ASME BTH-1–2014
(Revision of ASME BTH-1–2011)

Design of Below-the-Hook Lifting Devices

AN AMERICAN NATIONAL STANDARD

The American Society of Mechanical Engineers

INTENTIONALLY LEFT BLANK

Errata
to
ASME BTH-1–2014
Design of Below-the-Hook Lifting Devices

On page 13, in the nomenclature for eq. (3-17), the second definition for C_{LTB} has been corrected by errata to replace the square root sign in the numerator with parentheses. The correct equation is shown below.

$$C_{LTB} = \frac{2.00\,(EI_x/GJ)}{(L_b/b_f)^2} + 0.275 \leq 1.00$$

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
Two Park Avenue, New York, NY 10016-5990

January 2015

J0175E

ASME BTH-1–2014

(Revision of ASME BTH-1–2011)

Design of Below-the-Hook Lifting Devices

AN AMERICAN NATIONAL STANDARD

The American Society of Mechanical Engineers

Two Park Avenue • New York, NY • 10016 USA

Date of Issuance: October 29, 2014

The next edition of this Standard is scheduled for publication in 2017. This Standard will become effective 1 year after the Date of Issuance.

ASME issues written replies to inquiries concerning interpretations of technical aspects of this Standard. Interpretations may be accessed as they are issued at http://go.asme.org/interpretations. They are published on the ASME Web site under the Committee Pages at http://cstools.asme.org/.

Errata to codes and standards may be posted on the ASME Web site under the Committee Pages to provide corrections to incorrectly published items, or to correct typographical or grammatical errors in codes and standards. Such errata shall be used on the date posted.

The Committee Pages can be found at http://cstools.asme.org/. There is an option available to automatically receive an e-mail notification when errata are posted to a particular code or standard. This option can be found on the appropriate Committee Page after selecting "Errata" in the "Publication Information" section.

The American Society of Mechanical Engineers
Two Park Avenue, New York, NY 10016-5990

CONTENTS

Figures

Tables

Nonmandatory Appendices

FOREWORD

There have been many formal requests for interpretation of the limited structural design criteria stated within ASME B30.20, Below-the-Hook Lifting Devices, a safety standard. As a consequence, industry has for quite some time expressed a need for a comprehensive design standard for below-the-hook lifting devices that would complement the safety requirements of ASME B30.20. All editions of ASME B30.20 have included structural design criteria oriented toward the industrial manufacturing community requiring a minimum design factor of three, based on the yield strength of the material; recent editions have also included design criteria for the fatigue failure mode. However, members of the construction community expressed the need for design criteria more suitable to their operating conditions, including a lower design factor, and the necessity to address other failure modes such as fracture, shear and buckling, and design topics such as impact and fasteners.

A Design Task Group was created in 1997 to begin work on a design standard as a companion document to ASME B30.20. The ASME BTH Standards Committee on the Design of Below-the-Hook Lifting Devices was formed out of the Design Task Group and held its organizational meeting on December 5, 1999.

ASME BTH-1–2005, Design of Below-the-Hook Lifting Devices, contained five chapters: Scope and Definitions, Lifter Classifications, Structural Design, Mechanical Design, and Electrical Components. This Standard, intended for general industry and construction, sets forth two design categories for lifters based on the magnitude and variation of loading, and operating and environmental conditions. The two design categories provide different design factors for determining allowable static stress limits. Five Service Classes based on load cycles are provided. The Service Class establishes allowable stress range values for lifter structural members and design parameters for mechanical components. ASME BTH-1–2005 was approved by the American National Standards Institute on October 18, 2005.

ASME BTH-1–2008 incorporated editorial revisions and two new mechanical design sections for grip ratio and vacuum-lifting device design. ASME BTH-1–2008 was approved by the American National Standards Institute on September 17, 2008.

ASME BTH-1–2011 incorporated revisions throughout the Standard and the addition of a new mechanical design section for fluid power systems. ASME BTH-1–2011 was approved by the American National Standards Institute on September 23, 2011.

This revision of ASME BTH-1 includes a section on lifting magnets that has been incorporated into Chapter 4. Other technical revisions include new requirements for fluid pressure control and electrical system guarding. Along with these technical changes, the nonmandatory Commentary for each chapter was moved to its own respective Nonmandatory Appendix. ASME BTH-1–2014 was approved by the American National Standards Institute on June 24, 2014.

ASME BTH STANDARDS COMMITTEE
Design of Below-the-Hook Lifting Devices

(The following is the roster of the Committee at the time of approval of this Standard.)

STANDARDS COMMITTEE OFFICERS

D. R. Verenski, *Chair*
D. Duerr, *Vice Chair*
M. R. Gerson, *Secretary*

STANDARDS COMMITTEE PERSONNEL

S. Barton, Doc Magnet
G. A. Bond, Wood's Powr-Grip Co., Inc.
P. W. Boyd, The Boeing Co.
D. Duerr, 2DM Associates, Inc.
J. D. Edmundson, Konecranes/P&H
A. Faris, Ibberson Engineering Co.
M. R. Gerson, The American Society of Mechanical Engineers
K. M. Jankowski, Walker Magnetics
T. C. Mackey, Washington River Protection Solutions
D. L. McCabe, Babcock and Wilcox Co.
D. T. Rebbin, Bradley Lifting Corp.
J. E. Rea, *Alternate,* Bradley Lifting Corp.
J. W. Rowland III, Consultant
R. S. Stemp, Lampson International, LLC
P. D. Sweeney, General Dynamics, Electric Boat
P. E. Van Every, Tecnomagnete, Inc.
D. R. Verenski, Hunter Lift Ltd.
T. J. Brookbank, *Alternate,* Hunter Lift Ltd.

HONORARY MEMBERS

R. A. Dahlin, Walker Magnetics
J. W. Downs, Jr., Downs Crane and Hoist Co.

CORRESPONDENCE WITH THE BTH COMMITTEE

General. ASME Standards are developed and maintained with the intent to represent the consensus of concerned interests. As such, users of this Standard may interact with the Committee by requesting interpretations, proposing revisions, and attending Committee meetings. Correspondence should be addressed to:

Secretary, BTH Standards Committee
The American Society of Mechanical Engineers
Two Park Avenue
New York, NY 10016-5990
http://go.asme.org/Inquiry

Proposing Revisions. Revisions are made periodically to the Standard to incorporate changes that appear necessary or desirable, as demonstrated by the experience gained from the application of the Standard. Approved revisions will be published periodically.

The Committee welcomes proposals for revisions to this Standard. Such proposals should be as specific as possible, citing the paragraph number(s), the proposed wording, and a detailed description of the reasons for the proposal, including any pertinent documentation.

Interpretations. Upon request, the BTH Standards Committee will render an interpretation of any requirement of the Standard. Interpretations can only be rendered in response to a written request sent to the Secretary of the BTH Standards Committee at go.asme.org/Inquiry.

The request for an interpretation should be clear and unambiguous. It is further recommended that the inquirer submit his/her request in the following format:

Subject:	Cite the applicable paragraph number(s) and the topic of the inquiry.
Edition:	Cite the applicable edition of the Standard for which the interpretation is being requested.
Question:	Phrase the question as a request for an interpretation of a specific requirement suitable for general understanding and use, not as a request for an approval of a proprietary design or situation. The inquirer may also include any plans or drawings that are necessary to explain the question; however, they should not contain proprietary names or information.

Requests that are not in this format may be rewritten in the appropriate format by the Committee prior to being answered, which may inadvertently change the intent of the original request.

ASME procedures provide for reconsideration of any interpretation when or if additional information that might affect an interpretation is available. Further, persons aggrieved by an interpretation may appeal to the cognizant ASME Committee or Subcommittee. ASME does not "approve," "certify," "rate," or "endorse" any item, construction, proprietary device, or activity.

Attending Committee Meetings. The BTH Standards Committee regularly holds meetings and/or telephone conferences that are open to the public. Persons wishing to attend any meeting and/or telephone conference should contact the Secretary of the BTH Standards Committee. Future Committee meeting dates and locations can be found on the Committee Page at http://cstools.asme.org/.

ASME BTH-1–2014
SUMMARY OF CHANGES

Following approval by the ASME BTH Standards Committee and ASME, and after public review, ASME BTH-1–2014 was approved by the American National Standards Institute on June 24, 2014.

ASME BTH-1–2014 includes editorial changes, revisions, and corrections identified by a margin note, **(14)**.

Page	*Location*	*Change*
1	1-2	Second sentence of last paragraph revised
2, 3	1-4.6	Revised
	1-5.1	Definitions of *cycle, load; design factor; fatigue; lifting attachment; limit state; qualified person; stress, maximum;* and *stress, minimum* revised
	1-5.2	Definitions of *brittle fracture* revised and *unbraced length* deleted
	1-5.3	Definitions of *grip ratio* and *gripping force* deleted
4–7	1-6.1	Nomenclature for F_u, L_b, M_1, M_2, N, N_d, and N_{eq} revised, and I_x added
	1-6.2	Nomenclature for F_H revised, $GR_{\min}$ and μ_{SF} deleted, and F_s added
	1-7	ANSI/NFPA 70 updated
10	3-1.3	3-1.3.1 and 3-1.3.2 designations added
	3-1.4	Revised
12, 13	3-2.3.1	Title revised
	3-2.3.2	Revised
	3-2.3.3	Revised
	3-2.3.4	Revised
16	3-3.3.4	Revised
17	3-3.4.1	Revised
18	3-4.2	Revised
19–32	Table 3-4.3-1	Last row added
	3-4.6	F_{TH} and N revised
	Table 3-4.4-1	Revised in its entirety
39	4-9	Revised in its entirety
	Figure 4-9.2-1	Added
40	4-11.5	Added

Page	*Location*	*Change*
	4-11.6	Previous para. 4-11.5 redesignated as 4-11.6
	4-12	Added
42, 43	5-3.8	Revised
	5-4	Title revised
	5-4.6	Revised
	5-4.10	Added
	5-6.2	Revised
	5-6.3	Revised
45–48	Nonmandatory Appendix A	Added
49, 50	Nonmandatory Appendix B	Added
51–59	Nonmandatory Appendix C	Added
60–62	Nonmandatory Appendix D	Added
63, 64	Nonmandatory Appendix E	Added

INTENTIONALLY LEFT BLANK

DESIGN OF BELOW-THE-HOOK LIFTING DEVICES

Chapter 1
Scope, Definitions, and References

1-1 PURPOSE

This Standard sets forth design criteria for ASME B30.20, Below-the-Hook Lifting Devices. This Standard serves as a guide to designers, manufacturers, purchasers, and users of below-the-hook lifting devices.

(14) 1-2 SCOPE

This Standard provides minimum structural and mechanical design and electrical component selection criteria for ASME B30.20, Below-the-Hook Lifting Devices.

The provisions in this Standard apply to the design or modification of below-the-hook lifting devices. Compliance with requirements and criteria that may be unique to specialized industries and environments is outside the scope of this Standard.

Lifting devices designed to this Standard shall comply with ASME B30.20, Below-the-Hook Lifting Devices. ASME B30.20 includes provisions that apply to the marking, construction, installation, inspection, testing, maintenance, and operation of below-the-hook lifting devices.

The provisions defined in this Standard address the most common and broadly applicable aspects of the design of below-the-hook lifting devices. A qualified person shall determine the appropriate methods to be used to address design issues that are not explicitly covered in the standard so as to provide design factors and/or performance consistent with the intent of this Standard.

1-3 NEW AND EXISTING DEVICES

The effective date of this Standard shall be one year after its date of issuance. Lifting devices manufactured after the effective date shall conform to the requirements of this Standard.

When a lifter is being modified, its design shall be reviewed relative to this Standard, and the need to meet this Standard shall be evaluated by the manufacturer or a qualified person.

1-4 GENERAL REQUIREMENTS

1-4.1 Design Responsibility

Lifting devices shall be designed by, or under the direct supervision of, a qualified person.

1-4.2 Units of Measure

A dual unit format is used. Values are given in U.S. Customary units as the primary units followed by the International System of Units (SI) in parentheses as the secondary units. The values stated in U.S. Customary units are to be regarded as the standard. The SI units in the text have been directly (softly) converted from U.S. Customary units.

1-4.3 Design Criteria

All below-the-hook lifting devices shall be designed for specified rated loads, load geometry, Design Category (see section 2-2), and Service Class (see section 2-3). Resolution of loads into forces and stress values affecting structural members, mechanical components, and connections shall be performed by an accepted analysis method.

1-4.4 Analysis Methods

The allowable stresses and stress ranges defined in this Standard are based on the assumption of analysis by classical strength of material methods (models), although other analysis methods may be used. The analysis techniques and models used by the qualified person shall accurately represent the loads, material properties, and device geometry; stress values resulting from the analysis shall be of suitable form to permit correlation with the allowable stresses defined in this Standard.

1-4.5 Material

The design provisions of this Standard are based on the use of carbon, high strength low-alloy, or heat treated constructional alloy steel for structural members and many mechanical components. Other materials may be used, provided the margins of safety and fatigue life are equal to or greater than those required by this Standard.

All ferrous and nonferrous metal used in the fabrication of lifting device structural members and mechanical components shall be identified by an industry-wide or written proprietary specification.

(14) 1-4.6 Welding

All welding designs and procedures for lifters fabricated from steel, except for the design strength of welds, shall be in accordance with the requirements of AWS D14.1/D14.1M. The design strength of welds shall be as defined in para. 3-3.4. When conflicts exist between AWS D14.1/D14.1M and this Standard, the requirements of this Standard shall govern.

Welding of lifters fabricated from metals other than steel shall be performed in accordance with a suitable welding specification as determined by a qualified person, provided the quality and inspection requirements are equal to or greater than those required by this Standard.

1-4.7 Temperature

The design provisions of this Standard are considered applicable when the temperature of the lifter structural or mechanical component under consideration is within the range of 25°F to 150°F (−4°C to 66°C). When the temperature of the component is beyond these limits, special additional design considerations may be required. These considerations may include choosing a material that has better cold-temperature or high-temperature properties, limiting the design stresses to a lower percentage of the allowable stresses, or restricting use of the lifter until the component temperature falls within the stated limits.

The design provisions for electrical components are considered applicable when ambient temperatures do not exceed 104°F (40°C). Lifters expected to operate in ambient temperatures beyond this limit shall have electrical components designed for the higher ambient temperature.

1-5 DEFINITIONS

The paragraph given after the definition of a term refers to the paragraph where the term is first used.

(14) 1-5.1 Definitions — General

ambient temperature: the temperature of the atmosphere surrounding the lifting device (para. 1-4.7).

below-the-hook lifting device (lifting device, lifter): a device used for attaching a load to a hoist. The device may contain components such as slings, hooks, and rigging hardware that are addressed by ASME B30 volumes or other standards (section 1-1).

cycle, load: one sequence of loading defined by a range between minimum and maximum stress (para. 1-5.1).

design: the activity in which a qualified person creates devices, machines, structures, or processes to satisfy a human need (section 1-1).

design factor: the ratio of the limit state stress(es) or strength of an element to the permissible internal stress(es) or forces created by the external force(s) that acts upon the element (para. 1-6.1).

fatigue: the process of progressive localized permanent material damage that may result in cracks or complete fracture after a sufficient number of load cycles (para. 1-5.1).

fatigue life: the number of load cycles of a specific type and magnitude that a member sustains before failure (para. 1-4.5).

hoist: a machinery unit that is used for lifting and lowering (para. 1-5.1).

lifting attachment: a load supporting device that is bolted or permanently attached to the lifted load, such as lifting lugs, padeyes, trunnions, and similar appurtenances (Nonmandatory Appendix A, section A-2).

limit state: a condition in which a structure or component becomes unfit for service, such as brittle fracture, plastic collapse, excessive deformation, durability, fatigue, or instability, and is judged either to be no longer useful for its intended function *(serviceability limit state)* or to be unsafe *(strength limit state)* (para. 1-5.1).

load(s), applied: external force(s) acting on a structural member or machine element due to the rated load, dead load, and other forces created by the operation and geometry of the lifting device (para. 1-5.2).

load, dead: the weights of the parts of the lifting device (para. 1-5.1).

load, rated: the maximum load for which the lifting device is designated by the manufacturer (para. 1-4.3).

manufacturer: the person, company, or agency responsible for the design, fabrication, or performance of a below-the-hook lifting device or lifting device component (section 1-1).

mechanical component: a combination of one or more machine elements along with their framework, fastenings, etc., designed, assembled, and arranged to support, modify, or transmit motion, including, but not limited to, the pillow block, screw jack, coupling, clutch, brake, gear reducer, and adjustable speed transmission (para. 1-4.3).

modification: any change, addition to, or reconstruction of a lifter component (section 1-2).

qualified person: a person who, by possession of a recognized degree in an applicable field or certificate of professional standing, or who, by extensive knowledge, training and experience, has successfully demonstrated the ability to solve or resolve problems relating to the subject matter and work (section 1-2).

rigging hardware: a detachable load supporting device such as a shackle, link, eyebolt, ring, swivel, or clevis (para. 1-5.1).

serviceability limit state: limiting condition affecting the ability of a structure to preserve its maintainability, durability, or function of machinery under normal usage (para. 1-5.1).

shall: indicates that the rule is mandatory and must be followed (section 1-2).

should: indicates that the rule is a recommendation, the advisability of which depends on the facts in each situation (para. 2-2.1).

sling: an assembly to be used for lifting when connected to a hoist or lifting device at the sling's upper end and when supporting a load at the sling's lower end (para. 1-5.1).

strength limit state: limiting condition affecting the safety of the structure, in which the ultimate load carrying capacity is reached (para. 1-5.1).

stress concentration: localized stress considerably higher than average (even in uniformly loaded cross sections of uniform thickness) due to abrupt changes in geometry or localized loading (para. 3-4.1).

stress, maximum: highest algebraic stress per load cycle (para. 1-5.1).

stress, minimum: lowest algebraic stress per load cycle (para. 1-5.1).

stress range: algebraic difference between maximum and minimum stress. Tension stress is considered to have the opposite algebraic sign from compression stress (para. 1-4.4).

structural member: a component or rigid assembly of components fabricated from structural shape(s), bar(s), plate(s), forging(s), or casting(s) (para. 1-4.3).

(14) 1-5.2 Definitions for Chapter 3

block shear: a mode of failure in a bolted or welded connection that is due to a combination of shear and tension acting on orthogonal planes around the minimum net failure path of the connecting elements (para. 3-3.2).

brittle fracture: abrupt cleavage with little or no prior ductile deformation (para. 1-5.1).

compact section: a structural member cross-section that can develop a fully plastic stress distribution before the onset of local buckling (para. 3-2.3.1).

effective length: the equivalent length Kl used in compression formulas (para. 1-5.2).

effective length factor: the ratio between the effective length and the unbraced length of the member measured between the centers of gravity of the bracing members (para. 1-6.1).

effective net tensile area: portion of the gross tensile area that is assumed to carry the design tension load at the member's connections or at location of holes, cutouts, or other reductions of cross-sectional area (para. 3-2.1).

effective width: the reduced width of a plate which, with an assumed uniform stress distribution, produces the same effect on the behavior of a structural member as the actual plate width with its nonuniform stress distribution (para. 1-6.1).

faying surface: the plane of contact between two plies of a bolted connection (para. 1-5.2).

gross area: full cross-sectional area of the member (para. 3-2.1).

local buckling: the buckling of a compression element that may precipitate the failure of the whole member at a stress level below the yield stress of the material (para. 1-5.2).

noncompact section: a structural member cross section that can develop the yield stress in compression elements before local buckling occurs, but will not resist inelastic local buckling at strain levels required for a fully plastic stress distribution (para. 3-2.3.2).

prismatic member: a member with a gross cross section that does not vary along its length (para. 1-6.1).

prying force: a force due to the lever action that exists in connections in which the line of application of the applied load is eccentric to the axis of the bolt, causing deformation of the fitting and an amplification of the axial force in the bolt (para. 3-4.5).

slip-critical: a type of bolted connection in which shear is transmitted by means of the friction produced between the faying surfaces by the clamping action of the bolts (para. 1-6.1).

1-5.3 Definitions for Chapter 4 (14)

back-driving: a condition where the load imparts motion to the drive system (para. 4-5.5).

coefficient of static friction: the nondimensional number obtained by dividing the friction force resisting initial motion between two bodies by the normal force pressing the bodies together (para. 4-9.2).

drive system: an assembly of components that governs the starting, stopping, force, speed, and direction imparted to a moving apparatus (para. 1-5.3).

fluid power: energy transmitted and controlled by means of a pressurized fluid, either liquid or gas. The term applies to both hydraulics, which uses a pressurized liquid such as oil or water, and pneumatics, which uses compressed air or other gases (section 4-11).

L_{10} bearing life: the basic rating or specification life of a bearing (para. 4-6.2).

lock-up: a condition whereby friction in the drive system prevents back-driving (para. 4-5.5).

pitch diameter: the diameter of a sheave measured at the centerline of the rope (para. 4-2.2).

sheave: a grooved wheel used with a rope to change direction and point of application of a pulling force (para. 1-5.3).

sheave, equalizing: a sheave used to equalize tension in opposite parts of a rope. Because of its slight movement, it is not termed a *running sheave* (para. 4-2.3).

sheave, running: a sheave that rotates as the load is lifted or lowered (para. 1-5.3).

vacuum: pressure less than ambient atmospheric pressure (para. 1-5.3).

vacuum lifter: a below-the-hook lifting device for lifting and transporting loads using a holding force by means of vacuum (section 4-10).

vacuum pad: a device that applies a holding force on the load by means of vacuum (para. 4-10.1).

1-5.4 Definitions for Chapter 5

brake: a device, other than a motor, used for retarding or stopping motion of an apparatus by friction or power means (section 5-2).

control(s): a device used to govern or regulate the functions of an apparatus (para. 1-5.4).

control panel: an assembly of components that governs the flow of power to or from a motor or other equipment in response to a signal(s) from a control device(s) (para. 5-4.8).

control system: an assembly or group of devices that govern or regulate the operation of an apparatus (para. 5-3.1).

controller: a device or group of devices that govern, in a predetermined manner, the power delivered to the motor to which it is connected (section 5-4).

duty cycle:

$$\text{duty cycle} = \frac{\text{time on}}{\text{time on} + \text{time off}} \times 100$$

and is expressed as a percentage (para. 5-2.1).

EXAMPLE: 3 min on, 2 min off equals

$$\frac{3}{3+2} \times 100 = 60\%$$

electromagnet, externally powered: a lifting magnet suspended from a crane that requires power from a source external to the crane (para. 5-6.3).

ground (grounded): electrically connected to earth or to some conducting body that serves in place of the earth (section 5-5).

motor, electric: a rotating machine that transforms electrical energy into mechanical energy (section 5-2).

power supply, electrical: the specifications of the required or supplied electricity such as type (AC or DC), volts, amps, cycles, and phase (para. 5-1.3).

rectifier: a device for converting alternating current into direct current (section 5-4).

sensor(s): a device that responds to a physical stimulus and transmits the resulting signal (section 5-3).

switch: a device for making, breaking, or changing the connections in an electric circuit (para. 1-5.4).

switch, master: a manual switch that dominates the operation of contactors, relays, or other remotely operated devices (para. 5-3.1).

1-6 SYMBOLS

The paragraph given after the definition of a symbol refers to the paragraph where the symbol is first used. Each symbol is defined where it is first used.

NOTE: Some symbols may have different definitions within this Standard.

1-6.1 Symbols for Chapter 3 (14)

$2a$ = length of the nonwelded root face in the direction of the thickness of the tension-loaded plate, in. (mm) (para. 3-4.6)

A = cross-sectional area, in.^{2} (mm^2) (para. 3-2.3.1)

a = distance from the edge of the pinhole to the edge of the plate in the direction of the applied load (para. 3-3.3.1)

A_f = area of the compression flange, in.^{2} (mm^2) (para. 3-2.3.1)

A_s = tensile stress area, in.^{2} (mm^2) (para. 3-3.2)

A_v = total area of the two shear planes beyond the pinhole, in.^{2} (mm^2) (para. 3-3.3.1)

B = factor for bending stress in tees and double angles (para. 3-2.3.2)

b = width of a compression element, in. (mm) (Table 3-2.2-1)

b_e = actual net width of a pin-connected plate between the edge of the hole and the edge of the plate on a line perpendicular to the line of action of the applied load, in. (mm) (para. 3-3.3.1)

b_{eff} = effective width to each side of the pinhole, in. (mm) (para. 3-3.3.1)

b_f = width of the compression flange, in. (mm) (para. 3-2.3.2)

C_b = bending coefficient dependent upon moment gradient (para. 3-2.3.2)

C_c = column slenderness ratio separating elastic and inelastic buckling (para. 3-2.2)

C_f = stress category constant for fatigue analysis (para. 3-4.5)

C_{LTB} = lateral-torsional buckling strength coefficient (para. 3-2.3.2)

C_m = coefficient applied to bending term in interaction equation for prismatic member and dependent upon column curvature caused by applied moments (para. 3-2.4)

C_{mx}, C_{my} = coefficient applied to bending term in interaction equation about the x or y axis, as indicated (para. 3-2.4)

C_r = strength reduction factor for pin-connected plates (para. 3-3.3.1)

D = outside diameter of circular hollow section, in. (mm) (Table 3-2.2-1)

d = depth of the section, in. (mm) (para. 3-2.3.1); diameter of roller, in. (mm) (para. 3-3.1)

D_h = hole diameter, in. (mm) (para. 3-3.3.1)

D_p = pin diameter (para. 3-3.3.1)

E = modulus of elasticity
= 29,000 ksi (200 000 MPa) for steel (para. 3-2.2)

Exx = nominal tensile strength of the weld metal, ksi (MPa) (para. 3-3.4.1)

F_a = allowable axial compression stress, ksi (MPa) (para. 3-2.2)

f_a = computed axial compressive stress, ksi (MPa) (para. 3-2.4)

F_b = allowable bending stress, ksi (MPa) (para. 3-2.3.1)

F_{bx}, F_{by} = allowable bending stress about the x or y axis, as indicated, ksi (MPa) (para. 3-2.3.5)

f_{bx}, f_{by} = computed bending stress about the x or y axis, as indicated, ksi (MPa) (para. 3-2.3.5)

F_{cr} = allowable critical stress due to combined shear and normal stresses, ksi (MPa) (para. 3-2.5)

f_{cr} = critical stress, ksi (MPa) (para. 3-2.5)

F_e' = Euler stress for a prismatic member divided by the design factor, ksi (MPa) (para. 3-2.4)

F_{ex}', F_{ey}' = Euler stress about the x or y axis, as indicated, divided by the design factor, ksi (MPa) (para. 3-2.4)

F_p = allowable bearing stress, ksi (MPa) (para. 3-3.1)

F_r = compressive residual stress in flange, ksi (MPa) (Table 3-2.2-1)

F_{sr} = allowable stress range for the detail under consideration, ksi (MPa) (para. 3-4.6)

F_t = allowable tensile stress, ksi (MPa) (para. 3-2.1)

F_t' = allowable tensile stress for a bolt subjected to combined tension and shear stresses, ksi (MPa) (para. 3-3.2)

f_t = computed axial tensile stress, ksi (MPa) (para. 3-2.4)

F_{TH} = threshold value for F_{sr}, ksi (MPa) (para. 3-4.5)

F_u = specified minimum tensile strength, ksi (MPa) (para. 3-2.1)

F_v = allowable shear stress, ksi (MPa) (para. 3-2.3.6)

f_v = computed shear stress, ksi (MPa) (para. 3-2.5)

f_x, f_y = computed normal stress in the x or y direction, as indicated, ksi (MPa) (para. 3-2.5)

F_y = specified minimum yield stress, ksi (MPa) (para. 3-2.1)

F_{yf} = specified minimum yield stress of the flange, ksi (MPa) (Table 3-2.2-1)

F_{yw} = specified minimum yield stress of the web, ksi (MPa) (Table 3-2.2-1)

G = shear modulus of elasticity
= 11,200 ksi (77 200 MPa) for steel (para. 3-2.3.2)

h = clear depth of the plate parallel to the applied shear force at the section under investigation. For rolled shapes, this value may be taken as the clear distance between flanges less the fillet or corner radius, in. (mm) (para. 3-2.3.6)

I_x = major axis moment of inertia, in.^{4} (mm^4) (para. 3-2.3.2)

I_y = minor axis moment of inertia, in.^{4} (mm^4) (para. 3-2.3.2)

J = torsional constant, in.^{4} (mm^4) (para. 3-2.3.1)

K = effective length factor based on the degree of fixity at each end of the member (para. 3-2.2)

l = the actual unbraced length of the member, in. (mm) (para. 3-2.2)

L_b = distance between cross sections braced against twist or lateral displacement of the compression flange; for beams not braced against twist or lateral displacement, the greater of the maximum distance between supports or the distance between the two points of applied load that are farthest apart, in. (mm) (para. 3-2.3.2)

L_p = maximum laterally unbraced length of a bending member for which the full plastic bending capacity can be realized, uniform moment case (C_b = 1.0), in. (mm) (para. 3-2.3.1)

L_r = laterally unbraced length of a bending member above which the limit state will be lateral-torsional buckling, in. (mm) (para. 3-2.3.2)

M = allowable major axis moment for tees and double-angle members loaded in the plane of symmetry, kip-in. (N·mm) (para. 3-2.3.2)

m = number of slip planes in the connection (para. 3-3.2)

M_p = plastic moment, kip-in. (N·mm) (para. 3-2.3.1)

M_1 = smaller bending moment at the end of the unbraced length of a beam taken about the major axis of the member, kip-in. (N·mm) (para. 3-2.3.2)

M_2 = larger bending moment at the end of the unbraced length of a beam taken about the major axis of the member, kip-in. (N-mm) (para. 3-2.3.2)

N = desired design fatigue life in load cycles of the detail being evaluated (para. 3-4.6)

N_d = nominal design factor (para. 3-1.3)

N_{eq} = equivalent number of constant amplitude load cycles at stress range, S_{Rref} (para. 3-4.2)

n_i = number of cycles for the i^{th} portion of a variable amplitude loading spectrum (para. 3-4.2)

P_b = allowable single plane fracture strength beyond the pinhole, kips (N) (para. 3-3.3.1)

P_s = allowable shear capacity of a bolt in a slip-critical connection, kips (N) (para. 3-3.2)

P_t = allowable tensile strength through the pinhole, kips (N) (para. 3-3.3.1)

P_v = allowable double plane shear strength beyond the pinhole, kips (N) (para. 3-3.3.1)

R = distance from the center of the hole to the edge of the plate in the direction of the applied load, in. (mm) (para. 3-3.3.1) variable used in the cumulative fatigue analysis (para. 3-4.6); radius of edge of plate (Table 3-4.4-1)

r = radius of gyration about the axis under consideration, in. (mm) (para. 3-2.2), radius of curvature of the edge of the plate, in. (mm) (Nonmandatory Appendix C, para. C-3.3.1)

R_p = allowable bearing load on rollers, kips/in. (N/mm) (para. 3-3.1)

r_T = radius of gyration of a section comprising the compression flange plus one-third of the compression web area, taken about an axis in the plane of the web, in. (mm) (para. 3-2.3.2)

r_y = minor axis radius of gyration, in. (mm) (para. 3-2.3.1)

S_{Ri} = stress range for the i^{th} portion of variable amplitude loading spectrum, ksi (MPa) (para. 3-4.2)

S_{Rref} = reference stress range to which N_{eq} relates, ksi (MPa) (para. 3-4.2)

S_x = major axis section modulus, in.^{3} (mm^3) (para. 3-2.3.1)

t = thickness of the plate, in. (mm) (para. 3-2.3.3); thickness of a compression element, in. (mm) (Table 3-2.2-1)

t_p = thickness of the tension-loaded plate, in. (mm) (para. 3-4.6)

t_w = thickness of the web, in. (mm) (Table 3-2.2-1)

w = leg size of the reinforcing or contouring fillet, if any, in the direction of the thickness of the tension-loaded plate, in. (mm) (para. 3-4.6)

Z_x = major axis plastic modulus, in.^{3} (mm^3) (para. 3-2.3.1)

Z' = loss of length of the shear planein a pin-connected plate, in. (mm) (Nonmandatory Appendix C, para. C-3.3.1)

ϕ = shear plane locating angle for pin-connected plates (para. 3-3.3.1)

1-6.2 Symbols for Chapter 4 (14)

A = effective area of the vacuum pad enclosed between the pad and the material when the pad is fully compressed against the material surface to be lifted (para. 4-10.1)

C_r = basic dynamic load rating to theoretically endure one million revolutions, per bearing manufacturer, lb (N) (para. 4-6.3)

d = nominal shaft diameter or bearing inside diameter, in. (mm) (para. 4-6.4)

D_t = diametral pitch, in.$^{-1}$ (mm^{-1}) (para. 4-5.3)

F = face width of smaller gear, in. (mm) (para. 4-5.3)

F_a = axial component of the actual bearing load, lb (N) (para. 4-6.3)

F_H = minimum force on each side of the load, lb (N) (para. 4-9.2)

F_r = radial component of the actual bearing load, lb (N) (para. 4-6.3)

F_s = total support force created by the lifter, lb (N) (para. 4-9.2)

H = bearing power factor (para. 4-6.3)

K_A = fatigue stress amplification factor (para. 4-7.6.1)

K_{ST} = stress amplification factor for torsional shear [para. 4-7.6.3(b)]
K_{TB} = stress amplification factor for bending [para. 4-7.6.3(a)]
K_{TD} = stress amplification factor for direct tension [para. 4-7.6.3(a)]
L = bearing length, in. (mm) (para. 4-6.4)
L_G = allowable tooth load in bending, lb (N) (para. 4-5.3)
L_{10} = basic rating life exceeded by 90% of bearings tested, hr (para. 4-6.2)
N = rotational speed, rev./min (para. 4-6.3)
N_v = vacuum pad design factor based on orientation of load (para. 4-10.1)
P = average pressure, psi (MPa) (para. 4-6.4)
P_r = dynamic equivalent radial load, lb (N) (para. 4-6.3)
S = computed combined axial/bending stress, ksi (MPa) [para. 4-7.5(a)]
S_a = computed axial stress, ksi (MPa) [para. 4-7.5(a)]
S_{av} = portion of the computed tensile stress not due to fluctuating loads, ksi (MPa) [para. 4-7.6.3(d)]
S_b = computed bending stress, ksi (MPa) [para. 4-7.5(a)]
S_c = computed combined stress, ksi (MPa) [para. 4-7.5(c)]
S_e = fatigue (endurance) limit of polished, unnotched specimen in reversed bending, ksi (MPa) (para. 4-7.6.2)
S_{ec} = corrected fatigue (endurance) limit of shaft in reversed bending, ksi (MPa) (para. 4-7.6.2)
S_f = computed fatigue stress, ksi (MPa) [para. 4-7.6.3(a)]
S_R = portion of the computed tensile stress due to fluctuating loads, ksi (MPa) [para. 4-7.6.3(d)]
S_t = computed axial tensile stress, ksi (MPa) [para. 4-7.6.3(a)]
S_u = specified minimum ultimate tensile strength, ksi (MPa) [para. 4-7.5(a)]
S_y = specified minimum yield strength, ksi (MPa) [para. 4-7.6.3(d)]
UPC = calculated ultimate vacuum pad capacity (para. 4-10.1)
V = surface velocity of shaft, ft/min (m/sec) (para. 4-6.4)
V_p = minimum vacuum level specified at the pad (para. 4-10.1)
VPR = maximum calculated pad rating (para. 4-10.1)
W = bearing load, lb (N) (para. 4-6.4)
X = dynamic radial load factor per bearing manufacturer (para. 4-6.3)
Y = Lewis form factor (Table 4-5.3-1); dynamic axial load factor per bearing manufacturer (para. 4-6.3)
θ = angle of vacuum pad interface surface measured from horizontal (para. 4-10.1)
σ_y = specified minimum yield stress, psi (MPa) (para. 4-5.3)
τ = computed combined shear stress, ksi (MPa) [para. 4-7.5(b)]
τ_{av} = portion of the computed shear stress not due to the fluctuating loads, ksi (MPa) [para. 4-7.6.3(d)]
τ_f = computed combined fatigue shear stress, ksi (MPa) [para. 4-7.6.3(b)]
τ_R = portion of the computed shear stress due to fluctuating loads, ksi (MPa) [para. 4-7.6.3(d)]
τ_T = computed torsional shear stress, ksi (MPa) [para. 4-7.5(b)]
τ_V = computed transverse shear stress, ksi (MPa) [para. 4-7.5(b)]

1-7 REFERENCES (14)

The following is a list of publications referenced in this Standard.

ANSI/AGMA 2001-C95, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth[1]

Publisher: American Gear Manufacturers Association (AGMA), 500 Montgomery Street, Alexandria, VA 22314-1581 (www.agma.org)

AWS D14.1/D14.1M-2005, Specification for Welding of Industrial and Mill Cranes and Other Material Handling Equipment[1]

Publisher: American Welding Society (AWS), 8669 NW 36 Street, Doral, FL 33166 (www.aws.org)

ANSI/NFPA 70-2011, National Electrical Code[1]

Publisher: National Fire Protection Association (NFPA), 1 Batterymarch Park, Quincy, MA 02169-7471 (www.nfpa.org)

ASME B17.1-1967 (R2008), Keys and Keyseats

ASME B30.20-2013, Below-the-Hook Lifting Devices

Publisher: The American Society of Mechanical Engineers (ASME), Two Park Avenue, New York, NY 10016; Order Department: 22 Law Drive, P.O. Box 2900, Fairfield, NJ 07007-2900 (www.asme.org)

ASTM A325, Standard Specification for Structural Bolts, Steel, Heat Treated, 120/105 ksi Minimum Tensile Strength

[1] May also be obtained from the American National Standards Institute (ANSI), 25 West 43rd Street, New York, NY 10036.

ASTM A490, Standard Specification for Structural Bolts, Alloy Steel, Heat Treated, 150 ksi Minimum Tensile Strength

Publisher: American Society for Testing and Materials (ASTM International), 100 Barr Harbor Drive, West Conshohocken, PA 19428-2959 (www.astm.org)

DIN 6885-1, Drive Type Fastenings Without Taper Action; Parallel Keys, Keyways, Deep Pattern

Publisher: Deutsches Institut für Normung, e. V. (DIN), Am DIN-Platz Burggrafenstraße 6, 10787 Berlin, Germany (www.din.de)

ICS 2-2000 (R2005), Industrial Control and Systems: Controllers, Contactors, and Overload Relays Rated 600 Volts

ICS 6-1993 (R2001, R2006), Industrial Control and Systems: Enclosures

MG 1-2006, Revision 1-2007, Motors and Generators

Publisher: National Electrical Manufacturers Association (NEMA), 1300 North 17th Street, Rosslyn, VA 22209 (www.nema.org)

Pilkey, W. D., 2008, Peterson's Stress Concentration Factors, 3rd edition

Publisher: John Wiley & Sons, Inc., 111 River Street, Hoboken, NJ 07030-5774 (www.wiley.com)

Chapter 2
Lifter Classifications

2-1 GENERAL

A Design Category and Service Class shall be designated for each lifter.

2-1.1 Selection

The selection of a Design Category (static strength criteria) and Service Class (fatigue life criteria) described in sections 2-2 and 2-3 shall be based on the operating conditions (use) and expected life of the lifter.

2-1.2 Responsibility

The selection of Design Category and Service Class shall be the responsibility of a qualified person representing the owner, purchaser, or user of the lifting device. If not specified by the owner, purchaser, or user, the Design Category and Service Class shall be designated by the qualified person responsible for the design.

2-1.3 Identification

The Design Category and Service Class shall be marked on the lifter and appear on quotations, drawings, and documentation associated with the lifter.

2-1.4 Environment

All lifter components are assumed to operate within the temperature range defined in para. 1-4.7 and normal atmospheric conditions (free from excessive dust, moisture, and corrosive environments). Lifter components operating at temperatures outside the range specified in para. 1-4.7 may require additional consideration.

2-2 DESIGN CATEGORY

The design categories defined in paras. 2-2.1 and 2-2.2 provide for different design factors that establish the stress limits to be used in the design. The design factors are given in para. 3-1.3.

Table 2-3-1 Service Class

Service Class	Load Cycles
0	0–20,000
1	20,001–100,000
2	100,001–500,000
3	500,001–2,000,000
4	Over 2,000,000

Lifters shall be designed to Design Category B, unless a qualified person determines that Design Category A is appropriate.

2-2.1 Design Category A

(a) Design Category A should be designated when the magnitude and variation of loads applied to the lifter are predictable, where the loading and environmental conditions are accurately defined or not severe.

(b) Design Category A lifting devices shall be limited to Service Class 0.

(c) The nominal design factor for Design Category A shall be in accordance with para. 3-1.3.

2-2.2 Design Category B

(a) Design Category B should be designated when the magnitude and variation of loads applied to the lifter are not predictable, where the loading and environmental conditions are severe or not accurately defined.

(b) The nominal design factor for Design Category B shall be in accordance with para. 3-1.3.

2-3 SERVICE CLASS

The Service Class of the lifter shall be determined from Table 2-3-1 based on the specified fatigue life (load cycles). The selected Service Class establishes allowable stress range values for structural members (section 3-4) and design parameters for mechanical components (sections 4-6 and 4-7).

Chapter 3
Structural Design

3-1 GENERAL

3-1.1 Purpose

This chapter sets forth design criteria for prismatic structural members and connections of a below-the-hook lifting device.

3-1.2 Loads

Below-the-hook lifting devices shall be designed to resist the actual applied loads. These loads shall include the rated load, the weights of the individual components of the lifter, and other forces created by the operation of the lifter, such as gripping force or lateral loads. Resolution of these loads into member and connection forces shall be performed by an accepted structural analysis method.

(14) 3-1.3 Static Design Basis

3-1.3.1 Nominal Design Factors. The static strength design of a below-the-hook lifting device shall be based on the allowable stresses defined in sections 3-2 and 3-3. The minimum values of the nominal design factor, N_d, in the allowable stress equations shall be as follows:

N_d = 2.00 for Design Category A lifters
= 3.00 for Design Category B lifters

3-1.3.2 Other Design Conditions. Allowable stresses for design conditions not addressed herein shall be based on the following design factors:

(a) Design factors for Design Category A lifting devices shall be not less than 2.00 for limit states of yielding or buckling and 2.40 for limit states of fracture and for connection design.

(b) Design factors for Design Category B lifting devices shall be not less than 3.00 for limit states of yielding or buckling and 3.60 for limit states of fracture and for connection design.

(14) 3-1.4 Fatigue Design Basis

Members and connections subject to repeated loading shall be designed so that the maximum stress does not exceed the values given in sections 3-2 and 3-3, and the maximum range of stress does not exceed the values given in section 3-4. Members and connections subjected to fewer than 20,000 load cycles (Service Class 0) need not be analyzed for fatigue.

3-1.5 Curved Members

The design of curved members that are subjected to bending in the plane of the curve shall account for the increase in maximum bending stress due to the curvature, as applicable.

The stress increase due to member curvature need not be considered for flexural members that can develop the full plastic moment when evaluating static strength. This stress increase shall be considered when evaluating fatigue.

3-1.6 Allowable Stresses

All structural members, connections, and connectors shall be proportioned so the stresses due to the loads stipulated in para. 3-1.2 do not exceed the allowable stresses and stress ranges specified in sections 3-2, 3-3, and 3-4. The allowable stresses specified in these sections do not apply to peak stresses in regions of connections, provided the requirements of section 3-4 are satisfied.

3-2 MEMBER DESIGN

3-2.1 Tension Members

The allowable tensile stress, F_t, shall not exceed the value given by eq. (3-1) on the gross area nor the value given by eq. (3-2) on the effective net tensile area.

$$F_t = \frac{F_y}{N_d} \tag{3-1}$$

$$F_t = \frac{F_u}{1.20N_d} \tag{3-2}$$

where

F_u = specified minimum ultimate tensile strength
F_y = specified minimum yield stress

Refer to para. 3-3.3 for pinned connection design requirements.

3-2.2 Compression Members

The allowable axial compression stress, F_a, on the gross area where all of the elements of the section meet the noncompact provisions of Table 3-2.2-1 and when the largest slenderness ratio, Kl/r, is less than C_c is

$$F_a = \frac{\left[1 - \frac{(Kl/r)^2}{2C_c^2}\right]F_y}{N_d\left[1 + \frac{9(Kl/r)}{40C_c} - \frac{3(Kl/r)^3}{40C_c^3}\right]} \tag{3-3}$$

Table 3-2.2-1 Limiting Width–Thickness Ratios for Compression Elements

Description of Element	Width–Thickness Ratio	Limiting Width–Thickness Ratios: Compact	Limiting Width–Thickness Ratios: Noncompact
Flanges of I-shaped rolled beams and channels in flexure	b/t	$0.38\sqrt{E/F_y}$	$0.83\sqrt{E/F_L}$ [Note (1)]
Flanges of I-shaped hybrid or welded beams in flexure	b/t	$0.38\sqrt{E/F_{yf}}$	$0.95\sqrt{k_c E/F_L}$ [Notes (1), (2)]
Flanges projecting from built-up compression members	b/t	. . .	$0.64\sqrt{k_c E/F_y}$ [Note (2)]
Flanges of I-shaped sections in pure compression, plates projecting from compression elements, outstanding legs of pairs of angles in continuous contact; flanges of channels in pure compression	b/t	. . .	$0.56\sqrt{E/F_y}$
Legs of single angle struts; legs of double angle struts with separators; unstiffened elements, i.e., supported along one edge	b/t	. . .	$0.45\sqrt{E/F_y}$
Stems of tees	d/t	. . .	$0.75\sqrt{E/F_y}$
Flanges of rectangular box and hollow structural sections of uniform thickness subject to bending or compression; flange cover plates and diaphragm plates between lines of fasteners or welds	b/t	$1.12\sqrt{E/F_y}$	$1.40\sqrt{E/F_y}$
Unsupported width of cover plates perforated with a succession of access holes [Note (3)]	b/t	. . .	$1.86\sqrt{E/F_y}$
Webs in flexural compression [Note (4)]	h/t_w	$3.76\sqrt{E/F_y}$ [Note (5)]	$5.70\sqrt{E/F_y}$ [Note (5)]
Webs in combined flexural and axial compression	h/t_w	For $N_d f_a/F_y \le 0.125$ [Note (5)] $3.76\sqrt{\frac{E}{F_y}}\left(1 - 2.75\frac{N_d f_a}{F_y}\right)$ For $N_d f_a/F_y > 0.125$ [Note (5)] $1.12\sqrt{\frac{E}{F_y}}\left(2.33 - \frac{N_d f_a}{F_y}\right)$ $\ge 1.49\sqrt{E/F_y}$	$5.70\sqrt{\frac{E}{F_y}}\left(1 - 0.74\frac{N_d f_a}{F_y}\right)$ [Note (5)]
All other uniformly compressed stiffened elements; i.e., supported along two edges	b/t h/t_w	. . .	$1.49\sqrt{E/F_y}$
Circular hollow sections In axial compression In flexure	D/t	 . . . $0.07E/F_y$	 $0.11E/F_y$ $0.31E/F_y$

NOTES:

(1) The following values apply:

F_L = smaller of $(F_{yf} - F_r)$ or F_{yw}, ksi (MPa)
F_r = compressive residual stress in flange
= 10 ksi (69 MPa) for rolled shapes
= 16.5 ksi (114 MPa) for welded shapes

(2) The following values apply:

$$k_c = \frac{4}{\sqrt{h/t_w}} \text{ and } 0.35 \le k_c \le 0.763$$

(3) Assumes net area of plate at the widest hole.

(4) For hybrid beams, use the yield stress of the flange F_{yf}.

(5) Valid only when flanges are of equal size.

$$C_c = \sqrt{\frac{2\pi^2 E}{F_y}} \tag{3-4}$$

When Kl/r exceeds C_c, the allowable axial compressive stress on the gross section is

$$F_a = \frac{\pi^2 E}{1.15 N_d \,(Kl/r)^2} \tag{3-5}$$

where

E = modulus of elasticity
K = effective length factor based on the degree of fixity at each end of the member
l = the actual unbraced length of the member
r = radius of gyration about the axis under consideration

3-2.3 Flexural Members

(14) **3-2.3.1 Major Axis Bending of Compact Sections.** The allowable bending stress, F_b, for members with compact sections as defined by Table 3-2.2-1 symmetrical about, and loaded in, the plane of the minor axis, with the flanges continuously connected to the web or webs, and laterally braced at intervals not exceeding L_p as defined by eq. (3-7) for I-shape members and by eq. (3-8) for box members is

$$F_b = \frac{1.10 F_y}{N_d} \tag{3-6}$$

$$L_p = 1.76 r_y \sqrt{\frac{E}{F_y}} \le \frac{0.67E}{F_y\, d/A_f} \tag{3-7}$$

$$L_p = \frac{0.13 r_y E}{M_p}\sqrt{JA} \tag{3-8}$$

where

A = cross-sectional area
A_f = area of the compression flange
d = depth of the section
J = torsional constant
M_p = plastic moment
= $F_y Z_x \le 1.5 F_y S_x$ for homogeneous sections
r_y = minor axis radius of gyration
S_x = major axis section modulus
Z_x = major axis plastic modulus

For circular tubes with compact walls as defined by Table 3-2.2-1 or square tubes or square box sections with compact flanges and webs as defined by Table 3-2.2-1 and with the flanges continuously connected to the webs, the allowable bending stress is given by eq. (3-6) for any length between points of lateral bracing.

(14) **3-2.3.2 Major Axis and Minor Axis Bending of Compact Sections With Unbraced Length Greater Than L_p and Noncompact Sections.** The allowable bending stress for members with compact or noncompact sections as defined by Table 3-2.2-1, loaded through the shear center, bent about either the major or minor axis, and laterally braced at intervals not exceeding L_r for major axis bending as defined by eq. (3-10) for I-shape members and by eq. (3-11) for box members is given by eq. (3-9). For channels bent about the major axis, the allowable bending stress is given by eq. (3-17).

$$F_b = \frac{F_y}{N_d} \tag{3-9}$$

$$L_r = \sqrt{\frac{3.19 r_T^2 E C_b}{F_y}} \tag{3-10}$$

$$L_r = \frac{2 r_y E \sqrt{JA}}{F_y S_x} \tag{3-11}$$

$$C_b = 1.75 + 1.05(M_1/M_2) + 0.3(M_1/M_2)^2 \le 2.3 \tag{3-12}$$

where M_1 is the smaller and M_2 is the larger bending moment at the ends of the unbraced length, taken about the major axis of the member, and where M_1/M_2 is positive when M_1 and M_2 have the same sign (reverse curvature bending). C_b may be conservatively taken as unity. When the bending moment at any point within an unbraced length is larger than that at both ends of this length, C_b shall be taken as unity [see eq. (3-12)].

For I-shape members and channels bent about the major axis and with unbraced lengths that fall in the ranges defined by either eq. (3-13) or (3-15), the allowable bending stress in tension is given by eq. (3-9). For an I-shape member for which the unbraced length of the compression flange falls into the range defined by eq. (3-13), the allowable bending stress in compression is the larger of the values given by eqs. (3-14) and (3-17). For an I-shape member for which the unbraced length of the compression flange falls into the range defined by eq. (3-15), the allowable bending stress in compression is the larger of the values given by eqs. (3-16) and (3-17). Equation (3-17) is applicable only to sections with a compression flange that is solid, approximately rectangular in shape, and that has an area not less than the tension flange. For channels bent about the major axis, the allowable compressive stress is given by eq. (3-17).

$$\sqrt{\frac{3.19 E C_b}{F_y}} \le \frac{L_b}{r_T} \le \sqrt{\frac{17.59 E C_b}{F_y}} \tag{3-13}$$

$$F_b = \left[1.10 - \frac{F_y (L_b/r_T)^2}{31.9 E C_b}\right]\frac{F_y}{N_d} \le \frac{F_y}{N_d} \tag{3-14}$$

$$\frac{L_b}{r_T} > \sqrt{\frac{17.59 E C_b}{F_y}} \tag{3-15}$$

$$F_b = C_{LTB} \frac{\pi^2 E C_b}{N_d \,(L_b/r_T)^2} \le \frac{F_y}{N_d} \tag{3-16}$$

For any value of L_b/r_T

$$F_b = C_{LTB}\frac{0.66EC_b}{N_d(L_b d/A_f)} \le \frac{F_y}{N_d} \quad (3\text{-}17)$$

where

b_f = width of the compression flange

C_{LTB} = 1.00 for beams braced against twist or lateral displacement of the compression flange at the ends of the unbraced length

= $\frac{2.00\sqrt{EI_x/GJ}}{(L_b/b_f)^2} + 0.275 \le 1.00$ for beams not braced against twist or lateral displacement of the compression flange at the ends of the unbraced length

I_x = major axis moment of inertia

L_b = distance between cross-sections braced against twist or lateral displacement of the compression flange; for beams not braced against twist or lateral displacement, the greater of the maximum distance between supports or the distance between the two points of applied load that are farthest apart

r_T = radius of gyration of a section comprising the compression flange plus one-third of the compression web area, taken about an axis in the plane of the web

The allowable major axis moment, M, for tees and double-angle members loaded in the plane of symmetry is

$$M = C_{LTB}\frac{\pi}{N_d}\frac{\sqrt{E\,I_y GJ}}{L_b}\left(B + \sqrt{1 + B^2}\right) \le \frac{F_y a S_x}{N_d} \quad (3\text{-}18)$$

where

a = 1.0 if the stem is in compression

= 1.25 if the stem is in tension

B = $\pm 2.3(d/L_b)\sqrt{I_y/J}$

C_{LTB} = 1.00 for beams braced against twist or lateral displacement of the compression element at the ends of the unbraced length

= $\sqrt{\frac{0.25\sqrt{EI_x/GJ}}{L_b/b_f}} \le 1.00$ for beams not braced against twist or lateral displacement of the compression flange at the ends of the unbraced length if the stem is in tension

= $\sqrt{\frac{0.50\sqrt{EI_x/GJ}}{L_b/b_f}} \le 1.00$ for beams not braced against twist or lateral displacement of the compression flange at the ends of the unbraced length if the stem is in compression

G = shear modulus of elasticity

I_y = minor axis moment of inertia

The value B is positive when the stem is in tension and negative when the stem is in compression anywhere along the unbraced length.

3-2.3.3 Major Axis Bending of Solid Rectangular Bars. (14) The allowable bending stress for a rectangular section of depth, d, and thickness, t, is given as follows:

$$\frac{L_b d}{t^2} \le \frac{0.08E}{F_y} \quad (3\text{-}19)$$

$$F_b = \frac{1.25F_y}{N_d} \quad (3\text{-}20)$$

$$\frac{0.08E}{F_y} < \frac{L_b d}{t^2} \le \frac{1.9E}{F_y} \quad (3\text{-}21)$$

$$F_b = C_{LTB} \times C_b\left[1.52 - 0.274\left(\frac{L_b d}{t^2}\right)\frac{F_y}{E}\right]\frac{F_y}{N_d} \le \frac{1.25F_y}{N_d} \quad (3\text{-}22)$$

$$\frac{L_b d}{t^2} > \frac{1.9E}{F_y} \quad (3\text{-}23)$$

$$F_b = C_{LTB} \times \frac{1.9EC_b}{N_d(L_b d/t^2)} \le \frac{1.25F_y}{N_d} \quad (3\text{-}24)$$

where

C_{LTB} = 1.00 for beams braced against twist or lateral displacement of the compression element at the ends of the unbraced length

= $\frac{3.00\sqrt{EI_x/GJ}}{L_b/t} \le 1.00$ for beams not braced against twist or lateral displacement of compression element at the ends of unbraced length

3-2.3.4 Minor Axis Bending of Compact Sections, Solid Bars, and Rectangular Sections. (14) For doubly symmetric I- and H-shape members with compact flanges as defined by Table 3-2.2-1 continuously connected to the web and bent about their minor axes, solid round and square bars, and solid rectangular sections bent about their minor axes, the allowable bending stress is

$$F_b = \frac{1.25\,F_y}{N_d} \quad (3\text{-}25)$$

For rectangular tubes or box shapes with compact flanges and webs as defined by Table 3-2.2-1, with the flanges continuously connected to the webs, and bent about their minor axes, the allowable bending stress is given by eq. (3-6).

3-2.3.5 Biaxial Bending. Members other than cylindrical members subject to biaxial bending with no axial load shall be proportioned to satisfy eq. (3-26).

Cylindrical members subject to biaxial bending with no axial load shall be proportioned to satisfy eq. (3-27).

$$\frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1.0 \tag{3-26}$$

$$\frac{\sqrt{f_{bx}^2 + f_{by}^2}}{F_b} \leq 1.0 \tag{3-27}$$

f_{bx} or f_{by} = computed bending stress about the x or y axis, as indicated
F_{bx} or F_{by} = allowable bending stress about the x or y axis, as indicated, from para. 3-2.3

3-2.3.6 Shear on Bars, Pins, and Plates. The average shear stress F_v on bars, pins, and plates for which $h/t \leq 2.45\sqrt{E/F_y}$ shall not exceed

$$F_v = \frac{F_y}{N_d\sqrt{3}} \tag{3-28}$$

where
h = clear depth of the plate parallel to the applied shear force at the section under investigation. For rolled shapes, this value may be taken as the clear distance between flanges less the fillet or corner radius.
t = thickness of the plate

Methods used to determine the strength of plates subjected to shear forces for which $h/t > 2.45\sqrt{E/F_y}$ shall provide a design factor with respect to the limit state of buckling not less than the applicable value given in para. 3-1.3.

3-2.4 Combined Axial and Bending Stresses

Members subject to combined axial compression and bending stresses shall be proportioned to satisfy the following requirements:

(a) All members except cylindrical members shall satisfy eqs. (3-29) and (3-30) or (3-31).

(b) When $f_a/F_a \leq 0.15$, eq. (3-31) is permitted in lieu of eqs. (3-29) and (3-30).

$$\frac{f_a}{F_a} + \frac{C_{mx} f_{bx}}{\left(1 - \frac{f_a}{F_{ex}'}\right)F_{bx}} + \frac{C_{my} f_{by}}{\left(1 - \frac{f_a}{F_{ey}'}\right)F_{by}} \leq 1.0 \tag{3-29}$$

$$\frac{f_a}{F_y/N_d} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1.0 \tag{3-30}$$

$$\frac{f_a}{F_a} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1.0 \tag{3-31}$$

(c) Cylindrical members shall satisfy eqs. (3-32) and (3-33) or (3-34).

(d) When $f_a/F_a \leq 0.15$, eq. (3-34) is permitted in lieu of eqs. (3-32) and (3-33).

$$\frac{f_a}{F_a} + \frac{C_m\sqrt{f_{bx}^2 + f_{by}^2}}{\left(1 - \frac{f_a}{F_e'}\right)F_b} \leq 1.0 \tag{3-32}$$

$$\frac{f_a}{F_y/N_d} + \frac{\sqrt{f_{bx}^2 + f_{by}^2}}{F_b} \leq 1.0 \tag{3-33}$$

$$\frac{f_a}{F_a} + \frac{\sqrt{f_{bx}^2 + f_{by}^2}}{F_b} \leq 1.0 \tag{3-34}$$

(e) Members subject to combined axial tension and bending stresses shall be proportioned to satisfy the following equations. Equation (3-35) applies to all members except cylindrical members. Equation (3-36) applies to cylindrical members.

$$\frac{f_t}{F_t} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1.0 \tag{3-35}$$

$$\frac{f_t}{F_t} + \frac{\sqrt{f_{bx}^2 + f_{by}^2}}{F_b} \leq 1.0 \tag{3-36}$$

In eqs. (3-29) through (3-36),
F_a = allowable axial compressive stress from para. 3-2.2
f_a = computed axial compressive stress

$$F_e' = \frac{\pi^2 E}{1.15N_d(Kl/r)^2}$$

F_t = allowable tensile stress from para. 3-2.1
f_t = computed axial tensile stress

where the slenderness ratio, Kl/r, is that in the plane of bending under consideration

$$C_m = C_{mx} = C_{my} = 1.0$$

Lower values for C_m, C_{mx}, and C_{my} may be used if justified by analysis.

3-2.5 Combined Normal and Shear Stresses

Regions of members subject to combined normal and shear stresses shall be proportioned such that the critical stress f_{cr} computed with eq. (3-37) does not exceed the allowable stress F_{cr} defined in the equation.

$$f_{cr} = \sqrt{f_x^2 - f_x f_y + f_y^2 + 3f_v^2} \leq F_{cr} = \frac{F_y}{N_d} \tag{3-37}$$

where
F_{cr} = allowable critical stress due to combined shear and normal stresses
f_v = computed shear stress
f_x = computed normal stress in the x direction
f_y = computed normal stress in the y direction

3-2.6 Local Buckling

The width–thickness ratios of compression elements shall be less than or equal to the values given in Table 3-2.2-1 to be fully effective.

Methods used to determine the strength of slender compression elements shall provide a design factor with respect to the limit state of buckling no less than the applicable value given in para. 3-1.3.

3-3 CONNECTION DESIGN

3-3.1 General

In connection design, bolts shall not be considered as sharing stress in combination with welds. When the gravity axes of connecting, axially stressed members do not intersect at one point, provision shall be made for bending and shear stresses due to eccentricity in the connection.

The allowable bearing stress, F_p, on the contact area of milled surfaces, fitted bearing stiffeners, and other steel parts in static contact is

$$F_p = \frac{1.8F_y}{1.20N_d} \tag{3-38}$$

The allowable bearing load, R_p, in kips per inch of length (N/mm) on rollers is

$$R_p = \frac{a}{1.20N_d}\left(\frac{F_y - f}{20}\right)c \tag{3-39}$$

where

a = 1.2 if $d \leq 25$ in. (635 mm)
= 6.0 if $d > 25$ in. when using U.S. Customary units (F_y, ksi)
= 30.2 if $d > 635$ mm when using SI units (F_y, MPa)

c = d if $d \leq 25$ in. (635 mm)
= $\sqrt{d}$ if $d > 25$ in. (635 mm)

d = diameter of roller

f = 13 when using U.S. Customary units (F_y, ksi)
= 90 when using SI units (F_y, MPa)

F_y = lower yield stress of the parts in contact

3-3.2 Bolted Connections

A bolted connection shall consist of a minimum of two bolts. Bolt spacing and edge distance shall be determined by an accepted design approach so as to provide a minimum design factor of $1.20N_d$ with respect to fracture of the connected parts in tension, shear, or block shear.

The allowable tensile stress, F_t, of the bolt is

$$F_t = \frac{F_u}{1.20N_d} \tag{3-40}$$

The actual tensile stress, f_t, shall be based on the tensile stress area of the bolt and the bolt tension due to the applied loads as defined in para. 3-1.2.

The allowable shear stress, F_v, of the bolt is

$$F_v = \frac{0.62F_u}{1.20N_d} \tag{3-41}$$

The actual shear stress, f_v, shall be based on the gross area of the bolt if the shear plane passes through the bolt shank, or the root area if the shear plane passes through the threaded length of the bolt and the bolt shear due to the applied loads as defined in para. 3-1.2.

The allowable bearing stress, F_p, of the connected part on the projected area of the bolt is

$$F_p = \frac{2.40F_u}{1.20N_d} \tag{3-42}$$

where

F_u = the specified minimum ultimate tensile strength of the connected part

The allowable tensile stress, F_t', for a bolt subjected to combined tension and shear stresses is

$$F_t' = \sqrt{F_t^2 - 2.60f_v^2} \tag{3-43}$$

The allowable shear capacity, P_s, of a bolt in a slip-critical connection in which the faying surfaces are clean and unpainted is

$$P_s = m\frac{0.26A_sF_u}{1.20N_d} \tag{3-44}$$

where

A_s = tensile stress area

m = number of slip planes in the connection

The hole diameters for bolts in slip-critical connections shall not be more than $\frac{1}{16}$ in. (2 mm) greater than the bolt diameter. If larger holes are necessary, the capacity of the connection shall be reduced accordingly.

The slip resistance of connections in which the faying surfaces are painted or otherwise coated shall be determined by testing.

Bolts in slip-critical connections shall be tightened during installation to provide an initial tension equal to at least 70% of the specified minimum tensile strength of the bolt. A hardened flat washer shall be used under the part turned (nut or bolt head) during installation. Washers shall be used under both the bolt head and nut of ASTM A490 bolts when the connected material has a specified minimum yield stress less than 40 ksi (276 MPa). Only ASTM A325 or ASTM A490 bolts shall be used in slip-critical connections.

Bolted connections subjected to cyclic shear loading shall be designed as slip-critical connections unless the shear load is transferred between the connected parts by means of dowels, keys, or other close-fit elements.

3-3.3 Pinned Connections

3-3.3.1 Static Strength of the Plates. The strength of a pin-connected plate in the region of the pinhole shall be taken as the least value of the tensile strength of the effective area on a plane through the center of the pinhole perpendicular to the line of action of the applied load, the fracture strength beyond the pinhole on a single plane parallel to the line of action of the applied load, and the double plane shear strength beyond the pinhole parallel to the line of action of the applied load.

The allowable tensile strength through the pinhole, P_t, shall be calculated as follows:

$$P_t = C_r \frac{F_u}{1.20N_d} 2tb_{eff} \tag{3-45}$$

where

b_{eff} = effective width to each side of the pinhole

$$C_r = 1 - 0.275 \sqrt{1 - \frac{D_p^2}{D_h^2}} \tag{3-46}$$

where

D_h = hole diameter

D_p = pin diameter

The value of C_r may be taken as 1.00 for values of D_p/D_h greater than 0.90.

The effective width shall be taken as the smaller of the values calculated as follows:

$$b_{eff} = 4t \le b_e \tag{3-47}$$

$$b_{eff} = b_e 0.6 \frac{F_u}{F_y} \sqrt{\frac{D_h}{b_e}} \le b_e \tag{3-48}$$

where

b_e = actual width of a pin-connected plate between the edge of the hole and the edge of the plate on a line perpendicular to the line of action of the applied load

The width limit of eq. (3-47) does not apply to plates that are stiffened or otherwise prevented from buckling out of plane.

The allowable single plane fracture strength beyond the pinhole P_b is

$$P_b = C_r \frac{F_u}{1.20N_d}\left[1.13\left(R - \frac{D_h}{2}\right) + \frac{0.92b_e}{1 + b_e/D_h}\right]t \tag{3-49}$$

where

R = distance from the center of the hole to the edge of the plate in the direction of the applied load

The allowable double plane shear strength beyond the pinhole P_v is

$$P_v = \frac{0.70F_u}{1.20\ N_d} A_v \tag{3-50}$$

where

A_v = total area of the two shear planes beyond the pinhole

$$A_v = 2\left[a + \frac{D_p}{2}(1 - \cos\phi)\right]t \tag{3-51}$$

where

a = distance from the edge of the pinhole to the edge of the plate in the direction of the applied load, and

$$\phi = 55 \frac{D_p}{D_h} \tag{3-52}$$

3-3.3.2 Combined Stresses. If a pinhole is located at a point where significant stresses are induced from member behavior such as tension or bending, local stresses from the function as a pinned connection shall be combined with the gross member stresses in accordance with paras. 3-2.4 and 3-2.5.

3-3.3.3 Fatigue Loading. The average tensile stress on the net area through the pinhole shall not exceed the limits defined in para. 3-4.3 for Stress Category E.

Pinholes in connections designed for Service Classes 1 through 4 shall be drilled, reamed, or otherwise finished to provide a maximum surface roughness of 500 μin. (12.5 μm) around the inside surface of the hole.

3-3.3.4 Bearing Stress. The bearing stress between the pin and the plate, based on the projected area of the pin, shall not exceed the value given by eq. (3-53), where F_y is the yield stress of the pin or plate, whichever is smaller. The bearing stress between the pin and the plate in connections that will rotate under load for a large number of load cycles (Service Class 1 or higher) shall not exceed the value given by eq. (3-54). **(14)**

$$F_p = \frac{1.25F_y}{N_d} \tag{3-53}$$

$$F_p = \frac{0.63F_y}{N_d} \tag{3-54}$$

3-3.3.5 Pin-to-Hole Clearance. Pin-to-hole clearance in connections that will rotate under load or that will experience load reversal in service for a large number of load cycles (Service Class 1 or higher) shall be as required to permit proper function of the connection.

3-3.3.6 Pin Design. Shear forces and bending moments in the pin shall be computed based on the geometry of the connection. Distribution of the loads between the plates and the pin may be assumed to be uniform or may account for the effects of local deformations.

3-3.4 Welded Connections

(14) **3-3.4.1 General.** For purposes of this section, fillet or groove welds loaded parallel to the axis of the weld shall be designed for shear forces. Groove welds loaded perpendicular to the axis of the weld shall be designed for tension or compression forces. Welded connection design shall provide adequate access for depositing the weld metal. The strength of a weld is governed by either the base material or the deposited weld material as follows:

(a) The design strength of groove welds subject to tension or compression shall be equal to the effective area of the weld multiplied by the allowable stress of the base metal defined in section 3-2.

(b) The design strength of fillet or partial-joint-penetration groove welds subject to shear shall be equal to the effective area of the weld multiplied by the allowable stress F_v given by eq. (3-55). Stresses in the base metal shall not exceed the limits defined in section 3-2.

$$F_v = \frac{0.60Exx}{1.20N_d} \quad (3\text{-}55)$$

where

Exx = nominal tensile strength of the weld metal

(c) The design strength of complete-joint-penetration groove welds subject to shear shall be based on the strength of the base metal.

(d) *Combination of Welds.* If two or more of the general types of welds (paras. 3-3.4.2 through 3-3.4.4) are combined in a single joint, the effective capacity of each shall be separately computed with reference to the axis of the group in order to determine the allowable capacity of the combination.

Effective areas and limitations for groove, fillet, plug, and slot welds are indicated in paras. 3-3.4.2 through 3-3.4.4.

3-3.4.2 Groove Welds. Groove welds may be either complete-joint-penetration or partial-joint-penetration type welds. The effective weld area for either type is defined as the effective length of weld multiplied by the effective throat thickness.

The effective length of any groove weld is the length over which the weld cross-section has the proper effective throat thickness. Intermittent groove welds are not permitted.

The effective throat thickness is the minimum distance from the root of the groove to the face of the weld, less any reinforcement (usually the depth of groove). For a complete-penetration groove weld, the effective throat thickness is the thickness of the thinner part joined. In partial-penetration groove welds, the effective throat thickness for J- or U-grooves and for bevel or V-grooves with a minimum angle of 60 deg is the depth of groove. For V-grooves from 45 deg to 60 deg, the effective throat thickness is the depth of groove minus ⅛ in. (3 mm).

Table 3-3.4.2-1 Minimum Effective Throat Thickness of Partial-Penetration Groove Welds

Material Thickness of Thicker Part Joined, in. (mm)	Minimum Effective Throat Thickness, in. (mm)
To ¼ (6)	⅛ (3)
Over ¼ (6) to ½ (13)	3/16 (5)
Over ½ (13) to ¾ (19)	¼ (6)
Over ¾ (19) to 1½ (38)	5/16 (8)
Over 1½ (38) to 2¼ (57)	⅜ (10)
Over 2¼ (57) to 6 (150)	½ (13)
Over 6 (150)	⅝ (16)

GENERAL NOTE: The effective throat does not need to exceed the thickness of the thinner part joined.

The minimum partial-penetration groove weld effective throat thickness is given in Table 3-3.4.2-1. The minimum throat thickness is determined by the thicker part joined. However, in no case shall the effective throat thickness be less than the size required to transmit the calculated forces.

For bevel and V-groove flare welds, the effective throat thickness is based on the radius of the bar or bend to which it is attached and the flare weld type. For bevel welds, the effective throat thickness is 5/16 times the radius of the bar or bend. For V-groove welds, the effective throat thickness is ½ times the radius of the bar or bend.

3-3.4.3 Fillet Welds. Fillet weld size is specified by leg width, but stress is determined by effective throat thickness. The effective throat of a fillet weld shall be the shortest distance from the root to the face of the weld. In general, this effective throat thickness is considered to be on a 45-deg angle from the leg and have a dimension equal to 0.707 times the leg width. The effective weld area of a fillet weld is defined as the effective length of weld multiplied by the effective throat thickness.

The effective length of a fillet weld shall be the overall length of the full-size fillet including end returns. Whenever possible, a fillet weld shall be terminated with end returns. The minimum length of end returns shall be two times the weld size. These returns shall be in the same plane as the rest of the weld.

The minimum effective length of a fillet weld shall be four times the specified weld size, or the weld size shall be considered not to exceed one-fourth of the effective weld length.

For fillet welds in holes or slots, the effective length shall be the length of the centerline of the weld along the plane through the center of the weld throat. The effective weld area shall not exceed the cross-sectional area of the hole or slot.

Table 3-3.4.3-1 Minimum Sizes of Fillet Welds

Material Thickness of Thicker Part Joined, in. (mm)	Minimum Size of Fillet Weld, in. (mm)
To $\frac{1}{4}$ (6)	$\frac{1}{8}$ (3)
Over $\frac{1}{4}$ (6) to $\frac{1}{2}$ (13)	$\frac{3}{16}$ (5)
Over $\frac{1}{2}$ (13) to $\frac{3}{4}$ (19)	$\frac{1}{4}$ (6)
Over $\frac{3}{4}$ (19)	$\frac{5}{16}$ (8)

The minimum fillet weld size shall not be less than the size required to transmit calculated forces nor the size given in Table 3-3.4.3-1. These tabulated sizes do not apply to fillet weld reinforcements of partial- or complete-joint-penetration welds.

The maximum fillet weld size is based on the thickness of the connected parts. Along edges of materials of thickness less than $\frac{1}{4}$ in. (6 mm), the weld size shall not exceed the thickness of the material. Along edges where the material thickness is $\frac{1}{4}$ in. (6 mm) or greater, the weld size shall not be greater than the material thickness minus $\frac{1}{16}$ in. (2 mm).

Intermittent fillet welds may be used to transfer calculated stress across a joint or faying surface when the strength required is less than that developed by a continuous fillet weld of the smallest permitted size and to join components of built-up members. The effective length of any intermittent fillet shall not be less than four times the weld size with a minimum of $1\frac{1}{2}$ in. (38 mm). Intermittent welds shall be made on both sides of the joint for at least 25% of its length. The maximum spacing of intermittent fillet welds is 12 in. (300 mm).

In lap joints, the minimum amount of lap shall be five times the thickness of the thinner part joined, but not less than 1 in. (25 mm). Where lap joints occur in plates or bars that are subject to axial stress, both lapped parts shall be welded along their ends.

Fillet welds shall not be used in skewed T-joints that have an included angle of less than 60 deg or more than 135 deg. The edge of the abutting member shall be beveled, when necessary, to limit the root opening to $\frac{1}{8}$ in. (3 mm) maximum.

Fillet welds in holes or slots may be used to transmit shear in lap joints or to prevent the buckling or separation of lapped parts and to join components of built-up members. Fillet welds in holes or slots are not to be considered plug or slot welds.

3-3.4.4 Plug and Slot Welds. Plug and slot welds may be used to transmit shear in lap joints or to prevent buckling of lapped parts and to join component parts of built up members. The effective shear area of plug and slot welds shall be considered as the nominal cross-sectional area of the hole or slot in the plane of the faying surface.

The diameter of the hole for a plug weld shall not be less than the thickness of the part containing it plus $\frac{5}{16}$ in. (8 mm) rounded up to the next larger odd $\frac{1}{16}$ in. (2 mm), nor greater than the minimum diameter plus $\frac{1}{8}$ in. (3 mm) or $2\frac{1}{4}$ times the thickness of the weld, whichever is greater. The minimum center-to-center spacing of plug welds shall be four times the diameter of the hole.

The length of the slot for a slot weld shall not exceed 10 times the thickness of the weld. The width of the slot shall meet the same criteria as the diameter of the hole for a plug weld. The ends of the slot shall be semicircular or shall have the corners rounded to a radius of not less than the thickness of the part containing it, except for those ends that extend to the edge of the part. The minimum spacing of lines of slot welds in a direction transverse to their length shall be four times the width of the slot. The minimum center-to-center spacing in a longitudinal direction on any line shall be two times the length of the slot.

The thickness of plug or slot welds in material $\frac{5}{8}$ in. (16 mm) or less in thickness shall be equal to the thickness of the material. In material over $\frac{5}{8}$ in. (16 mm) thick, the weld thickness shall be at least one-half the thickness of the material but not less than $\frac{5}{8}$ in. (16 mm).

3-4 FATIGUE DESIGN

3-4.1 General

When applying the fatigue design provisions defined in this section, calculated stresses shall be based upon elastic analysis and stresses shall not be amplified by stress concentration factors for geometrical discontinuities.

3-4.2 Lifter Classifications (14)

Lifter classifications shall be as given in Chapter 2. These classifications are based on use of the lifter at loads of varying magnitude, as discussed Nonmandatory Appendix C. In reality, actual use of the lifter may differ, possibly significantly, from the defined load spectra. If sufficient lift data are known or can be assumed, the equivalent number of constant amplitude load cycles can be determined using eq. (3-56).

$$N_{eq} = \Sigma \left(\frac{S_{Ri}}{S_{Rref}}\right)^3 n_i \tag{3-56}$$

where

N_{eq} = equivalent number of constant amplitude load cycles at stress range S_{Rref}

n_i = number of load cycles for the i^{th} portion of a variable amplitude loading spectrum

S_{Ri} = stress range for the i^{th} portion of a variable amplitude loading spectrum

Table 3-4.3-1 Allowable Stress Ranges, ksi (MPa) (14)

Stress Category (From Table 3-4.4-1)	Service Class			
	1	2	3	4
A	63 (435)	37 (255)	24 (165)	24 (165)
B	49 (340)	29 (200)	18 (125)	16 (110)
B′	39 (270)	23 (160)	15 (100)	12 (80)
C	35 (240)	21 (145)	13 (90)	10 (70) [Note (1)]
D	28 (190)	16 (110)	10 (70)	7 (50)
E	22 (150)	13 (90)	8 (55)	5 (34)
E′	16 (110)	9 (60)	6 (40)	3 (20)
F	15 (100)	12 (80)	9 (60)	8 (55)
G	16 (110)	9 (60)	7 (48)	7 (48)

NOTE:
(1) Flexural stress range of 12 ksi (80 MPa) permitted at the toe of stiffener welds on flanges.

S_{Rref} = reference stress range to which N_{eq} relates. This is usually, but not necessarily, the maximum stress range considered.

3-4.3 Allowable Stress Ranges

The maximum stress range shall be that given in Table 3-4.3-1.

Tensile stresses in the base metal of all load-bearing structural elements, including shafts and pins, shall not exceed the stress ranges for Stress Category A.

3-4.4 Stress Categories

The Stress Category can be determined from the joint details given in Table 3-4.4-1.

3-4.5 Tensile Fatigue in Threaded Fasteners

High strength bolts, common bolts, and threaded rods subjected to tensile fatigue loading shall be designed so that the tensile stress calculated on the tensile stress area due to the combined applied load and prying forces do not exceed the design stress range computed using eq. (3-57). The factor C_f shall be taken as 3.9×10^8. The threshold stress, F_{TH}, shall be taken as 7 ksi (48 MPa).

For joints in which the fasteners are pretensioned to at least 70% of their minimum tensile strength, an analysis of the relative stiffness of the connected parts and fasteners shall be permitted to determine the tensile stress range in the fasteners due to the cyclic loads. Alternately, the stress range in the fasteners shall be assumed to be equal to the stress on the net tensile area due to 20% of the absolute value of the design tensile load. If the fasteners are not pretensioned to at least 70% of their minimum tensile strength, then all tension shall be assumed to be carried exclusively by the fasteners.

3-4.6 Cumulative Fatigue Analysis (14)

If a more refined component fatigue analysis than provided by the four Service Classes given in Chapter 2 is desired, eq. (3-57) may be used to obtain the allowable stress range for any number of load cycles for the Stress Categories given in Table 3-4.4-1.

$$F_{sr} = R\left(\frac{C_f q}{N}\right)^{ex} \geq F_{TH} \tag{3-57}$$

where $R = 1$, except as follows:

(a) for Stress Category C′ when stresses are in ksi,

$$R = \frac{0.65 - 0.59\left(\frac{2a}{t_p}\right) + 0.72\left(\frac{w}{t_p}\right)}{t_p^{0.167}} \leq 1.0$$

(b) for Stress Category C′ when stresses are in MPa,

$$R = \frac{1.12 - 1.01\left(\frac{2a}{t_p}\right) + 1.24\left(\frac{w}{t_p}\right)}{t_p^{0.167}} \leq 1.0$$

(c) for Stress Category C″ when stresses are in ksi,

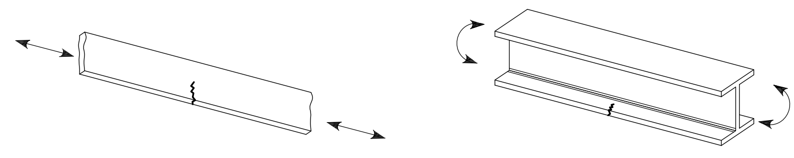
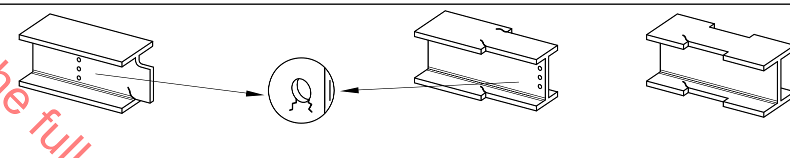
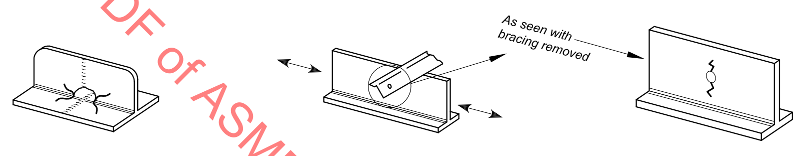
$$R = \frac{0.06 + 0.72\left(\frac{w}{t_p}\right)}{t_p^{0.167}} \leq 1.0$$

(d) for Stress Category C″ when stresses are in MPa,

$$R = \frac{0.10 + 1.24\left(\frac{w}{t_p}\right)}{t_p^{0.167}} \leq 1.0$$

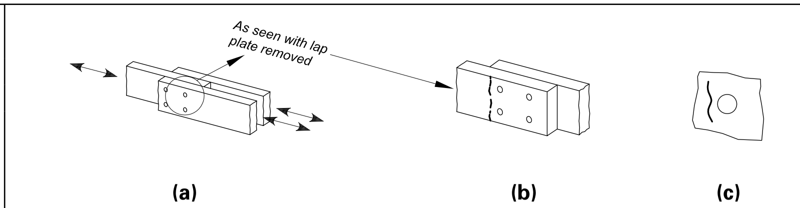
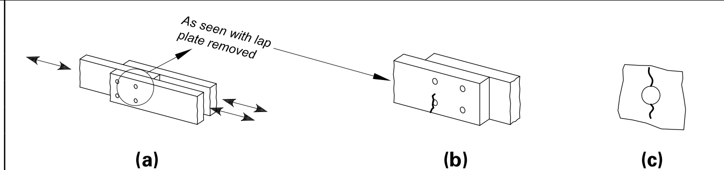
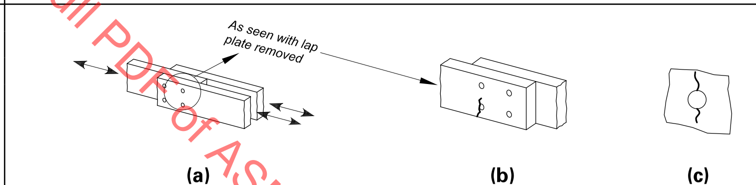
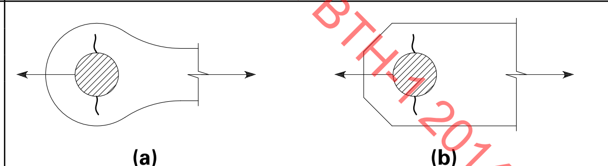
(14)

Table 3-4.4-1 Fatigue Design Parameters

Description	Stress Cate- gory	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 1 — Plain Material Away From Any Welding					
1.1 Base metal, except noncoated weathering steel, with rolled or cleaned surface. Flame-cut edges with surface roughness value of 1,000 μin. (25 μm) or less, but without re-entrant corners.	A	250×10^8	24 (165)	Away from all welds or structural connections	 (a) (b)
1.2 Noncoated weathering steel base metal with rolled or cleaned surface. Flame-cut edges with surface roughness value of 1,000 μin. (25 μm) or less, but without re-entrant corners.	B	120×10^8	16 (110)	Away from all welds or structural connections	
1.3 Member with drilled or reamed holes. Member with re-entrant corners at copes, cuts, block-outs or other geometrical discontinuities made to requirements of AISC (2010) Appendix 3, except weld access holes.	B	120×10^8	16 (110)	At any external edge or hole perimeter	 (a) (b) (c)
1.4 Rolled cross sections with weld access holes made to requirements of AISC (2010) Section J1.6 and Appendix 3. Members with drilled or reamed holes containing bolts for attachment of light bracing where there is a small longitudinal component of brace force.	C	44×10^8	10 (69)	At re-entrant corner of weld access hole or at any small hole (may contain bolt for minor connections)	 (a) (b) (c)

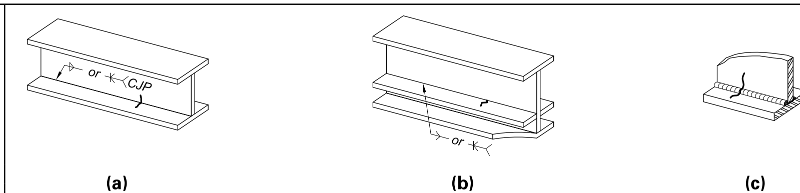
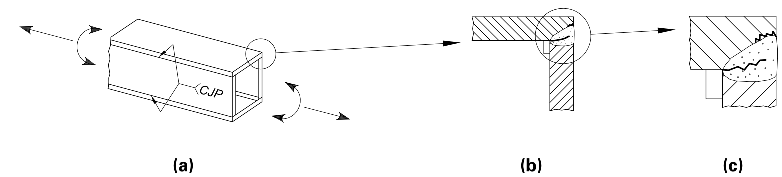
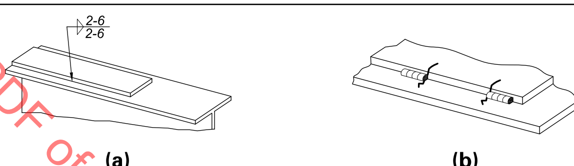
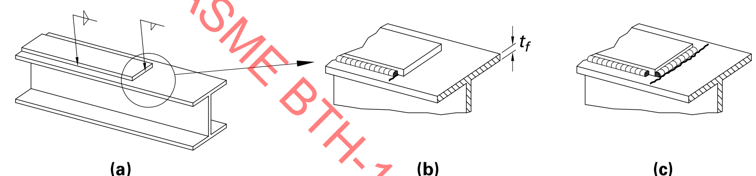
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Cate-gory	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 2 — Connected Material In Mechanically Fastened Joints					
2.1 Gross area of base metal in lap joints connected by high-strength bolts in joints satisfying all requirements for slip-critical connections.	B	120×10^8	16 (110)	Through gross section near hole	 *(Note: figures are for slip-critical bolted connections)*
2.2 Base metal at net section of high-strength bolted joints, designed on the basis of bearing resistance, but fabricated and installed to all requirements for slip-critical connections.	B	120×10^8	16 (110)	In net section originating at side of hole	 *(Note: figures are for bolted connections designed to bear, meeting the requirements of slip-critical connections)*
2.3 Base metal at the net section of other mechanically fastened joints except eyebars and pin plates.	D	22×10^8	7 (48)	In net section originating at side of hole	 *(Note: figures are for snug-tightened bolts, rivets, or other mechanical fasteners)*
2.4 Base metal at net section of eyebar head or pin plate.	E	11×10^8	4.5 (31)	In net section originating at side of hole	

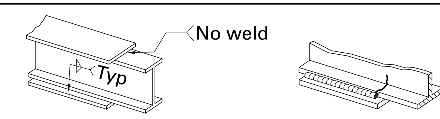
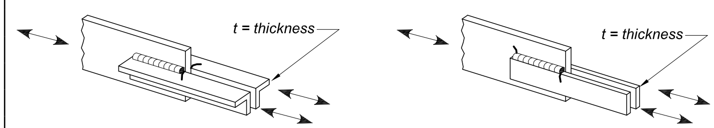
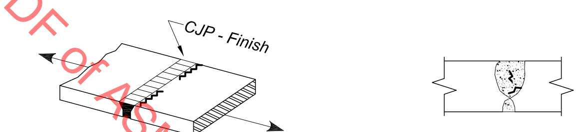
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 3 — Welded Joints Joining Components of Built-Up Members					
3.1 Base metal and weld metal in members without attachments built-up of plates or shapes connected by continuous longitudinal complete-joint-penetration groove welds, back gouged and welded from second side, or by continuous fillet welds.	B	120×10^8	16 (110)	From surface or internal discontinuities in weld away from end of weld	
3.2 Base metal and weld metal in members without attachments built-up of plates or shapes connected by continuous longitudinal complete-joint-penetration groove welds with backing bars not removed, or by continuous partial-joint-penetration groove welds.	B′	61×10^8	12 (83)	From surface or internal discontinuities in weld, including weld attaching backing bars	
3.3 Base metal at weld metal termination of longitudinal welds at weld access holes in connected built-up members.	D	22×10^8	7 (48)	From the weld termination into the web or flange	
3.4 Base metal at ends of longitudinal intermittent fillet weld segments.	E	11×10^8	4.5 (31)	In connected material at start and stop locations of any weld deposit	
3.5 Base metal at ends of partial length welded cover plates narrower than the flange having square or tapered ends, with or without welds across the ends; and cover plates wider than the flange with welds across the ends. Flange thickness ≤ 0.8 in. (20 mm) Flange thickness > 0.8 in. (20 mm)	 E E′	 11×10^8 3.9×10^8	 4.5 (31) 2.6 (18)	In flange at toe of end weld or in flange at termination of longitudinal weld or in edge of flange with wide cover plates	

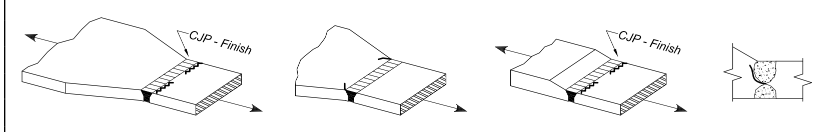
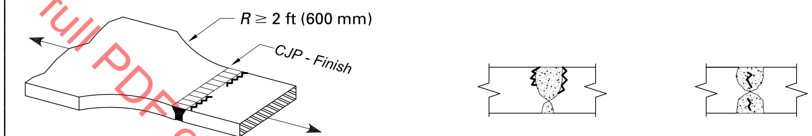
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 3 — Welded Joints Joining Components of Built-Up Members (Cont'd)					
3.6 Base metal at ends of partial length welded cover plates wider than the flange without welds across the ends.	E′	3.9×10^8	2.6 (18)	In edge of flange at end of cover plate weld	 (a) (b)
Section 4 — Longitudinal Fillet Welded End Connections					
4.1 Base metal at junction of axially loaded members with longitudinally welded end connections. Welds shall be on each side of the axis of the member to balance weld stresses. $t \le 0.5$ in. (12 mm) $t > 0.5$ in. (12 mm)	 E E′	 10×10^8 3.9×10^8	 4.5 (31) 2.6 (18)	Initiating from end of any weld termination extending into the base metal	 (a) (b)
Section 5 — Welded Joints Transverse to Direction of Stress					
5.1 Base metal and weld metal in or adjacent to complete joint penetration groove welded splices in rolled or welded cross sections with welds ground essentially parallel to the direction of stress and with soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M.	B	120×10^8	16 (110)	From internal discontinuities in filler metal or along the fusion boundary	 (a) (b)

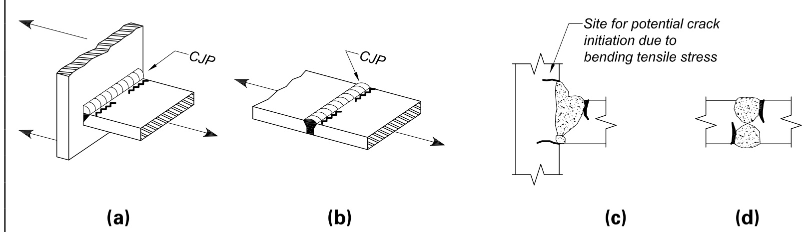
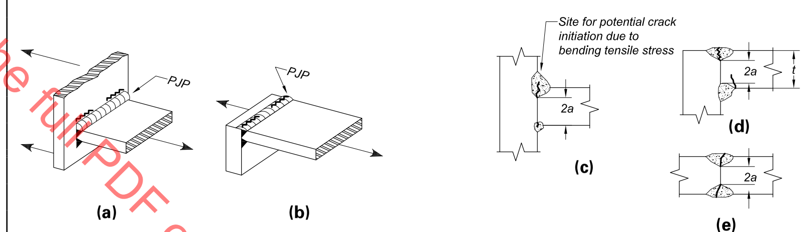
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 5 — Welded Joints Transverse to Direction of Stress (Cont'd)					
5.2 Base metal and weld metal in or adjacent to complete joint penetration groove welded splices with welds ground essentially parallel to the direction of stress at transitions in thickness or width made on a slope no greater than 1:2.5 and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M.				From internal discontinuities in filler metal or along fusion boundary or at start of transition when $F_y \geq 90$ ksi (620 MPa)	
$F_y < 90$ ksi (620 MPa)	B	120×10^8	16 (110)		
$F_y \geq 90$ ksi (620 MPa)	B′	61×10^8	12 (83)		
5.3 Base metal with $F_y \geq 90$ ksi (620 MPa) and weld metal in or adjacent to complete joint penetration groove welded splices with welds ground essentially parallel to the direction of stress at transitions in width made on a radius of not less than 2 ft (600 mm) with the point of tangency at the end of the groove weld and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M.	B	120×10^8	16 (110)	From internal discontinuities in filler metal or discontinuities along the fusion boundary	

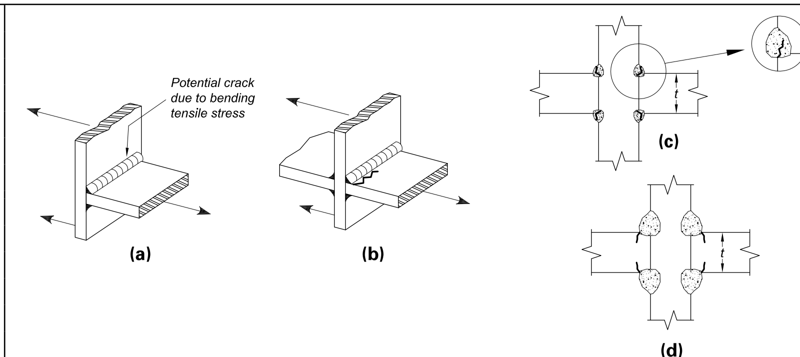
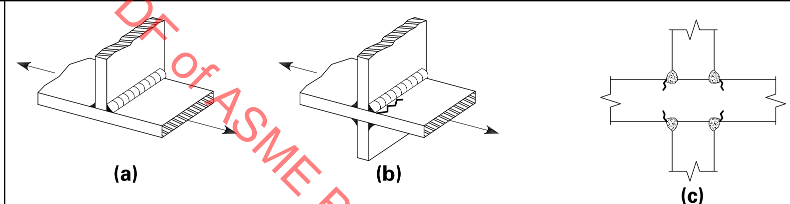
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

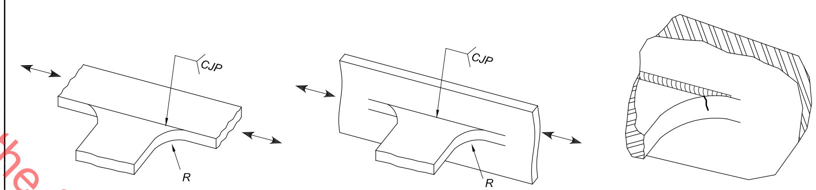
Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 5 — Welded Joints Transverse to Direction of Stress (Cont'd)					
5.4 Base metal and weld metal in or adjacent to the toe of complete joint penetration T or corner joints or splices, with or without transitions in thickness having slopes no greater than 1:2.5, when weld reinforcement is not removed and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M.	C	44 × 10^8	10 (69)	From surface discontinuity at toe of weld extending into base metal or into weld metal	
5.5 Base metal and weld metal at transverse end connections of tension-loaded plate elements using partial joint penetration groove welds in butt or T or corner joints, with reinforcing or contouring fillets, F_{SR} shall be the smaller of the toe crack or root crack allowable stress range.					
Crack initiating from weld toe:	C	44 × 10^8	10 (69)	Initiating from geometrical discontinuity at toe of weld extending into base metal.	
Crack initiating from weld root:	C′	eq. (3-57)	None provided	Initiating at weld root subject to tension extending into and through weld	

(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

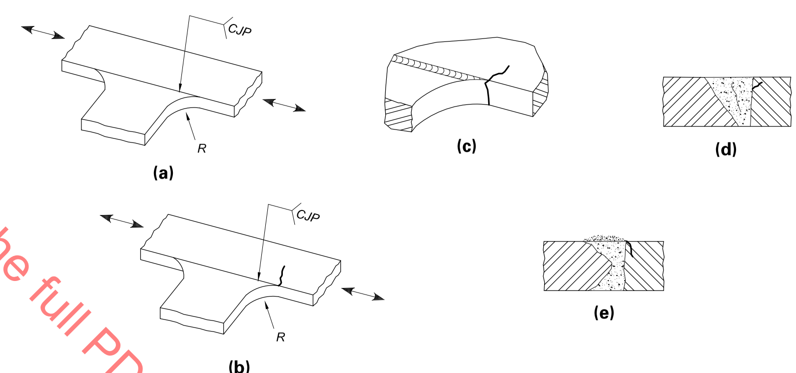
Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 5 — Welded Joints Transverse to Direction of Stress (Cont'd)					
5.6 Base metal and weld metal at transverse end connections of tension-loaded plate elements using a pair of fillet welds on opposite sides of the plate. F_{SR} shall be the smaller of the toe crack or root crack allowable stress range.					
Crack initiating from weld toe:	C	44×10^8	10 (69)	Initiating from geometrical discontinuity at toe of weld extending into base metal.	
Crack initiating from weld root:	C″	eq. (3-57)	None provided	Initiating at weld root subject to tension extending into and through weld.	
5.7 Base metal of tension loaded plate elements and on girders and rolled beam webs or flanges at toe of transverse fillet welds adjacent to welded transverse stiffeners.	C	44×10^8	10 (69)	From geometrical discontinuity at toe of fillet extending into base metal	

(14) **Table 3-4.4-1 Fatigue Design Parameters (Cont'd)**

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 6 — Base Metal at Welded Transverse Member Connections					
6.1 Base metal at details attached by complete joint penetration groove welds subject to longitudinal loading only when the detail embodies a transition radius *R* with the weld termination ground smooth and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M.				Near point of tangency of radius at edge of member	
$R \geq$ 24 in. (600 mm)	B	120×10^8	16 (110)		
24 in. (600 mm) > $R \geq$ 6 in. (150 mm)	C	44×10^8	10 (69)		
6 in. (150 mm) > $R \geq$ 2 in. (50 mm)	D	22×10^8	7 (48)		
2 in. (50 mm) > R	E	11×10^8	4.5 (31)		

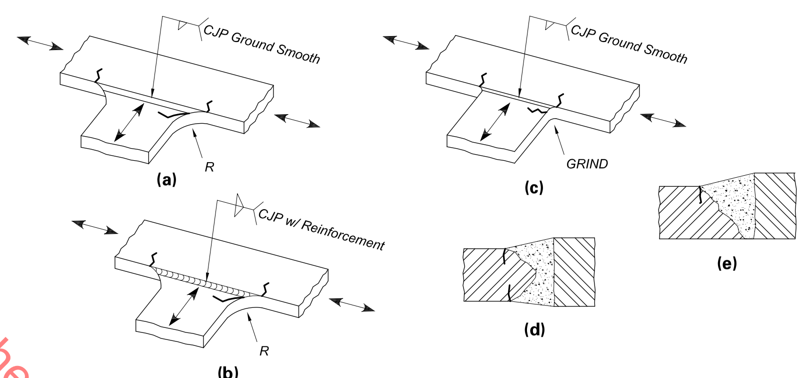
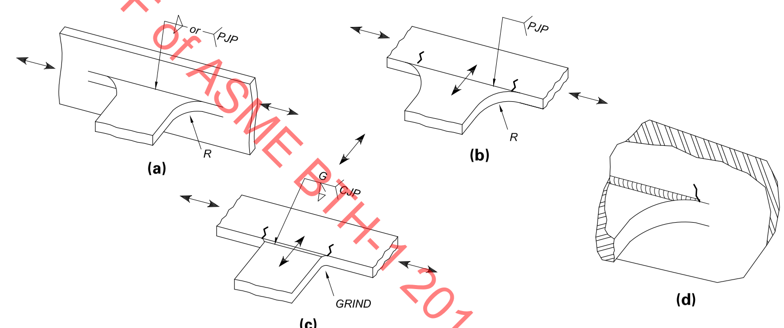
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 6 — Base Metal at Welded Transverse Member Connections (Cont'd)					
6.2 Base metal at details of equal thickness attached by complete joint penetration groove welds subject to transverse loading with or without longitudinal loading when the detail embodies a transition radius, *R*, with the weld termination ground smooth and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M:					
When weld reinforcement is removed:				Near points of tangency of radius or in the weld or at fusion boundary or member or attachment	
$R \geq 24$ in. (600 mm)	B	120×10^8	16 (110)		
24 in. (600 mm) > $R \geq 6$ in. (150 mm)	C	44×10^8	10 (69)		
6 in. (150 mm) > $R \geq 2$ in. (50 mm)	D	22×10^8	7 (48)		
2 in. (50 mm) > R	E	11×10^8	4.5 (31)		
When weld reinforcement is not removed:				At toe of the weld either along edge of member or the attachment	
$R \geq 24$ in. (600 mm)	C	44×10^8	16 (110)		
24 in. (600 mm) > $R \geq 6$ in. (150 mm)	C	44×10^8	10 (69)		
6 in. (150 mm) > $R \geq 2$ in. (50 mm)	D	22×10^8	7 (48)		
2 in. (50 mm) > R	E	11×10^8	4.5 (31)		

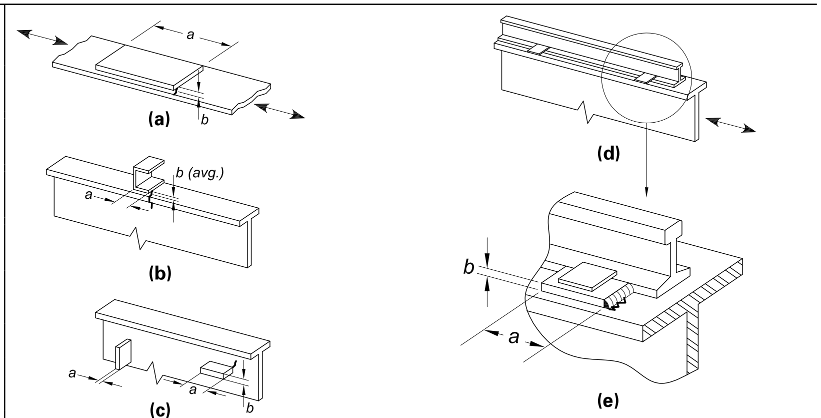
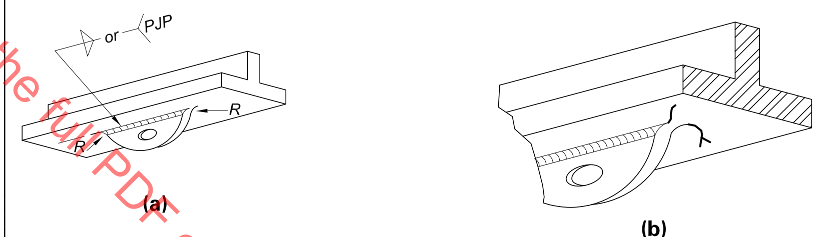
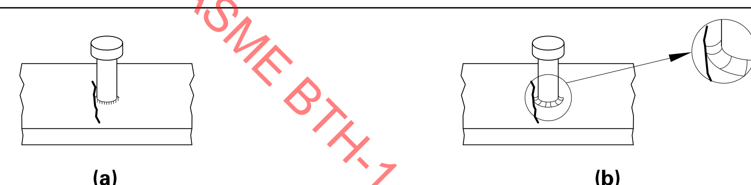
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 6 — Base Metal at Welded Transverse Member Connections (Cont'd)					
6.3 Base metal at details of unequal thickness attached by complete joint penetration groove welds subject to transverse loading with or without longitudinal loading when the detail embodies a transition radius, *R*, with the weld termination ground smooth and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of sub-clauses 6.12 or 6.13 of AWS D1.1/D1.1M:					
When weld reinforcement is removed: $R > 2$ in. (50 mm)	D	22×10^8	7 (48)	At toe of weld along edge of thinner material	
$R \leq 2$ in. (50 mm)	E	11×10^8	4.5 (31)	In weld termination in small radius	
When reinforcement is not removed: Any radius	E	11×10^8	4.5 (31)	At toe of weld along edge of thinner material	
6.4 Base metal subject to longitudinal stress at transverse members, with or without transverse stress, attached by fillet or partial-joint-penetration groove welds parallel to direction of stress when the detail embodies a transition radius, *R*, with weld termination ground smooth:				Initiating in base metal at the weld termination or at the toe of the weld extending into the base metal	
$R > 2$ in. (50 mm)	D	22×10^8	7 (48)		
$R \leq 2$ in. (50 mm)	E	11×10^8	4.5 (31)		

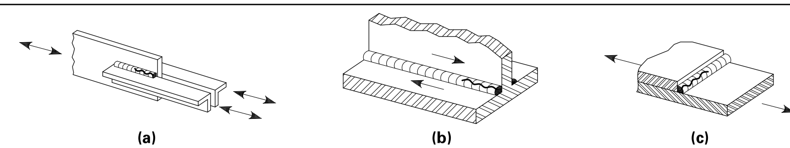
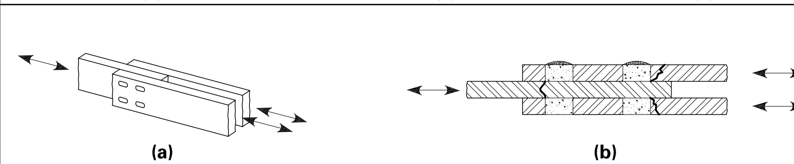
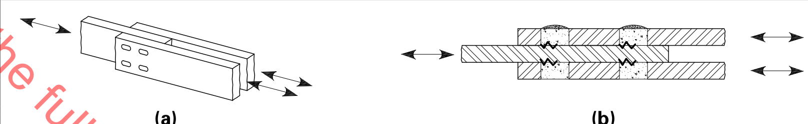
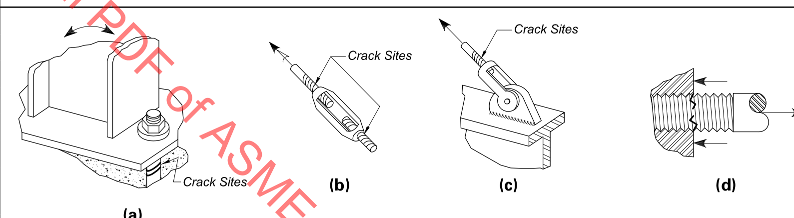
(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 7 — Base Metal at Short Attachments [Note (1)]					
7.1 Base metal subject to longitudinal loading at details with welds parallel or transverse to the direction of stress where the detail embodies no transition radius and with detail length in direction of stress, *a*, and thickness of attachment, *b*:				Initiating in base metal at the weld termination or at the toe of the weld extending into the base metal	
$a < 2$ in. (50 mm)	C	44×10^8	10 (69)		
2 in. (50 mm) $\le a \le$ lesser of 12*b* or 4 in. (100 mm)	D	22×10^8	7 (48)		
$a > 12b$ or 4 in. (100 mm), when *b* is ≤ 1 in. (25 mm)	E	11×10^8	4.5 (31)		
$a >$ lesser of 12*b* or 4 in. (100 mm), when *b* is > 1 in. (25 mm)	E′	3.9×10^8	2.6 (18)		
7.2 Base metal subject to longitudinal stress at details attached by fillet or partial joint peneration groove welds, with or without transverse load on detail, when the detail embodies a transition radius, *R*, with weld termination ground smooth:				Initiating in base metal at the weld termination, extending into the base metal	
$R > 2$ in. (50 mm)	D	22×10^8	7 (48)		
$R \le 2$ in. (50 mm)	E	11×10^8	4.5 (31)		
Section 8 — Miscellaneous					
8.1 Base metal at steel headed stud anchors attached by fillet or automatic stud welding.	C	44×10^8	10 (69)	At toe of weld in base metal	

(14)

Table 3-4.4-1 Fatigue Design Parameters (Cont'd)

Description	Stress Category	Constant, C_f	Threshold, F_{TH}, ksi (MPa)	Potential Crack Site Initiation	Illustrative Typical Examples
Section 8 — Miscellaneous (Cont'd)					
8.2 Shear on throat of continuous or intermittent longitudinal or transverse fillet welds.	F	150×10^{10}	8 (55)	Initiating at the root of the fillet weld, extending into the weld	
8.3 Base metal at plug or slot welds.	E	11×10^{8}	4.5 (31)	Initiating in the base metal at the end of the plug or slot weld, extending into the base metal	
8.4 Shear on plug or slot welds.	F	150×10^{10} eq. (3-57)	8 (55)	Initiating in the weld at the faying surface, extending into the weld	
8.5 Snug-tightened high-strength bolts, common bolts, threaded anchor rods and hanger rods with cut, ground or rolled threads. Stress range on tensile stress area due to live load plus prying action when applicable.	G	3.9×10^{8}	7 (48)	Initiating at the root of the threads, extending into the fastener	

NOTE:
(1) "Attachment" as used herein is defined as any steel detail welded to a member, which by its mere presence and independent of its loading, causes a discontinuity in the stress flow in the member and thus reduces the fatigue resistance.

Use the requirements for Stress Category C if $R = 1.0$.

$2a$ = length of the nonwelded root face in the direction of the thickness of the tension-loaded plate

C_f = constant from Table 3-4.4-1 for the Stress Category

$C_f(q)$ = 14.4×10^{11} for Stress Categories C, C′, and C″ when stresses are in MPa

ex = 0.167 for Stress Category F

= 0.333 for all Stress Categories except F

F_{sr} = allowable stress range for the detail under consideration. Stress range is the algebraic difference between the maximum stress and the minimum stress.

F_{TH} = threshold value for F_{sr} as given load in Table 3-4.4-1

N = desired design fatigue life in load cycles of the detail being evaluated. N is the expected number of constant amplitude stress range cycles and is to be provided by the owner. If no desired fatigue life is specified, a qualified person should use the threshold values, F_{TH}, as the allowable stress range, F_{sr}. For cumulative damage analysis of a varying amplitude load spectrum, an equivalent number of constant amplitude load cycles can be calculated using eq. (3-56).

q = 1.0 when stresses are in ksi

= 329 for all Stress Categories except F when stresses are in MPa, except as noted

= 110,000 for Stress Category F when stresses are in MPa, except as noted

t_p = thickness of the tension-loaded plate

w = leg size of the reinforcing or contouring fillet, if any, in the direction of the thickness of the tension-loaded plate

3-5 OTHER DESIGN CONSIDERATIONS

3-5.1 Impact Factors

The design of below-the-hook lifting devices does not normally require the use of an impact factor. The design factors established in this chapter are based on load spectra in which peak impact loads are equal to 50% of the maximum lifted load for Design Category A lifters and 100% of the maximum lifted load for Design Category B lifters. In the event that a lifter is expected to be subjected to impact loading greater than these values, a qualified person shall include an additional impact factor to account for such loads.

3-5.2 Stress Concentrations

Stress concentrations due to holes, changes in section, or similar details shall be accounted for when determining peak stresses in load-carrying elements subject to cyclic loading, unless stated otherwise in this chapter. The need to use peak stresses, rather than average stresses, when calculating static strength shall be determined by a qualified person based on the nature of the detail and the properties of the material being used.

3-5.3 Deflection

It is the responsibility of a qualified person to determine when deflection limits should be applied and to establish the magnitudes of those limits for the design of the mechanisms and structural elements of lifting devices.

Chapter 4
Mechanical Design

4-1 GENERAL

4-1.1 Purpose

This chapter sets forth design criteria for machine elements of a below-the-hook lifting device.

4-1.2 Relation to Chapter 3

Mechanical components of the lifting device that are stressed by the force(s) created during the lift or movement of the load shall be sized in accordance with this chapter and Chapter 3 of this Standard. The most conservative design shall be selected for use. All other mechanical components shall be designed to the requirements of this chapter.

4-2 SHEAVES

4-2.1 Sheave Material

Sheaves shall be fabricated of material specified by the lifting device manufacturer or qualified person.

4-2.2 Running Sheaves

Pitch diameter for running sheaves should not be less than 16 times the nominal diameter of the wire rope used. When the lifting device's sheaves are reeved into the sheaves on the hoist, the pitch diameter and configuration of the hoist shall be considered in the design.

4-2.3 Equalizing Sheaves

The pitch diameter of equalizing sheaves shall not be less than one-half of the diameter of the running sheaves, nor less than 12 times the wire rope diameter when using 6 × 37 class wire rope or 15 times the wire rope diameter when using 6 × 19 class wire rope.

4-2.4 Shaft Requirement

Sheave assemblies should be designed based on a removable shaft.

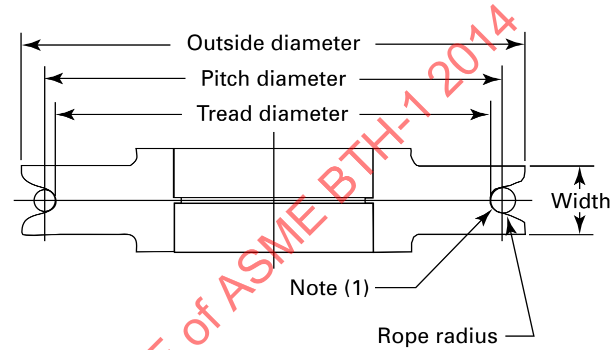
4-2.5 Lubrication

Means for lubricating sheave bearings shall be provided.

4-2.6 Sheave Design

Sheave grooves shall be smooth and free from surface irregularities that could cause wire rope damage. The groove radius of a new sheave shall be a minimum of 6% larger than the radius of the wire rope as shown in Fig. 4-2.6-1. The cross-sectional radius of the groove should form a close-fitting saddle for the size of the wire rope used, and the sides of the grooves should be tapered outwardly to assist entrance of the wire rope into the groove. Flange corners should be rounded, and rims should run true around the axis of rotation.

Fig. 4-2.6-1 Sheave Dimensions

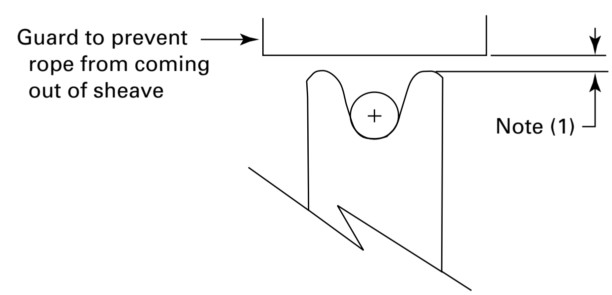


NOTE:
(1) Groove radius = rope radius × 1.06.

4-2.7 Sheave Guard

Sheaves shall be guarded to prevent inadvertent wire rope jamming or coming out of the sheave. The guard shall be placed within ⅛ in. (3 mm) to the sheave, or a distance of ⅜ times the wire rope diameter, whichever is smaller, as shown in Fig. 4-2.7-1.

Fig. 4-2.7-1 Sheave Gap



NOTE:
(1) ⅛ in. (3 mm) or a distance of ⅜ times the rope diameter, whichever is smaller.

4-3 WIRE ROPE

4-3.1 Relation to Other Standards

Wire rope reeved through the lifting device and the hoist shall conform to the requirements of the hoist.

4-3.2 Rope Selection

Wire rope shall be of a recommended construction for lifting service. The qualified person shall consider other factors (i.e., type of end connection, *D/d* ratio, sheave bearing friction, etc.) that affect the wire rope strength to ensure the 5:1 safety factor is maintained.

4-3.3 Environment

Wire rope material selection shall be appropriate for the environment in which it is to be used.

4-3.4 Fleet Angle

The wire rope fleet angle for sheaves should be limited to a 1 in 12 slope (4 deg, 45 min).

4-3.5 Rope Ends

Wire rope ends shall be attached to the lifting device in a manner to prevent disengagement during operation of the lifting device.

4-3.6 Rope Clips

Wire rope clips shall be drop-forged steel of the single-saddle (U-bolt) or double-saddle type. Malleable cast iron clips shall not be used. For spacing, number of clips, and torque values, refer to the clip manufacturer's recommendations. Wire rope clips attached with U-bolts shall have the U-bolt over the dead end of the wire rope and live rope resting in the clip saddle. Clips shall be tightened evenly to the recommended torque. After the initial load is applied to the wire rope, the clip nuts shall be retightened to the recommended torque to compensate for any decrease in wire rope diameter caused by the load.

4-4 DRIVE SYSTEMS

4-4.1 Drive Adjustment

Drive systems that contain belts, chains, or other flexible transmission devices should have provisions for adjustment.

4-4.2 Drive Design

The lifting device manufacturer or qualified person shall specify drive system components such as couplings, belts, pulleys, chains, sprockets, and clutches.

4-4.3 Commercial Components

Commercial components used in the drive system of a lifting device shall be sized so the maximum load rating specified by the manufacturer is not exceeded under worst case loadings.

4-4.4 Lubrication

Means for lubricating and inspecting drive systems shall be provided.

4-4.5 Operator Protection

All motion hazards associated with the operation of mechanical power transmission components shall be eliminated by design of the equipment or protection by a guard, device, safe distance, or safe location. All motion hazard guards shall

(a) prevent entry of hands, fingers, or other parts of the body into a point of hazard by reaching through, over, under, or around the guard

(b) not create additional motion hazards between the guard and the moving part

(c) utilize fasteners not readily removable by people other than authorized persons

(d) not cause any additional hazards, if openings are provided for lubrication, adjustment, or inspection

(e) reduce the likelihood of personal injury due to breakage of component parts

(f) be designed to hold the weight of a 200-lb (91-kg) person without permanent deformation, if used as a step

4-5 GEARING

4-5.1 Gear Design

The lifting device manufacturer or qualified person shall specify the types of gearing.

4-5.2 Gear Material

Gears and pinions shall be fabricated of material having adequate strength and durability to meet the requirements for the intended Service Class and manufactured to AGMA quality class 5 or better.

4-5.3 Gear Loading

The allowable tooth load in bending, L_G, of spur and helical gears is

$$L_G = \frac{\sigma_y F Y}{N_d D_t} \tag{4-1}$$

where

D_t = diametral pitch, in.$^{-1}$ (mm^{-1})
F = face width of smaller gear, in. (mm)
L_G = allowable tooth load in bending, lb (N)
N_d = design factor (per para. 3-1.3)
Y = Lewis form factor as defined in Table 4-5.3-1
σ_y = specified minimum yield stress, psi (MPa)

4-5.4 Relation to Other Standards

As an alternative to the Lewis formula in eq. (4-1), spur and helical gears may be based upon ANSI/AGMA 2001-C95, Fundamental Rating Factors

Table 4-5.3-1 Strength Factors for Calculating Load Capacity (American Standard Tooth Forms)

Number of Teeth	Strength Factors, *Y*, for Use With Diametral Pitch: $14\frac{1}{2}$ deg Composite and Involute	20 deg Full Depth Involute System	20 deg Stub-Tooth Involute System
12	0.210	0.245	0.311
13	0.220	0.261	0.324
14	0.226	0.276	0.339
15	0.236	0.289	0.348
16	0.242	0.295	0.361
17	0.251	0.302	0.367
18	0.261	0.308	0.377
19	0.273	0.314	0.386
20	0.283	0.320	0.393
21	0.289	0.327	0.399
22	0.292	0.330	0.405
24	0.298	0.336	0.415
26	0.307	0.346	0.424
28	0.314	0.352	0.430
30	0.320	0.358	0.437
34	0.327	0.371	0.446
38	0.336	0.383	0.456
43	0.346	0.396	0.462
50	0.352	0.408	0.474
60	0.358	0.421	0.484
75	0.364	0.434	0.496
100	0.371	0.446	0.506
150	0.377	0.459	0.518
300	0.383	0.471	0.534
Rack	0.390	0.484	0.550

GENERAL NOTE: The strength factors above are used in formulas containing diametral pitch. These factors are 3.1416 times those used in formulas based on circular pitch.

Table 4-6.2-1 L_{10} Life

Service Class	L_{10} Bearing Life, hr
0	2,500
1	10,000
2	20,000
3	30,000
4	40,000

and Calculation Methods for Involute Spur and Helical Gear Teeth.

4-5.5 Bevel and Worm Gears

Bevel and worm gearing shall be rated by the gear manufacturer with service factors appropriate for the specified Service Class of the lifting device. When back-driving could be a problem, due consideration shall be given to selecting a worm gear ratio to establish lock-up.

4-5.6 Split Gears

Split gears shall not be used.

4-5.7 Lubrication

Means shall be provided to allow for the lubrication and inspection of gearing.

4-5.8 Operator Protection

Exposed gearing shall be guarded per para. 4-4.5 with access provisions for lubrication and inspection.

4-5.9 Reducers

Gear reducer cases shall

(a) be oil-tight and sealed with compound or gaskets
(b) have an accessible drain plug
(c) have a means for checking oil level

4-6 BEARINGS

4-6.1 Bearing Design

The type of bearings shall be specified by the lifting device manufacturer or qualified person.

4-6.2 L_{10} Life

L_{10} bearing life for rolling element bearings shall equal or exceed the values given in Table 4-6.2-1 for the lifting device Service Class.

4-6.3 Bearing Loadings

The basic rating life, L_{10}, for a radial bearing is given by eq. (4-2).

$$L_{10} = \left(\frac{16{,}667}{N}\right)\left(\frac{C_r}{P_r}\right)^H \quad (4\text{-}2)$$

The basic dynamic load rating C_r for a bearing with L_{10} bearing life from Table 4-6.2-1 is determined by eqs. (4-3) and (4-4).

$$C_r = \frac{P_r(L_{10}N)^{\frac{1}{H}}}{16{,}667^{\frac{1}{H}}} \quad (4\text{-}3)$$

$$P_r = XF_r + YF_a \geq F_r \quad (4\text{-}4)$$

where

C_r = basic dynamic load rating to theoretically endure one million revolutions, per bearing manufacturer, lb (N)
F_a = axial component of the actual bearing load, lb (N)
F_r = radial component of the actual bearing load, lb (N)
H = 3 for ball bearings, 10/3 for roller bearings
L_{10} = basic rating life exceeded by 90% of bearings tested, hr
N = rotational speed, rev./min
P_r = dynamic equivalent radial load, lb (N)
X = dynamic radial load factor per bearing manufacturer
Y = dynamic axial load factor per bearing manufacturer

4-6.4 Sleeve and Journal Bearings

Sleeve or journal bearings shall not exceed pressure and velocity ratings as defined by eqs. (4-5) through (4-7). The manufacturers' values of P, V, and PV shall be used.

$$P = \frac{W}{dL} \quad (4\text{-}5)$$

$$V = \frac{\pi Nd}{c} \quad (4\text{-}6)$$

$$PV = \frac{\pi WN}{Lc} \quad (4\text{-}7)$$

where

c = 12 when using U.S. Customary units
= 60,000 when using SI units

d = nominal shaft diameter or bearing inside diameter, in. (mm)
L = bearing length, in. (mm)
P = average pressure, psi (MPa)
V = surface velocity of shaft, ft/min (m/s)
W = bearing load, lb (N)

4-6.5 Lubrication

Means shall be provided to lubricate bearings. Bearing enclosures should be designed to exclude dirt and prevent leakage of oil or grease.

4-7 SHAFTING

4-7.1 Shaft Design

Shafting shall be fabricated of material having adequate strength and durability suitable for the application. The shaft diameter and method of support shall be specified by the lifting device manufacturer or qualified person and satisfy the conditions of paras. 4-7.2 through 4-7.7.

4-7.2 Shaft Alignment

Alignment of the shafting to gearboxes, couplings, bearings, and other drive components shall meet or exceed the component manufacturer's specifications.

4-7.3 Operator Protection

Exposed shafting shall be guarded per para. 4-4.5 with access provisions for lubrication and inspection.

4-7.4 Shaft Details

Shafting, keys, holes, press fits, and fillets shall be designed for the forces encountered in actual operation under the worst case loading.

4-7.5 Shaft Static Stress

The nominal key size used to transmit torque through a shaft/bore interface shall be determined from Tables 4-7.5-1 and 4-7.5-2 based on the nominal shaft diameter.

Static stress on a shaft element shall not exceed the following values:

(a) axial or bending stress

$$S = S_a + S_b \leq 0.2S_u \tag{4-8}$$

where
S = computed combined axial/bending stress, ksi (MPa)
S_a = computed axial stress, ksi (MPa)
S_b = computed bending stress, ksi (MPa)
S_u = specified minimum ultimate tensile strength, ksi (MPa)

(b) shear stress

$$\tau = \tau_T + \tau_V \leq \frac{S_u}{5\sqrt{3}} = 0.1155S_u \tag{4-9}$$

where
τ = computed combined shear stress, ksi (MPa)
τ_T = computed torsional shear stress, ksi (MPa)
τ_V = computed transverse shear stress, ksi (MPa)

(c) Shaft elements subject to combined axial/bending and shear stresses shall be proportioned such that the combined stress does not exceed the following value:

$$S_c = \sqrt{S^2 + 3\tau^2} \leq 0.2S_u \tag{4-10}$$

where
S_c = computed combined stress, ksi (MPa)

4-7.6 Shaft Fatigue

Shafting subjected to fluctuating stresses such as bending in rotation or torsion in reversing drives shall be checked for fatigue. This check is in addition to the static checks in para. 4-7.5 and need only be performed at points of geometric discontinuity where stress concentrations exist such as holes, fillets, keys, and press fits. Appropriate geometric stress concentration factors for the discontinuities shall be determined by the lifting device manufacturer or qualified person from a reference such as *Peterson's Stress Concentration Factors* by W. D. Pilkey.

4-7.6.1 Fatigue Stress Amplification Factor. The fatigue stress amplification factor, K_A, based on Service Class shall be selected from Table 4-7.6.1-1.

4-7.6.2 Endurance Limit. The corrected bending endurance limit, S_{ec}, for the shaft material is

$$S_{ec} = 0.5S_e = 0.25S_u \tag{4-11}$$

where
S_e = fatigue (endurance) limit of polished, unnotched specimen in reversed bending, ksi (MPa)
S_{ec} = corrected fatigue (endurance) limit of shaft in reversed bending, ksi (MPa)

4-7.6.3 Fatigue Stress. Fatigue stress on a shaft element shall not exceed the following values:

(a) Direct axial and/or bending fatigue stress shall not exceed

$$S_f = (K_{TD})S_t + (K_{TB})S_b \leq \frac{S_{ec}}{K_A} \tag{4-12}$$

where
K_{TB} = stress amplification factor for bending
K_{TD} = stress amplification factor for direct tension

Table 4-7.5-1 Key Size Versus Shaft Diameter (ASME B17.1)

Nominal Shaft Diameter, in. Over	To	Nominal Key Size, in.
$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{32}$
$\frac{7}{16}$	$\frac{9}{16}$	$\frac{1}{8}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{16}$
$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$
$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{16}$
$1\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$
$1\frac{3}{4}$	$2\frac{1}{4}$	$\frac{1}{2}$
$2\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$
$2\frac{3}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$
$3\frac{1}{4}$	$3\frac{3}{4}$	$\frac{7}{8}$
$3\frac{3}{4}$	$4\frac{1}{2}$	1
$4\frac{1}{2}$	$5\frac{1}{2}$	$1\frac{1}{4}$
$5\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{1}{2}$

Table 4-7.5-2 Key Size Versus Shaft Diameter (DIN 6885-1)

Nominal Shaft Diameter, mm Over	To	Nominal Key Size, mm
6	8	2 × 2
8	10	3 × 3
10	12	4 × 4
12	17	5 × 5
17	22	6 × 6
22	30	8 × 7
30	38	10 × 8
38	44	12 × 8
44	50	14 × 9
50	58	16 × 10
58	65	18 × 11
65	75	20 × 12
75	85	22 × 14

Table 4-7.6.1-1 Fatigue Stress Amplification Factors

Service Class	Fatigue Stress Amplification Factor, K_A
0	1.015
1	1.030
2	1.060
3	1.125
4	1.250

S_f = computed fatigue stress, ksi (MPa)
S_t = computed axial tensile stress, ksi (MPa)

(*b*) Combined shear fatigue stress shall not exceed

$$\tau_f = (K_{ST})\tau \leq \frac{S_{ec}}{K_A\sqrt{3}} \quad (4\text{-}13)$$

where
K_{ST} = stress amplification factor for torsional shear
τ_f = computed combined fatigue shear stress, ksi (MPa)

(*c*) Combined axial/bending and shear fatigue stresses where all are fluctuating shall not exceed

$$S_f = \sqrt{(K_{TD}S_t + K_{TB}S_b)^2 + 3(K_{ST}\tau)^2} \leq \frac{S_{ec}}{K_A} \quad (4\text{-}14)$$

(*d*) Combined tensile and shear fatigue stresses where only part of the stresses are fluctuating shall not exceed

$$S_f = \sqrt{\left(S_{av}\frac{S_{ec}}{S_y} + K_TS_R\right)^2 + 3\left(\tau_{av}\frac{S_{ec}}{S_y} + K_{ST}\tau_R\right)^2} \leq \frac{S_{ec}}{K_A} \quad (4\text{-}15)$$

where
K_T = larger of either K_{TD} and K_{TB}
S_{av} = portion of the computed tensile stress not due to fluctuating loads, ksi (MPa)
S_R = portion of the computed tensile stress due to fluctuating loads, ksi (MPa)
S_y = specified minimum yield strength, ksi (MPa)
τ_{av} = portion of the computed shear stress not due to fluctuating loads, ksi (MPa)
τ_R = portion of the computed shear stress due to fluctuating loads, ksi (MPa)

4-7.7 Shaft Displacement

Shafts shall be sized or supported so as to limit displacements under load when necessary for proper functioning of mechanisms or to prevent excessive wear of components.

4-8 FASTENERS

4-8.1 Fastener Markings

All bolts, nuts, and cap screws shall have required ASTM or SAE grade identification markings.

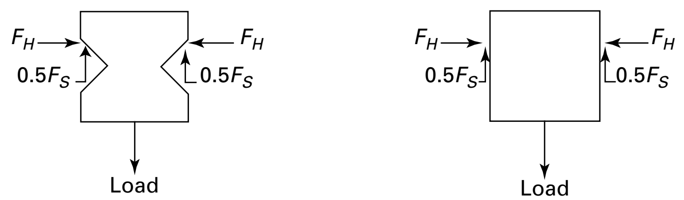
4-8.2 Fastener Selection

Fasteners for machine drives or other operational critical components shall use ASTM A325, SAE Grade 5, ASTM A490, or SAE Grade 8 bolts, cap screws, or equivalents.

4-8.3 Fastener Stresses

Bolt stress shall not exceed the allowable stress values established by eqs. (3-40) through (3-43) and para. 3-4.5.

Fig. 4-9.2-1 Illustration of Holding and Support Forces (14)



(a) Indentation Lifter **(b) Pressure-Gripping Lifter**

4-8.4 Fastener Integrity

Locknuts, double nuts, lock washers, chemical methods, or other means determined by the lifting device manufacturer or a qualified person shall be used to prevent the fastener from loosening due to vibration. Any loss of strength in the fastener caused by the locking method shall be accounted for in the design.

4-8.5 Fastener Installation

Fasteners shall be installed by an accepted method as determined by the lifting device manufacturer or a qualified person.

4-8.6 Noncritical Fasteners

Fasteners for covers, panels, brackets, or other noncritical components shall be selected by the lifting device manufacturer or a qualified person to meet the needs of the application.

(14) 4-9 GRIP SUPPORT FORCE

4-9.1 Purpose

This section sets forth requirements for the minimum support force for pressure-gripping (friction-type) and indentation-type lifters. Factors such as type and condition of gripping surfaces, environmental conditions, coefficients of friction, dynamic loads, and product temperature can affect the required support force and shall be considered during the design by a qualified person. In addition, lifters such as bar tongs and vertical axis coil grabs have other special load handling conditions (e.g., opening force) that should be considered.

4-9.2 Pressure-Gripping and Indentation Lifter Support Force

The coefficient of friction, static or dynamic as applicable, shall be determined by a qualified person through testing or from published data. The illustrations in Fig. 4-9.2-1 demonstrate the two ways friction forces may be applied.

$$F_S \geq 2.0 \times \text{load} \tag{4-16}$$

where

F_H = minimum force on each side of load, lb (N)

F_S = total support force created by lifter, lb (N)

load = weight of lifted load, lb (N)

4-10 VACUUM LIFTING DEVICE DESIGN

4-10.1 Vacuum Pad Capacity

(a) The ultimate pad capacity (*UPC*) shall be determined by eq. (4-17).

NOTE: Consistent units or unit conversions shall be used.

$$UPC = A\,V_p \tag{4-17}$$

where

A = effective area of the vacuum pad enclosed between the pad and the material when the pad is fully compressed against the material surface to be lifted

V_p = minimum vacuum specified at the pad

The value of V_p shall consider the altitude where the lifting device will be used.

(b) The *UPC* shall be reduced to a maximum vacuum pad rating (*VPR*).

$$VPR = UPC/N_v \tag{4-18}$$

where

N_v = 2 + 2 sin θ

θ = angle of vacuum pad interface surface measured from horizontal

The N_v value calculated in eq. (4-18) is for clean, flat, dry, nonporous surfaces, and shall be increased as required due to the surface conditions of interfacing materials as determined by a qualified person. Consideration should be given to conditions such as surface temperatures, contamination, torsion and bending loading of the vacuum pad, and tested vacuum pad performance.

4-10.2 Vacuum Preservation

The vacuum lifter shall incorporate a method to prevent the vacuum level under the pad(s) from decreasing more than 25% (starting from rated vacuum level) in 5 min without primary power and the vacuum pad(s) attached to a clean, dry, and nonporous surface at the rated load. Consideration should be given to conditions such as surface temperatures, contamination, torsion, and bending loads of the vacuum pad, tested vacuum pad performance, and surface conditions of interfacing materials. Unintended loss of power shall not disconnect the pad(s) from the vacuum preservation method.

4-10.3 Vacuum Indicator

A vacuum indicator shall be visible to the lifter operator during use and shall continue to function during an unintended loss of power. It shall indicate the presence of the minimum vacuum required for the rated load of the vacuum lifting device.

4-11 FLUID POWER SYSTEMS

4-11.1 Purpose

This section identifies requirements of fluid power systems and components for below-the-hook lifting devices.

4-11.2 Fluid Power Components

(a) The lifting device manufacturer or qualified person shall specify system components such as cylinders, pumps, valves, pipes, hoses, and tubes. Fluid power systems should be designed so that loss of the lifter power source(s), fluid loss, or control system failure will not result in uncontrolled movement of the load.

(b) Each hydraulic fluid power component shall be selected based on the manufacturer's rating and the maximum pressure applied to that component of the system, provided that the rating is based on a design factor equal to or greater than $1.67N_d$.

(c) Each pneumatic fluid power component shall be selected based on the maximum pressure applied to that component of the system and a rating equal to the manufacturer's rating divided by $0.50N_d$. Alternately, pneumatic fluid power components may be selected in accordance with para. 4-11.2(b).

(d) Components whose failure will not result in uncontrolled movement of the load may be selected based on the manufacturer's rating.

4-11.3 Power Source/Supply

Where the lifter uses an external fluid power source that is not part of the below-the-hook lifter, the supply requirements, which shall include the maximum sum of all fluid power components possible to actuate at one time, shall be detailed in the specifications.

4-11.4 Fluid Pressure Indication

If a change in fluid pressure could result in uncontrolled movement of the load, an indicator should be provided to allow the lifter operator to verify that the fluid pressure is sufficient during all stages of lifter use. Additional indicators may be necessary to allow monitoring of various systems. The fluid pressure indicator(s), if provided, shall be clearly visible or audible.

4-11.5 Fluid Pressure Control (14)

The fluid power system shall be equipped with a means to release stored energy and to verify that the system is at a zero-energy state. Hydraulic fluid shall not be discharged to atmosphere.

The system shall be designed to protect against pressures exceeding the rating of the system or any component.

4-11.6 System Guarding (14)

Fluid power tubing, piping, components, and indicators should be located or guarded to resist damage resulting from collision with other objects and whipping in the event of failure.

4-12 LIFTING MAGNETS (14)

(a) The control handle of a manually controlled permanent magnet shall include a device that will hold the magnetic circuit in either the "Load" or "Release" position to prevent inadvertent changes.

(b) Close proximity lifting magnets shall be designed to Design Category B (static strength criteria) and the proper Service Class (fatigue life criteria) selected for the number of load cycles.

(c) Remotely operated lifting magnets shall be designed to Design Category B (static strength criteria), and the proper Service Class (fatigue life criteria) selected for its number of load cycles.

(d) Lifting magnets should be weather resistant.

Chapter 5
Electrical Components

5-1 GENERAL

5-1.1 Purpose

This chapter sets forth selection criteria for electrical components of a below-the-hook lifting device.

5-1.2 Relation to Other Standards

Components of electrical equipment used to operate a below-the-hook lifting device shall conform to the applicable sections of ANSI/NFPA 70, National Electrical Code.

5-1.3 Power Requirements

The electrical power supply and control power requirements for operating a lifting device shall be detailed in the specifications. The supply requirements shall include the maximum full load amperage draw based on the operating conditions that will create the largest demand on the system.

5-2 ELECTRIC MOTORS AND BRAKES

5-2.1 Motors

Motors shall be reversible and have anti-friction bearings and totally enclosed frames. Motors used to operate hydraulic and vacuum equipment shall be continuous duty. Other motors used to operate a lifting device may be 30 min or 60 min intermittent duty, provided they can meet the required duty cycle of the lifter without overheating. Motors shall have torque characteristics suitable for the lifting device application and be capable of operating at the specified speed, load, and number of starts.

5-2.2 Motor Sizing

Motors shall be sized so the rated motor torque is not exceeded within the specified working range and/or rated load of the lifting device.

5-2.3 Temperature Rise

Temperature rise in motors shall be in accordance with NEMA MG 1 for the class of insulation and enclosure used. Unless otherwise specified, the lifting device manufacturer shall assume 104°F (40°C) ambient temperature.

5-2.4 Insulation

The minimum insulation rating of motors and brakes shall be Class B.

5-2.5 Brakes

Electric brakes shall be furnished whenever the lifted load could cause the gearing to back drive and allow unintended movement of the load. Brakes shall be electric release spring-set type. Brake torque shall hold a minimum of 150% rated motor torque or 150% of back driving torque, whichever is greater.

5-2.6 Voltage Rating

Motor and brake nameplate voltage shall be in accordance with NEMA MG 1 for the specified power supply. The installer/user shall ensure the voltage delivered to the terminals of the lifting device is within the tolerance set by NEMA.

5-3 OPERATOR INTERFACE

5-3.1 Locating Operator Interface

A qualified person shall choose a location for the operator interface in order to produce a safe and functional electrically powered lifting device. The lifting device specifications shall state the location of the operator interface chosen by a qualified person from the following options:

(a) push buttons or lever attached to the lifter
(b) pendant station push buttons attached to the lifter
(c) pendant station push buttons attached to the hoist or crane
(d) push buttons or master switches located in the crane cab
(e) handheld radio control or infrared transmitter
(f) automated control system

5-3.2 Unintended Operation

A qualified person shall choose the location and guarding of push buttons, master switches, or other operating devices that are used to open, drop, or release a load from a lifter. In order to inhibit unintentional operation of the lifter, one of the following options should be considered:

(a) Use two push buttons in series spaced such that they require two-handed operation in order to open, drop, or release a load from a lifter.

(b) Use one or more limit switches and/or sensors to confirm a load is lifted or suspended, in series with the open, drop, or release push button in order to inhibit open, drop, or release motion while the load is lifted.

(c) Use a mechanical guard or cover over the actuation device that requires two specific operations to activate the device.

5-3.3 Operating Levers

Cab operated master switches shall be spring return to neutral (off) position type, except that those for electromagnet or vacuum control shall be maintained type.

5-3.4 Control Circuits

Control circuit voltage of any lifter shall not exceed 150 volts AC or 300 volts DC.

5-3.5 Push-Button Type

Push buttons and control levers shall return to the "off" position when pressure is released by the operator, except for electromagnet or vacuum control which should be maintained type.

5-3.6 Push-Button Markings

Each push button, control lever, and master switch shall be clearly marked with appropriate legend plates describing resulting motion or function of the lifter.

5-3.7 Sensor Protection

Limit switches, sensors, and other control devices, if used, shall be located, guarded, and protected to inhibit inadvertent operation and damage resulting from collision with other objects.

(14) 5-3.8 Indicators

Indication or signal lights should be provided to indicate power is "on" or "off." If provided, the lights shall be located so that they are visible to the lifter operator. Multiple bulbs may be provided to avoid confusion due to a burned-out bulb.

(14) 5-4 CONTROLLERS AND AUXILIARY EQUIPMENT

5-4.1 Control Considerations

This section covers requirements for selecting and controlling the direction, speed, acceleration, and stopping of lifting device motors. Other control requirements such as limit switches, master switches, and push buttons are covered in section 5-3.

5-4.2 Control Location

Controls mounted on the lifting device shall be located, guarded, and designed for the environment and impacts expected.

5-4.3 Control Selection

A qualified person designated by the manufacturer and/or owner, purchaser, or user of a motor driven device shall determine the type and size of control to be used with the lifter for proper and safe operation. Control systems may be manual, magnetic, static, inverter (variable frequency), electric/electronic, or in combination.

5-4.4 Magnetic Control Contactors

Control systems utilizing magnetic contactors shall have sufficient size and quantity for starting, accelerating, reversing, and stopping the lifter. NEMA rated contactors shall be sized in accordance with NEMA ICS 2. Definite purpose contactors specifically rated for crane and hoist duty service or IEC contactors may be used for Service Classes 0, 1, and 2, provided the application does not exceed the contactor manufacturer's published rating. Reversing contactors shall be interlocked.

5-4.5 Static and Inverter Controls

Control systems utilizing static or inverter assemblies shall be sized with due consideration of motor, rating, drive requirements, service class, duty cycle, and application in the control. If magnetic contactors are included within the static assembly, they shall be rated in accordance with para. 5-4.4.

5-4.6 Lifting Magnet Controllers (14)

(a) Provisions shall be made for maintaining the control switch in position per section 5-3.2 to protect it from unintended operation.

(b) Loss of the crane or magnet control signal shall not result in de-energizing the lifting magnet.

5-4.7 Rectifiers

Direct current powered lifters may incorporate a single-phase full wave bridge rectifier for diode logic circuitry to reduce the number of conductors required between the lifter and the control. The rectifier shall be selenium or silicon type, sized to withstand the stalled current of the motor. Silicon type rectifiers shall employ transient suppressors to protect the rectifier from voltage spikes.

5-4.8 Electrical Enclosures

Control panels shall be enclosed and shall be suitable for the environment and type of controls. Enclosure types shall be in accordance with NEMA ICS 6 classifications.

5-4.9 Branch Circuit Overcurrent Protection

Control systems for motor powered lifters shall include branch circuit overcurrent protection as specified in ANSI/NFPA 70. These devices may be part of the hoisting equipment from which the lifter is suspended, or may be incorporated as part of the lifting device.

5-4.10 System Guarding (14)

Electrical components shall be guarded or located so that persons or objects cannot inadvertently come into contact with energized components under normal operating conditions.

5-5 GROUNDING

Electrically operated lifting devices shall be grounded in accordance with ANSI/NFPA 70.

5-5.1 Grounding Method

Special design considerations shall be taken for lifters with electronic equipment. Special wiring, shielding, filters, and grounding may need to be considered to account for the effects of electromagnetic interference (EMI), radio frequency interference (RFI), and other forms of emissions.

5-6 POWER DISCONNECTS

5-6.1 Disconnect for Powered Lifter

Control systems for motor powered lifters shall include a power disconnect switch as specified in ANSI/NFPA 70. This device may be part of the hoisting equipment from which the lifter is suspended, or may be incorporated as part of the lifting device.

(14) 5-6.2 Disconnect for Vacuum Lifter

(a) Hoisting equipment using an externally powered vacuum lifter shall have a separate vacuum lifter circuit switch of the enclosed type and shall be capable of being locked in the open (off) position. The provision for locking or adding a lock to the disconnecting means shall be installed on or at the switch or circuit breaker used as the disconnecting means and shall remain in place with or without the lock installed. Portable means for adding a lock to the switch or circuit breaker shall not be permitted.

(b) The vacuum lifter disconnect switch, when required by NFPA 70, shall be connected on the line side (power supply side) of the hoisting equipment disconnect switch.

(14) 5-6.3 Disconnect for Magnet

(a) Hoisting equipment with an externally powered electromagnet shall have a separate magnet circuit switch of the enclosed type and shall be capable of being locked in the open (off) position. The provision for locking or adding a lock to the disconnecting means shall be installed on or at the switch or circuit breaker used as the disconnecting means and shall remain in place with or without the lock installed. Portable means for adding a lock to the switch or circuit breaker shall not fbe permitted. Means for discharging the inductive energy of the magnet shall be provided.

(b) The magnet lifter disconnect switch, when required by NFPA 70, shall be connected on the line side (power supply side) of the hoisting equipment disconnect switch. Power supplied to lifting magnets from DC generators can be disconnected by disabling the external power source connected to the generator, or by providing a circuit switch that disconnects excitation power to the generator and removes all power to the lifting magnet.

5-6.4 Generator Supplied Magnets

Power supplied to magnets from DC generators can be disconnected by disabling the external power source connected to the generator, or by providing a circuit switch that disconnects excitation power to the generator and removes all power to the magnet.

5-7 BATTERIES

5-7.1 Battery Condition Indicator

Battery operated lifters or lifting magnets shall contain a device indicating existing battery conditions.

5-7.2 Enclosures

Battery enclosures or housings for wet cell batteries shall be vented to prevent accumulation of gases.

5-7.3 Battery Alarm

Battery backup systems shall have an audible or visible signal to warn the lifter operator when the primary power is being supplied by the backup battery(ies).

INTENTIONALLY LEFT BLANK

NONMANDATORY APPENDIX A COMMENTARY FOR CHAPTER 1: SCOPE, DEFINITIONS, AND REFERENCES[1]

(14)

A-1 PURPOSE

This Standard has been developed in response to the need to provide clarification of the intent of ASME B30.20 with respect to the structural design of below-the-hook lifting devices. Since the first edition of ASME B30.20 in 1986, users have requested interpretations of the construction (structural design) requirements stated therein. The level of detail required to provide adequate answers to the questions submitted extends beyond that which can be covered by interpretations of a B30 safety standard.

A-2 SCOPE

ASME BTH-1 addresses only design requirements. As such, this Standard should be used in conjunction with ASME B30.20, which addresses safety requirements. ASME BTH-1 does not replace ASME B30.20. The design criteria set forth are minimum requirements that may be increased at the discretion of the lifting device manufacturer or a qualified person.

The design of lifting attachments may be addressed by existing industry design standards. In the absence of such design standards, a qualified person should determine if the provisions of ASME BTH-1 are applicable.

A-3 NEW AND EXISTING DEVICES

It is not the intent of this Standard to require retrofitting of existing lifting devices.

A-4 GENERAL REQUIREMENTS

A-4.1 Design Responsibility

Although always implied, this provision now explicitly states that the design of below-the-hook lifting devices is the responsibility of a qualified person. This requirement has been established in recognition of the impact that the performance of a lifting device has on workplace safety, the complexity of the design process, and the level of knowledge and training required to competently design lifting devices.

A-4.2 Units of Measure

The requirements of this Standard are presented wherever possible in a manner that is dimensionally independent, thus allowing application of these requirements using either U.S. Customary units (USCU) or International System of Units (SI). U.S. Customary units are the primary units used in this Standard.

A-4.3 Design Criteria

The original ASME B30.20 structural design requirements defined a lifting device only in terms of its rated load. Later editions established fatigue life requirements by reference to AWS D14.1/D14.1M. ASME BTH-1 now defines the design requirements of a lifter in terms of the rated load, Design Category, and Service Class to better match the design of the lifter to its intended service. An extended discussion of the basis of the Design Categories and Service Classes can be found in Nonmandatory Appendices B and C (commentaries for Chapters 2 and 3, respectively).

A-4.4 Analysis Methods

The allowable stresses defined in Chapters 3 and 4 have been developed based on the presumption that the actual stresses due to design loads will be computed using classical methods. Such methods effectively compute average stresses acting on a structural or mechanical element.

Consideration of the effects of stress concentrations is not normally required when determining the static strength of a lifter component (see Nonmandatory Appendix C, para. C-5.2). However, the effects of stress concentrations are most important when determining fatigue life. Lifting devices often are constructed with discontinuities or geometric stress concentrations such as pin and bolt holes, notches, inside corners, and shaft keyways that act as initiation sites for fatigue cracks.

Analysis of a lifting device with discontinuities using linear finite element analysis will typically show peak stresses that indicate failure, where failure is defined as the point at which the applied load reaches the loss of function load (or limit state) of the part or device under consideration. This is particularly true when evaluating static strength. While the use of such methods is not prohibited, modeling of the device and interpretation of the results demand suitable expertise to ensure the

[1] This Nonmandatory Appendix contains commentary that may assist in the use and understanding of Chapter 1. Paragraphs in this Appendix correspond with paragraphs in Chapter 1.

requirements of this Standard are met without creating unnecessarily conservative limits for static strength and fatigue life.

A-4.5 Material

The design provisions in Chapters 3 and 4 are based on practices and research for design using carbon, high-strength low-alloy, and heat-treated constructional alloy steels. Some of the equations presented are empirical and may not be directly applicable to use with other materials. Both ferrous and nonferrous materials, including the constructional steels, may be used in the mechanical components described in Chapter 4.

Industry-wide specifications are those from organizations such as ASTM International (ASTM), American Iron and Steel Institute (AISI), and Society of Automotive Engineers (SAE). A proprietary specification is one developed by an individual manufacturer.

A-4.6 Welding

AWS D14.1/D14.1M is cited as the basis for weld design and welding procedures. This requirement is in agreement with CMAA #70 and those established by ASME B30.20. Because of the requirement for nondestructive examination of Class 1 and 2 weld joints, AWS D14.1/D14.1M was selected over the more commonly known AWS D1.1 (refer to AWS D14.1/D14.1M, section 10.8). Fabricators that utilize personnel and procedures that are qualified under earlier editions of AWS D14.1/D14.1M, AWS D1.1, or Section IX of the ASME Boiler and Pressure Vessel Code are qualified to perform duties under AWS D14.1/D14.1M, provided that they meet any additional requirements that are mandated by AWS D14.1/D14.1M (refer to AWS D14.1/D14.1M, para. 9.1.4). The allowable stresses for welds are modified in this Standard to provide the higher design factors deemed necessary for lifting devices.

A-4.7 Temperature

The temperature limits stated are based on the following. Historically, tension brittle failures have occurred during hydrotest in pressure vessels fabricated from low carbon steel at temperatures as high as 50°F (10°C). Flaws in steel plate material were the primary cause of these failures. With tighter production processes, closer metallurgical control, and better quality checks in current practice, the risk of such failure is reduced. Thus, the Committee selected the 25°F (−4°C) temperature as a reasonable lower limit. This lower temperature limit is also consistent with recommendations made by AISC (2003).

The Committee selected the upper temperature limit as a reasonable maximum temperature of operation in a summer desert environment. Data from the ASME Boiler and Pressure Vessel Code material design tables indicate that some carbon steels have already begun to decline in both yield stress and allowable tension stress at 200°F (93°C). Some materials decline by as much as 4.6%, but most are less. A straight-line interpolation between the tabulated values for materials at 100°F (38°C) and 200°F (93°C) in this reference gives acceptable stress values that have minimal degradation at 150°F (66°C).

In some industrial uses, lifting devices can be subjected to temperatures in excess of 1,000°F (540°C). At these temperatures, the mechanical properties of most materials are greatly reduced over those at ambient. If the exposure is prolonged and cyclic in nature, the creep rupture strength of the material, which is lower than the simple elevated temperature value, must be used in determining the design rated load and life of the device.

Of importance when evaluating the effects of temperature is the temperature of the lifter component rather than the ambient temperature. A lifter may move briefly through an area of frigid air without the temperature of the steel dropping to the point of concern. Likewise, a lifter that handles very hot items may have some components that become heated due to contact.

A-5 DEFINITIONS

This section presents a list of definitions applicable to the design of below-the-hook lifting devices. Definitions from the ASME Safety Codes and Standards Lexicon and other engineering references are used wherever possible. The defined terms are divided into general terms (para. 1-5.1) that are considered broadly applicable to the subject matter and into groups of terms that are specific to each chapter of the Standard.

A-6 SYMBOLS

The symbols used in this Standard are generally in conformance with the notation used in other design standards that are in wide use in the United States, such as the AISC specification (AISC, 1989) and the crane design specifications published by AIST and CMAA (AIST Technical Report No. 6 and CMAA #70, respectively). Where notation did not exist, unique symbols are defined herein and have been selected to be clear in meaning to the user.

A-7 REFERENCES

ASME BTH-1 is structured to be a stand-alone standard to the greatest extent practical. However, some areas are best suited to be covered by reference to established industry standards. Section 1-7 lists codes, standards, and other documents that are cited within the main body of this Standard and provides the names and addresses of the publishers of those documents.

Each chapter of this Standard has a related Nonmandatory Appendix that explains, where necessary, the basis of the provisions of that chapter. All publications cited in these Nonmandatory Appendices are listed below. These references are cited for information only.

Cornell, C. A., 1969, "A Probability-Based Structural Code," *ACI Journal,* Vol. 66, No. 12

Publisher: American Concrete Institute (ACI), 38800 Country Club Drive, Farmington Hills, MI 48331 (www.concrete.org)

Ellifritt, D. S., Wine, G., Sputo, T., and Samuel, S., 1992, "Flexural Strength of WT Sections," *Engineering Journal,* Vol. 29, No. 2

"Engineering FAQs Section 4.4.2," (www.aisc.org, 2003)

Guide for the Analysis of Guy and Stiffleg Derricks, 1974

Load and Resistance Factor Design Specification for Structural Steel Buildings, 1994 and 2000

Specification for Structural Steel Buildings, 2010

Specification for Structural Steel Buildings — Allowable Stress Design and Plastic Design, 1989

Yura, J. A., and Frank, K. H., 1985, "Testing Method to Determine the Slip Coefficient for Coatings Used in Bolted Connections," *Engineering Journal*, Vol. 22, No. 3

Publisher: American Institute of Steel Construction (AISC), 1 East Wacker Drive, Suite 700, Chicago, IL 60601-2001 (www.aisc.org)

Madsen, J., November 1941, "Report of Crane Girder Tests," *Iron and Steel Engineer*

Technical Report No. 6, Specification for Electric Overhead Traveling Cranes for Steel Mill Service, 2000

Publisher: Association for Iron & Steel Technology (AIST), 186 Thorn Hill Road, Warrendale, PA 15086 (www.aist.org)

ANSI B15.1-2008 (Reaffirmation of ASME B15.1-2000), Safety Standards for Mechanical Power Transmission Apparatus (Withdrawn)

Publisher: Association for Manufacturing Technology (AMT), 7901 Westpark Drive, McLean, VA 22102-4206 (www.amtonline.org)

ANSI/ABMA 9-1990 (R2000), Load Rating and Fatigue Life for Ball Bearings[2]

ANSI/ABMA 11-1990 (R1999), Load Rating and Fatigue Life for Roller Bearings[2]

Publisher: American Bearing Manufacturers Association (ABMA), 2025 M Street, NW, Washington, D.C. 20036 (www.abma-dc.org)

ANSI/AGMA 2001-C95, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth[2]

Publisher: American Gear Manufacturers Association (AGMA), 500 Montgomery Street, Alexandria, VA 22314-1582 (www.agma.org)

[2] May also be obtained from the American National Standards Institute (ANSI), 25 West 43rd Street, New York, NY 10036.

ANSI/AWS D14.1-1997, Specification for Welding of Industrial and Mill Cranes and Other Material Handling Equipment[2]

AWS D1.1-2010, Structural Welding Code — Steel

Publisher: American Welding Society (AWS), 8669 NW 36 Street, Doral, FL 33166 (www.aws.org)

ANSI/NFPA 70-2011, National Electrical Code[2]

ANSI/NFPA 79-2002, Electrical Standard for Industrial Machinery[2]

Publisher: National Fire Protection Association (NFPA), 1 Batterymarch Park, Quincy, MA 02169 (www.nfpa.org)

API RP 2A-WSD, 2000, Planning, Designing, and Constructing Fixed Offshore Platforms — Working Stress Design

Publisher: American Petroleum Institute (API), 1220 L Street, NW, Washington, DC 20005-4070 (www.api.org)

ASME B17.1-1967 (R1998), Keys and Keyseats

ASME B30.2-2001, Overhead and Gantry Cranes (Top Running Bridge, Single or Multiple Girder, Top Running Trolley Hoist)

ASME B30.20-2003, Below-the-Hook Lifting Devices

ASME Boiler and Pressure Vessel Code, Section II, Part D, Properties, 2001 Edition, 2002 Addenda

ASME Boiler and Pressure Vessel Code, Section IX, Welding and Brazing Qualifications, 2001 Edition, 2002 Addenda

ASME HST-4–1999, Performance Standard for Overhead Electric Wire Rope Hoists

Bibber, L. C., Hodge, J. M., Altman, R. C., and Doty, W. D., 1952, "A New High-Yield-Strength Alloy Steel for Welded Structures," *Transactions,* Vol. 74, Part 3

Publisher: The American Society of Mechanical Engineers (ASME), Two Park Avenue, New York, NY 10016-5990; Order Department: 22 Law Drive, P.O. Box 2900, Fairfield, NJ 07007-2900 (www.asme.org)

Bjorhovde, R., Galambos, T. V., and Ravindra, M. K., 1978, "LRFD Criteria for Steel Beam-Columns," *Journal of the Structural Division*, Vol. 104, No. ST9

Duerr, D., 2006, "Pinned Connection Strength and Behavior," *Journal of Structural Engineering*, Vol. 132, No. 2

Dux, P. F., and Kitipornchai, S. (1990). "Buckling of Suspended I-Beams." *Journal of Structural Engineering*, 116(7), 1877–1891

Fisher, J. W., Galambos, T. V., Kulak, G. L., and Ravindra, M. K., 1978, "Load and Resistance Design Criteria for Connectors," *Journal of the Structural Division,* Vol. 104, No. ST9

Galambos, T. V., and Ravindra, M. K., 1978, "Properties of Steel for Use in LRFD," *Journal of the Structural Division,* Vol. 104, No. ST9

Kitipornchai, S., and Trahair, N. S., 1980, "Buckling Properties of Monosymmetric I-Beams," *Journal of the Structural Division*, Vol. 109, No. ST5

McWhorter, J. C., Wetencamp, H. R., and Sidebottom, O. M., April 1971, "Finite Deflections of Curved Beams," *Journal of the Engineering Mechanics Division*, Vol. 97, No. EM2

Ravindra, M. K., and Galambos, T. V., 1978, "Load and Resistance Factor Design for Steel," *Journal of the Structural Division*, Vol. 104, No. ST9

Yura, J. A., Galambos, T. V., and Ravindra, M. K., 1978, "The Bending Resistance of Steel Beams," *Journal of the Structural Division*, Vol. 104, No. ST9

Publisher: American Society of Civil Engineers (ASCE), 1801 Alexander Bell Drive, Reston, VA 20191-4400 (www.asce.org)

Lyse, I., and Godfrey, H. J., 1933, "Shearing Properties and Poisson's Ratio of Structural and Alloy Steels," *Proceedings*

Publisher: American Society for Testing and Materials (ASTM International), 100 Barr Harbor Drive, P.O. Box C700, West Conshohocken, PA 19428-2959 (www.astm.org)

Specification No. 70-2004, Specifications for Top Running Bridge & Gantry Type Multiple Girder Electric Overhead Traveling Cranes

Specification No. 74-2004, Specifications for Top Running & Under Running Single Girder Electric Traveling Cranes Utilizing Under Running Trolley Hoist

Publisher: Crane Manufacturers Association of America, Inc. (CMAA), 8720 Red Oak Boulevard, Charlotte, NC 28217 (www.mhia.org/cmaa)

DIN 6885-1 (1968), Drive Type Fastenings Without Taper Action; Parallel Keys, Keyways, Deep Pattern

Publisher: Deutsches Institut für Normung, e. V. (DIN) Am DIN-Platz Burggrafenstraße 6, 10787 Berlin, Germany (www.din.de)

SAE J1078-1994, A Recommended Method of Analytically Determining the Competence of Hydraulic Telescopic Cantilevered Crane Booms

Publisher: Society of Automotive Engineers (SAE International), 400 Commonwealth Drive, Warrendale, PA 15096-0001 (www.sae.org)

U.S. Department of Defense, 1998, DOD Handbook MILHDBK-1038, Weight Handling Equipment

29 CFR 1910.179, Overhead and Gantry Cranes

Publisher: Superintendent of Documents, U.S. Government Printing Office (GPO), 732 N. Capitol Street, NW Washington, D.C. 20401 (www.gpo.gov)

Wire Rope Users Manual, 3rd edition, 1993

Publisher: Wire Rope Technical Board (WRTB), 7011A Manchester Boulevard, Suite 178, Alexandria, VA 22310 (www.wireropetechnicalboard.org)

Avallone, E. A., and Baumeister, T., eds., 1987, *Marks' Standard Handbook for Mechanical Engineers*, 9th edition, McGraw-Hill, Inc., New York, NY

Blodgett, O. W., 1966, *Design of Welded Structures*, The James F. Lincoln Arc Welding Foundation, Cleveland, OH

Boresi, A. P., and Sidebottom, O. M., 1985, *Advanced Mechanics of Materials*, 4th edition, John Wiley & Sons, Inc., New York, NY

Galambos, T. V., ed., 1998, *Guide to Stability Design Criteria for Metal Structures*, 5th edition, John Wiley & Sons, Inc., New York, NY

Kulak, G. L., Fisher, J. W., and Struik, J. H. A., 1987, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd edition, John Wiley & Sons, Inc., New York, NY

Melcon, M. A., and Hoblit, F. M., 1953, "Developments in the Analysis of Lugs and Shear Pins," *Product Engineering*, Vol. 24, No. 6, pp. 160–170, McGraw-Hill, Inc., New York, NY

Pilkey, W. D., 2008, *Peterson's Stress Concentration Factors*, 3rd edition, John Wiley & Sons, Inc., New York, NY

Shigley, J. E., and Mischke, C. R., 2001, *Mechanical Engineering Design*, 6th edition, McGraw-Hill, Inc., New York, NY

Tolbert, R. N., 1970, "A Photoelastic Investigation of Lug Stresses and Failures," Master's Thesis, Vanderbilt University, Nashville, TN

Wilson, W. M., 1934, *The Bearing Value of Rollers*, Bulletin No. 263, University of Illinois Engineering Experiment Station, Urbana, IL

Young, W. C., and Budynas, R. G., 2002, *Roark's Formulas for Stress and Strain*, 7th edition, McGraw-Hill, Inc., New York, NY

NONMANDATORY APPENDIX B COMMENTARY FOR CHAPTER 2: LIFTER CLASSIFICATIONS[1]

(14)

B-1 GENERAL

B-1.1 Selection

The selection of a Design Category and Service Class allows the strength and useful life of the lifter to be matched to the needs of the user. A qualified person or manufacturer must assure that the Design Category and Service Class specified for a particular lifter are appropriate for the intended use so as to provide a design with adequate structural reliability and expected service life.

B-1.3 Identification

The purpose of this requirement is to ensure that the designer, manufacturer, and end user are aware of the assigned Design Category and Service Class. Typically, documents that require the indicated markings may include top level drawings, quotations, calculations, and manuals.

B-1.4 Environment

Ambient operating temperature limits are intended only to be a guideline. The component temperature of each part of the lifter must be considered when the device is operating in an environment outside the limits defined in para. 1-4.7. The effects of dust, moisture, and corrosive atmospheric substances on the integrity and performance of a lifter cannot be specifically defined. These design considerations must be evaluated and accounted for by the lifting device manufacturer or qualified person.

B-2 DESIGN CATEGORY

When selecting a Design Category, consideration shall be given to all operations that will affect the lifting device design. The discussions of the Design Categories below and in Nonmandatory Appendix C, para. C-1.3 refer to considerations given to unintended overloads in development of the design factors. These comments are in no way to be interpreted as permitting a lifting device to be used above its rated load under any circumstances other than for load testing in accordance with ASME B30.20 or other applicable safety standards or regulations.

[1] This Nonmandatory Appendix contains commentary that may assist in the use and understanding of Chapter 2. Paragraphs in this Appendix correspond with paragraphs in Chapter 2.

B-2.1 Design Category A

The design factor specified in Chapter 3 for Design Category A lifters is based on presumptions of rare and only minor unintended overloading, mild impact loads during routine use, and a maximum impact multiplier of 50%. These load conditions are characteristic of use of the lifter in work environments where the weights of the loads being handled are reasonably well known, and the lifting operations are conducted in a controlled manner. Typical characteristics of the application for this Design Category include lifts at slow speeds utilizing a well-maintained lifting device under the control of a lift supervisor and experienced crane operator. This Design Category should not be used in any environment where severe conditions or use are present.

Design Category A is intended to apply to lifting devices used in controlled conditions. Practical considerations of various work environments indicate that the high numbers of load cycles that correspond to Service Class 1 and higher commonly equate to usage conditions under which the design factor of Design Category A is inappropriate. Thus, the use of Design Category A is restricted to lifting device applications with low numbers of load cycles (Service Class 0).

B-2.2 Design Category B

The design factor specified in Chapter 3 for Design Category B lifters is based on presumptions (compared to Design Category A) of a greater uncertainty in the weight of the load being handled, the possibility of somewhat greater unintended overloads, rougher handling of the load, which will result in higher impact loads, and a maximum impact multiplier of 100%. These load conditions are characteristic of use of the lifter in work environments where the weights of the loads being handled may not be well known, and the lifting operations are conducted in a more rapid, production-oriented manner. Typical characteristics of the application for this Design Category include rough usage and lifts in adverse, less controlled conditions. Design Category B will generally be appropriate for lifters that are to be used in severe environments. However, the Design Category B design factor does not necessarily account for all adverse environmental effects.

Table B-3-1 Service Class Life

Load Cycles per Day	Desired Life, yr 1	5	10	20	30
5	0	0	0	1	1
10	0	0	1	1	2
25	0	1	1	2	2
50	0	1	2	2	3
100	1	2	2	3	3
200	1	2	3	3	4
300	2	3	3	4	4
750	2	3	4	4	4
1,000	2	3	4	4	4

B-3 SERVICE CLASS

Design for fatigue involves an economic decision between desired life and cost. The intent is to provide the owner with the opportunity for more economical designs for the cases where duty service is less severe. A choice of five Service Classes is provided. The load cycle ranges shown in Table 2-3-1 are consistent with the requirements of AWS D14.1/D14.1M.

Table B-3-1 may assist in determining the required Service Class based on load cycles per day and service life desired.

NONMANDATORY APPENDIX C COMMENTARY FOR CHAPTER 3: STRUCTURAL DESIGN[1]

(14)

C-1 GENERAL

C-1.1 Purpose

The member allowable stresses defined in Chapter 3 have generally been derived based on the assumption of the members being prismatic. Design of tapered members may require additional considerations. References such as AISC (2000), Appendix F3, and Blodgett (1966), Section 4.6 may be useful for the design of tapered members.

C-1.2 Loads

The structural members and mechanical components of a below-the-hook lifting device are to be designed for the forces imposed by the lifted load (a value normally equal to the rated load), the weight of the device's parts, and any forces such as gripping or lateral forces that result from the function of the device. The inclusion of lateral forces in this paragraph is intended to refer to calculated lateral forces that occur as a result of the intended or expected use of the lifter. This provision is not intended to require the use of an arbitrary lateral load in lifter design. For most designs, an added impact allowance is not required. This issue is discussed further in paras. C-1.3 and C-5.1.

C-1.3 Static Design Basis

The static strength design provisions defined in Chapter 3 have been derived using a probabilistic analysis of the static and dynamic loads to which lifters may be subjected and the uncertainties with which the strength of the lifter members and connections may be calculated. The load and strength uncertainties are related to a design factor N_d using eq. (C-1) (Cornell, 1969; Shigley and Mischke, 2001).

$$N_d = \frac{1 + \beta\sqrt{V_R^2 + V_S^2 - \beta^2 V_R^2 V_S^2}}{1 - \beta^2 V_R^2} \tag{C-1}$$

The term V_R is the coefficient of variation of the element strength. Values of the coefficient of variation for different types of structural members and connections have been determined in an extensive research program sponsored by the American Iron and Steel Institute (AISI) and published in a series of papers in the September 1978 issue (Vol. 104, No. ST9) of the *Journal of the Structural Division* from the American Society of Civil Engineers. Maximum values of V_R equal to 0.151 for strength limits of yielding or buckling and 0.180 for strength limits of fracture and for connection design were taken from this research and used for development of the BTH design factors.

The term V_S is the coefficient of variation of the spectrum of loads to which the lifter may be subjected. The BTH Committee developed a set of static and dynamic load spectra based on limited crane loads research and the experience of the Committee members.

Design Category A lifters are considered to be used at relatively high percentages of their rated loads. Due to the level of planning generally associated with the use of these lifters, the likelihood of lifting a load greater than the rated load is considered small and such overloading is not likely to exceed 5%. The distribution of lifted loads relative to rated load is considered to be as shown in Table C-1.3-1.

A similar distribution was developed for dynamic loading. AISC (1974) reports the results of load tests performed on stiffleg derricks in which dynamic loading to the derrick was measured. Typical dynamic loads were approximately 20% of the lifted load, and the upper bound dynamic load was about 50% of the lifted load. Tests on overhead cranes (Madsen, 1941) showed somewhat less severe dynamic loading. Given these published data and experience-based judgments, a load spectrum was established for dynamic loading (see Table C-1.3-2).

A second dynamic load spectrum was developed for a special case of Design Category A. Some manufacturers of heavy equipment such as power generation machinery build lifters to be used for the handling of their equipment. As such, the lifters are used at or near 100% of rated load for every lift, but due to the nature of those lifts, the dynamic loading can reasonably be expected to be somewhat less than the normal Design Category A lifters. The distribution developed for this special case is shown in Table C-1.3-2.

The range of total loads was developed by computing the total load (static plus dynamic) for the combination of the spectra shown in Tables C-1.3-1 and C-1.3-2. The appropriate statistical analysis yielded loading coefficients of variation of 0.156 for the standard design spectrum and 0.131 for the special case.

[1] This Nonmandatory Appendix contains commentary that may assist in the use and understanding of Chapter 3. Paragraphs in this Appendix correspond with paragraphs in Chapter 3.