and λ_s. If $\sigma_c > 1.5S_c$, replace E_c with $E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$ and recalculate k_c and λ_c.

(2) Configurations b and c - This option may only be used when $\sigma_s \le S_{PS,\ s}$. In Step 4, replace E_s with $E_s^* = E_s\sqrt{1.5S_s/\sigma_s}$ and recalculate k_s and λ_s.

(3) Configurations e and f - This option may only be used when $\sigma_c \le S_{PS,\ c}$. In Step 4, replace E_c with $E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$ and recalculate k_c and λ_c.

After making the above changes, perform Steps 5 and 7, and recalculate the tubesheet bending stress σ given in Step 8. If $\sigma \leq 2S$, the assumed tubesheet thickness h is acceptable and the design is complete. Otherwise, the design shall be reconsidered by using Option 1 or 2.

4.18.7.5 Calculation Procedure for Simply Supported U-Tube Tubesheets.

4.18.7.5.1 Scope. This procedure describes how to use the rules of 4.18.7.4 when the effect of the stiffness of the integral channel and/or shell is not considered.

4.18.7.5.2 Conditions of Applicability. This calculation procedure applies only when the tubeheet is integral with the shell or channel (Configurations a, b, c, e, and f).

4.18.7.5.3 Calculation Procedure. The calculation procedure outlined in 4.18.7.4 shall be performed accounting for the following modifications.

(a) Perform 4.18.7.4, Steps 1 through 9.

(b) Perform 4.18.7.4, Step 10 except as follows:

(1) The shell (Configuration a, b, and c) is not required to meet a minimum length requirement.

(2) The channel (Configurations a, e, and f) is not required to meet a minimum length requirements.

(3) Acceptance Criteria

(-a) Configuration a: If $\sigma_s \leq S_{PS,\ s}$ and $\sigma_c \leq S_{PS,\ c}$, then the shell and channel are acceptable. Otherwise increase the thickness of the overstressed component(s) (shell and/or channel) and return to , 4.18.7.4, Step 1.

(-b) Configuration b and c: If $\sigma_s \leq S_{PS,\ s}$ then the shell is acceptable. Otherwise increase the thickness of the shell and return to 4.18.7.4, Step 1.

(-c) Configuration e and f: If $\sigma_c \leq S_{PS,\ c}$ then the channel is acceptable. Otherwise increase the thickness of the channel and return to 4.18.7.4, Step 1.

(c) Do not perform 4.18.7.4, Step 11.

(d) Repeat 4.18.7.4, Steps 1 through 8 with the following changes until the tubesheet criteria have been met:

(1) 4.18.7.4, Step 4

(-a) Configurations a, b, and c: $\beta_s = 0$, $k_s = 0$, $\lambda_s = 0$, $\delta_s = 0$

(-b) Configurations a, e, and f: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$.

(2) 4.18.7.4, Step 7 $M = |M_o|$

4.18.8 RULES FOR THE DESIGN OF FIXED TUBESHEETS

4.18.8.1 Scope. These rules cover the design of tubesheets for fixed tubesheet heat exchangers. The tubesheets may have one of the four configurations shown in Figure 4.18.5.

(a) Configuration a - tubesheet integral with shell and channel.

(b) Configuration b - tubesheet integral with shell and gasketed with channel, extended as a flange.

(c) Configuration c - tubesheet integral with shell and gasketed with channel, not extended as a flange.

(d) Configuration d - tubesheet gasketed with shell and channel, extended or not extended as a flange.

4.18.8.2 Conditions of Applicability. The two tubesheets shall have the same thickness, material and edge conditions.

(21) **4.18.8.3 Design Considerations.**

(a) It is generally not possible to determine, by observation, the most severe condition of coincident pressure, temperature and differential thermal expansion. Thus, it is necessary to evaluate all the anticipated loading conditions to ensure that the worst load combination has been considered in the design. The user shall specify all the conditions that may govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, shell, channel, tube-to-tubesheet joint). The loading conditions shall include, but not be limited to, normal operating, start-up, shutdown, cleaning, and upset conditions.

(1) For each of these conditions, the following loading cases shall be considered to determine the effective pressure P_e to be used in the design equations:

(-a) Design Loading Cases. Table 4.18.7 provides the load combinations required to evaluate the heat exchanger for the design condition.

(-b) Operating Condition Cases. Table 4.18.8 provides the load combinations required to evaluate the heat exchanger for each operating condition x.

(2) When differential design pressure is specified by the user, the design shall be based only on design loading case 3 and operating loading cases 3 and 4 for each specified operating condition. If the tube side is the higher-pressure side, P_t shall be the tube side design pressure and P_s shall be P_t less the differential design pressure. For the design of common elements, P_t and P_s shall be determined as follows:

(-a) If the tube side is the higher-pressure side, P_t shall be the tube-side design pressure and P_s shall be P_t less the differential design pressure.

(-b) If the shell side is the higher-pressure side, P_s shall be the shell-side design pressure and P_t shall be P_s less the differential design pressure.

For the operating loading cases, the differential pressure and the individual operating pressures shall not exceed the values used for design

(3) The designer should take appropriate consideration of the stresses resulting from the pressure test required by 4.1.6.2 and Part 8.

(b) The elastic moduli, yield strengths, and allowable stresses shall be taken at the design temperatures for the design loading cases and may be taken at the operating metal temperature of the component under consideration for operating condition x.

(c) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits and that the resulting tube-to-tubesheet joint load is acceptable.

(d) Because any increase of tubesheet thickness may lead to overstresses in the tubes, shell, channel, or tube-to-tubesheet joint, a final check shall be performed, using in the equations the nominal thickness of tubesheet, tubes, shell, and channel, in both corroded and uncorroded conditions.

(e) The designer shall consider the effect of radial differential thermal expansion between the tubesheet and integral shell or channel (Configurations a, b, and c) in accordance with 4.18.8.7.

(f) The designer may consider the tubesheet as simply supported in accordance with 4.18.8.8.

4.18.8.4 Calculation Procedure. (21)

Step 1. Determine D_o, μ, μ^*, and h'_g from 4.18.6.4(a). For the operating loading cases, $h'_g = 0$. Calculate the following quantities.

$$a_o = \frac{D_o}{2} \tag{4.18.54}$$

$$a_c = \frac{D_c}{2} \quad \text{Configuration a} \tag{4.18.55}$$

$$a_c = \frac{G_c}{2} \quad \text{Configurations b, c, d} \tag{4.18.56}$$

$$a_s = \frac{D_s}{2} \quad \text{Configurations a, b, c} \tag{4.18.57}$$

$$a_s = \frac{G_s}{2} \quad \text{Configuration d} \tag{4.18.58}$$

$$\rho_s = \frac{a_s}{a_o} \tag{4.18.59}$$

$$\rho_c = \frac{a_c}{a_o} \tag{4.18.60}$$

$$x_s = 1 - N_t\left(\frac{d_t}{2a_o}\right)^2 \tag{4.18.61}$$

$$x_t = 1 - N_t\left(\frac{d_t - 2t_t}{2a_o}\right)^2 \tag{4.18.62}$$

Step 2. Calculate the following parameters.

(a) The shell axial stiffness and the tube axial stiffness.

$$K_s = \frac{\pi \ t_s(D_s + t_s)\ E_s}{L} \tag{4.18.63}$$

$$K_t = \frac{\pi \ t_t(d_t - t_t)\ E_t}{L} \tag{4.18.64}$$

(b) The stiffness factors.

$$K_{s,t} = \frac{K_s}{N_t K_t} \tag{4.18.65}$$

$$J = \frac{K_j}{K_J + K_S} \tag{4.18.66}$$

(c) The shell coefficients for Configurations a, b, and c.

$$\beta_s = \frac{\left[12\left(1 - \nu_s^2\right)\right]^{0.25}}{\left[\left(D_s + t_s\right) t_s\right]^{0.5}} \tag{4.18.67}$$

$$k_s = \frac{\beta_s E_s t_s^3}{6\left(1 - \nu_s^2\right)} \tag{4.18.68}$$

$$\lambda_s = \frac{6\ D_s k_s}{h^3}\left(1 + h\beta_s + \frac{h^2\beta_s^2}{2}\right) \tag{4.18.69}$$

$$\delta_s = \frac{D_s^2}{4E_s\, t_s}\left(1 - \frac{\nu_s}{2}\right) \tag{4.18.70}$$

For Configuration d, $\beta_s = k_s = \lambda_s = \delta_s = 0$.

(d) The channel coefficients for Configuration a.

$$\beta_c = \frac{\left[12\ \left(1 - \nu_c^2\right)\right]^{0.25}}{\left[\left(D_c + t_c\right)t_c\right]^{0.5}} \tag{4.18.71}$$

$$k_c = \frac{\beta_c E_c t_c^3}{6\left(1 - \nu_c^2\right)} \tag{4.18.72}$$

$$\lambda_c = \frac{6\ D_c k_c}{h^3}\left(1 + h\beta_c + \frac{h^2{\beta_c}^2}{2}\right) \tag{4.18.73}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(1 - \frac{\nu_c}{2}\right) \qquad \text{for a cylinder} \tag{4.18.74}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(\frac{1 - \nu_c}{2}\right) \qquad \text{for a hemispherical head} \tag{4.18.75}$$

For Configurations b, c, and d, $\beta_c = k_c = \lambda_c = \delta_c = 0$.

Step 3. Calculate h/p. Determine E^*/E and v^* using 4.18.6.4(b). Calculate X_a.

$$X_a = \left[\frac{24\left(1 - (v^*)^2\right) N_t E_t t_t \left(d_t - t_t\right) a_o^2}{E^* L h^3} \right]^{0.25} \tag{4.18.76}$$

Using the calculated value of X_a enter either Table 4.18.3 or Figure 4.18.6 to determine Z_d, Z_v, Z_w, and Z_m.

Step 4. Calculate the following parameters

$$K = \frac{A}{D_o} \tag{4.18.77}$$

$$F = \frac{(1 - v^*)\left(\lambda_s + \lambda_c + E \ln K\right)}{E^*} \tag{4.18.78}$$

$$\Phi = \left(1 + v^*\right)F \tag{4.18.79}$$

$$Q_1 = \frac{\rho_s - 1 - \Phi Z_v}{1 + \Phi Z_m} \tag{4.18.80}$$

$$Q_{z1} = \frac{(Z_d + Q_1 Z_w)\ X_a^4}{2} \tag{4.18.81}$$

$$Q_{z2} = \frac{(Z_v + Q_1 Z_m)\ X_a^4}{2} \tag{4.18.82}$$

$$U = \frac{\left[Z_w + \left(\rho_s - 1\right)Z_m\right]X_a^4}{1 + \Phi Z_m} \tag{4.18.83}$$

Step 5. Calculate the following quantities.

(a) γ for the operating loading cases. For the design loading cases, $\gamma = 0$.

$$\gamma = \left[\alpha_{t,m}\left(T_{t,m} - T_a\right) - \alpha_{s,m}\left(T_{s,m} - T_a\right)\right]L \tag{4.18.84}$$

(b) ω_s, $\omega_s{}^*$, ω_c, and $\omega_c{}^*$.

$$\omega_s = \rho_s k_s \beta_s \delta_s \left(1 + h\beta_s\right) \tag{4.18.85}$$

$$\omega_s{}^* = \frac{a_o^2\left(\rho_s^2 - 1\right)\left(\rho_s - 1\right)}{4} - \omega_s \tag{4.18.86}$$

$$\omega_c = \rho_c k_c \beta_c \delta_c \left(1 + h\beta_c\right) \tag{4.18.87}$$

$$\omega_c{}^* = a_o^2\left[\frac{\left(\rho_c^2 + 1\right)\left(\rho_c - 1\right)}{4} - \frac{\left(\rho_s - 1\right)}{2}\right] - \omega_c \tag{4.18.88}$$

(c) γ_b

$$\gamma_b = 0 \qquad \text{Configuration a} \tag{4.18.89}$$

$$\gamma_b = \frac{G_c - C}{D_o} \qquad \text{Configuration b} \tag{4.18.90}$$

$$\gamma_b = \frac{G_c - G_1}{D_o} \qquad \text{Configuration c} \tag{4.18.91}$$

$$\gamma_b = \frac{G_c - G_s}{D_o} \qquad \text{Configuration d} \tag{4.18.92}$$

Step 6. For each loading case, calculate the effective pressure.

$$P_e = \frac{JK_{s,t}\left(P_s' - P_t' + P_\gamma + P_W + P_{\text{rim}}\right)}{1 + JK_{s,t}\left[Q_{Z1} + \left(\rho_s - 1\right)Q_{Z2}\right]} \tag{4.18.93}$$

where

$$P_s' = \left(x_s + 2\left(1 - x_s\right)\nu_t + \frac{2}{K_{s,t}}\left(\frac{D_s}{D_o}\right)^2 \nu_s - \frac{\rho_s^2 - 1}{JK_{s,t}} - \frac{\left(1 - J\right)}{2JK_{s,t}}\frac{\left[D_J^2 - D_s^2\right]}{D_o^2}\right)P_s \tag{4.18.94}$$

$$P_t' = \left(x_t + 2\left(1 - x_t\right)\nu_t + \frac{1}{JK_{s,t}}\right)P_t \tag{4.18.95}$$

$$P_\gamma = \frac{N_t K_t \gamma}{\pi \ a_o^2} \tag{4.18.96}$$

$$P_W = -\frac{U\gamma_b W^*}{2\pi a_o^2} \tag{4.18.97}$$

$$P_{\text{rim}} = -\frac{U\left(\omega_s^* P_s - \omega_c^* P_t\right)}{a_o^2} \tag{4.18.98}$$

Step 7. For each loading case check the bending stress.

(a) Calculate Q_2

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t\right) + \frac{\gamma_b W^*}{2\pi}}{1 + \Phi Z_m} \tag{4.18.99}$$

(b) Calculate the tubesheet bending stress

(1) If $P_e \neq 0$, calculate Q_3

$$Q_3 = Q_1 + \frac{2Q_2}{P_e a_o^2} \tag{4.18.100}$$

For each loading case, determine coefficient F_m from either Table 4.18.3 or Figures 4.18.7 and 4.18.8. Calculate the tubesheet maximum bending stress

$$\sigma = \left(\frac{1.5F_m}{\mu^*}\right)\left(\frac{2a_o}{h - h_g'}\right)^2 P_e \tag{4.18.101}$$

(2) If $P_e = 0$, calculate the tubesheet maximum bending stress

$$\sigma = \frac{6Q_2}{\mu^*\left(h - h_g'\right)^2} \tag{4.18.102}$$

(c) Acceptance Criteria

For the design loading cases, if $|\sigma| \leq 1.5S$, and for the operating loading cases, if $|\sigma| \leq S_{PS}$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to Step 1.

Step 8. For each loading case, check the average shear stress in the tubesheet at the outer edge of the perforated region, if required.

(a) Calculate the average shear stress.

If $\left| P_e \right| \leq \frac{2\mu h}{a_o} \min \left[0.8S,\ 0.533S_y \right]$, the shear stress is not required to be calculated; proceed to Step 9. Otherwise:

$$\tau = \left(\frac{1}{4\mu} \right) \left(\frac{1}{h} \left\{ \frac{4A_p}{C_p} \right\} \right) P_e \tag{4.18.104}$$

(b) Acceptance Criteria

If $|\tau| \leq \min(0.8S, 0.533S_y)$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness h and return to Step 1.

Step 9. Check the tube stress and tube-to-tubesheet joint design for each loading case.

(a) Check the axial tube stress.

(1) For each loading case, determine coefficients $F_{t,min}$ and $F_{t,max}$ from Table 4.18.4 and calculate the two extreme values of tube stress, $\sigma_{t,1}$ and $\sigma_{t,2}$, $\sigma_{t,1}$ and $\sigma_{t,2}$ may be positive or negative.

When $P_e \neq 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s} \left[\left(P_s x_s - P_t x_t \right) - P_e F_{t,\min} \right] \tag{4.18.105}$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s} \left[\left(P_s x_s - P_t x_t \right) - P_e F_{t,\max} \right] \tag{4.18.106}$$

When $P_e = 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s} \left[\left(P_s x_s - P_t x_t \right) - \frac{2Q_2}{a_o^2} F_{t,\min} \right] \tag{4.18.107}$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s} \left[\left(P_s x_s - P_t x_t \right) - \frac{2Q_2}{a_o^2} F_{t,\max} \right] \tag{4.18.108}$$

(2) Determine $\sigma_{t,\max}$

$$\sigma_{t,\max} = \max \left[\left| \sigma_{t,1} \right|, \left| \sigma_{t,2} \right| \right] \tag{4.18.109}$$

(b) Acceptance Criteria

For the design loading cases, if $\sigma_{,\max} > S$, and for the operating loading cases, if $\sigma_{\max} > 2S$, reconsider the tube design and return to Step 1.

Otherwise, proceed to (c).

(c) Check the tube-to-tubesheet joint design.

(1) Calculate the largest tube-to-tubesheet joint load, W_t

$$W_t = \sigma_{t,\max} \pi \left(d_t - t_t \right) t_t \tag{4.18.110}$$

(2) Determine the maximum allowable load for the tube-to-tubesheet joint design, $L_{\max}$. For tube-to-tubesheet joints with full-strength welds, $L_{\max}$ shall be determined in accordance with 4.21.2.2. For tube-to-tubesheet joints with partial-strength welds, $L_{\max}$ shall be determined in accordance with 4.21.2.3 or 4.21.3, as applicable. For all other tube joints, $L_{\max}$ shall be determined in accordance with 4.21.3.

(3) Acceptance Criteria

If $W_t > L_{\max}$, tube-to-tubesheet joint design shall be reconsidered.

If $W_t \leq L_{\max}$, tube-to-tubesheet joint design is acceptable. Proceed to (d).

(d) If $\sigma_{t,1}$ or $\sigma_{t,2}$ is negative, proceed to (f).

(e) If $\sigma_{t,1}$ and $\sigma_{t,2}$ are positive, the tube design is acceptable. Proceed to Step 10.

(f) Check the tubes for buckling.

(1) Calculate the largest equivalent unsupported buckling length of the tube l_t considering the unsupported tube spans l and their corresponding method of support defined by the parameter k.

$$l_t = kl \tag{4.18.111}$$

(2) Determine the maximum permissible buckling stress limit for the tubes.

$$S_{tb} = \min\left[\left\{\frac{\pi^2 E_t}{F_s F_t^2}\right\}, S_t\right] \qquad C_t \le F_t \tag{4.18.112}$$

$$S_{tb} = \min\left[\left\{\frac{S_{y,t}}{F_s}\left(1 - \frac{F_t}{2\ C_t}\right)\right\}, S_t\right] \qquad C_t > F_t \tag{4.18.113}$$

where

$$C_t = \sqrt{\frac{2\pi^2 E_t}{S_{y,t}}} \tag{4.18.114}$$

$$F_t = \frac{l_t}{r_t} \tag{4.18.115}$$

$$r_t = \frac{\sqrt{d_t^2 + (d_t - 2t_t)^2}}{4} \tag{4.18.116}$$

When $P_e \neq 0$

$$F_s = \min\left\{\max\left[\left(3.25 - 0.25(Z_d + Q_3 Z_w)X_a^4\right), 1.25\right], 2.0\right\} \tag{4.18.117}$$

When $P_e = 0$

$$F_s = 1.25 \tag{4.18.118}$$

(3) Determine $\sigma_{t,\min}$

$$\sigma_{t,\min} = \min\left[\sigma_{t,1}, \sigma_{t,2}\right] \tag{4.18.119}$$

(4) Acceptance Criteria

If $|\sigma_{t,\min}| > S_{tb}$, reconsider the tube design and return to Step 1.

If $|\sigma_{t,\min}| \le S_{tb}$ the tube design is acceptable. Proceed to Step 10.

Step 10. Perform this Step for each loading case.

(a) Calculate the axial membrane stress $\sigma_{s,m}$ in each different shell section. For shell sections integral with the tubesheet having a different material and/or thickness than the shell, refer to 4.18.8.5.

$$\sigma_{s,m} = \frac{a_o^2\left[P_e + \left(\rho_s^2 - 1\right)(P_s - P_t)\right]}{(D_s + t_s)t_s} + \frac{a_s^2 P_t}{(D_s + t_s)t_s} \tag{4.18.120}$$

(b) Acceptance Criteria

(1) For the design loading cases, if $|\sigma_{s,m}| > S_s E_{s,w}$, and for the operating loading cases, if $|\sigma_{s,m}| > S_{PS,s}$ reconsider the shell design and return to Step 1.

(2) If $\sigma_{s,m}$ is negative, proceed to (c) below.

(3) If $\sigma_{s,m}$ is positive, the shell design is acceptable. Configurations a, b, and c: Proceed to Step 11. Configuration d: the calculation procedure is complete.

(c) Determine the maximum allowable longituidinal compressive stress, $S_{s,b}$

(1) If $|\sigma_{s,m}| > S_{s,b}$, reconsider the shell design and return to Step 1

(2) If $|\sigma_{s,m}| \le S_{s,b}$, the shell design is acceptable. Configuration a, b, and c: Proceed to Step 11. Configuration d: The calculation procedure is complete.

Step 11. For each loading case, check the stresses in the shell and/or channel when integral with the tubesheet (Configurations a, b, and c).

(a) Shell Stresses (Configurations a, b and c) - The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{s,m}$, the bending stress, $\sigma_{s,b}$, and total axial stress, σ_s, in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{a_o^2\left[P_e + \left(\rho_s^2 - 1\right)\left(P_s - P_t\right)\right]}{\left(D_s + t_s\right)t_s} + \frac{a_s^2 P_t}{\left(D_s + t_s\right)t_s} \tag{4.18.121}$$

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s \delta_s P_s + \frac{6\left(1 - \left(\nu^*\right)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_s}{2}\right)H\right] \tag{4.18.122}$$

$$H = P_e\left(Z_v + Z_m Q_1\right) + \frac{2Z_m Q_2}{a_o^2} \tag{4.18.123}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right| \tag{4.18.124}$$

(b) Channel Stresses (Configuration a) - When the channel is cylindrical, it shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{c,m}$, the bending stress, $\sigma_{c,b}$, and total axial stress, σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{a_c^2 P_t}{\left(D_c + t_c\right)t_c} \tag{4.18.125}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c \delta_c P_t - \frac{6\left(1 - \left(\nu^*\right)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)H\right] \tag{4.18.126}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right| \tag{4.18.127}$$

(c) Acceptance Criteria

(1) Configuration a - For the design loading cases, if $\sigma_s \le 1.5S_s$ and $\sigma_c \le 1.5S_c$, and for the operating loading cases, if $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to Step 12.

(2) Configurations b and c - For the design loading cases, if $\sigma_s \le 1.5S_s$, and for the operating loading cases, if $\sigma_s \le S_{PS,s}$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to Step 12.

Step 12. The tubesheet design shall be reconsidered. One or a combination of the following three options may be used.

(a) Option 1 - Increase the assumed tubesheet thickness h and return to Step 1.

(b) Option 2 - Increase the integral shell and/or channel thickness as follows and return to Step 1.

(1) Configurations a, b, and c - If $\sigma_s > 1.5S_s$, increase the shell thickness t_s. It is permissible to increase the shell thickness adjacent to the tubesheet only (see Figure 4.18.9.)

(2) Configuration a - If $\sigma_c > 1.5S_c$, increase the channel thickness t_c.

(c) Option 3 - Perform the elastic–plastic calculation procedure as defined in 4.18.8.6 only when the conditions of applicability stated in 4.18.8.6(b) are satisfied.

4.18.8.5 Calculation Procedure for Effect of Different Shell Material and Thickness Adjacent to the Tubesheet.

(a) Scope

(1) This procedure describes how to use the rules of 4.18.8.4 when the shell has a different thickness and/or a different material adjacent to the tubesheet (see Figure 4.18.9).

(2) Use of this procedure may result in a smaller tubesheet thickness and should be considered when optimization of the tubesheet thickness or shell stress is desired.

(b) Conditions of Applicability - This calculation procedure applies only when the shell is integral with the tubesheet (Configurations a, b, and c).

(c) Calculation Procedure - The calculation procedure outlined in 4.18.8.4 shall be performed with the following modifications.

(1) The shell shall have a thickness of $t_{s,1}$ for a minimum length of $1.8\sqrt{D_s t_{s,1}}$ adjacent to the tubesheets.

(2) In 4.18.8.4, Step 2, replace the equation for K_s with

$$K_s{}^* = \frac{\pi\left(D_s + t_s\right)}{\left(\frac{L - l_1 - l_1'}{E_s t_s}\right) + \left(\frac{l_1 + l_1'}{E_{s,1} t_{s,1}}\right)} \tag{4.18.128}$$

Calculate $K_{s,t}$ and J, replacing K_s with $K_s{}^*$, and calculate β_s, k_s, and δ_s, replacing t_s with $t_{s,1}$ and E_s with $E_{s,1}$.

(3) In 4.18.8.4, Step 5, replace the equation for γ with

$$\gamma^* = \left(T_{t,m} - T_a\right)\alpha_{t,m}L - \left(T_{s,m} - T_a\right)\left[\alpha_{s,m}\left(L - l_1 - l_1'\right) + \alpha_{s,m,1}\left(l_1 + l_1'\right)\right] \tag{4.18.129}$$

(4) In 4.18.8.4, Step 6, calculate P_γ, replacing γ with γ^*.

(5) In 4.18.8.4, Step 10, calculate $\sigma_{s,m}$, replacing t_s with $t_{s,1}$. Replace S with $S_{s,1}$, and $S_{s,b}$ with $S_{s,b,1}$.

(6) In 4.18.8.4, Step 11, calculate $\sigma_{s,m}$ and $\sigma_{s,b}$, replacing t_s with $t_{s,1}$, and E with $E_{s,1}$. Replace S_s with $S_{s,1}$, and $S_{PS,s}$ with $S_{SP,s,1}$.

(7) If the elastic–plastic calculation procedure of 4.18.8.6 is being performed, replace $S_{y,s}$ with $S_{y,s,1}$, $S_{PS,s}$ with $S_{SP,s,1}$, and E with $E_{s,1}$ in this calculation.

(8) If the radial thermal expansion procedure of 4.18.8.7 is being performed, replace t_s with $t_{s,1}$, and E with $E_{s,1}$ in this calculation.

4.18.8.6 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint.

(a) Scope - This procedure describes how to use the rules of 4.18.8.4 when the effect of plasticity at the shell-tubesheet and/or channel-tubesheet joint is to be considered.

(1) If the discontinuity stresses at the shell-tubesheet and/or channel-tubesheet joint exceed the allowable stress limits, the thickness of the shell, channel, or tubesheet may be increased to meet the stress limits given in 4.18.8.4. As an alternative, when the calculated tubesheet stresses are within the allowable stress limits, but either or both of the calculated shell or channel total stresses exceed their allowable stress limits, one additional “elastic–plastic solution” calculation may be performed.

(2) This calculation permits a reduction of the shell and/or channel modulus of elasticity, where it affects the rotation of the joint, to reflect the anticipated load shift resulting from plastic action at the joint. The reduced effective modulus has the effect of reducing the shell and/or channel stresses in the elastic–plastic calculation; however, due to load shifting this usually leads to an increase in the tubesheet stress. In most cases, an elastic–plastic calculation using the appropriate reduced shell or channel modulus of elasticity results in a design where the calculated tubesheet stresses are within the allowable stress limits.

(b) Conditions of Applicability

(1) This procedure shall not be used at temperatures where the time-dependent properties govern the allowable stress.

(2) This procedure applies only for the design loading cases.

(3) This procedure applies to Configuration a when $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$.

(4) This procedure applies to Configurations b and c when $\sigma_s \le S_{PS,s}$.

(5) This procedure may only be used once for each iteration of tubesheet, shell, and channel thickness and change of materials.

(c) Calculation Procedure - After the calculation procedure given in 4.18.8.4 (Steps 1 through 11) has been performed for the elastic solution, an elastic–plastic calculation using the referenced steps from 4.18.8.4 shall be performed in accordance with the following procedure for each applicable loading case. Except for those quantities modified below, the quantities to be used for the elastic–plastic calculation shall be the same as those calculated for the corresponding elastic loading case.

(1) Define the maximum permissible bending stress limit in the shell and channel.

$$S_s{}^* = \min\left[S_{y,s}, \left(\frac{S_{PS,s}}{2}\right)\right] \qquad \text{Configurations a, b, c} \tag{4.18.130}$$

$$S_c^* = \min\left[S_{y,c}, \left(\frac{S_{PS,c}}{2}\right)\right] \quad \text{Configuration a} \quad (4.18.131)$$

(2) Using bending stresses $\sigma_{s,b}$ and $\sigma_{c,b}$ calculated in Step 11 for the elastic solution, determine fact_s and fact_c as follows.

$$\text{fact}_s = \min\left[\left(1.4 - \frac{0.4\left|\sigma_{s,b}\right|}{S_s^*}\right), 1.0\right] \quad \text{Configurations a, b, c} \quad (4.18.132)$$

$$\text{fact}_c = \min\left[\left(1.4 - \frac{0.4\left|\sigma_{c,b}\right|}{S_c^*}\right), 1.0\right] \quad \text{Configuration a} \quad (4.18.133)$$

(3) For Configuration a, if $\text{fact}_s = 1.0$ and $\text{fact}_c = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (4). For Configurations b and c, if $\text{fact}_s = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (4).

(4) Calculate reduced values of E_s and E_c as follows:

$$E_s^* = E_s \cdot \text{fact}_s \quad \text{Configurations a, b, c} \quad (4.18.134)$$

$$E_c^* = E_c \cdot \text{fact}_c \quad \text{Configuration a} \quad (4.18.135)$$

(5) In 4.18.8.4, Step 2, recalculate k_s, and λ_s by replacing E_s with $E_s{}^*$, and k_c and λ_c by replacing E_c with $E_c{}^*$.

(6) In 4.18.8.4, Step 4, recalculate F, Φ, Q_1, Q_{Z1}, Q_{Z2}, and U.

(7) In 4.18.8.4, Step 6, recalculate P_W, P_{rim}, and P_e.

(8) In 4.18.8.4, Step 7, recalculate Q_2, Q_3, and F_m, as applicable, and the tubesheet bending stress σ. If $|\sigma| \le 1.5S$, the design is acceptable and the calculation procedure is complete. Otherwise, the unit geometry shall be reconsidered.

4.18.8.7 Calculation Procedure for Effect of Radial Differential Thermal Expansion Adjacent to the Tubesheet.

(a) Scope

(1) This procedure describes how to use the rules of 4.18.8.4 when the effect of radial differential thermal expansion between the tubesheet and integral shell or channel is to be considered.

(2) This procedure shall be used when cyclic or dynamic reactions due to pressure or thermal variations are specified.

(3) This procedure shall be used when specified by the user. The user shall provide the Manufacturer with the data necessary to determine the required tubesheet, channel, and shell metal temperatures.

(4) Optionally, the designer may use this procedure to consider the effect of radial differential thermal expansion even when it is not required by (2) or (3).

(b) Conditions of Applicability - This calculation procedure applies only when the tubesheet is integral with the shell or channel (Configurations a, b, and c).

(c) Calculation Procedure - The calculation procedure outlined in 4.18.8.4 and 4.18.8.5, if applicable, shall be performed only for the operating loading cases, accounting for the following modifications.

Table 4.18.9 provides the load combinations required to evaluate the heat exchanger for each operating condition x.

(1) Determine the average temperature of the unperforated rim T_r.

$$T_r = \frac{T' + T_s' + T_c'}{3} \quad \text{Configuration a} \quad (4.18.136)$$

$$T_r = \frac{T' + T_s'}{2} \quad \text{Configurations b, c} \quad (4.18.137)$$

For conservative values of $P_s{}^*$ and $P_c{}^*$, $T_r = T'$ may be used.

(2) Determine the average temperature of the shell $T_s{}^*$ and channel $T_c{}^*$ at their junction to the tubesheet using the equations shown below.

$$T_s^* = \frac{T_s' + T_r}{2} \quad \text{Configurations a, b, c} \quad (4.18.138)$$

$$T_c^* = \frac{T_c' + T_r}{2} \quad \text{Configuration a} \tag{4.18.139}$$

For conservative values of P_s^* and P_c^*, $T_s^* = T_s'$ and $T_c^* = T_c'$ may be used.

(3) Calculate P_s^* and P_c^*.

$$P_s^* = \frac{E_s t_s \left[\alpha_s'\left(T_s^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]}{a_s} \quad \text{Configurations a, b, c} \tag{4.18.140}$$

$$P_c^* = \frac{E_c t_c \left[\alpha_c'\left(T_c^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]}{a_c} \quad \text{Configuration a} \tag{4.18.141}$$

$$P_c^* = 0.0 \quad \text{Configurations b, c} \tag{4.18.142}$$

(4) Calculate P_ω.

$$P_\omega = \frac{U\left(\omega_s P_s^* - \omega_c P_c^*\right)}{a_o^2} \tag{4.18.143}$$

(5) In 4.18.8.4, Step 6, replace the equation for P_e with:

$$P_e = \frac{J K_{s,t}\left(P_s' - P_t' + P_\gamma + P_\omega + P_W + P_{rim}\right)}{1 + J K_{s,t}\left[Q_{Z1} + \left(\rho_s - 1\right) Q_{Z2}\right]} \tag{4.18.144}$$

(6) In 4.18.8.4, Step 7, replace the equation for Q_2 with:

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t\right) - \left(\omega_s P_s^* - \omega_c P_c^*\right) + \dfrac{\gamma_b W^*}{2\pi}}{1 + \Phi Z_m} \tag{4.18.145}$$

(7) In 4.18.8.4, Step 11, replace the equations for $\sigma_{s,b}$ and $\sigma_{c,b}$ with the following equations where H is given by eq. (4.18.122).

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s\left(\delta_s P_s + \frac{a_s^2 P_s^*}{E_s t_s}\right) + \frac{6\left(1 - \nu^{*2}\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_s}{2}\right)H\right] \tag{4.18.146}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\left(\delta_c P_t + \frac{a_c^2 P_c^*}{E_c t_c}\right) - \frac{6\left(1 - \nu^{*2}\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)H\right] \tag{4.18.147}$$

4.18.8.8 Calculation Procedure for Simply Supported Fixed Tubesheets.

4.18.8.8.1 Scope. This procedure describes how to use the rules of 4.18.8.4 when the effect of the stiffness of the integral channel and/or shell is not considered.

4.18.8.8.2 Conditions of Applicability. This calculation applies only when the tubesheet is integral with the shell or channel (Configurations a, b, and c).

4.18.8.8.3 Calculation Procedure. The calculation procedure outlined in 4.18.8.4 shall be performed accounting for the following modifications;

(a) Perform 4.18.8.4, Steps 1 through 10.

(b) Perform 4.18.8.4, Step 11 except as follows:

(1) The shell (Configuration a, b, and c) is not required to meet a minimum length requirement. The shell is exempt from the minimum length requirement in 4.18.8.5(c)(1).

(2) The channel (Configuration a) is not required to meet a minimum length requirement.

(3) Acceptance Criteria

(-a) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, then the shell and channel are acceptable. Otherwise increase the thickness of the overstressed component(s) (shell and/or channel) and return to 4.18.8.4, Step 1.

(-b) Configuration b and c: If $\sigma_s \leq S_{PS,s}$ then the shell is acceptable. Otherwise increase the thickness of the shell and return to 4.18.8.4, Step 1.

(c) Do not perform 4.18.8.4, Step 12

(d) Repeat 4.18.8.4, Steps 1 through 7 for the design loading cases, with the following changes to 4.18.8.4, Step 2 until the tubesheet stress criteria have been met:

(1) Configurations a, b, and c: $\beta_s = 0$, $k_s = 0$, $\lambda_s = 0$, $\delta_s = 0$.

(2) Configuration a: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$.

4.18.8.9 Calculation Procedure for Kettle Shell Exchangers With Fixed Tubesheets.

4.18.8.9.1 Scope. This procedure describes how to use the rules of 4.18.8.4 when an eccentric cone and small cylinder exist between the large shell side cylinder and the tubesheet on both sides.

4.18.8.9.2 Conditions of Applicability.

(a) The two eccentric cones are identical in geometry and material.

(b) The small shell cylinders adjacent to the tubesheet are identical in geometry and material. They shall meet the length requirements of 4.18.8.4, Step 11(a) unless the simply supported rules of 4.18.8.8 are applied. The rules of 4.18.8.5 shall not be used. The rules of 4.18.8.7 may be used only if the length requirements of 4.18.8.4, Step 11(a) are met by the small shell cylinders.

(c) This procedure applies only when $\theta_{ecc} \leq 60$ deg. This procedure accounts for the stiffness and loadings in the shell of the eccentric cones used in the design of the tubesheet. This procedure does not evaluate the acceptability of the shell-to-cone transition. Other requirements in this Division pertaining to shell-to-cone transitions shall be satisfied [e.g., 4.2.5.3(f), 4.3.11, and 4.4.13].

(d) This procedure applies only when

$$0.5 \leq \frac{L_{ecc}}{D_{ecc,S}} \leq 1.5$$

(e) This procedure applies only when $D_{ecc,L} \leq 2.17 D_{ecc,S}$.

(f) These rules assume that an expansion joint, if present, is located in the small shell cylinder.

(g) For cone-to-cylinder junctions, use the following for the design cases (pressure-only cases) in 4.3.11, 4.3.12, 4.4.13, or 4.4.14, as applicable. The cone-to-cylinder junctions do not need to be evaluated for the operating cases (cases including differential thermal expansion).

$$X'_L = \sigma_{ecc,L,m} t_{ecc} \cos\left(\theta_{ecc}\right) - \frac{P_s D_{ecc,L}}{4}$$

$$X'_S = \sigma_{ecc,S,m} t_{ecc} \cos\left(\theta_{ecc}\right) - \frac{P_s D_{ecc,S}}{4}$$

Substitute the following for eq. (4.3.50):

$$X_L = \frac{F_L}{2\pi R_L} \pm \frac{M_L}{\pi R_L^2} + X'_L$$

Substitute the following for eq. (4.3.59):

$$X_S = \frac{F_S}{2\pi R_S} \pm \frac{M_S}{\pi R_S^2} + X'_S$$

4.18.8.9.3 Calculation Procedure. The calculation procedure outlined in 4.18.8.4 shall be performed accounting for the following modifications:

(a) Perform 4.18.8.4, Step 2 with the following changes:

$$K_{ecc} = 0.8\frac{\pi t_{ecc}\left(D_{ecc,S} + t_{ecc}\right)E_{ecc}}{L_{ecc}}$$

$$K_{s,L} = \frac{\pi t_{s,L}\left(D_{s,L} + t_{s,L}\right)E_{s,L}}{L_{s,L}}$$

$$K_s = \frac{\pi t_s\left(D_s + t_s\right)E_s}{L_s}$$

$$K_s^* = \frac{K_s K_{s,L} K_{ecc}}{2K_{ecc}K_{s,L} + 2K_{s,L}K_s + K_s K_{ecc}}$$

$$K_{s,t} = \frac{K_s^*}{N_t K_t}$$

$$J = \frac{1}{1 + \dfrac{K_s^*}{K_J}}$$

(b) Perform 4.18.8.4, Step 5 with the following change:

$$\gamma = \alpha_{t,m}\left(T_{t,m} - T_a\right)L - \alpha_{s,m,L}\left(T_{s,m} - T_a\right)L_{s,L} - 2\alpha_{ecc,m}\left(T_{s,m} - T_a\right)L_{ecc} - 2\alpha_{s,m}\left(T_{s,m} - T_a\right)L_s$$

(c) Perform 4.18.8.4, Step 6 with the following changes; use $\nu_s{}^*$ instead of ν_s:

$$A_s = D_s\left(D_s + t_s\right)$$

$$A_{s,L} = D_{s,L}\left(D_{s,L} + t_{s,L}\right)$$

$$\Delta_{ecc} = D_{ecc,L} - D_{ecc,S}$$

$$\nu_s^* = \frac{2K_s^*}{A_s}\left\{\frac{A_s\nu_s}{K_s} + \left[\nu_{ecc} - \frac{L_{ecc}{}^2 - 2\nu_{ecc}L_{ecc}{}^2 - 3\Delta_{ecc}{}^2}{8L_{ecc}{}^3}\left(\Delta_{ecc}{}^2 + L_{ecc}{}^2\right)^{0.5}\right]\frac{\left(D_{ecc,L} + D_{ecc,S}\right)\left(D_{ecc,S} + t_{ecc}\right)}{5K_{ecc}} + \frac{A_{s,L}\nu_{s,L}}{2K_{s,L}}\right\}$$

(d) Perform 4.18.8.4, Step 10 with the following changes:

(1) Calculate the axial membrane stress for the small cylinder.

$$\sigma_{s,m} = \frac{a_o^2}{t_s\left(D_s + t_s\right)}\left[P_e + \left(\frac{D_s^2}{4a_o^2} - 1\right)\left(P_s - P_t\right)\right] + \frac{D_s^2}{4t_s\left(D_s + t_s\right)}P_t$$

(2) Calculate the axial membrane stress for the eccentric cone at the small end.

$$\sigma_{ecc,S,m} = \frac{a_o^2}{t_{ecc}\left(D_{ecc,S} + t_{ecc}\right)\cos\left(\theta_{ecc}\right)}\left[P_e + \left(\frac{D_{ecc,S}{}^2}{4a_o^2} - 1\right)\left(P_s - P_t\right)\right] + \frac{D_{ecc,S}{}^2}{4t_{ecc}\left(D_{ecc,S} + t_{ecc}\right)\cos\left(\theta_{ecc}\right)}P_t$$

(3) Calculate the axial membrane stress for the eccentric cone at the large end.

$$\sigma_{ecc,L,m} = \frac{a_o^2}{t_{ecc}\left(D_{ecc,L} + t_{ecc}\right)\cos\left(\theta_{ecc}\right)}\left[P_e + \left(\frac{D_{ecc,L}{}^2}{4a_o^2} - 1\right)\left(P_s - P_t\right)\right] + \frac{D_{ecc,L}{}^2}{4t_{ecc}\left(D_{ecc,L} + t_{ecc}\right)\cos\left(\theta_{ecc}\right)}P_t$$

(4) Calculate the axial membrane stress for the large cylinder.

$$\sigma_{s,L,m} = \frac{a_o^2}{t_{s,L}\left(D_{s,L} + t_{s,L}\right)}\left[P_e + \left(\frac{D_{s,L}^{\ 2}}{4a_o^2} - 1\right)\left(P_s - P_t\right)\right] + \frac{D_{s,L}^{\ 2}}{4t_{s,L}\left(D_{s,L} + t_{s,L}\right)}P_t$$

(5) Acceptance Criteria

(-a) Design loading case acceptance criteria: $|\sigma_{s,m}| \leq S_s E_{s,w}$ and $|\sigma_{ecc,S,m}| \leq S_{ecc} E_{ecc,w}$ and $|\sigma_{ecc,L,m}| \leq S_{ecc} E_{ecc,w}$ and $|\sigma_{s,L,m}| \leq S_{s,L} E_{s,L,w}$

(-b) Operating loading case acceptance criteria: $|\sigma_{s,m}| \leq S_{PS,s}$ and $|\sigma_{ecc,S,m}| \leq S_{PS,ecc}$ and $|\sigma_{ecc,L,m}| \leq S_{PS,ecc}$ and $|\sigma_{s,L,m}| \leq S_{PS,s,L}$

(-c) If axial membrane stress is negative (design and operating): $|\sigma_{s,m}| \leq S_{s,b}$ and $|\sigma_{ecc,S,m}| \leq S_{ecc,b}$ and $|\sigma_{ecc,L,m}| \leq S_{ecc,b}$ and $|\sigma_{s,L,m}| \leq S_{s,L,b}$

If the acceptance criteria is not satisfied, reconsider the design of the failing components and return to (a).

4.18.9 RULES FOR THE DESIGN OF FLOATING TUBESHEETS

4.18.9.1 Scope. (21)

(a) These rules cover the design of tubesheets for floating tubesheet heat exchangers that have one stationary tubesheet and one floating tubesheet. Three types of floating tubesheet heat exchangers are covered as shown in Figure 4.18.10.

(1) Sketch (a), immersed floating head;

(2) Sketch (b), externally sealed floating head;

(3) Sketch (c), internally sealed floating tubesheet.

(b) Stationary tubesheets may have one of the six configurations shown in Figure 4.18.11.

(1) Configuration a: tubesheet integral with shell and channel;

(2) Configuration b: tubesheet integral with shell and gasketed with channel, extended as a flange;

(3) Configuration c: tubesheet integral with shell and gasketed with channel, not extended as a flange;

(4) Configuration d: tubesheet gasketed with shell and channel, extended or not extended as a flange;

(5) Configuration e: tubesheet gasketed with shell and integral with channel, extended as a flange;

(6) Configuration f: tubesheet gasketed with shell and integral with channel, not extended as a flange.

(c) Floating tubesheets may have one of the four configurations shown in Figure 4.18.12.

(1) Configuration A: tubesheet integral;

(2) Configuration B: tubesheet gasketed, extended as a flange;

(3) Configuration C: tubesheet gasketed, not extended as a flange;

(4) Configuration D: tubesheet internally sealed.

(d) These rules may be used to design a single-pass floating tubesheet heat exchanger with an immersed floating head, provided the design of the nozzle between the floating head and shell cover accounts for the axial differential thermal expansion.

CAUTION: Addition of the nozzle between the floating head and the shell cover may change the tube force balance, which should be considered in the design.

4.18.9.2 Conditions of Applicability.

The two tubesheets shall have the same thickness and material.

4.18.9.3 Design Considerations. (21)

(a) The calculation shall be performed for the stationary end and for the floating end of the exchanger. Since the edge configurations of the stationary and floating tubesheets are different, the data may be different for each set of calculations. However the conditions of applicability given in 4.18.9.2 must be maintained. For the stationary end, diameters A, C, D_s, D_c, G_s, G_c, G_1 and the thickness t_c shall be taken from Figure 4.18.11. For the floating end, diameters A, C, D_c, G_c and the thickness t_c shall be taken from Figure 4.18.12, and the radial shell dimension a_s shall be taken equal to a_c

(b) It is generally not possible to determine, by observation, the most severe condition of coincident pressure, temperature and radial differential thermal expansion. Thus, it is necessary to evaluate all the anticipated loading conditions to ensure that the worst load combination has been considered in the design. The various loading conditions to be considered shall include the normal operating conditions, the startup conditions, the shutdown conditions, and the upset conditions, which may govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, shell, channel, tube-to-tubesheet joint).

(1) For each of these conditions, the following loading cases shall be considered to determine the effective pressure P_e to be used in the design equations:

(-a) Design Loading Cases. Table 4.18.7 provides load combinations required to evaluate the heat exchanger for the design condition.

(-b) Operating Loading Cases. The operating loading cases are only required when the effect of radial differential thermal expansion is to be considered (see 4.18.9.6).

(2) When differential design pressure is specified by the user, the design shall be based only on design loading case 3 and operating loading cases 3 and 4 for each specified operating condition. For the design of common elements, P_t and P_s shall be determined as follows:

(-a) If the tube side is the higher-pressure side, P_t shall be the tube-side design pressure and P_s shall be P_t less the differential design pressure.

(-b) If the shell side is the higher-pressure side, P_s shall be the shell-side design pressure and P_t shall be P_s less the differential design pressure.

For the operating loading cases, the differential pressure and the individual operating pressures shall not exceed the values used for design.

(3) The designer should take appropriate consideration of the stresses resulting from the pressure test required by 4.1.6.2 and Part 8.

(c) The elastic moduli, yield strengths, and allowable stresses shall be taken at the design temperatures for the design loading cases and may be taken at the operating temperature of the component under consideration for operating condition x.

(d) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits and that the resulting tube-to-tubesheet joint load is acceptable.

(e) The designer shall consider the effect of radial differential thermal expansion between the tubesheet and integral shell or channel (Configurations a, b, c, e, f, and A) in accordance with 4.18.9.6.

(f) The designer may consider the tubesheet as simply supported in accordance with 4.18.9.7.

(21) **4.18.9.4 Calculation Procedure.** The procedure for the design of tubesheets for a floating tubesheet heat exchanger is as follows. Calculations shall be performed for both the stationary tubesheet and the floating tubesheet.

Step 1. Determine D_o, μ, μ^*, and h'_g from 4.18.6.4(a). For the operating loading cases, $h'_g = 0$ Calculate the following quantities.

$$a_o = \frac{D_o}{2} \tag{4.18.148}$$

$$a_s = \frac{D_s}{2} \quad \text{Configurations a, b, c} \tag{4.18.149}$$

$$a_s = \frac{G_s}{2} \quad \text{Configurations d, e, f} \tag{4.18.150}$$

$$a_s = a_c \quad \text{Configurations A, B, C, D} \tag{4.18.151}$$

$$a_c = \frac{D_c}{2} \quad \text{Configurations a, e, f, A} \tag{4.18.152}$$

$$a_c = \frac{G_c}{2} \quad \text{Configurations b, c, d, B, C} \tag{4.18.153}$$

$$a_c = \frac{A}{2} \quad \text{Configuration D} \tag{4.18.154}$$

$$\rho_s = \frac{a_s}{a_o} \tag{4.18.155}$$

$$\rho_c = \frac{a_c}{a_o} \tag{4.18.156}$$

$$x_s = 1 - N_t\left(\frac{d_t}{2a_o}\right)^2 \tag{4.18.157}$$

$$x_t = 1 - N_t\left(\frac{d_t - 2t_t}{2a_o}\right)^2 \tag{4.18.158}$$

Step 2. Calculate the shell and channel coefficients.

(a) The shell coefficients for Configurations a, b and c.

$$\beta_s = \frac{\left[12\left(1 - \nu_s^2\right)\right]^{0.25}}{\left[\left(D_s + t_s\right)t_s\right]^{0.5}} \tag{4.18.159}$$

$$k_s = \frac{\beta_s E_s t_s^3}{6\left(1 - \nu_s^2\right)} \tag{4.18.160}$$

$$\lambda_s = \frac{6D_s k_s}{h^3}\left(1 + h\beta_s + \frac{h^2\beta_s^2}{2}\right) \tag{4.18.161}$$

$$\delta_s = \frac{D_s^2}{4E_s t_s}\left(1 - \frac{\nu_s}{2}\right) \tag{4.18.162}$$

For Configurations d, e, f, A, B, C, and D, $\beta_s = k_s = \lambda_s = \delta_s = 0$.

(b) The channel coefficients for Configurations a, e, f, and A.

$$\beta_c = \frac{\left[12\left(1 - \nu_c^2\right)\right]^{0.25}}{\left[\left(D_c + t_c\right)t_c\right]^{0.5}} \tag{4.18.163}$$

$$k_c = \frac{\beta_c E_c t_c^3}{6\left(1 - \nu_c^2\right)} \tag{4.18.164}$$

$$\lambda_c = \frac{6D_c k_c}{h^3}\left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right) \tag{4.18.165}$$

$$\delta_c = \frac{D_c^2}{4E_c t_c}\left(1 - \frac{\nu_c}{2}\right) \quad \text{for a cylinder} \tag{4.18.166}$$

$$\delta_c = \frac{D_c^2}{4E_c t_c}\left(\frac{1 - \nu_c}{2}\right) \quad \text{for a hemispherical head} \tag{4.18.167}$$

For Configurations b, c, d, B, C, and D, $\beta_c = k_c = \lambda_c = \delta_c = 0$.

Step 3. Calculate h/p. Determine E^*/E and ν^* using 4.18.6.4(b). Calculate X_a.

$$X_a = \left[\frac{24\left(1 - \left(\nu^*\right)^2\right)N_t E_t t_t\left(d_t - t_t\right)a_o^2}{E^* L h^3}\right]^{0.25} \tag{4.18.168}$$

Using the calculated value of X_a enter either Table 4.18.3 or Figure 4.18.6 to determine Z_d, Z_v, Z_w, and Z_m.

Step 4. Calculate the following parameters.

$$K = \frac{A}{D_o} \tag{4.18.169}$$

$$F = \frac{\left(1 - v^*\right)\left(\lambda_s + \lambda_c + E \ln K\right)}{E^*} \tag{4.18.170}$$

$$\Phi = \left(1 + v^*\right)F \tag{4.18.171}$$

$$Q_1 = \frac{\rho_s - 1 - \Phi \, Z_v}{1 + \Phi \, Z_m} \tag{4.18.172}$$

Step 5. Calculate the following quantities.
(a) ω_s, $\omega_s{}^*$, ω_c, and $\omega_c{}^*$.

$$\omega_s = \rho_s k_s \beta_s \delta_s \left(1 + h\beta_s\right) \tag{4.18.173}$$

$$\omega_s^* = \frac{a_o^2 \left(\rho_s^2 - 1\right)\left(\rho_s - 1\right)}{4} - \omega_s \tag{4.18.174}$$

$$\omega_c = \rho_c k_c \beta_c \delta_c \left(1 + h\beta_c\right) \tag{4.18.175}$$

$$\omega_c{}^* = a_o^2 \left[\frac{\left(\rho_c^2 + 1\right)\left(\rho_c - 1\right)}{4} - \frac{\left(\rho_s - 1\right)}{2}\right] - \omega_c \tag{4.18.176}$$

(b) γ_b.

$$\gamma_b = 0.0 \qquad \text{Configurations a, A, D} \tag{4.18.177}$$

$$\gamma_b = \frac{G_c - C}{D_o} \qquad \text{Configurations b, B} \tag{4.18.178}$$

$$\gamma_b = \frac{G_c - G_1}{D_o} \qquad \text{Configurations c, C} \tag{4.18.179}$$

$$\gamma_b = \frac{G_c - G_s}{D_o} \qquad \text{Configuration d} \tag{4.18.180}$$

$$\gamma_b = \frac{C - G_s}{D_o} \qquad \text{Configuration e} \tag{4.18.181}$$

$$\gamma_b = \frac{G_1 - G_s}{D_o} \qquad \text{Configuration f} \tag{4.18.182}$$

Step 6. For each loading case, calculate the effective pressure P_e.
(a) For an exchanger with an immersed floating head [see Figure 4.18.10, sketch (a)]:

$$P_e = P_s - P_t \tag{4.18.183}$$

(b) For an exchanger with an externally sealed floating head [see Figure 4.18.10, sketch (b)]:

$$P_e = P_s\left(1 - \rho_s^2\right) - P_t \tag{4.18.184}$$

(c) For an exchanger with an internally sealed floating tubesheet [see Figure 4.18.10, sketch (c)]:

$$P_e = (P_s - P_t)(1 - \rho_s^2) \tag{4.18.185}$$

Step 7. For each loading case check the bending stress.

(a) Calculate Q_2.

$$Q_2 = \frac{(\omega_s{}^*P_s - \omega_c{}^*P_t) + \frac{\gamma_b W^*}{2\pi}}{1 + \Phi Z_m} \tag{4.18.186}$$

(b) Calculate the tubesheet bending stress.

(1) If $P_e \neq 0$, calculate Q_3.

$$Q_3 = Q_1 + \frac{2Q_2}{P_e a_o^2} \tag{4.18.187}$$

For each loading case, determine coefficient F_m from either Table 4.18.3 or Figures 4.18.7 and 4.18.8. Calculate the maximum tubesheet bending stress.

$$\sigma = \left(\frac{1.5F_m}{\mu^*}\right)\left(\frac{2a_o}{h - h'_g}\right)^2 P_e \tag{4.18.188}$$

(2) If $P_e = 0$, calculate the maximum tubesheet bending stress.

$$\sigma = \frac{6Q_2}{\mu^*(h - h'_g)^2} \tag{4.18.189}$$

(c) Acceptance Criteria

For the design loading cases, if $|\sigma| \leq 1.5S$, and for the operating loading cases, if $|\sigma| \leq S_{PS}$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to Step 1.

For Configurations a, b, c, d, e, and f, proceed to Step 8. For Configuration A, proceed to Step 10. For Configurations B, C, and D, the calculation is complete.

Step 8. For each loading case, check the average shear stress in the tubesheet at the outer edge of the perforated region, if required.

(a) Calculate the average shear stress.

If $\left|P_e\right| \leq \frac{2\mu h}{a_o}\min\left[0.8S,\ 0.533S_y\right]$, the shear stress is not required to be calculated; proceed to Step 9. Otherwise:

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right)P_e$$

(b) Acceptance Criteria

If $\tau \leq \min(0.8S,\ 0.533S_y)$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness h and return to Step 1.

Step 9. Check the tube stress and tube-to-tubesheet joint design for each loading case.

(a) Check the axial tube stress.

(1) For each loading case, determine the coefficients $F_{t,\min}$ and $F_{t,\max}$ from Table 4.18.4 and calculate the two extreme values of tube stress, $\sigma_{t,1}$ and $\sigma_{t,2}$. $\sigma_{t,1}$ and $\sigma_{t,2}$ may be positive or negative.

When $P_e \neq 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[(P_s x_s - P_t x_t) - P_e F_{t,\min}\right] \tag{4.18.192}$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[(P_s x_s - P_t x_t) - P_e F_{t,\max}\right] \tag{4.18.193}$$

When $P_e = 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - \frac{2Q_2}{a_0^2} F_{t,\min}\right] \tag{4.18.194}$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - \frac{2Q_2}{a_0^2} F_{t,\max}\right] \tag{4.18.195}$$

(2) Determine $\sigma_{t,\max}$

$$\sigma_{t,\max} = \max\left[\left|\sigma_{t,1}\right|, \left|\sigma_{t,2}\right|\right] \tag{4.18.196}$$

(b) Acceptance Criteria

For the design loading cases, if $\sigma_{t.\max} > S_t$, and for the operating loading cases, if $\sigma_{t.\max} > 2S_t$ reconsider the design and return to Step 1.

Otherwise, proceed to (c).

(c) Check the tube-to-tubesheet joint design.

(1) Calculate the largest tube-to-tubesheet joint load, W_t

$$W_t = \sigma_{t,\max}\pi\left(d_t - t_t\right)t_t \tag{4.18.197}$$

(2) Determine the maximum allowable load for the tube-to-tubesheet joint design, $L_{\max}$. For tube-to-tubesheet joints with full-strength welds, $L_{\max}$ shall be determined in accordance with 4.21.2.2. For tube-to-tubesheet joints with partial-strength welds, $L_{\max}$ shall be determined in accordance with 4.21.2.3 or 4.21.3, as applicable. For all other tube joints, $L_{\max}$ shall be determined in accordance with 4.21.3.

(3) Acceptance Criteria

If $W_t > L_{\max}$, tube-to-tubesheet joint design shall be reconsidered.

If $W_t \le L_{\max}$, tube-to-tubesheet joint design is acceptable. Proceed to (d).

(d) If $\sigma_{t,1}$ or $\sigma_{2,1}$ is negative, proceed to (f).

(e) If $\sigma_{t,1}$ and $\sigma_{2,1}$ are positive, the tube design is acceptable. Proceed to Step 10.

(f) Check the tubes for buckling.

(1) Calculate the largest equivalent unsupported buckling length of the tube l_t considering the unsupported tube spans l and their corresponding method of support defined by the parameter k.

$$l_t = kl \tag{4.18.198}$$

(2) Determine the maximum permissible buckling stress limit S_{tb} for the tubes.

$$S_{tb} = \min\left[\left\{\frac{\pi^2 E_t}{F_s F_t^2}\right\}, S_t\right] \qquad C_t \le F_t \tag{4.18.199}$$

$$S_{tb} = \min\left[\left\{\frac{S_{y,t}}{F_s}\left(1 - \frac{F_t}{2C_t}\right)\right\}, S_t\right] \qquad C_t > F_t \tag{4.18.200}$$

where

$$F_q = \frac{\left(Z_d + Q_3 Z_v\right)X_a^4}{2} \tag{4.18.201}$$

$$C_t = \sqrt{\frac{2\pi^2 E_t}{S_{y,t}}} \tag{4.18.202}$$

$$F_t = \frac{l_t}{r_t} \tag{4.18.203}$$

$$r_t = \frac{\sqrt{d_t^2 + (d_t - 2t_t)^2}}{4} \tag{4.18.204}$$

When $P_e \neq 0$

$$F_s = \min\left\{\max\left[\left(3.25 - 0.25\left(Z_d + Q_3 Z_w\right)X_a^4\right), 1.25\right], 2.0\right\} \tag{4.18.205}$$

When $P_e = 0$

$$F_s = 1.25 \tag{4.18.206}$$

(3) Determine $\sigma_{t,\min}$

$$\sigma_{t,\min} = \min\left[\sigma_{t,1}, \sigma_{t,2}\right] \tag{4.18.207}$$

(4) Acceptance Criteria

If $|\sigma_{t,\min}| > S_{tb}$, reconsider the tube design and return to Step 1.

If $|\sigma_{t,\min}| \leq S_{tb}$, the tube design is acceptable. Proceed to Step 10.

Step 10. For each loading case, check the stresses in the shell and/or channel integral with the tubesheet.

(a) Configurations a, b, and c - The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{s,m}$, the bending stress, $\sigma_{s,b}$, and total axial stress, σ_s, in the shell at its junction to the tubesheet, where H is given by Eq. (4.18.123).

$$\sigma_{s,m} = \frac{a_o^2\left[P_e + \left(\rho_s^2 - 1\right)\left(P_s - P_t\right)\right]}{\left(D_s + t_s\right)t_s} + \frac{a_s^2 P_t}{\left(D_s + t_s\right)t_s} \tag{4.18.208}$$

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s \delta_s P_s + \frac{6\left(1 - (\nu^*)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_s}{2}\right)H\right] \tag{4.18.209}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right| \tag{4.18.210}$$

(b) Configurations a, e, f, and A - A cylindrical channel shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{c,m}$, the bending stress, $\sigma_{c,b}$, and total axial stress, σ_c, in the channel at its junction to the tubesheet, where H is given by Eq. (4.18.123).

$$\sigma_{c,m} = \frac{a_c^2 P_t}{\left(D_c + t_c\right)t_c} \tag{4.18.211}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c \delta_c P_t - \frac{6\left(1 - (\nu^*)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)H\right] \tag{4.18.212}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right| \tag{4.18.213}$$

(c) Acceptance Criteria

(1) Configuration a - For the design loading cases, if $\sigma_s \leq 1.5S_s$ and $\sigma_c \leq 1.5S_c$, and for the operating loading cases, if $\sigma_s \leq S_{PS,s}$ and $\sigma_c \leq S_{PS,c}$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to Step 11.

(2) Configurations b and c - For the design loading cases, if $\sigma_s \leq 1.5S_s$, and for the operating loading cases, if $\sigma_s \leq S_{PS,s}$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to Step 11.

(3) Configurations e, f, and A - For the design loading cases, if $\sigma_c \leq 1.5S_c$, and for the operating loading cases, if $\sigma_c \leq S_{PS,c}$, the channel design is acceptable and the calculation procedure is complete. Otherwise, proceed to Step 11.

Step 11. The design shall be reconsidered. One or a combination of the following three options may be used.

(a) Option 1 - Increase the assumed tubesheet thickness h and return to Step 1.

(b) Option 2 - Increase the integral shell and/or channel thickness and return to Step 1.

(1) Configurations a, b, and c - If $\sigma_s \leq 1.5S_s$, increase the shell thickness t_s.

(2) Configurations a, e, f, and A - If $\sigma_c \leq 1.5S_c$, increase the channel thickness t_c.

(c) Option 3 - Perform the elastic–plastic calculation procedure as defined in 4.18.9.5 only when the conditions of applicability stated in 4.18.9.5(b) are satisfied.

4.18.9.5 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint.

(a) Scope - This procedure describes how to use the rules of 4.18.9.4 when the effect of plasticity at the shell-tubesheet and/or channel-tubesheet joint is to be considered.

(1) When the calculated tubesheet stresses are within the allowable stress limits, but either or both of the calculated shell or channel total stresses exceed their allowable stress limits, an additional "elastic–plastic solution" calculation may be performed.

(2) This calculation permits a reduction of the shell and/or channel modulus of elasticity, where it affects the rotation of the joint, to reflect the anticipated load shift resulting from plastic action at the joint. The reduced effective modulus has the effect of reducing the shell and/or channel stresses in the elastic–plastic calculation; however, due to load shifting this usually leads to an increase in the tubesheet stress. In most cases, an elastic–plastic calculation using the appropriate reduced shell or channel modulus of elasticity results in a design where the calculated tubesheet stresses are within the allowable stress limits.

(b) Conditions of Applicability

(1) This procedure shall not be used at temperatures where the time-dependent properties govern the allowable stress.

(2) This procedure applies only for the design loading cases.

(3) This procedure applies to Configuration a when $\sigma_s \leq S_{PS,s}$ and $\sigma_c \leq S_{PS,c}$.

(4) This procedure applies to Configurations b and c when $\sigma_s \leq S_{PS,s}$.

(5) This procedure applies to Configurations e, f, and A when $\sigma_s \leq S_{PS,c}$.

(6) This procedure may only be used once for each iteration of tubesheet, shell, and channel thickness and change of materials.

(c) Calculation Procedure - After the calculation procedure given in 4.18.9.4 (Steps 1 through 10) has been performed for the elastic solution, an elastic–plastic calculation using the referenced steps from 4.18.9.4 shall be performed in accordance with the following procedure for each applicable loading case. Except for those quantities modified below, the quantities to be used for the elastic–plastic calculation shall be the same as those calculated for the corresponding elastic loading case.

(1) Define the maximum permissible bending stress limit in the shell and channel.

$$S_s^* = \min\left[S_{y,s}, \left(\frac{S_{PS,s}}{2}\right)\right] \quad \text{Configurations a, b, c} \tag{4.18.214}$$

$$S_c^* = \min\left[S_{y,c}, \left(\frac{S_{PS,c}}{2}\right)\right] \quad \text{Configurations a, e, f, A} \tag{4.18.215}$$

(2) Using bending stresses $\sigma_{s,b}$ and $\sigma_{c,b}$ calculated in Step 10 for the elastic solution, determine fact_s and fact_c as follows.

$$\text{fact}_s = \min\left[\left(1.4 - \frac{0.4\left|\sigma_{s,b}\right|}{S_s^*}\right), 1.0\right] \quad \text{Configurations a, b, c} \tag{4.18.216}$$

$$\text{fact}_c = \min\left[\left(1.4 - \frac{0.4\left|\sigma_{c,b}\right|}{S_c^*}\right), 1.0\right] \quad \text{Configurations a, e, f, A} \tag{4.18.217}$$

(3) For Configuration a, if fact_s = 1.0 and fact_c = 1.0, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to 4.18.9.5(c)(4). For Configurations b and c, if fact_s = 1.0, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to 4.18.9.5(c)(4).

(4) Calculate reduced values of E_s and E_c as follows:

$$E_s^* = E_s \cdot \text{fact}_s \qquad \text{Configurations a, b, c} \tag{4.18.218}$$

$$E_c^* = E_c \cdot \text{fact}_c \qquad \text{Configurations a, e, f, A} \tag{4.18.219}$$

(5) In 4.18.8.4, Step 2, recalculate k_s and λ_s by replacing E_s with E_s*, and k_c and λ_c by replacing E_c with E_c*.

(6) In 4.18.8.4, Step 4, recalculate F, Φ, and Q_1.

(7) In 4.18.8.4, Step 7, recalculate Q_2, Q_3, F_m, as applicable, and the tubesheet bending stress σ. If $|\sigma| \le 1.5S$, the design is acceptable and the calculation procedure is complete. Otherwise, the unit geometry shall be reconsidered.

4.18.9.6 Calculation Procedure for Effect of Radial Differential Thermal Expansion Adjacent to the Tubesheet.

(a) Scope

(1) This procedure describes how to use the rules of 4.18.9.4 when the effect of radial differential thermal expansion between the tubesheet and integral shell or channel is to be considered.

(2) This procedure shall be used when cyclic or dynamic reactions due to pressure or thermal variations are specified.

(3) This procedure shall be used when specified by the user. The user shall provide the Manufacturer with the data necessary to determine the required tubesheet, channel, and shell metal temperatures.

(4) Optionally, the designer may use this procedure to consider the effect of radial differential thermal expansion even when it is not required by 4.18.9.6(a)(2) or 4.18.9.6(a)(3).

(b) Conditions of Applicability - This calculation procedure applies only when the tubesheet is integral with the shell or channel (Configurations a, b, c, e, f, and A).

(c) Calculation Procedure - The calculation procedure outlined in 4.18.9.4 shall be performed only for the operating loading cases, accounting for the following modifications.

(1) Determine the average temperature of the unperforated rim T_r.

$$T_r = \frac{T' + T_s' + T_c'}{3} \qquad \text{Configuration a} \tag{4.18.220}$$

$$T_r = \frac{T' + T_s'}{2} \qquad \text{Configurations b, c} \tag{4.18.221}$$

$$T_r = \frac{T' + T_c'}{2} \qquad \text{Configurations e, f, A} \tag{4.18.222}$$

For conservative values of P_s* and P_c*, $T_r = T'$ may be used.

(2) Determine the average temperature of the shell T_s* and channel T_c* at their junction to the tubesheet using the equations shown below.

$$T_s^* = \frac{T_s' + T_r}{2} \qquad \text{Configurations a, b, c} \tag{4.18.223}$$

$$T_c^* = \frac{T_c' + T_r}{2} \qquad \text{Configurations a, e, f, A} \tag{4.18.224}$$

For conservative values of P_s* and P_c*, $T_s^* = T_s'$ and $T_c^* = T_c'$ may be used.

(3) Calculate P_s* and P_c*.

$$P_s^* = \frac{E_s t_s\left[\alpha_s'\left(T_s^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]}{a_s} \qquad \text{Configurations a, b, c} \tag{4.18.225}$$

$$P_s^* = 0 \qquad \text{Configurations e, f, A} \tag{4.18.226}$$

$$P_c^* = \frac{E_c t_c \left[\alpha'_c \left(T_c^* - T_a \right) - \alpha' \left(T_r - T_a \right) \right]}{a_c} \qquad \text{Configurations a, e, f, A} \tag{4.18.227}$$

$$P_c^* = 0 \qquad \text{Configurations b, c} \tag{4.18.228}$$

(4) In Step 7, replace the equation for Q_2 with:

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t \right) - \left(\omega_s P_s^* - \omega_c P_c^* \right) + \frac{\gamma_b W^*}{2\pi}}{1 + \Phi Z_m} \tag{4.18.229}$$

(5) In Step 10, replace the equations for $\sigma_{s,b}$ and $\sigma_{c,b}$ with the following equations, where H is given by eq. (4.18.122).

$$\sigma_{s,b} = \frac{6k_s}{t_s^2} \left[\beta_s \left(\delta_s P_s + \frac{a_s^2 P_s^*}{E_s t_s} \right) + \frac{6\left(1 - \left(\nu^* \right)^2 \right)}{E^*} \left(\frac{a_o^3}{h^3} \right) \left(1 + \frac{h \beta_s}{2} \right) H \right] \tag{4.18.230}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2} \left[\beta_c \left(\delta_c P_t + \frac{a_c^2 P_c^*}{E_c t_c} \right) + \frac{6\left(1 - \left(\nu^* \right)^2 \right)}{E^*} \left(\frac{a_o^3}{h^3} \right) \left(1 + \frac{h \beta_c}{2} \right) H \right] \tag{4.18.231}$$

4.18.9.7 Calculation Procedure for Simply Supported Floating Tubesheets.

4.18.9.7.1 Scope. This procedure describes how to use the rules of 4.18.9.4 when the effect of the stiffness of the integral channel and/or shell is not considered.

4.18.9.7.2 Conditions of Applicability. This calculation applies only when the tubesheet is integral with the shell or channel (Configurations a, b, c, e, f, and A.)

4.18.9.7.3 Calculation Procedure. The calculation procedure outlined in 4.18.9.4 shall be performed accounting for the following modifications;

(a) Perform Steps 1 through 9.

(b) Perform Step 10 except as follows:

(1) The shell (Configuration a, b, and c) is not required to meet a minimum length requirement.

(2) The channel (Configurations a, e, f, and A) is not required to meet a minimum length requirement.

(3) Acceptance Criteria

(-a) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, then the shell and channel are acceptable. Otherwise increase the thickness of the overstressed component(s) (shell and/or channel) and return to Step 1.

(-b) Configuration b and c: If $\sigma_s \le S_{PS,s}$ then the shell is acceptable. Otherwise increase the thickness of the shell and return to Step 1.

(-c) Configuration e, f and A: If $\sigma_c \le S_{PS,c}$, then the channel is acceptable. Otherwise increase the thickness of the channel and return to Step 1.

(c) Do not perform Step 11

(d) Repeat Steps 1 through 7 for the design loading cases, with the following changes to Step 2 until the tubesheet stress criteria have been met:

(1) Configurations a, b, and c: $\beta_s = 0$, $k_s = 0$, $\lambda_s = 0$, $\delta_s = 0$.

(2) Configurations a, e, f, and A: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$.

(21) 4.18.10 TUBE-TO-TUBESHEET WELDS

DELETED

4.18.11 BELLOWS EXPANSION JOINTS

Bellows expansion joints shall be designed in accordance with 4.19, as applicable. The expansion joint shall be designed for the axial displacement range over all load cases from one of the following equations for the axial displacement over the length of the thin-walled bellows element. Note that these may be used for flanged-and-flued or flanged-only expansion joints when the expansion joint analysis method uses the displacement over the expansion element only [see 4.18.12.1(c)].

(a) For heat exchangers with constant shell thickness and material, use the following equation:

$$\Delta_J = \frac{\sigma_{s,m}\left[t_s\left(D_s + t_s\right)\pi\right]}{JK_s} + \frac{\pi}{8}\frac{D_J^2 - D_s^2}{K_J}P_s$$

(b) For heat exchangers that have a different shell thickness and/or material adjacent to the tubesheet per 4.18.8.5, use the following equation:

$$\Delta_J = \frac{\sigma_{s,m}\left[t_{s,1}\left(D_s + t_{s,1}\right)\pi\right]}{JK_s^*} + \frac{\pi}{8}\frac{D_J^2 - D_s^2}{K_J}P_s$$

4.18.12 FLEXIBLE SHELL ELEMENT EXPANSION JOINTS

4.18.12.1 Design.

(a) Flexible shell element expansion joints shall be designed in accordance with 4.20, as applicable.

(b) The higher stress limits shown in Table 4.18.5 may be applied in lieu of those in 4.20.5(a). These limits allow the expansion joint to yield, which decreases its stiffness. All calculations must be performed in both the corroded and uncorroded condition. To apply these limits it shall be shown that

(1) the design of the other components of the heat exchanger (i.e., tubesheet, tubes, shell, channel, etc.) is acceptable considering the decreased stiffness of the expansion joint. This may be accomplished by performing an additional evaluation of all of the components of the exchanger for design loading cases 1 through 4 (when $P_{sd,\min}$ and $P_{td,\min}$ are both zero, design loading case 4 does not need to be considered) with zero expansion joint stiffness. In 4.18.8, this may be accomplished by replacing the equation for P_e in 4.18.8.4, Step 6 with

$$P_e = \left[1 - \frac{1}{2}\left(\rho_s^2 + \frac{D_J^2}{D_o^2}\right)\right]P_s - P_t$$

(2) the rotational stiffness at the expansion joint corners and torus is not necessary to meet the stress limits for annular plates and straight flanges for the design loading cases shown in Table 4.18.5. This may be accomplished by modeling the corners and torus as simply supported to determine the stress in the annular plates and straight flanges.

(c) Displacements arising from pressure and differential thermal expansion shall be calculated for use in the expansion joint analysis. The length over which the displacement is taken is dependent upon the expansion joint analysis method. If the expansion joint analysis method utilizes displacements over the length of the expansion joint only, use the appropriate equation from 4.18.11. If the expansion joint analysis method utilizes displacements over the length between the inner tubesheet faces, L, use the appropriate equation from below.

(1) For heat exchangers with a constant shell thickness and material, use one of the following:

(-a) If the expansion joint analysis includes thermal expansion effects

$$\Delta_s^T = \frac{\sigma_{s,m}\left[t_s\left(D_s + t_s\right)\pi\right]}{JK_s} + L\alpha_{s,m}\left(T_{s,m} - T_a\right) - \frac{\pi D_s^2 P_s}{2K_s}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_s^2}{K_J}P_s$$

(-b) If the expansion joint analysis does not include thermal expansion effects

$$\Delta_s^M = \frac{\sigma_{s,m}\left[t_s\left(D_s + t_s\right)\pi\right]}{JK_s} - \frac{\pi D_s^2 P_s}{2K_s}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_s^2}{K_J}P_s$$

(2) For heat exchangers that have a different shell thickness and/or material adjacent to the tubesheet per 4.18.8.5, use one of the following:

(-a) If the expansion joint analysis includes thermal expansion effects

$$\Delta_s^T = \frac{\sigma_{s,m}\left[t_{s,1}\left(D_s + t_{s,1}\right)\pi\right]}{JK_s^*} + \left[\left(L - \ell_1 - \ell_1'\right)\alpha_{s,m} + \left(\ell_1 + \ell_1'\right)\alpha_{s,m,1}\right]\left(T_{s,m} - T_a\right) - \frac{\pi D_s^2 P_s}{2K_s^*}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_s^2}{K_J}P_s$$

(-b) If the expansion joint analysis does not include thermal expansion effects

$$\Delta_s^M = \frac{\sigma_{s,m}\left[t_{s,1}\left(D_s + t_{s,1}\right)\pi\right]}{JK_s^*} - \frac{\pi D_s^2 P_s}{2K_s^*}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_s^2}{K_J}P_s$$

4.18.13 PRESSURE TEST REQUIREMENTS

(a) The shell side and the tube side of the heat exchanger shall be subjected to a pressure test in accordance with 4.1 and Part 8.

(b) Shipping bars on bellows expansion joints may be required to maintain assembly length during shipment and vessel fabrication. Shipping bars shall not be engaged or otherwise provide any restraint of the expansion joint during vessel pressure testing and operation [see 4.19.3.1(c) and 4.19.3.1(d)].

4.18.14 HEAT EXCHANGER MARKING AND REPORTS

4.18.14.1 Required Marking. The marking of heat exchangers shall be in accordance with Annex 2-F using the specific requirements for combination units (multi-chamber vessels). When the markings are grouped in one location and abbreviations for each chamber are used, they shall be as follows:

(a) The chambers shall be abbreviated SHELL for shell side and TUBES for tube side. This abbreviation shall precede the appropriate design data. For example, use the following for the shell side maximum allowable working pressure and for the tube side maximum allowable working pressure:

(1) SHELL FV & 2 000 kPa (FV & 300 psi) at 280°C (500°F)

(2) TUBES 1 000 kPa (150 psi) at 175°C (350°F)

(b) When the markings are different for each chamber, the chambers shall be abbreviated with a S for shell side and T for the tube side. For example, use "F-T" for forged construction on the tube side.

4.18.14.2 Supplemental Marking. A supplemental tag or marking shall be supplied on the heat exchanger to caution the user if there are any restrictions on the design, testing, or operation of the heat exchanger. The marking shall meet the requirements of 2-F.7, except that height of the characters for the caution required by (b) shall be at least 3 mm ($^1\!/_8$ in.) high. Supplemental marking shall be required for, but not limited to, the following:

(a) Common Elements - Shell-and-tube heat exchangers are combination units as defined in 4.1.8.1, and the tubes and tubesheets are common elements. The following marking is required when the common elements are designed for conditions less severe than the design conditions for which its adjacent chambers are stamped.

(1) Differential Pressure Design - When common elements such as tubes and tubesheets are designed for a differential design pressure, the heat exchanger shall be marked "Differential Design" in addition to meeting the requirements of 4.1.8.1. If the tubes and tubesheets are designed for a differential pressure of 150 psi, an example of the marking would be:

DIFFERENTIAL DESIGN: TUBES & TUBESHEETS 150 psi

(2) Mean Metal Temperature Design - When common elements such as tubes and tubesheets are designed for a maximum mean metal design temperature that is less than the maximum of the shell side and tube side design temperatures, the heat exchanger shall be marked "Max Mean Metal Temp" in addition to meeting the requirements of 4.1.8.1. If the tubes are designed for a maximum mean metal temperature of 400°F, an example of the marking would be:

MAX MEAN METAL TEMP: TUBES 400°F

(b) Fixed Tubesheet Heat Exchangers - Fixed tubesheet heat exchangers shall be marked with the following caution: "The heat exchanger design has been evaluated for the range of conditions listed on Form A-4 of the MDR. It shall be reevaluated for conditions outside this range before being operated at them."

4.18.14.3 Manufacturer's Data Reports.

(a) Common Elements. When common elements such as tubes and tubesheets are designed for a differential pressure or a mean metal temperature, or both, that is less severe than the design conditions for which its adjacent chambers are stamped, the data for each common element that differs from the data for the corresponding chamber shall be indicated as required in 4.1.8.2 in the "Remarks" section of the Manufacturer's Data Report.

(b) Fixed Tubesheet Heat Exchangers. For each design and operating condition, the following information shall be indicated on Form A-4 of the Manufacturer's Data Report Supplementary Sheet for Shell-and-Tube Heat Exchangers. The operating conditions may be combined on this form where they are bounded by the operating pressure range, maximum metal temperatures, and axial differential thermal expansion range.

(1) Name of Condition. The first condition shown shall be the design condition. If there is more than one design condition or a differential pressure design condition, multiple lines may be used. Each different operating condition or range of operating conditions shall be listed.

(2) Design/Operating Pressure Ranges. Range of shell side and tube side pressures for each condition listed.

(3) Design/Operating Metal Temperatures. For each condition, the temperature at which the allowable stress was taken for the shell, channel, tube, and tubesheet shall be listed. Any metal temperature between the MDMT and the listed temperature is permitted, provided the resulting axial differential thermal expansion is within the listed range.

(4) Axial Differential Thermal Expansion Range. The minimum and maximum axial differential thermal expansion for each operating condition listed. If the minimum value is positive, zero shall be used for the minimum value. If the maximum value is negative, zero shall be used for the maximum value. Within the listed range of operating temperature and pressure, any combination of shell and tube axial mean metal temperatures is permitted, provided the resulting axial differential thermal expansion is within the listed range.

4.18.15 NOMENCLATURE (21)

(a) Nomenclature for tubesheet extension (4.18.5).

D_E = maximum of the shell and channel gasket inside diameters, but not less than the maximum of the shell and channel flange inside diameters
G = diameter of gasket load reaction
= G_c for U-tube tubesheet Configuration b
= G_s for U-tube tubesheet Configuration e
= G_c for fixed tubesheet Configuration b
= G_c for stationary tubesheet Configuration b of a floating tubesheet exchanger
= G_s for stationary tubesheet Configuration e of a floating tubesheet exchanger
= G_c for floating tubesheet Configuration B of a floating tubesheet exchanger
= G_c or G_s for tubesheet Configuration d when applicable (eg. hydrotest)
h_g = gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction (see 4.16)
h_r = minimum required thickness of the tubesheet extension
P_s = shell side design pressure. For shell side vacuum, use a negative value for P_s
P_t = tube side design pressure. For tube side vacuum, use a negative value for P_t
S_a = allowable stress from Annex 3-A for the material of the tubesheet extension at ambient temperature
S_{fe} = allowable stress from Annex 3-A for tubesheet extension material at tubesheet extension design temperature
W = flange design bolt load
= W_g for gasket seating conditions
= W_o for operating conditions

(b) Nomenclature for determining tubesheet characteristics (see 4.18.6)

A_L = total area of untubed lanes, $A_L = U_{L1}L_{L1} + U_{L2}L_{L2} + \dots + U_{Ln}L_{Ln}$ (limited to $4D_op$)
A_p = total area enclosed by C_p
c_t = tubesheet corrosion allowance on the tube side, $c_t = 0$ for the uncorroded condition
C_p = perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outer most tubes (see Figure 4.18.14)
D_o = equivalent diameter of outer tube limit circle
d = diameter of tube hole
d_t = nominal outside diameter of tubes
d^* = effective tube hole diameter
E = modulus of elasticity for tubesheet material at tubesheet design temperature
E_{tT} = modulus of elasticity for tube material at tubesheet design temperature, T
E^* = effective modulus of elasticity of tubesheet in perforated region, T
h = tubesheet thickness
h_g = tube side pass partition groove depth

h'_g = effective tube side pass partition groove depth
$L_{L1}, L_{L2}, \ldots$ = length(s) of untubed lane(s)
l_{tx} = expanded length of tube in tubesheet, $0 \le l_{tx} \le h$. An expanded tube-to-tubesheet joint is produced by applying pressure inside the tube such that contact is established between the tube and tubesheet. In selecting an appropriate value of expanded length, the designer shall consider the degree of initial expansion, differences in thermal expansion, or other factors that could result in loosening of the tubes within the tubesheet
p = tube pitch
p^* = effective tube pitch
r_o = radius to outermost tube hole center
S = allowable stress from Annex 3-A for tubesheet material at tubesheet design temperature,
S_{tT} = allowable stress from Annex 3-A for tube material at tubesheet design temperature,. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85
T = tubesheet design temperature
t_t = nominal tube wall thickness
$U_{L1}, U_{L2}, \ldots$ = center-to-center distance(s) between adjacent tube rows of untubed lane(s), but not to exceed $4p$
μ = basic ligament efficiency for shear
μ^* = effective ligament efficiency for bending
ν^* = effective Poisson's ratio in perforated region of tubesheet
ρ = tube expansion depth ratio

(c) Nomenclature for the design of U-tube tubesheets (see 4.18.7)
A = outside diameter of tubesheet, except as limited by 4.18.4(b)
C = bolt circle diameter (see 4.16)
D_c = inside channel diameter
D_s = inside shell diameter
E = modulus of elasticity for tubesheet material at design temperature
E_c = modulus of elasticity for channel material at design temperature
E_s = modulus of elasticity for shell material at design temperature
G_1 = midpoint of contact between flange and tubesheet
G_c = diameter of channel gasket load reaction (see 4.16)
G_s = diameter of shell gasket load reaction (see 4.16)
h = tubesheet thickness
P_s = shell-side design pressure. For shell-side vacuum, use a negative value for P_s
P_t = tube side design pressure. For tube side vacuum, use a negative value for P_t
$P_{td,\max}$ = maximum tube side design pressure
$P_{td,\min}$ = minimum tube side design pressure (negative if vacuum is specified, otherwise zero)
$P_{sd,\max}$ = maximum shell side design pressure
$P_{sd,\min}$ = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)
S = allowable stress from Annex 3-A for tubesheet material at tubesheet design temperature
S_c = allowable stress from Annex 3-A for channel material at design temperature. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85
S_s = allowable stress from Annex 3-A for shell material at design temperature. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85
S_y = yield strength from Annex 3-D for tubesheet material at tubesheet design temperature
$S_{y,c}$ = yield strength from Annex 3-D for channel material at design temperature
$S_{y,s}$ = yield strength from Annex 3-D for shell material at design temperature
$S_{PS,c}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for channel material at design temperature
$S_{PS,s}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for shell material at design temperature
t_c = channel thickness
t_s = shell thickness
W_c = channel flange design bolt load for the gasket seating condition (see 4.16)
W_s = shell flange design bolt load for the gasket seating condition (see 4.16)

W_{max} = MAX[(W_c), (W_s)]
W_{dc} = channel flange design bolt load (see W_o, 4.16)
W_{ds} = shell flange design bolt load (see W_o, 4.16)
$W_{d,max}$ = MAX[(W_{dc}), (W_{ds})]
W^* = tubesheet design bold load determined in accordance with Table 4.18.6
ν_c = Poisson's ratio of channel material
ν_s = Poisson's ratio of shell material

(d) Nomenclature for the design of fixed or floating tubesheets (see 4.18.8 and 4.18.9)
A = outside diameter of tubesheet, except as limited by 4.18.4(b)
a_c = radial channel dimension
a_o = equivalent radius of outer tube limit circle
a_s = radial shell dimension
C = bolt circle diameter (see 4.16)
D_c = inside channel diameter
$D_{ecc,L}$ = eccentric cone inside diameter at the large end (see Figure 4.18.17)
$D_{ecc,S}$ = eccentric cone inside diameter at the small end (see Figure 4.18.17)
D_J = inside diameter of the expansion joint at its convolution height
D_s = inside shell diameter
$D_{s,L}$ = large cylinder inside diameter (see Figure 4.18.17)
d_t = nominal outside diameter of tubes
E = modulus of elasticity for tubesheet material at T
E_c = modulus of elasticity for channel material at T_c
E_{ecc} = modulus of elasticity for eccentric cone material at T_s
$E_{ecc,w}$ = joint efficiency (longitudinal stress) for eccentric cone
E_s = modulus of elasticity for shell material at T_s
$E_{s,1}$ = modulus of elasticity for shell material adjacent to the tubesheet at T_s
$E_{s,L}$ = modulus of elasticity for large cylinder material at T_s
$E_{s,L,w}$ = joint efficiency (longitudinal stress) for large cylinder
$E_{s,w}$ = joint efficiency (longitudinal stress) for shell
E_t = modulus of elasticity for tube material at T_t
fact_c = factor used in the elastic–plastic analysis to account for any yielding of the channel
fact_s = factor used in the elastic–plastic analysis to account for any yielding of the shell
G_1 = midpoint of contact between flange and tubesheet
G_c = diameter of channel gasket load reaction (see 4.16)
G_s = diameter of shell gasket load reaction (see 4.16)
h = tubesheet thickness
J = ratio of expansion joint to shell axial rigidity (J = 1.0 if no expansion joint)
K_j = axial rigidity of expansion joint, total force/elongation
k = constant accounting for the method of support for the unsupported tube span under consideration
= 0.6 for unsupported spans between two tubesheets
= 0.8 for unsupported spans between a tubesheet and a tube support
= 1.0 for unsupported spans between two tube supports
l = unsupported tube span under consideration
l_1, l'_1 = lengths of shell thickness $t_{s,1}$ adjacent to the tubesheets
L = tube length between inner tubesheet faces, $L = L_t - 2h$
L_{ecc} = eccentric cone shortest length from small end to large end (see Figure 4.18.17)
L_s = axial length of small cylinder (see Figure 4.18.17)
$L_{s,L}$ = axial length of large cylinder (see Figure 4.18.17)
L_t = tube length between outer tubesheet faces
N_t = number of tubes
P_e = effective pressure acting on tubesheet
P_s = shell side design or operating pressure, as applicable. For shell side vacuum, use a negative value for P_s
$P_{sd,max}$ = maximum shell side design pressure
$P_{sd,min}$ = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)
$P_{sox,max}$ = max. (0, maximum shell-side operating pressure for operating condition x)
$P_{sox,min}$ = min. (0, minimum shell-side operating pressure for operating condition x)

P_t = tube side design or operating pressure, as applicable. For tube side vacuum, use a negative value for P_t
$P_{td,\max}$ = maximum tube side design pressure
$P_{td,\min}$ = minimum tube side design pressure (negative if vacuum is specified, otherwise zero)
$P_{tox,\max}$ = max. (0, maximum tube-side operating pressure for operating condition x)
$P_{tox,\min}$ = min. (0, minimum tube-side operating pressure for operating condition x)
S = allowable stress from Annex 3-A for tubesheet material at T
S_c = allowable stress from Annex 3-A for channel material at T_c. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85
S_{ecc} = allowable stress from Annex 3-A for eccentric cone material at T_s
$S_{ecc,b}$ = maximum allowable longitudinal compressive stress in accordance with 4.4.12.3 for eccentric cone material at T_s
S_{PS} = allowable primary plus secondary stress evaluated using 4.1.6.3 for tubesheet material at temperature T
$S_{PS,c}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for channel material at temperature T_c
$S_{PS,ecc}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for eccentric cone material at T_s
$S_{PS,s}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for shell material at temperature T_s
$S_{PS,s,1}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for shell material adjacent to the tubesheet at temperature T_s
$S_{PS,s,L}$ = allowable primary plus secondary stress evaluated using 4.1.6.3 for large cylinder material at T_s
S_s = allowable stress from Annex 3-A for shell material at T_s. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85
$S_{s,1}$ = allowable stress from Annex 3-A for shell material adjacent to the tubesheets at T_s
$S_{s,b}$ = maximum allowable longitudinal stress in accordance with 4.4.12.2 for the shell
$S_{s,b,1}$ = maximum allowable longitudinal stress in accordance with 4.4.12.2 for the shell adjacent to the tubesheets
$S_{s,L}$ = allowable stress from Annex 3-A for large cylinder material at T_s
$S_{s,L,b}$ = maximum allowable longitudinal compressive stress in accordance with 4.4.12.2 for large cylinder material at T_s
S_t = allowable stress from Annex 3-A for tube material at T_t. For a welded pipe or tube, use the allowable stress from Annex 3-A for the equivalent seamless product. When the allowable stress for the equivalent seamless product is not available, divide the allowable stress of the welded product by 0.85
S_y = yield strength from Annex 3-D for tubesheet material at T
$S_{y,c}$ = yield strength from Annex 3-D for channel material at T_c
$S_{y,s}$ = yield strength from Annex 3-D for shell material at T_s
$S_{y,s,1}$ = yield strength from Annex 3-D for shell material adjacent to the tubesheets at T_s
$S_{y,t}$ = yield strength from Annex 3-D for tube material at T_t
T = tubesheet design temperature for the design condition or operating metal temperature for operating condition x, as applicable
T' = tubesheet metal temperature at the rim (see Figure 4.18.18)
T_a = ambient temperature, 20°C (70°F)
T_c = channel design temperature for the design condition or operating metal temperature for operating condition x, as applicable
t_c = channel thickness
T'_c = channel metal temperature at the tubesheet
T'_{cx} = channel metal temperature at the tubesheet for operating condition x
t_{ecc} = eccentric cone wall thickness (see Figure 4.18.17)
T_s = shell design temperature for the design condition or operating metal temperature for operating condition x, as applicable
t_s = shell thickness
T'_s = shell metal temperature at the tubesheet
$t_{s,1}$ = shell thickness adjacent to the tubesheets
$t_{s,L}$ = large cylinder wall thickness (see Figure 4.18.17)
$T_{s,m}$ = mean shell metal temperature along shell length
$T_{s,mx}$ = shell axial mean metal temperature for operating condition x, as applicable
T'_{sx} = shell metal temperature at the tubesheet for operating condition x

T_t = tube design temperature for the design condition or operating metal temperature for operating condition x, as applicable
t_t = nominal tube wall thickness
$T_{t,m}$ = mean tube metal temperature along tube length
$T_{t,mx}$ = tube axial mean metal temperature for operating condition x, as applicable
T'_x = tubesheet metal temperature at the rim for operating condition x
W^* = tubesheet design bolt load determined in accordance with Table 4.18.6
W_c = channel flange design bolt load for the gasket seating condition (see 4.16)
W_{dc} = channel flange design bolt load (see W_o, 4.16)
$W_{d\max}$ = MAX[(W_{dc}), (W_{ds})]
W_{ds} = shell flange design bolt load (see W_o, 4.16)
$W_{\max}$ = MAX[(W_c), (W_s)]
W_s = shell flange design bolt load for the gasket seating condition (see 4.16)
W_t = tube-to-tubesheet joint load
x = 1,2,3,...n, integer denoting applicable operating condition under consideration, (e.g., normal operating, start-up, shutdown, cleaning, upset)
X_L' = equivalent line load acting on the large end cylinder due to heat exchanger constraint
X_S' = equivalent line load acting on the small end cylinder due to heat exchanger constraint
α' = mean coefficient of thermal expansion of tubesheet material at T'
α'_c = mean coefficient of thermal expansion of channel material at T'_c
$\alpha_{ecc,m}$ = mean coefficient of thermal expansion of eccentric cone material at $T_{s,m}$
α'_s = mean coefficient of thermal expansion of shell material at T'_s
$\alpha_{s,m}$ = mean coefficient of thermal expansion of shell material at $T_{s,m}$
$\alpha_{s,m,1}$ = mean coefficient of thermal expansion of shell material adjacent to the tubesheets at $T_{s,m}$
$\alpha_{s,m,L}$ = mean coefficient of thermal expansion of large cylinder material at $T_{s,m}$
$\alpha_{t,m}$ = mean coefficient of thermal expansion of tube material at $T_{t,m}$
γ = axial differential thermal expansion between tubes and shell
Δ_J = axial displacement over the length of the thin-walled bellows element (see 4.18.11)
Δ_s = shell axial displacement over the length between the inner tubesheet faces, L [see 4.18.12.1(c)]
θ_{ecc} = eccentric cone half-apex angle, deg (see Figure 4.18.17)
ν = Poisson's ratio of tubesheet material
ν_c = Poisson's ratio of channel material
ν_{ecc} = Poisson's ratio of eccentric cone material
ν_s = Poisson's ratio of shell material
$\nu_{s,L}$ = Poisson's ratio of large cylinder material
ν_t = Poisson's ratio of tube material

4.18.16 TABLES

Table 4.18.1
Effective Elastic Modulus and Poisson's Ratio for a Perforated Plate With an Equilateral Triangular Hole Pattern

h/p	A_0	A_1	A_2	A_3	A_4	B_0	B_1	B_2	B_3	B_4
0.10	0.0353	1.2502	−0.0491	0.3604	−0.6100	−0.0958	0.6209	−0.8683	2.1099	−1.6831
0.15	...	...	...	...	...	0.8897	−9.0855	36.1435	−59.5425	35.8223
0.25	0.0135	0.9910	1.0080	−1.0498	0.0184	0.7439	−4.4989	12.5779	−14.2092	5.7822
0.50	0.0054	0.5279	3.0461	−4.3657	1.9435	0.9100	−4.8901	12.4325	−12.7039	4.4298
1.00	...	...	...	...	...	0.9923	−4.8759	12.3572	−13.7214	5.7629
2.00	−0.0029	0.2126	3.9906	−6.1730	3.4307	0.9966	−4.1978	9.0478	−7.9955	2.2398

GENERAL NOTES:
(a) $E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$
(b) $\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$
(c) These coefficients are only valid for $0.1 \le \mu^* \le 0.6$. Data for the range $0.1 \le \mu^* \le 1.0$ is provided in Annex 5-E.
(d) If $h/p < 0.1$, use $h/p = 0.1$.
(e) If $h/p > 2.0$, use $h/p = 2.0$.

Table 4.18.2
Effective Elastic Modulus and Poisson's Ratio for a Perforated Plate With a Square Hole Pattern

h/p	A_0	A_1	A_2	A_3	A_4	B_0	B_1	B_2	B_3	B_4
0.10	0.0676	1.5756	−1.2119	1.7715	−1.2628	−0.0791	0.6008	−0.3468	0.4858	−0.3606
0.15	...	...	...	...	...	0.3345	−2.8420	10.9709	−15.8994	8.3516
0.25	0.0250	1.9251	−3.5230	6.9830	−5.0017	0.4296	−2.6350	8.6864	−11.5227	5.8544
0.50	0.0394	1.3024	−1.1041	2.8714	−2.3994	0.3636	−0.8057	2.0463	−2.2902	1.1862
1.00	...	...	...	...	...	0.3527	−0.2842	0.4354	−0.0901	−0.1590
2.00	0.0372	1.0314	−0.6402	2.6201	−2.1929	0.3341	0.1260	−0.6920	0.6877	−0.0600

GENERAL NOTES:
(a) $E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$
(b) $\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$
(c) These coefficients are only valid for $0.1 \le \mu^* \le 0.6$. Data for the range $0.1 \le \mu^* \le 1.0$ is provided in Annex 5-E.
(d) If $h/p < 0.1$, use $h/p = 0.1$.
(e) If $h/p > 2.0$, use $h/p = 2.0$.

Table 4.18.3
Evaluation of Z_a, Z_d, Z_v, Z_w, Z_m, and F_m

Evaluation of Kelvin Functions ber, bei, ber', and bei' Relative to x [Note (1)]

$$ber(x) = \sum_{n=0}^{n=m-1} (-1)^n \frac{(x/2)^{4n}}{(2n)!^2} = 1 - \frac{(x/2)^4}{(2)!^2} + \frac{(x/2)^8}{(4)!^2} - \frac{(x/2)^{12}}{(6)!^2} + \ldots$$

$$bei(x) = \sum_{n=1}^{n=m} (-1)^{n-1} \frac{(x/2)^{4n-2}}{(2n-1)!^2} = \frac{(x/2)^2}{(1)!^2} - \frac{(x/2)^6}{(3)!^2} + \frac{(x/2)^{10}}{(5)!^2} - \ldots$$

$$ber'(x) = \sum_{n=1}^{n=m} (-1)^n \frac{2n(x/2)^{4n-1}}{(2n)!^2} = -\frac{2(x/2)^3}{(2)!^2} + \frac{4(x/2)^7}{(4)!^2} - \frac{6(x/2)^{11}}{(6)!^2} + \ldots$$

$$bei'(x) = \sum_{n=1}^{n=m} (-1)^{n-1} \frac{(2n-1)(x/2)^{4n-3}}{(2n-1)!^2} = \frac{(x/2)^1}{(1)!^2} - \frac{3(x/2)^5}{(3)!^2} + \frac{5(x/2)^9}{(5)!^2} - \ldots$$

Ψ_i Functions for Determination of Z_a, Z_d, Z_v, Z_w, Z_m, and F_m Relative to x

$$\Psi_1(x) = bei(x) + \left(\frac{1-\nu^*}{x}\right) ber'(x)$$

$$\Psi_2(x) = ber(x) - \left(\frac{1-\nu^*}{x}\right) bei'(x)$$

Table 4.18.3
Evaluation of Z_a, Z_d, Z_v, Z_w, Z_m, and F_m (Cont'd)

Evaluation of Z_a, Z_d, Z_v, Z_w, Z_m at X_a

$$Z_d = \frac{ber(X_a)\Psi_2(X_a) + bei(X_a)\Psi_1(X_a)}{X_a^3 Z_a}$$

$$Z_v = \frac{ber'(X_a)\Psi_2(X_a) + bei'(X_a)\Psi_1(X_a)}{X_a^2 Z_a}$$

$$Z_w = \frac{ber'(X_a)ber(X_a) + bei'(X_a)bei(X_a)}{X_a^2 Z_a}$$

$$Z_m = \frac{\left[ber'(X_a)\right]^2 + \left[bei'(X_a)\right]^2}{X_a Z_a}$$

where

$Z_a = bei'(X_a)\psi_2(X_a) - ber'(X_a)\psi_1(X_a)$

Evaluation of F_m From $0 \to X_a$

Calculate the functions $Q_m(x)$ and $Q_v(x)$ relative to x.

$$Q_m(x) = \frac{bei'(X_a)\Psi_2(x) - ber'(X_a)\Psi_1(x)}{Z_a}$$

$$Q_v(x) = \frac{\Psi_1(X_a)\Psi_2(x) - \Psi_2(X_a)\Psi_1(x)}{X_a Z_a}$$

For each loading case (note that Q_3 is a dependent on the load case being evaluated), calculate $F_m(x)$ relative to x.

$$F_m(x) = \frac{Q_v(x) + Q_3 \cdot Q_m(x)}{2}$$

F_m is the maximum of the absolute value of $F_m(x)$ as x varies from $0 \to X_a$ such that $0 \le x \le X_a$.

$$F_m = \max\left[\left|F_m(x)\right|\right]$$

NOTE:
(1) Use $m = 4 + X_a/2$ terms (rounded to the nearest integer) to obtain an adequate approximation of the Kelvin Functions and their derivatives.

Table 4.18.4
Evaluation of $F_{t,\min}$ and $F_{t,\max}$

Equations for the Determination of $F_{t,\min}$ and $F_{t,\max}$

Calculate the Kelvin functions, the Ψ_i functions and Z_a from Table 4.18.3.
Calculate functions $Z_d(x)$ and $Z_w(x)$ relative to x.

$$Z_d(x) = \frac{\Psi_2(X_a) \cdot ber(x) + \Psi_1(X_a) \cdot bei(x)}{X_a^3 \cdot Z_a}$$

$$Z_w(x) = \frac{ber'(X_a) \cdot ber(x) + bei'(X_a) \cdot bei(x)}{X_a^2 \cdot Z_a}$$

For each loading case, calculate $F_t(x)$ relative to x.

When $P_e \neq 0$

$$F_t(x) = \left[Z_d(x) + Q_3 \cdot Z_w(x)\right] \cdot \frac{X_a^4}{2}$$

When $P_e = 0$

$$F_t(x) = Z_w(x) \cdot \frac{X_a^4}{2}$$

Calculate the minimum and maximum values, $F_{t,\min}$ and $F_{t,\max}$, of $F_t(x)$ as x varies from $0 \rightarrow X_a$ such that $0 \le x \le X_a$. $F_{t,\min}$ and $F_{t,\max}$ may be positive or negative.

$$F_{t,\min} = \min\left[F_t(x)\right]$$

$$F_{t,\max} = \max\left[F_t(x)\right]$$

Table 4.18.5
Flexible Shell Element Expansion Joint Load Cases and Stress Limits

	Loading Case	Shell Side Pressure, P_s	Tube Side Pressure, P_t	Differential Thermal Expansion	Maximum Stress: Membrane: Corners and Torus	Maximum Stress: Membrane Plus Bending: Corners and Torus	Maximum Stress: Membrane Plus Bending: Annular Plates	Maximum Stress: Membrane Plus Bending: Straight Flanges
Design	1	$P_{sd,\min}$	$P_{td,\max}$	No	$1.5S$	S_{PS}	$1.5S$	$1.5S$
	2	$P_{sd,\max}$	$P_{td,\min}$	No	$1.5S$	S_{PS}	$1.5S$	$1.5S$
	3	$P_{sd,\max}$	$P_{td,\max}$	No	$1.5S$	S_{PS}	$1.5S$	$1.5S$
	4	$P_{sd,\min}$	$P_{td,\min}$	No	$1.5S$	S_{PS}	$1.5S$	$1.5S$
Operating	1	None	P_{tox}	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}
	2	P_{sox}	None	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}
	3	P_{sox}	P_{tox}	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}
	4	None	None	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}

Table 4.18.6
Tubesheet Effective Bolt Load, W^*

Configuration	Design Loading Case 1	2	3	4	Operating Loading Case 1–4
a	0	0	0	0	0
b	W_{dc}	0	W_{dc}	0	W_c
c	W_{dc}	0	W_{dc}	0	W_c
d	W_{dc}	W_{ds}	$W_{d\ max}$	0	W_{max}
e	0	W_{ds}	W_{ds}	0	W_s
f	0	W_{ds}	W_{ds}	0	W_s
A	0	0	0	0	0
B	W_{dc}	0	W_{dc}	0	W_c
C	W_{dc}	0	W_{dc}	0	W_c
D	0	0	0	0	0

Table 4.18.7
Load Combinations Required to Evaluate the Heat Exchanger for the Design Condition

Design Loading Case	Shell Side Design Pressure, P_s	Tube Side Design Pressure, P_t
1	$P_{sd,min}$	$P_{td,max}$
2	$P_{sd,max}$	$P_{td,min}$
3	$P_{sd,max}$	$P_{td,max}$
4	$P_{sd,min}$	$P_{td,min}$

GENERAL NOTE: When $P_{sd,min}$ and $P_{td,min}$ are both zero, design loading case 4 does not need to be considered.

Table 4.18.8
Load Combinations Required to Evaluate the Heat Exchanger for Each Operating Condition x

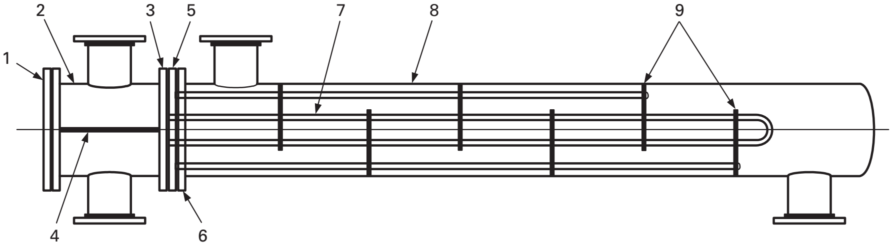
Operating Loading Case	Operating Pressure: Shell Side, P_s	Operating Pressure: Tube Side, P_t	Axial Mean Metal Temperature: Tubes, $T_{t,m}$	Axial Mean Metal Temperature: Shell, $T_{s,m}$
1	$P_{sox,\ min}$	$P_{tox,\ max}$	$T_{t,mx}$	$T_{s,mx}$
2	$P_{sox,\ max}$	$P_{tox,\ min}$	$T_{t,mx}$	$T_{s,mx}$
3	$P_{sox,\ max}$	$P_{tox,\ max}$	$T_{t,mx}$	$T_{s,mx}$
4	$P_{sox,\ min}$	$P_{tox,\ min}$	$T_{t,mx}$	$T_{s,mx}$

Table 4.18.9
Load Combinations Required to Evaluate the Heat Exchanger for Each Operating Condition x

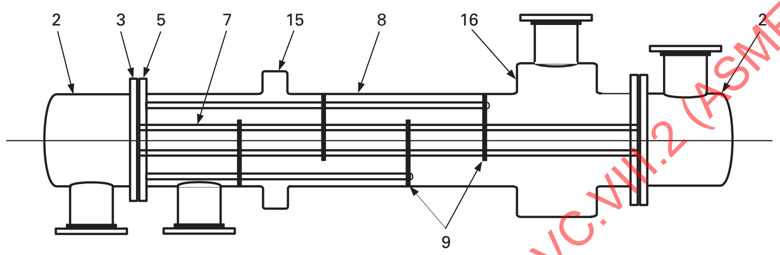
Operating Loading Case	Operating Pressure: Shell Side, P_s	Operating Pressure: Tube Side, P_t	Axial Mean Metal Temperature: Tubes, $T_{t,m}$	Axial Mean Metal Temperature: Shell, $T_{s,m}$	Metal Temperature: Tubesheet at the RIM, T'	Metal Temperature: Channel at Tubesheet, T'_c	Metal Temperature: Shell at Tubesheet, T'_s
1	$P_{sox,\ min}$	$P_{tox,\ max}$	$T_{t,mx}$	$T_{s,mx}$	T'_x	T'_{cx}	T'_{sx}
2	$P_{sox,\ max}$	$P_{tox,\ min}$	$T_{t,mx}$	$T_{s,mx}$	T'_x	T'_{cx}	T'_{sx}
3	$P_{sox,\ max}$	$P_{tox,\ max}$	$T_{t,mx}$	$T_{s,mx}$	T'_x	T'_{cx}	T'_{sx}
4	$P_{sox,\ min}$	$P_{tox,\ min}$	$T_{t,mx}$	$T_{s,mx}$	T'_x	T'_{cx}	T'_{sx}

4.18.17 FIGURES

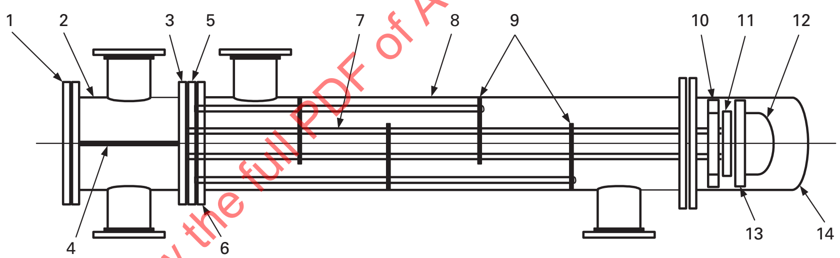
Figure 4.18.1
Terminology of Heat Exchanger Components



(a) U-Tube Heat Exchanger



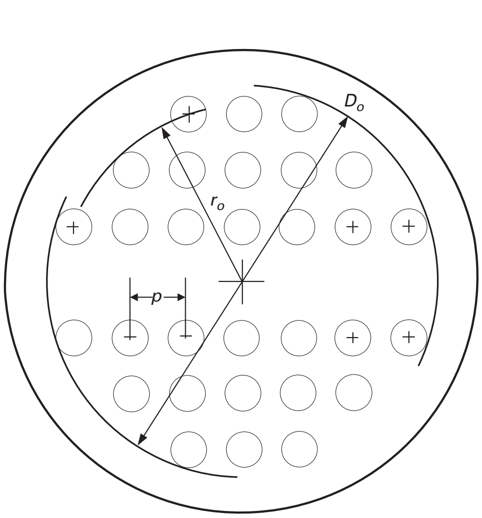
(b) Fixed Tubesheet Heat Exchanger



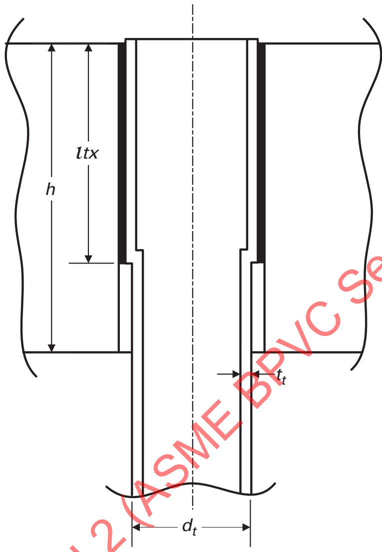
(c) Floating Tubesheet Heat Exchanger

Legend:
1 = channel cover (bolted flat cover)
2 = channel
3 = channel flange
4 = pass partition
5 = stationary tubesheet
6 = shell flange
7 = tubes
8 = shell
9 = baffles or support plates
10 = floating head backing device
11 = floating tubesheet
12 = floating head
13 = floating head flange
14 = shell cover
15 = expansion joint
16 = distribution or vapor belt

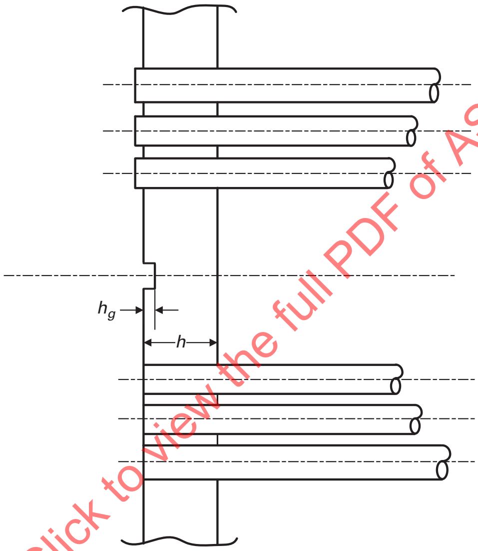
Figure 4.18.2
Tubesheet Geometry



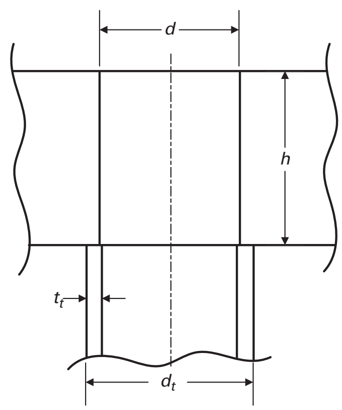
(a) Tubesheet Layout



(b) Expanded Tube Joint



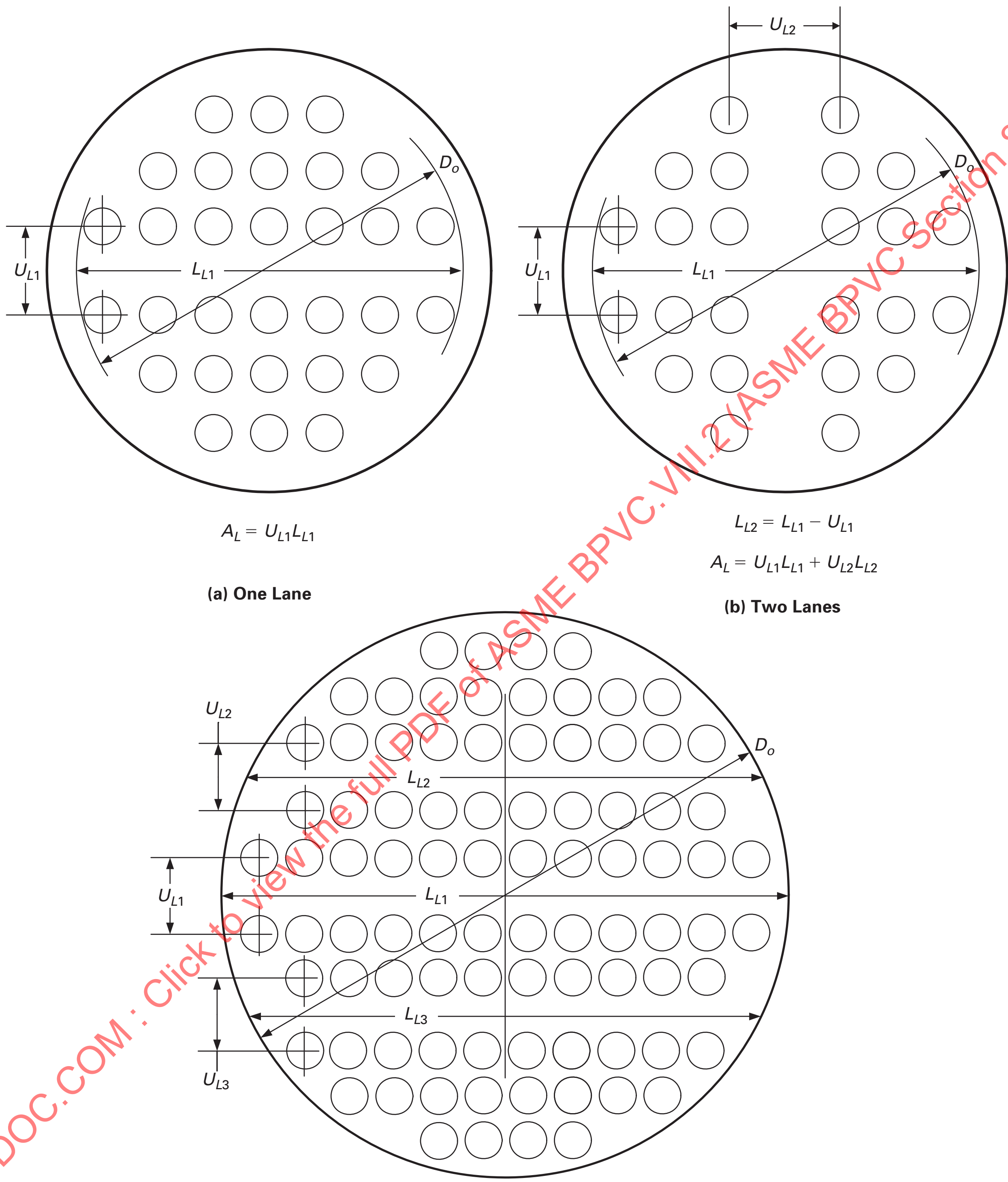
(c) Tube Side Pass Partition Groove Depth



(d) Tubes Welded to Back Side of Tubesheet [Note (1)]

NOTE:
(1) $d_t - 2t_t \le d < d_t$

Figure 4.18.3
Typical Untubed Lane Configurations



$A_L = U_{L1}L_{L1}$

(a) One Lane

$L_{L2} = L_{L1} - U_{L1}$

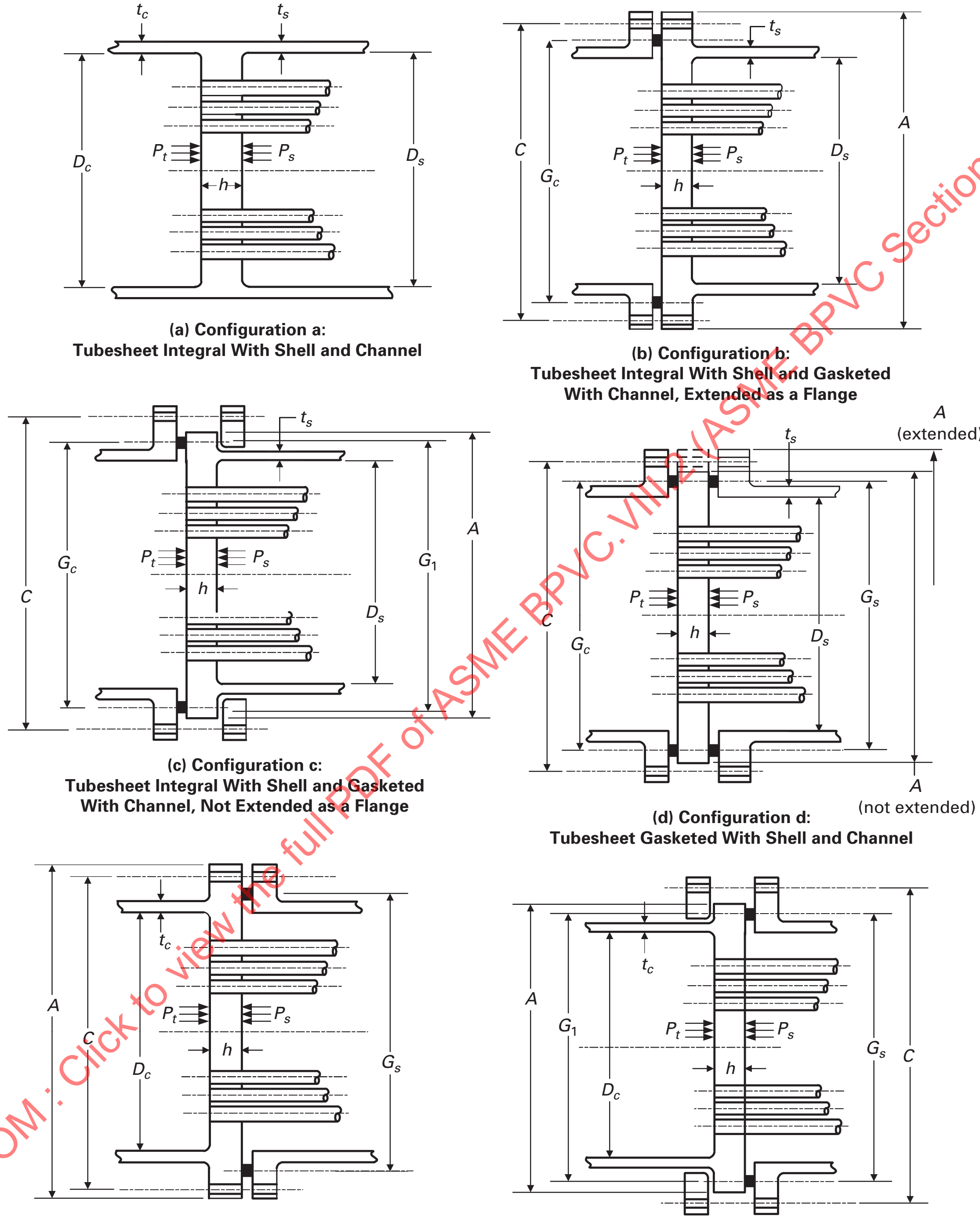
$A_L = U_{L1}L_{L1} + U_{L2}L_{L2}$

(b) Two Lanes

$A_L = U_{L1}L_{L1} + U_{L2}L_{L2} + U_{L3}L_{L3}$

(c) Three Lanes

Figure 4.18.4
U-Tube Tubesheet Configurations



(a) Configuration a:
Tubesheet Integral With Shell and Channel

(b) Configuration b:
Tubesheet Integral With Shell and Gasketed With Channel, Extended as a Flange

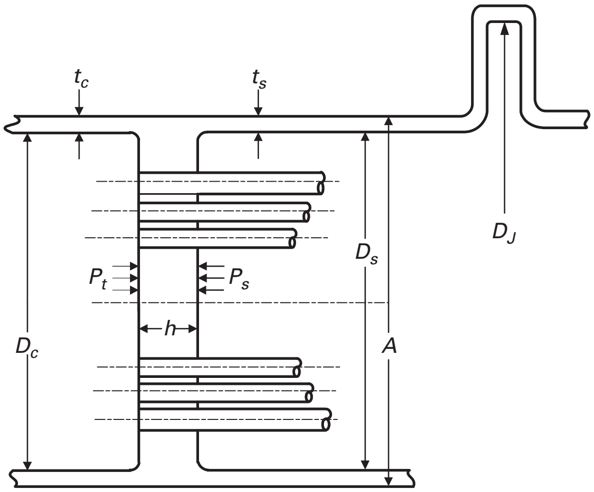
(c) Configuration c:
Tubesheet Integral With Shell and Gasketed With Channel, Not Extended as a Flange

(d) Configuration d:
Tubesheet Gasketed With Shell and Channel

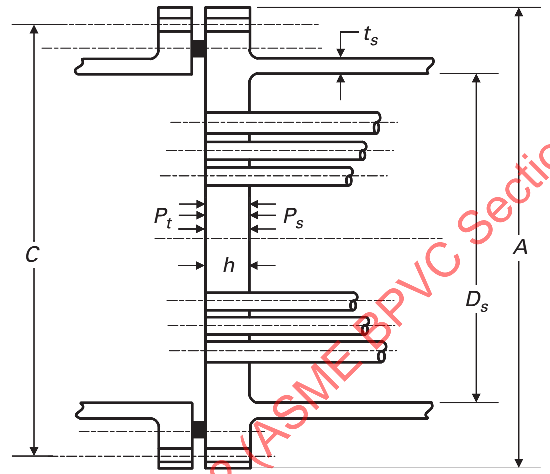
(e) Configuration e:
Tubesheet Gasketed With Shell and Integral With Channel, Extended as a Flange

(f) Configuration f:
Tubesheet Gasketed With Shell and Integral With Channel, Not Extended as a Flange

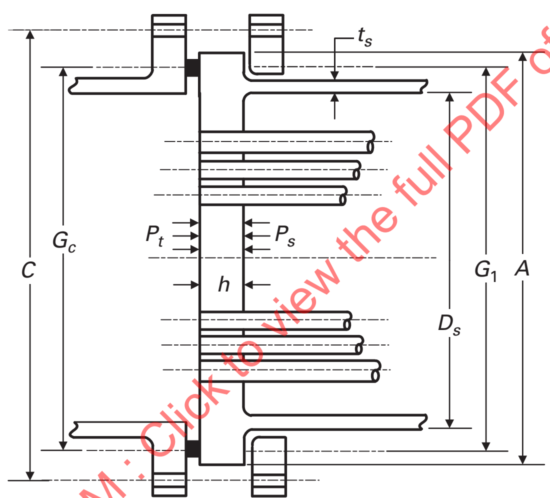
Figure 4.18.5
Fixed Tubesheet Configurations



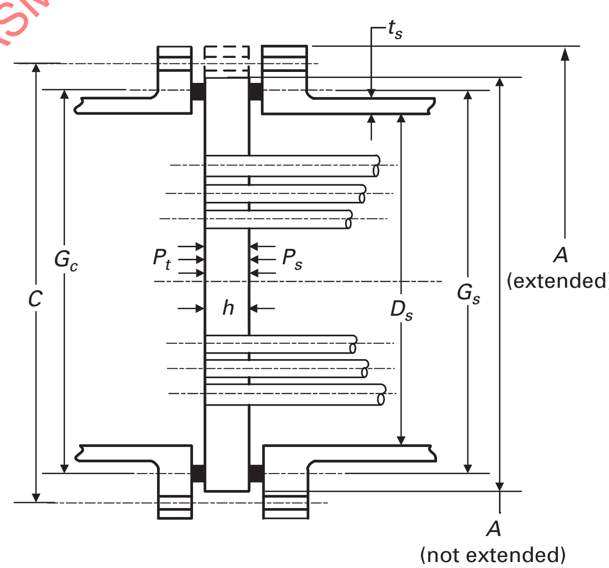
(a) Configuration a:
Tubesheet Integral With Shell and Channel



(b) Configuration b:
Tubesheet Integral With Shell and Gasketed With Channel, Extended as a Flange

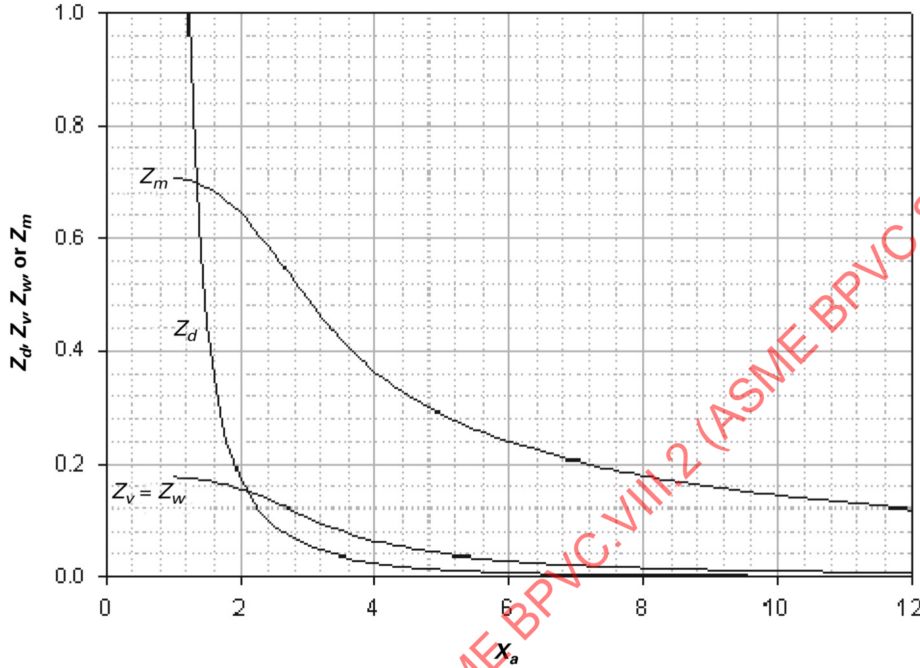


(c) Configuration c:
Tubesheet Integral With Shell and Gasketed With Channel, Not Extended as a Flange



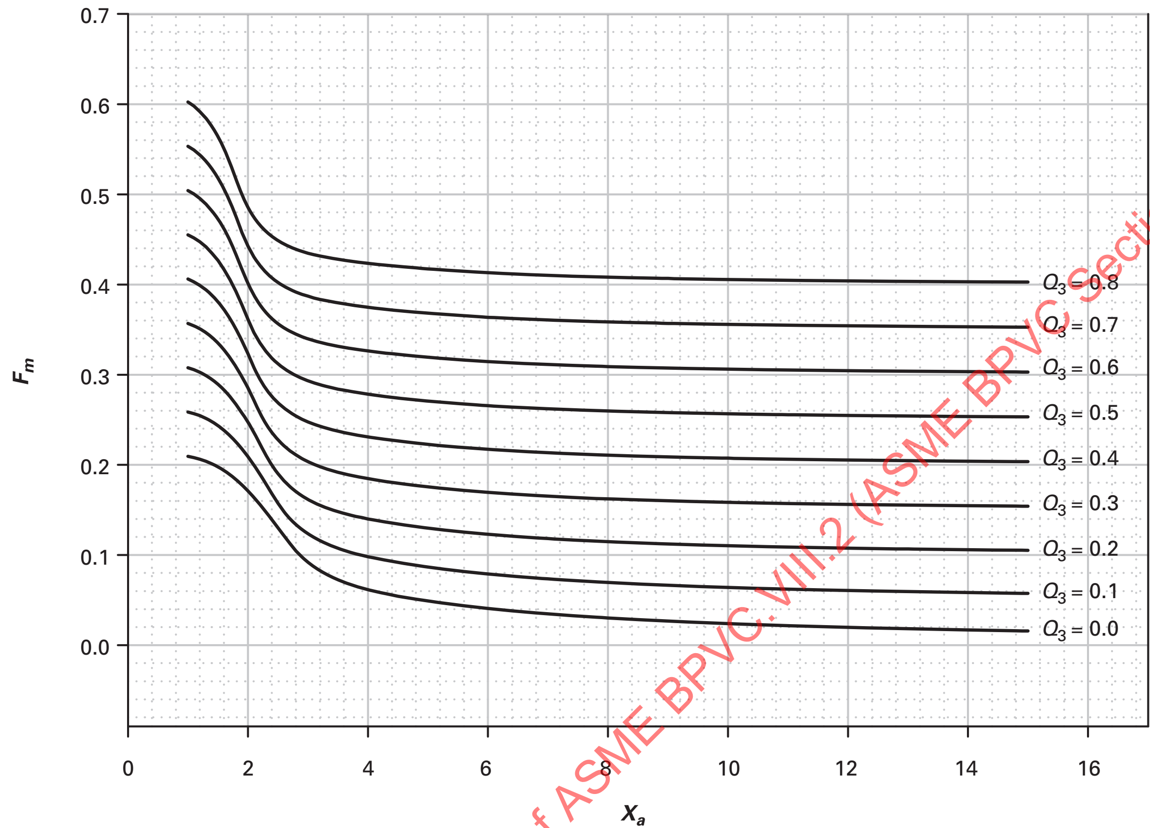
(d) Configuration d:
Tubesheet Gasketed With Shell and Channel

Figure 4.18.6
Z_d, Z_v, Z_w, and Z_m Versus X_a



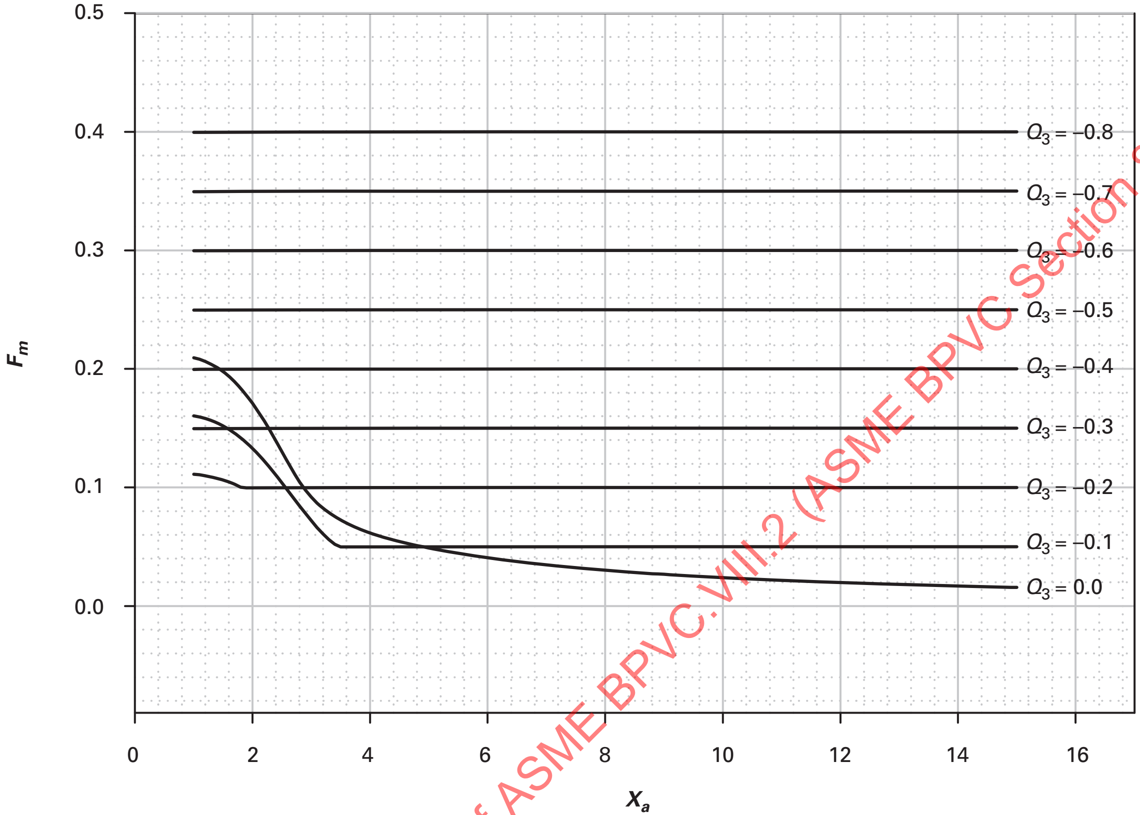
GENERAL NOTES:
(a) The curves for Z_d, Z_v, Z_w, and Z_m are valid for $v^* = 0.4$. These curves are sufficiently accurate for other values of v^*.
(b) If $X_a > 12$, Z_d, Z_v, Z_w, and Z_m shall be calculated using Table 4.18.3.

Figure 4.18.7
F_m Versus X_a ($0.0 \le Q_3 \le 0.8$)



GENERAL NOTES:
(a) The curves for F_m are valid for $v^* = 0.4$. These curves sufficiently accurate for other values of v^*.
(b) If X_a and Q_3 are beyond the values in the curves, F_m shall be calculated using Table 4.18.3.

Figure 4.18.8
F_m Versus X_a ($-0.8 \le Q_3 \le 0.0$)



GENERAL NOTES:
(a) The curves for F_m are valid for $v^* = 0.4$. These curves sufficiently accurate for other values of v^*.
(b) If X_a and Q_3 are beyond the values in the curves, F_m shall be calculated using Table 4.18.3.

Figure 4.18.9
Different Shell Thickness and/or Material Adjacent to the Tubesheets

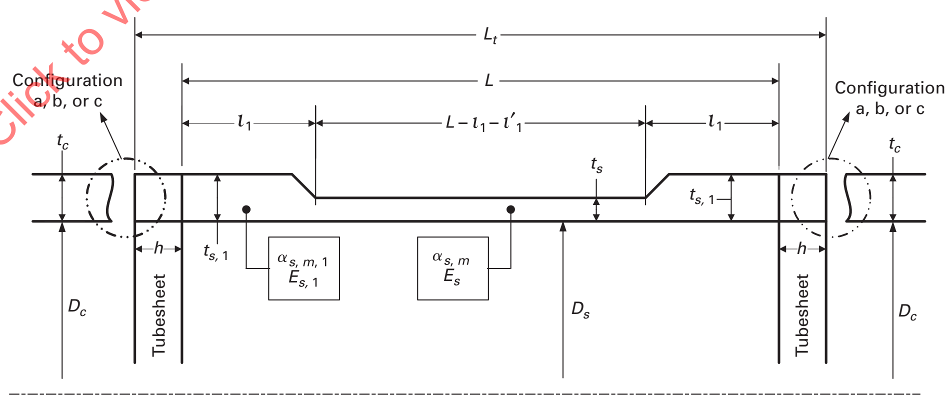
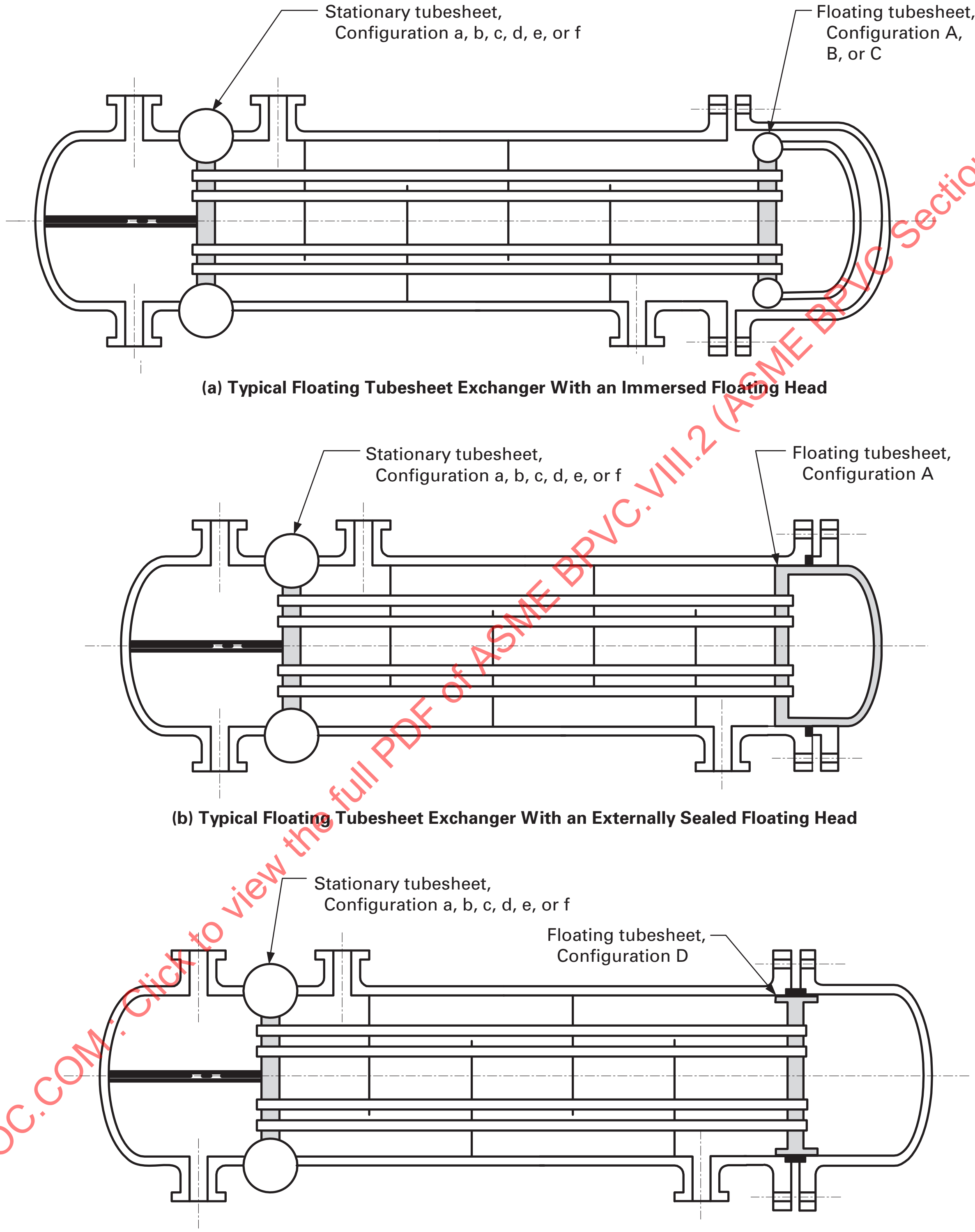


Figure 4.18.10
Floating Tubesheet Heat Exchangers

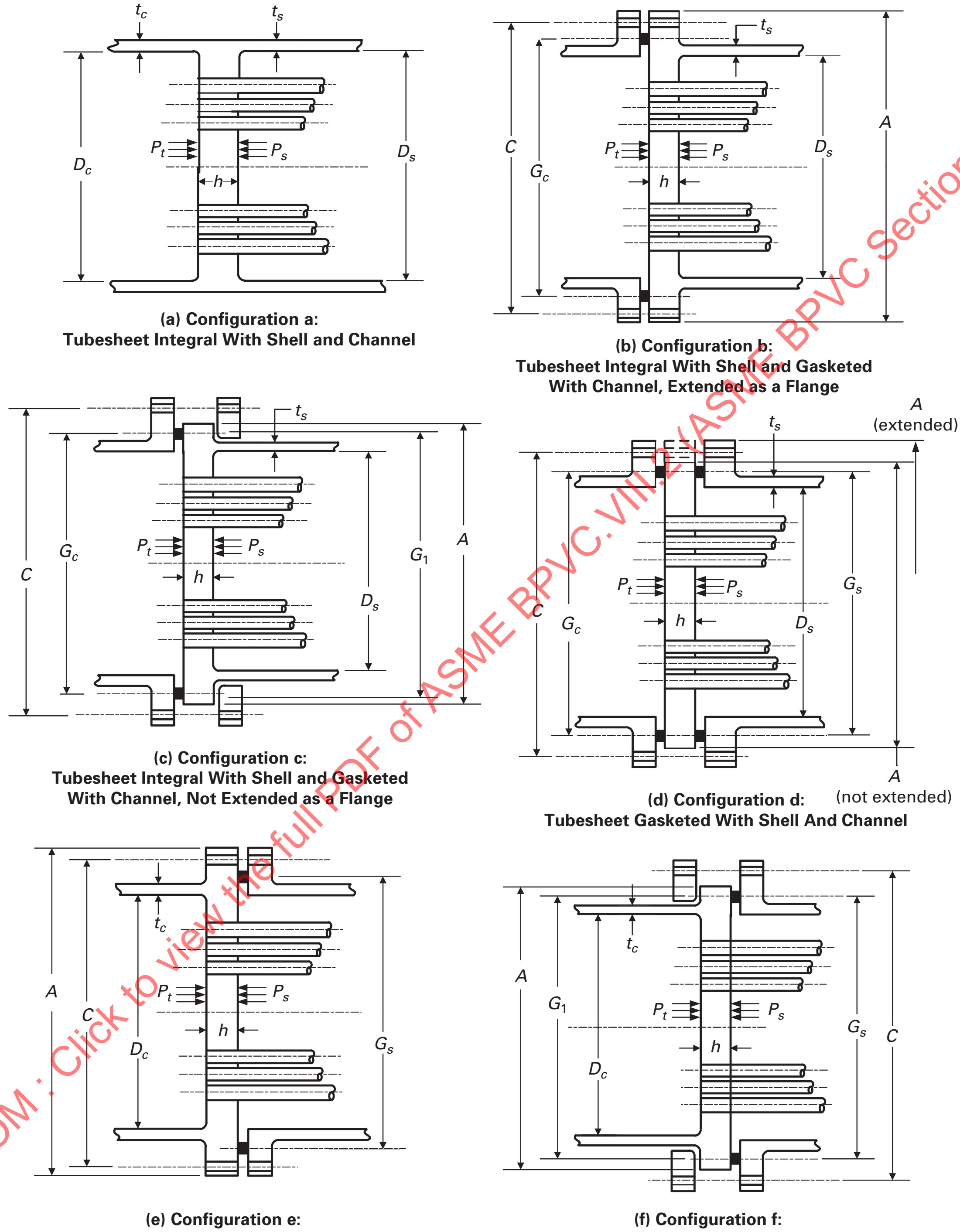


(a) Typical Floating Tubesheet Exchanger With an Immersed Floating Head

(b) Typical Floating Tubesheet Exchanger With an Externally Sealed Floating Head

(c) Typical Floating Tubesheet Exchanger With an Internally Sealed Floating Tubesheet

Figure 4.18.11
Stationary Tubesheet Configurations



(a) Configuration a:
Tubesheet Integral With Shell and Channel

(b) Configuration b:
Tubesheet Integral With Shell and Gasketed With Channel, Extended as a Flange

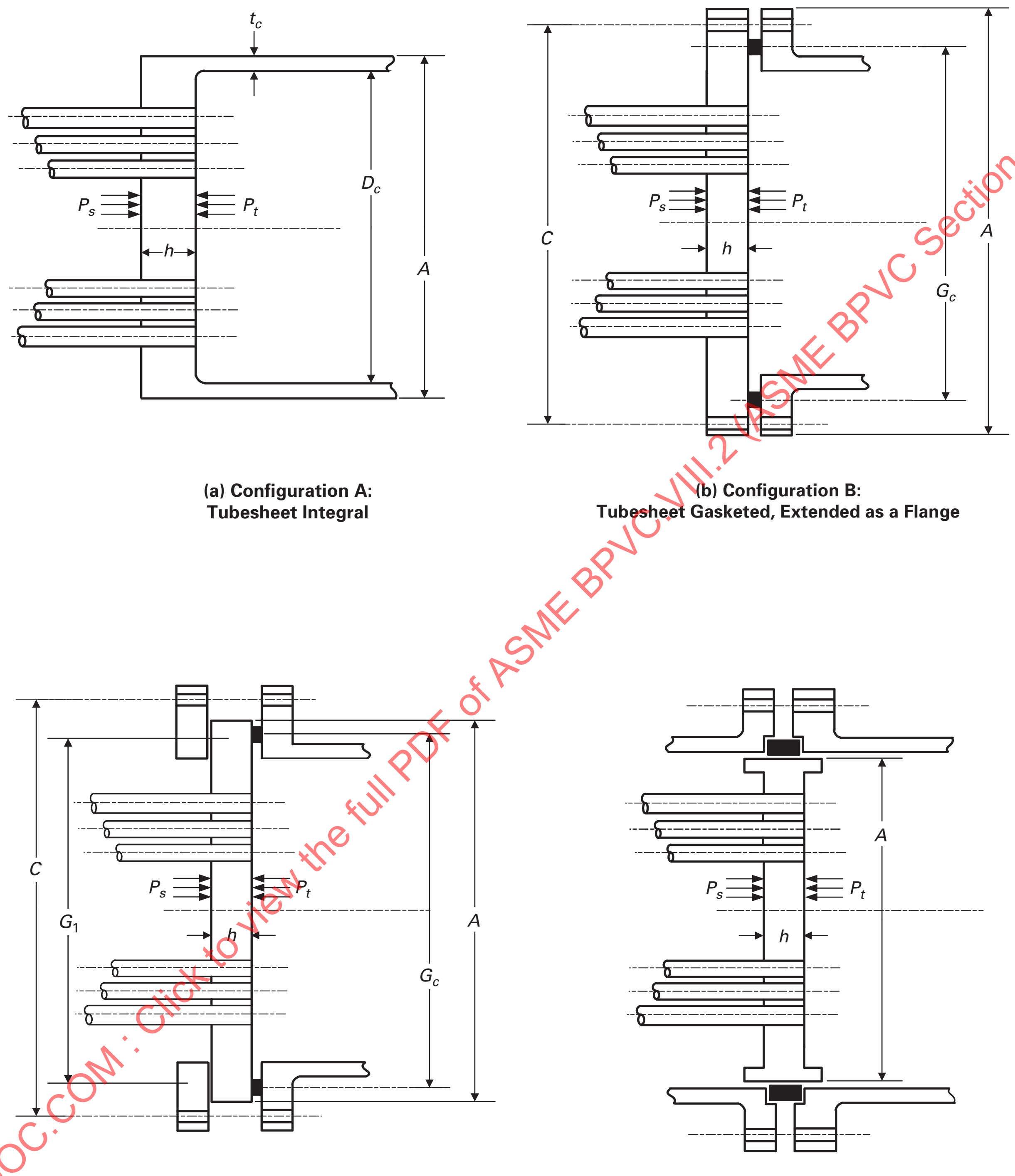
(c) Configuration c:
Tubesheet Integral With Shell and Gasketed With Channel, Not Extended as a Flange

(d) Configuration d:
Tubesheet Gasketed With Shell And Channel

(e) Configuration e:
Tubesheet Gasketed With Shell and Integral With Channel, Extended as a Flange

(f) Configuration f:
Tubesheet Gasketed With Shell and Integral With Channel, Not Extended as a Flange

Figure 4.18.12
Floating Tubesheet Configurations



(a) Configuration A:
Tubesheet Integral

(b) Configuration B:
Tubesheet Gasketed, Extended as a Flange

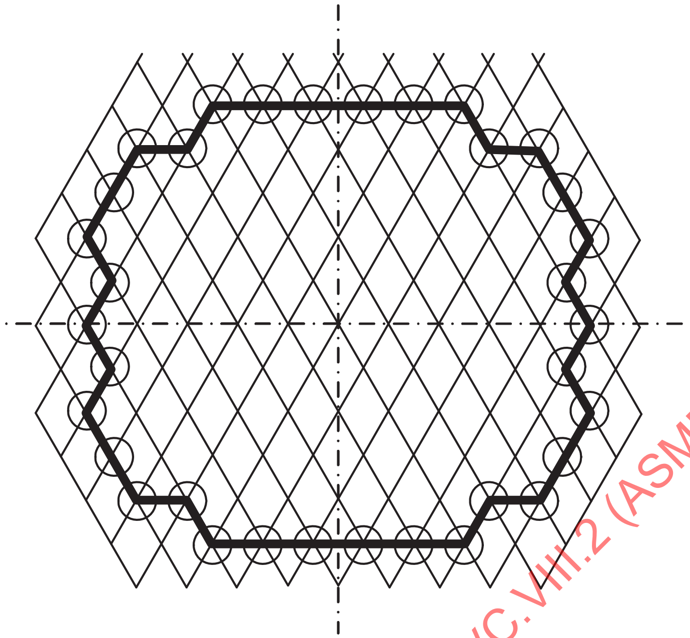
(c) Configuration C:
Tubesheet Gasketed, Not Extended as a Flange

(d) Configuration D:
Tubesheet Internally Sealed

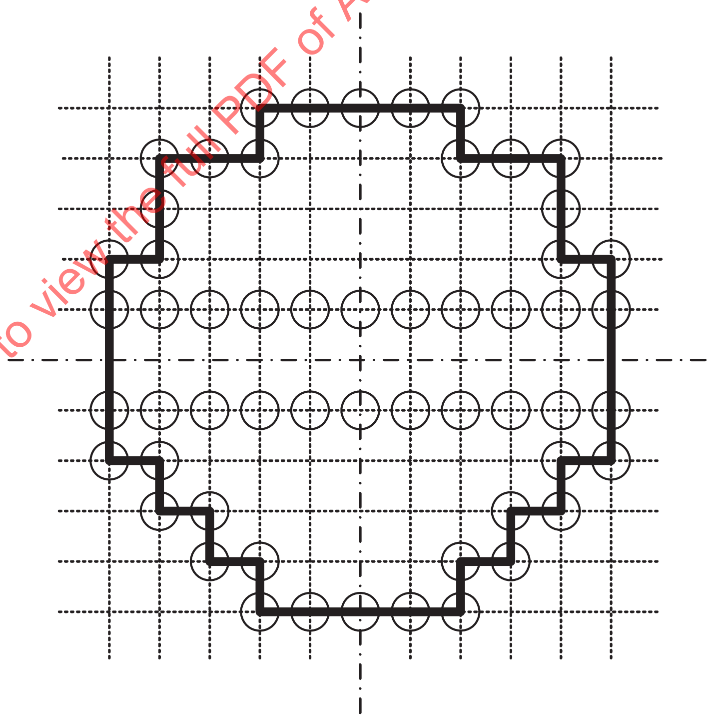
Figure 4.18.13
Some Acceptable Types of Tube-to-Tubesheet Strength Welds (21)

DELETED

Figure 4.18.14
Tube Layout Perimeter



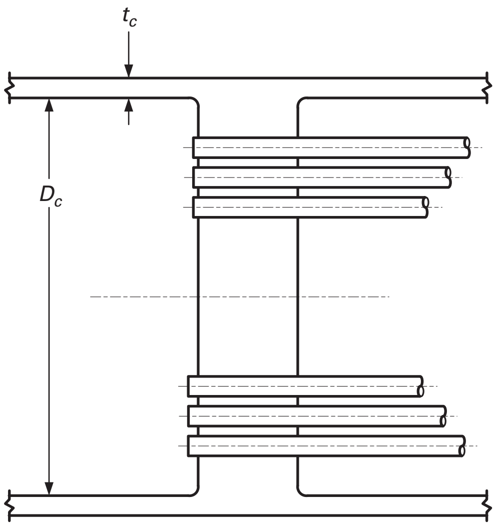
(a) Equilateral Triangular Pattern



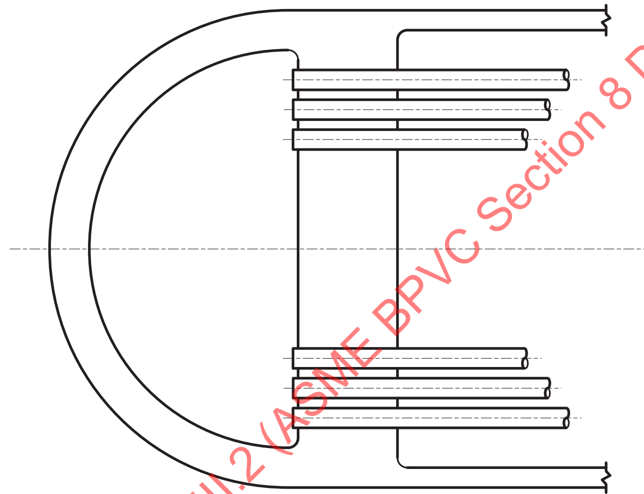
(b) Square Pattern

GENERAL NOTE: C_p (perimeter) is the length of the heavy line.

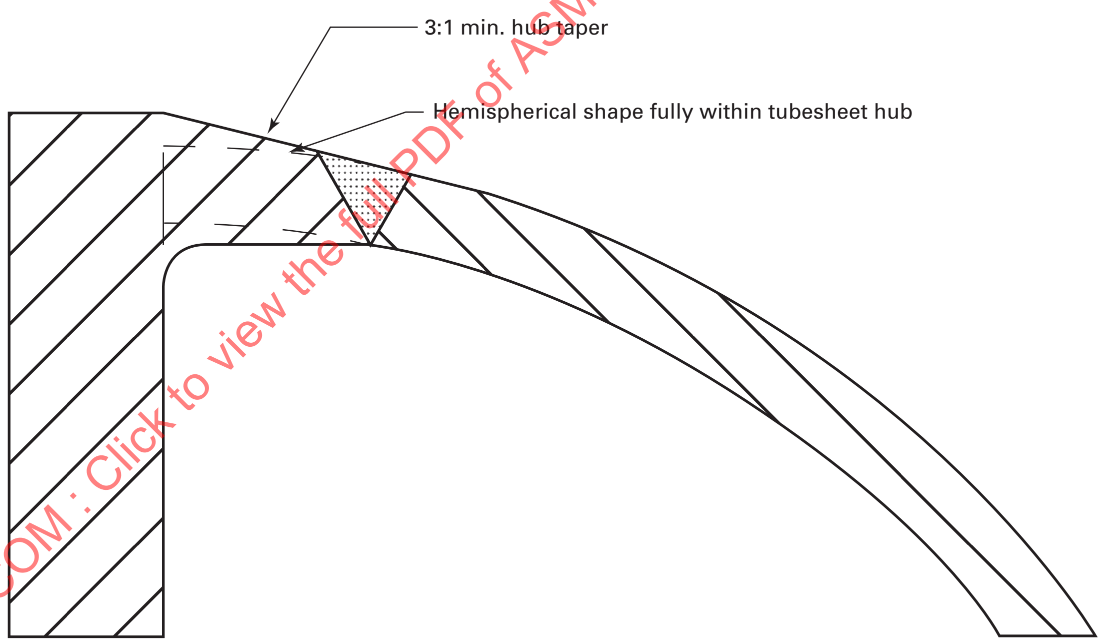
Figure 4.18.15
Integral Channels



(a) Cylindrical Channel [Note (1)]



(b) Hemispherical Channel [Note (2)]



(c) Hemispherical Channel With Tubesheet Hub Thicker Than Channel

NOTES:

(1) Length of the cylinder shall be $\geq 1.8\sqrt{D_c t_c}$.

(2) Head shall be 180 deg with no intervening cylinders. These rules also apply to channels integral with tubesheets having extensions.

Figure 4.18.16
Some Representative Configurations Describing the Minimum Required Thickness of the Tubesheet Flanged Extension, h_r

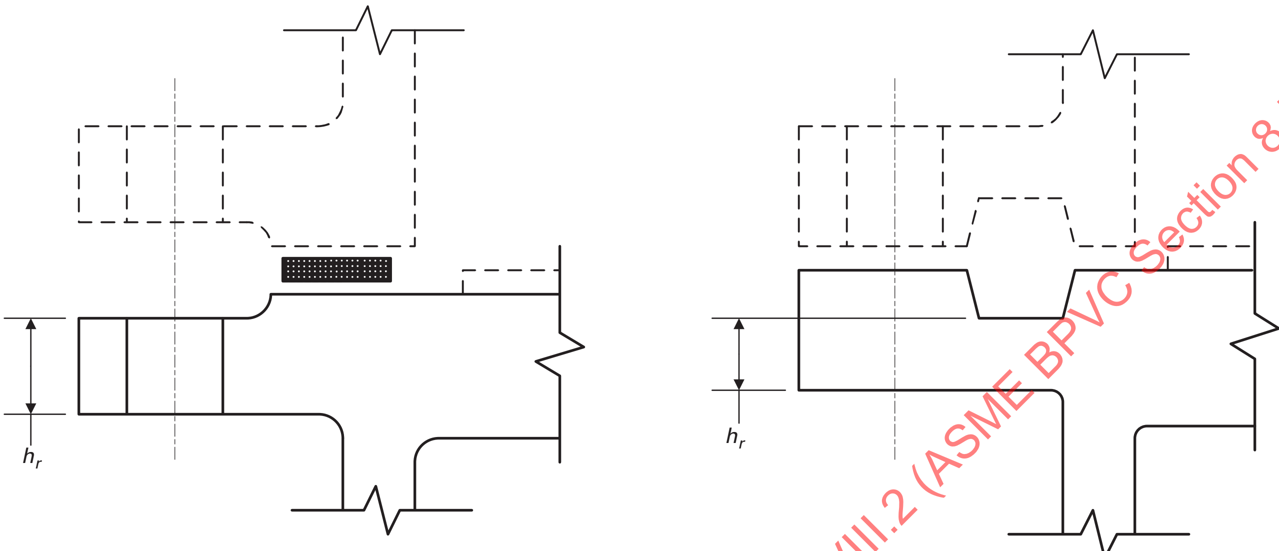


Figure 4.18.17
Kettle Shell

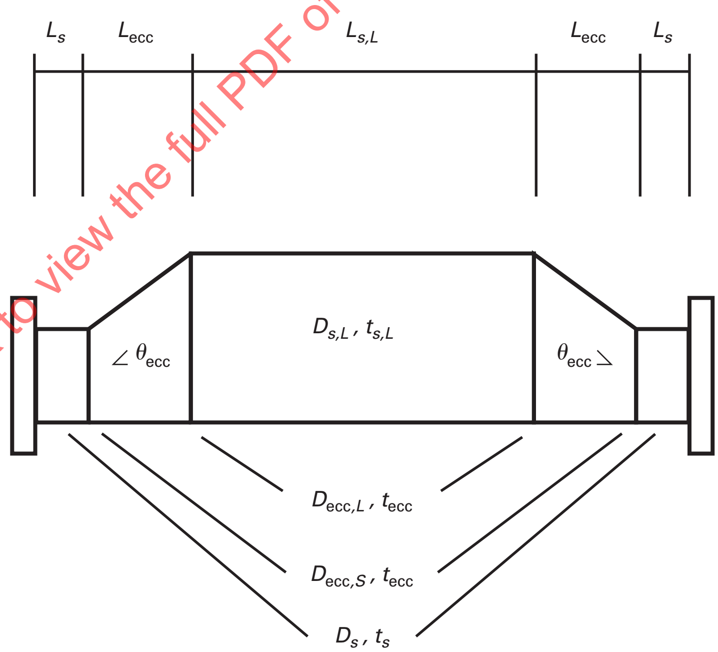
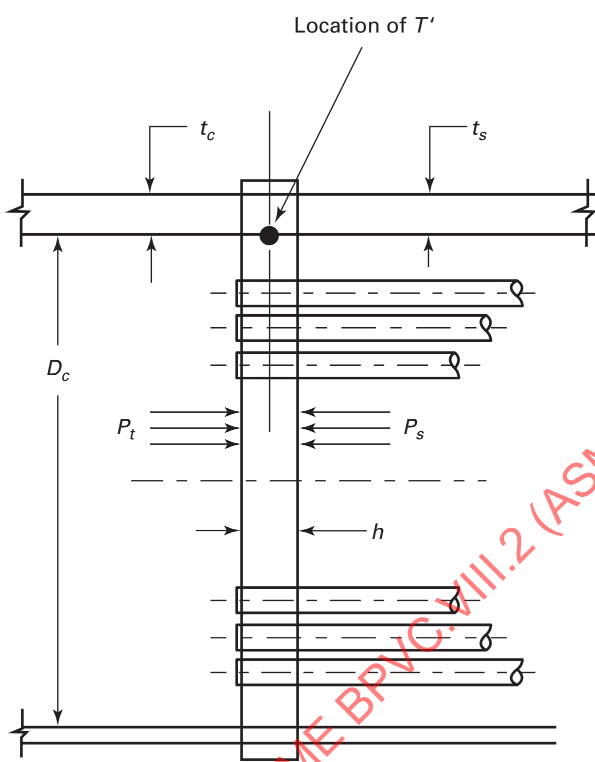


Figure 4.18.18
Location of Tubesheet Metal Temperature, T', at the Rim



4.19 DESIGN RULES FOR BELLOWS EXPANSION JOINTS

4.19.1 SCOPE

(a) The rules in 4.19 cover the minimum requirements for the design of bellows expansion joints used as an integral part of heat exchangers or other pressure vessels. These rules apply to single or multiple layer bellows expansion joints, unreinforced, reinforced or toroidal, as shown in Figure 4.19.1, subject to internal or external pressure and cyclic displacement. The bellows shall consist of single or multiple identically formed convolutions. They may be as formed (not heat treated), or annealed (heat treated). The suitability of an expansion joint for the specified design pressure, temperature, and axial displacement shall be determined by the methods described herein.

(b) The rules in 4.19 cover the common types of bellows expansion joints but are not intended to limit the configurations or details to those illustrated or otherwise described herein. Designs that differ from those covered in 4.19 [e.g., asymmetric geometries or loadings, or temperatures exceeding those in 4.19.2(e)] shall be in accordance with 4.1.1.2.

4.19.2 CONDITIONS OF APPLICABILITY

The design rules of this paragraph are applicable only when the following conditions of applicability are satisfied:

(a) The bellows length shall be such that: $Nq \le 3D_b$.

(b) The bellows nominal thickness shall be such that: $nt \le 5$ mm (0.2 in.).

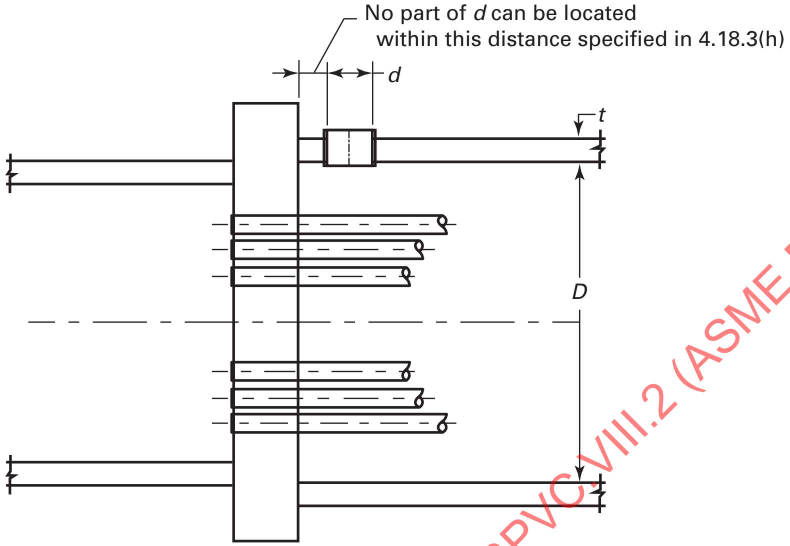
(c) The number of plies shall be such that: $n \le 5$.

(d) The displacement shall be essentially axial. However angular and/or lateral deflection inherent in the fit-up of the expansion joint to the pressure vessel is permissible, provided the amount is specified and is included in the expansion joint design (see 4.19.3.1(d)).

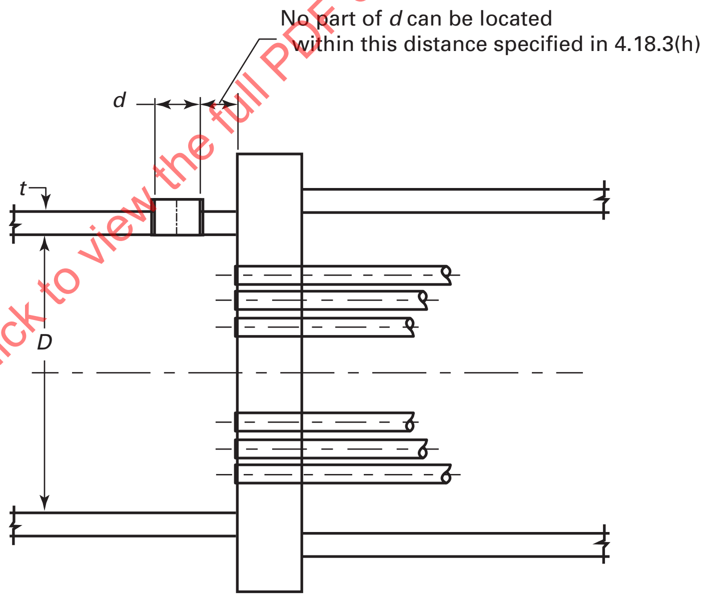
(e) These rules are valid for design temperatures up to the temperatures shown in Table 4.19.1.

(f) The fatigue equations given in Table 4.19.6, Table 4.19.8, and Table 4.19.11 are valid for austenitic chromium-nickel steels, UNS N066XX and UNS N04400. For other materials, the fatigue evaluation shall meet the requirements of 4.19.3.2(c).

Figure 4.18.19
Nozzles Adjacent to Tubesheets



(a)



(b)

(g) The length of the cylindrical shell on each side of the bellows shall not be less than $1.8\sqrt{D_s t_s}$. The length shall be taken from the beginning of the end convolution (point A in Figure 4.19.2) except that for internally attached toroidal bellows, the length shall be taken from the extremity of the shell (point B in Figure 4.19.2).

4.19.3 DESIGN CONSIDERATIONS

4.19.3.1 General.

(a) Expansion joints shall be designed to provide flexibility for thermal expansion and also to function as a pressure-containing element.

(b) The vessel Manufacturer shall specify the design conditions and requirements for the detailed design and manufacture of the expansion joint. Use of Specification Sheet Form 4.19.1 or 4.19.2 is recommended.

(c) In all vessels with integral expansion joints, the hydrostatic end force caused by pressure and/or the joint spring force shall be resisted by adequate restraint elements (e.g., exchanger tubes or shell, external restraints, anchors, etc.). The primary stress in these restraining elements shall satisfy 4.1.6.1.

(d) The expansion joints shall be provided with bars or other suitable members for maintaining the proper overall length dimension during shipment and vessel fabrication. During a heat exchanger pressure test, these bars or members shall not carry load or limit expansion joint movement. Expansion bellows shall not be extended, compressed, rotated, or laterally offset to accommodate connecting parts, which are not properly aligned, unless the design considers such movements (see 4.19.8). Care should be taken to ensure that any torsional loads applied to expansion joints are kept to a minimum to prevent high shear stresses that may be detrimental to their use. If torsional loads are present or expected, they shall be considered in the design (see 4.19.3.3).

(e) The minimum thickness limitations of 4.1.2 do not apply to bellows designed to this paragraph.

(f) Bellows longitudinal weld seams shall have a joint efficiency of 1.0.

(g) Bellows circumferential attachment welds, shell or shell weld ends, and collars shall be in accordance with Figure 4.19.11, as applicable.

(h) The elastic moduli, yield strength, and allowable stresses shall be taken at the design temperatures. However, when performing the fatigue evaluation in accordance with 4.19.5.7 (unreinforced bellows), 4.19.6.7 (reinforced bellows) and 4.19.7.7 (toroidal bellows), it is permitted to use the operating metal temperature instead of the design temperature.

4.19.3.2 Fatigue.

(a) Cumulative Damage - If there are two or more types of stress cycles, which produce significant stresses, their cumulative effect shall be evaluated as given below.

(1) Procedure

(-a) Designate the specified number of times each type of stress cycle of types 1, 2, 3, etc., of stress range S_{t1}, S_{t2}, S_{t3}, etc., will be repeated during the life of the expansion joint as n_1, n_2, n_3, etc., respectively.

(-b) For each value S_{t1}, S_{t2}, S_{t3}, etc., use the applicable design fatigue curve to determine the maximum number of repetitions which would be allowable if this type of cycle were the only one acting. Call these values N_1, N_2, N_3, etc.

(-c) For each type of stress cycle, calculate the usage factors U_1, U_2, U_3 etc., from $U_1 = n_1/N_1$, $U_2 = n_2/N_2$, $U_3 = n_3/N_3$, or for the ith type of stress cycle:

$$U_i = \frac{n_i}{N_i} \tag{4.19.1}$$

(-d) Calculate the cumulative usage factor U by summing the individual factors, or $U = U_1 + U_2 + U_3 + \ldots + U_m$.

(-e) The cumulative usage factor U for the total number of stress cycles, m, shall not exceed 1.0, or

$$\sum_{i=1}^{m} U_i = \sum_{i=1}^{m} \frac{n_i}{N_i} \leq 1.0 \tag{4.19.2}$$

(2) Cycle Counting. Stresses to be used for cycle counting shall be based on the total equivalent axial displacement of each convolution, Δq_e or Δq_c, at the top and bottom of each cycle, as determined in 4.19.8.5, not the range, Δq, determined in 4.19.8.6. Only the displacements shall be taken into account; pressure shall be neglected. The total equivalent axial displacement range, Δq, to be used for the calculation of the total stress range due to cyclic displacement, S_t, in the fatigue evaluation in 4.19.5.7, 4.19.6.7, or 4.19.7.7 shall be deduced from the stress range, S_q, obtained.

(-a) Concurrent Conditions. When determining n_1, n_2, n_3, etc., and S_{q1}, S_{q2}, S_{q3}, etc., consideration shall be given to the superposition of cycles of various origins that produces a total stress range greater than the stress ranges of the individual cycles. For example, if one type of stress cycle produces 1,000 cycles of a stress variation from −1,000 psi to +60,000 psi and another type of stress cycle produces 10,000 cycles of a stress variation from −1,000 psi to −50,000 psi, the two types of cycles to be considered are defined by the following parameters:

(-1) Type 1 cycle: n_1 = 1,000; S_{q1} = |60,000 − (−1,000)| + |−50,000 − (−1,000)| = 110,000 psi

(-2) Type 2 cycle: n_2 = 10,000 − 1,000 = 9,000; S_{q2} = |0| + |−50,000 − (−1,000)| = 49,000 psi

(-b) Independent Conditions. When no superposition of cycles can occur, cycle counting shall be simply based on the stress ranges of the individual cycles. For example, if one type of stress cycle produces 1,000 cycles of a stress variation from −1,000 psi to +60,000 psi and another type of stress cycle produces 10,000 cycles of a stress variation from −1,000 psi to −50,000 psi, the two types of cycles to be considered are defined by the following parameters:

(-1) Type 1 cycle: n_1 = 1,000; S_{q1} = |60,000 − (−1,000)| = 61,000 psi

(-2) Type 2 cycle: n_2 = 10,000; S_{q2} = |−50,000 − (−1,000)| = 49,000 psi

(-c) Alternatively, when the cyclic displacement history is known, cycle counting may be performed by the Rainflow Method described in Annex 5-B or an equivalent method.

(-d) If only the overall number of cycles of each range is known, or in case of doubt, cycle counting shall be performed considering concurrent conditions.

(b) Fatigue Correlation Testing - Fatigue curves in 4.19.5.7, 4.19.6.7, or 4.19.7.7 may be used to design a bellows only if they have been correlated with actual bellows test results obtained by proof or strain gage testing (see Annex 5-F) by the bellows Manufacturer to demonstrate predictability of cyclic life on a consistent series of bellows of the same basic design (convolution shape, reinforcement, number of plies, etc.) and forming process. Annealed and as-formed bellows are considered as separate designs.

(1) The substantiation of the fatigue curves shall be based on data obtained from five separate tests on bellows of the same basic design. When substantiating bellows designs with more than two convolutions in series, the test data shall have been obtained from bellows with a minimum of three convolutions. The effect of pressure shall be considered in the fatigue tests. For each test data pair (S_t, N), two results shall be computed and compared to the applicable fatigue curve: one result with the number of cycles divided by a design factor of 2.6 (S_t, $N/2.6$) and the other result with the equivalent stress divided by a design factor of 1.25 ($S_t/1.25$, N). For a result to be accepted, it must be above the applicable fatigue curve. If all the results meet the acceptance criterion, the substantiation shall be considered valid. If any result does not meet the acceptance criterion, a retest of five additional bellows shall be made. If all the results of the retest, including design factors, meet the acceptance criterion, the substantiation shall be considered valid. Otherwise, a specific fatigue curve shall be established as described in (c) and used for the fatigue design of the bellows. The original test and retest results shall be taken into account to establish the specific fatigue curve.

(2) When S_t along with the other appropriate factors are used in the cycle life equations in 4.19.5.7, 4.19.6.7, or 4.19.7.7, the specified number of fatigue cycles, N_{spe}, shall be less than the calculated number of cycles to failure based on the data obtained by testing. The allowable number of fatigue cycles, N_{alw}, may not be increased above that obtained from the equations in these paragraphs, regardless of the test results.

(3) The test results shall be available for review by the Inspector.

(4) The substantiation of the fatigue curve used by the bellows Manufacturer for a bellows design having shown a history of safe use can be waived, provided that the manufacturing process remains unchanged.

(c) Fatigue Curves for Other Materials - For materials other than those specified in the applicable rules, 4.19.5.7, 4.19.6.7, or 4.19.7.7, specific fatigue curves shall be built. The Manufacturer shall determine the fatigue curve for the material intended for the bellows. This fatigue curve shall not be used for temperatures above the temperature shown in Table 4.19.1 for the tested material. Annealed and as-formed bellows shall be considered as being built with different materials. Different forming methods may have either individual curves established for each method or a single curve established by incorporating test results obtained from at least two bellows formed by each different anticipated forming method. The procedure applied to determine the fatigue curve shall be as described in (1) through (4). The test results with the subsequent calculations used to determine the fatigue curve shall be available for review by the Inspector.

(1) A minimum of 25 fatigue tests shall be carried out. Each bellows in the test group shall have a minimum of three convolutions and varying geometries, including inside diameter, convolution profile, and thickness. A minimum of three different heats of the intended material shall be used.

(2) Each bellows in the test group shall be submitted to three to five different amplitudes of axial movement with a constant internal pressure applied. To ensure that the equivalent fatigue stress, S_t, is due primarily to cyclic displacement and not to pressure, the pressure-induced component stress shall not be higher than 30% of the equivalent fatigue stress.

(3) The test results shall be obtained by proof or strain gage testing (see Annex 5-F) at room temperature.

(4) The fatigue curves shall be determined as follows:

(-a) The best fit curve for the relation between the number of measured cycles to failure and the equivalent fatigue stress, S_t, calculated according to 4.19.5.7, 4.19.6.7, or 4.19.7.7, as applicable, shall be determined and expressed as

$$N = \left(\frac{K_0}{S_t - K'_0}\right)^2 \tag{4.19.3}$$

(-b) The curve shall then be adjusted such that all the test results are on or above the curve. The curve is now expressed as

$$N = \left(\frac{K_1}{S_t - K'_1}\right)^2 \tag{4.19.4}$$

(-c) The final fatigue curve shall be the lower bound curve of the curve obtained by applying a factor of 2.6 on numbers of cycles, expressed as

$$N = \left(\frac{K_2}{S_t - K'_2}\right)^2 \tag{4.19.5}$$

and of the curve obtained by applying a factor of 1.25 on stresses, expressed as

$$N = \left(\frac{K_3}{S_t - K'_3}\right)^2 \tag{4.19.6}$$

where

$K_2 = K_1/\sqrt{2.6}$
$K'_2 = K'_1$
$K_3 = K_1/1.25$
$K'_3 = K'_1/1.25$

4.19.3.3 Torsion. The shear stress due to torsion shall satisfy the criterion in (c) based either on the torsional load or the twist angle:

(a) The shear stress due to torsional load, M_z, is as follows:

$$\tau_z = \frac{2\left|M_z\right|}{\pi n t D_b^2} \tag{4.19.7}$$

(b) The shear stress due to twist angle, θ_z, expressed in radians is as follows:

$$\tau_z = \frac{\left|\theta_z\right| G_b D_b}{2 N L_{dt}} \tag{4.19.8}$$

where

$$G_b = \frac{E_b}{2\left(1 + \nu_b\right)} \tag{4.19.9}$$

For U-shaped bellows, $L_{dt} = A\ /nt_p$ with A and t_p defined in Table 4.19.2.

(c) The shear stress shall satisfy the following equation:

$$\tau_z \le 0.25S \tag{4.19.10}$$

4.19.4 MATERIALS

Pressure-restraining component materials including the restraining elements covered by 4.19.3.1(c) shall comply with the requirements of Part 3.

4.19.5 DESIGN OF U-SHAPED UNREINFORCED BELLOWS

4.19.5.1 Scope. These rules cover the design of bellows having unreinforced U-shaped convolutions. The bellows can be attached to the shell either externally or internally. Each half convolution consists of a sidewall and two quarter tori of nearly the same radius (at the crest and root of the convolution), in the neutral position, so that the convolution profile presents a smooth geometrical shape as shown in Figure 4.19.1, sketch (a).

4.19.5.2 Conditions of Applicability. The following conditions of applicability apply in addition to those listed in 4.19.2.

(a) A variation of 10% between the crest convolution radius r_{ic} and the root convolution radius r_{ir} is permitted (see Figure 4.19.3).

(b) The torus radius shall be such that: $r_i \geq 3t$, where: $r_i = (r_{ic} + r_{ir})/2$. A smaller torus radius may be used, provided that the rules of 4.19.3.2(b) are followed and the increased bending stress due to curvature is accounted for in the fatigue correlation testing.

(c) The off-set angle of the sidewalls, α, in the neutral position shall be such that: $-15 \text{ deg} \leq \alpha \leq +15 \text{ deg}$ (see Figure 4.19.3).

(d) The convolution height shall be such that: $w \leq D_b/3$.

(e) The type of attachment to the shell (external or internal) shall be the same on both sides.

(f) For internally attached bellows, the length of the shell on each side of the bellows having thickness t_s shall be at least equal to $L_{sm} = 1.8\sqrt{D_s t_s}$.

4.19.5.3 Internal Pressure. The required stress calculations and acceptance criteria for U-shaped unreinforced bellows are given in Table 4.19.2.

4.19.5.4 Column Instability Due to Internal Pressure.

(a) The allowable internal design pressure to avoid column instability is given by the following equation.

$$P_{sc} = \frac{0.34\pi K_b}{Nq} \tag{4.19.11}$$

(b) The internal pressure shall satisfy the following equation.

$$P \leq P_{sc} \tag{4.19.12}$$

4.19.5.5 In-Plane Instability Due to Internal Pressure.

(a) The allowable internal design pressure based on in-plane instability is given by the following equation.

$$P_{si} = \frac{AS_y^*(\pi - 2)}{D_m q\left[1 + 2\delta^2 + \left(1 - 2\delta^2 + 4\delta^4\right)^{0.5}\right]^{0.5}} \tag{4.19.13}$$

where

$$\delta = \frac{S_4}{3S_{2,I}} \tag{4.19.14}$$

(b) S_y^* is the effective yield strength at design temperature (unless otherwise specified) of bellows material in the as-formed or annealed condition. If S_y^* is not available in material standards, the values shown in the following equations shall be used where S_y is the yield strength of bellows material at design temperature, given by Annex 3-D. Higher values of S_y^* may be used if justified by representative tests.

$$S_y^* = 2.3S_y \quad \text{for as-formed bellows} \tag{4.19.15}$$

$$S_y^* = 0.75S_y \quad \text{for annealed bellows} \tag{4.19.16}$$

(c) The internal pressure shall satisfy the following equation.

$$P \le P_{si} \tag{4.19.17}$$

4.19.5.6 External Pressure Strength.

(a) External pressure capacity - The rules of 4.19.5.3 shall be applied taking P as the absolute value of the external pressure. When the expansion bellows is subjected to vacuum, the design shall be performed assuming that only the internal ply resists the pressure. The pressure stress equations of 4.19.5.3 shall be applied with $n = 1$.

(b) Instability due to external pressure - The design shall be performed according to the rules of 4.4 by replacing the bellows with an equivalent cylinder, using an equivalent outside diameter, D_{eq}, and an equivalent thickness, t_{eq}, given by the following equations where I_{xx} is the moment of inertia (see Figure 4.19.4).

$$D_{\text{eq}} = D_b + w + 2t_{\text{eq}} \tag{4.19.18}$$

$$t_{\text{eq}} = \left(\frac{12\left(1 - v_b^2\right)I_{xx}}{q}\right)^{1/3} \tag{4.19.19}$$

(c) If $L_t = 0$, I_{xx} is given by the following equation.

$$I_{xx} = nt_p\left[\frac{(2w - q)^3}{48} + 0.4q(w - 0.2q)^2\right] \tag{4.19.20}$$

4.19.5.7 Fatigue Evaluation.

(a) The meridional membrane and bending stresses due to the total equivalent axial displacement range Δq of each convolution is given by the following equations.

$$S_5 = \frac{E_b t_p^2 \Delta q}{2w^3 C_f} \tag{4.19.21}$$

$$S_6 = \frac{5E_b t_p \Delta q}{3w^2 C_d} \tag{4.19.22}$$

C_f and C_d are evaluated using Tables 4.19.4 and 4.19.5, respectively.

(b) The total stress range due to cyclic displacement is given by the following equation.

$$S_t = 0.7\left[S_3 + S_4\right] + \left[S_5 + S_6\right] \tag{4.19.23}$$

(c) Calculation of allowable number of cycles

(1) The specified number of cycles N_{spe} shall be specified in consideration of the anticipated number of cycles expected to occur during the operating life of the bellows. The allowable number of cycles shall satisfy the following equation.

$$N_{\text{alw}} \ge N_{\text{spe}} \tag{4.19.24}$$

(2) The allowable number of cycles, N_{alw}, shall be calculated using the equations in Table 4.19.6. These equations are valid for:

(-a) austenitic chromium-nickel stainless steels, UNS N066XX and UNS N04400, for metal temperatures not exceeding 425°C (800°F)

(-b) U-shaped unreinforced bellows, as-formed or annealed

(-c) basic designs and manufacturing processes that have successfully undergone fatigue correlation testing per 4.19.3.2(b)

(3) For other materials, the allowable number of cycles, N_{alw}, shall be calculated using the equations in Table 4.19.6, replacing the constants with those of curves determined according to 4.19.3.2(c).

(4) If the bellows is subjected to different cycles of pressure or displacement, such as those produced by start-up or shutdown, their cumulative damage shall be considered as in 4.19.3.2(a).

4.19.5.8 Axial Stiffness. The theoretical axial stiffness of a bellows comprised of N convolutions may be evaluated by the following equation. This equation is valid only in the elastic range. Outside of the elastic range lower values can be used, based upon Manufacturer's experience or representative test results.

$$K_b = \frac{\pi E_b D_m}{2\left(1 - \nu_b^2\right) C_f} \left(\frac{n}{N}\right) \left(\frac{t_p}{w}\right)^3 \tag{4.19.25}$$

4.19.6 DESIGN OF U-SHAPED REINFORCED BELLOWS

4.19.6.1 Scope. These rules cover the design of bellows having U-shaped convolutions with rings to reinforce the bellows against internal pressure. The bellows shall be attached to the shell externally. Each half convolution consists of a sidewall and two quarter tori of nearly the same radius (at the crest and root of the convolution), in the neutral position, so that the convolution profile presents a smooth geometrical shape as shown in Figure 4.19.1, sketch (b).

4.19.6.2 Conditions of Applicability. The conditions in 4.19.5.2 apply, with the exception of conditions 4.19.5.2(e) and 4.19.5.2(f).

4.19.6.3 Internal Pressure. The required stress calculations and acceptance criteria for U-shaped reinforced bellows are given in Table 4.19.7.

4.19.6.4 Column Instability Due to Internal Pressure.

(a) The allowable internal design pressure to avoid column instability is given by the following equation.

$$P_{sc} = \frac{0.3\pi K_b}{Nq} \tag{4.19.26}$$

(b) The internal pressure shall satisfy the following equation.

$$P \leq P_{sc} \tag{4.19.27}$$

4.19.6.5 In-Plane Instability Due to Internal Pressure. Reinforced bellows are not prone to in-plane instability.

4.19.6.6 External Pressure Strength.

(a) External pressure capacity - The rules of 4.19.5.3 relative to unreinforced bellows shall be applied taking P as the absolute value of the external pressure. When the expansion bellows is subjected to vacuum, the design shall be performed assuming that only the internal ply resists the pressure. The pressure stress equations of 4.19.5.3 shall be applied with $n = 1$.

(b) Instability due to external pressure - The circumferential instability of a reinforced bellows shall be calculated in the same manner as for unreinforced bellows (see 4.19.5.6(b))

4.19.6.7 Fatigue Evaluation.

(a) The meridional membrane and bending stresses due to the total equivalent axial displacement range Δq of each convolution are given by the following equations.

$$S_5 = \frac{1}{2} \frac{E_b t_p^2}{\left(w - 4C_r r_m\right)^3 C_f} \Delta q \tag{4.19.28}$$

$$S_6 = \frac{5}{3} \frac{E_b t_p}{\left(w - 4C_r r_m\right)^2 C_d} \Delta q \tag{4.19.29}$$

C_f and C_d are evaluated using Tables 4.19.4 and 4.19.5, respectively.

(b) The total stress range due to cyclic displacement is given by eq. (4.19.23).

(c) The fatigue evaluation of a reinforced bellows shall be calculated in the same manner as for unreinforced bellows (see 4.19.5.7(c)), except that the allowable number of cycles shall be calculated using the equations in Table 4.19.8.

4.19.6.8 Axial Stiffness. The theoretical axial stiffness of a bellows comprised of N convolutions is given by the following equation.

$$K_b = \frac{\pi}{2\left(1 - v_b^2\right)} \frac{n}{N} E_b D_m \left(\frac{t_p}{w - 4C_r r_m}\right)^3 \frac{1}{C_f} \tag{4.19.30}$$

This equation is valid only in the elastic range. Outside of the elastic range lower values can be used, based upon Manufacturer's experience or representative test results.

4.19.7 DESIGN OF TOROIDAL BELLOWS

4.19.7.1 Scope. These rules cover the design of bellows having toroidal convolutions. The bellows can be attached to the shell either externally or internally. Each convolution consists of a torus of radius r as shown in Figure 4.19.1, sketch (c).

4.19.7.2 Conditions of Applicability. The following conditions of applicability apply in addition to those listed in 4.19.2:

(a) The type of attachment to the shell (external or internal) shall be the same on both sides.

(b) Distance L_g shall be less than $0.75r$ in the maximum extended position.

(c) For internally attached bellows, the length of the shell on each side of the bellows having thickness t_s shall be at least equal to $L_{sm} = 1.8\sqrt{D_s t_s}$.

4.19.7.3 Internal Pressure. The required stress calculations and acceptance criteria for toroidal bellows are given in Table 4.19.9.

4.19.7.4 Column Instability Due to Internal Pressure.

(a) The allowable internal design pressure to avoid column instability is given by the following equation.

$$P_{sc} = \frac{0.15\pi K_b}{Nr} \tag{4.19.31}$$

(b) The internal pressure shall satisfy the following equation.

$$P \le P_{sc} \tag{4.19.32}$$

4.19.7.5 In-Plane Instability Due to Internal Pressure. Toroidal bellows are not prone to in-plane instability.

4.19.7.6 External Pressure Strength.

(a) Toroidal bellows designed per the rules of this Division are suitable for external design pressures up to 103 kPa (15 psi) or full vacuum. For external design pressures greater than 103 kPa (15 psi), see 4.1.1.2.

(b) Instability due to external pressure is not covered by the present rules.

4.19.7.7 Fatigue Evaluation.

(a) The meridional membrane and bending stresses due to the total equivalent axial displacement range Δq of each convolution are given by the following equations.

$$S_5 = \frac{E_b t_p^2 B_1 \Delta q}{34.3 r^3} \tag{4.19.33}$$

$$S_6 = \frac{E_b t_p B_2 \Delta q}{5.72 r^2} \tag{4.19.34}$$

B_1 and B_2 are evaluated using Table 4.19.10.

(b) The total stress range due to cyclic displacement is given by the following equation.

$$S_t = 3S_3 + S_5 + S_6 \tag{4.19.35}$$

(c) The fatigue evaluation of a toroidal bellows shall be calculated in the same manner as for unreinforced bellows [see 4.19.5.7(c)], except that the allowable number of cycles shall be calculated using the equations in Table 4.19.11.

4.19.7.8 Axial Stiffness. The theoretical axial stiffness of a bellows comprised of N convolutions is given by the following equation.

$$K_b = \frac{E_b D_m B_3}{12\left(1 - {v_b}^2\right)}\left(\frac{n}{N}\right)\left(\frac{t_p}{r}\right)^3 \tag{4.19.36}$$

B_3 is evaluated using Table 4.19.10. This equation is valid only in the elastic range. Outside of the elastic range lower values can be used, based upon Manufacturer's experience or representative test results.

4.19.8 BELLOWS SUBJECTED TO AXIAL, LATERAL, OR ANGULAR DISPLACEMENTS

4.19.8.1 General. The purpose of this paragraph is to determine the equivalent axial displacement of an expansion bellows subjected at its ends to:

(a) an axial displacement from the neutral position: x in extension ($x > 0$) or in compression ($x < 0$)

(b) a lateral deflection from the neutral position: y, ($y > 0$)

(c) an angular rotation from the neutral position: θ, ($\theta > 0$)

4.19.8.2 Axial Displacement.

(a) When the ends of the bellows are subjected to an axial displacement x (see Figure 4.19.5), the equivalent axial displacement per convolution is given by the following equation. In this equation, x shall be taken as positive for extension ($x > 0$) and negative for compression ($x < 0$). Values of x in extension and compression may be different.

$$\Delta q_x = \frac{x}{N} \tag{4.19.37}$$

(b) The corresponding axial force F_x applied to the ends of the bellows is given by the following equation.

$$F_x = K_b x \tag{4.19.38}$$

4.19.8.3 Lateral Deflection.

(a) When the ends of the bellows are subjected to a lateral deflection y (see Figure 4.19.6), the equivalent axial displacement per convolution is given by the following equation where y shall be taken positive.

$$\Delta q_y = \frac{3\ D_m y}{N\left(Nq + x\right)} \tag{4.19.39}$$

(b) The corresponding lateral force F_y applied to the ends of the bellows is given by the following equation.

$$F_y = \frac{3K_b D_m^2 y}{2\left(Nq + x\right)^2} \tag{4.19.40}$$

(c) The corresponding moment M_y applied to the ends of the bellows is given by the following equation.

$$M_y = \frac{3K_b D_m^2 y}{4\left(Nq + x\right)} \tag{4.19.41}$$

4.19.8.4 Angular Rotation.

(a) When the ends of the bellows are subjected to an angular rotation θ (see Figure 4.19.7), the equivalent axial displacement per convolution is given by the following equation where θ, expressed in radians, shall be taken positive.

$$\Delta_{q\theta} = \frac{D_m \theta}{2\ N} \tag{4.19.42}$$

(b) The corresponding moment M_θ applied to the ends of the bellows is given by the following equation.

$$M_\theta = \frac{K_b D_m^2 \theta}{8} \tag{4.19.43}$$

4.19.8.5 Total Equivalent Axial Displacement Per Convolution. Axial displacement leads to uniform deformation of the convolutions. Lateral deflection and angular rotation lead to nonuniform deformation of the convolutions with one side extended and the other side compressed as shown in Figures 4.19.6 and 4.19.7. The total equivalent axial displacements per convolution, on the extended side and the compressed side, are given by the following equations:

$$\Delta q_e = \Delta q_x + \Delta q_y + \Delta q_\theta \qquad \text{(extended side)} \tag{4.19.44}$$

$$\Delta q_c = \Delta q_x - \Delta q_y - \Delta q_\theta \qquad \text{(compressed side)} \tag{4.19.45}$$

NOTE: In case of axial displacement only, $\Delta q_e = \Delta q_c = \Delta q_x$.

4.19.8.6 Total Equivalent Axial Displacement Range Per Convolution. (21)

(a) Bellows installed without cold spring - If the bellows is subjected to displacements from the neutral position ($x_0 = 0, y_0 = 0, \theta_0 = 0$) to the operating position (x_1, y_1, θ_1) (see Figure 4.19.8), the total equivalent axial displacements per convolution, on the extended side and the compressed side, for the initial and operating positions and the total equivalent axial displacement range are given by the following equations:

Initial Position:

$$\Delta q_{e,0} = 0.0 \qquad \text{(extended side)} \tag{4.19.46}$$

$$\Delta q_{c,0} = 0.0 \qquad \text{(compressed side)} \tag{4.19.47}$$

Operating Position:

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \qquad \text{(extended side)} \tag{4.19.48}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \qquad \text{(compressed side)} \tag{4.19.49}$$

Total Equivalent Axial Displacement Range:

$$\Delta q = \max\left[\left|\Delta q_{e,1}\right|, \left|\Delta q_{c,1}\right|\right] \tag{4.19.50}$$

NOTE: In case of axial displacement only, $\Delta q = |\Delta q_{x,1}|$.

(b) Bellows installed with cold spring - If the bellows is subjected to displacements from an initial position (x_0, y_0, θ_0), which is not the neutral position to the operating position (x_1, y_1, θ_1) (see Figure 4.19.9), the total equivalent axial displacements per convolution, on the extended side and the compressed side, for the initial and operating positions and the total equivalent axial displacement range are given by the following equations:

Initial Position:

$$\Delta q_{e,0} = \Delta q_{x,0} + \Delta q_{y,0} + \Delta q_{\theta,0} \qquad \text{(extended side)} \tag{4.19.51}$$

$$\Delta q_{c,0} = \Delta q_{x,0} - \Delta q_{y,0} - \Delta q_{\theta,0} \qquad \text{(compressed side)} \tag{4.19.52}$$

Operating Position:

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \qquad \text{(extended side)} \tag{4.19.53}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \qquad \text{(compressed side)} \tag{4.19.54}$$

Total Equivalent Axial Displacement Range:

$$\Delta q = \max\left[\left|\Delta q_{e,1} - \Delta q_{c,0}\right|, \left|\Delta q_{c,1} - \Delta q_{e,0}\right|\right] \tag{4.19.55}$$

Alternatively, if the neutral position for lateral deflection and angular rotation is not passed between the initial position and the operating position, the total equivalent axial displacement range may be written as

$$\Delta q = \max\left[\left|\Delta q_{e,1} - \Delta q_{e,0}\right|, \left|\Delta q_{c,1} - \Delta q_{c,0}\right|\right] \tag{4.19.56}$$

NOTE: In case of axial displacement only, $\Delta q = |\Delta q_{x,1} - \Delta q_{x,0}|$.

(c) Bellows operating between two operating positions - If the bellows is subjected to displacements from operating position number 1 (x_1, y_1, θ_1) to the operating position number 2 (x_2, y_2, θ_2) (see Figure 4.19.10), the total equivalent axial displacements per convolution, on the extended side and the compressed side, for operating positions number 1 and 2 and the total equivalent axial displacement range are given by the following equations. An initial cold spring (initial position 0) has no effect on the results.

Position Number 1:

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \qquad \text{(extended side)} \tag{4.19.57}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \qquad \text{(compressed side)} \tag{4.19.58}$$

Position Number 2:

$$\Delta q_{e,2} = \Delta q_{x,2} + \Delta q_{y,2} + \Delta q_{\theta,2} \qquad \text{(extended side)} \tag{4.19.59}$$

$$\Delta q_{c,2} = \Delta q_{x,2} - \Delta q_{y,2} - \Delta q_{\theta,2} \qquad \text{(compressed side)} \tag{4.19.60}$$

Total Equivalent Axial Displacement Range:

$$\Delta q = \max\left[\left|\Delta q_{e,2} - \Delta q_{c,1}\right|, \left|\Delta q_{c,2} - \Delta q_{e,1}\right|\right] \tag{4.19.61}$$

Alternatively, if the neutral position for lateral deflection and angular rotation is not passed between operating positions 1 and 2, the total equivalent axial displacement range may be written as

$$\Delta q = \max\left[\left|\Delta q_{e,2} - \Delta q_{e,1}\right|, \left|\Delta q_{c,2} - \Delta q_{c,1}\right|\right] \tag{4.19.62}$$

NOTE: In case of axial displacement only, $\Delta q = |\Delta q_{x,2} - \Delta q_{x,1}|$.

4.19.9 PRESSURE TEST DESIGN REQUIREMENTS

The designer shall consider the possibility of instability of the bellows due to internal pressure if the test pressure exceeds the value determined using the following applicable equation. In such a case, the designer shall redesign the bellows to satisfy the test condition.

For unreinforced bellows

$$P_{t,s} = 1.5 \min(P_{sc}, P_{si}) \tag{4.19.63}$$

For reinforced and toroidal bellows

$$P_{t,s} = 1.5\, P_{sc} \tag{4.19.64}$$

4.19.10 MARKING AND REPORTS

(a) The expansion joint Manufacturer, whether the vessel Manufacturer or a parts Manufacturer, shall have a valid ASME Code U2 Certificate of Authorization and shall complete the appropriate Data Report in accordance with Part 2.

(b) The Manufacturer responsible for the expansion joint design shall include the following additional data and statements on the appropriate Data Report:

(1) Axial movement (±), associated design life in cycles, and associated loading condition, if applicable;

(2) Spring rate; and

(3) That the expansion joint has been constructed to the rules of this paragraph.

(c) A parts Manufacturer shall identify the vessel for which the expansion joint is intended on the Partial Data Report.

(d) Markings shall not be stamped on the flexible elements of the expansion joint.

4.19.11 NOMENCLATURE

A = cross-sectional metal area of one convolution.
A_f = cross-sectional metal area of one reinforcing fastener
A_r = cross-sectional metal area of one bellows reinforcing member for U-shaped bellows and cross-sectional metal area of one reinforcing collar for toroidal bellows based on length L_r
A_{rt} = cross-sectional metal area of one reinforcing collar for toroidal bellows based on overall length
A_{tc} = cross-sectional metal area of one tangent collar
A_{ts} = cross-sectional metal area of shell based on length L_s
B_1, B_2, B_3 = stress and stiffness coefficients used for toroidal bellows.
C_1, C_2 = coefficients used to determine the coefficients C_p, C_f, C_d
C_3 = coefficient used to determine the coefficients B_1, B_2, B_3
C_p, C_f, C_d = stress coefficients for U-shaped convolutions.
C_r = convolution height factor for reinforced bellows.
C_{wc} = longitudinal weld joint efficiency of tangent collar.
C_{wr} = longitudinal weld joint efficiency of reinforcing member.
C_{ws} = longitudinal weld joint efficiency of shell.
D_b = inside diameter of bellows convolution and end tangents.
D_c = mean diameter of tangent collar.
D_{eq} = equivalent outside diameter.
D_m = mean diameter of bellows convolution.
D_r = mean diameter of reinforcing collar for toroidal bellows.
D_s = inside diameter of cylindrical shell or weld end on which the bellows is attached.
E_0 = modulus of elasticity of bellows material at room temperature.
E_b = modulus of elasticity of bellows material at design temperature.
E_c = modulus of elasticity of collar material at design temperature.
E_f = modulus of elasticity of reinforcing fastener material at design temperature.
E_r = modulus of elasticity of reinforcing ring member material at design temperature.
E_s = modulus of elasticity of shell or weld end material at design temperature
G_b = modulus of rigidity of bellows material at design temperature.
I_{xx} = moment of inertia of one convolution cross section relative to the axis passing by the center of gravity and parallel to the axis of the bellows.
k = factor considering the stiffening effect of the attachment weld and the end convolution on the pressure capacity of the end tangent
K_0, K_1, K_2, K_3 = coefficients determined by best curve fit of bellows fatigue test data.
K'_0, K'_1, K'_2, K'_3 = coefficients determined by best curve fit of bellows fatigue test data.
K_b = bellows axial stiffness.
K_f = forming method factor
= 1.0 for expanding mandrel or roll forming
= 0.6 for hydraulic, elastomeric, or pneumatic tube forming
K_g = fatigue strength reduction factor.
L_c = bellows collar length.
L_d = length from attachment weld to the center of the first convolution for reinforced bellows and for externally attached toroidal bellows.
L_{dt} = developed length of one convolution.
L_f = effective length of one reinforcing fastener. Distance between mating face of the bolt head and mid-thickness of the nut or distance between mid-thickness of the two nuts, as applicable.
L_g = maximum distance across the inside opening of a toroidal convolution considering all movements.
L_r = effective reinforcing collar length.

$$= \frac{1}{3}\sqrt{D_r t_r}$$

L_{rt} = overall length of reinforcing collar.
L_s = effective shell length.

$$= \frac{1}{3}\sqrt{\left(D_s + t_s\right)t_s}$$

L_{sm} = minimum required shell length having thickness t_s.
L_t = end tangent length.
M_z = torsional load.
N = number of convolutions.
n = number of plies.
N_{alw} = allowable number of fatigue cycles.
N_{spe} = specified number of fatigue cycles.
P = design pressure.

NOTE: If the specified design pressure of the bellows is significantly greater than the required design pressure of the vessel, use of the larger specified design pressure may adversely affect the allowable number cycles that the bellows can experience.

P_{sc} = allowable internal design pressure to avoid column instability.
P_{si} = allowable internal design pressure to avoid in-plane instability.
q = convolution pitch as shown in Figure 4.19.1.
r = mean radius of toroidal bellows convolution.
R = ratio of the internal pressure force resisted by the bellows to the internal pressure force resisted by the reinforcement
r_i = average internal torus radius of U-shaped bellows convolution (see 4.19.5.2).
r_{ic} = crest convolution internal radius.
r_{ir} = root convolution internal radius.
r_m = mean radius of U-shaped bellows convolution.
S = allowable stress from Annex 3-A of bellows material at design temperature.
S_y^* = effective yield strength of the bellows material at the design temperature.
S_1 = circumferential membrane stress in bellows tangent, due to pressure P.
S_2 = circumferential membrane stress in bellows, due to pressure P.
S_3 = meridional membrane stress in bellows, due to pressure P.
S_4 = meridional bending stress in bellows, due to pressure P.
S_5 = meridional membrane stress in bellows, due to total equivalent axial displacement range Δq.
S_6 = meridional bending stress in bellows, due to total equivalent axial displacement range Δq.
S_c = allowable stress from Annex 3-A of collar material at design temperature.
S_f = allowable stress from Annex 3-A of reinforcing fastener material at design temperature.
S_q = total stress range due to cyclic displacement.
S_r = allowable stress from Annex 3-A of reinforcing ring member material at design temperature.
S_s = allowable stress from Annex 3-A of shell material at design temperature.
S_t = total stress range due to cyclic displacement corrected by internal pressure.
S_y = yield strength of the bellows material at the design temperature.
S_1' = circumferential membrane stress in collar, due to pressure P.
S_1''' = circumferential membrane stress in shell, due to pressure P, for internally attached bellows.
S_2' = circumferential membrane stress in reinforcing member, due to pressure P.
S_2'' = membrane stress in fastener, due to pressure P.
t = nominal thickness of one ply.
t_c = collar thickness.
t_{eq} = equivalent wall thickness
t_p = thickness of one ply, corrected for thinning during forming.
t_r = reinforcing collar thickness.
t_s = nominal thickness of shell or weld end.
U = usage factor.
ν_b = Poisson's ratio of bellows material.
w = convolution height.
Y_{sm} = yield strength multiplier depending upon material.
Δq = total equivalent axial displacement range per convolution.
ε_f = bellows forming strain.
α = offset angle of the U-shape bellows side-wall.
θ_z = twist angle between the two extreme points of the end convolutions.
τ_z = shear stress due to torsional load or twist angle.

4.19.12 TABLES

Table 4.19.1
Maximum Design Temperatures for Application of the Rules of 4.19

Table in Which Material Is Listed	Maximum Temperature, °C	Maximum Temperature, °F
3-A.3	425	800
3-A.4	150	300
3-A.6	425	800
3-A.7	315	600

Table 4.19.2
Stress Calculations and Acceptability Criteria for U-Shaped Unreinforced Bellows Subject to Internal Pressure

Stress Calculation

Circumferential membrane stress in end tangent for externally attached bellows

$$S_1 = \frac{P\left(D_b + nt\right)^2 L_t E_b k}{2\left[nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k\right]}$$

Circumferential membrane stress in collar for externally attached bellows

$$S_1' = \frac{P D_c^2 L_t E_c k}{2\left[nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k\right]}$$

Circumferential membrane stress in shell for internally attached bellows

$$S_1''' = \frac{1}{2}\frac{\left(D_s + t_s\right)^2\left(L_s + 0.5q\right)E_s}{nt\left(D_b + nt\right)L_t E_b + t_s\left(D_s + t_s\right)L_s E_s}P$$

Circumferential membrane stress in bellows convolutions:
for end convolutions of externally attached bellows when k is less than 1.0

$$S_{2,E} = \frac{1}{2}\frac{\left[qD_m + L_t\left(D_b + nt\right)\right]E_b}{\left(A + ntL_t\right)E_b + t_c L_c E_c}P$$

for intermediate convolutions

$$S_{2,I} = \frac{PqD_m}{2A}$$

Meridional membrane stress

$$S_3 = \frac{Pw}{2nt_p}$$

Meridional bending stress

$$S_4 = \frac{PC_p}{2n}\left(\frac{w}{t_p}\right)^2$$

where

$$A = \left[2\pi r_m + 2\sqrt{\left(\frac{q}{2} - 2r_m\right)^2 + \left(w - 2r_m\right)^2}\right]nt_p$$

$$r_m = r_i + \frac{nt}{2}$$

$$D_c = D_b + 2nt + t_c$$

$$D_m = D_b + w + nt$$

$$k = \min\left[\left(\frac{L_t}{1.5\sqrt{D_b t}}\right), 1.0\right] \qquad t_p = t\sqrt{\frac{D_b}{D_m}}$$

Acceptance Criteria

$S_1 \le S$	for externally attached bellows
$S_1' \le C_{wc}S_c$	for externally attached bellows
$S_1''' \le C_{ws}S_s$	for internally attached bellows
$S_{2,E} \le$	for externally attached bellows
$S_{2,I} \le S$	
$S_3 + S_4 \le K_m S$	

GENERAL NOTES:
(a) $K_m = 1.5Y_{sm}$ for as-formed bellows and $K_m = 1.5$ for annealed bellows.
(b) $Y_{sm} = 1 + 9.94(K_f\varepsilon_f) - 7.59(K_f\varepsilon_f)^2 - 2.4(K_f\varepsilon_f)^3 + 2.21(K_f\varepsilon_f)^4$ for austenitic stainless steel.

Table 4.19.2
Stress Calculations and Acceptability Criteria for U-Shaped Unreinforced Bellows Subject to Internal Pressure (Cont'd)

GENERAL NOTES (CONT'D):

$Y_{sm} = 1 + 6.8(K_f\varepsilon_f) - 9.11(K_f\varepsilon_f)^2 + 9.73(K_f\varepsilon_f)^3 - 6.43(K_f\varepsilon_f)^4$ for nickel alloys.

$Y_{sm} = 1.0$ for other materials.

If Y_{sm} is less than 1.0, then $Y_{sm} = 1.0$.

If Y_{sm} is greater than 2.0, then $Y_{sm} = 2.0$.

(c) $\varepsilon_f = \sqrt{\left[\ln\left(1 + \frac{2w}{D_b}\right)\right]^2 + \left[\ln\left(1 + \frac{nt_p}{2r_m}\right)\right]^2}$ for bellows formed from cylinders with an inside diameter of D_b if forming is performed 100% to the outside of the initial cylinder.

$\varepsilon_f = \sqrt{\left[\ln\left(1 + \frac{w}{D_b}\right)\right]^2 + \left[\ln\left(1 + \frac{nt_p}{2r_m}\right)\right]^2}$ for bellows formed from cylinders with an inside diameter of D_m if forming is performed 50% to the inside and 50% to the outside of the initial cylinder.

(d) C_p is evaluated using Table 4.19.3.

Table 4.19.3
Method to Determine Coefficient C_p

C_2	α_0	α_1	α_2	α_3	α_4	α_5
			$C_1 \le 0.3$			
0.2	1.001	−0.448	−1.244	1.932	−0.398	−0.291
0.4	0.999	−0.735	0.106	−0.585	1.787	−1.022
0.6	0.961	−1.146	3.023	−7.488	8.824	−3.634
0.8	0.955	−2.708	7.279	14.212	−104.242	133.333
1.0	0.95	−2.524	10.402	−93.848	423.636	−613.333
1.2	0.95	−2.296	1.63	16.03	−113.939	240
1.4	0.95	−2.477	7.823	−49.394	141.212	−106.667
1.6	0.95	−2.027	−5.264	48.303	−139.394	160
2.0	0.95	−2.073	−3.622	29.136	−49.394	13.333
2.5	0.95	−2.073	−3.622	29.136	−49.394	13.333
3.0	0.95	−2.073	−3.622	29.136	−49.394	13.333
3.5	0.95	−2.073	−3.622	29.136	−49.394	13.333
4.0	0.95	−2.073	−3.622	29.136	−49.394	13.333
			$C_1 > 0.3$			
C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.001	−0.448	−1.244	1.932	−0.398	−0.291
0.4	0.999	−0.735	0.106	−0.585	1.787	−1.022
0.6	0.961	−1.146	3.023	−7.488	8.824	−3.634
0.8	0.622	1.685	−9.347	18.447	−15.991	5.119
1.0	0.201	2.317	−5.956	7.594	−4.945	1.299
1.2	0.598	−0.99	3.741	−6.453	5.107	−1.527
1.4	0.473	−0.029	−0.015	−0.03	0.016	0.016
1.6	0.477	−0.146	−0.018	0.037	0.097	−0.067
2.0	0.935	−3.613	9.456	−13.228	9.355	−2.613
2.5	1.575	−8.646	24.368	−35.239	25.313	−7.157
3.0	1.464	−7.098	17.875	−23.778	15.953	−4.245
3.5	1.495	−6.904	16.024	−19.6	12.069	−2.944
4.0	2.037	−11.037	28.276	−37.655	25.213	−6.716

GENERAL NOTES:

(a) $C_1 = \frac{2r_m}{w}$ $\quad 0 \le C_1 \le 1$

Table 4.19.3
Method to Determine Coefficient C_p (Cont'd)

GENERAL NOTES (CONT'D):

(b) $C_2 = \frac{1.82 r_m}{\sqrt{D_m t_p}}$ $\quad 0.2 \le C_2 \le 4.0$

(c) $C_p = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$

(d) A plot of C_p versus C_1 and C_2 is shown in Figure 4.19.12.

Table 4.19.4
Method to Determine Coefficient C_f

C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.006	2.375	−3.977	8.297	−8.394	3.194
0.4	1.007	1.82	−1.818	2.981	−2.43	0.87
0.6	1.003	1.993	−5.055	12.896	−14.429	5.897
0.8	1.003	1.338	−1.717	1.908	0.02	−0.55
1.0	0.997	0.621	−0.907	2.429	−2.901	1.361
1.2	1	0.112	−1.41	3.483	−3.044	1.013
1.4	1	−0.285	−1.309	3.662	−3.467	1.191
1.6	1.001	−0.494	−1.879	4.959	−4.569	1.543
2.0	1.002	−1.061	−0.715	3.103	−3.016	0.99
2.5	1	−1.31	−0.829	4.116	−4.36	1.55
3.0	0.999	−1.521	−0.039	2.121	−2.215	0.77
3.5	0.998	−1.896	1.839	−2.047	1.852	−0.664
4.0	1	−2.007	1.62	−0.538	−0.261	0.249

GENERAL NOTES:

(a) $C_1 = \frac{2 r_m}{w}$ $\quad 0 \le C_1 \le 1$

(b) $C_2 = \frac{1.82 r_m}{\sqrt{D_m t_p}}$ $\quad 0.2 \le C_2 \le 4.0$

(c) $C_f = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$

(d) A plot of C_f versus C_1 and C_2 is shown in Figure 4.19.13.

Table 4.19.5
Method to Determine Coefficient C_d

C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1	1.151	1.685	−4.414	4.564	−1.645
0.4	0.999	1.31	0.909	−2.407	2.273	−0.706
0.6	1.003	2.189	−3.192	5.928	−5.576	2.07
0.8	1.005	1.263	5.184	−13.929	13.828	−4.83
1.0	1.001	0.953	3.924	−8.773	10.44	−4.749
1.2	1.002	0.602	2.11	−3.625	5.166	−2.312
1.4	0.998	0.309	1.135	−1.04	1.296	−0.087
1.6	0.999	0.12	0.351	−0.178	0.942	−0.115
2.0	1	−0.133	−0.46	1.596	−1.521	0.877
2.5	1	−0.323	−1.118	3.73	−4.453	2.055
3.0	1	−0.545	−0.42	1.457	−1.561	0.71
3.5	1	−0.704	−0.179	0.946	−1.038	0.474
4.0	1.001	−0.955	0.577	−0.462	0.181	0.08

Table 4.19.5
Method to Determine Coefficient C_d (Cont'd)

GENERAL NOTES:

(a) $C_1 = \frac{2r_m}{w} \qquad 0 \le C_1 \le 1$

(b) $C_2 = \frac{1.82r_m}{\sqrt{D_m t_p}} \qquad 0.2 \le C_2 \le 4.0$

(c) $C_d = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$

(d) A plot of C_d versus C_1 and C_2 is shown in Figure 4.19.14.

Table 4.19.6
Allowable Number of Cycles for U-Shaped Unreinforced Bellows

Stress Range Criteria	Allowable Number of Cycles
SI Units	
$K_g(E_o/E_b)S_t \ge 448$ MPa	$N_{alw} = \left(\frac{35\ 850}{K_g(E_o/E_b)S_t - 264}\right)^2$
257.2 MPa $< K_g(E_o/E_b)S_t < 448$ MPa	$N_{alw} = \left(\frac{46\ 200}{K_g(E_o/E_b)S_t - 211}\right)^2$
$K_g(E_o/E_b)S_t \le 257.2$ MPa	$N_{alw} = 10^6$ cycles
U.S. Customary Units	
$K_g(E_o/E_b)S_t \ge 65{,}000$ psi	$N_{alw} = \left(\frac{5.2(10)^6}{K_g(E_o/E_b)S_t - 38{,}300}\right)^2$
$37{,}300$ psi $< K_g(E_o/E_b)S_t < 65{,}000$ psi	$N_{alw} = \left(\frac{6.7(10)^6}{K_g(E_o/E_b)S_t - 30{,}600}\right)^2$
$K_g(E_o/E_b)S_t \le 37{,}300$ psi	$N_{alw} = 10^6$ cycles

GENERAL NOTES:

(a) In the above equations, K_g is the fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range for K_g is $1.0 \le K_g \le 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. If the expansion bellows does not have circumferential welds and satisfies all of the design and examination requirements of this paragraph, a $K_g = 1.0$ may be used.

(b) The allowable number of cycles given in this table includes a reasonable design margin (2.6 on cycles and 1.25 on stress) and represents the maximum number of cycles for the operating condition considered. Therefore an additional design margin should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

Table 4.19.7
Stress Calculations and Acceptability Criteria for U-Shaped Reinforced Bellows Subject to Internal Pressure

Bellows	Reinforcing Ring Member and Fastener
Stress Calculation	
Circumferential membrane stress in end tangent $$S_1 = \frac{1}{2}\frac{(D_b + nt)^2 L_d E_b}{(ntL_t + A/2)(D_b + nt)E_b + A_{tc}D_cE_c}P$$	Circumferential membrane stress in ring member $$S_2' = \frac{PD_mq}{2A_r}\left(\frac{1}{R_1 + 1}\right)$$
Circumferential membrane stress in collar $$S_1' = \frac{1}{2}\frac{(D_c)^2 L_d E_c}{(ntL_t + A/2)(D_b + nt)E_b + A_{tc}D_cE_c}P$$	where $R_1 = \dfrac{AE_b}{A_rE_r}$
Circumferential membrane stress in convolutions $$S_2 = \frac{PD_mq}{2A}\left(\frac{R}{R + 1}\right)$$	
Meridional membrane stress $$S_3 = 0.85\frac{w - 4C_r r_m}{2nt_p}P$$	Membrane stress in fastener
Meridional bending stress $$S_4 = \frac{0.85}{2n}\left(\frac{w - 4C_r r_m}{t_p}\right)^2 C_pP$$	$$S_2'' = \frac{PD_mq}{2A_f}\left(\frac{1}{R_2 + 1}\right)$$
where $C_r = 0.3 - \left(\dfrac{100}{1048P^{1.5} + 320}\right)^2$ P in MPa $= 0.3 - \left(\dfrac{100}{0.6P^{1.5} + 320}\right)^2$ P in psig	where $R_2 = \dfrac{AE_b}{D_m}\left(\dfrac{L_f}{A_fE_f} + \dfrac{D_m}{A_rE_r}\right)$
Acceptance Criteria	
$S_1 \le S$ $S_1' \le C_{wc}S_c$ $S_2 \le S$ $S_3 + S_4 \le K_mS$	$S_2' \le C_{wr}S_r$ ring member $S_2'' \le S_f$ fastener

GENERAL NOTES:
(a) k, A, D_c, D_m, t_p, K_m, and r_m are evaluated using the equations in Table 4.19.2.
(b) C_p is evaluated using Table 4.19.3.
(c) $R = R_1$ for integral reinforcing ring members and R_2 for reinforcing ring members joined by fasteners.
(d) In the case of reinforcing members which are made in sections, and joined by fasteners in tension, the equation for S_2 assumes that the structure used to retain the fastener does not bend so as to permit the reinforcing member to expand diametrically. In addition, the end reinforcing members must be restrained against the longitudinal annular pressure load of the bellows.
(e) In the case of equalizing rings, the equation for S'_2 provides only the simple membrane stress and does not include the bending stress caused by the eccentric fastener location. Elastic analysis and/or actual tests can determine these stresses.

Table 4.19.8
Allowable Number of Cycles for U-Shaped Reinforced Bellows

Stress Range Criteria	Allowable Number of Cycles
SI Units	
$K_g(E_o/E_b)S_t \geq 567$ MPa	$N_{alw} = \left(\dfrac{45\ 505}{K_g(E_o/E_b)S_t - 344}\right)^2$
326.1 MPa $< K_g(E_o/E_b)S_t < 567$ MPa	$N_{alw} = \left(\dfrac{58\ 605}{K_g(E_o/E_b)S_t - 267.5}\right)^2$
$K_g(E_o/E_b)S_t \leq 326.1$ MPa	$N_{alw} = 10^6$ cycles
U.S. Customary Units	
$K_g(E_o/E_b)S_t \geq 82{,}200$ psi	$N_{alw} = \left(\dfrac{6.6(10)^6}{K_g(E_o/E_b)S_t - 48{,}500}\right)^2$
$47{,}300$ psi $< K_g(E_o/E_b)S_t < 82{,}200$ psi	$N_{alw} = \left(\dfrac{8.5(10)^6}{K_g(E_o/E_b)S_t - 38{,}800}\right)^2$
$K_g(E_o/E_b)S_t \leq 47{,}300$ psi	$N_{alw} = 10^6$ cycles

GENERAL NOTES:

(a) In the above equations, K_g is the fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range for K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg. welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. If the expansion bellows does not have circumferential welds and satisfies all of the design and examination requirements of this paragraph, $K_g = 1.0$ may be used.

(b) The allowable number of cycles given in this table includes a reasonable design margin (2.6 on cycles and 1.25 on stress) and represents the maximum number of cycles for the operating condition considered. Therefore an additional design margin should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

Table 4.19.9
Stress Calculations and Acceptability Criteria for Toroidal Bellows Subject to Internal Pressure

Stress Calculation

Circumferential membrane stress in end tangent

$$S_1 = \frac{1}{2}\frac{\left(D_b + nt\right)^2 L_d E_b}{D_c E_c A_{tc}} P \text{ for externally attached bellows}$$

Circumferential membrane stress in tangent collar

$$S_1' = \frac{1}{2}\frac{D_c L_d}{A_{tc}} P \text{ for externally attached bellows}$$

Circumferential membrane stress in shell

$$S_1''' = \frac{(D_s + t_s)(L_s + 0.5L_g + nt)}{2A_{ts}} P \text{ for internally attached bellows}$$

Circumferential membrane stress in bellows convolutions

$$S_2 = \frac{Pr}{2nt_p}$$

Circumferential membrane stress in reinforcing collar

$$S_2' = \frac{D_r(L_{rt} + L_g + 2nt)}{2A_{rt}} P \text{ if } L_{rt} \le \frac{2}{3}\sqrt{D_r t_r}$$

$$S_2' = \frac{D_r(L_r + 0.5L_g + nt)}{2A_r} P \text{ if } L_{rt} > \frac{2}{3}\sqrt{D_r t_r}$$

Meridional membrane stress in bellows convolutions

$$S_3 = \frac{Pr}{nt_p}\left(\frac{D_m - r}{D_m - 2r}\right)$$

where

$D_c = D_b + 2nt + t_c$

$t_p = t\sqrt{\frac{D_b}{D_m}}$

Acceptance Criteria

$S_1 \le S$ for externally attached bellows
$S_1' \le C_{wc} S_c$ for externally attached bellows
$S_1''' \le C_{ws} S_s$ for internally attached bellows
$S_2 \le S$
$S_2' \le C_{wr} S_r$
$S_3 \le S$

Table 4.19.10
Stress and Axial Stiffness Coefficients for Toroidal Bellows

C_3	B_1	B_2	B_3
0	1.0	1.0	1.0
1	1.1	1.0	1.1
2	1.4	1.0	1.3
3	2.0	1.0	1.5
4	2.8	1.0	1.9
5	3.6	1.0	2.3
6	4.6	1.1	2.8
7	5.7	1.2	3.3
8	6.8	1.4	3.8
9	8.0	1.5	4.4
10	9.2	1.6	4.9
11	10.6	1.7	5.4
12	12.0	1.8	5.9
13	13.2	2.0	6.4
14	14.7	2.1	6.9
15	16.0	2.2	7.4
16	17.4	2.3	7.9
17	18.9	2.4	8.5
18	20.3	2.6	9.0
19	21.9	2.7	9.5
20	23.3	2.8	10.0

GENERAL NOTE: Equations for B_1, B_2, and B_3 are shown below.

$$B_1 = \frac{1.00404 + 0.028725C_3 + 0.18961C_3^2 - 0.00058626C_3^3}{1 + 0.14069C_3 - 0.0052319C_3^2 + 0.00029867C_3^3 - 6.2088(10)^{-6}C_3^4}$$

$$B_2 = 1.0 \qquad \text{for} \qquad C_3 \le 5$$

$$B_2 = \frac{0.049198 - 0.77774C_3 - 0.13013C_3^2 + 0.080371C_3^3}{1 - 2.81257C_3 + 0.63815C_3^2 + 0.0006405C_3^3} \qquad \text{for} \qquad C_3 > 5$$

$$B_3 = \frac{0.99916 - 0.091665C_3 + 0.040635C_3^2 - 0.0038483C_3^3 + 0.00013392C_3^4}{1 - 0.1527C_3 + 0.013446C_3^2 - 0.00062724C_3^3 + 1.4374(10)^{-5}C_3^4}$$

where

$$C_3 = \frac{6.61r^2}{D_m t_p}$$

Table 4.19.11
Allowable Number of Cycles for Toroidal Bellows

Stress Range Criteria	Allowable Number of Cycles
SI Units	
$K_g(E_o/E_b)S_t \geq 448$ MPa	$N_{alw} = \left(\dfrac{35\ 850}{K_g(E_o/E_b)S_t - 264}\right)^2$
$257.2\text{ MPa} < K_g(E_o/E_b)S_t < 448$ MPa	$N_{alw} = \left(\dfrac{46\ 200}{K_g(E_o/E_b)S_t - 211}\right)^2$
$K_g(E_o/E_b)S_t \leq 257.2$ MPa	$N_{alw} = 10^6$ cycles
U.S. Customary Units	
$K_g(E_o/E_b)S_t \geq 65{,}000$ psi	$N_{alw} = \left(\dfrac{5.2(10)^6}{K_g(E_o/E_b)S_t - 38{,}300}\right)^2$
$37{,}300\text{ psi} < K_g(E_o/E_b)S_t < 65{,}000$ psi	$N_{alw} = \left(\dfrac{6.7(10)^6}{K_g(E_o/E_b)S_t - 30{,}600}\right)^2$
$K_g(E_o/E_b)S_t \leq 37{,}300$ psi	$N_{alw} = 10^6$ cycles

GENERAL NOTES:
(a) In the above equations, K_g is the fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range for K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. If the expansion bellows does not have circumferential welds and satisfies all of the design and examination requirements of this paragraph, $K_g = 1.0$ may be used.
(b) The allowable number of cycles given in this table includes a reasonable design margin (2.6 on cycles and 1.25 on stress) and represents the maximum number of cycles for the operating condition considered. Therefore an additional design margin should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

4.19.13 FIGURES

Figure 4.19.1
Typical Bellows Expansion Joints

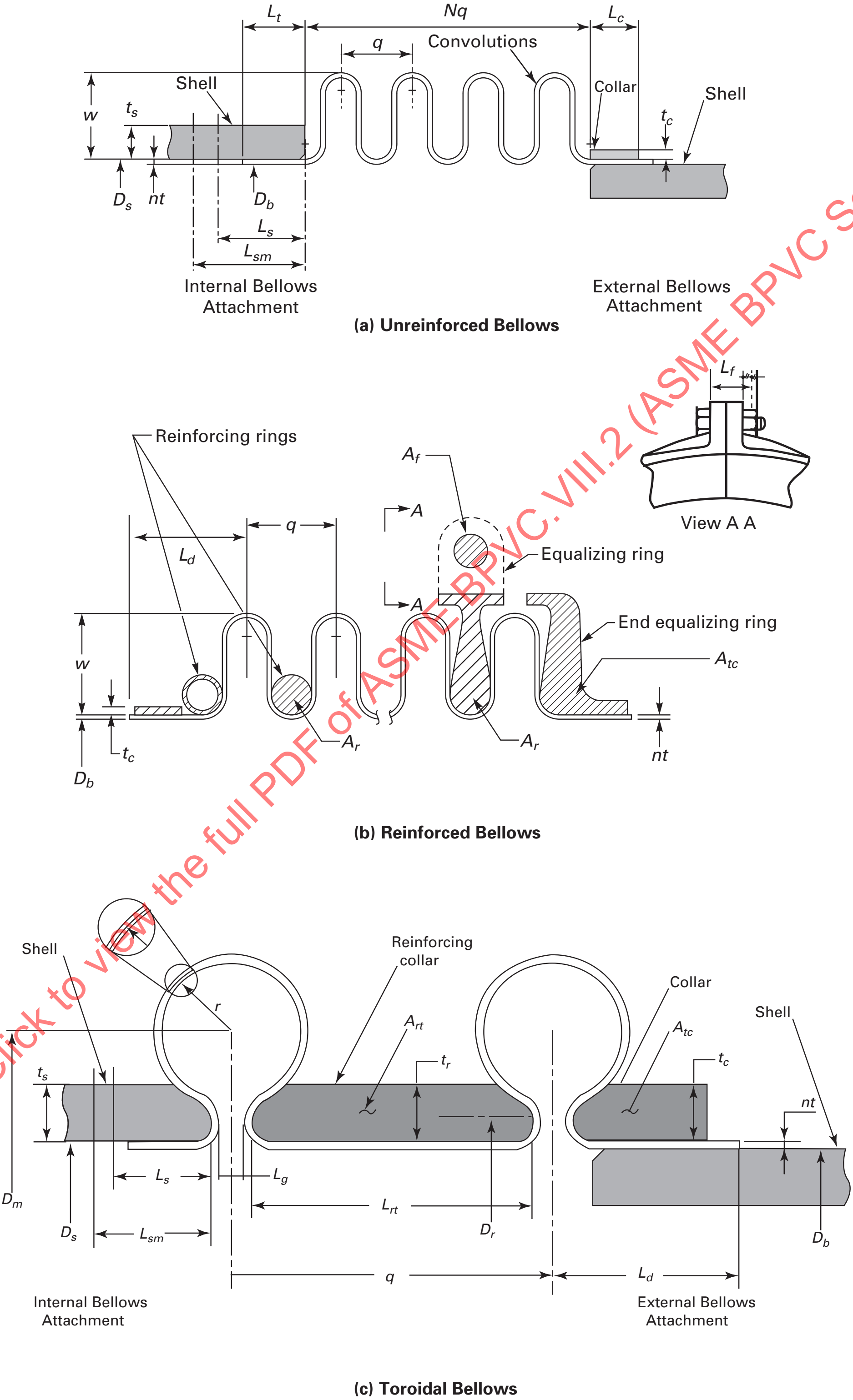
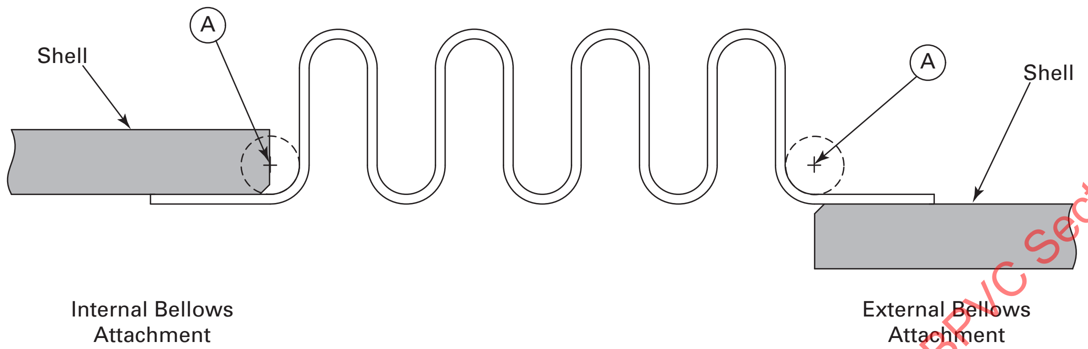
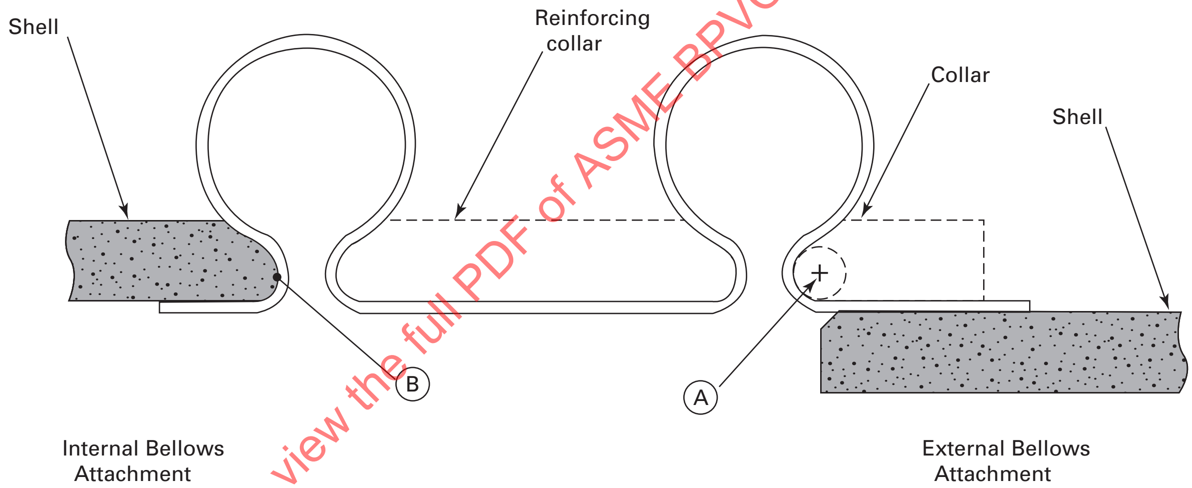


Figure 4.19.2
Starting Points for the Measurement of the Length of Shell on Each Side of Bellows



(a) U-Shaped Bellows



(b) Toroidal Bellows

Figure 4.19.3
Possible Convolution Profile in Neutral Position

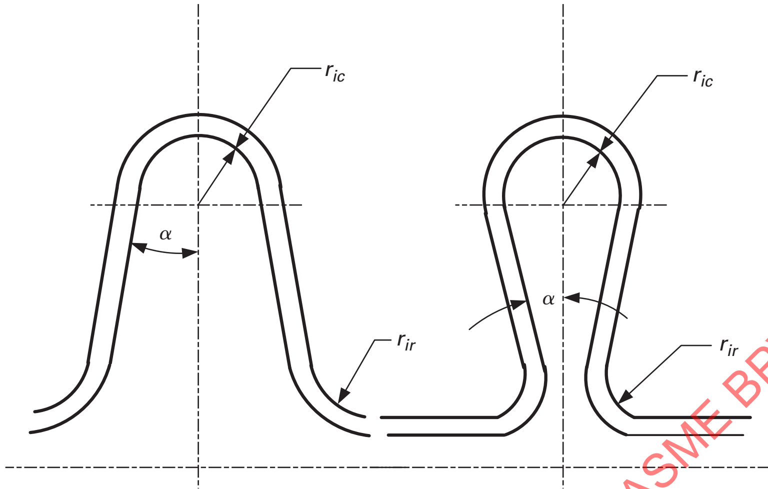


Figure 4.19.4
Dimensions to Determine I_{xx}

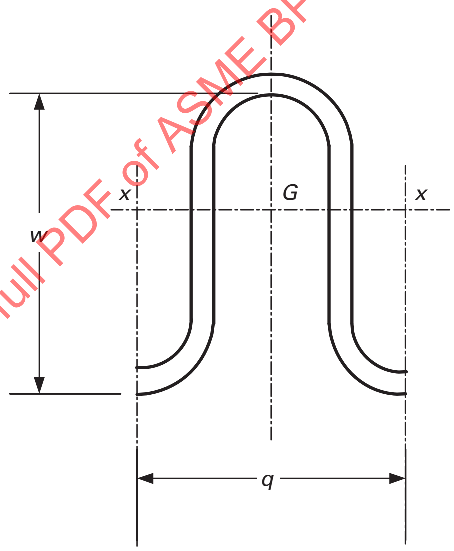


Figure 4.19.5
Bellows Subjected to an Axial Displacement *x*

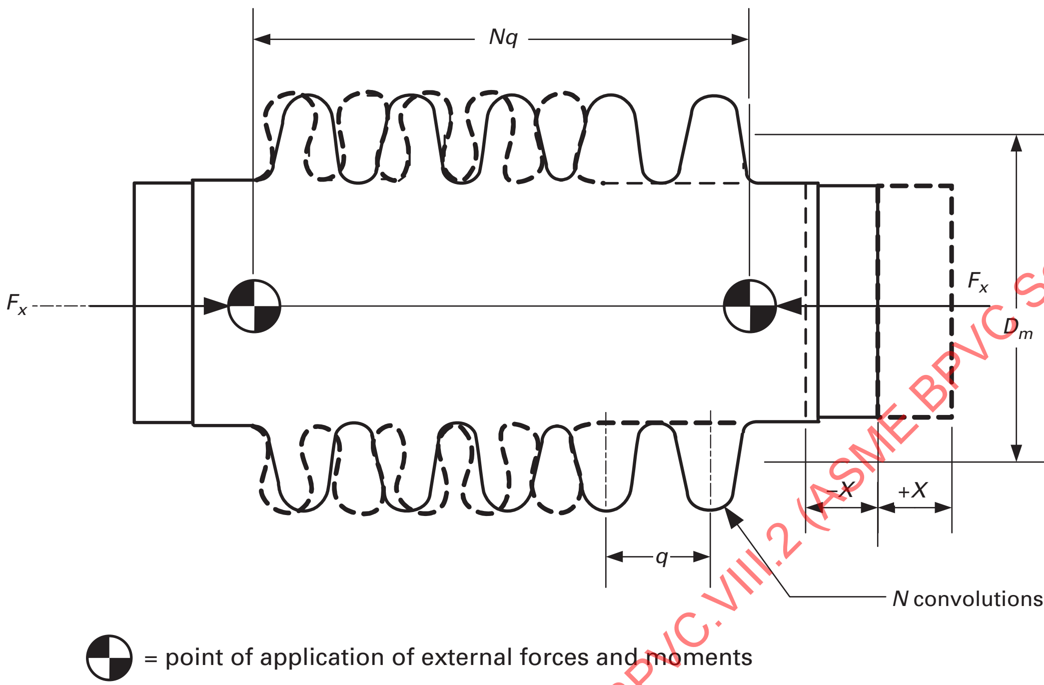


Figure 4.19.6
Bellows Subjected to a Lateral Deflection *y*

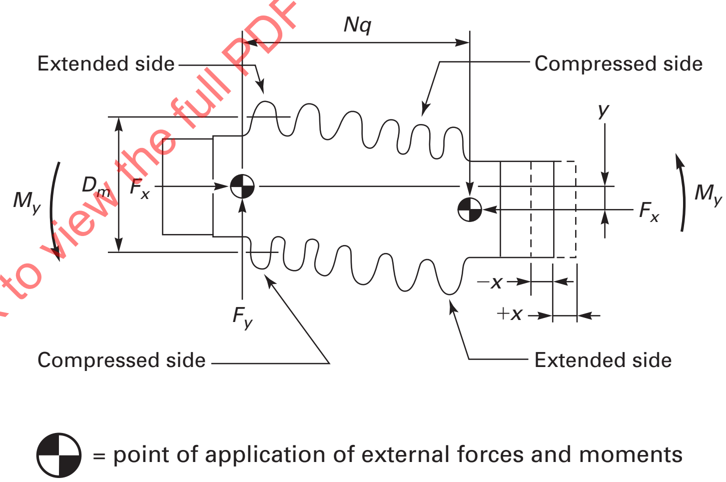


Figure 4.19.7
Bellows Subjected to an Angular Rotation θ

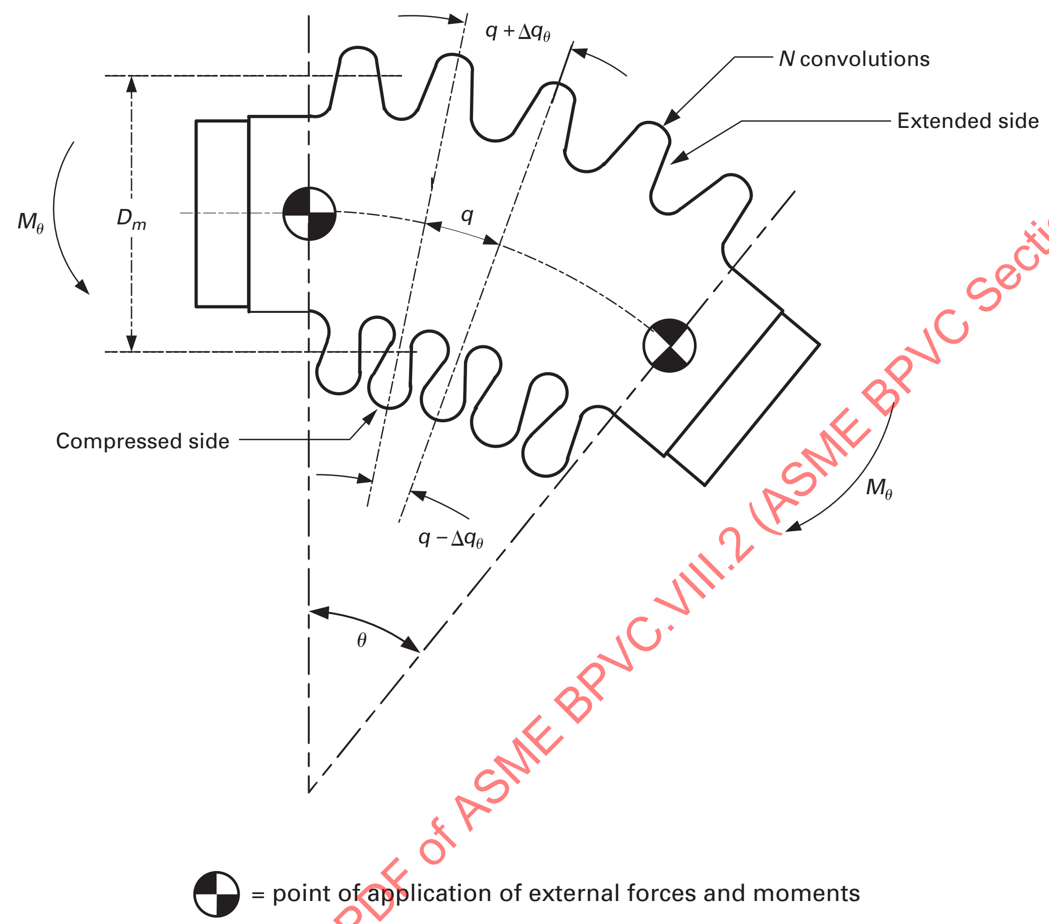
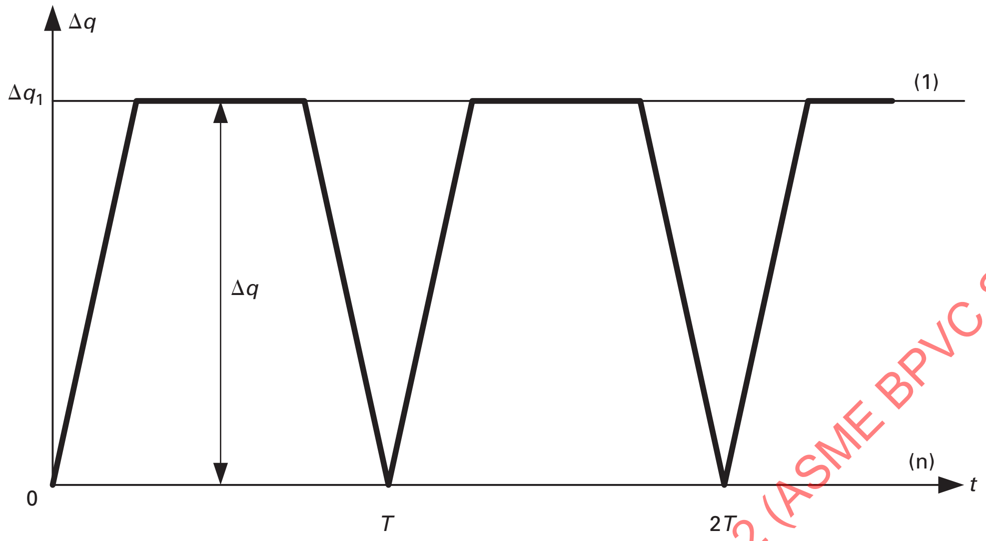
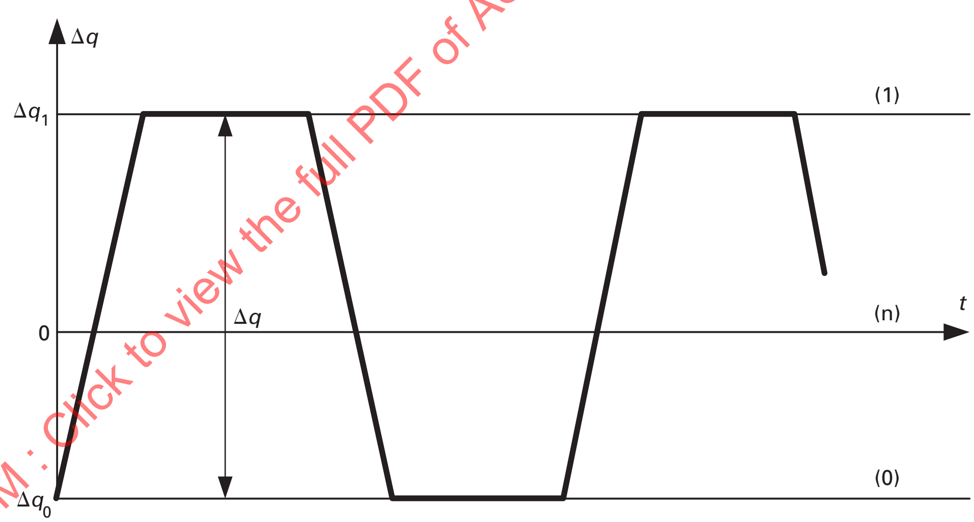


Figure 4.19.8
Cyclic Displacements



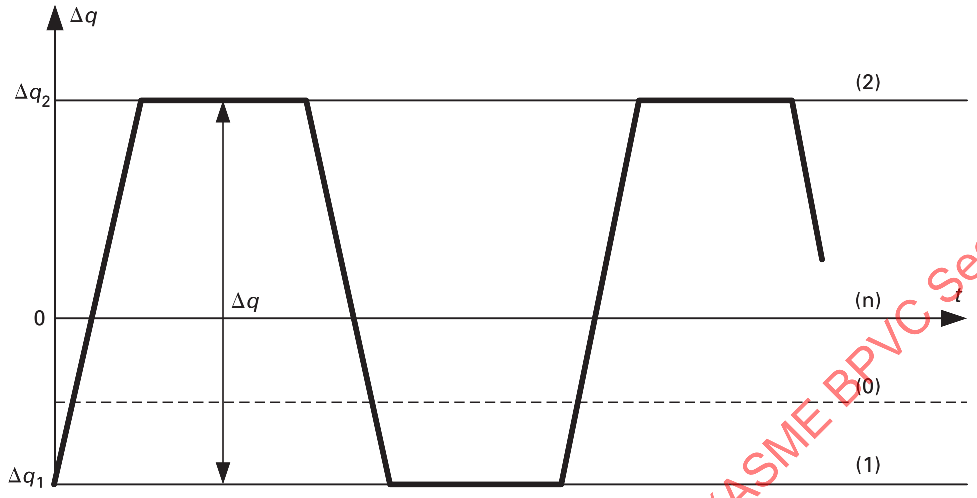
Legend:
(1) = operating position, Δq_1
(n) = neutral position

Figure 4.19.9
Cyclic Displacements



Legend:
(0) = initial position, Δq_0
(1) = operating position, Δq_1
(n) = neutral position

Figure 4.19.10
Cyclic Displacements



Legend:
(0) = initial position 0
(1) = operating position 1
(2) = operating position 2
(n) = neutral position

Figure 4.19.11
Some Typical Expansion Bellows Attachment Welds

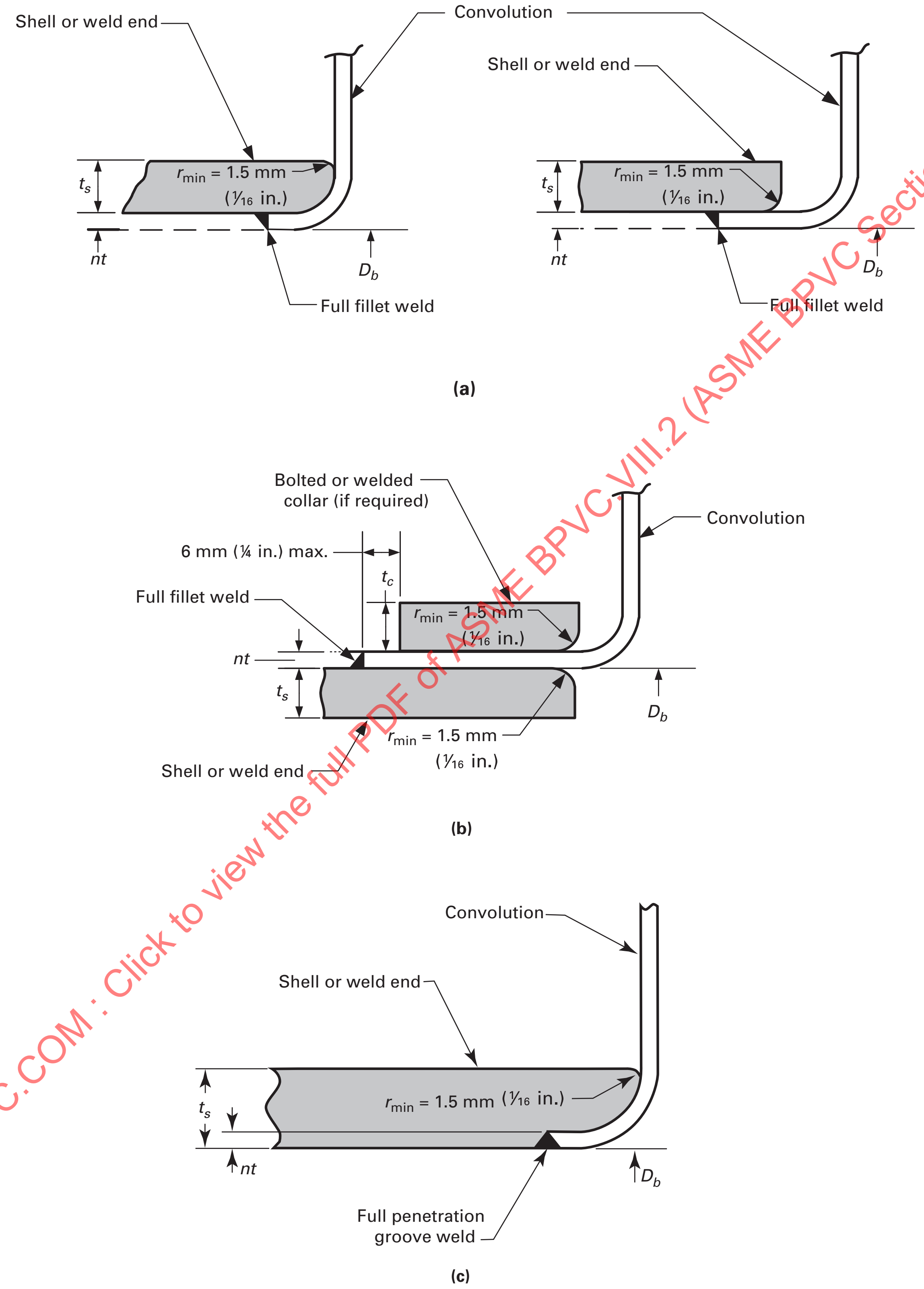


Figure 4.19.12
C_p Versus C_1 and C_2

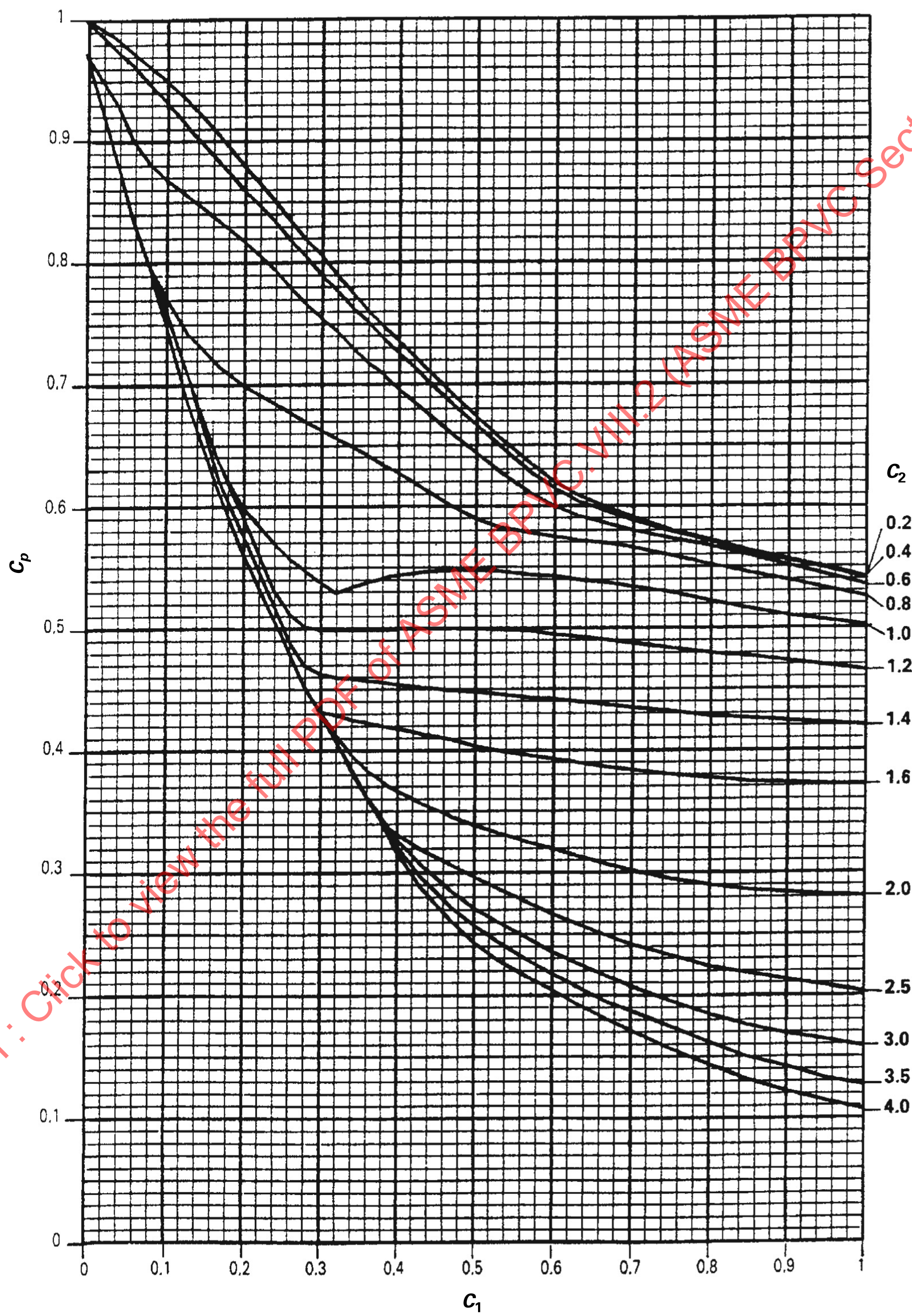


Figure 4.19.13
C_f Versus C_1 and C_2

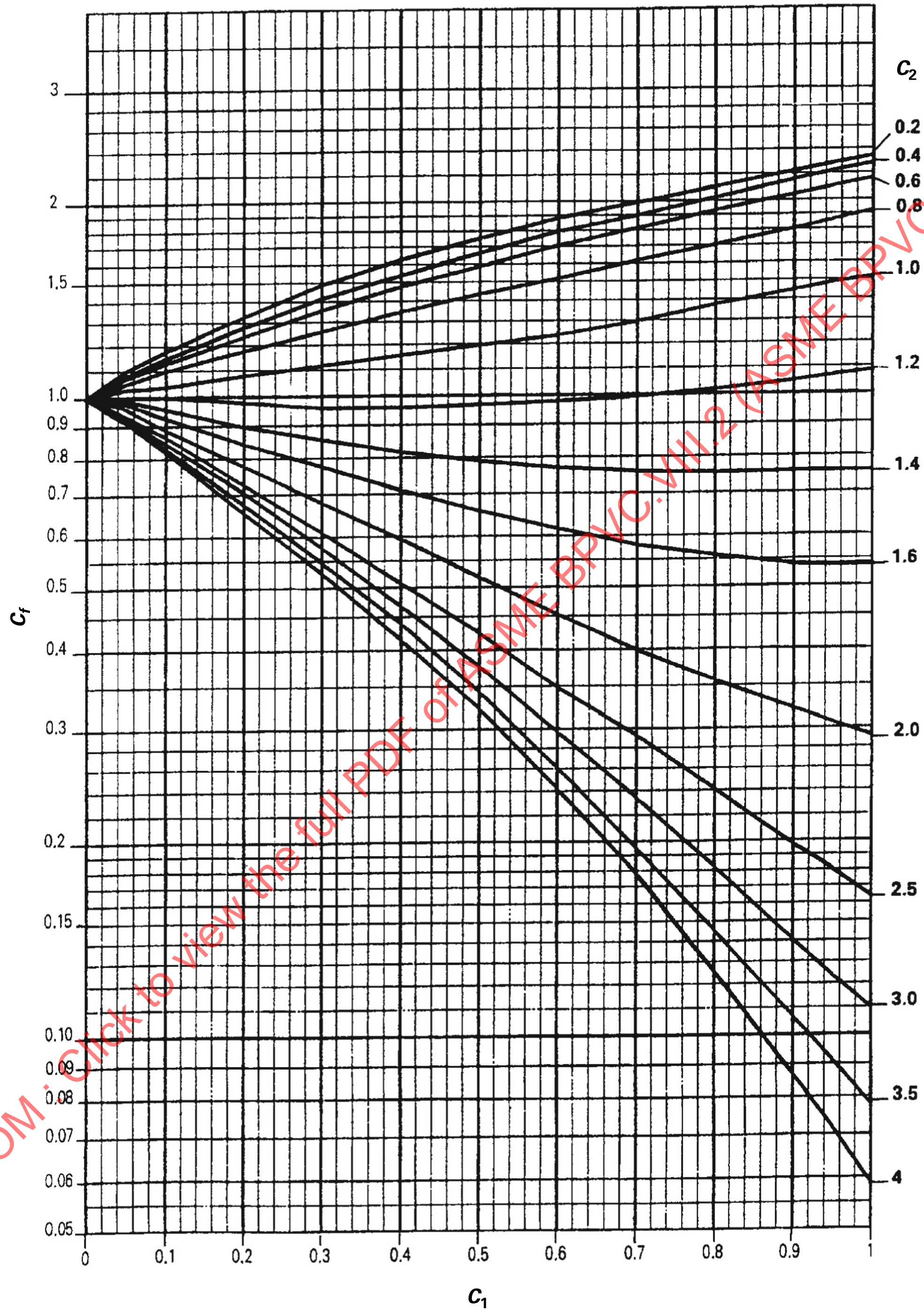
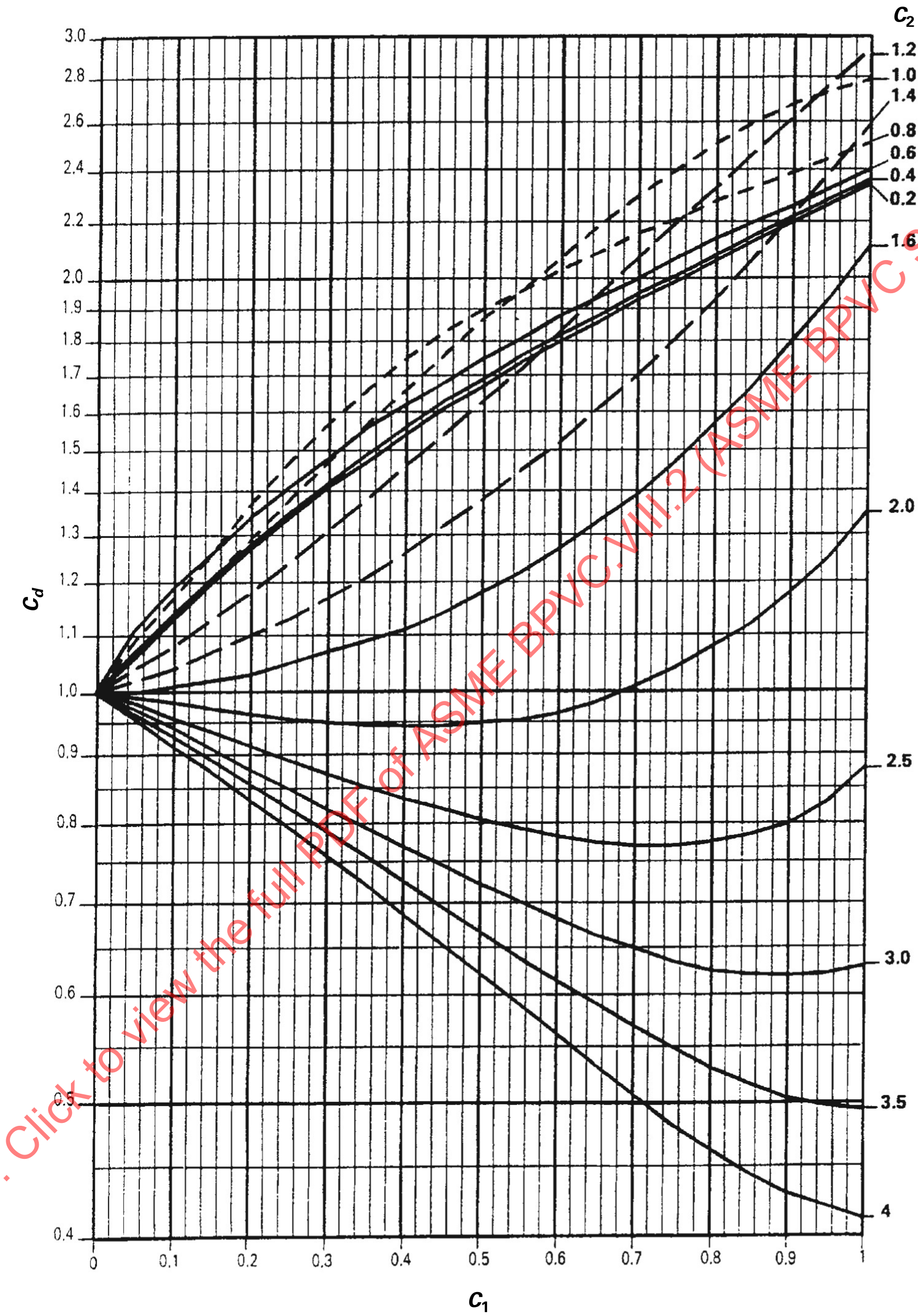


Figure 4.19.14
C_d Versus C_1 and C_2



4.19.14 SPECIFICATION SHEETS

(21)

4.19.1 Metric Form Specification Sheet For ASME Section VIII, Division 2 Bellows Expansion Joints, Metric Units

Date:______/________/_________

1. Item Number:__________________
2. Drawing/Tag/Serial/Job Number:________________
3. Quantity:__________________
4. Size: ________ OD _________ID mm

Applicable ASME Code Edition:__________________

Vessel Class:__________________________

Vessel Manufacturer:__________________________

Vessel Owner:______________________________

Installation Location:__________________________

Expansion Joint Overall Length:_______________ mm

5. Internal Pressure: Design ________ MPa
6. External Pressure: Design ________ MPa
7. Vessel Manufacturer Hydrotest Pressure Internal ________ MPa External ________ MPa

8. Temperature | Design ________ °C Operating ________ °C | Upset ________ °C
9. Vessel Rating | MAWP ________ MPa MDMT ________ °C | Installed Position: Horz. Vert.

10. Design Movements [Note (1)]:

Axial Compression: (−)_______mm Axial Extension: (+)_______mm Lateral:_______mm Angular: ________ deg

11. Specified Number of Cycles: ____________

12. Design Torsion: Moment ____________ N-mm or Twist Angle: _______________ deg

13. Shell Material: _____________ Bellows Material: _____________

14. Shell Thickness: _____________ mm Shell Corrosion Allowance: Internal:_______ mm External:_________ mm

15. Shell Radiography: Spot / Full

16. End Preparation: Square Cut Outside Bevel Inside Bevel Double Bevel (Describe in Line 24 if special)

17. Heat Exchanger Tube Length Between Inner Tubesheet Faces: _______ mm

18. Maximum Bellows Spring Rate: No Yes – ______________________ N/mm

19. Internal Liner: No Yes – Material____________________________

20. Drain Holes in Liner: No Yes – Quantity/Size:______________________

21. Liner Flush with Shell ID: No Yes – Telescoping Liners? No Yes

22. External Cover: No Yes – Material:______________________

23. Pre-Production Approvals Required: No Yes – Drawings / Bellows Calculations / Weld Procedures

24. Additional Requirements (i.e., bellows pre-set, ultrasonic examination, etc.):

NOTE:

(1) For multiple movements, design movements (line 10) can be replaced by operating movements, which should then be described under "Additional Requirements" (line 24). For each one of them, axial compression or axial extension, lateral deflection and angular rotation at each extremity of cycle, together with the specified number of cycles, should be indicated. When known, the order of occurrence of the movements should also be indicated.

(07/21)

(21)

4.19.2 U.S. Customary Form Specification Sheet For ASME Section VIII, Division 2 Bellows Expansion Joints, U.S. Customary Units

Date:______/________/_________

Applicable ASME Code Edition:_________________

Vessel Class:_________________________

1. Item Number:__________________

Vessel Manufacturer:_________________________

2. Drawing/Tag/Serial/Job Number:________________

Vessel Owner:_____________________________

3. Quantity:__________________

Installation Location:________________________

4. Size: ________ OD _________ID in.

Expansion Joint Overall Length:_______________ in.

5. Internal Pressure: Design ________ psig

6. External Pressure: Design ________ psig

7. Vessel Manufacturer Hydrotest Pressure Internal ________ psig External ________ psig

8. Temperature Design ________ °F Operating ________ °F Upset ________ °F

9. Vessel Rating MAWP ________ psig MDMT ________ °F Installed Position: Horz. Vert.

10. Design Movements [Note (1)]:

Axial Compression: (−) _______ in. Axial Extension: (+) ________ in. Lateral: _______ in. Angular: ________ deg

11. Specified Number of Cycles: _____________

12. Design Torsion: Moment _____________ in.-lb or Twist Angle: _______________ deg

13. Shell Material: _____________

Bellows Material: _____________

14. Shell Thickness: _____________ in. Shell Corrosion Allowance: Internal: _______ in. External:_________ in.

15. Shell Radiography: Spot / Full

16. End Preparation: Square Cut Outside Bevel Inside Bevel Double Bevel (Describe in Line 24 if special)

17. Heat Exchanger Tube Length Between Inner Tubesheet Faces: _______ in.

18. Maximum Bellows Spring Rate: No Yes – _______________________ lb/in.

19. Internal Liner: No Yes – Material___________________________

20. Drain Holes in Liner: No Yes – Quantity/Size:______________________

21. Liner Flush with Shell ID: No Yes – Telescoping Liners? No Yes

22. External Cover: No Yes – Material:__________________________

23. Pre-Production Approvals Required: No Yes – Drawings/Bellows Calculations/Weld Procedures

24. Additional Requirements (i.e., bellows pre-set, ultrasonic examination, etc.):

NOTE:

(1) For multiple movements, design movements (line 10) can be replaced by operating movements, which should then be described under "Additional Requirements" (line 24). For each one of them, axial compression or axial extension, lateral deflection and angular rotation at each extremity of cycle, together with the specified number of cycles, should be indicated. When known, the order of occurrence of the movements should also be indicated.

(07/21)

4.20 DESIGN RULES FOR FLEXIBLE SHELL ELEMENT EXPANSION JOINTS

4.20.1 SCOPE

(a) The rules in 4.20 cover the minimum requirements for the design of flexible shell element expansion joints used as an integral part of heat exchangers or other pressure vessels. These rules apply to single-layer flexible shell element expansion joints shown in Figure 4.20.1 and are limited to applications involving axial displacement only. The suitability of an expansion joint for the specified design pressure, temperature, and axial displacement shall be determined by the methods described herein.

(b) The rules in 4.20 cover the common types of flexible shell element expansion joints but are not intended to limit the configurations or details to those illustrated or otherwise described herein. Designs that differ from those covered in 4.20 (e.g., multilayer, asymmetric geometries or loadings having a thick liner or other attachments) shall be in accordance with 4.1.1.2.

4.20.2 CONDITIONS OF APPLICABILITY

(a) For carbon and low-alloy steels, the minimum thickness, exclusive of corrosion allowance, shall be 3 mm (0.125 in.) for all pressure-retaining components.

(b) For high-alloy and nonferrous steels, the minimum thickness shall conform to the requirements of 4.1.2.

(c) The knuckle radius, r_a or r_b, of any formed element shall not be less than three times the element thickness, t, as shown in Figure 4.20.1.

(d) Extended straight flanges between the inner torus and the shell and between both outer tori are permissible. An outer shell element between the outer tori is permissible. Extended straight flanges between the inner torus and the shell, between the outer tori and an outer shell element, and between both outer tori that do not have an intermediate outer shell element with lengths in excess of $0.5\sqrt{Rt_f}$ shall satisfy all the requirements of 4.3.3.

(e) Nozzles or other attachments located in the outer straight flange or outer shell element shall satisfy the axial spacing requirements of Figure 4.20.2.

4.20.3 DESIGN CONSIDERATIONS

(a) Expansion joints shall be designed to provide flexibility for thermal expansion and also to function as pressure-containing elements.

(b) The vessel Manufacturer shall specify the design conditions and requirements for the detailed design and manufacture of the expansion joint.

(c) Thinning of any flexible element as a result of forming operations shall be considered in the design and specification of material thickness.

(d) In all vessels with integral expansion joints, the hydrostatic end force caused by the pressure or the joint spring force, or both, shall be resisted by adequate restraining elements (e.g., exchanger tubes, tubesheets or shell, external restraints, anchors, etc.). The primary stress in these restraining elements shall satisfy 4.1.6.1.

(e) If expansion joint flexible elements are to be extended, compressed, rotated, or laterally offset to accommodate connecting parts that are not properly aligned, such movements shall be considered in the design.

(f) Elastic moduli, yield strengths, and allowable stresses shall be taken at the design temperatures. However, for cases involving thermal loading, it is permitted to use the operating metal temperature instead of the design temperature.

4.20.4 MATERIALS

Materials for pressure-retaining components shall comply with the requirements of Part 3.

4.20.5 DESIGN

The expansion joint design shall conform to the requirements of this Part and the additional requirements listed below.

(a) Except as permitted by 4.18.12.1(b), the design of the expansion joint flexible elements shall satisfy the following stress limits. These stress limits shall be met in both the corroded and uncorroded conditions.

(b) Mechanical Loads Only. Mechanical loads include pressure and pressure-induced axial deflection. The maximum stress in the expansion joint is limited to $1.5S$.

(c) Thermally Induced Displacements Only. The maximum stress in the expansion joint is limited to S_{PS}.

(d) Mechanical Loads Plus Thermally Induced Displacements. The maximum stress in the expansion joint is limited to S_{PS}.

(e) The calculation of the individual stress components and their combination shall be performed by a method of stress analysis that can be shown to be appropriate for expansion joints.

(f) The spring rate of the expansion joint assembly shall be determined either by calculation or by testing.

4.20.6 MARKING AND REPORTS

(a) The expansion joint Manufacturer, whether the vessel Manufacturer or a parts Manufacturer, shall have a valid ASME Code U2 Certificate of Authorization and shall complete the appropriate Data Report in accordance with Part 2.

(b) The Manufacturer responsible for the expansion joint design shall include the following additional data and statements on the appropriate Data Report:

(1) axial movement (±) and associated loading condition, if applicable

(2) new and corroded spring rates

(3) the expansion joint has been constructed to the rules of this paragraph

(c) A parts Manufacturer shall identify the vessel for which the expansion joint is intended on the Partial Data Report.

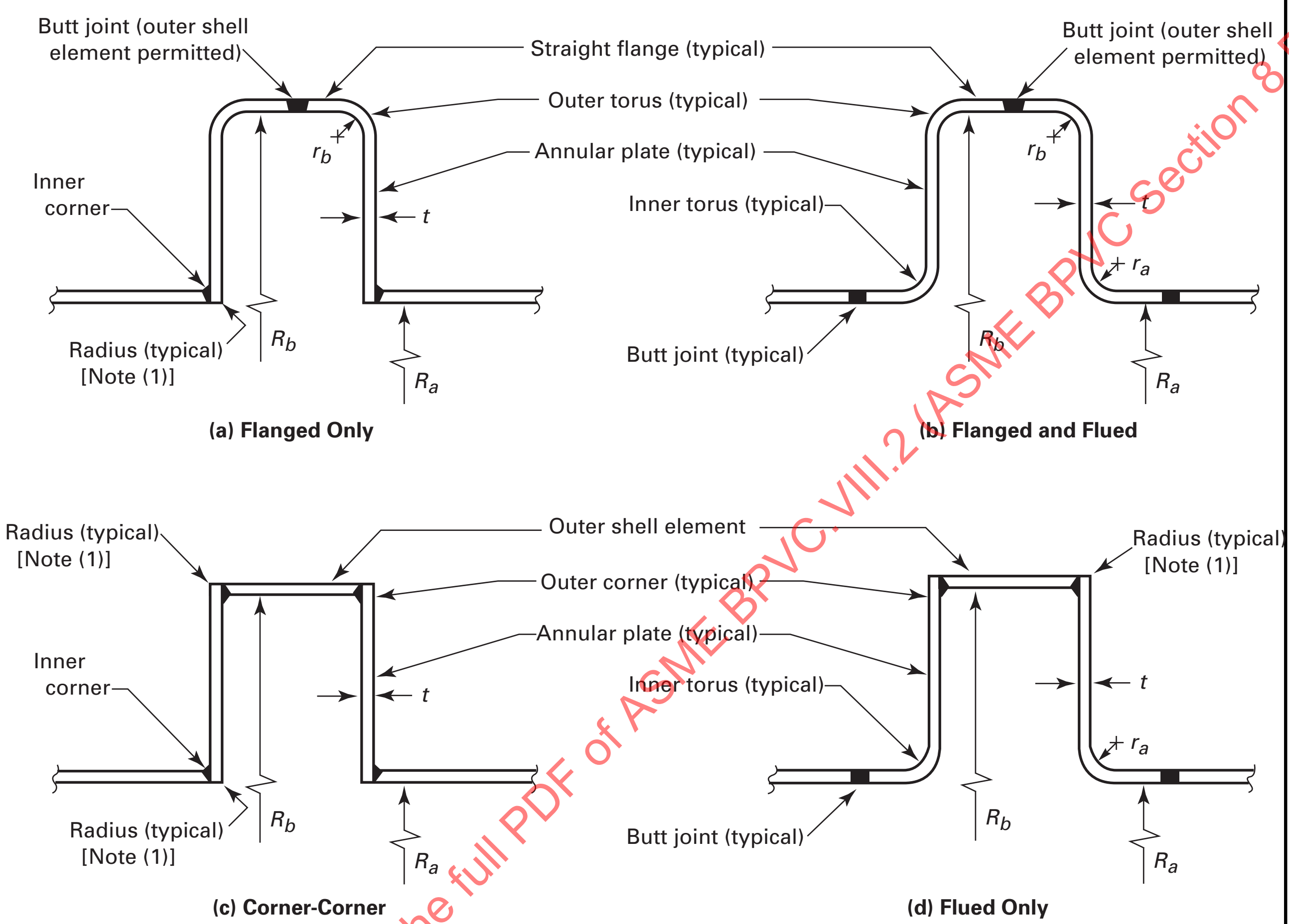
(d) Markings shall not be stamped on the flexible elements of the expansion joint.

4.20.7 NOMENCLATURE

R = uncorroded inside radius of expansion joint straight flange at the point of consideration
= R_a at the inner torus
= R_b at the outer torus
r_a = outside radius of inner torus
r_b = inside radius of outer torus
S = allowable stress from Annex 3-A for expansion joint material at temperature, T
S_{PS} = allowable primary plus secondary stress evaluated using 4.1.6.3 for expansion joint material at temperature, T
S_y = allowable yield strength from Annex 3-D for expansion joint material at temperature, T
T = expansion joint design temperature
t = thickness of expansion joint flexible element
t_f = uncorroded thickness of expansion joint straight flange
t_o = uncorroded thickness of expansion joint outer shell element

4.20.8 FIGURES

Figure 4.20.1
Typical Flexible Shell Element Expansion Joints



Legend:

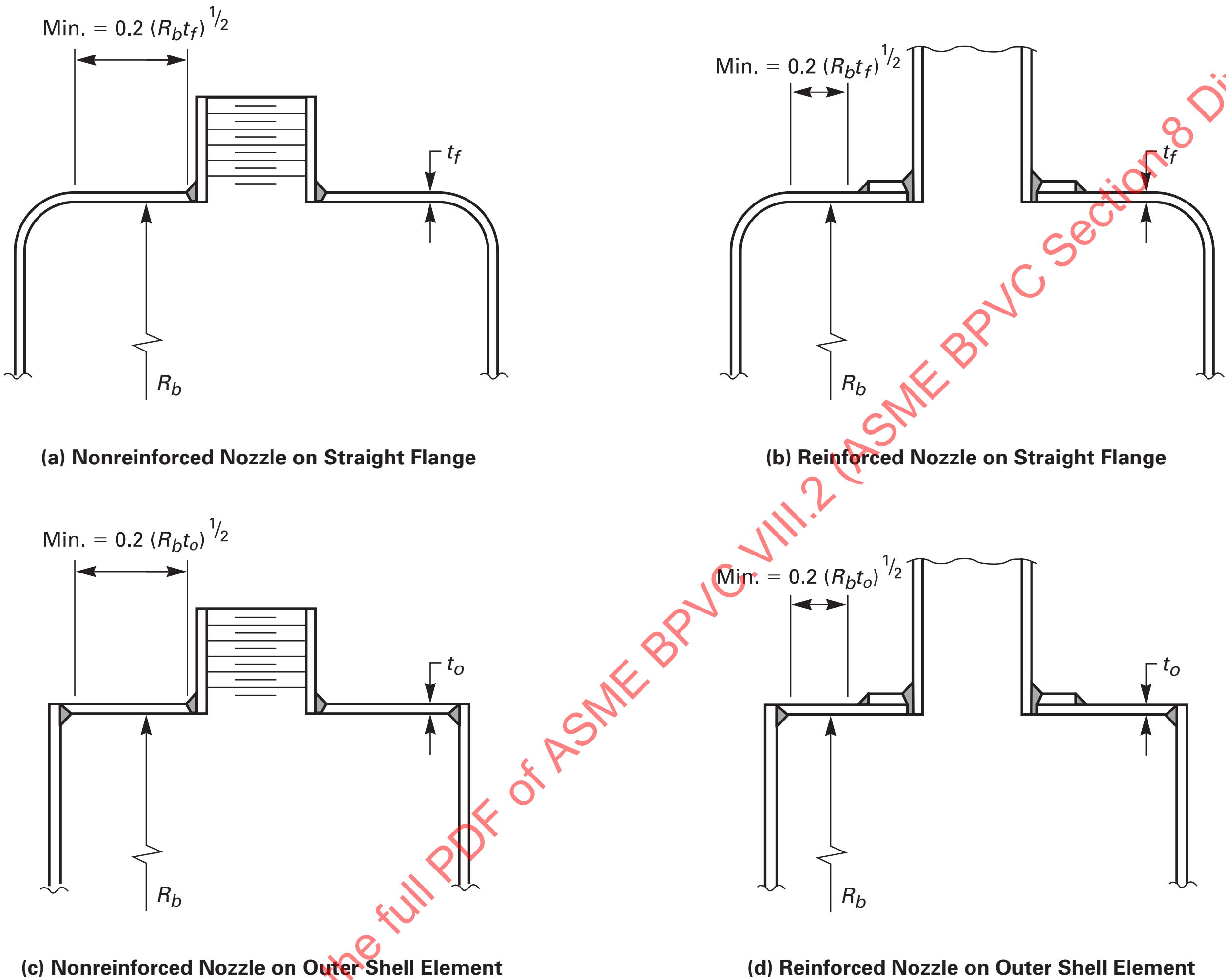
R_a, R_b = inside radius of expansion joint straight flange, shell, or outer shell element

t = thickness of expansion joint flexible element

GENERAL NOTE: r_a, r_b ≥ 3t

NOTE:
(1) Where the term "Radius" appears, provide a 3-mm ($^1/_8$-in.) minimum blend radius.

Figure 4.20.2
Typical Nozzle Attachment Details Showing Minimum Length of Straight Flange or Outer Shell Element



Legend:
R_b = inside radius of expansion joint straight flange
t_f = uncorroded thickness of expansion joint straight flange
t_o = uncorroded thickness of expansion joint outer shell element

(21) 4.21 TUBE-TO-TUBESHEET JOINT STRENGTH

4.21.1 SCOPE

4.21.1.1 General.

(a) Tubes used in the construction of heat exchangers or similar apparatus may be considered to act as stays that support or contribute to the strength of tubesheets in which they are engaged. Tube-to-tubesheet joints shall be capable of transferring the applied tube loads. The design of tube-to-tubesheet joints depends on the type of joint, degree of examination, and shear load tests, if performed.

(b) Tube-to-tubesheet joints, except for as exempted in (c) and (d) below, shall have their strength determined by one of the following sections. Either 4.21.2 or 4.21.3 may be used at the discretion of the designer and as applicable. For designs that are within the scope of both sections, compliance with both is not mandatory.

(c) Back-face welded joints, such as shown in Figure 4.18.2(d), are not covered.

(d) Determination of tube-to-tubesheet joint strength is not mandatory for U-Tube tubesheets (see 4.18.7).

(e) Some combinations of tube and tubesheet materials, when welded, result in welded joints having lower ductility than required in the material specifications. Appropriate tube-to-tubesheet joint geometry, welding method, and/or heat treatment shall be used with these materials to minimize this effect.

(f) In the selection of joint type, consideration shall be given to the mean metal temperature of the joint at operating temperatures and differential thermal expansion of the tube and tubesheet which may affect the joint integrity. The following provisions apply for establishing maximum operating temperature for tube-to-tubesheet joints.

(1) Tube-to-tubesheet joints where the maximum allowable axial load is controlled by the weld shall be limited to the maximum temperature for which there are allowable stresses for the tube or tubesheet material in Annex 3-A. Tube-to-tubesheet joints in this category are any of the following:

(-a) those with or without expansion complying with the following:

(-1) Tube-to-tubesheet joints having full-strength welds as defined in accordance with 4.21.1.2(a) shall be designed in accordance with 4.21.2.2 and do not require shear load testing

(-2) Tube-to-tubesheet joints having partial-strength welds as defined in accordance to 4.21.1.2(b) and shall be designed in accordance with 4.21.2.3 and do not require shear load testing.

(-b) those welded and expanded joints, such as Joint Types f, g, and h from Table 4.21.1, where the maximum allowable axial load is determined in accordance with 4.21.3.2 and is controlled by the weld

(-c) those welded-only joints, such as Joint Types a, b, b-1, and e from Table 4.21.1, where the maximum allowable load is determined in accordance with 4.21.3.2

(2) Tube-to-tubesheet joints where the maximum allowable axial load is determined in accordance with 4.21.3.2 considering friction only, such as Joint Types i, j, and k from Table 4.21.1, or is controlled by friction in welded and expanded joints, such a Joint Types f, g, and h from Table 4.21.1, shall be limited to temperatures as determine by the following:

(-a) The operating temperature of the tube-to-tubesheet joint shall be within the tube and tubesheet time-independent properties as provided in Annex 3-A.

(-b) The maximum operating temperature is based on the interface pressure that exists between the tube and tubesheet. The maximum operating temperature is limited such that the interface pressure due to expanding the tube at joint fabrication plus the interface pressure due to differential thermal expansion, (P_o + P_T), does not exceed 58% of the smaller of the tube or tubesheet yield strength in Annex 3-D at the operating temperature for $P_T \geq 0$. For $P_T < 0$, where due to differential thermal expansion the tube expands less than the tubesheet, the maximum operating temperature is limited such that the factor f_T remains of sufficient magnitude for the design loads. The interface pressure due to expanding the tube at fabrication or the interface pressure due to differential thermal expansion may be determined analytically or experimentally.

(-c) As an alternate to (-b) above, when the tube expands less than or equal to the tubesheet, joint acceptability shall be determined by shear load test described in 4.21.3.3. Two sets of specimens shall be tested. The first set shall be tested at the proposed operating temperature. The second set shall be tested at room temperature after heat soaking at the proposed operating temperature for 24 hr. The proposed operating temperature is acceptable if the provisions of 4.21.3.5 are satisfied.

(g) The Manufacturer shall prepare written procedures for joints that are expanded (whether welded and expanded or expanded only) for joint strength (see Annex 4-E). The Manufacturer shall establish the variables that affect joint repeatability in these procedures. The procedures shall provide detailed descriptions or sketches of enhancements, such as grooves, serrations, threads, and coarse machining profiles. The Manufacturer shall make these written procedures available to the Authorized Inspector.

4.21.1.2 Definitions.

(a) Full-Strength Weld — A full-strength tube-to-tubesheet weld is one in which the design strength is equal to or greater than the axial tube strength, Ft. When the weld in a tube-to-tubesheet joint meets the requirements of 4.21.2.2, it is a full-strength weld and the joint does not require qualification by shear load testing. Such a weld also provides tube joint leak tightness.

(b) Partial-Strength Weld — A partial-strength weld is one in which the design strength is based on the mechanical and thermal axial tube loads (in either direction) that are determined from the actual design conditions. The maximum allowable axial load of this weld may be determined in accordance 4.21.2.3 or 4.21.3. When the weld in a tube-to-tubesheet joint meets the requirements of 4.21.2.3, the joint does not require qualification by shear load testing. Such a weld also provides tube joint leak tightness.

(c) Seal Weld — A tube-to-tubesheet seal weld is one used to supplement an expanded tube joint to ensure leak tightness. Its size has not been determined based on axial tube loading.

(d) Welded-Only Joint — A tube-to-tubesheet joint that is made by welding the end of the tube to the tubesheet.

(e) Welded-and-Expanded Joint — A tube-to-tubesheet joint that is made by both welding and expanding.

(f) Expanded Joint — A tube-to-tubesheet joint that is made by expanding the tube into the tube hole in a manner that produces a determinable allowable axial joint load.

(g) Tube hole enhancement — A groove or other modification of the tube hole surface that increases the allowable axial joint load.

4.21.2 JOINT STRENGTH BY CALCULATION

4.21.2.1 Scope. These rules provide a basis for establishing weld sizes and allowable joint loads by calculation for full-strength and partial-strength tube-to-tubesheet welds. These rules apply to welded-only joints and welded-and-expanded joints in which the strength of the expansion is not considered. These rules cover the welds shown in Figure 4.21.1.

4.21.2.2 Full-Strength Welds. Full-strength welds shown in Figure 4.21.1 shall conform to the following requirements:

(a) The size of a full-strength weld shall be determined in accordance with 4.21.2.4.

(b) The maximum allowable axial load in either direction on a tube-to-tubesheet joint with a full-strength weld shall be $L_{\max} = kF_t$.

4.21.2.3 Partial-Strength Welds. Partial-strength welds shown in Figure 4.21.1 shall conform to the following requirements:

(a) The size of a partial-strength weld shall be determined in accordance with 4.21.2.4.

(b) The maximum allowable axial load in either direction on a tube-to-tubesheet joint with a partial-strength weld shall be $L_{\max} = k(F_f + F_g)$, but not greater than kF_t.

4.21.2.4 Weld Size Design Equations.

(a) The size of tube-to-tubesheet strength welds shown in Figure 4.21.1 shall conform to the following requirements:

(1) For fillet welds shown in Figure 4.21.1, sketch (a), the following applies:

(-a) Calculate the minimum required length of the fillet weld leg.

$$a_r = \sqrt{(0.75d_o)^2 + 2.73t(d_o - t)f_w f_d} - 0.75d_o \tag{4.21.1}$$

(-b) For full-strength welds, $a_f \geq \max[a_r, 1.4t]$.

(-c) For partial-strength welds, $a_f \geq a_r$.

(2) For groove welds shown in Figure 4.21.1, sketch (b), the following applies:

(-a) Calculate the minimum required length of the groove weld leg.

$$a_r = \sqrt{(0.75d_o)^2 + 1.76t(d_o - t)f_w f_d} - 0.75d_o \tag{4.21.2}$$

(-b) For full-strength welds, $a_g \geq \max[a_r, t]$.

(-c) For partial-strength welds, $a_g \geq a_r$.

(3) For combined groove and fillet welds shown in Figure 4.21.1, sketch (c) where $a_f = a_g$ the following applies:

(-a) Calculate the minimum required length of the combined weld legs.

$$a_r = 2\left[\sqrt{(0.75d_o)^2 + 1.07t(d_o - t)f_w f_d} - 0.75d_o\right] \tag{4.21.3}$$

(-b) For full-strength welds, $a_c \geq \max[a_r, 1.2t]$.
(-c) For partial-strength welds, $a_c \geq a_r$.
(-d) Calculate a_f and a_g using the following equations:

$$a_f = \frac{a_c}{2} \tag{4.21.4}$$

$$a_g = \frac{a_c}{2} \tag{4.21.5}$$

(4) For combined groove and fillet welds shown in Figure 4.21.1, sketch (d) where a_f is not equal to a_g, a_r the following applies:
(-a) Choose a_g and calculate the minimum required length of the fillet weld leg.

$$a_r = \sqrt{\left(0.75d_o\right)^2 + 2.73t\left(d_o - t\right)f_w f_d f_f} - 0.75d_o \tag{4.21.6}$$

(-b) For full-strength welds $a_c \geq \max[(a_r + a_g), 1.4t - 0.4a_g]$.
(-c) For partial-strength welds $a_c \geq (a_r + a_g)$.
(-d) Calculate a_f using the following equation:

$$a_f = a_c - a_g \tag{4.21.7}$$

(5) For inset fillet welds shown in Figure 4.21.1, sketch (e), the following applies:
(-a) Calculate the minimum required length of the fillet weld leg.

$$a_r = 0.75d_o - \sqrt{\left(0.75d_o\right)^2 - 2.73t\left(d_o - t\right)f_w f_d} \tag{4.21.8}$$

(-b) Full-strength welds are not possible with this configuration.
(-c) For partial-strength welds, $t \geq a_f \geq a_r$. If $a_r > t$, joint load cannot be calculated in accordance with this section. See 4.21.3.
(6) For combined groove and inset fillet welds shown in Figure 4.21.1, sketch (f), a_r the following applies:
(-a) Choose a_f and calculate the minimum required length of the groove weld.

$$a_r = \sqrt{\left(0.75d_o\right)^2 + 1.76t\left(d_o - t\right)f_w f_d f_g} - 0.75d_0 \tag{4.21.9}$$

(-b) For full-strength welds, $a_c \geq \max[(a_r + a_f), (t + 0.3a_f)]$.
(-c) For partial-strength welds, $a_c \geq (a_f + a_f)$.
(b) In weld strength factors used in (a) above shall be calculated using the following equations:

$$f_d = 1.0 \qquad \text{for full-strength welds} \tag{4.21.10}$$

$$f_d = \frac{F_d}{F_t} \qquad \text{for partial-strength welds} \tag{4.21.11}$$

$$f_f = 1 - \frac{F_g}{f_d F_t} \tag{4.21.12}$$

$$f_g = 1 - \frac{F_f}{\left(f_d F_t\right)} \tag{4.21.13}$$

$$f_w = \frac{S}{S_w} \tag{4.21.14}$$

where

$$F_f = 0.55\pi a_f\left(d_o + 0.67a_f\right)S_w \text{ for face fillet welds shown in Figure 4.21.1(a), (c), (d)} \quad (4.21.15)$$

where

$$F_f = 0.55\pi a_f\left(d_o - 0.67a_f\right)S_w \text{ for inset fillet welds as shown in Figure 4.21.1(e) \& (f)} \quad (4.21.16)$$

where

$$F_g = 0.85\pi a_g\left(d_o + 0.67a_g\right)S_w \quad (4.21.17)$$

$$F_t = \pi t\left(d_o - t\right)S \quad (4.21.18)$$

4.21.3 JOINT STRENGTH FACTORS

4.21.3.1 Scope. These rules provide a basis for establishing allowable joint loads through the use of strength factors. Some acceptable geometries and combinations of welded and mechanical joints are described in Table 4.21.1. Some acceptable types of welded joints are illustrated in Figure 4.21.2.

(a) Geometries, including variations in tube pitch, fastening methods, and combinations of fastening methods, not described or shown, may be used, provided qualification tests have been conducted and applied in compliance with the procedures in 4.21.3.3 and 4.21.3.4.

(b) Materials for welded tube-to-tubesheet joints that do not meet the requirements of Part 6, but in all other respects meet the requirements of this Division, may be used if qualification tests of the tube-to-tubesheet joint have been conducted and applied in compliance with the procedures in 4.21.3.3 and 4.21.3.4.

4.21.3.2 Maximum Axial Loads. The maximum allowable axial load in either direction on tube-to-tubesheet joints shall be determined in accordance with the following:

$$L_{max} = kA_tSf_r \qquad \text{Joint Types a, b, e} \quad (4.21.19)$$

$$L_{max} = \min\left(kA_tSf_{re},\ kA_tS\right) \qquad \text{Joint Types f, g, h} \quad (4.21.20)$$

$$L_{max} = \min\left(kA_tSf_ef_rf_yf_T,\ kA_tS\right) \qquad \text{Joint Types i, j, k} \quad (4.21.21)$$

where

$$A_t = \pi\left(d_o - t\right)t \quad (4.21.22)$$

$$f_T = \frac{P_o + P_T}{P_o} \quad (4.21.23)$$

The following equations may be used to calculate P_o and P_T:

$$P_e = S_{y,t}\frac{t + r_o\left(\dfrac{S_y}{S_{y,t}}\right)}{t + r_o}\left(1.945 - 1.384\frac{d_i}{d_o}\right) \quad (4.21.24)$$

$$P_o = P_e\left[1 - \left(\frac{d_i}{d_o}\right)^2\right] - \frac{2}{\sqrt{3}}S_{y,t}\left[\ln\frac{d_o}{d_i}\right] \tag{4.21.25}$$

$$R_m = r_o - \frac{t}{2} \tag{4.21.26}$$

$$P_T = \frac{\frac{R_m}{d_o}E_t\left[\alpha_t d_o\left(T - T_a\right) - \alpha d_o\left(T - T_a\right)\right]}{\left(\frac{d_o^{\,2}}{t} - R_m\right) + R_m\left(2.9\frac{E_t}{E} - 0.3\right)} \tag{4.21.27}$$

4.21.3.3 Shear Load Test.

4.21.3.3.1 Flaws in the specimen may affect results. If any test specimen develops flaws, the retest provisions of 4.21.3.3.11 shall govern.

4.21.3.3.2 If any test specimen fails because of mechanical reasons, such as failure of testing equipment or improper specimen preparation, it may be discarded and another specimen taken from the same heat.

4.21.3.3.3 The shear load test subjects a full-size specimen of the tube joint under examination to a measured load sufficient to cause failure. In general, the testing equipment and methods are given in the Methods of Tension Testing of Metallic Materials (ASTM E 8). Additional fixtures for shear load testing of tube-to- tubesheet joints are shown in Figure 4.21.3.

4.21.3.3.4 The test block simulating the tubesheet may be circular, square, or rectangular in shape, essentially in general conformity with the tube pitch geometry. The test assembly shall consist of an array of tubes such that the tube to be tested is in the geometric center of the array and completely surrounded by at least one row of adjacent tubes. The test block shall extend a distance of at least one tubesheet ligament beyond the edge of the peripheral tubes in the assembly.

4.21.3.3.5 All tubes in the test block array shall be from the same heat and shall be installed using identical procedures.

(a) The finished thickness of the test block may be less but not greater than the tubesheet it represents. For expanded joints, made with or without welding, the expanded area of the tubes in the test block may be less but not greater than that for the production joint to be qualified.

(b) The length of the tube used for testing the tube joint need only be sufficient to suit the test apparatus. The length of the tubes adjacent to the tube joint to be tested shall be not less than the thickness of the test block to be qualified.

4.21.3.3.6 The procedure used to prepare the tube-to-tubesheet joints in the test specimens shall be the same as used for production.

4.21.3.3.7 The tube-to-tubesheet joint specimens shall be loaded until mechanical failure of the joint or tube occurs. The essential requirement is that the load be transmitted axially.

4.21.3.3.8 Any speed of testing may be used, provided load readings can be determined accurately.

4.21.3.3.9 The reading from the testing device shall be such that the applied load required to produce mechanical failure of the tube-to-tubesheet joint can be determined.

4.21.3.3.10 For determining $f_{r,\text{test}}$ for joint types listed in Table 4.21.1, a minimum of three specimens shall constitute a test. The value of $f_{r,\text{test}}$ shall be calculated in accordance with 4.21.3.4.1 using the lowest value of L_{test}. In no case shall the value of $f_{r,\text{test}}$ using a three specimen test exceed the value of $f_{r,\text{test}}$ given in Table 4.21.1. If the value of $f_{r,\text{test}}$ so determined is less than the value for $f_{r,\text{test}}$ given in Table 4.21.1, retesting shall be performed in accordance with 4.21.3.3.11, or a new three specimen test shall be performed using a new joint configuration or fabrication procedure. All previous test data shall be rejected. To use a value of $f_{r,\text{test}}$ greater than the value given in Table 4.21.1, a nine specimen test shall be performed in accordance with 4.21.3.3.11.

4.21.3.3.11 For joint types not listed in Table 4.21.1, to increase the value of $f_{r,\text{test}}$ for joint types listed in Table 4.21.1, or to retest joint types listed in Table 4.21.1, the tests to determine $f_{r,\text{test}}$ shall conform to the following:

(a) A minimum of nine specimens from a single tube shall be tested. Additional tests of specimens from the same tube are permitted, provided all test data are used in the determination of $f_{r,\text{test}}$. If a change in the joint design or its manufacturing procedure is necessary to meet the desired characteristics, complete testing of the modified joint shall be performed.

(b) In determining the value of $f_{r,\text{test}}$, the mean value of L_{test} shall be determined and the standard deviation, sigma, about the mean shall be calculated. The value of $f_{r,\text{test}}$ shall be calculated using the value of L_{test} corresponding to −2 sigma, using the applicable equation in 4.21.3.4. In no case shall $f_{r,\text{test}}$ exceed 1.0.

4.21.3.3.12 Once shear load tests have been successfully completed for a tube-to-tubesheet joint design, the Manufacturer that produced the test specimen may use the calculated $f_{r,\text{test}}$ for any production tube-to-tubesheet joint design that the Manufacturer produces having the same geometry, material nominal composition, specified ultimate tensile strength, and fabrication procedure used for the shear load test specimen. The fabrication procedure shall contain or reference the test qualification information required by 4-E.5.2 and/or Section IX, QW-193.1, as applicable.

4.21.3.4 Acceptance Standards for Joint Efficiency Factor Determined by Test.

4.21.3.4.1 The value of $f_{r,\text{test}}$ determined by testing shall be calculated as follows:

$$f_{r,\text{test}} = \frac{L_{\text{test}}}{A_t S_T} \qquad \text{Joint Types a, b, b-1, e} \tag{4.21.28}$$

$$f_{r,\text{test}} = \frac{L_{\text{test}}}{A_t S_T f_e f_y} \qquad \text{Joint Types f, g, h, i, j, k} \tag{4.21.29}$$

4.21.3.4.2 The value of $f_{r,\text{test}}$ shall be used for $f_{r,\text{test}}$ in the equation for $L_{\max}$.

4.21.3.5 Acceptance Standards for Proposed Operating Temperatures Determined by Test. The proposed operating conditions shall be acceptable if both of the following conditions are satisfied:

$$L_{1,\text{test}} \ge A_t f_e f_y S_T \left(S_u / S_{ua}\right) \tag{4.21.30}$$

$$L_{2,\text{test}} \ge A_t f_e f_y S_T \tag{4.21.31}$$

4.21.4 NOMENCLATURE

- a_c = length of the combined weld(s) measured parallel to the longitudinal axis of the tube at its outside diameter. For fillet only welds, $a_c = a_f$. For groove only welds, $a_c = a_g$. These dimensions are illustrated in Figures 4.21.1 and 4.21.2.
- a_f = fillet weld leg. For unequal leg fillets, a_f shall be the size of the smaller of the two legs.
- a_g = depth of groove weld. Depth can be achieved by chamfer or non-chamfer groove.
- a_r = minimum required length of the weld(s) under consideration
- A_t = tube cross-sectional area
- d_i = nominal tube inside diameter
- d_o = nominal tube outside diameter
- E = modulus of elasticity for tubesheet material at T
- E_t = modulus of elasticity for tube material at T
- F_d = design strength, but not greater than F_t
- f_d = ratio of the design strength to the tube strength
- f_e = factor for the length of the expanded portion of the tube. $f_e = \min[(l/d_o), 1.0]$ for tube joints made with expanded tubes in tube holes without enhancement and $f_e = 1.0$ for tube joints made with expanded tubes in tube holes with enhancement. An expanded joint is a joint between the tube and tubesheet produced by applying an expanding force inside the portion of the tube to be engaged in the tubesheet. The expanding force shall be set to values necessary to effect sufficient residual interface pressure between the tube and hole for joint strength.
- F_f = fillet weld strength, but not greater than F_t

f_f = ratio of the fillet weld strength to the design strength
F_g = groove weld strength, but not greater than F_t
f_g = ratio of the groove weld strength to the design strength
f_r = factor to define the efficiency of joint, set equal to the value of $f_{r,\text{test}}$ or $f_{r,\text{notest}}$. $f_{r,\text{test}}$ is equal to the value calculated from results of test in accordance with 4.21.3.4 or as tabulated in Table 4.21.1, whichever is less, except as permitted in 4.21.3.3.11. $f_{r,\text{notest}}$ is equal to maximum allowable value without qualification test in accordance with Table 4.21.1
F_t = axial tube strength
$f_{r,\text{test}}$ = factor to define the efficiency of joint established in a test
$f_{r,\text{notest}}$ = factor to define the efficiency of joint established without a test
f_{re} = factor for the overall efficiency of welded and expanded joints. This is the maximum of the efficiency of the weld alone, $f_r(b)$, and the net efficiency of the welded and expanded joint.
= $\max[f_e f_r f_y f_T, f_r(b)]$
f_T = factor to account for the increase or decrease of tube joint strength due to radial differential thermal expansion at the tube-to-tubesheet joint. Acceptable values of f_T may range from 0 to greater than 1. When the f_T value is negative, it shall be set to 0.
f_w = weld strength factor
f_y = factor for differences in the mechanical properties of tubesheet and tube materials. $f_y = \min[(S_y/S_{y\ t}), 1.0]$ for expanded joints. When f_y is less than 0.60, qualification tests in accordance with 4.21.3.3 and 4.21.3.4 are required.
k = tube load factor
= 1.0 for loads due to pressure-induced axial forces
= 1.0 for loads due to thermally induced or pressure plus thermally induced axial forces on welded-only joints where the thickness through the weld throat is less than the nominal tube wall thickness t
= 2.0 for loads due to thermally induced or pressure plus thermally induced axial forces on all other tube-to-tubesheet joints
l = expanded tube length
$L_{\max}$ = maximum allowable axial load in either direction on the tube-to-tubesheet joint
L_{test} = axial load at which failure of the test specimens occur
$L_{1,\text{test}}$ = lowest axial load at which failure occurs at operating temperature
$L_{2,\text{test}}$ = lowest axial load at which failure of heat soaked specimen tested at room temperature occurs
P_e = tube expanding pressure
P_o = interface pressure between the tube and tubesheet that remains after expanding the tube at fabrication. This pressure may be established analytically or experimentally, but shall consider the effect of change in material strength at operating temperature.
P_T = interface pressure between the tube and tubesheet due to differential thermal growth. This pressure may be established analytically or experimentally.
R_m = mean tube radius
r_o = tube outside radius
S = allowable stress from Annex 3-A for the tube at the design temperature. For a welded tube, S is the equivalent allowable stress for a seamless tube.
S_T = tensile strength for tube material from the material test report
S_t = allowable stress from Annex 3-A of the material to which the tube is welded (see 3.3.7.4)
S_u = tensile strength for tube material at operating temperature from Annex 3-D
S_{ua} = tensile strength for tube material at room temperature from Annex 3-D
S_w = allowable stress in weld, $S_w = [S, S_t]$
S_y = tubesheet specified minimum yield strength at the design temperature from Annex 3-D
$S_{y,t}$ = tube specified minimum yield strength at the design temperature from Annex 3-D
T = tubesheet design temperature
T_a = ambient temperature
t = nominal tube wall thickness
α = mean coefficient of thermal expansion of tubesheet material at T
α_t = mean coefficient of thermal expansion of tube material at T

4.21.5 TABLES

Table 4.21.1
Efficiencies for Welded and/or Expanded Tube-to-Tubesheet Joints

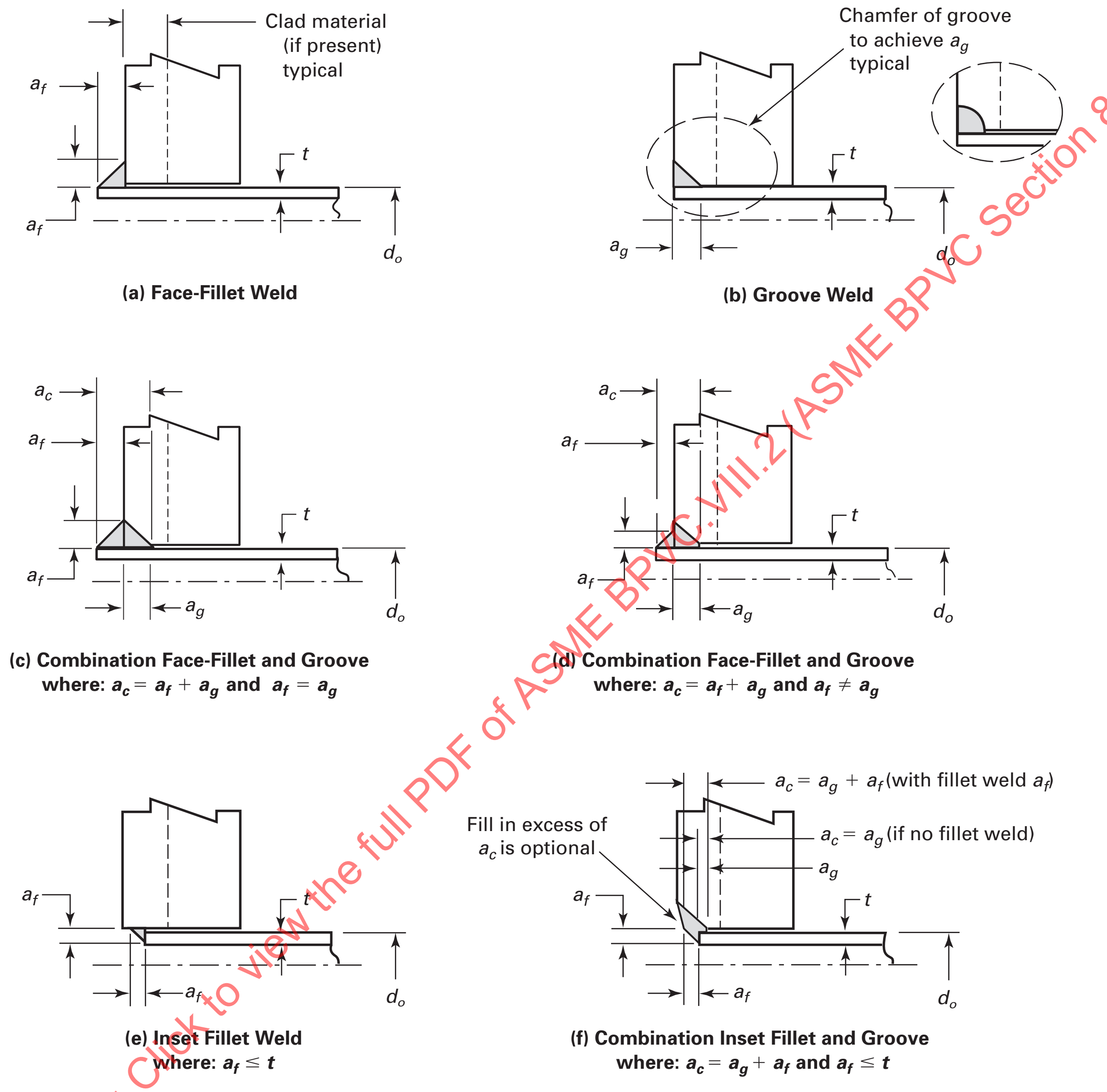
Joint Type	Description [Note (1)]	Notes	$f_{r,test}$ [Note (2)]	$f_{r,no\ test}$
a	Welded only, total weld size $(0.7a_f + a_g) \geq t$	(4)	1.00	...
b	Welded only, total weld size $(0.7a_f + a_g) < t$	(4)	0.70	...
e	Welded, total weld size $(0.7a_f + a_g) \geq t$, and expanded	(3)	1.00	0.80
f	Welded, total weld size $(0.7a_f + a_g) < t$, and expanded Enhanced with two or more grooves	(3), (5), (6), (7), (8)	0.95	0.75
g	Welded, total weld size $(0.7a_f + a_g) < t$, and expanded Enhanced with single groove	(3), (5), (6), (7), (8)	0.85	0.65
h	Welded, total weld size $(0.7a_f + a_g) < t$, and expanded Not enhanced	(3), (5), (6)	0.70	0.50
i	Expanded Enhanced with two or more grooves	(5), (6), (7), (8)	0.90	0.70
j	Expanded Enhanced with single groove	(5), (6), (7), (8)	0.80	0.65
k	Expanded Not enhanced	(5), (6)	0.60	0.50

NOTES:
(1) For joint types involving more than one fastening method, the sequence used in the joint description does not necessarily indicate the order in which the operations are performed.
(2) The use of the $f_{r,\ test}$ factor requires qualification in accordance with 4.21.3.3 and 4.21.3.4.
(3) The value of $f_{r,\ no\ test}$ applies only to material combinations as provided for under Section IX. For material combinations not provided for under Section IX, f_r shall be determined by test in accordance with 4.21.3.3 and 4.21.3.4.
(4) $f_{r,\ no\ test}$, is not permitted for welded only joint types in accordance with Section 4.21.3. Refer to 4.21.2..
(5) If $d_o/(d_o - 2t) < 1.05$ or $d_o/(d_o - 2t) > 1.410$, f_r shall be determined by test in accordance with 4.21.3.3 and 4.21.3.4.
(6) If the nominal pitch (center-to-center distance of adjacent tube holes) is less than $d_o + 2t$, f_r shall be determined by test in accordance with 4.21.3.3 and 4.21.3.4.
(7) The Manufacturer may use other means to enhance the strength of expanded joints, provided however, that the joints are tested in accordance with 4.21.3.3 and 4.21.3.4.
(8) For explosive and hydraulic expansion, grooves shall be a minimum of $1.1[(d_o - t)t]^{0.5}$ wide. For explosively or hydraulically expanded joints with single grooves meeting this requirement, f_r for Joint Type f may be used in lieu of that for Joint Type g, and f_r for Joint Type i may be used in lieu of that for Joint Type j, as applicable.

4.21.6 FIGURES

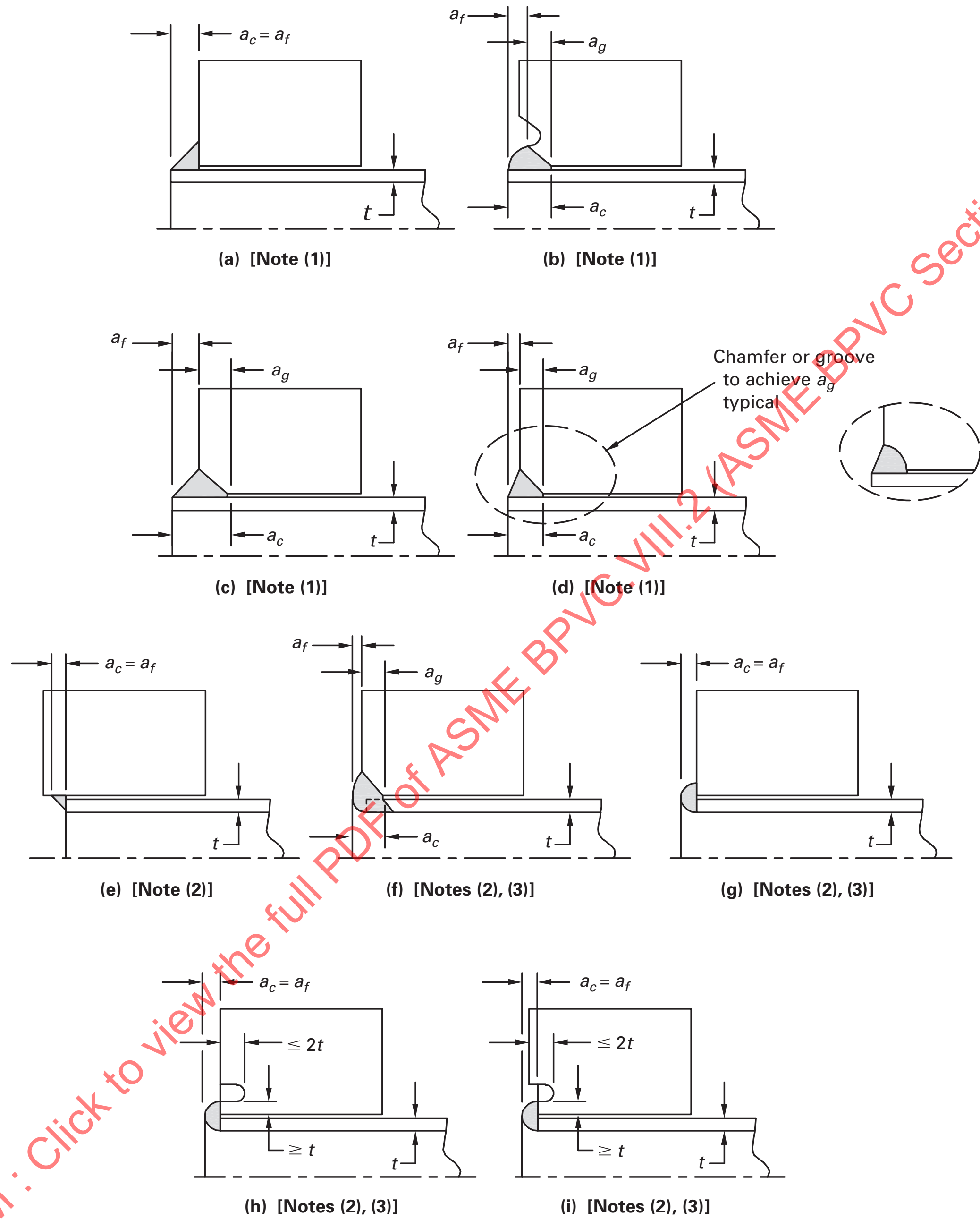
(21)

Figure 4.21.1
Tube-to-Tubesheet Joints Acceptable to Determine Joint Strength by Calculation



(21)

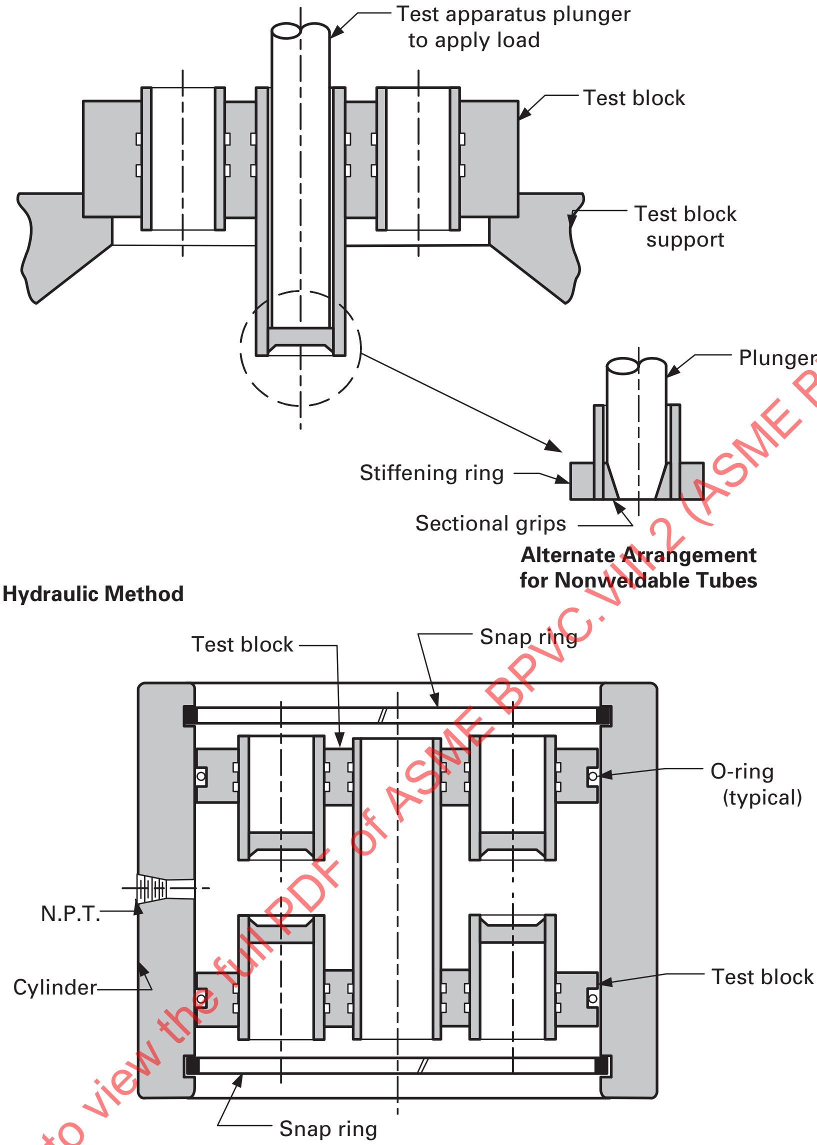
Figure 4.21.2
Some Acceptable Types of Tube-to-Tubesheet Joints



NOTES:
(1) Sketches (a) through (d) show some acceptable weld geometries where thickness through the weld throat may be sized ≥ t.
(2) Sketches (e) through (i) show some acceptable weld geometries where thickness through the weld throat is less than t.
(3) For these geometries where weld length cannot be established by dimensions prior to weld, dimension a_c shall be verified during WPS qualification per QW-193 and design strength shall be calculated in accordance with 4.21.3.2.

(21)

Figure 4.21.3
Typical Test Fixtures for Expanded or Welded Tube-to-Tubesheet Joints



ANNEX 4-A

(Currently Not Used)

ANNEX 4-B
GUIDE FOR THE DESIGN AND OPERATION OF QUICK-ACTUATING (QUICK-OPENING) CLOSURES

(Informative)

4-B.1 INTRODUCTION

4-B.1.1 This Annex provides guidance in the form of recommendations for the installation, operation, and maintenance of quick-actuating closures. This guidance is primarily for the use of the Owner and the user. The safety of the quick-actuating closure is the responsibility of the user. This includes the requirement for the user to provide training for all operating personnel, follow safety procedures, periodically inspect the closure, provide scheduled maintenance, and have all necessary repairs made in a timely fashion.

4-B.1.2 This Annex also contains guidance for use by the Designer. The rules specific to the design and construction of quick-actuating closures are found in 4.8 of this division.

4-B.1.3 The Manufacturer should supply the Owner a copy(ies) of the Installation, Operational, and Maintenance Manual for the quick-actuating closure which should, as a minimum, address the requirements described in this Annex. The Owner should supply a copy of the Installation, Operational, and Maintenance Manual to the user.

4-B.2 RESPONSIBILITIES

4-B.2.1 It is the responsibility of the user to ensure that the sensing and safety devices and the equipment specified by the Manufacturer are properly installed before initial operation, and maintained during subsequent operation. Provision of written operation and maintenance procedures and training of personnel are also the responsibility of the Owner or user.

4-B.2.2 The user must not remove any devices furnished or specified by the Manufacturer of the vessel, and any repairs or replacements must be the same as, or equal to, the original equipment furnished or specified by the Manufacturer.

4-B.2.3 The rules of this Annex do not require these devices to be supplied by the Manufacturer of the vessel or of the quick-actuating closure.

4-B.3 DESIGN

4-B.3.1 Code rules cannot be written to address each specific design; therefore, engineering judgment exercised by a qualified designer with the necessary experience is required to achieve a safe design. Because of the multiple requirements imposed on the design, it should be prepared by a designer with suitable experience and training in the design of quick-actuating closures.

4-B.3.2 The design must be safe, reliable, and allow for quick and safe opening and closing. Therefore, sensing and safety devices and equipment are integral and vitally important parts of the closure, and are to be furnished or specified by the Manufacturer of the vessel or quick-actuating closure. These devices must never be removed by the user.

4-B.3.3 It should be noted that there is a higher likelihood of personnel being close to the vessel and the closure when accidents during opening occur, especially those due to violations of operating procedures. An example is attempting to pry open the closure when they believe the vessel has been depressurized and it may not be.

4-B.3.4 The passive safety features described below can help to protect against such actions, but most can still be subverted. Protection against subversion of safety features is covered under Inspection, Training, and Administrative Controls.

4-B.3.5 Structural Elements in the vessel and the closure require design margins. However, it is also important to provide the suggested features listed below, for erroneous opening.

(a) Passive Actuation - A passively actuated safety feature or device does not require the operator to take any action to provide safety. An example is a pressure relief valve in a vessel or a pressure-actuated locking device in a quick-actuating closure.

(b) Redundancy - A redundant safety feature or device is one of two or more features or devices that perform the same safety function. Two pressure-actuated locking devices in parallel are an example application to quick-actuating closures. Another example is two or more independent holding elements, the failure of one of which does not reduce the capability to withstand pressure loadings below an acceptable level.

(c) Fail-Safe Behavior - If a device or element fails, it should fail in a safe mode. An example applicable to quick-actuating closures is a normally-closed electrical interlock that stays locked if power fails.

(d) Multiple Lines of Defense - This can consist of any combination of two or more items from the list above. They should consist, at the very least, of warnings or alarms.

4-B.3.6 Pressure controls and sensors that operate well at 350 kPa to 700 kPa (50 psi to 100 psi) or at much greater pressure do not operate well at very low pressure. For example, they may not sense a small, static head of hot fluid. Certain accidents can occur because of the release of hot fluid under static head alone, or under very low pressure. To protect against such accidents, separate controls and sensors may be used to maintain operating pressure on the one hand, and others may be required to prevent inappropriate opening at low pressures.

4-B.3.7 It may be necessary or desirable to utilize electrical or electronic devices and interlocks. If these are used, careful detailed installation, operating, and maintenance instructions (see following) shall be required.

4-B.3.8 The effects of repetitive loading shall be considered. There are two phenomena that are of major concern. The first is the wear produced by repetitive actuation of the mechanism. This can generally be mitigated by routine maintenance. The second is fatigue damage produced in the vessel or in the closure by repetitive actuation of the mechanism or by repetitive pressurization and depressurization.

4-B.3.9 The code does not provide explicit guidance for the evaluation or mitigation of wear. As well as proper maintenance, the selection of suitable materials for mating wear surfaces and control of contact stresses is necessary during the design process to properly control wear.

4-B.4 INSTALLATION

4-B.4.1 The manufacturer shall provide clear instructions for the installation of the quick-actuating closure itself and any adjustments that are necessary in the field. An example is adjustment of wedges or clamps.

4-B.4.2 Instructions, preferably including schematics and drawings, shall be provided for the installation, adjustment, and checkout of interlocks and warning devices.

4-B.4.3 Maintenance

4-B.4.4 Vessels with quick-actuating closures are commonly installed in industrial environments subject to dirt, moisture, abrasive materials, etc. These environmental factors are detrimental to safe and reliable operation of mechanical, electrical, and electronic sensors and safety devices. Therefore, the user should establish a suitable cleaning and maintenance interval, and a means to verify that the equipment has been properly cleaned and maintained.

4-B.4.5 Accidents have occurred because gaskets have stuck, and have released suddenly when pried open. Many soft gaskets (60-70 Shore A Scale) have a combined shelf life and operating life of as little as six months. Aging can change the properties of the gasket material and change the gasket dimensions, impeding its proper function.

4-B.5 INSPECTION

4-B.5.1 It is recommended that the user inspect the completed installation including the pressure gauges before it is permitted to operate. Records of this inspection should be retained.

4-B.5.2 It is recommended that the user establish and document a periodic in-service inspection program, and that this program be followed and the results documented.

4-B.6 TRAINING

4-B.6.1 Many accidents involving quick-actuating closures have occurred because the operators have been unfamiliar with the equipment or its safety features. The greater safety inherent in current designs has sometimes been produced by the use of sophisticated mechanical, electrical or electronic control devices. In order to make these features produce the maximum safety, personnel should be properly trained in their operation and maintenance.

4-B.6.2 Note that accidents may occur because hot fluid remains present in the vessel at atmospheric pressure of 15 kPa to 20 kPa (2 psig to 3 psig). If the vessel is forced open while under this pressure, then injuries may occur. Such specific accident sources should be guarded against by training and by administrative procedures. It is important that sound written operating procedures, understandable by the operating personnel and multi-lingual if necessary, exist for the quick-actuating closure, and that the operators be trained in the proper use of all interlocks, sensing devices, and manual closure mechanisms.

4-B.6.3 Provision of written operation and maintenance procedures and training of personnel are the responsibility of the user.

4-B.6.4 As part of the training program, testing should be performed to ensure that the trainee understands the material he or she is trained in. Records should be retained by the user.

4-B.7 ADMINISTRATIVE CONTROLS

The user should provide administrative controls covering training, cleanliness, operation, periodic inspection, and maintenance of equipment with quick-actuating closures. Records should be retained by the user.

ANNEX 4-C
BASIS FOR ESTABLISHING ALLOWABLE LOADS FOR TUBE-TO-TUBESHEET JOINTS

(21)

DELETED

ANNEX 4-D
GUIDANCE TO ACCOMMODATE LOADINGS PRODUCED BY DEFLAGRATION

(Normative)

4-D.1 SCOPE

When an internal vapor-air or dust-air deflagration is defined by the user or his designated agent as a load condition to be considered in the design, this Annex provides guidance for the designer to enhance the ability of a pressure vessel to withstand the forces produced by such conditions.

4-D.2 GENERAL

Deflagration is the propagation of a combustion zone at a velocity that is less than the speed of sound in the unreacted medium, whereas detonation is the propagation of a combustion zone at a velocity that is greater than the speed of sound in the unreacted medium. A detonation can produce significant dynamic effects in addition to pressure increases of great magnitude and very short duration, and is outside the scope of this Annex. This Annex only addresses the lower and slower loadings produced by deflagrations that propagate in a gas-phase.

The magnitude of the pressure rise produced inside the vessel by a deflagration is predictable with reasonable certainty. Unvented deflagration pressures can be predicted with more certainty then vented deflagration pressures. Methods are provided in the references listed in 4-D.5 to bound this pressure rise. Other methods may also be used to determine pressure rise.

4-D.3 DESIGN LIMITATIONS

The limits of validity for deflagration pressure calculations are described in References [1] and [2].

4-D.4 DESIGN CRITERIA

4-D.4.1 SAFETY MARGIN

As described in NFPA-69, (see Reference [1]), a vessel may be designed to withstand the loads produced by a deflagration:

(a) without significant permanent deformation; or

(b) without rupture (see Reference [3]).

A decision between these two design criteria should be made by the user or his designated agent based upon the likelihood of the occurrence and the consequences of significant deformation. It is noted that either (a) or (b) above will result in stresses for a deflagration that are larger than the basic Code allowable stress listed in Section II, Part D. Because of this, appropriate design details and nondestructive examination requirements shall be agreed upon between the user and designer.

These two criteria are very similar in principle to the Level C and Level D criteria, respectively, contained in Section III, Subsection NB for use with Class 1 vessels, (see References [4] and [5]). The limited guidance in NFPA 69 requires the application of technical judgments made by knowledgeable designers experienced in the selection and design of appropriate details. The Level C and Level D criteria in Section III provide the detailed methodology for design and analysis. The successful use of either NFPA 69 or Section III criteria for deflagration events requires the selection of materials for construction the will not fail because of brittle fracture during the deflagration pressure excursions.

4-D.4.2 LIKELIHOOD OF OCCURRENCE

For vapor-air and dust-air combustion, various methods of reducing the likelihood of occurrence are described in Reference [2]. It is good engineering practice to minimize the likelihood of occurrence of these events, regardless of the capability of the vessel to withstand them.

4-D.4.3 CONSEQUENCES OF OCCURRENCE

In deciding between designing to prevent significant permanent deformation (see 4-D.4.1(a)) or designing to prevent rupture (see 4-D.4.1(b)), the consequences of significant distortion of the pressure boundary should be considered. Either the aforementioned NFPA or Section III design criteria may be used: Each has been used successfully.

4-D.4.4 STRAIN CONCENTRATION

When developing a design to withstand either of the criteria sited above, the designer should avoid creating weak sections in the vessel at which strain can be concentrated. Examples of design details to avoid are partial penetration pressure boundary welds, cone to cylinder junctions without transition knuckles, large openings in heads or cylindrical shells which require special design consideration etc.

4-D.5 REFERENCES

[1] National Fire Protection Association (NFPA) 69, Standard on Explosion Prevention Systems, Chapter 5, Deflagration Pressure Containment, issue effective with the applicable Addenda of the ASME Boiler and Pressure Vessel Code.
[2] National Fire Protection Association (NFPA) 68, Guide for the Venting of Deflagrations, issue effective with the applicable Addenda of the ASME Boiler and Pressure Vessel Code.
[3] B.F. Langer, PVRC Interpretive Report of Pressure Vessel Research, Section 1- Design Considerations, 1.4 Bursting Strength, Welding Research Council Bulletin 95, April 1964.
[4] ASME Boiler and Pressure Vessel Code, Section III, Division 1, NB-3224, Level C Service Limits.
[5] ASME Boiler and Pressure Vessel Code, Section III, Division 1, NB-3225 and Appendix F, Level D Service Limits.

ANNEX 4-E
TUBE EXPANDING PROCEDURES AND QUALIFICATION

(Informative)

4-E.1 GENERAL

This Annex establishes requirements for procedure specifications for expanded tube-to-tubesheet joints

(a) designed using the test joint efficiencies, f_r (test), listed in Table 4.21.1;

(b) designed using the no-test joint efficiencies, f_r (no test), listed in Table 4.21.1; and

(c) used in tubesheets designed in accordance with the rules of 4.18 when the effective tube hole diameter depends upon the expanded depth of the tube ($\rho > 0$).

Leak tightness of expanded joints is not a consideration in 4.18 and Annex 4-C, and is therefore not considered in this Annex.

4-E.2 SCOPE

The rules in this Annex apply to preparation and qualification of tube expanding procedures for the types of expanding processes permitted in this Annex.

4-E.3 TERMS AND DEFINITIONS

Some of the more common terms relating to tube expanding are as follows:

explosive expanding: uniform pressure expanding in which the force of an explosion is applied to the length of tube to be expanded.

groove: an annular machined depression in a tube hole.

hybrid expanding: hydroexpanding or explosive expanding to a percent wall reduction that ensures maintenance of tube-hole contact, followed by roller expanding to the final percent wall reduction.

hydroexpanding: uniform pressure expanding in which hydraulic pressure is applied to the length of tube to be expanded.

near contact kinetic expanding: see *explosive expanding*.

parallel tube roller: tube rolling tool in which the taper angle of the mandrel and the taper angle of the hardened pins are approximately equal and opposite, thereby causing the pins to bear uniformly on the tube surface.

percent wall reduction: reduction in tube wall thickness due to expanding, expressed as a percent of the measured thickness of the tube.

progressive rolling: step rolling in which the first step begins at or near the front face of the tubesheet and successive steps progress toward the rear face.

prosser: see *segmental expander*.

prossering: expanding tubes with a segmental expander.

regressive rolling: step rolling in which the first step begins at or near the rear face of the tubesheet and successive steps progress toward the front face.

roller expanding: expanding by inserting a tube rolling tool into a tube aligned with a tube hole.

segmental expander: thick-walled, flanged cylinder with a tapered interior wall, cut axially into segments and held together by bands. A mandrel with a reverse taper in contact with the taper of the interior of the cylinder is thrust forward, forcing the segments outward to contact and expand the tube. The flange bears against the tube end or tubesheet face to maintain the position of the expander relative to the tube.

self-feeding rolling tool: tube rolling tool with the slots in the cage at an angle with the tool centerline such that rotating the mandrel in a clockwise direction causes the tool to feed into the tube and reversing the direction causes it to back out.

serrations: parallel, narrow grooves machined in a tube hole or on the exterior of a tube end.

step rolling: tube rolling in which successive, overlapping applications of the tube roller are applied in order to roll tubes into tubesheets thicker than approximately 2 in. (50 mm).

torque control: an electronic, hydraulic control or cam-operated reversing mechanism that causes a rolling tool driver to reverse direction when a preset level of torque is reached.

transition zone: region of an expanded joint in which the expanded part of the tube transitions to the unexpanded part.

tube end enhancement: treatment to that part of the tube O.D. to be expanded into a tubesheet hole to increase the strength of the expanded tube-to-tubesheet joint.

tube expanding: process of expanding a tube to a fully plastic state into contact with the surrounding metal of a tube hole that creates residual interface pressure between the tube and tube hole when the expanding tool is withdrawn.

tube hole enhancement: treatment to the tube hole to increase the strength of an expanded tube-to-tubesheet joint. Enhancements may be by means of grooves or serrations.

tube rolling tool: tool consisting of a slotted cylindrical cage that holds hardened pins into which a hardened tapered mandrel is thrust and rotated, to expand the tube.

two-stage expanding: explosive, hydraulic, or roller expanding in which in the first stage all the tubes are expanded into firm contact with the holes, followed by a second stage of expanding to the final specified percent wall reduction.

uniform pressure expanding: tube expanding by applying force equally on the surfaces of the length of tube to be expanded.

4-E.4 TUBE EXPANDING PROCEDURE SPECIFICATION (TEPS)

A TEPS is a written document that provides the tube expander operator with instructions for making production tube-to-tubesheet joint expansions in accordance with Code requirements (see Form TEXP-1). The Manufacturer is responsible for ensuring that production tube expanding is performed in accordance with a qualified TEPS that meets the requirements of 4-E.7.

NOTE: The instructions for completing Form TEXP-1 are provided in Table TEXP-1. The instructions are identified by parenthesized numbers corresponding to circled numbers in the form.

The TEPS shall address, as a minimum, the specific variables, both essential and nonessential, as provided in 4-E.7.1 for each process to be used in production expanding.

4-E.5 TUBE EXPANDING PROCEDURE QUALIFICATION

The purpose for qualifying a TEPS is to demonstrate that the expanded joint proposed for construction will be suitable for its intended application. The tube expanding procedure qualification establishes the suitability of the expanded joint, not the skill of the tube expander operator.

4-E.5.1 NO TEST QUALIFICATION

Tube expanding procedures not required to be qualified by 4-E.5.2 may be used for expanded tube joints meeting 4-E.1(b) or 4-E.1(c) without a qualification test, provided the Manufacturer maintains records indicating that the tube joints expanded using the tube expanding procedures were successfully tested in accordance with Part 8.

4-E.5.2 TEST QUALIFICATION

Tube expanding procedures to be used for expanded tube joints meeting 4-E.1(a) shall be qualified by the Manufacturer in accordance with the requirements of 4.21.1.1 and 4.21.3.3, and the qualification shall be documented in accordance with 4-E.5.3.

4-E.5.3 TUBE EXPANDING PROCEDURE QUALIFICATION RECORD (TEPQR) FOR TEST JOINT EFFICIENCIES

The TEPQR documents what occurred during expanding the test specimen and the results of the testing in accordance with the requirements of 4.21.1.1 and 4.21.3.3. In addition, the TEPQR shall document the essential variables and other specific information identified in 4-E.7 for each process used.

4-E.6 TUBE EXPANDING PERFORMANCE QUALIFICATION (TEPQ)

The purpose of performing a TEPQ is to demonstrate that the operator of the equipment is qualified to make an expanded joint of the type specified in the TEPS.

4-E.6.1 NO TEST QUALIFICATION

A tube expander operator not required to be qualified by 4-E.6.2 is qualified to expand tube joints meeting 4-E.1(b) or 4-E.1(c), provided the Manufacturer maintains records indicating that tube joints expanded by the operator were successfully tested in accordance with Part 8.

4-E.6.2 TEST QUALIFICATION

A tube expander operator is qualified to expand tube joints using tube expanding procedures that have been qualified in accordance with 4-E.5.2, provided the operator, under the direction of the Manufacturer, has prepared at least one specimen that meets the requirements of 4.21.1.1 and 4.21.3.3 for the applicable procedure.

4-E.7 TUBE EXPANDING VARIABLES

Variables are subdivided into essential variables that apply to all expanding processes, and essential and nonessential variables that apply to each expanding process. Essential variables are those in which a change, as described in specific variables, is considered to affect the mechanical properties of the expanded joint, and shall require requalification of the TEPS. Nonessential variables are those that may be changed at the Manufacturer's discretion and are included in the TEPS for instruction purposes.

4-E.7.1 ESSENTIAL VARIABLES FOR ALL EXPANDING PROCESSES

The following essential variables shall be specified for all expanding processes. The Manufacturer may define additional essential variables.

(a) method of measuring and controlling tube hole diameter
(b) limit of percentage of tube holes that deviate from the specified diameter tolerance and maximum tolerance of hole-diameter deviation
(c) limiting ratio of tube diameter to tube wall thickness
(d) minimum ratio of tubesheet thickness to tube diameter
(e) minimum ratio of drilling pitch to tube diameter
(f) details of tube and/or tube hole treatments for joint strength enhancement, including surface finish of tube holes, tube-hole and tube end serrations, and tube hole annular grooves
(g) tube-to-hole diametral clearance prior to expanding (fit)
(h) range of modulus of elasticity of tube material
(i) range of modulus of elasticity of tubesheet material
(j) range of specified minimum tube yield stresses listed in Annex 3-D
(k) maximum permissible increase of tube yield stress above the minimum yield stress specified in Annex 3-D
(l) specified minimum tubesheet yield stress listed in Annex 3-D
(m) minimum ratio of tubesheet to tube yield stress;[2] a ratio below 0.6 requires shear load testing

[2] Manufacturers are cautioned to calculate the minimum ratio based upon mill test values of the tube and tubesheet.

(n) minimum and maximum percent wall reduction[3]

(o) for welded tube joints where tubes are to be expanded after welding, the method of fixing tube position before welding, the setback from the front face of the tubesheet to onset of expanding, the treatment of weld and tube-end shrinkage before inserting the expanding mandrel, and any post-expansion heat treatment

(p) for tubes to be expanded before welding, the procedure to be used to remove all traces of lubricants and moisture from the surfaces to be welded

(q) distance from front face of tubesheet to commencement of expanding

(r) distance from rear face of tubesheet to end of expanding

(s) unrolled length between front and rear expansion

(t) lubrication and cooling of the expanding mandrel

(u) measured actual amount of expansion

(v) range of tube wall thickness

4-E.7.2 ESSENTIAL VARIABLES FOR ROLLER EXPANDING

The following are essential variables for roller expanding:

(a) tool driver type (electrical, air, hydraulic), power or torque rating

(b) number and length of overlapping steps

(c) direction of rolling (progressive or regressive)

(d) speed of rotation

(e) tool type (parallel or nonparallel)

(f) cage and pin length

(g) number of pins in the cage

(h) cage slot angle or tool manufacturer's tool number

(i) frequency of verifying percent wall reduction

(j) for tubes to be expanded after welding, amount of setback before expanding mandrel insertion due to weld and tube-end shrinkage

4-E.7.3 ESSENTIAL VARIABLES FOR HYDRAULIC EXPANDING

The following are essential variables for hydraulic expanding:

(a) hydraulic mandrel details or mandrel manufacturer's mandrel number(s)

(b) hydraulic expanding pressure

(c) precision of pressure control

(d) number of applications of hydraulic pressure

(e) permissible + and − deviation from specified hydraulic expanding pressure

4-E.7.4 ESSENTIAL VARIABLES FOR EXPLOSIVE EXPANDING

The following are essential variables for explosive expanding:

(a) number of applications of explosive force

(b) number of tubes to be simultaneously expanded

(c) tube supports in surrounding holes

(d) post-expanding tube-end cleaning

(e) size of the explosive load

(f) buffer material

(g) outside diameter of the buffer material

(h) inside diameter of the buffer material

(i) theoretical expanded O.D. of the tube based on original cross-sectional area and expanded I.D. of the tube as compared to the tubesheet hole diameter

[3] The Manufacturer may correlate rolling torque, hydraulic expanding pressure, or explosive charge with shear load tests. For explosive expanding, the Manufacturer may correlate interference of fit.

4-E.7.5 ESSENTIAL VARIABLES FOR HYBRID EXPANDING

The essential variables for hybrid expanding are the variables listed in 4-E.7.4 for the initial explosive expanding or 4-E.7.3 for the initial hydraulic expanding and the following:

(a) the range of percent wall reduction to be achieved by the initial expanding

(b) the range of total percent wall reduction to be achieved by the initial expanding and final rolling

4-E.7.6 NONESSENTIAL VARIABLES

The Manufacturer shall specify nonessential variables for each process.

FORM TEXP-1 TUBE EXPANDING PROCEDURE SPECIFICATION (TEPS)

Company Name:				By:	
Tube Expanding Procedure Specification No.		Date		Supporting TEPQR No.(s)	
Revision No.		Date			
Expanding Process(es)				Driver Type(s)	
(Rolling, Hydroexpanding, Explosive Expanding, Hybrid Expanding)				(Electric, Air, Hydraulic, Hydroexpanded, Explosive)	

JOINTS

Measurement and Control of Tube Hole		Tube Pitch	
Tube Hole Diameter and Tolerance		Maximum Tube to Hole Clearance Before Expanding	
Ratio Tube Diameter/Tube Wall Thickness		Minimum Ratio Drilling Pitch/Tube Diameter	
Maximum % Wall Reduction		Minimum % Wall Reduction	
Maximum Permissible Deviation from Specified Hole Diameter		Maximum Permissible % of Holes that Deviate	
Details of Tube End Hole Enhancement and/or Tube End Enhancement		Minimum Ratio Tubesheet Thickness/Tube Diameter	
Method of Fixing Tubes in Position		Length of Expansion	
Setback from Front Tubesheet Face Before Start of Expanding		Setback from Rear Tubesheet Face after Expanding	
Method of Removing Weld Droop		Method of Tube End and Hole Cleaning	
Other Joint Details:			

EXPANDING EQUIPMENT

Manufacturer(s), Model No.(s), Range of Tube Diameters and Thicknesses, Maximum Torque Output or Pressure.

Expanding Tool Model and Description			
Expanded Length per Application of Expanding Mandrel		No. of Applications/ Expanded Length	
Torque or Pressure Calibration System and Frequency		Explosive Charge and No.(s) of Applications	

PROPERTIES

Range of Tube Elastic Modulus			Range of Plate Elastic Modulus		
Range of Tube Yield Stress (Mill Test Report Values)	Min.			Max.	
Range of Tubesheet Yield Stress (Mill Test Report Values)	Min.			Max.	
Minimum Tubesheet Yield Stress/Tube Yield Stress Note: Values below 0.6 require shear load testing					

TUBES

Diameter Range		Thickness Range		Maximum Ratio Tube Diameter/Thickness	
Material Specifications					

TUBESHEETS

Thickness Range		Minimum Ratio of Tubesheet Thickness to Tube Diameter	
Material Specifications			

REMARKS

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FORM TEXP-1 TUBE EXPANDING PROCEDURE SPECIFICATION (TEPS)

1	Company Name: 1			By: 2	
2	Tube Expanding Procedure Specification No.	3	Date 4	Supporting TEPQR No.(s)	5
3	Revision No.	6	Date 7		
4	Expanding Process(es)	8	Driver Type(s)	9	
JOINTS					
5	Measurement and Control of Tube Hole	10	Tube Pitch	11	
6	Tube Hole Diameter and Tolerance	12	Maximum Tube to Hole Clearance Before Expanding	13	
7	Ratio Tube Diameter/Tube Wall Thickness	14	Minimum Ratio Drilling Pitch/Tube Diameter	15	
8	Maximum % Wall Reduction	16	Minimum % Wall Reduction	17	
9	Maximum Permissible Deviation from Specified Hole Diameter	18	Maximum Permissible % of Holes that Deviate	19	
10	Details of Tube End Hole Enhancement and/or Tube End Enhancement	20	Minimum Ratio Tubesheet Thickness/Tube Diameter	21	
11	Method of Fixing Tubes in Position	22	Length of Expansion	23	
12	Setback from Front Tubesheet Face Before Start of Expanding	24	Setback from Rear Tubesheet Face After Expanding	25	
13	Method of Removing Weld Droop	26	Method of Tube End and Hole Cleaning	27	
14	Other Joint Details:	28			
EXPANDING EQUIPMENT					
15	Manufacturer(s), Model No.(s), Range of Tube Diameters and Thicknesses, Maximum Torque Output or Pressure. 29				
16	Expanding Tool Model and Description	30			
17	Expanded Length per Application of Expanding Mandrel	31	No. of Applications/ Expanded Length	32	
18	Torque or Pressure Calibration System and Frequency	33	Explosive Charge and No.(s) of Applications	34	
PROPERTIES					
19	Range of Tube Elastic Modulus	35	Range of Plate Elastic Modulus	36	
20	Range of Tube Yield Stress (mill test report values)	Min. 37	Max. 38		
21	Range of Tubesheet Yield Stress (mill test report values)	Min. 39	Max. 40		
22	Minimum Tubesheet Yield Stress/Tube Yield Stress NOTE: Values below 0.6 require shear load testing.	41			
TUBES					
23	Diameter Range	42	Thickness Range	43	Maximum Ratio Tube Diameter/Thickness 44
24	Material Specifications	45			
TUBESHEETS					
25	Thickness Range	46	Minimum Ratio of Tubesheet Thickness to Tube Diameter	47	
26	Material Specifications	48			
27	**REMARKS:**	49, 50, 51			

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Table TEXP-1
Instructions for Filling Out TEPS Form

Ref. to Circled Nos. in Form TEXP-1	Explanation of Information to Be Provided
(1)	Show Manufacturer's name and address.
(2)	Show TEPS author's names.
(3)	Show Manufacturer's TEPS number.
(4)	Show applicable date of TEPS.
(5)	Insert number of supporting Tube Expanding Procedure Qualification Record (TEPQR).
(6)	Show revision number if any.
(7)	Insert date of revision if any.
(8)	Describe expanding process as torque-controlled expanding, hydraulic expanding, or explosive expanding. If hybrid expanding is to be performed, describe sequence, e.g., "hybrid expanding (hydraulic expanding to 3% wall reduction followed by torque-controlled roller expanding to 6% to 8% total wall reduction)."
(9)	Describe as hydraulic, explosive, air-driven torque controlled, electric torque controlled, or hydraulic torque controlled drive. If hybrid expanded, describe as hydraulic or explosive expanded + torque controlled air, torque controlled electric, or torque controlled hydraulic torque controlled drive.
(10)	Describe measuring equipment, e.g., "go–no/go gage," "internal 3 point micrometer," or similar measuring device. All equipment used for measurements shall be calibrated.
(11)	Minimum centerline distance between tube holes.
(12)	Show hole size and plus/minus tolerance.
(13)	Show diametrical clearance, e.g., 0.014 in. (for minimum of 96%) and 0.022 in. (for maximum of 4%).
(14)	Minimum and maximum ratio of tube O.D. to tube wall (O.D./t) for this TEPS.
(15)	Fill in nominal ratio of drilling pitch to tube diameter.
(16)	Fill in maximum percent wall reduction to which the TEPS applies.
(17)	Fill in minimum percent wall reduction to which the TEPS applies.
(18)	Enter maximum permissible deviation of hole from specified drilling size and tolerance, e.g., 0.01 in.
(19)	Enter maximum percent of holes that may deviate by the amount shown in (18).
(20)	Describe enhancements for joint strength, e.g., "two $^1/_8$ in. wide × $^1/_{64}$ in. grooves set 1 in. from inlet face with $^1/_2$ in. land between."
(21)	Fill in the maximum and minimum ratios of tubesheet thickness to tube diameter.
(22)	Describe how the tube will be fixed in position before expanding, e.g., "nose roll" or "hydraulically preset."
(23)	Fill in the length of tube end to be expanded into the hole, e.g., "tubesheet thickness - $^3/_{16}$ in." If hybrid expansion is to be performed, show length of expansion for each step.
(24)	Fill in the distance from the front face of the tubesheet to the point where expanding will begin.
(25)	Fill in the distance from the rear face of the tubesheet to the point where expanding will end.
(26)	If tube is welded to front face of tubesheet, describe how any weld metal that impedes access of the expanding tool(s) will be removed.
(27)	Describe how tube ends will be cleaned before expanding, e.g., "solvent wash and clean with felt plugs."
(28)	Describe any other pertinent details, e.g., "tubes to be welded to front face of tubesheet before expanding."
(29)	Show expanding tool manufacturer, e.g., name hydraulic expanding system or model no., "range of tube diameters $^1/_2$ in. to 2 in., range of thicknesses 0.028 in. to 0.109 in., maximum hydraulic pressure 60,000 psi."
(30)	Fill in roller expanding tool or hydraulic mandrel number. If explosive expanding, fill in drawing number that describes the charges. If hybrid expanding, show this information for Steps 1 and 2.
(31)	Describe expanded length per application, e.g., "2 in. (roller length)."
(32)	Show number of applications of expanding tool, e.g., "two applications required for roll depth." If hydraulic or explosive expanding, show length of expansion per application of hydraulic expanding pressure or explosive charge, e.g., "tubesheet thickness - $^5/_8$ in."
(33)	Describe the system used to calibrate and control the rolling torque and frequency of verification. Alternatively, describe the use of production control holes and expansions.
(34)	Describe the explosive charge and whether it will be single- or two-stage explosive expansion.
(35)	List the minimum and maximum elastic modulus of the tubes for this TEPS.
(36)	List the minimum and maximum elastic modulus of the tubesheet(s) for this TEPS.
(37)	List minimum permissible tube yield stress.

Table TEXP-1
Instructions for Filling Out TEPS Form (Cont'd)

Ref. to Circled Nos. in Form TEXP-1	Explanation of Information to Be Provided
(38)	List maximum permissible tube yield stress.
(39)	List minimum permissible tubesheet yield stress.
(40)	List maximum permissible tubesheet yield stress.
(41)	Show the minimum ratio of tubesheet to tube yield stresses.
(42)	List the range of tube diameters to which this TEPS applies.
(43)	List the range of tube thicknesses to which this TEPS applies.
(44)	Show the maximum ratio of tube diameter to thickness to which this TEPS applies.
(45)	Show the tube specification number, e.g., “SA-688 TP304N.”
(46)	Show the range of tubesheet thicknesses to which this TEPS applies, e.g., 1 in. to 5 in.
(47)	Show the minimum ratio of tubesheet thickness to tube diameter to which this TEPS applies.
(48)	Show the tubesheet material specification numbers, e.g., “SA-350 LF2.”
(49)	Describe pertinent job-specific information.
(50)	Describe such things as bundle setup and sequence of expansion operation. Refer to drawing numbers and manufacturer’s standards as appropriate.
(51)	Refer to any attachment or supplement to the TEPS form.

FORM TEXP-2 SUGGESTED FORMAT FOR TUBE-TO-TUBESHEET EXPANDING PROCEDURE QUALIFICATION RECORD FOR TEST QUALIFICATION (TEPQR)

Company name ______________________

Procedure Qualification Record number ______________ Date ______________

TEPS no. ______________________

Expanding process(es) ______________
(Rolling, hydroexpanding, explosive expanding, hybrid expanding)

Driver types ______________
(Electric, air-driven, hydraulic, other)

Expanded tube length ______________
(If there is a gap in the expanded zone, record the total expanded length)

Tube pitch ______________

Joints (Annex 4-E, 4-E.7)

Sketch of Test Array

Tubesheet Material(s)

Material spec. ______________ Type or grade ______________

Diameter and thickness of test specimen ______________ Hole diameter and pitch arrangement ______________

No. and location of joints to be tested ______________

No. and description of annular grooves ______________

Hole surface finish ______________

Yield stress (from mill test report) ______________

Other ______________

Testing Apparatus ______________
(Manufacturer, type, calibration date)

Rate of loading to avoid impact ______________
[Maximum $^1/_2$ in. (13 mm) per minute]

Tube Material(s) ______________

Material spec. ______________ Type or grade ______________

Diameter and thickness (min./avg.) ______________

Yield stress (from mill test report) ______________

Other ______________

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FORM TEXP-2 (Back)

Shear Load Test (See Annex 4-C, Figure 4-C.2)

Tube No.	Position in Array	Diameter	Thickness	Cross-Sectional Area	Test Temp.	L_1 (test)	Manner of Failure

Tube No.	Position in Array	Diameter	Thickness	Cross-Sectional Area	Ambient Temp.	L_2 (test)	Manner of Failure

Mean value of L_1 (test) ____________ Mean value of L_2 (test) ____________

Standard deviation ____________ Standard deviation ____________ Satisfactory (see Annex 4-C, 4-C.5) ____________

f_r (test) (see Annex 4-C, 4-C.4) ____________

Operator's name ____________ Clock no. ____________

Manufacturer ____________

Date ____________ By ____________

Remarks:

07/15

PART 5
DESIGN BY ANALYSIS REQUIREMENTS

5.1 GENERAL REQUIREMENTS

5.1.1 SCOPE

5.1.1.1 The design requirements for application of the design-by-analysis methodology of this Division are described in Part 5. Detailed design procedures utilizing the results from a stress analysis are provided to evaluate components for plastic collapse, local failure, buckling, and cyclic loading. Supplemental requirements are provided for the analysis of bolts, perforated plates and layered vessels. Procedures are also provided for design using the results from an experimental stress analysis, and for fracture mechanics evaluations.

5.1.1.2 The design-by-analysis requirements are organized based on protection against the failure modes listed below. The component shall be evaluated for each applicable failure mode. If multiple assessment procedures are provided for a failure mode, only one of these procedures must be satisfied to qualify the design of a component.

(a) Protection Against Plastic Collapse – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules.

(b) Protection Against Local Failure – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules. It is not necessary to evaluate the protection against local failure, 5.3, if the component design is in accordance with Part 4 (e.g., component wall thickness and weld detail per 4.2).

(c) Protection Against Collapse From Buckling – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules and the applied loads result in a compressive stress field.

(d) Protection Against Failure From Cyclic Loading – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules and the applied loads are cyclic. In addition, these requirements can also be used to qualify a component for cyclic loading where the thickness and size of the component are established using the design-by-rule requirements of Part 4.

5.1.1.3 The design-by-analysis procedures in Part 5 may only be used if the allowable stress from Annex 3-A evaluated at the design temperature is governed by time-independent properties unless otherwise noted in a specific design procedure. If the allowable stress from Annex 3-A evaluated at the design temperature is governed by time-dependent properties and the fatigue screening criteria of 5.5.2.2 are satisfied, the elastic stress analysis procedures in 5.2.2, 5.3.2, 5.6, 5.7.1, 5.7.2, and 5.8 may be used.

5.1.1.3.1 Class 1. The design-by-analysis procedures in Part 5 shall not be used unless allowed by 4.1.1.2.1.

5.1.2 NUMERICAL ANALYSIS

5.1.2.1 The design-by-analysis rules in Part 5 are based on the use of results obtained from a detailed stress analysis of a component. Depending on the loading condition, a thermal analysis to determine the temperature distribution and resulting thermal stresses may also be required.

5.1.2.2 Procedures are provided for performing stress analyses to determine protection against plastic collapse, local failure, buckling, and cyclic loading. These procedures provide the necessary details to obtain a consistent result with regards to development of loading conditions, selection of material properties, post-processing of results, and comparison to acceptance criteria to determine the suitability of a component.

5.1.2.3 Recommendations on a stress analysis method, modeling of a component, and validation of analysis results are not provided. While these aspects of the design process are important and shall be considered in the analysis, a detailed treatment of the subject is not provided because of the variability in approaches and design processes. However, an accurate stress analysis including validation of all results shall be provided as part of the design.

5.1.2.4 The following material properties for use in the stress analysis shall be determined using the data and material models in Part 3.

(a) Physical properties – Young's Modulus, thermal expansion coefficient, thermal conductivity, thermal diffusivity, density, Poisson's ratio

(b) Strength Parameters – Allowable stress, minimum specified yield strength, minimum specified tensile strength

(c) Monotonic stress–strain Curve – elastic perfectly plastic and elastic–plastic true stress–strain curve with strain hardening

(d) Cyclic stress–strain Curve – Stabilized true stress–strain amplitude curve

5.1.3 LOADING CONDITIONS

5.1.3.1 All applicable applied loads on the component shall be considered when performing a design-by-analysis. Supplemental loads shall be considered in addition to the applied pressure in the form of applicable load cases. If the load case varies with time, a loading histogram shall be developed to show the time variation of each specific load. The load case definition shall be included in the User's Design Specification. An overview of the supplemental loads and loading conditions that shall be considered in a design are shown in Table 5.1.

(**21**) **5.1.3.2** Load case combinations shall be considered in the analysis. Typical load descriptions are provided in Table 5.2. Load case combinations for elastic analysis, limit-load analysis, and elastic–plastic analysis are shown in Tables 5.3, 5.4, and 5.5, respectively. In evaluating load cases involving the pressure term, P, the effects of the pressure being equal to zero shall be considered. The applicable load case combinations shall be considered in addition to any other combinations defined in the User's Design Specification. The factors for wind loading, W, in Table 5.3, Design Load Combinations, and in Tables 5.4 and 5.5, Required Factored Load Combinations, are based on ASCE/SEI 7 wind maps and probability of occurrence. If a different recognized standard for wind loading is used, the User's Design Specification shall cite the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7. If a different recognized standard for earthquake loading is used, the User's Design Specification shall cite the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7.

5.1.3.3 If any of the loads vary with time, a loading histogram shall be developed to show the time variation of each specific load. The loading histogram shall include all significant operating temperatures, pressures, supplemental loads, and the corresponding cycles or time periods for all significant events that are applied to the component. The following shall be considered in developing the loading histogram.

(a) The number of cycles associated with each event during the operation life, these events shall include start-ups, normal operation, upset conditions, and shutdowns.

(b) When creating the histogram, the history to be used in the assessment shall be based on the anticipated sequence of operation. When it is not possible or practical to develop a histogram based on the actual sequence of operation, a histogram may be used that bounds the actual operation. Otherwise, the cyclic evaluation shall account for all possible combinations of loadings.

(c) Applicable loadings such as pressure, temperature, supplemental loads such as weight, support displacements, and nozzle reaction loadings.

(d) The relationship between the applied loadings during the time history.

5.2 PROTECTION AGAINST PLASTIC COLLAPSE

5.2.1 OVERVIEW

5.2.1.1 Three alternative analysis methods are provided for evaluating protection against plastic collapse. A brief description of these analysis methodologies is provided below.

(a) Elastic Stress Analysis Method – Stresses are computed using an elastic analysis, classified into categories, and limited to allowable values that have been conservatively established such that a plastic collapse will not occur.

(b) Limit-Load Method – A calculation is performed to determine a lower bound to the limit load of a component. The allowable load on the component is established by applying design factors to the limit load such that the onset of gross plastic deformations (plastic collapse) will not occur.

(c) Elastic–Plastic Stress Analysis Method – A collapse load is derived from an elastic–plastic analysis considering both the applied loading and deformation characteristics of the component. The allowable load on the component is established by applying design factors to the plastic collapse load.

5.2.1.2 For components with a complex geometry and/or complex loading, the categorization of stresses requires significant knowledge and judgment. This is especially true for three-dimensional stress fields. Application of the limit-load or elastic–plastic analysis methods in 5.2.3 and 5.2.4, respectively, is recommended for cases where the categorization process may produce ambiguous results.

5.2.1.3 The use of elastic stress analysis combined with stress classification procedures to demonstrate structural integrity for heavy-wall ($R/t \le 4$) pressure-containing components, especially around structural discontinuities, may produce non-conservative results and is not recommended. The reason for the non-conservatism is that the nonlinear stress distributions associated with heavy wall sections are not accurately represented by the implicit linear stress distribution utilized in the stress categorization and classification procedure. The misrepresentation of the stress distribution is enhanced if yielding occurs. For example, in cases where calculated peak stresses are above yield over a through thickness dimension which is more than five percent of the wall thickness, linear elastic analysis may give a non-conservative result. In these cases, the elastic–plastic stress analysis procedures in 5.2.3 or 5.2.4 shall be used.

5.2.1.4 The structural evaluation procedures based on elastic stress analysis in 5.2.2 provide an approximation of the protection against plastic collapse. A more accurate estimate of the protection against plastic collapse of a component can be obtained using elastic–plastic stress analysis to develop limit and plastic collapse loads. The limits on the general membrane equivalent stress, local membrane equivalent stress and primary membrane plus primary bending equivalent stress in 5.2.2 have been placed at a level which conservatively assures the prevention of collapse as determined by the principles of limit analysis. These limits need not be satisfied if the requirements of 5.2.3 or 5.2.4 are satisfied.

5.2.2 ELASTIC STRESS ANALYSIS METHOD

5.2.2.1 Overview. To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value. The basis of the categorization procedure is described below.

(a) A quantity known as the equivalent stress is computed at locations in the component and compared to an allowable value of equivalent stress to determine if the component is suitable for the intended design conditions. The equivalent stress at a point in a component is a measure of stress, calculated from stress components utilizing a yield criterion, which is used for comparison with the mechanical strength properties of the material obtained in tests under uniaxial load.

(b) The maximum distortion energy yield criterion shall be used to establish the equivalent stress. In this case, the equivalent stress is equal to the von Mises equivalent stress given by eq. (5.1)

$$S_e = \sigma_e = \frac{1}{\sqrt{2}}\left[\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2\right]^{0.5} \tag{5.1}$$

5.2.2.2 Stress Categorization. The three basic equivalent stress categories and associated limits that are to be satisfied for plastic collapse are defined below. The terms general primary membrane stress, local primary membrane stress, primary bending stress, secondary stress, and peak stress used for elastic analysis are defined in the following paragraphs. The design loads to be evaluated and the allowable stress limits are provided in Table 5.3.

(a) General Primary Membrane Equivalent Stress (P_m)

(1) The general primary membrane equivalent stress (see Figure 5.1) is the equivalent stress, derived from the average value across the thickness of a section, of the general primary stresses produced by the design internal pressure and other specified mechanical loads but excluding all secondary and peak stresses.

(2) Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

(b) Local Primary Membrane Equivalent Stress (P_L)

(1) The local primary membrane equivalent stress (see Figure 5.1) is the equivalent stress, derived from the average value across the thickness of a section, of the local primary stresses produced by the design pressure and specified mechanical loads but excluding all secondary and peak stresses. A region of stress in a component is considered as local if the distance over which the equivalent stress exceeds $1.1S$ does not extend in the meridional direction more than $\sqrt{Rt}$.

(2) Regions of local primary membrane stress that exceed $1.1S$ shall be separated in the meridional direction by a distance greater than or equal to $1.25\sqrt{(R_1 + R_2)(t_1 + t_2)}$. Discrete regions of local primary membrane stress, such as those resulting from concentrated loads on support brackets, where the membrane stress exceeds $1.1S$, shall be spaced so that there is not an overlapping area in which the membrane stress exceeds $1.1S$.

(3) Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

(c) Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress ($P_L + P_b$)

(1) The Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress (see Figure 5.1) is the equivalent stress, derived from the highest value across the thickness of a section, of the linearized general or local primary membrane stresses plus primary bending stresses produced by design pressure and other specified mechanical loads but excluding all secondary and peak stresses.

(2) Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

5.2.2.3 Linearization of Stress Results for Stress Classification. Results from an elastic stress analysis can be used to compute the equivalent linearized membrane and bending stresses for comparison to the limits in 5.2.2.4 using the methods described in Annex 5-A.

5.2.2.4 Assessment Procedure. To determine the acceptability of a component, the computed equivalent stresses given in 5.2.2.2 for a component subject to loads shall not exceed the specified allowable values. A schematic illustrating the categorization of equivalent stresses and their corresponding allowable values is shown in Figure 5.1. The following procedure is used to compute and categorize the equivalent stress at a point in a component (see 5.2.2.3), and to determine the acceptability of the resulting stress state.

Step 1. Determine the types of loads acting on the component. In general, separate load cases are analyzed to evaluate "load-controlled" loads such as pressure and externally applied reactions due to weight effects and "strain-controlled" loads resulting from thermal gradients and imposed displacements. The loads to be considered in the design shall include, but not be limited to, those given in Table 5.1. The load combinations that shall be considered for each loading condition shall include, but not be limited to those given in Table 5.3.

Step 2. At the point on the vessel that is being investigated, calculate the stress tensor (six unique components of stress) for each type of load. Assign each of the computed stress tensors to one or to a group of the categories defined below. Assistance in assigning each stress tensor to an appropriate category for a component can be obtained by using Figure 5.1 and Table 5.6. Note that the equivalent stresses Q and F do not need to be determined to evaluate protection against plastic collapse. However, these components are needed for fatigue and ratcheting evaluations that are based on elastic stress analysis (see 5.5.3 and 5.5.6, respectively).

(a) General primary membrane equivalent stress – P_m

(b) Local primary membrane equivalent stress – P_L

(c) Primary bending equivalent stress – P_b

(d) Secondary equivalent stress – Q

(e) Additional equivalent stress produced by a stress concentration or a thermal stress over and above the nominal ($P + Q$) stress level – F

Step 3. Sum the stress tensors (stresses are added on a component basis) assigned to each equivalent stress category. The final result is a stress tensor representing the effects of all the loads assigned to each equivalent stress category. Note that in applying STEPs in this paragraph, a detailed stress analysis performed using a numerical method such as finite element analysis typically provides a combination of $P_L + P_b$ and $P_L + P_b + Q + F$ directly.

(a) If a load case is analyzed that includes only "load-controlled" loads (e.g., pressure and weight effects), the computed equivalent stresses shall be used to directly represent the P_m, $P_L + P_b$, or $P_L + P_b + Q$. For example, for a vessel subject to internal pressure with an elliptical head; P_m equivalent stresses occur away from the head to shell junction, and P_L and $P_L + P_b + Q$ equivalent stresses occur at the junction.

(b) If a load case is analyzed that includes only "strain-controlled" loads (e.g., thermal gradients), the computed equivalent stresses represent Q alone; the combination $P_L + P_b + Q$ shall be derived from load cases developed from both "load-controlled" and "strain-controlled" loads.

(c) If the stress in category F is produced by a stress concentration or thermal stress, the quantity F is the additional stress produced by the stress concentration in excess of the nominal membrane plus bending stress. For example, if a plate has a nominal primary membrane equivalent stress of S_e, and has a fatigue strength reduction characterized by a factor K_f, then: $P_m = S_e$, $P_b = 0$, $Q = 0$, and $F = P_m(K_f - 1)$. The total equivalent stress is $P_m + F$.

Step 4. Determine the principal stresses of the sum of the stress tensors assigned to the equivalent stress categories, and compute the equivalent stress using Eq. (5.1).

Step 5. To evaluate protection against plastic collapse, compare the computed equivalent stress to their corresponding allowable values (see 5.2.2.2).

The allowable limit on local primary membrane and local primary membrane plus bending, S_{PL}, is computed as the larger of the quantities shown below.

(a) 1.5 times the tabulated allowable stress for the material from Annex 3-A

(b) S_y for the material from Annex 3-A, except that the value from (a) shall be used when the ratio of the minimum specified yield strength to ultimate tensile strength exceeds 0.70, or the value of S is governed by time-dependent properties as indicated in Annex 3-A

$$P_m \le S \tag{5.2}$$

$$P_L \le S_{PL} \tag{5.3}$$

$$(P_L + P_b) \le S_{PL} \tag{5.4}$$

5.2.2.5 Test Condition for Components Designed Using Elastic Stress Analysis Method. The test condition for components designed using the elastic stress analysis method, for a selected test pressure, P_T, greater than or equal to the minimum test pressure, shall be evaluated using Design Load Combination (18) from Table 5.3 and the limits listed in 4.1.6.2. It is not required to evaluate any stress categories not listed in 4.1.6.2 (e.g., primary local, secondary, or peak) in the test condition. In the case where the limits in 4.1.6.2 are not achieved, the selected test pressure shall be reduced, but no lower than the minimum test pressure. The requirements of 5.4 shall be satisfied for the test condition at the selected test pressure, P_T.

5.2.3 LIMIT-LOAD ANALYSIS METHOD

5.2.3.1 Overview.

(a) Limit-load analysis addresses the failure modes of ductile rupture and the onset of gross plastic deformation (plastic collapse) of a structure. As defined in the following paragraphs, it provides one option to protect a vessel or component from plastic collapse. It is to be applied to single or multiple static loading, applied in any specified order. Limit-load analysis provides an alternative to elastic analysis and stress linearization and the satisfaction of primary stress limits in 5.2.2.2.

(b) Displacements and strains indicated by a limit analysis solution have no physical meaning. If the User's Design Specification requires a limit on such variables, the procedures in 5.2.4 shall be used to satisfy these requirements.

(c) Protection against plastic collapse using limit-load analysis is based on the theory of limit analysis that defines a lower bound to the limit load of a structure as the solution of a numerical model with the following properties:

(1) The material model is elastic-perfectly plastic with a specified yield strength.

(2) The strain-displacement relations are those of small displacement theory.

(3) Equilibrium is satisfied in the undeformed configuration.

5.2.3.2 Limitations. The following limitations apply equally to limit-load analysis and to primary stress limits of 5.2.2.

(a) The effect of strain-controlled loads resulting from prescribed non-zero displacements and temperature fields is not considered.

(b) Components that experience reduction in stiffness with deformation, e.g., a pipe elbow under in-plane bending, shall be evaluated using 5.2.4.

5.2.3.3 Numerical Analysis. The limit load is the load that causes overall structural instability. In practice, an estimate of the limit load is obtained using a numerical analysis technique (e.g., finite element method) by incorporating an elastic-perfectly-plastic material model and small displacement theory to obtain a solution. The estimated limit load is the maximum load before overall structural instability occurs. Structural instability is indicated by the inability to achieve an equilibrium solution for a small increase in load (i.e., the solution will not converge).

5.2.3.4 Acceptance Criteria. The acceptability of a component using a limit-load analysis is determined by satisfying the following two criteria.

(a) Global Criteria – The estimated global plastic collapse load is established by performing a limit-load analysis of the component subject to the specified loading conditions. The estimated plastic collapse load is taken as the maximum load before overall structural instability occurs. The concept of Load and Resistance Factor Design (LRFD) is used as an alternative to the rigorous computation of a plastic collapse load to design a component. In this procedure, factored loads that include a design factor to account for uncertainty, and the resistance of the component to these factored loads is determined using a limit-load analysis (see Table 5.4).

(b) Service Criteria – Service criteria as provided by the Owner/User that limit the potential for unsatisfactory performance shall be satisfied at every location in the component when subject to the design loads. The service criteria shall satisfy the requirements of 5.2.4.3(b) using the procedures in 5.2.4.

5.2.3.5 Assessment Procedure. The following assessment procedure is used to determine the acceptability of a component using a limit-load analysis.

Step 1. Develop a numerical model of the component including all relevant geometry characteristics. The model used for the analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads. The model need not be accurate for small details, such as small holes, fillets, corner radii, and other stress raisers, but should otherwise correspond to commonly accepted practice.

Step 2. Define all relevant loads and applicable load cases. The loads to be considered in the analysis shall include, but not be limited to, those given in Table 5.1.

Step 3. An elastic perfectly plastic material model with small displacement theory shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized. The yield strength defining the plastic limit shall equal $1.5S$.

Step 4. Determine the load case combinations to be used in the analysis using the information from Step 2 in conjunction with Table 5.4. Each of the indicated load cases shall be evaluated. The effects of one or more loads not acting shall be investigated. Additional load cases for special conditions not included in Table 5.4 shall be considered, as applicable.

Step 5. Perform a limit-load analysis for each of the load case combinations defined in Step 4. If convergence is achieved, the component is stable under the applied loads for this load case. Otherwise, the component configuration (i.e., thickness) shall be modified or applied loads reduced and the analysis repeated. Note that if the applied loading results in a compressive stress field within the component, buckling may occur, and the effects of imperfections, especially for shell structures, should be considered in the analysis (see 5.4).

5.2.3.6 Test Condition for Components Designed Using Limit-Load Analysis Method. The test condition for components designed using the limit-load analysis method, for a selected test pressure, P_T, greater than or equal to the minimum test pressure, shall be evaluated by following the Test Condition, Required Factored Load Combination given in Table 5.4, where β_T shall be obtained from Table 4.1.3 for the appropriate test medium (hydrostatic or pneumatic) and the applicable class. In the case where the analysis does not converge (see 5.2.3.5, Step 5), the selected test pressure shall be reduced, but no lower than the minimum test pressure. The requirements of 5.4 shall be satisfied for the test condition at the selected test pressure, P_T.

5.2.4 ELASTIC–PLASTIC STRESS ANALYSIS METHOD

5.2.4.1 Overview.

(a) Protection against plastic collapse is evaluated by determining the plastic collapse load of the component using an elastic–plastic stress analysis. The allowable load on the component is established by applying a design factor to the calculated plastic collapse load.

(b) Elastic–plastic stress analysis provides a more accurate assessment of the protection against plastic collapse of a component relative to the criteria in 5.2.2 and 5.2.3 because the actual structural behavior is more closely approximated. The redistribution of stress that occurs as a result of inelastic deformation (plasticity) and deformation characteristics of the component are considered directly in the analysis.

5.2.4.2 Numerical Analysis. The plastic collapse load is the load that causes overall structural instability. In practice, an estimate of the plastic collapse load can be obtained using a numerical analysis technique (e.g., finite element method) by incorporating an elastic–plastic material model to obtain a solution. The effects of non-linear geometry shall be considered in this analysis. The estimated plastic collapse load is the maximum load before overall structural instability occurs. Structural instability is indicated by the inability to achieve an equilibrium solution for a small increase in load (i.e., the solution will not converge).

5.2.4.3 Acceptance Criteria. The acceptability of a component using an elastic–plastic analysis is determined by satisfying the following two criteria.

(a) Global Criteria – The estimated global plastic collapse load is established by performing an elastic–plastic analysis of the component subject to the specified loading conditions. The estimated plastic collapse load is taken as the maximum load before overall structural instability occurs. The concept of Load and Resistance Factor Design (LRFD) is used as an alternate to the rigorous computation of a plastic collapse load to design a component. In this procedure, factored loads that include a design factor to account for uncertainty, and the resistance of the component to these factored loads are determined using an elastic–plastic analysis (see Table 5.5).

(b) Service Criteria – Service criteria that limit the potential for unsatisfactory performance shall be satisfied at every location in the component when subject to the design loads (see Table 5.5). Examples of service criteria are limits on the rotation of a mating flange pair to avoid possible flange leakage concerns and limits on tower deflection that may cause operational concerns. In addition, the effect of deformation of the component on service performance shall be evaluated at the design load combinations. This is especially important for components that experience an increase in resistance (geometrically stiffen) with deformation under applied loads such as elliptical or torispherical heads subject to internal

pressure loading. The plastic collapse criteria may be satisfied but the component may have excessive deformation at the derived design conditions. In this case, the design loads may have to be reduced based on a deformation criterion. Examples of some of the considerations in this evaluation are the effect of deformation on:

(1) piping connections or,

(2) misalignment of trays, platforms and other internal or external appurtenances, and

(3) interference with adjacent structures and equipment.

If applicable, the service criteria shall be specified in the User's Design Specification.

5.2.4.4 Assessment Procedure. The following assessment procedure is used to determine the acceptability of a component using an elastic–plastic stress analysis.

Step 1. Develop a numerical model of the component including all relevant geometry characteristics. The model used for the analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads. In addition, refinement of the model around areas of stress and strain concentrations shall be provided. The analysis of one or more numerical models may be required to ensure that an accurate description of the stress and strains in the component is achieved.

Step 2. Define all relevant loads and applicable load cases. The loads to be considered in the design shall include, but not be limited to, those given in Table 5.1.

Step 3. An elastic plastic material model shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized if plasticity is anticipated. A material model that includes hardening or softening, or an elastic-perfectly plastic model may be utilized. A true stress–strain curve model that includes temperature dependent hardening behavior is provided in Annex 3-D. When using this material model, the hardening behavior shall be included up to the true ultimate stress and perfect plasticity behavior (i.e., the slope of the stress–strain curves is zero) beyond this limit. The effects of non-linear geometry shall be considered in the analysis.

Step 4. Determine the load case combinations to be used in the analysis using the information from Step 2 in conjunction with Table 5.5. Each of the indicated load cases shall be evaluated. The effects of one or more loads not acting shall be investigated. Additional load cases for special conditions not included in Table 5.5 shall be considered, as applicable.

Step 5. Perform an elastic–plastic analysis for each of the load cases defined in Step 4. If convergence is achieved, the component is stable under the applied loads for this load case. Otherwise, the component configuration (i.e., thickness) shall be modified or applied loads reduced and the analysis repeated. Note that if the applied loading results in a compressive stress field within the component, buckling may occur, and an evaluation in accordance with 5.4 may be required.

5.2.4.5 Test Condition for Components Designed Using Elastic–Plastic Stress Analysis Method. The test condition for components designed using the elastic–plastic stress analysis method, for a selected test pressure, P_T, greater than or equal to the minimum test pressure, shall be evaluated by following the Test Condition, Required Factored Load Combination given in Table 5.5, where β_T shall be obtained from Table 4.1.3 for the appropriate test medium (hydrostatic or pneumatic) and the applicable class. In the case where the analysis does not converge (see 5.2.4.4, Step 5), the selected test pressure shall be reduced, but no lower than the minimum test pressure. The requirements of 5.4 shall be satisfied for the test condition.

5.3 PROTECTION AGAINST LOCAL FAILURE

5.3.1 OVERVIEW

5.3.1.1 In addition to demonstrating protection against plastic collapse as defined in 5.2, the applicable local failure criteria below shall be satisfied for a component. These requirements apply to all components where the thickness and configuration of the component are established by using design-by-analysis rules. It is not necessary to evaluate protection against local failure (5.3), if the component design is in accordance with Part 4 (e.g., component wall thickness and weld detail per 4.2).

5.3.1.2 Two analysis methodologies are provided for evaluating protection against local failure under applied design loads. When protection against plastic collapse is satisfied by the method in 5.2.3, either method listed below is acceptable.

(a) The analysis procedures in 5.3.2 provide an approximation of the protection against local failure based on the results of an elastic analysis.

(b) A more accurate estimate of the protection against local failure of a component can be obtained using the elastic–plastic stress analysis procedures in 5.3.3.

5.3.2 ELASTIC ANALYSIS — TRIAXIAL STRESS LIMIT

The algebraic sum of the three linearized primary principal stresses from Design Load Combination (1) of Table 5.3 shall be used for checking this criterion.

$$(\sigma_1 + \sigma_2 + \sigma_3) \le 4S \tag{5.5}$$

5.3.3 ELASTIC–PLASTIC ANALYSIS — LOCAL STRAIN LIMIT

5.3.3.1 The following procedure shall be used to evaluate protection against local failure for a sequence of applied loads.

Step 1. Perform an elastic–plastic stress analysis based on the load case combinations for the local criteria given in Table 5.5. The effects of non-linear geometry shall be considered in the analysis.

Step 2. For each point in the component, determine the principal stresses, σ_1, σ_2, σ_2, the equivalent stress, σ_e, using Eq. (5.1) and the equivalent plastic strain, ε_{peq}.

Step 3. Determine the limiting triaxial strain, ε_L, using eq. (5.6) where ε_{Lu}, m_2, and α_{sl} are determined from Table 5.7.

$$\varepsilon_L = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right)\left(\left\{\frac{(\sigma_1+\sigma_2+\sigma_3)}{3\sigma_e}\right\} - \frac{1}{3}\right)\right] \tag{5.6}$$

Step 4. Determine the forming strain ε_{cf} based on the material and fabrication method in accordance with Part 6. If heat treatment is performed in accordance with Part 6, the forming strain may be assumed to be zero.

Step 5. Determine if the strain limit is satisfied. The component is acceptable for the specified load case if eq. (5.7) is satisfied for each point.

$$\varepsilon_{peq} + \varepsilon_{cf} \le \varepsilon_L \tag{5.7}$$

5.3.3.2 If a specific loading sequence is to be evaluated in accordance with the User's Design Specification, a strain limit damage calculation procedure may be required. This procedure may also be used in lieu of the procedure in 5.3.3.1. In this procedure, the loading path is divided into k load increments and the principal stresses, $\sigma_{1,k}$, $\sigma_{2,k}$, σ_{3k}, equivalent stress, $\Delta_{e,k}$, and change in the equivalent plastic strain from the previous load increment, $\Delta\varepsilon_{peq,k}$, are calculated for each load increment. The strain limit for the Δkth load increment, $\varepsilon_{L,k}$, is calculated using eq. (5.8) where ε_{Lu}, m_2, and α_{sl} are determined from Table 5.7. The strain limit damage for each load increment is calculated using eq. (5.9) and the strain limit damage from forming, $D_{\varepsilon\ form}$, is calculated using eq. (5.10). If heat treatment is performed in accordance with Part 6, the strain limit damage from forming is assumed to be zero. The accumulated strain limit damage is calculated using eq. (5.11). The location in the component is acceptable for the specified loading sequence if this equation is satisfied.

$$\varepsilon_{L,k} = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right)\left(\left\{\frac{\sigma_{1,k}+\sigma_{2,k}+\sigma_{3,k}}{3\sigma_{e,k}}\right\} - \frac{1}{3}\right)\right] \tag{5.8}$$

$$D_{\varepsilon,k} = \frac{\Delta\varepsilon_{peq,k}}{\varepsilon_{L,k}} \tag{5.9}$$

$$D_{\varepsilon form} = \frac{\varepsilon_{cf}}{\varepsilon_{Lu} \cdot \exp\left[-\frac{1}{3}\left(\frac{\alpha_{sl}}{1+m_2}\right)\right]} \tag{5.10}$$

$$D_{\varepsilon} = D_{\varepsilon form} + \sum_{k=1}^{M} D_{\varepsilon,k} \le 1.0 \tag{5.11}$$

5.4 PROTECTION AGAINST COLLAPSE FROM BUCKLING

5.4.1 DESIGN FACTORS

5.4.1.1 In addition to evaluating protection against plastic collapse as defined in 5.2, a design factor for protection against collapse from buckling shall be satisfied to avoid buckling of components with a compressive stress field under applied design loads.

5.4.1.2 The design factor to be used in a structural stability assessment is based on the type of buckling analysis performed. The following design factors shall be the minimum values for use with shell components when the buckling loads are determined using a numerical solution (i.e., bifurcation buckling analysis or elastic–plastic collapse analysis).

(a) Type 1 – If a bifurcation buckling analysis is performed using an elastic stress analysis without geometric nonlinearities in the solution to determine the pre-stress in the component, a minimum design factor of $\Phi_B = 2/B_{cr}$ shall be used (see 5.4.1.3). In this analysis, the pre-stress in the component is established based on Design Load Combinations (1) through (9) in Table 5.3.

(b) Type 2 – If a bifurcation buckling analysis is performed using an elastic–plastic stress analysis with the effects of non-linear geometry in the solution to determine the pre-stress in the component, a minimum design factor of $\Phi_B = 1.667/\beta_{cr}$ shall be used (see 5.4.1.3). In this analysis, the pre-stress in the component is established based on Design Load Combinations (1) through (9) in Table 5.3.

(c) Type 3 – If a collapse analysis is performed in accordance with 5.2.4, and imperfections are explicitly considered in the analysis model geometry, the design factor is accounted for in the factored load combinations in Table 5.5. It should be noted that a collapse analysis can be performed using elastic or plastic material behavior. If the structure remains elastic when subject to the applied loads, the elastic–plastic material model will provide the required elastic behavior, and the collapse load will be computed based on this behavior.

5.4.1.3 The capacity reduction factors, β_{cr}, shown below shall be used unless alternative factors can be developed from published information.

(a) For unstiffened or ring stiffened cylinders and cones under axial compression

$$\beta_{cr} = 0.207 \qquad \text{for} \quad \frac{D_o}{t} \geq 1\,247 \tag{5.12}$$

$$\beta_{cr} = \frac{338}{389 + \frac{D_o}{t}} \qquad \text{for} \quad \frac{D_o}{t} < 1\,247 \tag{5.13}$$

(b) For unstiffened and ring stiffened cylinders and cones under external pressure

$$\beta_{cr} = 0.80 \tag{5.14}$$

(c) For spherical shells and spherical, torispherical, elliptical heads under external pressure

$$\beta_{cr} = 0.124 \tag{5.15}$$

5.4.2 Numerical Analysis. If a numerical analysis is performed to determine the buckling load for a component, all possible buckling mode shapes shall be considered in determining the minimum buckling load for the component. Care should be taken to ensure that simplification of the model does not result in exclusion of a critical buckling mode shape. For example, when determining the minimum buckling load for a ring-stiffened cylindrical shell, both axisymmetric and non-axisymmetric buckling modes shall be considered in determination of the minimum buckling load.

5.5 PROTECTION AGAINST FAILURE FROM CYCLIC LOADING

5.5.1 OVERVIEW

5.5.1.1 A fatigue evaluation shall be performed if the component is subject to cyclic operation. The evaluation for fatigue is made on the basis of the number of applied cycles of a stress or strain range at a point in the component. The allowable number of cycles should be adequate for the specified number of cycles as given in the User's Design Specification.

5.5.1.2 Screening criteria are provided in 5.5.2 that can be used to determine if fatigue analysis is required as part of a design. If the component does not satisfy the screening criteria, a fatigue evaluation shall be performed using the techniques in 5.5.3, 5.5.4 or 5.5.5.

5.5.1.3 Fatigue curves are typically presented in two forms: fatigue curves that are based on smooth bar test specimens and fatigue curves that are based on test specimens that include weld details of quality consistent with the fabrication and inspection requirements of this Division.

(a) Smooth bar fatigue curves may be used for components with or without welds. The welded joint curves shall only be used for welded joints.

(b) The smooth bar fatigue curves are applicable up to the maximum number of cycles given on the curves. The welded joint fatigue curves do not exhibit an endurance limit and are acceptable for all cycles.

(c) If welded joint fatigue curves are used in the evaluation, and if thermal transients result in a through-thickness stress difference at any time that is greater than the steady-state difference, the number of design cycles shall be determined as the smaller of the number of cycles for the base metal established using either 5.5.3 or 5.5.4, and for the weld established in accordance with 5.5.5.

5.5.1.4 Stresses and strains produced by any load or thermal condition that does not vary during the cycle need not be considered in a fatigue analysis if the fatigue curves utilized in the evaluation are adjusted for mean stresses and strains. The design fatigue curves referenced in 5.5.3 and 5.5.4 are based on smooth bar test specimens and are adjusted for the maximum possible effect of mean stress and strain; therefore, an adjustment for mean stress effects is not required. The fatigue curves referenced in 5.5.5 are based on welded test specimens and include explicit adjustments for thickness and mean stress effects.

5.5.1.5 Under certain combinations of steady-state and cyclic loadings there is a possibility of ratcheting. A rigorous evaluation of ratcheting normally requires an elastic–plastic analysis of the component; however, under a limited number of loading conditions, an approximate analysis can be utilized based on the results of an elastic stress analysis, see 5.5.6.

5.5.1.6 Protection against ratcheting shall be considered for all operating loads listed in the User's Design Specification and shall be performed even if the fatigue screening criteria are satisfied (see 5.5.2). Protection against ratcheting is satisfied if one of the following three conditions is met:

(a) The loading results in only primary stresses without any cyclic secondary stresses.

(b) Elastic Stress Analysis Criteria – Protection against ratcheting is demonstrated by satisfying the rules of 5.5.6.

(c) Elastic–Plastic Stress Analysis Criteria – Protection against ratcheting is demonstrated by satisfying the rules of 5.5.7.

5.5.1.7 If a fatigue analysis is required, the effects of joint alignment (see 6.1.6.1) and weld peaking (see 6.1.6.3) in shells and heads shall be considered in the determination of the applicable stresses.

5.5.2 SCREENING CRITERIA FOR FATIGUE ANALYSIS

5.5.2.1 Overview.

(a) The provisions of this paragraph can be used to determine if a fatigue analysis is required as part of the vessel design. The screening options to determine the need for fatigue analysis are described below. If any one of the screening options is satisfied, then a fatigue analysis is not required as part of the vessel design.

(1) Provisions of 5.5.2.2, Experience with comparable equipment operating under similar conditions.

(2) Provisions of 5.5.2.3, Method A based on the materials of construction (limited applicability), construction details, loading histogram, and smooth bar fatigue curve data.

(3) Provisions of 5.5.2.4, Method B based on the materials of construction (unlimited applicability), construction details, loading histogram, and smooth bar fatigue curve data.

(b) The fatigue exemption in accordance with this paragraph is performed on a component or part basis. One component (integral) may be exempt, while another component (non-integral) is not exempt. If any one component is not exempt, then a fatigue evaluation shall be performed for that component.

(c) If the specified number of cycles is greater than $(10)^6$, then the screening criteria are not applicable and a fatigue analysis is required.

5.5.2.2 Fatigue Analysis Screening Based on Experience With Comparable Equipment. If successful experience over a sufficient time frame is obtained with comparable equipment subject to a similar loading histogram and addressed in the User's Design Specification (see 2.2.3.1(f)), then a fatigue analysis is not required as part of the vessel design. When evaluating experience with comparable equipment operating under similar conditions as related to the design and service contemplated, the possible harmful effects of the following design features shall be evaluated.

(a) The use of non-integral construction, such as the use of pad type reinforcements or of fillet welded attachments, as opposed to integral construction

(b) The use of pipe threaded connections, particularly for diameters in excess of 70 mm (2.75 in.)

(c) The use of stud bolted attachments

(d) The use of partial penetration welds

(e) Major thickness changes between adjacent members

(f) Attachments and nozzles in the knuckle region of formed heads

5.5.2.3 Fatigue Analysis Screening, Method A. The following procedure can only be used for materials with a specified minimum tensile strength that is less than or equal to 552 MPa (80,000 psi).

Step 1. Determine a load history based on the information in the User's Design Specification. The load history should include all cyclic operating loads and events that are applied to the component.

Step 2. Based on the load history in Step 1, determine the expected (design) number of full-range pressure cycles including startup and shutdown, and designate this value as $N_{\Delta FP}$.

Step 3. Based on the load history in Step 1, determine the expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction, and designate this value as $N_{\Delta PO}$. Pressure cycles in which the pressure variation does not exceed these percentages of the design pressure and pressure cycles caused by fluctuations in atmospheric conditions do not need to be considered in this evaluation.

Step 4. Based on the load history in Step 1, determine the effective number of changes in metal temperature difference between any two adjacent points, ΔT_E, as defined below, and designate this value as $N_{\Delta TE}$. The effective number of such changes is determined by multiplying the number of changes in metal temperature difference of a certain magnitude by the factor given in Table 5.8, and by adding the resulting numbers. In calculating the temperature difference between adjacent points, conductive heat transfer shall be considered only through welded or integral cross sections with no allowance for conductive heat transfer across un-welded contact surfaces (i.e., vessel shell and reinforcing pad).

(a) For surface temperature differences, points are considered to be adjacent if they are within the distance L computed as follows: for shells and dished heads in the meridional or circumferential directions,

$$L = 2.5\sqrt{Rt} \tag{5.16}$$

and for flat plates,

$$L = 3.5a \tag{5.17}$$

(b) For through-the-thickness temperature differences, adjacent points are defined as any two points on a line normal to any surface on the component.

Step 5. Based on the load history in Step 1, determine the number of temperature cycles for components involving welds between materials having different coefficients of thermal expansion that causes the value of $(\alpha_1 - \alpha_2)\Delta T$ to exceed 0.00034, and designate this value as $N_{\Delta T\alpha}$.

Step 6. If the expected number of operating cycles from Steps 2, 3, 4 and 5 satisfy the criterion in Table 5.9, then a fatigue analysis is not required as part of the vessel design. If this criterion is not satisfied, then a fatigue analysis is required as part of the vessel design. Examples of non-integral attachments are: screwed-on caps, screwed-in plugs, shear ring closures, fillet welded attachments, and breech lock closures.

5.5.2.4 Fatigue Analysis Screening, Method B. The following procedure can be used for all materials.

Step 1. Determine a load history based on the information in the User's Design Specification. The load histogram should include all significant cyclic operating loads and events that the component will be subjected. Note, in eq. (5.18), the number of cycles from the applicable design fatigue curve (see Annex 3-F) evaluated at a stress amplitude of S_e is defined as $N(S_e)$. Also in eqs. (5.19) through (5.23), the stress amplitude from the applicable design fatigue curve (see Annex 3-F) evaluated at N cycles is defined as $S_a(N)$.

Step 2. Determine the fatigue screening criteria factors, C_1 and C_2, based on the type of construction in accordance with Table 5.10, see 4.2.5.6(j).

Step 3. Based on the load histogram in Step 1, determine the design number of full-range pressure cycles including startup and shutdown, $N_{\Delta FP}$. If the following equation is satisfied, proceed to Step 4; otherwise, a detailed fatigue analysis of the vessel is required.

$$N_{\Delta FP} \le N\left(C_1 S\right) \tag{5.18}$$

Step 4. Based on the load histogram in Step 1, determine the maximum range of pressure fluctuation during normal operation, excluding startups and shutdowns, ΔP_N, and the corresponding number of significant cycles, $N_{\Delta P}$. Significant pressure fluctuation cycles are defined as cycles where the pressure range exceeds $S_{as}/C_1 S$ times the design pressure. If the following equation is satisfied, proceed to Step 5; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta P_N \le \frac{P}{C_1}\left(\frac{S_a\left(N_{\Delta P}\right)}{S}\right) \tag{5.19}$$

Step 5. Based on the load histogram in Step 1, determine the maximum temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation, ΔT_N, and the corresponding number of cycles, $N_{\Delta TN}$. If the following equation is satisfied, proceed to Step 6; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_N \leq \left(\frac{S_a(N_{\Delta TN})}{C_2 E_{ym} \alpha} \right) \tag{5.20}$$

Step 6. Based on the load histogram in Step 1, determine the maximum range of temperature difference fluctuation, ΔT_R, between any two adjacent points (see 5.5.2.3, Step 4) of the vessel during normal operation, excluding startups and shutdowns, and the corresponding number of significant cycles, $\Delta N_{\Delta TR}$. Significant temperature difference fluctuation cycles for this Step are defined as cycles where the temperature range exceeds $S_{as}/C_2 E_{ym}\alpha$. If the following equation is satisfied, proceed to Step 7; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_R \leq \left(\frac{S_a(N_{\Delta TR})}{C_2 E_{ym} \alpha} \right) \tag{5.21}$$

Step 7. Based on the load histogram in Step 1, determine the range of temperature difference fluctuation between any two adjacent points (see 5.5.2.3, Step 4) for components fabricated from different materials of construction during normal operation, ΔT_M, and the corresponding number of significant cycles, $N_{\Delta TM}$. Significant temperature difference fluctuation cycles for this Step are defined as cycles where the temperature range exceeds $S_{as}/[C_2(E_{y1}\alpha_1 - E_{y2}\alpha_2)]$. If the following equation is satisfied, proceed to Step 8; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_M \leq \left(\frac{S_a(N_{\Delta TM})}{C_2\left(E_{y1}\alpha_1 - E_{y2}\alpha_2\right)} \right) \tag{5.22}$$

Step 8. Based on the load histogram in Step 1, determine the equivalent stress range computed from the specified full range of mechanical loads, excluding pressure but including piping reactions, ΔS_{ML}, and the corresponding number of significant cycles, $N_{\Delta S}$. Significant mechanical load range cycles for this Step are defined as cycles where the stress range exceeds S_{as}. If the total specified number of significant load fluctuations exceeds the maximum number of cycles defined on the applicable fatigue curve, the S_{as} value corresponding to the maximum number of cycles defined on the fatigue curve shall be used. If the following equation is satisfied a fatigue analysis is not required; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta S_{ML} \leq S_a(N_{\Delta S}) \tag{5.23}$$

5.5.3 FATIGUE ASSESSMENT — ELASTIC STRESS ANALYSIS AND EQUIVALENT STRESSES

5.5.3.1 Overview.

(a) An effective total equivalent stress amplitude is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the effective total equivalent stress amplitude, defined as one-half of the effective total equivalent stress range $(P_L + P_b + Q + F)$ calculated for each cycle in the loading histogram.

(b) The primary plus secondary plus peak equivalent stress (see Figure 5.1) is the equivalent stress, derived from the highest value across the thickness of a section, of the combination of all primary, secondary, and peak stresses produced by specified operating pressures and other mechanical loads and by general and local thermal effects and including the effects of gross and local structural discontinuities. Examples of load case combinations for this stress category for typical pressure vessel components are shown in Table 5.3.

5.5.3.2 Assessment Procedure. The following procedure can be used to evaluate protection against failure due to cyclic loading based on the effective total equivalent stress amplitude.

Step 1. Determine a load history based on the information in the User's Design Specification and the methods in Annex 5-B. The load history should include all significant operating loads and events that are applied to the component. If the exact sequence of loads is not known, alternatives should be examined to establish the most severe fatigue damage, see Step 6.

Step 2. For a location in the component subject to a fatigue evaluation, determine the individual stress–strain cycles using the cycle counting methods in Annex 5-B. Define the total number of cyclic stress ranges in the histogram as M.

Step 3. Determine the equivalent stress range for the kth cycle counted in Step 2.

(a) If the effective alternating equivalent stress is computed using eq. (5.30), then determine the stress tensor at the start and end points (time points ${}^m t$ and ${}^n t$, respectively) for the kth cycle counted in Step 2, Determine the local thermal stress at time points ${}^m t$ and ${}^n t$, ${}^m\sigma^{LT}_{ij,k}$ and ${}^n\sigma^{LT}_{ij,k}$, respectively, as described in Annex 5-C. The component stress ranges between time points ${}^m t$ and ${}^n t$ and the effective equivalent stress ranges for use in eq. (5.30) are calculated using eqs. (5.24) through (5.27).

$$\Delta\sigma_{ij,k} = \left({}^m\sigma_{ij,k} - {}^m\sigma^{LT}_{ij,k}\right) - \left({}^n\sigma_{ij,k} - {}^n\sigma^{LT}_{ij,k}\right) \tag{5.24}$$

$$\left(\Delta S_{p,k} - \Delta S_{LT,k}\right) = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma^2_{12,k} + \Delta\sigma^2_{13,k} + \Delta\sigma^2_{23,k}\right)\right]^{0.5} \tag{5.25}$$

$$\Delta\sigma^{LT}_{ij,k} = {}^m\sigma^{LT}_{ij,k} - {}^n\sigma^{LT}_{ij,k} \tag{5.26}$$

$$\Delta S_{LT,k} = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma^{LT}_{11,k} - \Delta\sigma^{LT}_{22,k}\right)^2 + \left(\Delta\sigma^{LT}_{11,k} - \Delta\sigma^{LT}_{33,k}\right)^2 + \left(\Delta\sigma^{LT}_{22,k} - \Delta\sigma^{LT}_{33,k}\right)^2\right]^{0.5} \tag{5.27}$$

(b) If the effective alternating equivalent stress is computed using eq. (5.36), then determine the stress tensor at the start and end points (time points ${}^m t$ and ${}^n t$, respectively) for the kth cycle counted in Step 2, The component stress ranges between time points ${}^m t$ and ${}^n t$, and the effective equivalent stress range for use in eq. (5.36) are given by eqs. (5.28) and (5.29), respectively.

$$\Delta\sigma_{ij,k} = {}^m\sigma_{ij,k} - {}^n\sigma_{ij,k} \tag{5.28}$$

$$\Delta S_{p,k} = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma^2_{12,k} + \Delta\sigma^2_{13,k} + \Delta\sigma^2_{23,k}\right)\right]^{0.5} \tag{5.29}$$

Step 4. Determine the effective alternating equivalent stress amplitude for the kth cycle using the results from Step 3.

$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \left(\Delta S_{p,k} - \Delta S_{LT,k}\right) + K_{v,k} \cdot \Delta S_{LT,k}}{2} \tag{5.30}$$

(a) If the local notch or effect of the weld is accounted for in the numerical model, then $K_f = 1.0$ in eqs. (5.30) and (5.36). If the local notch or effect of the weld is not accounted for in the numerical model, then a fatigue strength reduction factor, K_f, shall be included. Recommended values for fatigue strength reduction factors for welds are provided in Tables 5.11 and 5.12.

(b) The fatigue penalty factor, $K_{e,k}$, in eqs. (5.30) and (5.36) is evaluated using the following equations where the parameters and are determined from Table 5.13 and S_{PS} and $\Delta S_{n,k}$ are defined in 5.5.6.1. For $K_{e,k}$ values greater than 1.0, the simplified elastic–plastic criteria of 5.5.6.2 shall be satisfied.

$$K_{e,k} = 1.0 \qquad \text{for} \quad \Delta S_{n,k} \le S_{PS} \tag{5.31}$$

$$K_{e,k} = 1.0 + \frac{(1-n)}{n(m-1)}\left(\frac{\Delta S_{n,k}}{S_{PS}} - 1\right) \qquad \text{for} \quad S_{PS} < \Delta S_{n,k} < mS_{PS} \tag{5.32}$$

$$K_{e,k} = \frac{1}{n} \qquad \text{for} \quad \Delta S_{n,k} \ge mS_{PS} \tag{5.33}$$

(c) The Poisson correction factor, $K_{v,k}$ in eq. (5.30) is computed using eq. (5.34).

$$K_{v,k} = \left(\frac{1 - v_e}{1 - v_p}\right) \tag{5.34}$$

where

$$v_p = \max\left[0.5 - 0.2\left(\frac{S_{y,k}}{S_{a,k}}\right), v_e\right] \tag{5.35}$$

(d) The Poisson correction factor, $K_{v,k}$, in eq. (5.34) need not be used if the fatigue penalty factor, $K_{e,k}$, is used for the entire stress range (including $\Delta S_{LT,k}$). In this case, eq. (5.30) becomes:

$$S_{\text{alt},k} = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2} \tag{5.36}$$

Step 5. Determine the permissible number of cycles, N_k, for the alternating equivalent stress computed in Step 4. Fatigue curves based on the materials of construction are provided in Annex 3-F, 3-F.1.

Step 6. Determine the fatigue damage for the kth cycle, where the actual number of repetitions of the kth cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.37}$$

Step 7. Repeat Steps 3 through 6 for all stress ranges, M, identified in the cycle counting process in Step 2.

Step 8. Compute the accumulated fatigue damage using the following equation. The location in the component is acceptable for continued operation if this equation is satisfied.

$$D_f = \sum_{k=1}^{M} D_{f,k} \leq 1.0 \tag{5.38}$$

Step 9. Repeat Steps 2 through 8 for each point in the component subject to a fatigue evaluation.

5.5.3.3 In Step 4 of 5.5.3.2, $K_{e,k}$ may be calculated using one of the following methods.

(a) Method 1 – The equivalent total strain range from elastic–plastic analysis and the equivalent total strain range from elastic analysis for the point of interest as given below.

$$K_{e,k} = \frac{\Delta\varepsilon_{\text{eff},k}}{\Delta\varepsilon_{el,k}} \tag{5.39}$$

where

$$\Delta\varepsilon_{\text{eff},k} = \frac{\Delta S_{P,k}}{E_{ya,k}} + \Delta\varepsilon_{\text{peq},k} \tag{5.40}$$

$$\Delta\varepsilon_{el,k} = \frac{\Delta S_{P,k}}{E_{ya,k}} \tag{5.41}$$

$$\Delta\varepsilon_{\text{peq},k} = \frac{\sqrt{2}}{3}\left[\left(\Delta p_{11,k} - \Delta p_{22,k}\right)^2 + \left(\Delta p_{22,k} - \Delta p_{33,k}\right)^2 + \left(\Delta p_{33,k} - \Delta p_{11,k}\right)^2 + 1.5\left(\Delta p_{12,k}^2 + \Delta p_{23,k}^2 + \Delta p_{31,k}^2\right)\right]^{0.5} \tag{5.42}$$

The stress range $\Delta S_{P,k}$ is given by eq. (5.29) and is the elastic–plastic value in eq. (5.40) and the elastic value in eq. (5.41). The component stress and plastic strain ranges (differences between the components at the start and end points of the cycle) for the kth cycle are designated as $\Delta\sigma_{ij,k}$ and $\Delta p_{ij,k}$, respectively. eq. (5.42) is a special form of the effective plastic strain equation based on engineering shear strains (typical FEA output and twice the tensorial shear strain values).

(b) Method 2 – The alternate plasticity adjustment factors and alternating equivalent stress may be computed using Annex 5-C.

5.5.3.4 In lieu of a detailed stress analysis, stress indices may be used to determine peak stresses around a nozzle opening in accordance with Annex 5-D.

5.5.4 FATIGUE ASSESSMENT — ELASTIC–PLASTIC STRESS ANALYSIS AND EQUIVALENT STRAINS

5.5.4.1 Overview.

(a) The Effective Strain Range is used to evaluate the fatigue damage for results obtained from an elastic–plastic stress analysis. The Effective Strain Range is calculated for each cycle in the loading histogram using either cycle-by-cycle analysis or the Twice Yield Method. For the cycle-by-cycle analysis, a cyclic plasticity algorithm with kinematic hardening shall be used.

(b) Twice Yield Method is an elastic–plastic stress analysis performed in a single loading step, based on a specified stabilized cyclic stress range-strain range curve and a specified load range representing a cycle. Stress and strain ranges are the direct output from this analysis. This method is performed in the same manner as a monotonic analysis and does not require cycle-by-cycle analysis of unloading and reloading. The Twice Yield Method can be used with an analysis program without cyclic plasticity capability.

(c) For the calculation of the stress range and strain range of a cycle at a point in the component, a stabilized cyclic stress–strain curve and other material properties shall be used based on the average temperature of the cycle being evaluated for each material of construction. The cyclic curve may be that obtained by test for the material, or that which is known to have more conservative cyclic behavior to the material that is specified. Cyclic stress–strain curves are also provided in 3-D.4 of Annex 3-D for certain materials and temperatures. Other cyclic stress–strain curves may be used that are known to be either more accurate for the application or lead to more conservative results.

5.5.4.2 Assessment Procedure. The following procedure can be used to evaluate protection against failure due to cyclic loading using elastic–plastic stress analysis.

Step 1. Determine a load history based on the information in the User's Design Specification and the methods in Annex 5-B. The load history should include all significant operating loads and events that are applied to the component.

Step 2. For a location in the component subject to a fatigue evaluation, determine the individual stress–strain cycles using the cycle counting methods in Annex 5-B. Define the total number of cyclic stress ranges in the histogram as M.

Step 3. Determine the loadings at the start and end points of the kth cycle counted in Step 2. Using these data, determine the loading ranges (differences between the loadings at the start and end points of the cycle).

Step 4. Perform elastic–plastic stress analysis for the kth cycle. For cycle-by-cycle analysis, constant-amplitude loading is cycled using cyclic stress amplitude-strain amplitude curve (5.5.4.1). For the Twice Yield Method, the loading at the start point of the cycle is zero and the loading at the end point is the loading range determined in Step 3. The cyclic stress range-strain range curve is used (5.5.4.1). For thermal loading, the loading range in Twice-Yield Method may be applied by specifying the temperature field at the start point for the cycle as an initial condition, and applying the temperature field at the end point for the cycle in a single loading step.

Step 5. Calculate the Effective Strain Range for the kth cycle.

$$\Delta\varepsilon_{\mathrm{eff},k} = \frac{\Delta S_{p,k}}{E_{ya,k}} + \Delta\varepsilon_{\mathrm{peq},k} \tag{5.43}$$

where the stress range $\Delta S_{p,k}$ is given by eq. (5.29) and $\Delta\mathcal{E}_{peq,k}$ is given by eq. (5.42).

The component stress and plastic strain ranges (differences between the components at the start and end points of the cycle) for the kth cycle are designated as $\Delta\sigma_{ij,k}$ and $\Delta p_{ij,k}$, respectively. However, since a range of loading is applied in a single load step with the Twice Yield Method, the calculated maximum equivalent plastic strain range, $\Delta\varepsilon_{peq,k}$ and the von Mises equivalent stress range $\Delta S_{p,k}$ defined above are typical output variables that can be obtained directly from a stress analysis.

Step 6. Determine the effective alternating equivalent stress for the kth cycle.

$$S_{\mathrm{alt},k} = \frac{E_{ya,k} \cdot \Delta\varepsilon_{\mathrm{eff},k}}{2} \tag{5.44}$$

Step 7. Determine the permissible number of cycles, N_k, for the alternating equivalent stress computed in Step 6. Fatigue curves based on the materials of construction are provided in Annex 3-F, 3-F.1,

Step 8. Determine the fatigue damage for the kth cycle, where the actual number of repetitions of the kth cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.45}$$

Step 9. Repeat Steps 3 through 8 for all stress ranges, M, identified in the cycle counting process in Step 2.

Step 10. Compute the accumulated fatigue damage using the following equation. The location in the component is acceptable for continued operation if this equation is satisfied.

$$\sum_{k=1}^{M} D_{f,k} \le 1.0 \tag{5.46}$$

Step 11. Repeat Steps 2 through 10 for each point in the component subject to a fatigue evaluation.

5.5.5 FATIGUE ASSESSMENT OF WELDS — ELASTIC ANALYSIS AND STRUCTURAL STRESS

5.5.5.1 Overview.

(a) An equivalent structural stress range parameter is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the structural stress that is a function of the membrane and bending stresses normal to the hypothetical crack plane. This method is recommended for evaluation of welded joints that have not been machined to a smooth profile. Weld joints with controlled smooth profiles may be evaluated using 5.5.3 or 5.5.4.

(b) Fatigue cracks at pressure vessel welds are typically located at the toe of a weld. For as-welded and weld joints subject to post weld heat treatment, the expected orientation of a fatigue crack is along the weld toe in the through-thickness direction, and the structural stress normal to the expected crack is the stress measure used to correlate fatigue life data. For fillet welded components, fatigue cracking may occur at the toe of the fillet weld or the weld throat, and both locations shall be considered in the assessment. It is difficult to accurately predict fatigue life at the weld throat due to variability in throat dimension, which is a function of the depth of the weld penetration. It is recommended to perform sensitivity analysis where the weld throat dimension is varied.

5.5.5.2 Assessment Procedure. The following procedure can be used to evaluate protection against failure due to cyclic loading using the equivalent structural stress range.

Step 1. Determine a load history based on the information in the User's Design Specification and the histogram development methods in Annex 5-B. The load history should include all significant operating loads and events that are applied to the component.

Step 2. For a location at a weld joint subject to a fatigue evaluation, determine the individual stress–strain cycles using the cycle counting methods in Annex 5-B. Define the total number of cyclic stress ranges in the histogram as M.

Step 3. Determine the elastically calculated membrane and bending stress normal to the assumed hypothetical crack plane at the start and end points (time points ${}^{m}t$ and ${}^{n}t$, respectively) for the kth cycle counted in Step 2. Using this data, calculate the membrane and bending stress ranges between time points ${}^{m}t$ and ${}^{n}t$, and the maximum, minimum and mean stress.

$$\Delta\sigma^{e}_{m,k} = \left({}^{m}\sigma^{e}_{m,k} + {}^{m}P_k\right) - \left({}^{n}\sigma^{e}_{m,k} + {}^{n}P_k\right) \tag{5.47}$$

$$\Delta\sigma^{e}_{b,k} = {}^{m}\sigma^{e}_{b,k} - {}^{n}\sigma^{e}_{b,k} \tag{5.48}$$

$$\sigma_{\max,k} = \max\left[\left({}^{m}\sigma^{e}_{m,k} + {}^{m}P_k + {}^{m}\sigma^{e}_{b,k}\right), \left({}^{n}\sigma^{e}_{m,k} + {}^{n}P_k + {}^{n}\sigma^{e}_{b,k}\right)\right] \tag{5.49}$$

$$\sigma_{\min,k} = \min\left[\left({}^{m}\sigma^{e}_{m,k} + {}^{m}P_k + {}^{m}\sigma^{e}_{b,k}\right), \left({}^{n}\sigma^{e}_{m,k} + {}^{n}P_k + {}^{n}\sigma^{e}_{b,k}\right)\right] \tag{5.50}$$

$$\sigma_{\mathrm{mean},k} = \frac{\sigma_{\max,k} + \sigma_{\min,k}}{2} \tag{5.51}$$

Step 4. Determine the elastically calculated structural stress range for the kth cycle, $\Delta\sigma^{e}_{k}$, using eq. (5.52).

$$\Delta\sigma^{e}_{k} = \Delta\sigma^{e}_{m,k} + \Delta\sigma^{e}_{b,k} \tag{5.52}$$

Step 5. Determine the elastically calculated structural strain, $\Delta\varepsilon^{e}_{k}$, from the elastically calculated structural stress, $\Delta\sigma^{e}_{k}$, using eq. (5.53)

$$\Delta\varepsilon^{e}_{k} = \frac{\Delta\sigma^{e}_{k}}{E_{ya,k}} \tag{5.53}$$

The corresponding local nonlinear structural stress and strain ranges, $\Delta\sigma_k$ and $\Delta\varepsilon_k$, respectively, are determined by simultaneously solving Neuber's Rule, eq. (5.54), and a model for the material hysteresis loop stress–strain curve given by eq. (5.55), see Annex 3-D, 3-D.4.

$$\Delta\sigma_k \cdot \Delta\varepsilon_k = \Delta\sigma^{e}_{k} \cdot \Delta\varepsilon^{e}_{k} \tag{5.54}$$

$$\Delta\varepsilon_k = \frac{\Delta\sigma_k}{E_{ya,k}} + 2\left(\frac{\Delta\sigma_k}{2K_{css}}\right)^{\frac{1}{n_{css}}} \tag{5.55}$$

The structural stress range computed solving eqs. (5.54) and (5.55) is subsequently modified for low-cycle fatigue using eq. (5.56).

$$\Delta\sigma_k = \left(\frac{E_{ya,k}}{1-\nu^2}\right)\Delta\varepsilon_k \tag{5.56}$$

NOTE: The modification for low-cycle fatigue should always be performed because the exact distinction between high-cycle fatigue and low-cycle fatigue cannot be determined without evaluating the effects of plasticity which is a function of the applied stress range and cyclic stress–strain curve. For high cycle fatigue applications, this procedure will provide correct results, i.e., the elastically calculated structural stress will not be modified.

Step 6. Compute the equivalent structural stress range parameter for the kth cycle using the following equations. In eq. (5.57), for SI Units, the thickness, t, stress range, $\Delta\sigma_k$, and the equivalent structural stress range parameter, $\Delta S_{ess,k}$, are in mm, MPa, and MPa/(mm)$^{(2-m_{ss})/2m_{ss}}$, respectively, and for U.S. Customary Units, the thickness, t, stress range, $\Delta\sigma_k$, and the equivalent structural stress range parameter, $\Delta S_{ess,k}$, are in ksi, ksi, and in ksi/(inches)$^{(2-m_{ss})/2m_{ss}}$, respectively.

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}} \tag{5.57}$$

where

$$m_{ss} = 3.6 \tag{5.58}$$

$$t_{ess} = 16 \text{ mm } (0.625 \text{ in.}) \quad \text{for} \quad t \le 16 \text{ mm } (0.625 \text{ in.}) \tag{5.59}$$

$$t_{ess} = t \quad \text{for} \quad 16 \text{ mm } (0.625 \text{ in.}) < t < 150 \text{ mm } (6 \text{ in.}) \tag{5.60}$$

$$t_{ess} = 150 \text{ mm } (6 \text{ in.}) \quad \text{for} \quad t \ge 150 \text{ mm } (6 \text{ in.}) \tag{5.61}$$

$$I^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b,k} - 0.17R_{b,k}^2}{1.007 - 0.306R_{b,k} - 0.178R_{b,k}^2} \tag{5.62}$$

$$R_{b,k} = \frac{\left|\Delta\sigma_{b,k}^e\right|}{\left|\Delta\sigma_{m,k}^e\right| + \left|\Delta\sigma_{b,k}^e\right|} \tag{5.63}$$

$$f_{M,k} = \left(1 - R_k\right)^{\frac{1}{m_{ss}}} \quad \text{for} \quad \sigma_{\text{mean},k} \ge 0.5S_{y,k}, \text{ and } R_k > 0, \text{ and } \left|\Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e\right| \le 2S_{y,k} \tag{5.64}$$

$$f_{M,k} = 1.0 \quad \text{for} \quad \sigma_{\text{mean},k} < 0.5S_{y,k}, \text{ or } R_k \le 0, \text{ or } \left|\Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e\right| > 2S_{y,k} \tag{5.65}$$

$$R_k = \frac{\sigma_{\min,k}}{\sigma_{\max,k}} \tag{5.66}$$

Step 7. Determine the permissible number of cycles, N_k, based on the equivalent structural stress range parameter for the kth cycle computed in Step 6. Fatigue curves for welded joints are provided in Annex 3-F, 3-F.2.

Step 8. Determine the fatigue damage for the kth cycle, where the actual number of repetitions of the kth cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.67}$$

Step 9. Repeat Steps 6 through 8 for all stress ranges, M, identified in the cycle counting process in Step 3.

Step 10. Compute the accumulated fatigue damage using the following equation. The location along the weld joint is suitable for continued operation if this equation is satisfied.

$$D_f = \sum_{i=1}^{M} D_{f,k} \leq 1.0 \tag{5.68}$$

Step 11. Repeat Steps 5 through 10 for each point along the weld joint that is subject to a fatigue evaluation.

5.5.5.3 Assessment Procedure Modifications. The assessment procedure in 5.5.5.2 may be modified as shown below.

(a) Multiaxial Fatigue – If the structural shear stress range is not negligible, i.e., $\Delta\tau_k > \Delta\sigma_k/3$, a modification should be made when computing the equivalent structural stress range. Two conditions need to be considered:

(1) If $\Delta\sigma_k$ and $\Delta\tau_k$ are out of phase, the equivalent structural stress range $\Delta S_{ess,k}$ in eq. (5.57) should be replaced by:

$$\Delta S_{ess,k} = \frac{1}{F(\delta)}\left[\left(\frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}}\right)^2 + 3\left(\frac{\Delta\tau_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I_\tau^{\frac{1}{m_{ss}}}}\right)^2\right]^{0.5} \tag{5.69}$$

where

$$I_\tau^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b\tau,k} - 0.17R_{b\tau,k}^2}{1.007 - 0.306R_{b\tau,k} - 0.178R_{b\tau,k}^2} \tag{5.70}$$

$$R_{b\tau,k} = \frac{\left|\Delta\tau_{b,k}^e\right|}{\left|\Delta\tau_{m,k}^e\right| + \left|\Delta\tau_{b,k}^e\right|} \tag{5.71}$$

$$\Delta\tau_k = \Delta\tau_{m,k}^e + \Delta\tau_{b,k}^e \tag{5.72}$$

$$\Delta\tau_{m,k}^e = {}^m\tau_{m,k}^e - {}^n\tau_{m,k}^e \tag{5.73}$$

$$\Delta\tau_{b,k}^e = {}^m\tau_{b,k}^e - {}^n\tau_{b,k}^e \tag{5.74}$$

In eq. (5.69), $F(\delta)$ is a function of the out-of-phase angle between $\Delta\sigma_k$ and $\Delta\tau_k$ if both loading modes can be described by sinusoidal functions, or:

$$F(\delta) = \frac{1}{\sqrt{2}}\left[1 + \left[1 - \frac{12 \cdot \Delta\sigma_k^2 \cdot \Delta\tau_k^2 \cdot \sin^2[\delta]}{\left[\Delta\sigma_k^2 + 3\Delta\tau_k^2\right]^2}\right]^{0.5}\right]^{0.5} \tag{5.75}$$

A conservative approach is to ignore the out-of-phase angle and recognize the existence of a minimum possible value for $F(\delta)$ in eq. (5.75) given by:

$$F(\delta) = \frac{1}{\sqrt{2}} \tag{5.76}$$

(2) If $\Delta\sigma_k$ and $\Delta\tau_k$ are in-phase the equivalent structural stress range $\Delta S_{ess,k}$ is given by eq. (5.69) with $F(\delta) = 1.0$.

(b) Weld Quality – If a defect exists at the toe of a weld that can be characterized as a crack-like flaw, i.e., undercut, and this defect exceeds the value permitted by Part 7, then a reduction in fatigue life shall calculated by substituting the value of $I^{1/m_{ss}}$ in Eq. (5.62) with the value given by eq. (5.77). In this equation, a is the depth of the crack-like flaw at the weld toe. eq. (5.77) is valid only when $a/t \le 0.1$.

$$\frac{1}{I^{m_{ss}}} = \frac{1.229 - 0.365R_{b,k} + 0.789\left(\frac{a}{t}\right) - 0.17R_{b,k}^2 + 13.771\left(\frac{a}{t}\right)^2 + 1.243R_{b,k}\left(\frac{a}{t}\right)}{1 - 0.302R_{b,k} + 7.115\left(\frac{a}{t}\right) - 0.178R_{b,k}^2 + 12.903\left(\frac{a}{t}\right)^2 - 4.091R_{b,k}\left(\frac{a}{t}\right)} \quad (5.77)$$

5.5.6 RATCHETING ASSESSMENT — ELASTIC STRESS ANALYSIS

5.5.6.1 Elastic Ratcheting Analysis Method.

(a) To evaluate protection against ratcheting the following limit shall be satisfied.

$$\Delta S_{n,k} \le S_{PS} \quad (5.78)$$

(b) The primary plus secondary equivalent stress range, $\Delta S_{n,k}$, is the equivalent stress range, derived from the highest value across the thickness of a section, of the combination of linearized general or local primary membrane stresses plus primary bending stresses plus secondary stresses ($P_L + P_b + Q$), produced by specified operating pressure and other specified mechanical loads and by general thermal effects. The effects of gross structural discontinuities but not of local structural discontinuities (stress concentrations) shall be included. Examples of this stress category for typical pressure vessel components are shown in Table 5.6. Load case combinations to be considered for this stress category for typical pressure vessel components are shown in Table 5.3.

(c) The maximum range of this equivalent stress is limited to S_{PS}. The quantity S_{PS} represents a limit on the primary plus secondary equivalent stress range and is defined in (d). In the determination of the maximum primary plus secondary equivalent stress range, it may be necessary to consider the effects of multiple cycles where the total stress range may be greater than the stress range of any of the individual cycles. In this case, the value of S_{PS} may vary with the specified cycle, or combination of cycles, being considered since the temperature extremes may be different in each case. Therefore, care shall be exercised to assure that the applicable value of S_{PS} for each cycle, or combination of cycles, is used (see 5.5.3).

(d) The allowable limit on the primary plus secondary stress range, S_{PS}, is computed as the larger of the quantities shown below.

(1) Three times the average of the S values for the material from Annex 3-A at the highest and lowest temperatures during the operational cycle.

(2) Two times the average of the S_y values for the material from Annex 3-D at the highest and lowest temperatures during the operational cycle, except that the value from (1) shall be used when the ratio of the minimum specified yield strength to ultimate tensile strength exceeds 0.70 or the value of S is governed by time-dependent properties as indicated in Annex 3-A.

5.5.6.2 Simplified Elastic–Plastic Analysis. The equivalent stress limit on the range of primary plus secondary equivalent stress in 5.5.6.1 may be exceeded, provided all of the following are true:

(a) The range of primary plus secondary membrane plus bending equivalent stress, excluding thermal stress, is less than S_{PS}.

(b) The value of the alternating stress range in 5.5.3.2, Step 4 is multiplied by the factor $K_{e,k}$ (see eqs. (5.31) through (5.33), or 5.5.3.3).

(c) The material of the component has a ratio of the specified minimum yield strength to specified minimum tensile strength of less than or equal to 0.80.

(d) The component meets the secondary equivalent stress range requirements of 5.5.6.3.

5.5.6.3 Thermal Stress Ratcheting Assessment. The allowable limit on the secondary equivalent thermal stress range to prevent ratcheting, when applied in conjunction with a steady-state general or local primary membrane equivalent stress, is determined below. This procedure can only be used with an assumed linear or parabolic distribution of a secondary stress range (e.g., thermal stress).

Step 1. Determine the ratio of the primary membrane stress to the specified minimum yield strength from Annex 3-D, at the average temperature of the cycle.

$$X = \left(\frac{P_m}{S_y}\right) \tag{5.79}$$

Step 2. Compute the secondary membrane equivalent stress range, ΔQ_m, using elastic analysis methods.

Step 3. Compute the secondary membrane plus bending equivalent thermal stress range, ΔQ_{mb}, using elastic analysis methods.

Step 4. Determine the allowable limit on the secondary membrane plus bending equivalent thermal stress range, S_{Qmb}.

(a) For a secondary equivalent thermal stress range with a linear variation through the wall thickness

$$S_{Qmb} = S_y\left(\frac{1}{X}\right) \quad \text{for} \quad 0 < X < 0.5 \tag{5.80}$$

$$S_{Qmb} = 4.0S_y(1 - X) \quad \text{for} \quad 0.5 \le X \le 1.0 \tag{5.81}$$

(b) For a secondary equivalent stress range from thermal loading with a parabolic constantly increasing or decreasing variation through the wall thickness

$$S_{Qmb} = S_y\left(\frac{1}{0.1224 + 0.9944X^2}\right) \quad \text{for} \quad 0.0 < X < 0.615 \tag{5.82}$$

$$S_{Qmb} = 5.2S_y(1 - X) \quad \text{for} \quad 0.615 \le X \le 1.0 \tag{5.83}$$

Step 5. Determine the allowable limit on the secondary membrane equivalent thermal stress range, S_{Qm}.

$$S_{Qm} = 2.0S_y(1 - X) \quad \text{for} \quad 0 < X < 1.0 \tag{5.84}$$

Step 6. To demonstrate protection against ratcheting, the following two criteria shall be satisfied:

$$\Delta Q_m \le S_{Qm} \tag{5.85}$$

$$\Delta Q_{mb} \le S_{Qmb} \tag{5.86}$$

5.5.6.4 Progressive Distortion of Non-Integral Connections. Screwed-on caps, screwed-in plugs, shear ring closures, and breech lock closures are examples of non-integral connections that are subject to failure by bell-mouthing or other types of progressive deformation. If any combination of applied loads produces yielding, such joints are subject to ratcheting because the mating members may become loose at the end of each complete operating cycle and may start the next cycle in a new relationship with each other, with or without manual manipulation. Additional distortion may occur in each cycle so that interlocking parts, such as threads, can eventually lose engagement. Therefore primary plus secondary equivalent stresses that produce slippage between the parts of a non-integral connection in which disengagement could occur as a result of progressive distortion, shall be limited to the minimum specified yield strength at temperature, S_y, or evaluated using the procedure in 5.5.7.2.

5.5.7 RATCHETING ASSESSMENT — ELASTIC–PLASTIC STRESS ANALYSIS

5.5.7.1 Overview. To evaluate protection against ratcheting using elastic–plastic analysis, an assessment is performed by application, removal and re application of the applied loadings. If protection against ratcheting is satisfied, it may be assumed that progression of the stress–strain hysteresis loop along the strain axis cannot be sustained with cycles and that the hysteresis loop will stabilize. A separate check for plastic shakedown to alternating plasticity is not required. The following assessment procedure can be used to evaluate protection against ratcheting using elastic–plastic analysis.

5.5.7.2 Assessment Procedure.

Step 1. Develop a numerical model of the component including all relevant geometry characteristics. The model used for analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads.

Step 2. Define all relevant loads and applicable load cases (see Table 5.1).

Step 3. An elastic perfectly plastic material model shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized. The yield strength defining the plastic limit shall be the minimum specified yield strength at temperature from Annex 3-D. The effects of non-linear geometry shall be considered in the analysis.

Step 4. Perform an elastic–plastic analysis for the applicable loading from Step 2 for a number of repetitions of a loading event (see Annex 5-B), or, if more than one event is applied, of two events that are selected so as to produce the highest likelihood of ratcheting.

Step 5. The ratcheting criteria below shall be evaluated after application of a minimum of three complete repetitions of the cycle. Additional cycles may need to be applied to demonstrate convergence. If any one of the following conditions is met, the ratcheting criteria are satisfied. If the criteria shown below are not satisfied, the component configuration (i.e., thickness) shall be modified or applied loads reduced and the analysis repeated.

(a) There is no plastic action (i.e., zero plastic strains incurred) in the component.

(b) There is an elastic core in the primary-load-bearing boundary of the component.

(c) There is not a permanent change in the overall dimensions of the component. This can be demonstrated by developing a plot of relevant component dimensions versus time between the last and the next to the last cycles.

5.6 SUPPLEMENTAL REQUIREMENTS FOR STRESS CLASSIFICATION IN NOZZLE NECKS

The following classification of stresses shall be used for stress in a nozzle neck. The classification of stress in the shell shall be in accordance with 5.2.2.2.

(a) Within the limits of reinforcement given by 4.5, whether or not nozzle reinforcement is provided, the following classification shall be applied.

(1) A P_m classification is applicable to equivalent stresses resulting from pressure-induced general membrane stresses as well as stresses, other than discontinuity stresses, due to external loads and moments including those attributable to restrained free end displacements of the attached pipe.

(2) A P_L classification shall be applied to local primary membrane equivalent stresses derived from discontinuity effects plus primary bending equivalent stresses due to combined pressure and external loads and moments including those attributable to restrained free end displacements of the attached pipe.

(3) A $P_L + P_b + Q$ classification (see 5.5.2) shall apply to primary plus secondary equivalent stresses resulting from a combination of pressure, temperature, and external loads and moments, including those due to restrained free end displacements of the attached pipe.

(b) Outside of the limits of reinforcement given in 4.5, the following classification shall be applied.

(1) A P_m classification is applicable to equivalent stresses resulting from pressure-induced general membrane stresses as well as the average stress across the nozzle thickness due to externally applied nozzle axial, shear, and torsional loads other than those attributable to restrained free end displacement of the attached pipe.

(2) A $P_L + P_b$ classification is applicable to the equivalent stresses resulting from adding those stresses classified as P_m to those due to externally applied bending moments except those attributable to restrained free end displacement of the pipe.

(3) A $P_L + P_b + Q$ classification (see 5.5.2) is applicable to equivalent stresses resulting from all pressure, temperature, and external loads and moments, including those attributable to restrained free end displacements of the attached pipe.

(c) Beyond the limits of reinforcement, the S_{PS} limit on the range of primary plus secondary equivalent stress may be exceeded as provided in 5.5.6.2, except that in the evaluation of the range of primary plus secondary equivalent stress, $P_L + P_b + Q$, stresses resulting from the restrained free end displacements of the attached pipe may also be excluded. The range of membrane plus bending equivalent stress attributable solely to the restrained free end displacements of the attached piping shall be less than S_{PS}.

5.7 SUPPLEMENTAL REQUIREMENTS FOR BOLTS

5.7.1 DESIGN REQUIREMENTS

(a) The number and cross-sectional area of bolts required to resist the design pressure shall be determined in accordance with the procedures of 4.16. The allowable bolt stress shall be obtained from Part 3.

(b) When sealing is effected by a seal weld instead of a gasket, the gasket factor, m, and the minimum gasket seating stress, y, may be taken as zero.

(c) When gaskets are used for pre-service testing only, the design is satisfactory if the above requirements are satisfied for m and y factors equal to zero, and the requirements of 5.7.1 and 5.7.2 are satisfied when the appropriate m and y factors are used for the test gasket.

5.7.2 SERVICE STRESS REQUIREMENTS

Actual service stress in bolts, such as those produced by the combination of preload, pressure, and differential expansion, may be higher than the allowable stress values given in Annex 3-A.

(a) The maximum value of service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two times the allowable stress values in 3-A.2.2 of Annex 3-A.

(b) The maximum value of service stress, except as restricted by 5.7.3.1(b) at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations, shall not exceed three times the allowable stress values in 3-A.2 of Annex 3-A. When the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion, the stress measure used in the evaluation shall be the equivalent stress as defined in eq. (5.1).

5.7.3 FATIGUE ASSESSMENT OF BOLTS

5.7.3.1 The suitability of bolts for cyclic operation shall be determined in accordance with the following procedures unless the vessel on which they are installed meets all the conditions of 5.5.2 (a fatigue analysis is not required).

(a) Bolts made of materials which have minimum specified tensile strengths of less than 689 MPa (100,000 psi) shall be evaluated for cyclic operation using the method in 5.5.3, using the applicable design fatigue curves (see Annex 3-F), and, unless it can be shown by analysis or test that a lower value is appropriate, the fatigue strength reduction factor used in the evaluation shall not be less than 4.0.

(b) High strength alloy steel bolts and studs shall be evaluated for cyclic operation using the methodology in 5.5.3 with the applicable design fatigue curve of Annex 3-F, provided all of the following are true:

(1) The material is one of the following: SA-193 Grade B7 or B16, SA-320 Grade L43, SA-540 Grades B23 and B24, heat treated in accordance with Section 5 of SA-540.

(2) The maximum value of the service stress at the periphery of the bolt cross section (resulting from direct tension plus bending and neglecting stress concentrations) shall not exceed $2.7S$, if the higher of the two fatigue design curves for high strength bolting given in Annex 3-F is used (the $2S$ limit for direct tension is unchanged).

(3) The threads shall be of a V-Type, having a minimum thread root radius no smaller than 0.076 mm (0.003 in.).

(4) The fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

(5) The fatigue strength reduction factor used in the evaluation shall not be less than 4.0.

5.7.3.2 The bolts shall be acceptable for the specified cyclic operation application of loads and thermal stresses, provided the fatigue damage fraction, D_f, is less than or equal to 1.0 (see 5.5.3).

5.8 SUPPLEMENTAL REQUIREMENTS FOR PERFORATED PLATES

Perforated plates may be analyzed using any of the procedures in this Part if the holes are explicitly included in the numerical model used for the stress analysis. An elastic stress analysis option utilizing the concept of an effective solid plate is described Annex 5-E.

5.9 SUPPLEMENTAL REQUIREMENTS FOR LAYERED VESSELS

The equations developed for solid wall cylindrical shells, spherical shells, or heads as expressed in this Part may be applied to layered cylindrical shells, spherical shells or heads, provided that in-plane shear force on each layer is adequately supported by the weld joint. In addition, consideration shall be given to the construction details in the zones of load application. In order to assure solid wall equivalence for layered cylindrical shells, spherical shells, or heads as described above, all cylindrical shells, spherical shells, or heads subjected to radial forces and/or longitudinal bending moments due to discontinuities or externally applied loads shall have all layers adequately bonded together to resist any longitudinal shearing forces resulting from the radial forces and/or longitudinal bending moments acting on the sections. For example, the use of the girth weld to bond layers together is shown in Figures 5.2, 5.3, and 5.4. The required width of the attachment weld at the midpoint of the weld depth is given by eq. (5.87).

$$w = 1.88\left(\frac{M_o}{t \cdot S}\right) \tag{5.87}$$

In eq. (5.87), the parameter M_o is the longitudinal bending moment per unit length of circumference existing at the weld junction of a layered cylindrical shells, spherical shells, or head. This parameter is determined from a stress analysis considering the pressure loading and all externally applied loads such as M_1, Q_1, and F_1.

5.10 EXPERIMENTAL STRESS ANALYSIS

Requirements for determining stresses in parts using experimental stress analysis are provided in Annex 5-F.

5.11 FRACTURE MECHANICS EVALUATIONS

Fracture mechanics evaluations performed to determine the MDMT in accordance with 3.11.2.8 shall be in accordance with API/ASME FFS-1. Residual stresses resulting from welding shall be considered along with primary and secondary stresses in all fracture mechanics calculations.

5.12 DEFINITIONS

1. Bending Stress: The variable component of normal stress, the variation may or may not be linear across the section thickness.

2. Bifurcation Buckling: The point of instability where there is a branch in the primary load versus displacement path for a structure.

3. Event: The User's Design Specification may include one or more events that produce fatigue damage. Each event consists of loading components specified at a number of time points over a time period and is repeated a specified number of times. For example, an event may be the startup, shutdown, upset condition, or any other cyclic action. The sequence of multiple events may be specified or random.

4. Cycle: A cycle is a relationship between stress and strain that is established by the specified loading at a location in a vessel or component. More than one stress–strain cycle may be produced at a location, either within an event or in transition between two events, and the accumulated fatigue damage of the stress–strain cycles determines the adequacy for the specified operation at that location. This determination shall be made with respect to the stabilized stress–strain cycle.

5. Cyclic Loading: A service in which fatigue becomes significant due to the cyclic nature of the mechanical and/or thermal loads. A screening criteria is provided in 5.5.2 that can be used to determine if a fatigue analysis should be included as part of the vessel design.

6. Fatigue: The conditions leading to fracture under repeated or fluctuating stresses having a maximum value less than the tensile strength of the material.

7. Fatigue Endurance Limit: The maximum stress below which a material can undergo 10^{11} alternating stress cycles without failure.

8. Fatigue Strength Reduction Factor: A stress intensification factor which accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. It is the ratio of the fatigue strength of a component without a discontinuity or weld joint to the fatigue strength of that same component with a discontinuity or weld joint. Values for some specific cases are empirically determined (e.g., socket welds). In the absence of experimental data, the stress intensification factor can be developed from a theoretical stress concentration factor derived from the theory of elasticity or based on the guidance provided in Tables 5.11 and 5.12.

9. Fracture Mechanics: An engineering discipline concerned with the behavior of cracks in materials. Fracture mechanics models provide mathematical relationships for critical combinations of stress, crack size and fracture toughness that lead to crack propagation. Linear Elastic Fracture Mechanics (LEFM) approaches apply to cases where crack propagation occurs during predominately elastic loading with negligible plasticity. Elastic–Plastic Fracture Mechanics (EPFM) methods are suitable for materials that undergo significant plastic deformation during crack propagation.

10. Gross Structural Discontinuity: A source of stress or strain intensification that affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples of gross structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

11. Local Primary Membrane Stress: Cases arise in which a membrane stress produced by pressure, or other mechanical loading associated with a primary and/or a discontinuity effect would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress.

12. Local Structural Discontinuity: A source of stress or strain intensification which affects a relatively small volume of material and does not have a significant effect on the overall stress or strain pattern, or on the structure as a whole. Examples are small fillet radii, small attachments, and partial penetration welds.

13. Membrane Stress: The component of normal stress that is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration.

14. Normal Stress: The component of stress normal to the plane of reference. Usually the distribution of normal stress is not uniform through the thickness of a part.

15. Operational Cycle: An operational cycle is defined as the initiation and establishment of new conditions followed by a return to the conditions that prevailed at the beginning of the cycle. Three types of operational cycles are considered: the startup-shutdown cycle, defined as any cycle which has atmospheric temperature and/or pressure as one of its extremes and normal operating conditions as its other extreme; the initiation of, and recovery from, any emergency or upset condition or pressure test condition that shall be considered in the design; and the normal operating cycle, defined as any cycle between startup and shutdown which is required for the vessel to perform its intended purpose.

16. Peak Stress: The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress that is not highly localized falls into this category if it is of a type that cannot cause noticeable distortion. Examples of peak stress are: the thermal stress in the austenitic steel cladding of a carbon steel vessel, the thermal stress in the wall of a vessel or pipe caused by a rapid change in temperature of the contained fluid, and the stress at a local structural discontinuity.

17. Primary Stress: A normal or shear stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses which considerably exceed the yield strength will result in failure or at least in gross distortion. A thermal stress is not classified as a primary stress. Primary membrane stress is divided into general and local categories. A general primary membrane stress is one that is distributed in the structure such that no redistribution of load occurs as a result of yielding. Examples of primary stress are general membrane stress in a circular cylindrical or a spherical shell due to internal pressure or to distributed live loads and the bending stress in the central portion of a flat head due to pressure. Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a primary and/or a discontinuity effect would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress. Finally a primary bending stress can be defined as a bending stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium of external and internal forces and moments.

18. Ratcheting: A progressive incremental inelastic deformation or strain that can occur in a component subjected to variations of mechanical stress, thermal stress, or both (thermal stress ratcheting is partly or wholly caused by thermal stress). Ratcheting is produced by a sustained load acting over the full cross section of a component, in combination with a strain controlled cyclic load or temperature distribution that is alternately applied and removed. Ratcheting causes cyclic straining of the material, which can result in failure by fatigue and at the same time produces cyclic incremental growth of a structure, which could ultimately lead to collapse.

19. Secondary Stress: A normal stress or a shear stress developed by the constraint of adjacent parts or by self-constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur and failure from one application of the stress is not to be expected. Examples of secondary stress are a general thermal stress and the bending stress at a gross structural discontinuity.

20. Shakedown: Caused by cyclic loads or cyclic temperature distributions which produce plastic deformations in some regions of the component when the loading or temperature distribution is applied, but upon removal of the loading or temperature distribution, only elastic primary and secondary stresses are developed in the component, except in small areas associated with local stress (strain) concentrations. These small areas shall exhibit a stable hysteresis loop, with no indication of progressive deformation. Further loading and unloading, or applications and removals of the temperature distribution shall produce only elastic primary and secondary stresses.

21. Shear Stress: The component of stress tangent to the plane of reference.

22. Stress Concentration Factor: The ratio of the maximum stress to the average section stress or bending stress.

23. Stress Cycle: A stress cycle is a condition in which the alternating stress difference goes from an initial value through an algebraic maximum value and an algebraic minimum value and then returns to the initial value. A single operational cycle may result in one or more stress cycles.

24. Thermal Stress: A self-balancing stress produced by a non-uniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally should under a change in temperature. For the purpose of establishing allowable stresses, two types of thermal stress are recognized, depending on the volume or area in which distortion takes place. A general thermal stress that is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. Therefore this type is classified as a secondary stress. Examples of general thermal stress are: the stress produced by an axial temperature distribution in a cylindrical shell, the stress produced by the temperature difference between a nozzle and the shell to which It is attached, and the equivalent linear stress produced by the radial temperature distribution in a cylindrical shell. A Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as local stresses. Examples of local thermal stresses are the stress in a small hot spot in a vessel wall, the difference between the non-linear portion of a through-wall temperature gradient in a cylindrical shell, and the thermal stress in a cladding material that has a coefficient of expansion different from that of the base metal.

5.13 NOMENCLATURE

a = radius of hot spot or heated area within a plate or the depth of a flaw at a weld toe, as applicable.
α = thermal expansion coefficient of the material at the mean temperature of two adjacent points, the thermal expansion coefficient of material evaluated at the mean temperature of the cycle, or the cone angle, as applicable.
α_1 = thermal expansion coefficient of material 1 evaluated at the mean temperature of the cycle.
α_2 = thermal expansion coefficient of material 2 evaluated at the mean temperature of the cycle.
α_{sl} = material factor for the multiaxial strain limit.
β = elastic–plastic load factor for Class 1 or Class 2 construction (see Table 4.1.3)
β_{cr} = capacity reduction factor.
β_T = test load factor for hydrostatic or pneumatic test and for Class 1 or Class 2 construction (see Table 4.1.3)
C_1 = factor for a fatigue analysis screening based on Method B.
C_2 = factor for a fatigue analysis screening based on Method B.
D_f = is cumulative fatigue damage.
$D_{f,k}$ = is fatigue damage for the kth cycle.
D_ε = cumulative strain limit damage.
$D_{\varepsilon\mathrm{form}}$ = strain limit damage from forming.
$D_{\varepsilon,k}$ = strain limit damage for the kth loading condition.
$\Delta e_{ij,k}$ = change in total strain range components minus the free thermal strain at the point under evaluation for the kth cycle.
$\Delta\varepsilon_k$ = local nonlinear structural strain range at the point under evaluation for the kth cycle.
$\Delta{\varepsilon^e}_k$ = elastically calculated structural strain range at the point under evaluation for the kth cycle.
$\Delta\varepsilon_{el,k}$ = equivalent strain range for the kth cycle, computed from elastic analysis.
$\Delta\varepsilon_{ij,k}$ = component strain range for the kth cycle, computed using the total strain less the free thermal strain
$\Delta\varepsilon_{peq,k}$ = equivalent plastic strain range for the kth loading condition or cycle.
$\Delta\varepsilon_{\mathrm{eff},k}$ = Effective Strain Range for the kth cycle.
$\Delta p_{ij,k}$ = change in plastic strain range components at the point under evaluation for the kth loading condition or cycle.
ΔP_N = maximum design range of pressure associated with $N_{\Delta P}$.
$\Delta S_{n,k}$ = primary plus secondary equivalent stress range.
$\Delta S_{P,k}$ = range of primary plus secondary plus peak equivalent stress for the kth cycle.
$\Delta S_{LT,k}$ = local thermal equivalent stress for the kth cycle.
$\Delta S_{ess,k}$ = equivalent structural stress range parameter for the kth cycle.
ΔS_{ML} = equivalent stress range computed from the specified full range of mechanical loads, excluding pressure but including piping reactions.
ΔQ = range of secondary equivalent stress.

ΔT = operating temperature range.
ΔT_E = effective number of changes in metal temperature between any two adjacent points.
ΔT_M = temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TM}$.
ΔT_N = temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TM}$.
ΔT_R = temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TM}$.
$\Delta \sigma_i$ = stress range associated with the principal stress in the ith direction.
$\Delta \sigma_{ij}$ = stress tensor range.
$\Delta \sigma_k$ = local nonlinear structural stress range at the point under evaluation for the kth cycle.
$\Delta {\sigma^e}_k$ = elastically calculated structural stress range at the point under evaluation for the kth cycle.
$\Delta {\sigma^e}_{b,k}$ = elastically calculated structural bending stress range at the point under evaluation for the kth cycle.
$\Delta \sigma_{ij,k}$ = stress tensor range at the point under evaluation for the kth cycle.
$\Delta {\sigma^e}_{m,k}$ = elastically calculated structural membrane stress range at the point under evaluation for the kth cycle.
$\Delta \tau_k$ = structural shear stress range at the point under evaluation for the kth cycle.
$\Delta {\tau^e}_{b,k}$ = elastically calculated bending component of the structural shear stress range at the point under evaluation for the kth cycle.
$\Delta {\tau^e}_{m,k}$ = elastically calculated membrane component of the structural shear stress range at the point under evaluation for the kth cycle.
δ = out-of-phase angle between $\Delta \sigma_k$ and Δ_{τ_k} for the kth cycle.
E_y = unmodified Young's modulus for plate material.
E_{yf} = value of modulus of elasticity on the fatigue curve being utilized.
$E_{ya,k}$ = value of modulus of elasticity of the material at the point under consideration, evaluated at the mean temperature of the kth cycle.
E_{y1} = Young's Modulus of material 1 evaluated at the mean temperature of the cycle.
E_{y2} = Young's Modulus of material 2 evaluated at the mean temperature of the cycle.
E_{ym} = Young's Modulus of the material evaluated at the mean temperature of the cycle.
ε_{cf} = cold forming strain.
ε_{Lu} = uniaxial strain limit.
$\varepsilon_{L,k}$ = limiting triaxial strain.
$f_{M,k}$ = mean stress correction factor for the kth cycle.
F = additional stress produced by the stress concentration over and above the nominal stress level resulting from operating loadings.
F_1 = externally applied axial force.
$F(\delta)$ = a fatigue modification factor based on the out-of-phase angle between $\Delta \sigma_k$ and Δ_{τ_k}.
I = correction factor used in the structural stress evaluation.
I_τ = correction factor used in the structural shear stress evaluation.
K_{css} = material parameter for the cyclic stress–strain curve model.
$K_{e,k}$ = fatigue penalty factor for the kth cycle.
$K_{v,k}$ = plastic Poisson's ratio adjustment for local thermal and thermal bending stresses for the kth cycle.
K_f = fatigue strength reduction factor used to compute the cyclic stress amplitude or range.
K_L = equivalent stress load factor.
K_m = ratio of peak stress in reduced ligament to the peak stress in normal ligament.
M = total number of stress ranges at a point derived from the cycle counting procedure.
M_o = longitudinal bending moment per unit length of circumference existing at the weld junction of layered spherical shells or heads due to discontinuity or external loads.
M_1 = externally applied bending moment.
m = material constant used for the fatigue knock-down factor used in the simplified elastic–plastic analysis.
m_{ij} = mechanical strain tensor, mechanical strain is defined as the total strain minus the free thermal strain.
m_{ss} = exponent used in a fatigue analysis based on the structural stress.
n = material constant used for the fatigue knock-down factor used in the simplified elastic–plastic analysis
n_k = actual number of repetitions of the kth cycle.
n_{css} = material parameter for the cyclic stress–strain curve model.
N_k = permissible number of cycles for the kth cycle.
$N(C_1\ S)$ = number of cycles from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at a stress amplitude of $C_1 S$.

$N(S_e)$ = number of cycles from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at a stress amplitude of S_e.
$N_{\Delta FP}$ = design number of full-range pressure cycles including startup and shutdown.
$N_{\Delta P}$ = number of significant cycles associated with ΔP_N.
$N_{\Delta PO}$ = expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction.
$N_{\Delta S}$ = number of significant cycles associated with ΔS_{ML}, significant cycles are those for which the range in temperature exceeds S_{as}.
$N_{\Delta T}$ = number of cycles associated with ΔT_N.
$N_{\Delta TE}$ = number of cycles associated with ΔT_E.
$N_{\Delta TM}$ = number of significant cycles associated with ΔT_M.
$N_{\Delta TR}$ = number of significant cycles associated with ΔT_R.
$N_{\Delta Ta}$ = number of temperature cycles for components involving welds between materials having different coefficients of expansion.
ν = Poisson's ratio.
Ω_P = load factor for pressure when combined with occasional load L, S_s, W, or E (see Table 5.2 for load parameter definitions).
= 1.0 unless otherwise specified in the User's Design Specification [see 2.2.2.1(e)].
$\Omega_P P$ = maximum anticipated operating pressure (internal or external) acting simultaneously with occasional load L, S_s, W, or E.
P = specified design pressure.
P_b = primary bending equivalent stress.
P_L = local primary membrane equivalent stress.
P_m = general primary membrane equivalent stress.
P_T = selected hydrostatic or pneumatic test pressure [see 8.2.1(c)].
${}^mP_k$ = component crack face pressure at time point ${}^mt$ for the kth cycle. The crack face pressure should be specified if the maximum value of the membrane plus bending stress used in the analysis occurs on a surface that is exposed to the fluid pressure. A conservative approach is to always specify the crack face pressure. The crack face pressure is based on the actual or operating pressure defined in the loading time history.
${}^nP_k$ = component crack face pressure at time point ${}^nt$ for the kth cycle. The crack face pressure should be specified if the maximum value of the membrane plus bending stress used in the analysis occurs on a surface that is exposed to the fluid pressure. A conservative approach is to always specify the crack face pressure. The crack face pressure is based on the actual or operating pressure defined in the loading time history.
Φ_B = design factor for buckling.
Q = secondary equivalent stress resulting from operating loadings.
Q_1 = externally applied shear force.
R = inside radius measured normal to the surface from the mid-wall of the shell to the axis of revolution, or the ratio of the minimum stress in the kth cycle to the maximum stress in the kth cycle, as applicable.
R_k = stress ratio for the kth cycle.
$R_{b,k}$ = ratio of the bending stress to the membrane plus bending stress.
$R_{b\tau,k}$ = ratio of the bending component of the shear stress to the membrane plus bending component of the shear stress.
RSF = computed remaining strength factor.
R_1 = mid-surface radius of curvature of region 1 where the local primary membrane stress exceeds $1.1S$.
R_2 = mid-surface radius of curvature of region 2 where the local primary membrane stress exceeds $1.1S$.
S = allowable stress based on the material of construction and design temperature.
S_a = alternating stress obtained from a fatigue curve for the specified number of operating cycles.
S_{as} = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at 1E6 cycles.
S_e = computed equivalent stress.
S_Q = allowable limit on the secondary stress range.
S_{PL} = allowable limit on the local primary membrane and local primary membrane plus bending stress categories (see 5.2.2.4.)
S_{PS} = allowable limit on the primary plus secondary stress range (see 5.5.6.)
S_y = minimum specified yield strength at the design temperature.
${S^L}_y$ = specified plastic limit for limit-load analysis.

$S_{a,k}$ = value of alternating stress obtained from the applicable design fatigue curve for the specified number of cycles of the kth cycle.
$S_{\mathrm{alt},k}$ = alternating equivalent stress for the kth cycle.
$S_{y,k}$ = yield strength of the material evaluated at the mean temperature of the kth cycle.
$S_a\ (N)$ = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at N cycles.
$S_a\ (N_{\Delta P})$ = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at $N_{\Delta P}$ cycles.
$S_a\ (N_{\Delta S})$ = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at $N_{\Delta S}$ cycles.
$S_a\ (N_{\Delta TN})$ = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at $N_{\Delta TN}$ cycles.
$S_a\ (N_{\Delta TM})$ = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at $N_{\Delta TM}$ cycles.
$S_a(N_{\Delta TR})$ = stress amplitude from the applicable design fatigue curve (see Annex 3-F, 3-F.1.3) evaluated at $N_{\Delta TR}$ cycles.
${}^m\sigma^e_{b,k}$ = elastically calculated bending stress at the point under evaluation for the kth cycle at the m point.
${}^n\sigma^e_{b,k}$ = elastically calculated bending stress at the point under evaluation for the kth cycle at the n point.
σ_e = von Mises stress.
$\sigma_{e,k}$ = von Mises stress for the kth loading condition.
σ_i = are the principal stress components.
$\sigma_{ij,k}$ = stress tensor at the point under evaluation for the kth cycle at the m point.
${}^m\sigma_{ij,k}$ = stress tensor at the point under evaluation for the kth cycle at the m point.
${}^n\sigma_{ij,k}$ = stress tensor at the point under evaluation for the kth cycle at the n point.
$\sigma^{LT}_{ij,k}$ = stress tensor due to local thermal stress at the location and time point under evaluation for the kth cycle.
${}^m\sigma^e_{m,k}$ = elastically calculated membrane stress at the point under evaluation for the kth cycle at the m point.
${}^n\sigma^e_{m,k}$ = elastically calculated membrane stress at the point under evaluation for the kth cycle at the n point.
$\sigma_{\max,k}$ = maximum stress in the kth cycle.
$\sigma_{\mathrm{mean},k}$ = mean stress in the kth cycle.
$\sigma_{\min,k}$ = minimum stress in the kth cycle.
σ_1 = principal stress in the 1-direction.
σ_2 = principal stress in the 2-direction.
σ_3 = principal stress in the 3-direction.
$\sigma_{1,k}$ = principal stress in the 1-direction for the kth loading condition.
$\sigma_{2,k}$ = principal stress in the 2-direction for the kth loading condition.
$\sigma_{3,k}$ = principal stress in the 3-direction for the kth loading condition.
t = minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable.
t_{ess} = structural stress effective thickness.
t_1 = minimum wall thickness associated with R_1.
t_2 = minimum wall thickness associated with R_2.
${}^m\tau^e_{b,k}$ = elastically calculated bending component of shear stress distribution at the point under evaluation for the kth cycle at the m point.
${}^n\tau^e_{b,k}$ = elastically calculated bending component of shear stress distribution at the point under evaluation for the kth cycle at the n point.
${}^m\tau^e_{m,k}$ = elastically calculated membrane component of shear stress distribution at the point under evaluation for the kth cycle at the m point.
${}^n\tau^e_{m,k}$ = elastically calculated membrane component of shear stress distribution at the point under evaluation for the kth cycle at the n point.
UTS = minimum specified ultimate tensile strength at room temperature.
w = required with of attachment.
X = maximum general primary membrane stress divided by the yield strength.
YS = minimum specified yield strength at room temperature.

5.14 TABLES

Table 5.1
Loads and Load Cases to Be Considered in a Design

Loading Condition	Design Loads
Pressure testing	*(1)* Dead load of component plus insulation, fireproofing, installed internals, platforms, and other equipment supported from the component in the installed position *(2)* Piping loads including pressure thrust *(3)* Applicable live loads excluding vibration and maintenance live loads *(4)* Pressure and fluid loads (water) for testing and flushing equipment and piping unless a pneumatic test is specified *(5)* Wind loads
Normal operation	*(1)* Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms, and other equipment supported from the component in the installed position *(2)* Piping loads including pressure thrust *(3)* Applicable live loads *(4)* Pressure and fluid loading during normal operation *(5)* Thermal loads
Normal operation plus occasional [Note (1)]	*(1)* Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms, and other equipment supported from the component in the installed *(2)* Piping loads including pressure thrust *(3)* Applicable live loads *(4)* Pressure and fluid loading during normal operation *(5)* Thermal loads *(6)* Wind, earthquake, or other occasional loads, whichever is greater *(7)* Loads due to wave action
Abnormal or start-up operation plus occasional [Note (1)]	*(1)* Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms, and other equipment supported from the component in the installed position *(2)* Piping loads including pressure thrust *(3)* Applicable live loads *(4)* Pressure and fluid loading associated with the abnormal or start-up conditions *(5)* Thermal loads *(6)* Wind loads

NOTE:
(1) Occasional loads are usually governed by wind and earthquake; however, other load types such as snow and ice loads may govern, see ASCE-7.

Table 5.2 Load Combination Parameters

Design Load Parameter	Operating Load Parameter [Note (1)]	Description
P	P_o	Internal and external specified design or operating pressure
P_S	P_{So}	Design or operating static head from liquid or bulk materials (e.g., catalyst)
P_T	...	Selected hydrostatic or pneumatic test pressure [see 8.2.1(d) or 8.3.1]. Used only in the pressure test load combinations.
D	D_o	Deadweight of the vessel, contents, and appurtenances at the location of interest, including the following: • weight of vessel including internals, supports (e.g., skirts, lugs, saddles, and legs), and appurtenances (e.g., platforms, ladders, etc.) • weight of vessel contents design, operating, and test conditions • refractory linings, insulation • static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping • transportation loads (the static forces obtained as equivalent to the dynamic loads experienced during normal operation of a transport vessel [see 1.2.1.3(b)]
L	L_o	• Appurtenance live loading • Effects of fluid momentum, steady state, and transient • Loads resulting from wave action
E	Same as design parameter	Earthquake loads (see 5.1.3.2)
W		Wind loads (see 5.1.3.2)
W_{pt}		The pressure test wind load case. The design wind speed for this case shall be specified by the Owner-User. Used only in the pressure test load combinations.
S_s		Snow loads
T	T_o	The self-restraining load case (i.e., thermal loads, applied displacements). This load case does not typically affect the collapse load, but should be considered in cases where elastic follow-up causes stresses that do not relax sufficiently to redistribute the load without excessive deformation.

NOTE:
(1) The operating load parameters do not necessarily represent single quantities, but rather each operating load parameter may represent multiple operating conditions as specified in the User's Design Specification [see 2.2.1.1(e) and 2.2.1.1(f)].

(21)

Table 5.3
Load Case Combinations and Allowable Stresses for an Elastic Analysis

Load Combinations [Note (1)]	Stress Assessment and Allowable Stress [Note (2)]: General Primary Membrane P_m	Local Primary Membrane P_L	Local Primary Membrane Plus Bending $P_L + P_b$	Range of Primary Plus Secondary $P_L + P_b + Q$	Range of Primary Plus Secondary Plus Peak $P_L + P_b + Q + F$
Design					
(1) $P + P_s + D$ (2) $P + P_s + D + L$ (3) $P + P_s + D + L + T$ (4) $P + P_s + D + S_s$ (5) $0.6D$ + ($0.6W$ or $0.7E$) [Note (3)] (6) $\Omega_p P + P_s + D$ + ($0.6W$ or $0.7E$) [Note (4)] (7) $\Omega_p P + P_s + D + 0.75(L + T) + 0.75S_s$ [Note (4)] (8) $\Omega_p P + P_s + D + 0.75(0.6W$ or $0.7E) + 0.75L + 0.75S_s$ [Note (4)] (9) Other design load combinations as defined in the User's Design Specification	S	S_{PL}	S_{PL}	Not applicable	
Operating					
(10) $P_o + P_{So} + D_o$ (11) $P_o + P_{So} + D_o + L_o$ (12) $P_o + P_{So} + D_o + L_o + T_o$ (13) $P_o + P_{So} + D_o + S_s$ (14) $P_o + P_{So} + D_o$ + ($0.6W$ or $0.7E$) (15) $P_o + P_{So} + D_o + 0.75(L_o + T_o) + 0.75S_s$ (16) $P_o + P_{So} + D_o + 0.75(0.6W$ or $0.7E) + 0.75L_o + 0.75S_s$ (17) Other operating load combinations as defined in the User's Design Specification	Not applicable			S_{PS}	$2S_a$ (see 5.5.1.4)
Pressure Test					
(18) $P_T + P_S + D + 0.6W_{pt}$	See 5.2.2.5	Not applicable	See 5.2.2.5	Not applicable	

GENERAL NOTE: Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

NOTES:
(1) The parameters used in the Load Combinations column are defined in Table 5.2.
(2) See Figure 5.1 for additional guidance on stress categories and limits on equivalent stress.
(3) This load combination addresses an overturning condition for foundation design. It does not apply to design of anchorage (if any) to the foundation. Refer to ASCE/SEI 7, 2.4.1, Exception 2 for an additional reduction to W that may be applicable.
(4) The product of $\Omega_p P$ is used in lieu of the design pressure, P, for evaluation of P_m, P_L, and $P_L + P_b$ stress assessments since it is unlikely the occasional loads would occur at the same time as the maximum design pressure, P.

Table 5.4
Load Case Combinations and Load Factors for a Limit-Load Analysis

Criteria	Required Factored Load Combinations
Design Conditions	
Global	*(1)* $1.5(P + P_s + D)$ *(2)* $1.3(P + P_s + D + T) + 1.7L + 0.54S_s$ *(3)* $1.3(P + P_s + D) + 1.7S_s + (1.1L \text{ or } 0.54W)$ *(4)* $1.3(P + P_s + D) + 1.1W + 1.1L + 0.54S_s$ *(5)* $1.3(P + P_s + D) + 1.1E + 1.1L + 0.21S_s)$
Local	See 5.3.1.2
Serviceability	Per User's Design Specification, if applicable; see Table 5.5
Test Condition	
Global	$\frac{1}{\beta_T}\left(P_T + P_S + D + 0.6W_{pt}\right)$
Serviceability	Per User's Design Specification, if applicable

GENERAL NOTES:
(a) The parameters used in the Design Load Combination column are defined in Table 5.2.
(b) See 5.2.3.4 for descriptions of global and serviceability criteria.
(c) S is the allowable membrane stress at the design temperature.
(d) S_T is the allowable membrane stress at the pressure test temperature.
(e) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Table 5.5
Load Case Combinations and Load Factors for an Elastic–Plastic Analysis

Criteria	Required Factored Load Combinations
Design Conditions	
Global	*(1)* $\beta(P + P_s + D)$ *(2)* $0.88\beta(P + P_S + D + T) + 1.13\beta L + 0.36\beta S_S$ *(3)* $0.88\beta(P + P_S + D) + 1.13\beta S_s + 0.71\beta L$ or $0.36\beta W$ *(4)* $0.88\beta(P + P_S + D) + 0.71\beta W + 0.71\beta L + 0.36\beta S_S$ *(5)* $0.88\beta(P + P_S + D) + 0.71\beta E + 0.71\beta L + 0.14\beta S_S$
Local	$1.7\left(P + P_s + D\right)$
Serviceability	Per User's Design Specification, if applicable; see 5.2.4.3(b)
Test Condition	
Global	$\frac{\beta}{1.5} \times \frac{1}{\beta_T}\left(P_T + P_S + D + 0.6W_{pt}\right)$
Serviceability	Per User's Design Specification, if applicable

GENERAL NOTES:
(a) The parameters used in the Design Load Combination column are defined in Table 5.2.
(b) See 5.2.4.3 for descriptions of global and serviceability criteria.
(c) S is the allowable membrane stress at the design temperature.
(d) S_T is the allowable membrane stress at the pressure test temperature.
(e) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Table 5.6
Examples of Stress Classification

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Any shell including cylinders, cones, spheres, and formed heads	Shell plate remote from discontinuities	Internal pressure	General membrane	P_m
			Gradient through plate thickness	Q
		Axial thermal gradient	Membrane	Q
			Bending	
	Near nozzle or other opening	Net-section axial force and/ or bending moment applied to the nozzle, and/or internal pressure	Local membrane	P_L
			Bending	Q
			Peak (fillet or corner)	F
	Any location	Temperature difference between shell and head	Membrane	Q
			Bending	
	Shell distortions such as out-of-roundness and dents	Internal pressure	Membrane	P_m
			Bending	Q
Cylindrical or conical shell	Any section across entire vessel	Net-section axial force, bending moment applied to the cylinder or cone, and/or internal pressure	Membrane stress averaged through the thickness, remote from discontinuities; stress component perpendicular to cross section	P_m
			Bending stress through the thickness; stress component perpendicular to cross section	P_b
	Junction with head or flange	Internal pressure	Membrane	P_L
			Bending	Q
Dished head or conical head	Crown	Internal pressure	Membrane	P_m
			Bending	P_b
	Knuckle or junction to shell	Internal pressure	Membrane	P_L [Note (1)]
			Bending	Q
Flat head	Center region	Internal pressure	Membrane	P_m
			Bending	P_b
	Junction to shell	Internal pressure	Membrane	P_L
			Bending	Q [Note (2)]
Perforated head or shell	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section)	P_m
			Bending (averaged through width of ligament., but gradient through plate)	P_b
			Peak	F
	Isolated or atypical ligament	Pressure	Membrane	Q
			Bending	F
			Peak	

Table 5.6
Examples of Stress Classification (Cont'd)

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Nozzle (see 5.6)	Within the limits of reinforcement given by 4.5	Pressure and external loads and moments, including those attributable to restrained free end displacements of attached piping	General membrane	P_m
			Bending (other than gross structural discontinuity stresses) averaged through nozzle thickness	
	Outside the limits of reinforcement given by 4.5	Pressure and external axial, shear, and torsional loads, including those attributable to restrained free end displacements of attached piping	General membrane	P_m
		Pressure and external loads and moments, excluding those attributable to restrained free end displacements of attached piping	Membrane	P_L
			Bending	P_b
		Pressure and all external loads and moments	Membrane	P_L
			Bending	Q
			Peak	F
	Nozzle wall	Gross structural discontinuities	Membrane	P_L
			Bending	Q
			Peak	F
		Differential expansion	Membrane	Q
			Bending	
			Peak	F
Cladding	Any	Differential expansion	Membrane	F
			Bending	
Any	Any	Radial temperature distribution [Note (3)]	Equivalent linear stress [Note (4)]	Q
			Nonlinear portion of stress distribution	F
Any	Any	Any	Stress concentration (notch effect)	F

NOTES:
(1) Consideration shall be given to the possibility of wrinkling and excessive deformation in vessels with large diameter-to-thickness ratio.
(2) If the bending moment at the edge is required to maintain the bending stress in the center region within acceptable limits, the edge bending is classified as P_b; otherwise, it is classified as Q.
(3) Consider possibility of thermal stress ratchet.
(4) Equivalent linear stress is defined as the linear stress distribution that has the same net bending moment as the actual stress distribution.

Table 5.7
Uniaxial Strain Limit for Use in Multiaxial Strain Limit Criterion

Material	Maximum Temperature	Uniaxial Strain Limit, ε_{Lu} [Note (1)], [Note (2)], [Note (3)]			α_{sl}
		m_2	Elongation Specified	Reduction of Area Specified	
Ferritic steel	480°C (900°F)	$0.60(1.00 - R)$	$2 \cdot ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	2.2
Stainless steel and nickel base alloys	480°C (900°F)	$0.75(1.00 - R)$	$3 \cdot ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	0.6
Duplex stainless steel	480°C (900°F)	$0.70(0.95 - R)$	$2 \cdot ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	2.2
Precipitation hardenable nickel base alloys	540°C (1,000°F)	$1.09(0.93 - R)$	$ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	2.2
Aluminum	120°C (250°F)	$0.52(0.98 - R)$	$1.3 \cdot ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	2.2
Copper	65°C (150°F)	$0.50(1.00 - R)$	$2 \cdot ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	2.2
Titanium and zirconium	260°C (500°F)	$0.50(0.98 - R)$	$1.3 \cdot ln\left[1 + \frac{E}{100}\right]$	$ln\left[\frac{100}{100 - RA}\right]$	2.2

NOTES:
(1) If the elongation and reduction in area are not specified, then $\varepsilon_{Lu} = m_2$. If the elongation or reduction in area is specified, then ε_{Lu} is the maximum number computed from columns 3, 4, or 5, as applicable.
(2) R is the ratio of the minimum specified yield strength divided by the minimum specified ultimate tensile strength.
(3) E is the % elongation and RA is the % reduction in area determined from the applicable material specification.

Table 5.8
Temperature Factors for Fatigue-Screening Criteria

Metal Temperature Differential		
°C	°F	Temperature Factor for Fatigue-Screening Criteria
28 or less	50 or less	0
29 to 56	51 to 100	1
57 to 83	101 to 150	2
84 to 139	151 to 250	4
140 to 194	251 to 350	8
195 to 250	351 to 450	12
Greater than 250	Greater than 450	20

GENERAL NOTES:
(a) If the weld metal temperature differential is unknown or cannot be established, a value of 20 shall be used.
(b) As an example illustrating the use of this table, consider a component subject to metal temperature differentials for the following number of thermal cycles:

Temperature Differential	Temperature Factor Based on Temperature Differential	Number of Thermal Cycles
28°C (50°F)	0	1,000
50°C (90°F)	1	250
222°C (400°F)	12	5

The effective number of thermal cycles due to changes in metal temperature is: $N_{\Delta TE} = 1{,}000(0) + 250(1) + 5(12) = 310$ cycles

Table 5.9
Fatigue-Screening Criteria for Method A

Type of Construction	Component Description	Fatigue-Screening Criteria
Integral construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 350$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 1{,}000$
Nonintegral construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 60$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 400$

Table 5.10
Fatigue-Screening Criteria Factors for Method B

Type of Construction	Component Description	Fatigue-Screening Criteria Factor	
		C_1	C_2
Integral construction	Attachments and nozzles in the knuckle region of formed heads	4	2.7
	All other components	3	2
Nonintegral construction	Attachments and nozzles in the knuckle region of formed heads	5.3	3.6
	All other components	4	2.7

Table 5.11
Weld Surface Fatigue-Strength-Reduction Factors

Weld Condition	Surface Condition	Quality Levels (See Table 5.12)						
		1	2	3	4	5	6	7
Full penetration	Machined	1.0	1.5	1.5	2.0	2.5	3.0	4.0
	As-welded	1.2	1.6	1.7	2.0	2.5	3.0	4.0
Partial penetration	Final surface machined	NA	1.5	1.5	2.0	2.5	3.0	4.0
	Final surface as-welded	NA	1.6	1.7	2.0	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0
Fillet	Toe machined	NA	NA	1.5	NA	2.5	3.0	4.0
	Toe as-welded	NA	NA	1.7	NA	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0

Table 5.12
Weld Surface Fatigue-Strength-Reduction Factors

Fatigue-Strength-Reduction Factor	Quality Level	Definition
1.0	1	Machined or ground weld that receives a full volumetric examination, and a surface that receives MT/PT examination and a VT examination
1.2	1	As-welded weld that receives a full volumetric examination, and a surface that receives MT/PT and VT examination
1.5	2	Machined or ground weld that receives a partial volumetric examination, and a surface that receives MT/PT examination and VT examination
1.6	2	As-welded weld that receives a partial volumetric examination, and a surface that receives MT/PT and VT examination
1.5	3	Machined or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
1.7	3	As-welded surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
2.0	4	Weld has received a partial or full volumetric examination, and the surface has received VT examination, but no MT/PT examination
2.5	5	VT examination only of the surface; no volumetric examination nor MT/PT examination
3.0	6	Volumetric examination only
4.0	7	Weld backsides that are nondefinable and/or receive no examination

GENERAL NOTES:
(a) Volumetric examination is RT or UT in accordance with Part 7.
(b) MT/PT examination is magnetic particle or liquid penetrant examination in accordance with Part 7.
(c) VT examination is visual examination in accordance with Part 7.
(d) See WRC Bulletin 432 for further information.

Table 5.13
Fatigue Penalty Factors for Fatigue Analysis

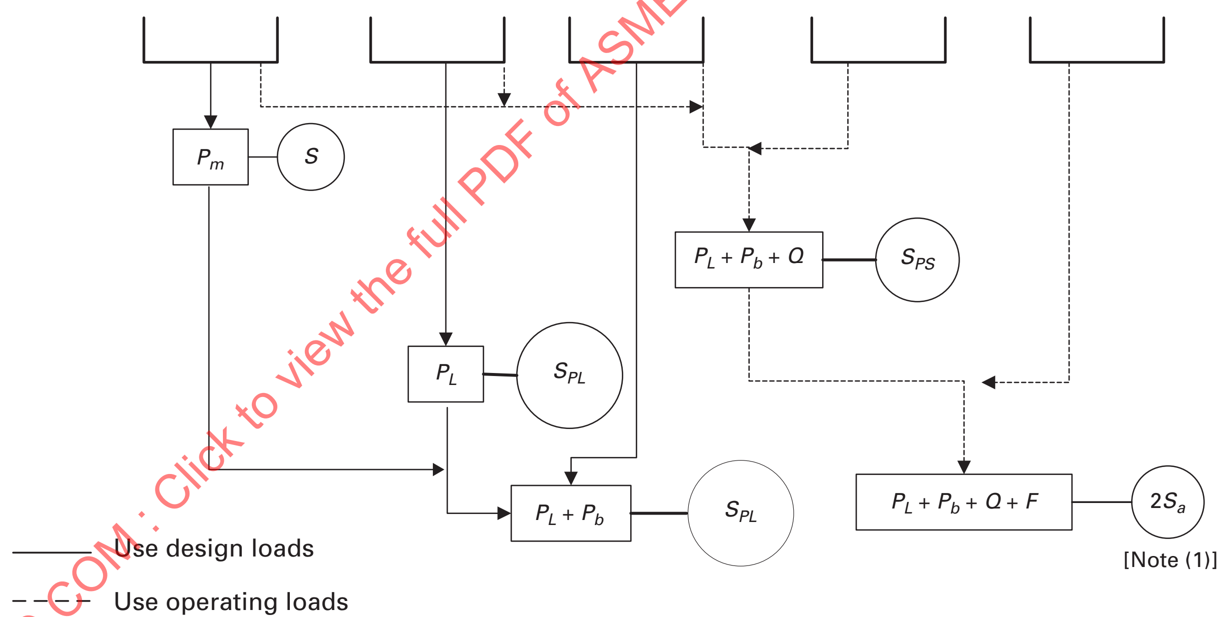
	K_e [Note (1)]		T_{max} [Note (2)]	
Material	m	n	°C	°F
Low alloy steel	2.0	0.2	371	700
Martensitic stainless steel	2.0	0.2	371	700
Carbon steel	3.0	0.2	371	700
Austenitic stainless steel	1.7	0.3	427	800
Nickel-chromium-iron	1.7	0.3	427	800
Nickel-copper	1.7	0.3	427	800

NOTES:
(1) Fatigue penalty factor.
(2) The fatigue penalty factor should be used only if all of the following are satisfied:
- The component is not subject to thermal ratcheting.
- The maximum temperature in the cycle is within the value in the table for the material.

5.15 FIGURES

Figure 5.1
Stress Categories and Limits of Equivalent Stress

Stress Category	Primary			Secondary Membrane plus Bending	Peak
	General Membrane	Local Membrane	Bending		
Description (For examples, see Table 5.2.)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations.	1. Increment added to primary or secondary stress by a concentration (notch). 2. Certain thermal stresses that may cause fatigue but not distortion of vessel shape.
Symbol	P_m	P_L	P_b	Q	F



NOTE:
(1) See 5.5.3.

Figure 5.2
Example of Girth Weld Used to Tie Layers for Solid Wall Equivalence

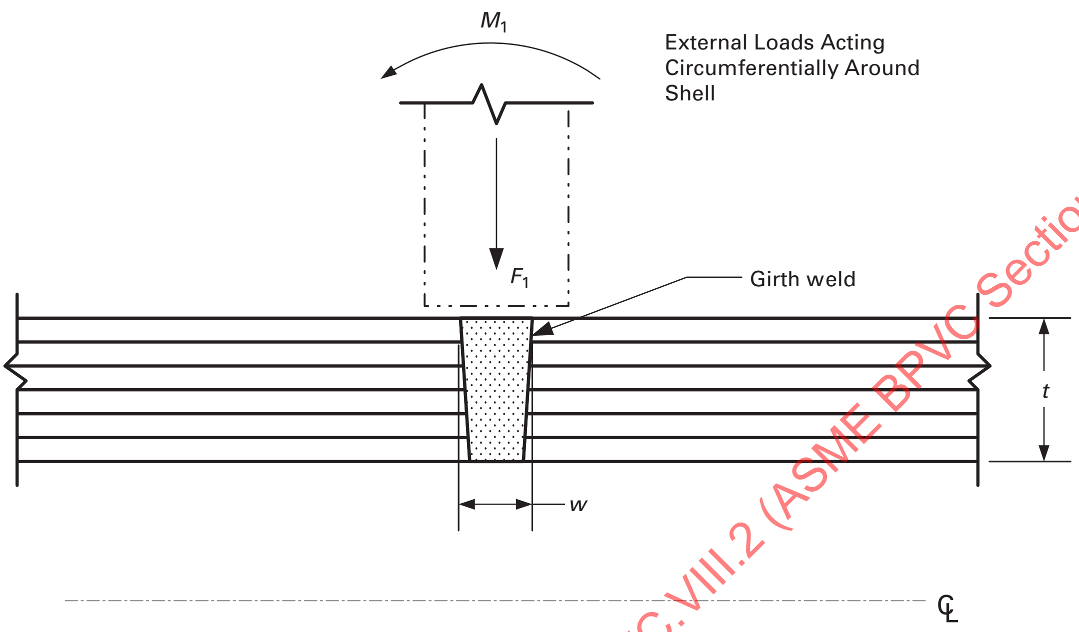


Figure 5.3
Example of Circumferential Butt Weld Attachment Between Layered Sections in Zone of Discontinuity

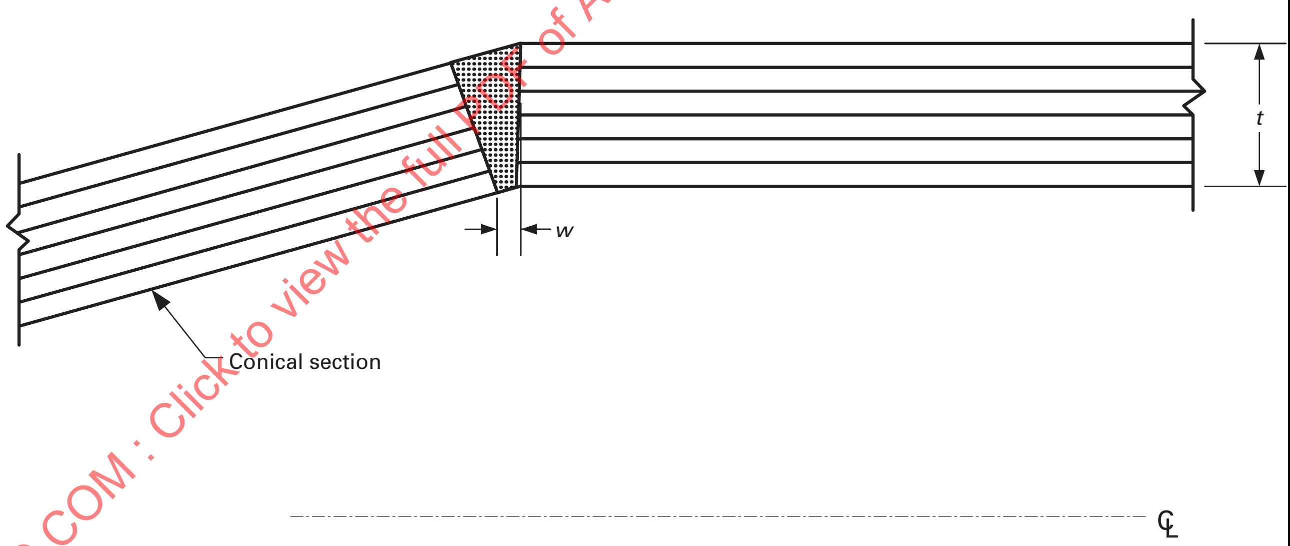
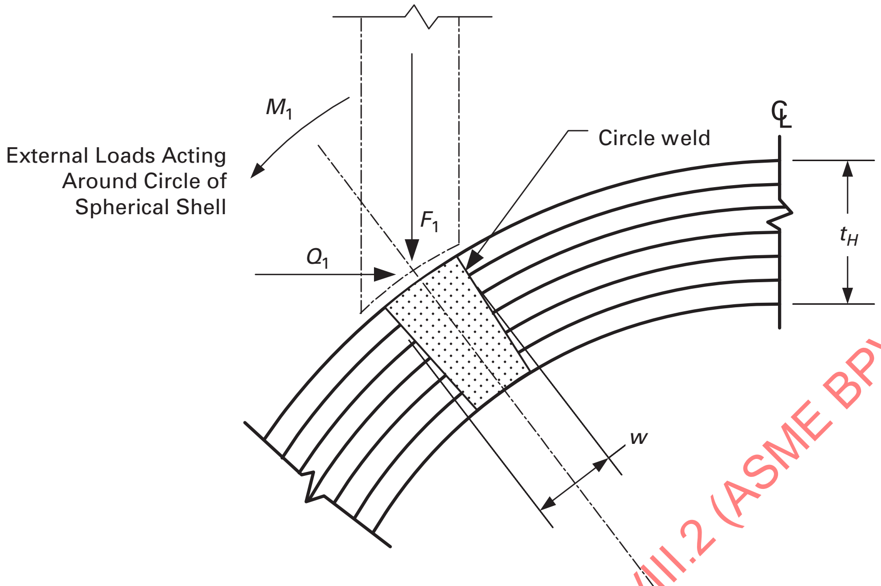


Figure 5.4
An Example of Circle Weld Used to Tie Layers for Solid Wall Equivalence



ANNEX 5-A
LINEARIZATION OF STRESS RESULTS FOR STRESS CLASSIFICATION

(Informative)

5-A.1 SCOPE

This Annex provides recommendations for post-processing of the results from an elastic finite element stress analysis for comparison to the limits in 5.2.2.

5-A.2 GENERAL

(a) In the finite element method, when continuum elements are used in an analysis, the total stress distribution is obtained. Therefore, to produce membrane and bending stresses, the total stress distribution shall be linearized on a stress component basis and used to calculate the equivalent stresses. If shell elements (shell theory) are used, then the membrane and bending stresses shall be obtained directly from shell stress resultants.

(b) Membrane and bending stresses are developed on cross sections through the thickness of a component. These sections are called stress classification planes (SCPs). In a planar geometry, a Stress Classification Line (SCL) is obtained by reducing two opposite sides of a SCP to an infinitesimal length. SCPs are flat planes that cut through a section of a component and SCLs are straight lines that cut through a section of a component. SCLs are surfaces when viewed in an axisymmetric or planar geometry. Examples of an SCP and SCL are given in Figure 5-A.1 and Figure 5-A.2.

(c) The following three approaches are provided for linearization of finite element results.

(1) Stress Integration Method – This method can be used to linearize stress results from continuum finite element models [Ref. WRC-429].

(2) Structural Stress Method Based on Nodal Forces – This method is based on processing of nodal forces, and has been shown to be mesh insensitive and correlate well with welded fatigue data [Ref. WRC-474].

(3) Structural Stress Method Based on Stress Integration – This method utilizes the Stress Integration Method, but restricts the set of elements that contribute to the line of nodes being processed.

(d) The Structural Stress Method based on Stress Integration is recommended unless another method can be shown to produce a more accurate assessment for the given component and loading condition. This method matches the Structural Stress Method Based on Nodal Forces, which is insensitive to mesh refinement. In addition, this method can be performed with post-processing tools typically provided by commercial finite element analysis software.

5-A.3 SELECTION OF STRESS CLASSIFICATION LINES

(a) Pressure vessels usually contain structural discontinuity regions where abrupt changes in geometry, material or loading occur. These regions are typically the locations of highest stress in a component. For the evaluation of failure modes of plastic collapse and ratcheting, Stress Classification Lines (SCLs) are typically located at gross structural discontinuities. For the evaluation of local failure and fatigue, SCLs are typically located at local structural discontinuities.

(b) For SCLs that span a material discontinuity (e.g., base metal with cladding), the SCL should include all materials and associated loadings. If one of the materials, such as cladding, is neglected for strength calculations, then only the base metal thickness should be used to calculate the membrane and bending stresses from the linearized forces and moments across the full section for the evaluation of plastic collapse.

(c) To most accurately determine the linearized membrane and bending stresses for comparison to elastic stress limits, the following guidelines should be followed. These guidelines can be used as a qualitative means to evaluate the applicability of different SCLs. Failure to comply with any of these criteria may not produce valid membrane and/or bending stresses. Application of the limit-load or elastic–plastic analysis methods in Part 5 is recommended for cases where elastic stress analysis and stress linearization may produce ambiguous results.

(1) SCLs should be oriented normal to contour lines of the stress component of highest magnitude. However, as this may be difficult to implement, similar accuracy can be obtained by orienting the SCL normal to the mid-surface of the cross section. SCL orientation guidelines are shown in Figure 5-A.3.

(2) Hoop and meridional component stress distributions on the SCL should be monotonically increasing or decreasing, except for the effects of stress concentration or thermal peak stresses, see Figure 5-A.3(b).

(3) The distribution of the through-thickness stress should be monotonically increasing or decreasing. For pressure loading the through-thickness stress should be equal to the compressive pressure on the applied surface, and approximately zero on the other surface defining the SCL (see Figure 5-A.3(c)). When the SCL is not perpendicular to the surfaces, this requirement will not be satisfied.

(4) The shear stress distribution should be parabolic and/or the stress should be low relative to the hoop and meridional stresses. Depending on the type of loading, the shear stress should be approximately zero on both surfaces defined by the SCL. Guidelines are provided in Figure 5-A.3(d).

(-a) The shear stress distribution along an SCL will approximate a parabolic distribution only when the inner and outer surfaces are parallel and the SCL is normal to the surfaces. If the surfaces are not parallel or an SCL is not normal to the surfaces, the appropriate shear distribution will not be obtained. However, if the magnitude of shear stress is small as compared to the hoop or meridional stresses, this orientation criterion can be waived.

(-b) When the shear stress distribution is approximately linear, the shear stress is likely to be significant.

(5) For pressure boundary components, the hoop or meridional stresses typically are the largest magnitude component stresses and are the dominant terms in the equivalent stress. Typically the hoop or meridional stresses deviate from a monotonically increasing or decreasing trend along an SCL if the SCL is skewed with respect to the interior, exterior, or mid surfaces. For most pressure vessel applications, the hoop or meridional stresses due to pressure should be nearly linear.

5-A.4 STRESS INTEGRATION METHOD

5-A.4.1 CONTINUUM ELEMENTS

5-A.4.1.1 Overview

Stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional continuum elements may be processed using the stress integration method. Stress components are integrated along SCLs through the wall thickness to determine the membrane and bending stress components. The peak stress components can be derived directly using this procedure by subtracting the membrane plus bending stress distribution from the total stress distribution. Using these components, the equivalent stress shall be computed per Eq. (5.1).

5-A.4.1.2 Stress Linearization Procedure

The methods to derive the membrane, bending, and peak components of a stress distribution are shown below, and in Figure 5-A.4. The component stresses used for the calculations shall be based on a local coordinate system defined by the orientation of the SCL, see Figure 5-A.2.

Step 1. Calculate the membrane stress tensor. The membrane stress tensor is the tensor comprised of the average of each stress component along the stress classification line, or:

$$\sigma_{ij,m} = \frac{1}{t}\int_0^t \sigma_{ij}dx \tag{5-A.1}$$

Step 2. Calculate the bending stress tensor.

(a) Bending stresses are calculated only for the local hoop and meridional (normal) component stresses, and not for the local component stress parallel to the SCL or in-plane shear stress.

(b) The linear portion of shear stress needs to be considered only for shear stress distributions that result in torsion of the SCL (out-of-plane shear stress in the normal-hoop plane, see Figure 5-A.2).

(c) The bending stress tensor is comprised of the linear varying portion of each stress component along the stress classification line, or:

$$\sigma_{ij,b} = \frac{6}{t^2}\int_0^t \sigma_{ij}\left(\frac{t}{2} - x\right)dx \tag{5-A.2}$$

Step 3. Calculate the peak stress tensor. The peak stress tensor is the tensor whose components are equal to:

$$\sigma_{ij,F}(x)\Big|_{x=0} = \sigma_{ij}(x)\Big|_{x=0} - \left(\sigma_{ij,m} + \sigma_{ij,b}\right) \tag{5-A.3}$$

$$\sigma_{ij,F}(x)\Big|_{x=t} = \sigma_{ij}(x)\Big|_{x=t} - \left(\sigma_{ij,m} - \sigma_{ij,b}\right) \tag{5-A.4}$$

Step 4. Calculate the three principal stresses at the ends of the SCL based on components of membrane and membrane plus bending stresses.

Step 5. Calculate the equivalent stresses using eq. (5.1) at the ends of the SCL based on components of membrane and membrane plus bending stresses.

5-A.4.2 SHELL ELEMENTS

5-A.4.2.1 Overview

Stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional shells are obtained directly from the analysis results. Using the component stresses, the equivalent stress shall be computed per eq. (5.1).

5-A.4.2.2 Stress Linearization Procedure

The methods to derive the membrane, bending, and peak components of a stress distribution are shown below.

(a) The membrane stress tensor is the tensor comprised of the average of each stress component along the stress classification line, or:

$$\sigma_{ij,m} = \frac{\sigma_{ij,\text{in}} + \sigma_{ij,\text{out}}}{2} \tag{5-A.5}$$

(b) The bending stress tensor is the tensor comprised of the linear varying portion of each stress component along the stress classification line, or:

$$\sigma_{ij,b} = \frac{\sigma_{ij,\text{in}} - \sigma_{ij,\text{out}}}{2} \tag{5-A.6}$$

(c) The peak stress tensor is the tensor whose components are equal to:

$$\sigma_{ij,F} = \left(\sigma_{ij,m} + \sigma_{ij,b}\right)\left(K_f - 1\right) \tag{5-A.7}$$

5-A.5 STRUCTURAL STRESS METHOD BASED ON NODAL FORCES

5-A.5.1 OVERVIEW

Stress results derived from a finite element analysis utilizing continuum or shell elements may be processed using the Structural Stress Method based on nodal forces. The mesh-insensitive structural stress method provides a robust procedure for capturing the membrane and bending stresses and can be directly utilized in fatigue design of welded joints. With this method, the structural stress normal to a hypothetical cracked plane at a weld is evaluated. For typical pressure vessel component welds, the choice of possible crack orientations is straightforward (e.g., toe of fillet weld). Two alternative calculation procedures for the structural stress method are presented for continuum elements; a procedure based on nodal forces and a procedure based on stress integration. A typical finite element continuum model and stress evaluation line for this type of analysis is shown in Figure 5-A.5.

5-A.5.2 CONTINUUM ELEMENTS

(a) Stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional continuum elements may be processed using the structural stress method and nodal forces as described below. The membrane and bending stresses can be computed from element nodal internal forces using the equations provided in Table 5-A.1. The process is illustrated in Figure 5-A.6. This method is recommended when internal force results can be obtained as part of the finite element output because the results are insensitive to the mesh density.

(b) When using three-dimensional continuum elements, forces and moments must be summed with respect to the mid-thickness of a member from the forces at nodes in the solid model at a through-thickness cross section of interest. For a second order element, three summation lines of nodes are processed along the element faces through the wall thickness. The process is illustrated in Figure 5-A.7.

(c) For a symmetric structural stress range, the two weld toes have equal opportunity to develop fatigue cracks. Therefore, the structural stress calculation involves establishing the equilibrium equivalent membrane and bending stress components with respect to one-half of the plate thickness. The equivalent structural stress calculation procedure for a symmetric stress state is illustrated in Figure 5-A.8.

5-A.5.3 SHELL ELEMENTS

(a) Stress results derived from a finite element analysis utilizing shell elements may be processed using the structural stress method and nodal forces. The membrane and bending stresses can be computed from element nodal internal forces using the equations provided in Table 5-A.2. A typical shell model is illustrated in Figure 5-A.9.

(b) When using three-dimensional shell elements, forces and moments with respect to the mid-thickness of a member must be obtained at a cross section of interest. The process is illustrated in Figure 5-A.10.

5-A.6 STRUCTURAL STRESS METHOD BASED ON STRESS INTEGRATION

As an alternative to the nodal force method above, stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional continuum elements may be processed using the Structural Stress Method Based on Stress Integration. This method utilizes the Stress Integration Method of 5-A.3, but restricts the set of elements that contribute to the line of nodes being processed. The elements applicable to the SCL for the region being evaluated shall be included in the post-processing, as is illustrated in Figure 5-A.11.

5-A.7 NOMENCLATURE

σ_s = structural stress
$\Delta\sigma_s$ = structural stress range.
f_i = line force at element location position i.
NF_j = nodal force at node j, normal to the section.
NF_{ij} = nodal force at node j, normal to the section, for element location position i.
NM_j = in-plane nodal moment at node j, normal to the section, for a shell element.
F_i = nodal force resultant for element location position i.
K_f = fatigue strength reduction factor used to compute the cyclic stress amplitude or range.
m_i = line moment at element location position i.
n = number of nodes in the through-wall thickness direction.
M_i = nodal moment resultant for element location position i.
σ_m = membrane stress.

σ_b = bending stress.
σ_{ij} = stress tensor at the point under evaluation.
$\sigma_{ij,m}$ = membrane stress tensor at the point under evaluation.
$\sigma_{ij,b}$ = bending stress tensor at the point under evaluation.
$\sigma_{ij,F}$ = peak stress component.
$\sigma_{ij,\text{in}}$ = stress tensor on the inside surface of the shell.
$\sigma_{ij,\text{out}}$ = stress tensor on the outside surface of the shell.
σ_{mi} = membrane stress for element location position i.
σ_{bi} = bending stress for element location position i.
r_j = radial coordinate of node j for an axisymmetric element.
s_j = local coordinate, parallel to the stress classification line, that defines the location of nodal force 'NF_j' relative to the mid-thickness of the section.
P = primary equivalent stress.
Q = secondary equivalent stress.
X_L = local X axis, oriented parallel to the stress classification line.
Y_L = local Y axis, oriented normal to the stress classification line.
X_g = global X axis.
Y_g = global Y axis.
t = minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable.
w = width of the element to determine structural stresses from Finite Element Analysis.
x = through-wall thickness coordinate.

5-A.8 TABLES

Table 5-A.1
Structural Stress Definitions for Continuum Finite Elements

Element Type	Membrane Stress	Bending Stress
Two-dimensional axisymmetric second order (8-node) continuum elements	$\sigma_m = \frac{1}{t}\Sigma\frac{NF_j}{2\pi r_j}$	$\sigma_b = \frac{6}{t^2}\Sigma\frac{NF_j \cdot S_j}{2\pi r_j}$
Two-dimensional second order plane stress or plane strain (8-node) continuum elements	$\sigma_m = \frac{1}{t}\Sigma\frac{NF_j}{w}$	$\sigma_b = \frac{6}{t^2}\Sigma\frac{NF_j \cdot S_j}{w}$
Three-dimensional second order (20-node) continuum elements	$\sigma_{mi} = \frac{f_i}{t}$ [Note (1)]	$\sigma_{bi} = \frac{6 \cdot m_i}{t^2}$ [Note (2)]

NOTES:
(1) f_i represents the line force corresponding to the element location positions (i = 1, 2, 3) along the element width, w; position i = 2 corresponds to the midside of the element (see Figure 5-A.7).

$$f_1 = \frac{3(6F_1 + 2F_3 - F_2)}{2w}$$

$$f_2 = \frac{-3(2F_1 + 2F_3 - 3F_2)}{4w}$$

$$f_3 = \frac{3(2F_1 + 6F_3 - F_2)}{2w}$$

In the above, F_1, F_2, and F_3 are the nodal force resultants (producing normal membrane stress to Section A-A) through the thickness and along the width, w, of the group of elements $F_i = \Sigma NF_{ij}$ summed over the nodes from j = 1, n (number of nodes in the through-thickness direction) at Section A-A (see Figure 5-A.7).

(2) m_i represents the line moment corresponding to the element location positions (i = 1, 2, 3) along the element width, w; position i = 2 corresponds to the midside of the element (see Figure 5-A.7).

$$m_1 = \frac{3(6M_1 + 2M_3 - M_2)}{2w}$$

$$m_2 = \frac{-3(2M_1 + 2M_3 - 3M_2)}{4w}$$

$$m_3 = \frac{3(2M_1 + 6M_3 - M_2)}{2w}$$

In the above, M_1, M_2, and M_3 are the nodal moment resultants (producing normal bending stress to Section A-A) calculated based on nodal forces with respect to the mid-thickness S_j along the width, w, of the group of elements $M_i = \Sigma NF_{ij} \cdot S_j$ summed over the nodes from j = 1, n (number of nodes in the through-thickness direction) at Section A-A (see Figure 5-A.7).

Table 5-A.2
Structural Stress Definitions for Shell or Plate Finite Elements

Element Type	Membrane Stress	Bending Stress
Three-dimensional second order (8-node) shell elements	$\sigma_{mi} = \frac{f_i}{t}$ [Note (1)]	$\sigma_{bi} = \frac{6 \cdot m_i}{t^2}$ [Note (2)]
Three-dimensional first order (4-node) shell elements	$\sigma_{mi} = \frac{f_i}{t}$ [Note (3)]	$\sigma_{bi} = \frac{6 \cdot m_i}{t^2}$ [Note (4)]
Axisymmetric linear and parabolic shell finite element	$\sigma_m = \frac{NF_j}{2\pi r_j t}$	$\sigma_b = \frac{6 \cdot NM_j}{2\pi r_j t^2}$

NOTES:

(1) f_i represents the force corresponding to the element location positions (i = 1, 2, 3) along the element width, w; position i = 2 corresponds to the midside of the element (see Figure 5-A.10).

$$f_1 = \frac{3\left(6NF_1 + 2NF_3 - NF_2\right)}{2w}$$

$$f_2 = \frac{-3\left(2NF_1 + 2NF_3 - 3NF_2\right)}{4w}$$

$$f_3 = \frac{3\left(2NF_1 + 6NF_3 - NF_2\right)}{2w}$$

In the above, NF_1, NF_2, and NF_3 are the internal nodal forces (in the direction normal to Section A-A) from the shell model along a weld (see Figure 5-A.10).

(2) m_i represents the moment corresponding to the element location positions (i = 1, 2, 3) along the element width, w; position i = 2 corresponds to the midside of the element (see Figure 5-A.10).

$$m_1 = \frac{3\left(6NM_1 + 2NM_3 - NM_2\right)}{2w}$$

$$m_2 = \frac{-3\left(2NM_1 + 2NM_3 - 3NM_2\right)}{4w}$$

$$m_3 = \frac{3\left(2NM_1 + 6NM_3 - NM_2\right)}{2w}$$

In the above, NM_1, NM_2, and NM_3 are the internal nodal moments (producing normal bending stresses to Section A-A) from the shell model along a weld (see Figure 5-A.10).

(3) f_i represents the force corresponding to the element corner node location positions (i = 1, 2) along the element width, w.

$$f_1 = \frac{2\left(2NF_1 - NF_2\right)}{w}$$

$$f_2 = \frac{2\left(2NF_2 - NF_1\right)}{w}$$

(4) m_i represents the moment corresponding to the element corner node location positions (i 0= 1, 2) along the element width, w.

$$m_1 = \frac{2\left(2NM_1 - NM_2\right)}{w}$$

$$m_2 = \frac{2\left(2NM_2 - NM_1\right)}{w}$$

5-A.9 FIGURES

Figure 5-A.1
Stress Classification Line (SCL) and Stress Classification Plane (SCP)

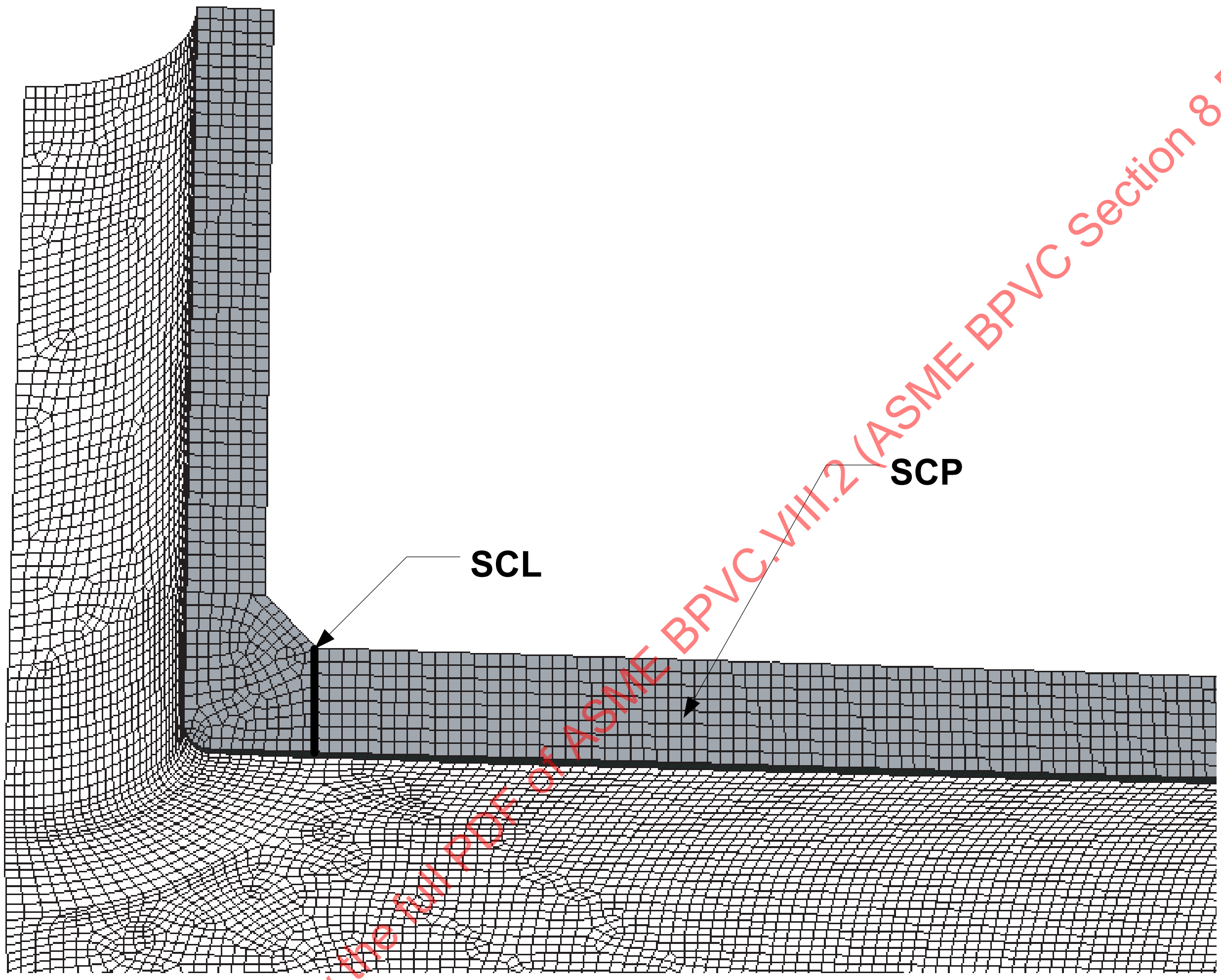
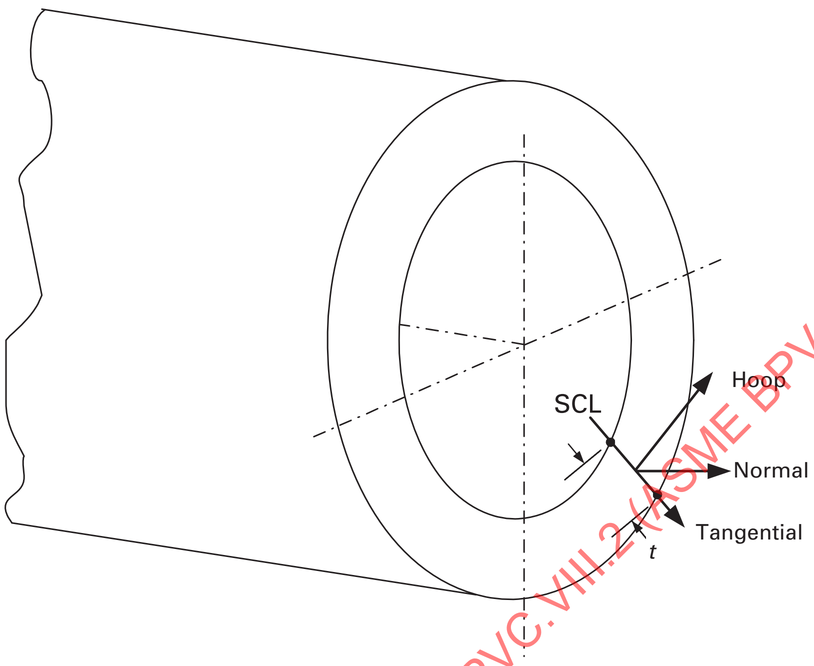
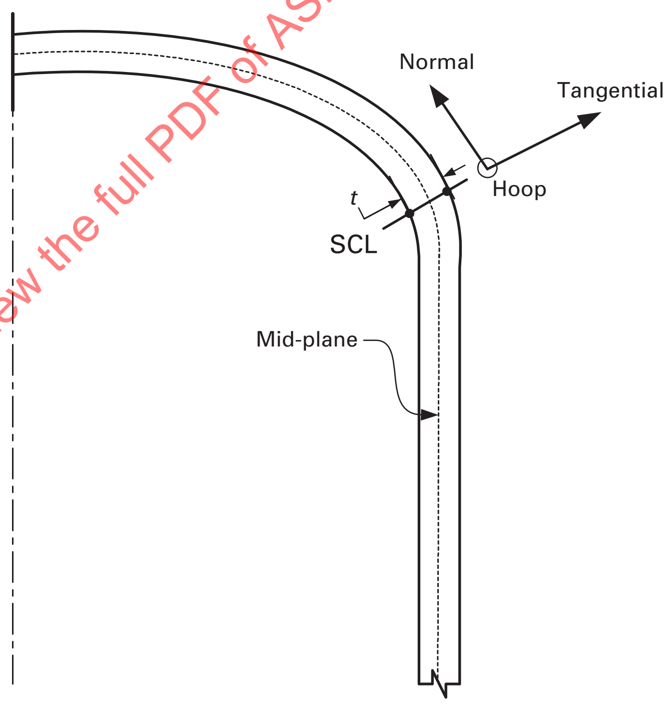


Figure 5-A.2
Stress Classification Lines (SCLs)

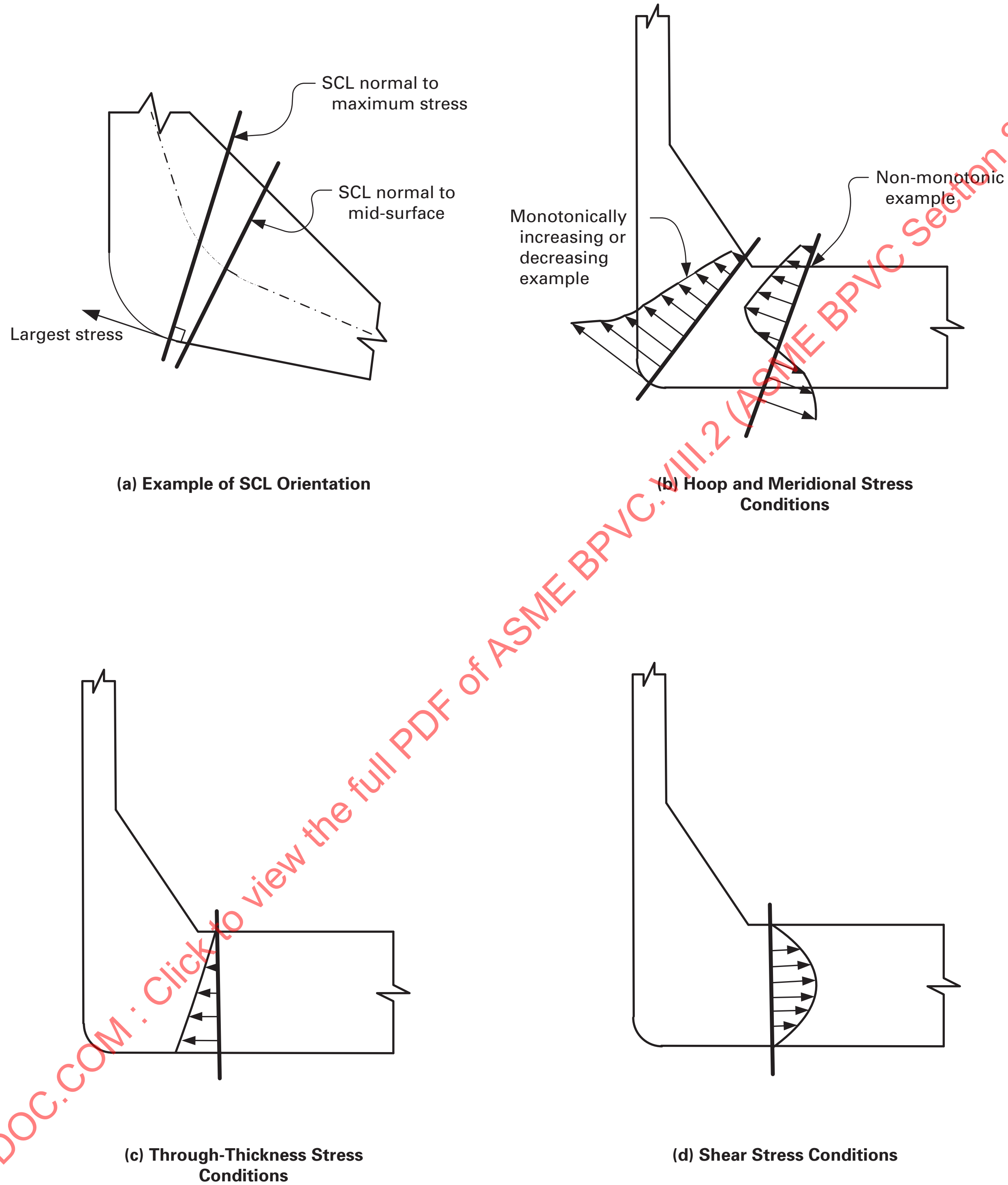


(a) SCL Orientation, Three-Dimensional Model



(b) SCL Orientation, Two-Dimensional Model

Figure 5-A.3
Stress Classification Line Orientation and Validity Guidelines



(a) Example of SCL Orientation

(b) Hoop and Meridional Stress Conditions

(c) Through-Thickness Stress Conditions

(d) Shear Stress Conditions

Figure 5-A.4
Computation of Membrane and Bending Equivalent Stresses by the Stress Integration Method Using the Results From a Finite Element Model With Continuum Elements

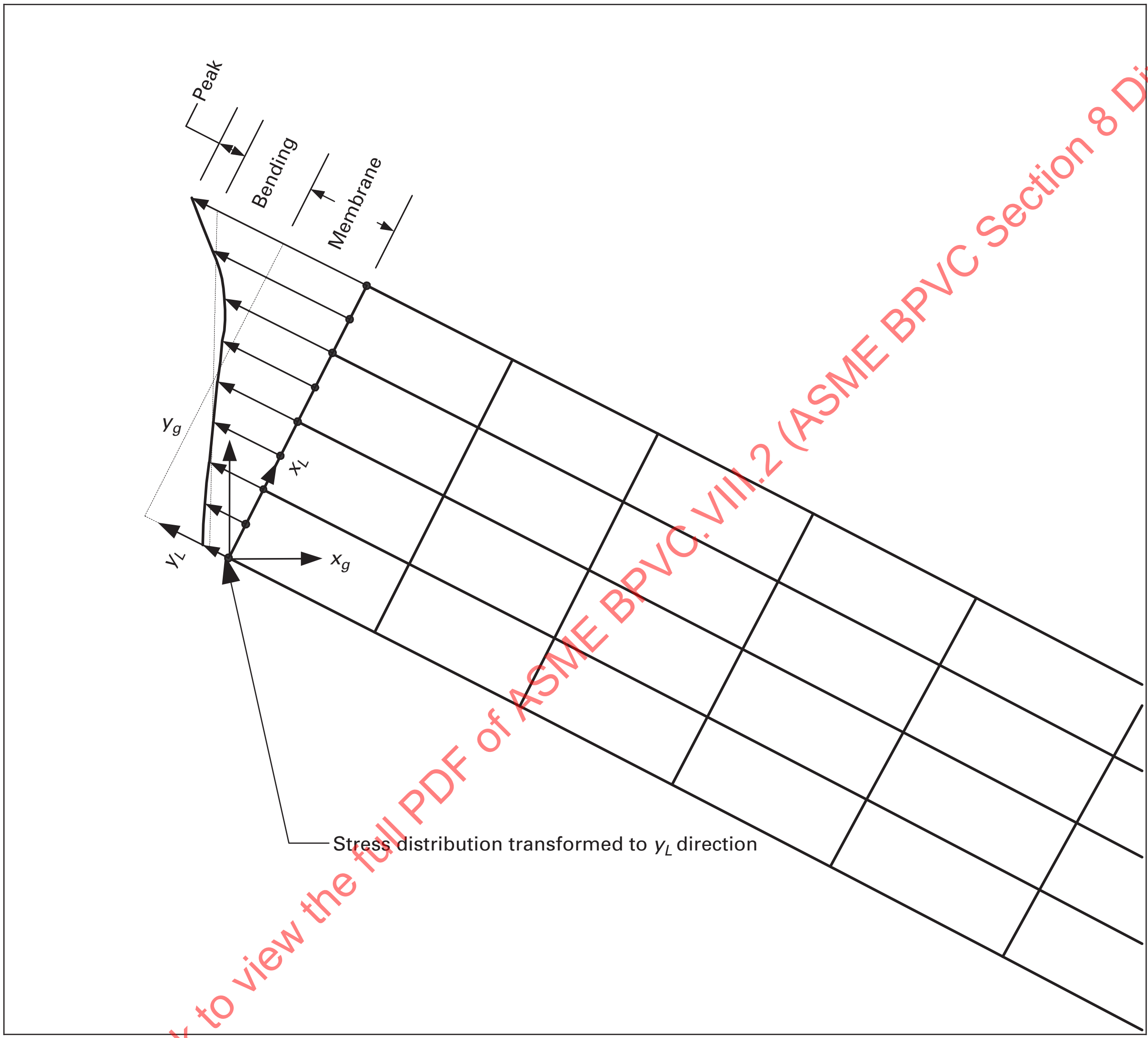
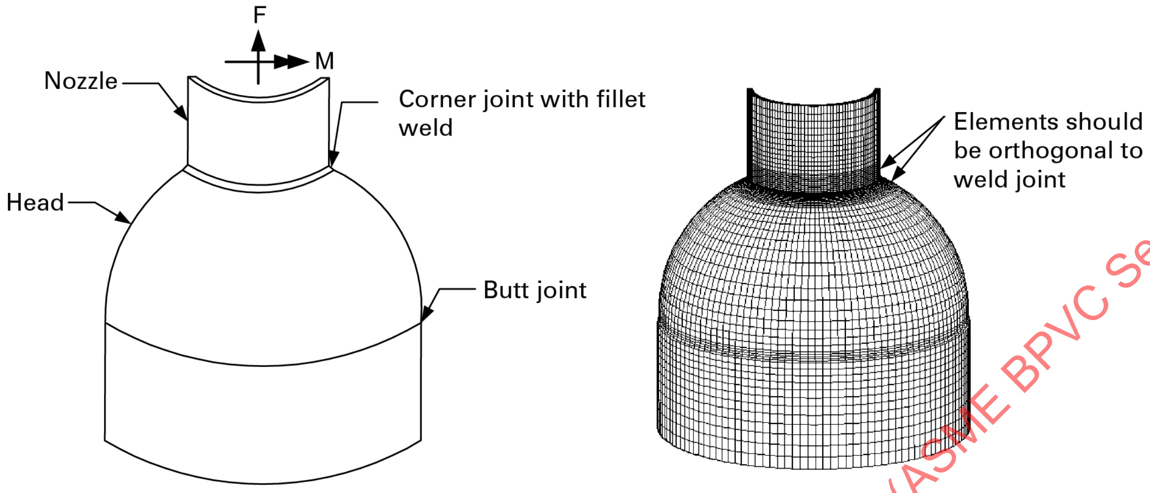
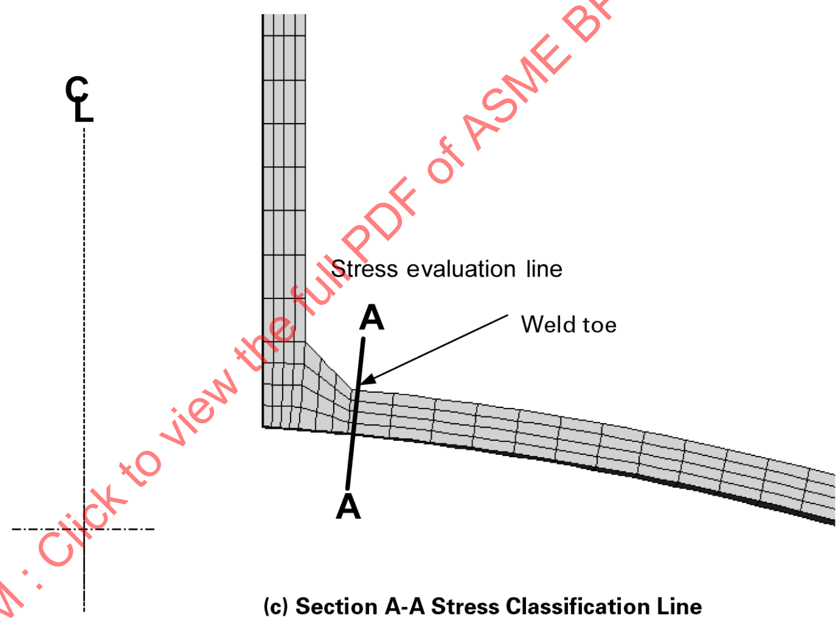


Figure 5-A.5
Continuum Finite Element Model Stress Classification Line for the Structural Stress Method



(a) Component Geometry **(b) Finite Elemental Mesh**



(c) Section A-A Stress Classification Line

Figure 5-A.6
Computation of Membrane and Bending Equivalent Stresses by the Structural Stress Method Using Nodal Force Results From a Finite Element Model With Continuum Elements

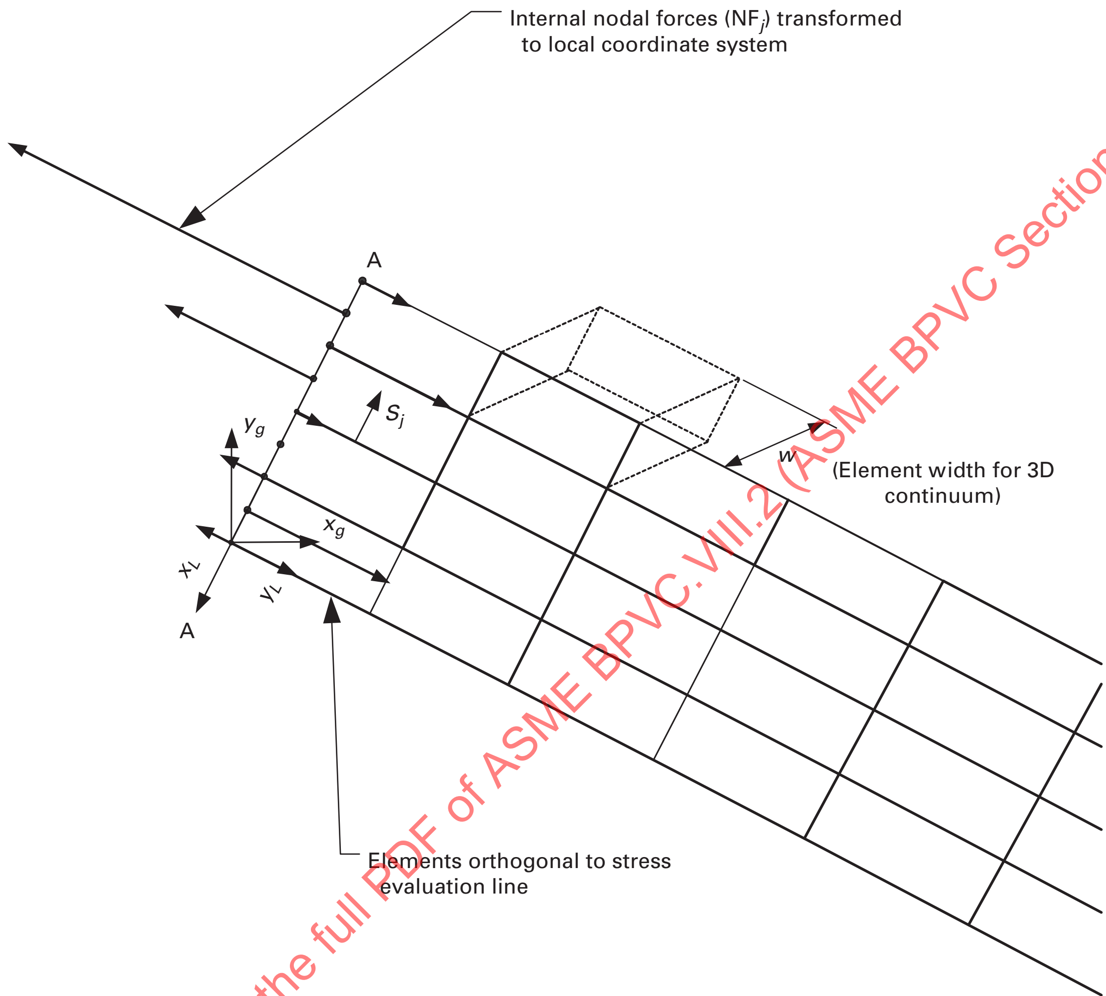


Figure 5-A.7
Processing Nodal Force Results With the Structural Stress Method Using the Results From a Finite Element Model With Three-Dimensional Second Order Continuum Elements

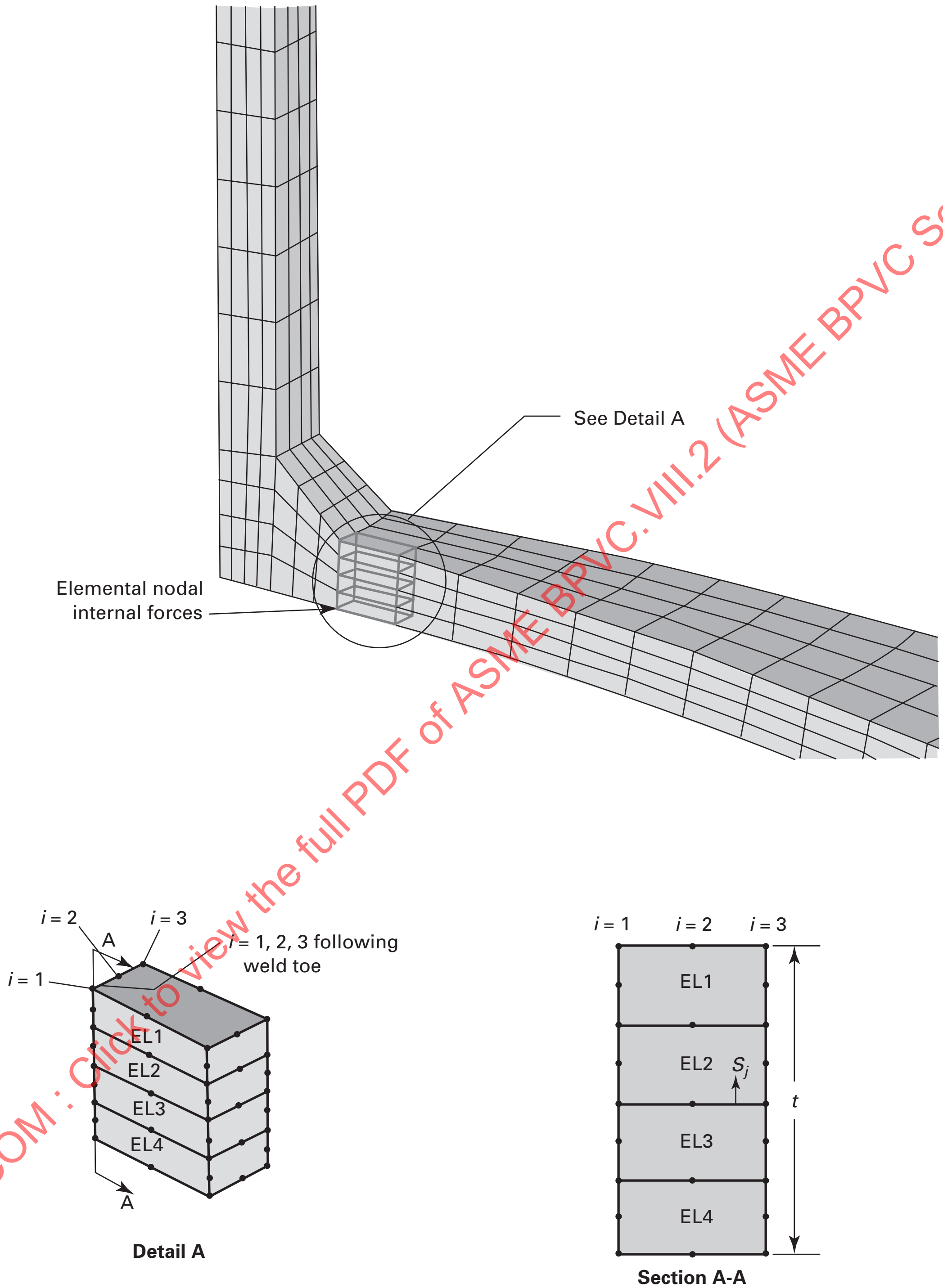
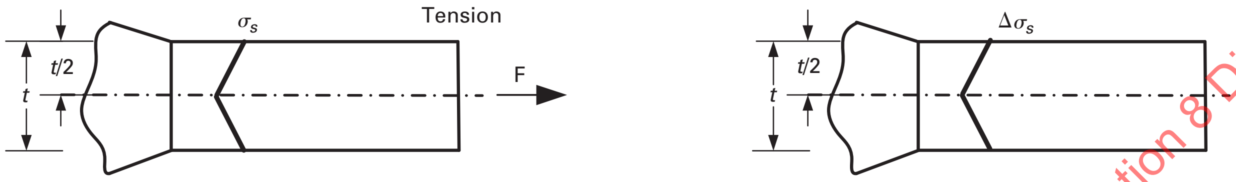
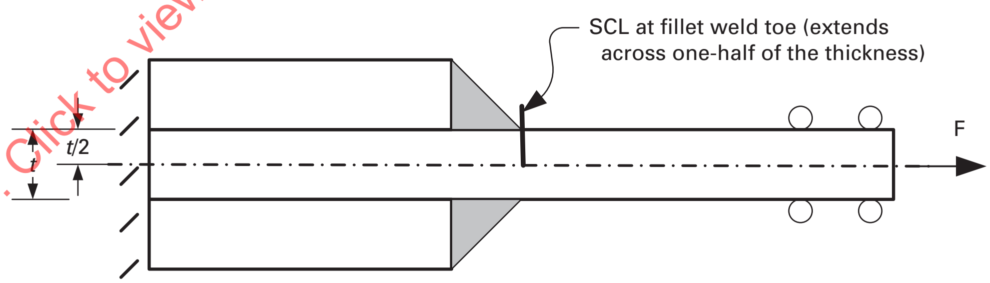


Figure 5-A.8
Processing Structural Stress Method Results for a Symmetric Structural Stress Range



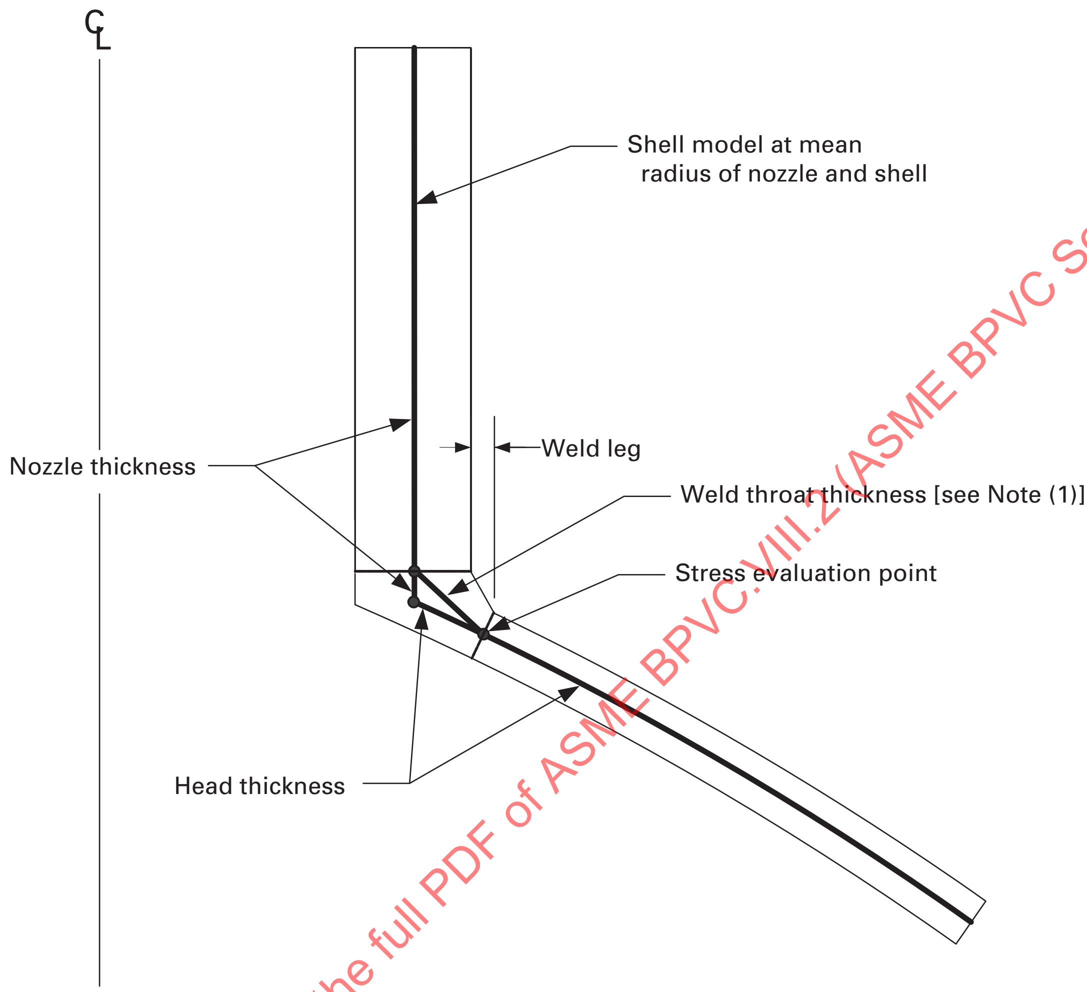
(a) Symmetric Structural Stress State (Symmetric Joint and Symmetric Loading)

(b) Antisymmetric Structural Stress State (Symmetric Joint and Antisymmetric Loading)



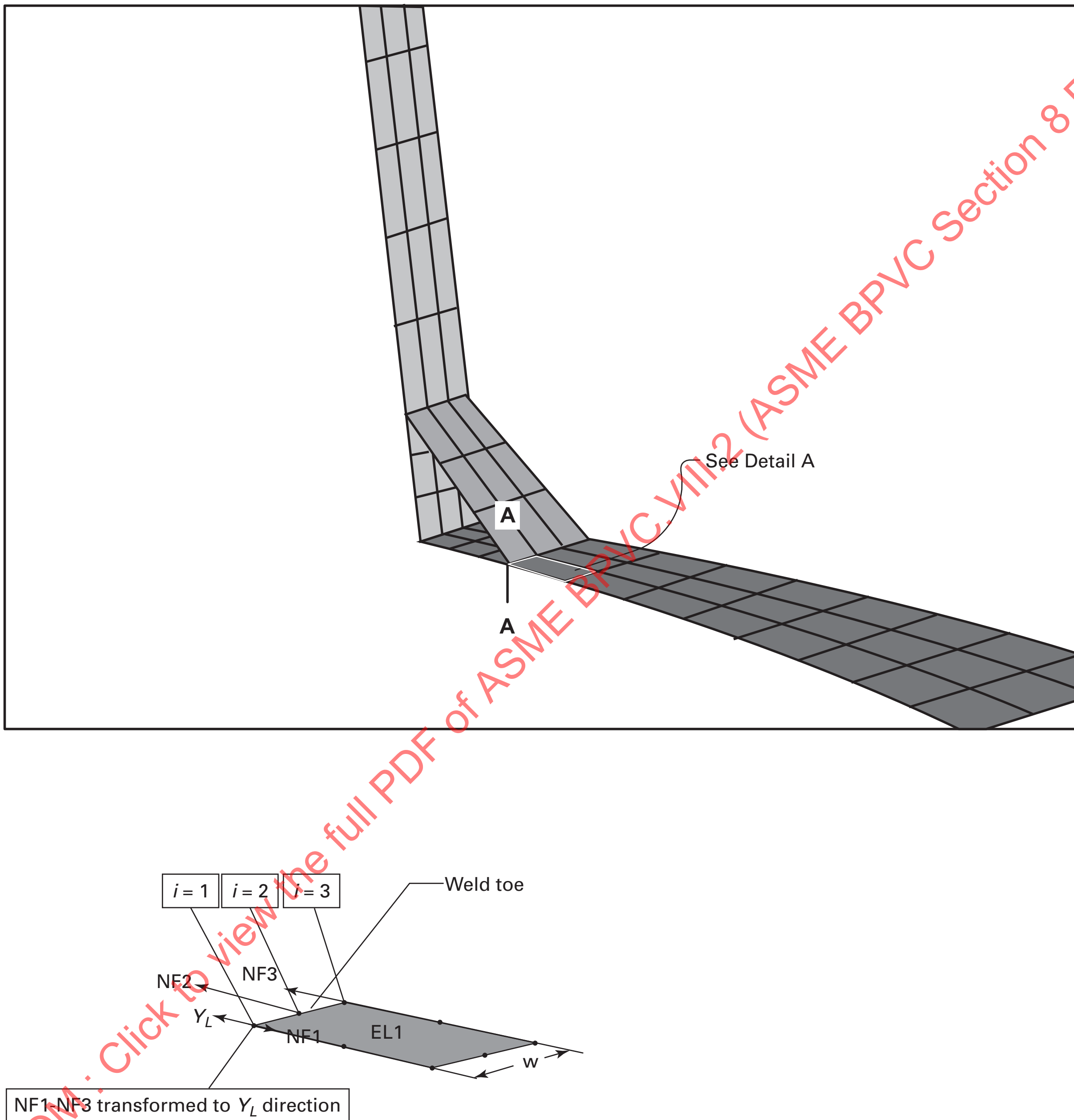
(c) Example of Symmetric Joint (Double Plate Lap Fillet Weld)

Figure 5-A.9
Computation of Membrane and Bending Equivalent Stresses by the Structural Stress Method Using the Results From a Finite Element Model With Shell Elements



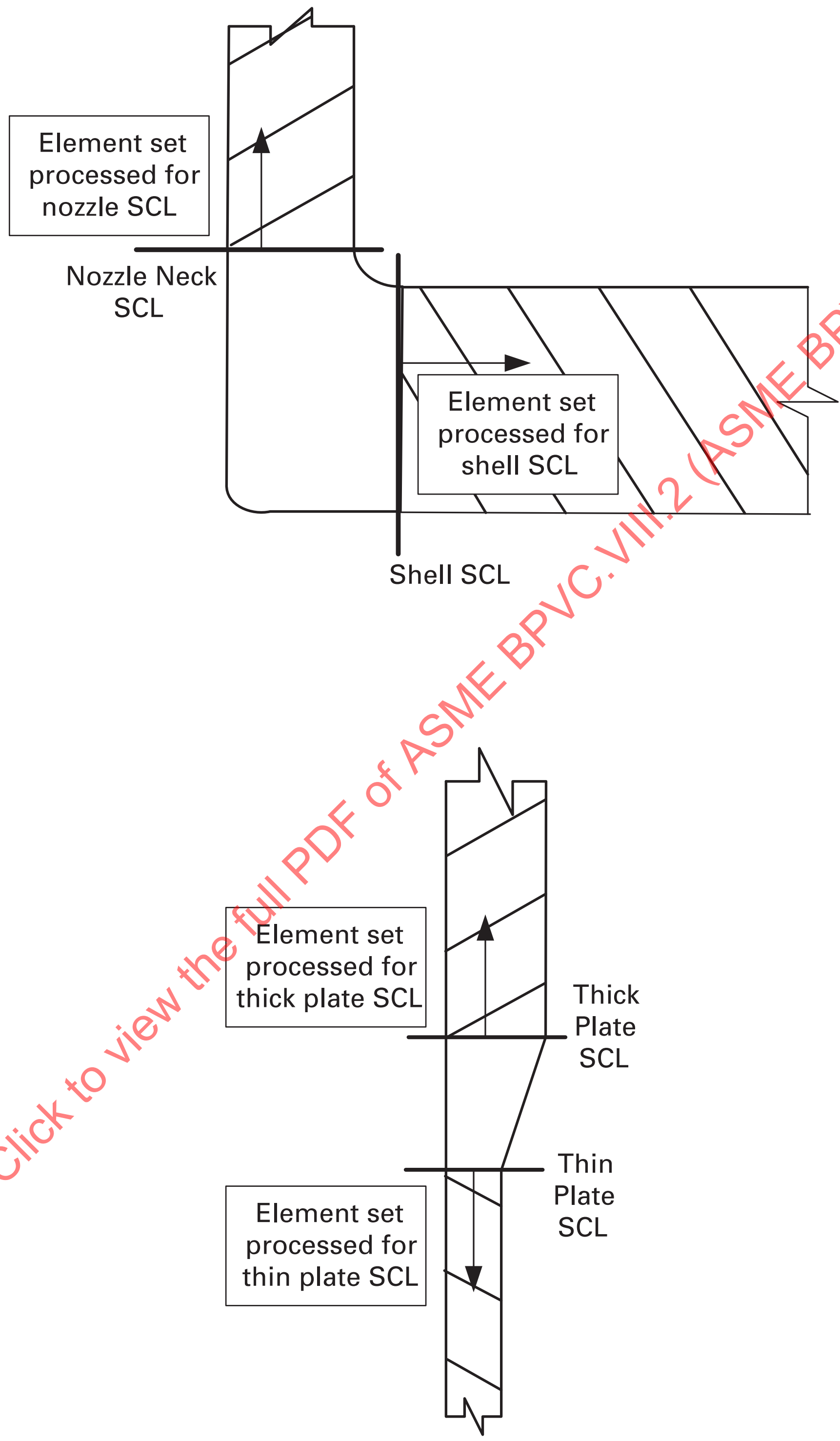
NOTE:
(1) The thickness and material properties of the shell element used to model the fillet weld should be established based on producing an equivalent stiffness of the actual fillet weld.

Figure 5-A.10
Processing Nodal Force Results With the Structural Stress Method Using the Results From a Finite Element Model With Three-Dimensional Second Order Shell Elements



Detail A

Figure 5-A.11
Element Sets for Processing Finite Element Nodal Stress Results With the Structural Stress Method Based on Stress Integration



ANNEX 5-B
HISTOGRAM DEVELOPMENT AND CYCLE COUNTING FOR FATIGUE ANALYSIS

(Informative)

5-B.1 GENERAL

This Annex contains cycle counting procedures required to perform a fatigue assessment for irregular stress or strain versus time histories. These procedures are used to break the loading history down into individual cycles that can be evaluated using the fatigue assessment rules of Part 5. Two cycle counting methods are presented in this Annex. An alternative cycle counting method may be used if agreed to by the Owner-User.

5-B.2 DEFINITIONS

The definitions used in this Annex are shown below.

(a) Event – The User's Design Specification may include one or more events that produce fatigue damage. Each event consists of loading components specified at a number of time points over a time period and is repeated a specified number of times. For example, an event may be the startup, shutdown, upset condition, or any other cyclic action. The sequence of multiple events may be specified or random.

(b) Cycle – A cycle is a relationship between stress and strain that is established by the specified loading at a location in a vessel or component. More than one stress–strain cycle may be produced at a location, either within an event or in transition between two events, and the accumulated fatigue damage of the stress–strain cycles determines the adequacy for the specified operation at that location. This determination shall be made with respect to the stabilized stress–strain cycle.

(c) Proportional Loading – During constant amplitude loading, as the magnitudes of the applied stresses vary with time, the size of Mohr's circle of stress also varies with time. In some cases, even though the size of Mohr's circle varies during cyclic loading, if the orientation of the principal axes remains fixed, the loading is called proportional. An example of proportional loading is a shaft subjected to in-phase torsion and bending, where the ratio of axial and torsional stress remains constant during cycling.

(d) Non-Proportional Loading – If the orientation of the principal axes are not fixed, but change orientation during cyclic loading, the loading is called non-proportional. An example of non-proportional loading is a shaft subjected to out-of-phase torsion and bending, where the ratio of axial and torsional stress varies continuously during cycling.

(e) Peak – The point at which the first derivative of the loading or stress histogram changes from positive to negative.

(f) Valley – The point at which the first derivative of the loading or stress histogram changes from negative to positive.

5-B.3 HISTOGRAM DEVELOPMENT

5-B.3.1

The loading histogram should be determined based on the specified loadings provided in the User's Design Specification. The loading histogram should include all significant operating loads and events that are applied to the component. The following should be considered in developing the loading histogram.

(a) The number of repetitions of each event during the operation life.

(b) The sequence of events during the operation life, if applicable.

(c) Applicable loadings such as pressure, temperature, supplemental loads such as weight, support displacements, and nozzle reaction loadings.

(d) The relationship between the applied loadings during the time history.

5-B.4 CYCLE COUNTING USING THE RAINFLOW METHOD

5-B.4.1

The Rainflow Cycle Counting Method (ASTM Standard No. E1049) is recommended to determine the time points representing individual cycles for the case of situations where the variation in time of loading, stress, or strain can be represented by a single parameter. This cycle counting method is not applicable for non-proportional loading. Cycles counted with the Rainflow Method correspond to closed stress–strain hysteresis loops, with each loop representing a cycle.

5-B.4.2 RECOMMENDED PROCEDURE

Step 1. Determine the sequence of peaks and valleys in the loading histogram. If multiple loadings are applied, it may be necessary to determine the sequence of peaks and valleys using a stress histogram. If the sequence of events is unknown, the worst case sequence should be chosen.

Step 2. Re-order the loading histogram to start and end at either the highest peak or lowest valley, so that only full cycles are counted. Determine the sequence of peaks and valleys in the loading history. Let X denote the range under consideration, and let Y denote the previous range adjacent to X.

Step 3. Read the next peak or valley. If out of data, go to Step 8.

Step 4. If there are less than 3 points, go to Step 3; If not, form ranges X and Y using the three most recent peaks and valleys that have not been discarded.

Step 5. Compare the absolute values of ranges X and Y.

(a) If $X < Y$ go to Step 3

(b) If $X \geq Y$ go to Step 6

Step 6. Count range as one cycle; discard the peak and valley of Y. Record the time points and loadings or component stresses, as applicable, at the starting and ending time points of the cycle.

Step 7. Return to Step 4 and repeat Steps 4 to 6 until no more time points with stress reversals remain.

Step 8. Using the data recorded for the counted cycles perform fatigue assessment in accordance with Part 5.

5-B.5 CYCLE COUNTING USING MAX–MIN CYCLE COUNTING METHOD

5-B.5.1 OVERVIEW

The Max-Min Cycle Counting Method is recommended to determine the time points representing individual cycles for the case of non-proportional loading. The cycle counting is performed by first constructing the largest possible cycle, using the highest peak and lowest valley, followed by the second largest cycle, etc., until all peak counts are used.

5-B.5.2 RECOMMENDED PROCEDURE

Step 1. Determine the sequence of peaks and valleys in the loading history. If some events are known to follow each other, group them together but otherwise arrange the random events in any order.

Step 2. Calculate the elastic stress components σ_{ij} produced by the applied loading at every point in time during each event at a selected location of a vessel. All stress components must be referred to the same global coordinate system. The stress analysis must include peak stresses at local discontinuities.

Step 3. Scan the interior points of each event and delete the time points at which none of the stress components indicate reversals (peaks or valleys).

Step 4. Using the stress histogram from Step 2, determine the time point with the highest peak or lowest valley. Designate the time point as ${}^{m}t$, and the stress components as ${}^{m}\sigma_{ij}$.

Step 5. If time point ${}^{m}t$ is a peak in the stress histogram, determine the component stress range between time point ${}^{m}t$ and the next valley in the stress histogram. If time point ${}^{m}t$ is a valley, determine the component stress range between time point ${}^{m}t$ and the next peak. Designate the next time point as ${}^{n}t$, and the stress components as ${}^{n}\sigma_{ij}$. Calculate the stress component ranges and the von Mises equivalent stress range between time points ${}^{m}t$ and ${}^{n}t$.

$$ {}^{mn}\Delta\sigma_{ij} = {}^{m}\sigma_{ij} - {}^{n}\sigma_{ij} \tag{5-B.1} $$

$$ {}^{mn}\Delta S_{range} = \frac{1}{\sqrt{2}}\left[\left({}^{mn}\Delta\sigma_{11} - {}^{mn}\Delta\sigma_{22}\right)^2 + \left({}^{mn}\Delta\sigma_{22} - {}^{mn}\Delta\sigma_{33}\right)^2 + \left({}^{mn}\Delta\sigma_{33} - {}^{mn}\Delta\sigma_{11}\right)^2 + 6\left({}^{mn}\Delta\sigma_{12}^2 + {}^{mn}\Delta\sigma_{23}^2 + {}^{mn}\Delta\sigma_{31}^2\right)\right]^{0.5} \tag{5-B.2} $$

Step 6. Repeat Step 5, for the current time point, ${}^{m}t$ and the time point of the next peak or valley in the sequence of the stress histogram. Repeat this process for every remaining time point in the stress histogram.

Step 7. Determine the maximum von Mises equivalent stress range obtained in Step 5 and record the time points ${}^m t$ and ${}^n t$ that define the start and end points of the kth cycle.

Step 8. Determine the event or events to which the time points ${}^m t$ and ${}^n t$ belong and record their specified number of repetitions as ${}^m N$ and ${}^n N$, respectively.

Step 9. Determine the number of repetitions of the kth cycle.

(a) If ${}^m N < {}^n N$: Delete the time point ${}^m t$ from those considered in Step 4, and reduce the number of repetitions at time point ${}^n t$ from ${}^n N$ to ${}^n N - {}^m N$.

(b) If ${}^m N > {}^n N$: Delete the time point ${}^n t$ from those considered in Step 4, and reduce the number of repetitions at time point ${}^m t$ from ${}^m N$ to ${}^m N - {}^n N$.

(c) If ${}^m N = {}^n N$: Delete both time points ${}^m t$ and ${}^n t$ from those considered in Step 4.

Step 10. Return to Step 4 and repeat Steps 4 to 10 until no more time points with stress reversals remain.

Step 11. Using the data recorded for the counted cycles, perform fatigue assessment in accordance with Part 5. Note that an elastic–plastic fatigue assessment (see 5.5.4) may be applied if ${}^{nm}\Delta S_{range}$ exceeds the yield point of the cyclic stress range-strain range curve of the material.

5-B.6 NOMENCLATURE

${}^{mn}\Delta S_{range}$ = von Mises equivalent stress range between time points ${}^m t$ and ${}^n t$.
σ_{ij} = stress tensor at the point under evaluation.
${}^m\sigma_{ij}$ = stress tensor at the point under evaluation at time point ${}^m t$.
${}^n\sigma_{ij}$ = stress tensor at the point under evaluation at time point ${}^n t$.
${}^{mn}\Delta\sigma_{ij}$ = stress component range between time points ${}^m t$ and ${}^n t$.
${}^{mn}\Delta\sigma_{11}$ = stress range associated with the normal stress component in the 1-direction between time points ${}^m t$ and ${}^n t$.
${}^{mn}\Delta\sigma_{22}$ = stress range associated with the normal stress component in the 2-direction between time points ${}^m t$ and ${}^n t$.
${}^{mn}\Delta\sigma_{33}$ = stress range associated with the normal stress component in the 3-direction between time points ${}^m t$ and ${}^n t$.
${}^{mn}\Delta\sigma_{12}$ = stress range associated with the shear stress component in the 1-direction between time points ${}^m t$ and ${}^n t$.
${}^{mn}\Delta\sigma_{13}$ = stress range associated with the shear stress component in the 2-direction between time points ${}^m t$ and ${}^n t$.
${}^{mn}\Delta\sigma_{23}$ = stress range associated with the shear stress component in the 3-direction between time points ${}^m t$ and ${}^n t$.
${}^m t$ = time point under consideration with the highest peak or lowest valley.
${}^n t$ = time point under consideration that forms a range with time point ${}^m t$.
${}^m N$ = specified number of repetitions of the event associated with time point ${}^m t$.
${}^n N$ = specified number of repetitions of the event associated with time point ${}^n t$.
X = absolute value of the range (load or stress) under consideration using the Rainflow Cycle Counting Method.
Y = absolute value of the adjacent range (load or stress) to previous X using the Rainflow Cycle Counting Method.

ANNEX 5-C
ALTERNATIVE PLASTICITY ADJUSTMENT FACTORS AND EFFECTIVE ALTERNATING STRESS FOR ELASTIC FATIGUE ANALYSIS

(Normative)

5-C.1 SCOPE

5-C.1.1 This Annex contains procedures for the determination of plasticity correction factors and effective alternating equivalent stress for elastic fatigue analysis. These procedures include a modified Poisson's ratio adjustment for local thermal and thermal bending stresses, a notch plasticity adjustment factor that is applied to thermal bending stresses, and a nonlocal plastic strain redistribution adjustment that is applied to all stresses except local thermal and thermal bending stresses. These procedures are an alternative to effective alternating stress calculations in Step 4 of 5.5.3.2.

5-C.2 DEFINITIONS

5-C.2.1 Thermal Bending Stress. Thermal bending stress is caused by the linear portion of the through-wall temperature gradient. Such stresses shall be classified as secondary stresses.

5-C.2.2 Local Thermal Stress. Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as peak stresses. Examples of local thermal stresses are the stress in a small hot spot in a vessel wall, the non-linear portion of a through-wall temperature gradient in a cylindrical shell, and the thermal stress in a cladding material that has a coefficient of expansion different from that of the base metal. Local thermal stresses are characterized by having two principal stresses that are approximately equal.

5-C.3 EFFECTIVE ALTERNATING STRESS FOR ELASTIC FATIGUE ANALYSIS

5-C.3.1 The effective total equivalent stress amplitude is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the effective total equivalent stress amplitude, defined as one-half of the effective total equivalent stress range ($P_L + P_b + Q + F$) calculated for each cycle in the loading histogram.

5-C.3.2 The following procedure shall be used to determine plasticity correction factors for elastic fatigue analysis and the effective alternating equivalent stress.

Step 1. At the point of interest, determine the following stress tensors and associated equivalent stresses at the start and end points (time points ${}^m t$ and ${}^n t$, respectively) for the kth cycle counted in Step 2 of 5.5.3.2.

(a) Calculate the component stress range between time points ${}^m t$ and ${}^n t$ and compute an equivalent stress range due to primary plus secondary plus peak stress as given below.

$$\Delta\sigma_{ij,k} = {}^m\sigma_{ij,k} - {}^n\sigma_{ij,k} \tag{5-C.1}$$

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\right]^{0.5} \tag{5-C.2}$$

(b) Using the linearized stress results due to primary plus secondary stresses, compute the component stress range using eq. (5-C.1). Compute the equivalent stress range using eq. (5-C.2) and designate this quantity as $\Delta S_{n,k}$.

(c) Determine the stress tensor due to local thermal and thermal bending stresses at the start and end points for the kth cycle. It may be difficult to calculate the local thermal stress from stress distributions obtained from numerical methods. If this is the case, the procedure below can be used to calculate the local thermal and thermal bending stresses due to a non-linear temperature distribution. This method is based on calculating a thermal stress difference range associated with the linearized temperature distribution along the SCL for the time steps of interest. Consistent with that method, consider the distribution of the temperature from numerical method as a function of the local through thickness direction. The temperature distribution for each time step can be separated into three parts.

(1) A constant temperature equal to the average of the temperature distribution

$$T_{\text{avg}} = \frac{1}{t}\int_0^t T dx \tag{5-C.3}$$

(2) The linearly varying portion of the temperature distribution

$$T_b = \frac{6}{t^2}\int_0^t T\left(\frac{t}{2} - x\right)dx \tag{5-C.4}$$

(3) The non-linear portion of the temperature distribution

$$T_p = T - \left(T_{\text{avg}} + 2T_b/t\right) \tag{5-C.5}$$

By assuming full suppression of the differential expansion of the cross-section, the associated local thermal stress parallel to the surface for each time step may be calculated as given below where T_p is given by eq. (5-C.5).

$$\sigma_{ij,k}^{LT} = \frac{-E\alpha\left[T - \left(T_{\text{avg}} + 2T_b/t\right)\right]}{1-\nu} \quad \text{for} \quad i = j = 1, 2 \tag{5-C.6}$$

$$\sigma_{ij,k}^{LT} = 0 \quad \text{for} \quad i \neq j \quad \text{and} \quad i = j = 3 \tag{5-C.7}$$

Using eqs. (5-C.6) and (5-C.7), determine the local thermal component stress ranges using eq. (5-C.1) and designate this quantity as $\Delta\sigma^{LT}{}_{ij,k}$. The thermal bending component stress range, $\Delta\sigma^{TB}{}_{ij,k}$ is determined by linearizing the through-wall stress distribution due to thermal effects only.

(d) Compute the equivalent stress ranges due to primary plus secondary plus peak stress minus the local thermal stress using eqs. (5-C.8) and (5-C.9).

$$\Delta\sigma_{ij,k} = \left({}^m\sigma_{ij,k} - {}^m\sigma_{ij,k}^{LT}\right) - \left({}^n\sigma_{ij,k} - {}^n\sigma_{ij,k}^{LT}\right) \tag{5-C.8}$$

$$\left(\Delta S_{P,k} - \Delta S_{LT,k}\right) = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\right]^{0.5} \tag{5-C.9}$$

(e) Compute the equivalent stress ranges due to local thermal plus thermal bending stress using eqs. (5-C.10) and (5-C.11).

$$\Delta\sigma_{ij,k} = \left({}^m\sigma_{ij,k}^{TB} + {}^m\sigma_{ij,k}^{LT}\right) - \left({}^n\sigma_{ij,k}^{TB} + {}^n\sigma_{ij,k}^{LT}\right) \tag{5-C.10}$$

$$\left(\Delta S_{LT,k} - \Delta S_{TB,k}\right) = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\right]^{0.5} \tag{5-C.11}$$

(f) If required, see eq. (5-C.32), compute the stress tensor due to nonthermal effects (all loadings except local thermal and thermal bending), $\sigma^{NT}{}_{ij,k}$, at the start and end points for the kth cycle.

Step 2. Determine the Poisson's ratio adjustment, $K_{\nu,k}$ to adjust local thermal and thermal bending stresses for the kth cycle based on the equivalent stress ranges in Step 1 using the following equations, (S_{PS} is defined in 5.5.6.1):

$$K_{\nu,k} = 1.0 \quad \text{for} \quad \Delta S_{P,k} \leq S_{PS} \tag{5-C.12}$$

$$K_{v,k} = 0.6\left[\frac{\left(\Delta S_{P,k} - S_{PS}\right)}{\left(\Delta S_{LT,k}\right) + \Delta S_{TB,k}}\right] + 1.0 \quad \text{for} \quad \Delta S_{P,k} > S_{PS} \quad \text{and} \quad \left(\Delta S_{LT,k} + \Delta S_{TB,k}\right) > \left(\Delta S_{P,k} - S_{PS}\right) \tag{5-C.13}$$

$$K_{v,k} = 1.6 \quad \text{for} \quad \Delta S_{P,k} > S_{PS} \quad \text{and} \quad \left(\Delta S_{LT,k} + \Delta S_{TB,k}\right) \leq \left(\Delta S_{P,k} - S_{PS}\right) \tag{5-C.14}$$

Step 3. Determine the nonlocal plastic strain redistribution adjustment, $K_{nl,k}$ to adjust all stresses except local thermal and thermal bending for the kth cycle. In these equations, the parameters m and n are defined in Table 5.13.

$$K_{nl,k} = 1.0 \quad \text{for} \quad \Delta S_{n,k} \leq S_{PS} \tag{5-C.15}$$

$$K_{nl,k} = 1.0 + \frac{(1-n)}{n(m-1)}\left(\frac{\Delta S_{n,k}}{S_{PS}} - 1\right) \quad \text{for} \quad S_{PS} < \Delta S_{n,k} < mS_{PS} \tag{5-C.16}$$

$$K_{nl,k} = \frac{1}{n} \quad \text{for} \quad \Delta S_{n,k} \geq mS_{PS} \tag{5-C.17}$$

Step 4. Determine the notch plasticity adjustment factor, $K_{np,k}$ based on the equivalent stress ranges in Step 1 to adjust thermal bending stresses to account for additional local strain concentration due to a geometric stress riser for the kth cycle. In these equations, the parameters n are defined in Table 5.13.
For numerical results used directly:

$$K_{np,k} = 1.0 \quad \text{for} \quad \left(\Delta S_{P,k} - \Delta S_{LT,k}\right) \leq S_{PS} \tag{5-C.18}$$

$$K_{np,k} = \min\left[K_1, K_2\right] \quad \text{for} \quad \left(\Delta S_{P,k} - \Delta S_{LT,k}\right) > S_{PS} \tag{5-C.19}$$

$$K_1 = \left[\left(\frac{\Delta S_{P,k} - \Delta S_{LT,k}}{\Delta S_{n,k}}\right)^{\left(\frac{1-n}{1+n}\right)} - 1.0\right] \cdot \left[\frac{\left(\Delta S_{P,k} - \Delta S_{LT,k}\right) - S_{PS}}{\left(\Delta S_{P,k} - \Delta S_{LT,k}\right)}\right] + 1.0 \tag{5-C.20}$$

$$K_2 = \frac{K_{nl,k}}{K_{v,k}} \tag{5-C.21}$$

For numerical results that are adjusted with a stress concentration factor (SCF):

$$K_{np,k} = 1.0 \quad \text{for} \quad \left(\Delta S_{n,k} \cdot \mathrm{SCF}\right) \leq S_{PS} \tag{5-C.22}$$

$$K_{np,k} = \min\left[K_1, K_2\right] \quad \text{for} \quad \left(\Delta S_{n,k} \cdot \mathrm{SCF}\right) > S_{PS} \tag{5-C.23}$$

$$K_1 = \left[(\mathrm{SCF})^{\left(\frac{1-n}{1+n}\right)} - 1.0\right] \cdot \left[\frac{\left(\Delta S_{n,k} \cdot \mathrm{SCF}\right) - S_{PS}}{\left(\Delta S_{n,k} \cdot \mathrm{SCF}\right)}\right] + 1.0 \tag{5-C.24}$$

$$K_2 = \frac{K_{nl,k}}{K_{v,k}} \tag{5-C.25}$$

Note that the SCF and $K_{np,k}$ values may be dependent upon the component stress direction.

Step 5. Apply the plasticity adjustment factors to the component stresses at the start and end points for the kth cycle.

(a) Compute the component stresses including plastic Poisson's ratio and notch plasticity adjustments as given below for time points ${}^{m}t$ and ${}^{n}t$. For numerical results used directly:

$$\left(\sigma_{ij}^{LT}\right)_{\text{adj}} = \sigma_{ij,k}^{LT} \cdot K_{v,k} \tag{5-C.26}$$

$$\left(\sigma_{ij}^{TB}\right)_{\text{adj}} = \sigma_{ij,k}^{TB} \cdot K_{v,k} \cdot K_{np,k} + \sigma_{ij,k}^{TB} \cdot \left(\text{SCF}_{\text{NUM}} - 1\right) \cdot K_{np,k} \tag{5-C.27}$$

For numerical results that are adjusted with a stress concentration factor (SCF):

$$\left(\sigma_{ij}^{LT}\right)_{\text{adj}} = \sigma_{ij,k}^{LT} \cdot K_{v,k} \cdot \text{SCF}_{LT} \tag{5-C.28}$$

$$\left(\sigma_{ij}^{TB}\right)_{\text{adj}} = \sigma_{ij,k}^{TB} \cdot K_{v,k} \cdot K_{np,k} \cdot \text{SCF} + \sigma_{ij,k}^{TB} \cdot \left(\text{SCF}_{\text{NUM}} - 1\right) \cdot K_{np,k} \tag{5-C.29}$$

(b) Compute the component stresses including nonlocal plastic strain redistribution adjustment as given below for time points ${}^{m}t$ and ${}^{n}t$. For numerical results used directly:

$$\left(\sigma_{ij}^{NT}\right)_{\text{adj}} = \left[\sigma_{ij,k} - \sigma_{ij,k}^{TB}\left(\text{SCF}_{\text{NUM}} - 1\right)\right] \cdot K_{np,k} \tag{5-C.30}$$

$$\text{SCF}_{\text{NUM}} = \frac{\left(\Delta S_{p,k} - \Delta S_{LT,k}\right)}{\Delta S_{n,k}} \tag{5-C.31}$$

For numerical results that are adjusted with a stress concentration factor (SCF):

$$\left(\sigma_{ij}^{NT}\right)_{\text{adj}} = \sigma_{ij,k}^{NT} \cdot K_{nl,k} \cdot \text{SCF} \tag{5-C.32}$$

Step 6. Compute the adjusted component stress ranges between time points ${}^{m}t$ and ${}^{n}t$ as given below.

$$\left(\Delta\sigma_{ij,k}\right)_{\text{adj}} = \left\{ {}^{m}\!\left[\left(\sigma_{ij}^{LT}\right)_{\text{adj}} + \left(\sigma_{ij}^{NT}\right)_{\text{adj}} + \left(\sigma_{ij}^{TB}\right)_{\text{adj}}\right] - {}^{n}\!\left[\left(\sigma_{ij}^{LT}\right)_{\text{adj}} + \left(\sigma_{ij}^{NT}\right)_{\text{adj}} + \left(\sigma_{ij}^{TB}\right)_{\text{adj}}\right] \right\} \tag{5-C.33}$$

Step 7. Compute the effective equivalent stress range using the adjusted component stress ranges from Step 6 and Eq. (5-C.2). Designate the adjusted effective equivalent stress range as $(\Delta S_{P,k})_{\text{adj}}$.

Step 8. Compute the effective alternating equivalent stress for the kth cycle as given below.

$$S_{\text{alt},k} = 0.5\left(\Delta S_{P,k}\right)_{\text{adj}} \tag{5-C.34}$$

5-C.4 NOMENCLATURE

α = thermal expansion coefficient of the material at the point under consideration, evaluated at the mean temperature of the kth cycle.

$\Delta S_{n,k}$ = primary plus secondary equivalent stress range for the kth cycle.

$\Delta S_{P,k}$ = range of primary plus secondary plus peak equivalent stress for the kth cycle.

$\Delta S_{LT,k}$ = primary plus secondary plus peak equivalent stress range due to local thermal effects for the kth cycle.

$\Delta S_{TB,k}$ = primary plus secondary plus peak equivalent stress range due to thermal bending effects for the kth cycle.

$\Delta S_{NT,k}$ = primary plus secondary plus peak equivalent stress range due to nonthermal effects for the kth cycle.

$(\Delta S_{P,k})_{\text{adj}}$ = adjusted range of primary plus secondary plus peak equivalent stress, including nonlocal strain redistribution, notch plasticity, and plastic Poisson's ratio adjustments for the kth cycle.

$\Delta\sigma_{ij,k}$ = stress component range between time points ${}^{m}t$ and ${}^{n}t$ for the kth cycle.

$\Delta\sigma^{LT}_{ij,k}$ = stress component range due to local thermal stress between time point ${}^{m}t$ and ${}^{n}t$ for the kth cycle.

$\Delta\sigma^{TB}_{ij,k}$ = stress component range due to thermal bending stress between time points ${}^{m}t$ and ${}^{n}t$ for the kth cycle.

adjusted stress tensor, including nonlocal strain redistribution, notch plasticity, and plastic Poisson's ratio adjustments at the location and time point under evaluation for the kth cycle.

E = Young's Modulus of the material evaluated at the mean temperature of the cycle.

K_1 = parameter used to compute $K_{np,k}$.

K_2 = parameter used to compute $K_{np,k}$.

$K_{nl,k}$ = nonlocal strain redistribution adjustment factor for the kth cycle.

$K_{np,k}$ = notch plasticity adjustment factor for the kth cycle.

$K_{v,k}$ = plastic Poisson's ratio adjustment factor for the kth cycle.
m = material constant used for the nonlocal strain redistribution adjustment factor, per Table 5.13.
n = material constant used for the nonlocal strain redistribution adjustment factor, per Table 5.13.
v = Poisson's ratio.
$S_{\mathrm{alt},k}$ = alternating equivalent stress for the kth cycle.
S_{PS} = allowable limit on the primary plus secondary stress range.
SCF = stress concentration factor.
SCF_{LT} = stress concentration factor applicable for local thermal stress.
$\mathrm{SCF}_{\mathrm{NUM}}$ = stress concentration factor determined from the numerical model.
${\sigma^{LT}}_{ij,k}$ = stress tensor due to local thermal stress at the location and time point under evaluation for the kth cycle.
${\sigma^{NT}}_{ij,k}$ = stress tensor due to nonthermal stress at the location and time point under evaluation for the kth cycle.
${\sigma^{TB}}_{ij,k}$ = stress tensor due to thermal bending stress due to the linearly varying portion of the temperature distribution at the location and time point under evaluation for the kth cycle.
${}^m\sigma_{ij,k}$ = stress tensor at the point under evaluation at time point ${}^m t$ for the kth cycle.
${}^n\sigma_{ij,k}$ = stress tensor at the point under evaluation at time point ${}^n t$ for the kth cycle.
${}^m{\sigma^{LT}}_{ij,k}$ = stress tensor due to local thermal stress at the location under evaluation at time point ${}^m t$ for the kth cycle.
${}^n{\sigma^{LT}}_{ij,k}$ = stress tensor due to local thermal stress at the location under evaluation at time point ${}^n t$ for the kth cycle.
${}^m{\sigma^{TB}}_{ij,k}$ = stress tensor due to thermal bending stress at the location under evaluation at time point ${}^m t$ for the kth cycle.
${}^n{\sigma^{TB}}_{ij,k}$ = stress tensor due to thermal bending stress at the location under evaluation at time point ${}^n t$ for the kth cycle.
$({\sigma^{LT}}_{ij})_{\mathrm{adj}}$ = adjusted stress tensor due to local thermal stress at the location and time point under evaluation for the kth cycle.
$({\sigma^{NT}}_{ij})_{\mathrm{adj}}$ = adjusted stress tensor due to nonthermal stress at the location and time point under evaluation for the kth cycle.
$({\sigma^{TB}}_{ij})_{\mathrm{adj}}$ = adjusted stress tensor due to thermal bending stress at the location and time point under evaluation for the kth cycle.
t = wall thickness.
${}^m t$ = time point under consideration with the highest peak or lowest valley.
${}^n t$ = time point under consideration that forms a range with time point ${}^m t$.
T = temperature distribution.
T_{avg} = average temperature component of temperature distribution T.
T_b = equivalent linear temperature component of temperature distribution T.
T_p = peak temperature component of temperature distribution T.
x = position through the wall thickness.
z = local coordinate for the temperature distribution.