SECTION 6 - TABLE OF CONTENTS- MECHANICAL DESIGN

SPECIFICATIONS

6.1 SCOPE

The provisions in this section apply to the design of the machinery for moving, aligning, and locking a movable bridge span. The section addresses the requirements for bascule, swing, and vertical lift movable spans.

COMMENTARY

C6.1

6.2 DEFINITIONS

Addendum - Portion of gear tooth outside (greater than) the pitch radius.

AGMA - American Gear Manufacturer's Association

Allowable Static Design Stress - Permissible value of stress for calculations involving components subjected to static loading.

Average (mean) Stress - One-half of the sum of the maximum and minimum stress.

Backlash - The smallest amount of space between the faces of mating gears.

Bevel Gear - Type of gear commonly used when shafts intersect and utilizes the concept of rolling cones.

Brittle - Materials designed against ultimate strength for which failure means fracture; easily broken snapped or cracked.

Contact Stress Failure - Failure of gear teeth based on projected area of contact.

Crank Pins - Joint between linkages where stress alternates between application and release.

Cyclic Stress - Stress range which follows a pattern over and over.

Dedendum - Portion of gear tooth from the root to the pitch line.

Deflector Sheaves - Component used on span drive vertical lift bridges to guide operating ropes from the top chord (horizontal) to the tower attachments (vertical).

Diametral Pitch - Index of gear tooth sizes that is defined as the number of teeth divided by the pitch diameter (in.).

Ductile - Materials designed against yield strength and failure is visible before fracture.

Enclosed Gearing - Gear set of which all moving elements are included in a given frame and cover that is dust-proof and oil-tight.

Endurance Limit Strength - Stress level at which completely reversing cyclic stress (fatigue) causes failure in one million (1×10^6) cycles; the ability to withstand fatigue loads.

Fatigue Failure - Point at which cyclic loading causes fracture or permanent deformation.

Fatigue Limit State - Limit state relating to cyclic stress and crack propagation.

Fatigue Strength - Ability to withstand cyclic loading.

Helical - A gear with a cylindrical pitch surface and teeth that are at an angle to the axis.

Herringbone - A type of helical gearing where half of the teeth are right-handed and the other half are left-handed.

Idler Gear - A gear that has the same number of teeth as the mating gear and, therefore, introduces no change in shaft RPM, but changes the rotational direction.

L-10 Life - Basic rating life of a component for 90% reliability based on load and speed data, typically given as a number of revolutions.

Lay - The manner in which the wires in a strand or the strands in a rope are helically positioned.

Mechanical Shrink-fit Assembly - Mechanical connection where assembly is performed by heating or cooling one element relative to the other, and when an equilibrium temperature is reached, an interference fit is produced.

Minimum Yield Strength - The lowest value of stress a material shows a specified limiting deviation from the proportionality of stress to strain.

Module - Metric index of gear tooth sizes that is defined as pitch diameter (mm) divided by number of teeth.

Open Gearing - Gear set that is not sealed and may have moving elements exposed to the environment.

Pitting Resistance/Wear/Surface Durability - AGMA terms used for rating gearing from the aspect of contact surface stress.

Service Limit State - Limit state relating to stress, deformation and cracking applied to normal operating loads.

Sheave - A pulley or wheel having a grooved rim, typically used for wire ropes on vertical lift bridges.

Spur Gear Teeth -Teeth on the cylindrical pitch surface of a gear that are parallel to the axis.

Stress Range - Maximum stress minus the minimum stress.

Uniaxial Tensile Stress - Stress acting along only one axis.

Yield Failure/Intermittent Overload - Overload condition for which yield failure may occur in spur gear teeth experiencing less than 100 cycles in its design life.

6.3 NOTATION

6.3.1 General

A	=	constant cross-sectional area (in.^{2}); projected area (in.^{2}) (6.6.1) (6.6.2.5)
a	=	constant for finding surface roughness factor (DIM) (6.6.3.2)
$\sqrt{a}$	=	Neuber constant for finding q (notch sensitivity) (in.$^{0.5}$) (6.7.3.2)
B	=	exponent quantity (6.7.5.2.2)
b	=	exponent for finding surface roughness factor (DIM); pinion/gear face width (mm); key width (rectangular or square) (in.) (6.6.3.2) (6.7.10.1)
^{o}C	=	degrees Celsius (6.7.5.2.2)
C_D	=	endurance limit modifying factor based on diameter (DIM) (6.6.3.2)
C_f	=	gear design surface finish factor (DIM) (6.7.5.2.3)
C_H	=	gear design hardness ratio factor (DIM) (6.7.5.2.3)
C_M	=	endurance limit modifying factor for miscellaneous condition (DIM) (6.6.3.2)
C_{or}	=	basic static load rating, element bearing (lb.) (6.7.7.2.4)
C_p	=	gear design elastic coefficient (psi$^{-0.5}$)(6.7.5.2.3)
C_R, C_S, C_T	=	reliability, surface, temperature factors, respectively (DIM) (6.6.3.2)
C_r	=	basic dynamic load rating, rolling element bearing (lb.) (6.7.7.2.2)

SPECIFICATIONS COMMENTARY

d = pitch diameter of pinion (in.); diameter of the wire rope (in.) (6.7.5.2.1) (C6.8.3.3.4)
D = diameter for use in various sliding bearings (in.); rolling bearings (in.); shaft diameter (in.); tread diameter of sheave rope grooves (in.) (6.6.2.5)&(6.7.7.1.2) (6.6.2.6) (6.6.3.2) (6.7.4.3) (6.8.3.3.4)
d_w = diameter of outer wires in the wire rope (in.) (6.8.3.3.4)
E_R = tensile modulus of elasticity of the wire rope (psi) (C6.8.3.3.6)
E_w = tensile modulus of elasticity of the steel wire (psi) (6.8.3.3.4)
°F = degrees Fahrenheit (6.7.5.2.2)
F = pinion/gear face width (in.) (6.7.5.2.2)
F_{oxy} = factored large bearing load (lb.) (6.7.7.2.4)
F_{ua} = applied axial load (lb.) (6.7.7.2.2)
F_{ur} = radial load (lb.) (6.7.7.1.2)
G = gear for J and I tables (DIM) (C6.7.5.2.2)
g = acceleration due to gravity = 32.2 (ft./s.^{2}) (6.7.4.3)
H_B = Brinell hardness number (BHN) (DIM) (6.6.4.2)
H_{BP}, H_{BG} = Brinell hardness number for pinion/gear (DIM) (6.7.5.2.3)
h = key height (in.) (6.7.10.1)
h_t = total gear tooth height (in.) (6.7.5.2.2)
I = gear design tooth geometry factor - surface durability (DIM) (6.7.5.2.3)
J = gear design tooth geometry factor - fatigue (DIM) (6.7.5.2.2)
K_B = gear design rim factor (DIM) (6.7.5.2.2)
K_F = fatigue stress concentration factor (normal stress) (DIM) (6.7.3.2)
K_{FS} = fatigue stress concentration factor (shear stress) (DIM) (6.7.3.2)
K_f = gear design stress correction factor (DIM) (6.7.5.2.4)
K_m = gear design load distribution factor (DIM) (6.7.5.2.2)
K_{my} = gear design load distribution factor for overload (DIM) (6.7.5.2.4)
K_o = gear design overload factor (DIM) (6.7.5.2.2)
K_R = gear design reliability factor (DIM) (6.7.5.2.2)
K_S = gear design tooth size factor (DIM) (6.7.5.2.2)
K_t = theoretical stress concentration factor (normal stress) (DIM) (6.7.3.2)
K_T = gear design temperature factor (DIM) (6.7.5.2.2)
K_{ts} = theoretical stress concentration factor (shear stress) (DIM) (6.7.3.2)
K_v = gear design velocity (dynamic) factor (DIM) (6.7.5.2.2)
K_y = gear design yield strength factor (DIM) (6.7.5.2.4)
k = radius of gyration (in.) (6.6.1)
L = length of shaft between supports (in.) (6.7.4.2)
L_{act} = actual length (in.) (6.6.1)
L_{eff} = effective length (in.) (6.6.1)
M_a = bending moment amplitude (lb.-in.) (6.7.4.1)
m_B = gear design backup ratio (DIM) (6.7.5.2.2)
m_t = gear design tooth module transverse (6.7.5.1)
N = gear design number of load cycles (DIM) (6.7.5.2.2)
n = rotational speed (RPM); rotational speed of bearing inner race (RPM) (6.6.2.5) (6.7.7.2.2)
n_c = critical shaft rotational speed (RPM) (6.7.4.3)
n_p = pinion speed (RPM) (C6.7.5.2.1)
N_p = gear design number of pinion teeth (DIM) (6.7.5.2.2)
n_S = static design factor (DIM) (6.6.1)
P = power which the gear transmits (hp); pinion for J and I tables; direct load on wire rope (lb.) (C6.7.5.2.1) (C6.7.5.2.2) (6.8.3.3.4)
P_{ac} = allowable transmitted power for gear design surface durability (hp) (C6.7.5.2.1)
P_{at} = allowable transmitted power for gear design fatigue (hp) (C6.7.5.2.1)
P_d = diametral pitch (in.$^{-1}$) (6.7.5.1)
P_o = operating loads, e.g., the larger of starting or inertial loads (lb.) (6.8.3.3.4)
P_{or} = factored radial design resistance (lb.) (6.7.7.2.4)
P_r = equivalent dynamic radial load for rolling element bearings (lb.) (6.7.7.2.2)

P_{ut}	=	wire rope minimum ultimate breaking load (lb.) (6.8.3.3.6)
p	=	allowable static design resistance in compression; bearing design pressure (psi) (6.6.1) (6.7.7.1.2)
Q_v	=	gear quality number (DIM) (6.7.5.2.2)
q	=	fatigue design notch sensitivity factor (DIM) (6.7.3.2)
R_a	=	surface arithmetic average roughness (μin.) (6.6.3.2)
R_b	=	bearing resistance (lb.) (6.6.2.5)
R_R	=	line bearing resistance on rollers (lb./in.) (6.6.2.6)
r	=	notch/fillet radius or radius of a hole (in.) (6.7.3.2)
S_{at}	=	gear design allowable stress - fatigue (psi) (6.6.4.2)
S_{ac}	=	gear design allowable stress - surface durability (psi) (6.6.4.3)
S_{ay}	=	gear design allowable yield (psi) (6.6.4.4)
S_F	=	safety factor for bending strength (DIM) (6.7.5.2.2)
S_H	=	surface durability design factor (DIM) (6.7.5.2.3)
T	=	torque, torsional moment (lb.-in.) (6.7.5.2.1)
T_m	=	mean-steady torque (lb.-in.) (6.7.4.1)
t_R	=	rim thickness (in.)(6.7.5.2.2)
V	=	velocity; journal surface speed (fpm) (6.7.7.1.2)
v_t	=	gear design pitch line velocity (fpm) (6.7.5.2.2)
W	=	weight (lb.) (6.7.4.3)
W_{max}	=	maximum peak transmitting gear force (lb.) (6.7.5.2.4)
W_t	=	tangential transmitting gear force (lb.) (6.7.5.2.1)
W_{tac}	=	factored surface durability resistance of spur gear teeth (lb.) (6.7.5.2.3)
W_{tat}	=	factored flexural resistance of spur gear teeth (lb.) (6.7.5.2.2)
X, X_o	=	bearing design factors (radial) (DIM) (6.7.7.2.2), (6.7.7.2.4)
Y, Y_o	=	bearing design factors thrust (axial) (DIM) (6.7.7.2.2), (6.7.7.2.4)
Y_N	=	gear design life factor - fatigue (DIM) (6.7.5.2.2)
Z_N	=	gear design life factor (DIM) (6.7.5.2.3)
α	=	factor for allowable bearing resistance (lb./in.•RPM); factor for allowable line bearing resistance (lb./in.); endurance limit factor (DIM) (6.6.2.5), (6.6.2.6), (6.6.3.2)
σ	=	normal stress (psi) (6.7.2.3)
σ'_a, σ'_m	=	Von Mises stresses (psi) (6.7.3.3.2)
σ_a	=	amplitude stress - fluctuating (psi) (6.7.3.3.1)
σ_b	=	maximum wire rope bending stress (psi) (6.8.3.3.4)
σ_e	=	endurance limit (psi) (6.6.3.2)
σ_m	=	mean or average stress (psi (6.7.3.3.1)
σ_{max}, σ_{min}	=	maximum and minimum cyclic stress (psi) (6.7.2.2) (6.7.3.3.1)
σ_t	=	maximum total stress in wire rope (psi) (6.8.3.3.4)
σ_{ut}	=	ultimate tensile strength (psi) (6.6.3.2)
$\sigma_y, \sigma_{yt}, \sigma_{yc}$	=	yield strength of material - min. (psi) (C6.6.1)
τ	=	shear stress (psi) (6.7.2.3)
τ_a, τ_m	=	amplitude and mean cyclic shear stresses (psi) (6.7.3.3.1)
τ_{max}	=	maximum shear stress resulting from applied loads (psi) (6.7.2.2)

6.4 GENERAL REQUIREMENTS

6.4.1 Machinery

6.4.1.1 LIMIT STATES AND RESISTANCE FACTORS

Unless otherwise stated, machinery design shall be based on the service and fatigue limit states using the loads and resistances specified herein.

C6.4.1.1

The design of bridge machinery in the United States is based on allowable working stress design, therefore, this section follows the accepted industry design practice. As

SPECIFICATIONS

These specifications require consideration of design equations to prevent fatigue failure of critical machine elements.

Where applicable, depending on the seismic design strategy chosen to comply with Article 3.4.3, some machinery may be required to resist seismic loads for which the extreme event limit state shall apply.

Unless specified otherwise, the resistance factors shall be applied to the general limit state Equation 1.3.2.1-1 shall be taken as:

- For the service limit state ....$\varphi = 1$
- For the fatigue limit state ....$\varphi = 1$
- For the overload limit state:
  - Forged, drawn, rolled, wrought steel ......$\varphi = 2.25$
  - Cast steel ....$\varphi = 3$
- For the extreme event limit state:
  - Forged, drawn, rolled, wrought steel ....$\varphi = 2.7$
  - Cast steel ....$\varphi = 3.6$

COMMENTARY

of this writing (1998), reliability-based design at the strength limit state is not possible given the dearth of necessary data. See commentary to Article 1.3.2.1 for further discussion.

Historically, the design of machinery was done by primarily static analysis. Open spur gear design now follows current AGMA design procedures.

6.4.1.2 GENERAL

The machinery for the movable bridge shall be of simple design and substantial construction. The arrangement of parts shall permit easy installation, adjustment, and replacement of worn or defective parts and shall be accessible for inspection, cleaning, lubricating, and repairing. On any machinery with liquid or grease reservoirs, suitable petcocks and/or tube extensions at drains shall be provided to facilitate fluid changes.

6.4.1.3 LOCATION OF MACHINERY

The location of the machinery shall be selected with consideration for easy access for repair and maintenance, or future removal and replacement. Unless there are compelling reasons to the contrary, the machinery shall be located on the stationary part of the bridge.

6.4.1.4 SUPPORT AND ANCHORAGE

The attachment of machinery to its supports shall be adequate to hold the parts in place under all conditions of service. Where practical, each group of machinery shall be mounted on a self-contained steel frame, base, bedplate, or other sufficiently rigid structural steel support.

The provisions of Articles 6.7.14.1 and 6.7.14.2 shall apply.

C6.4.1.4

A support shall be considered “sufficiently rigid” if its maximum deflection or distortion under allowable overloads does not cause misalignment of any component beyond design or rated misalignment capacity.

SPECIFICATIONS

6.4.2 Aligning and Locking of Movable Span

Movable bridges shall be equipped with span locks or other suitable mechanisms to accurately align the span ends to the approach roadway, both horizontally and vertically, and to secure the movable span in the closed position so that it cannot be displaced either horizontally or vertically under the action of traffic or other conditions of service. Effective end lifting and centering devices shall be used for swing bridges, and for bascule bridges centering devices may be used in conjunction with span locks. Span locks shall also be provided for vertical lift bridges where specified by the Owner.

Span locks shall be designed so that they cannot be engaged unless the movable portions of the span are within ½ in. of the proper position.

Where bascule, swing, or vertical lift bridges are normally, or seasonally left in the open position, span locks shall also be provided to hold the span in the fully open position.

6.4.3 Elevators

An elevator or elevators should be provided on any movable bridge.

When provided, elevators shall meet the requirements for passenger elevators of the ANSI/ASME Safety Code for Elevators and Escalators, ASME A17.3-1996, or latest revision, and applicable local codes.

The elevator cars shall be fully enclosed with solid sides and roof. They shall have a net floor area of not less than 12 ft.^{2} and a capacity of not less than 1,200 lb.

On tower-drive vertical lift bridges, consideration shall be given to providing an elevator in each tower designed to carry personnel and hand carried maintenance equipment between the roadway level, if possible, or the operator's house and the machinery houses at the tops of the towers. Intermediate stops may be provided where specified by the Owner.

Elevators shall be power-operated, with single automatic control permitting the car to be called from a station at any landing and sent to any landing from the car station.

6.5 DESIGN LOADING REQUIREMENTS

The provisions of Section 5 shall apply.

COMMENTARY

C6.4.2

For a double leaf bascule, the two leaves must be aligned vertically within ½ in. relative to each other for the locks to be driven. For a vertical lift span, the ½ in. would again apply to the vertical distance.

Swing bridges must have their centering device properly engaged prior to engaging end lifts. When the end lifts also serve to center the swing span, the ½ in. would apply to the horizontal misalignment measured at the end lifts.

6.6 RESISTANCE OF MACHINERY PARTS

6.6.1 Resistance at the Service Limit State

For commonly used materials, resistance values shall be computed using allowable static stresses in Table 1 which include the factors of safety and unsupported length provisions specified herein as applicable.

Table 6.6.1-1 - Allowable Static Stresses in psi

Material	AASHTO	ASTM	Tension	Compression	Fixed Bearing	Shear
Structural Steel	M 183	A 36	12,000	$12{,}000–55\,L_{eff}/k$	16,000	6,000
Forged Carbon Steel (except keys)	M 102	A 668 (CL D)	15,000	$15{,}000–65\,L_{eff}/k$	18,000	7,500
Forged Carbon Steel (keys)	M 102	A 668 (CL D)	-	-	15,000	7,500
Forged Alloy Steel	M 102	A 668 (CL G)	16,000	$16{,}000–70\,L_{eff}/k$	21,000	8,000
Cast Steel	M 103	A 27 (GR 485-250)	9,000	$10{,}000–45\,L_{eff}/k$	13,000	5,000
Cast Steel	-	A 148 (GR 620-415)	15,000	$15{,}000–65\,L_{eff}/k$	21,000	7,500
Bronze	M 107	B 22 ALLOY UNS C90500, C91100, C91300, C93700	7,000	7,000	-	-
Hot Rolled Steel Bar	M 255	A 675 (GR 515)	12,000	$12{,}000–55\,L_{eff}/k$	16,000	6,000

For materials not included in Table 1, resistance shall be determined by applying the remaining provisions of this article.

The minimum static design resistance at the service limit state shall be determined by applying the following factors of safety, n_S, to minimum tensile yield:

- Forged, drawn, rolled, wrought steel .... $n_S = 3$
- Cast steel .... $n_S = 4$

The static shear resistance shall be based upon one-half the allowable tensile design resistance.

C6.6.1

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For slender members in compression, the allowable static design resistance in compression, p, shall be determined as:

$$p = \frac{A\sigma_y}{n_s}\left(1 - 4.6x10^{-3}\frac{L_{eff}}{k}\right) \quad (6.6.1\text{-}1)$$

where:

A = cross-sectional area (in.^{2})

σ_y = minimum yield strength (psi)

L_{eff} = effective length, depending on end conditions (in.)

k = radius of gyration (in.)

In lieu of a more precise determination, the following approximate effective length may be used:

- Pinned-Pinned Ends Member .... $L_{eff.} = L_{act.}$
- Pinned-Fixed Ends Member............... $L_{eff.} = 0.80\ L_{act.}$
- Fixed-Fixed Ends Member .... $L_{eff.} = 0.65\ L_{act.}$
- Fixed-Free Ends Member .... $L_{eff.} = 2.1\ L_{act.}$

where:

L_{act} = actual length, end to end between supports of the compression member

Fixed bearing resistance shall be determined based upon the minimum static design factor of safety, n_S, specified herein and applied to the minimum specified yield strength of the component material.

- Forged, drawn, rolled, wrought steel ....$n_s = 2.5$
- Cast steel ....$n_s = 3$

6.6.2 Resistance of Components in Bearing at the Service Limit State

6.6.2.1 GENERAL

At the service limit state, the resistance of bearings other than roller bearings shall be designed for the lesser of allowable stresses from:

COMMENTARY

A slender column is one which may buckle at a nominal stress that is below yield.

It is noted that for steel and most other ductile materials, the yield strength in compression is the same as the yield strength in tension. That is, $\sigma_y = \sigma_{yt} = \sigma_{yc}$.

SPECIFICATIONS

- criteria to prevent permanent deformation as specified in Article 6.6.2.3 and 6.6.2.4, or
- criteria to prevent overheating and seizing as specified in Article 6.6.2.5.

6.6.2.2 BEARING ON COMPONENTS NOT SUBJECT TO MOTION

The provisions of Article 6.6.1 shall apply.

6.6.2.3 INTERMITTENT MOTION AND SLOW SPEEDS

For intermittent motion, and where speeds do not exceed 50 fpm, resistance shall be based on the following allowable bearing stresses, applied to the diametral projected area or net surface area for sliding surfaces:

- Pivots of swing bridges, hardened steel on AASHTO M 107 (ASTM B 22 Alloy UNS C91300) bronze disks ....3,000 psi
- Pivots of swing bridges, hardened steel on AASHTO M 107 (ASTM B 22 Alloy UNS C91100) bronze disks ....2,500 psi
- Trunnion bearings and counterweight sheave bearings, rolled or forged steel on AASHTO M 107 (ASTM B 22 Alloy UNS C91100) bronze:

  For loads while in motion....................1,500 psi
  For loads while at rest ....2,000 psi

- Shaft journals, rolled or forged steel on AASHTO M 107 (ASTM B 22 Alloy UNS C93700) bronze ....1,000 psi
- Wedges, cast steel on cast steel or structural steel

  For loads while in motion ....1,500 psi
  For loads while at rest....................2,000 psi

- Acme screws which transmit motion, rolled or forged steel on AASHTO M 107 (ASTM B 22 Alloy UNS C90500) bronze....................1,500 psi

For slowly rotating journals, as on trunnions, counterweight and deflector sheave bearings, and operating drum bearings, the bearing area used to determine resistance shall be taken as the gross projected bearing area, less the effective area of oil grooves.

COMMENTARY

C6.6.2.3

For materials other than those listed here, the maximum journal surface velocity (fpm) should not be exceeded.

Many materials used for plain journal bearings have a maximum allowable journal surface velocities in the range of 10 to 35 fpm. Refer to Bearing Manufacturer's catalogs. See also Table C6.7.7.1.2-1.

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COMMENTARY

For crank pins and similar joints with alternating application and release of stress, the bearing resistance determined as specified above may be doubled.

6.6.2.4 INTERMEDIATE SPEEDS

For intermediate motion at speeds exceeding 50 fpm, bearing resistance shall be determined using the gross projected bearing area and shall be based on the stresses specified below:

- Shaft journals, rolled or forged steel on AASHTO M 107 (ASTM B 22 Alloy UNS C93700) bronze .... 600 psi
- Thrust collars, rolled or forged steel on AASHTO M 107 (ASTM B 22 Alloy UNS C93700) bronze .... 200 psi

The provisions of Article 6.6.2.5 shall not be exceeded.

C6.6.2.4

The maximum journal surface velocity (fpm) is different for various plain bearing materials. Most plain journal bearings are intended for slow speed applications only. Use rolling element bearings for most intermediate and high speeds, based on journal surface velocity.

6.6.2.5 HEATING AND SEIZING

The maximum allowable bearing resistance, R_b (lb.), shall be taken as:

$$R_b = \alpha \left(\frac{A}{nD} \right) \qquad (6.6.2.5\text{-}1)$$

where:

A = planar or projected area over which the load is acting (in.^{2})

n = rotational speed - revolutions per minute (RPM)

D = diameter of journal or step bearing, or mean diameter of collar or screw (in.)

α = a factor specified herein (lb./in.·RPM)

The factor α shall be taken as follows:

- For sleeve bearing shaft, journals rolled or forged steel on bronze 250,000
- For step bearings, hardened steel on bronze 60,000
- For thrust collars, rolled or forged steel on bronze 50,000

C6.6.2.5

The purpose of this provision is to avoid heating and seizing on sleeve bearing shaft journals, step bearings for vertical shafts, thrust collars, and Acme thread power screws.

At high rotational speeds, rolling element bearings are strongly preferred. Use of plain bearings in this type of application can lead to excessive wear and shortened life.

SPECIFICATIONS

- For acme screws, rolled or forged steel on bronze 220,000

6.6.2.6 BEARING ON ROLLERS

The maximum allowable line bearing resistance, R_R per unit width (lb./in.) on rollers used in steel-steel interfaces shall be taken as:

$$R_R = \alpha \left(\frac{\sigma_y - 13{,}000}{20{,}000} \right) \quad (6.6.2.6\text{-}1)$$

where:

σ_y = minimum yield strength of the weaker material (psi)

D = diameter of the roller (in.)

α = factor specified herein (lb./in.)

	Diameters Up to 25 in.	Diameters 25 to 125 in.
For rollers in motion:	400 D	2,000 $\sqrt{D}$
For rollers at rest:	600 D	3,000 $\sqrt{D}$

For manufactured trunnion and counterweight sheave roller bearings, the provisions of Article 6.7.7.2.4 shall apply.

6.6.3 Design for the Fatigue Limit State

6.6.3.1 GENERAL

The loads generated from operation at full load torque or normal operating pressure shall be used unless otherwise specified.

For steel parts other than spur gears subjected to cyclic stresses, when the expected number of load cycles is expected to be more than one million, design shall be based on the endurance limit specified herein.

COMMENTARY

C6.6.2.6

The guidelines in this article are typically used for special steel rollers designed for the following uses: swing span rim bearing rollers, swing span balance wheels (center bearing), end lift rollers or rockers, span guide rollers, counterweight guide rollers, etc.

Equation 1 is based on steel-steel interface only. Use of Equation 1 for other materials would be erroneous.

For steel and most other ductile materials, the yield strength in compression is the same as the yield strength in tension.

C6.6.3.1

From field measurements of several movable bridges, it has been found that the operating machinery constant velocity running loads usually range from 40 percent to 70 percent of full-load torque. Where the movable span is likely to be subject to frequent operation under heavy wind or ice loading, a load factor of up to 1.5 may be applied.

Where mechanical components are subjected to high numbers of cyclic stresses during the expected life of the part, designing statically may not be adequate, and a component may fail by fatigue.

SPECIFICATIONS

COMMENTARY

The endurance limit is referred to as the constant amplitude fatigue threshold in the specifications for fixed bridges.

Nonferrous metal such as aluminum or copper alloys are examples of materials which do not exhibit endurance limits.

6.6.3.2 ENDURANCE LIMIT

For wrought Carbon and Alloy steels, and for Stainless Steels, subjected to cyclic stresses, the endurance limit shall be taken as:

$$\sigma_e = \alpha\,\sigma_{ut}\;(C_D\;C_S\;C_R\;\;C_T\;C_M\;) \qquad (6.6.3.2\text{-}1)$$

where:

σ_{ut} = specified minimum ultimate tensile strength (psi)

α = factor depending on material (DIM) taken as:

- for wrought carbon and alloy steel and for Ferritic stainless steels .... $\alpha = 0.50$
- for cast steels and for Austenitic stainless steels .... $\alpha = 0.40$

C_D = size factor (DIM) based on shaft diameter, D (in.) taken as:

- For $D \leq 0.3$ in., $C_D = 1$ (6.6.3.2-2)
- For $D > 0.3$ in., $C_D = (D / 7.6\,)^{-0.113}$ (6.6.3.2-3)

C_S = surface roughness factor (DIM) taken as:

$$C_S = a\;(\sigma_{ut}\,)^b \qquad (6.6.3.2\text{-}4)$$

C_R = reliability factor (DIM)

- When using minimum specified ultimate tensile strength, $C_R = 1$.
- When using Typical Ultimate Strength properties for the value of σ_{ut} , C_R shall be taken as:

Reliability Percent	C_R
90.0	0.897
99.0	0.814
99.9	0.753

C6.6.3.2

The equation for the endurance limit σ_e is a function of many factors, as shown herein.

For steel mechanical components subjected to fewer than one million, but more than 10,000 cycles of stress during its expected life or for materials that have no true endurance limit, design shall be based on a finite life fatigue resistance.

The value of α specified for Austenitic stainless steel is based upon Deutschman, 1975.

Since the shaft diameter is not initially known in design, estimate D in the 4 to 8 in. range and C_D in the 0.7 to 0.75 range. Alternatively, the diameter may be estimated by doing a static failure analysis, and finding a diameter based on the maximum stress, the yield strength of the material and an appropriate design factor.

For components other than round (shafts), refer to Norton, 1998 for equivalent diameter equations.

These values of C_R are based on an assumed standard deviation in ultimate tensile strength of 8 percent. Typical or average values for the ultimate tensile strength, σ_{ut}, are often given in AISI and ASM tables of properties.

SPECIFICATIONS

C_T = temperature factor, usually taken as 1, except for very high or low temperatures (Juvinall, 1967) (DIM)

C_M = any miscellaneous factors applicable to the details of a particular design (DIM)

Table 6.6.3.2-1 - Variables for Determining C_s (Surface Roughness Factor) (Shigley, 1989)

Condition	a	b
For a ground surface	1.34	-0.085
For a cold finished or smooth machined surface with $R_a \leq$ 32 µin.	2.7	-0.265
For a hot rolled or rough machined with $R_a >$ 32 µin., or as a heat treated surface	14.4	-0.718
For an as cast or as forged surface	39.9	-0.995

R_a = surface roughness factor taken as an arithmetic mean (µin.)

6.6.4 Resistance of Open Spur Gearing Using Allowable Stresses

6.6.4.1 GENERAL

Three criteria shall be satisfied in the design of open spur gears:

- failure of the teeth at the fatigue limit state,
- surface durability through pitting and wear resistance, and
- resistance for overload conditions.

6.6.4.2 SPUR GEAR BENDING RESISTANCE AT THE FATIGUE LIMIT STATE (S_{at})

Resistance shall be based upon the allowable bending stress S_{at} (psi) used in the equations specified in Article 6.7.5.2.2.

When through hardened steel gear teeth having a Brinell hardness between 180 and 400 are subjected to one-way bending during any revolution and use AGMA Grade 1 material, then S_{at} may be taken as:

COMMENTARY

Miscellaneous factors are associated with welding, shot peening, plating, corrosion, or other manufacturing or environmental factors required for the specific design conditions. Shot peening will give a C_M value greater than 1, while the others will give values less than 1.

See National Standard ANSI/ASME B46.1.

C6.6.4.2

See AGMA 2001-C95 or latest revision for the definition of Grade 1 steel.

SPECIFICATIONS

$$S_{at} = 77.3\ H_B + 12{,}800 \qquad (6.6.4.2\text{-}1)$$

where:

H_B = Brinell hardness for the teeth (DIM)

For gears subjected to two-way bending during a rotation, such as idler gears, use 70 percent of this value for S_{at} given above.

COMMENTARY

This could be extrapolated to an H_B value as low as 150.

AGMA defines an idler gear as any gear whose teeth see a stress reversal (two way reverse bending) every revolution. Operating drums on span drive vertical lift bridges may have idler gears, so that each hoisting drum rotates in the desired direction. Also a rack on a bascule span has teeth that see reverse bending every opening and closing cycle.

6.6.4.3 ALLOWABLE SPUR GEAR CONTACT/ DURABILITY/WEAR STRESSES AT THE FATIGUE LIMIT STATE

Resistance shall be based upon the allowable contact stress S_{ac} (psi) used in the equations specified in Article 6.7.5.2.3.

For through hardened steel gear teeth having a Brinell hardness between 180 and 400 and use AGMA Grade 1 material, S_{ac} may be taken as:

$$S_{ac} = 322\ H_B + 29{,}100 \qquad (6.6.4.3\text{-}1)$$

C6.6.4.3

See AGMA 2001-C95 or latest revision for the definition of Grade 1 steel.

This could be extrapolated to an H_B value as low as 150.

6.6.4.4 ALLOWABLE SPUR GEAR YIELD STRESSES FOR INTERMITTENT OVERLOAD

Resistance applicable to the overload limit state shall be as specified in Article 6.7.5.2.4 using the values of S_{ay} specified herein.

For annealed or normalized gear teeth having a Brinell hardness between 150 and 240, S_{ay} shall be taken as:

$$S_{ay} = 2\ H_B^{\ 2} - 300\ H_B + 31{,}000 \qquad (6.6.4.4\text{-}1)$$

For quenched and tempered gear teeth having a Brinell hardness between 180 and 400, S_{ay} shall be taken as:

$$S_{ay} = 482\ H_B - 32{,}800 \qquad (6.6.4.4\text{-}2)$$

C6.6.4.4

These criteria have intended to prevent yielding by bending of teeth under overloads.

6.6.5 Wire Rope Allowable Stresses

Counterweight ropes shall be proportional such that:

- The total tensile stress from axial load and bending does not exceed 22.2 percent of the specified minimum ultimate tensile strength of the rope.
- The tension from the direct load only does not exceed 12.5 percent of the specified minimum ultimate tensile strength of the rope.

C6.6.5

Refer to Article 6.8.3.3.4 for wire rope stress equations.

SPECIFICATIONS

COMMENTARY

For operating ropes, the respective corresponding limits shall be taken as 30 percent and 16.7 percent.

6.7 MECHANICAL MACHINERY DESIGN

6.7.1 General

Machinery shall be designed for loading from:

- the prime mover as specified in Article 5.7,
- holding as specified in Article 5.5, and
- braking as specified in Article 5.7.3.

6.7.2 Requirements for Design with Static Stresses

6.7.2.1 GENERAL

These provisions shall be taken to apply at the service limit state.

Stresses which are cyclic and changing shall not be treated as static unless the total number of stress cycles during the life of the part are less than about 10,000 cycles, or the range of stress is small in comparison to the average stress.

Determination of resistance based on allowable static stresses is specified in Article 6.6.1.

Where a component does not satisfy the criteria above, it shall be designed for the fatigue limit state as specified in Article 6.6.3.

C6.7.2.1

Static stresses are stresses that are uniform, usually caused by uniform loads acting on the stationary component.

6.7.2.2 UNIAXIAL NORMAL STRESS AND SHEAR STRESS

Where components comprised of ductile materials are subjected to static, steady loads, resistance to static uniaxial tension or bending shall satisfy the following:

$$\sigma_{max} \le \frac{\sigma_{yt}}{n_s} \qquad (6.7.2.2\text{-}1)$$

where:

σ_{yt} = specified minimum tensile yield stress (psi)

σ_{max} = maximum normal stress resulting from the applied loads (psi)

n_s = static factor of safety (DIM)

C6.7.2.2

It is not necessary to include stress concentration factors in the stress calculation when the failure mode is yielding.

For the values for n_s refer to Article 6.6.1.

SPECIFICATIONS

COMMENTARY

Where components comprised of ductile materials are subjected to static loads of direct shear or pure torsion, but are not subjected to axial or bending loads, resistance shall satisfy:

$$\tau_{max} \le \left(\frac{\sigma_{yt}}{2n_s} \right) \quad (6.7.2.2\text{-}2)$$

where:

τ_{max} = maximum shear stress resulting from the applied loads (psi)

6.7.2.3 COMBINED STRESSES

Resistance of components subjected to simultaneous loads producing uniaxial normal stress and shear stress shall satisfy:

$$\tau_{max} \le \frac{\sigma_{yt}}{2n_s} \quad (6.7.2.3\text{-}1)$$

in which:

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2} \right)^2 + \tau^2} \quad (6.7.2.3\text{-}2)$$

where:

σ = applied uniaxial normal stress due to loads producing tension, bending, or both (psi)

τ = applied shear stress due to loads producing torsion, direct shear, or both (psi)

C6.7.2.3

The most common case of combined stresses in machinery components is a combination of uniaxial normal, σ, and shear, τ, stresses, where the normal stress is caused usually by bending or axial loads and the shear stresses caused by torsion or direct shear.

For ductile materials and combined stresses, one of the most commonly used theories of failure by yielding is the maximum shear stress theory.

This is the maximum shear stress from Mohr's circle, due to combined stresses.

6.7.3 General Requirements for Design with Fluctuating Stresses at the Fatigue Limit State

6.7.3.1 GENERAL

The fatigue limit state shall be considered where a machinery component is subjected more than about 10,000 cycles of stress during the component's lifetime.

C6.7.3.1

For an indefinite life, and for steel components, it is possible to design parts that will not experience fatigue failure during its lifetime. The design process uses the modified endurance limit, σ_e as defined in Article 6.6.3.2.

SPECIFICATIONS

6.7.3.2 STRESS CONCENTRATION FACTORS - UNIAXIAL NORMAL STRESS AND SHEAR

Stress concentration factors for fluctuating stress conditions for normal stress, K_F, and shear stress, K_{FS}, may be determined as:

$$K_F = 1 + q (K_t - 1) \quad (6.7.3.2\text{-}1)$$

$$K_{FS} = 1 + q (K_{ts} - 1) \quad (6.7.3.2\text{-}2)$$

in which for ductile materials,

$$q = \frac{1}{\left(1 + \frac{\sqrt{a}}{\sqrt{r}}\right)} \quad (6.7.3.2\text{-}3)$$

where:

r = radius of notch or fillet (<u>in.</u>)

$\sqrt{a}$ = Neuber constant corresponding to the minimum specified ultimate tensile stress as specified in Table 1 (<u>in.</u>)$^{0.5}$

Table 6.7.3.2-1 - Value of Neuber Constant

σ_{ut} (psi)	$\sqrt{a}$ (in.)$^{0.5}$
60,000	0.108
90,000	0.070
120,000	0.049
140,000	0.039
180,000	0.024

COMMENTARY

C6.7.3.2

Stress concentrations are probably the most critical criteria to consider when designing to prevent fatigue failure. Stress concentration factors depend on the loading and the shape of the part. The most critical stress concentrations occur at locations of size or shape discontinuities, especially fillets at shoulders on shafts where the diameters change. Other locations of possible stress concentrations include keyways, other grooves, threads, holes, and similar discontinuities.

The values for K_t and K_{ts} come from figures by Peterson, or others (Pilkey, 1997). See representative figures given at the end of this section, Appendix A6.

For ductile materials only, the theoretical stress concentration factors K_t or K_{ts} from the graphs, are modified to fatigue stress concentration factors, K_F or K_{FS}, which are used in the fatigue design equations.

The equations for the fatigue stress concentration factor, K_F for bending or axial stresses and K_{FS} for torsional shear stresses, are dependent on q, the notch or fillet radius sensitivity factor, and K_t or K_{ts}, the theoretical stress concentration factors.

K_t depends on the type of loading, i.e., bending moment, axial force, torsional moment, the shape of the part, i.e., round, flat, the fillet radius size r as related to the smaller section size d, usually r/d, and the ratio of the dimensions at a change in cross-section, usually D/d.

The tables in Appendix A6 provide values of K_F or K_{FS} for analyzing parts with threads and keyways. There is no modification required, using q, since these values are given as fatigue stress concentration factors.

For a fillet or notch radius equal to <u>0.4 in</u>. Equation 3 yields the following values:

Table C6.7.3.2-1 - Value of Notch Sensitivity Factor

σ_{ut} (psi)	q
60,000	0.85
90,000	0.90
120,000	0.93
140,000	0. 94
180,000	0.96

SPECIFICATIONS

6.7.3.3 FATIGUE DESIGN

6.7.3.3.1 Mean and Amplitude Stresses

The provisions of this section require consideration of mean stresses, σ_m and τ_m, and amplitude stresses, σ_a and τ_a, which shall be determined as:

$$\sigma_a = \left(\frac{\sigma_{max} - \sigma_{min}}{2}\right) \quad (6.7.3.3.1\text{-}1)$$

$$\sigma_m = \left(\frac{\sigma_{max} + \sigma_{min}}{2}\right) \quad (6.7.3.3.1\text{-}2)$$

$$\tau_a = \left(\frac{\tau_{max} - \tau_{min}}{2}\right) \quad (6.7.3.3.1\text{-}3)$$

$$\tau_m = \left(\frac{\tau_{max} + \tau_{min}}{2}\right) \quad (6.7.3.3.1\text{-}4)$$

where:

σ_{max} = maximum applied normal stress (psi)

σ_{min} = minimum applied normal stress (psi)

τ_{max} = maximum applied shear stress (psi)

τ_{min} = minimum applied shear stress (psi)

6.7.3.3.2 Fatigue Failure Theory

Components subjected to loads producing, both uniaxial normal stresses and shear stresses, shall satisfy:

$$\frac{\sigma'_a}{\sigma_e} + \frac{\sigma'_m}{\sigma_{yt}} \leq 0.80 \quad (6.7.3.3.2\text{-}1)$$

in which:

$$\sigma'_a = \sqrt{(K_F \sigma_a)^2 + 3(K_{FS} \tau_a)^2} \quad (6.7.3.3.2\text{-}2)$$

$$\sigma'_m = \sqrt{(K_F \sigma_m)^2 + 3(K_{FS} \tau_m)^2} \quad (6.7.3.3.2\text{-}3)$$

COMMENTARY

C6.7.3.3.1

Stresses may vary during a cyclic stress condition from some minimum to some maximum stress, whether normal or shear stresses.

C6.7.3.3.2

The fatigue failure theory presented here will use the Soderberg failure criteria, which is a conservative theory that uses the endurance limit σ_e and the material tensile yield strength (minimum), σ_{yt} .

Another fatigue theory of failure, which is less conservative, known as the nominal mean stress method, uses K_F and K_{FS} only with the alternating, amplitude stresses, and not with the mean stresses.

When dealing with combined cyclic fluctuating stresses, both uniaxial normal (usually bending) and shear (usually torsion), it is necessary to calculate the Von Mises stress, given by Equations 2 and 3, which is an equivalent normal stress that is used in the fatigue design equations.

The terms σ'_a and σ'_m are the amplitude and mean Von Mises stresses, respectively.

With these equations, the fatigue stress concentration factors, K_F and K_{FS} are used both with the amplitude stresses and the mean stresses.

SPECIFICATIONS

where:

σ_a = amplitude normal stress specified in Article 6.7.3.3.1 (psi)

σ_e = endurance limit of a steel shaft specified in Article 6.6.3.2 (psi)

σ_m = mean normal stress specified in Article 6.7.3.3.1 (psi)

τ_a = amplitude shear stress specified in Article 6.7.3.3.1 (psi)

τ_m = mean shear stress specified in Article 6.7.3.3.1 (psi)

K_F = stress concentration factor for fluctuating normal stress specified in Article 6.7.3.2 (DIM)

K_{FS} = stress concentration factor for fluctuating shear stress specified in Article 6.7.3.2 (DIM)

6.7.4 Shafts, Trunnions, Machine Elements Subjected to Cyclic Stresses

6.7.4.1 SHAFT AND TRUNNION DIAMETER

Unless specified otherwise by the Owner, the design of shafts, trunnions, and other machinery parts subjected to more than 1 million cycles of reversed bending moment due to rotation in combination with a steady torsional moment shall satisfy:

$$\frac{32}{\pi d^3}\left[\frac{K_F M_a}{\sigma_e} + \frac{\sqrt{3}K_{FS}T_m}{2\sigma_{yt}}\right] \le 0.8 \qquad (6.7.4.1\text{-}1)$$

where:

K_F = fatigue stress concentration factor (bending) (DIM)

K_{FS} = fatigue stress concentration factor (torsion) (DIM)

M_a = amplitude bending moment (lb.-in.)

T_m = mean (steady) torsional moment (lb.-in.)

σ_e = endurance limit of the steel shaft specified in Article 6.6.3.2 (psi)

σ_{yt} = minimum tensile yield strength of the steel shaft (psi)

The diameter of shafts used for transmitting power for

COMMENTARY

C6.7.4.1

The previous editions of the AASHTO Movable Bridge Specifications were essentially devoid of any reference to the possibility of fatigue failure of shafts, trunnions, or similar machinery parts that are subjected to high numbers of stress cycles during their life, that can lead to failure especially at locations of high stress concentration.

For a shaft or trunnion of multiple diameters, it is necessary to analyze all crucial cross-sections.

For trunnion type bascule bridges, the trunnions experience a single one-way bending cycle for each complete bridge operation. Therefore, Equation 1 may be taken as:

$$\frac{32}{\pi d^3}\left[\frac{K_F M_a}{\sigma_e} + \frac{\sqrt{3}K_{FS}T_m}{2\sigma_{yt}}\right] \le 1 \qquad (C6.7.4.1\text{-}1)$$

SPECIFICATIONS

the operation of the bridge, or for shafts 48 in. or more in length forming part of the operating machinery or bridge lock system, shall be not less than 2.5 in. in diameter.

Shafts and trunnions with a diameter more than 8 in. shall have a hole bored lengthwise through the center. The hole diameter should be about 20 percent of the shaft diameter.

COMMENTARY

The requirement that large shafts and trunnions, i.e., greater than 8 in. diameter, have a hole of about 20 percent of the diameter bored lengthwise through the center of the shaft should not effect the calculation of the required diameter, giving less than 1 percent error.

Historically, these bores have been provided to reduce residual stresses and to remove nonhomogeneous material that may result from the forming process. Additionally, these bores allow for improved heat treating, as well as easier and more accurate field alignment.

6.7.4.2 SHAFT LENGTH AND DEFORMATION

For solid steel shafts supporting their own weight only, the unsupported length of the shaft shall not exceed:

$$L = \underline{80}\,(D^2)^{1/3} \quad (6.7.4.2\text{-}1)$$

where:

L = length of shaft between bearings (in.)

D = diameter of solid shaft (in.)

Where shafts are considered to be subject to misalignment resulting from the deflection of the supporting structure, they shall be made in noncontinuous lengths, and the arrangement should be such that only angular misalignment need be accounted for by the couplings. Each length of shaft should be supported by not more than two bearings.

Shafts shall be proportioned so that the angular twist in degrees per in. of length, under maximum torsional loads, shall not exceed:

- For typical shafts and where the shaft diameter exceeds 7.5 in. .... (0.05/D)
- Where less twist is desirable, as in shafts driving end-lifting devices ....(0.006)

where:

D = solid shaft diameter (in.)

C6.7.4.2

If shafts meet the requirements for couplings, offset misalignment is satisfied by a floating shaft.

SPECIFICATIONS

6.7.4.3 SHAFT CRITICAL SPEED

The maximum speed of a shaft shall not exceed 67 percent of the critical speed of any section of the shaft as specified herein.

For a solid steel floating shaft subjected to only its own mass, the critical speed in RPM shall be determined as:

$$n_c = 4.732 \times 10^6 \frac{D}{L^2} \tag{6.7.4.3-1}$$

where:

D = diameter of the solid shaft (in.)

L = distance between supports, usually flexible gear couplings (in.)

For a simply supported shaft with a concentrated mass at about center span, the critical shaft speed in RPM shall be determined as:

$$n_c = 1.55 \times 10^6 \left(\frac{D^2}{L^2}\right)\sqrt{\frac{L}{W}} \tag{6.7.4.3-2}$$

where:

D = diameter of the solid shaft (in.)

L = distance between supports, usually flexible gear coupling (in.)

W = weight of the concentrated mass (lb.)

6.7.4.4 SHAFTS INTEGRAL WITH PINIONS

Pinions should be forged integral with their shafts where the following conditions are satisfied:

- if the required shaft size is approximately equal to the root diameter of the teeth;

COMMENTARY

C6.7.4.3

Torque transmitted through line shafting should be kept as small as is practical in order to minimize the weight of line shafting and associated bearings, couplings, and supports. This results in higher line shaft speeds which must be verified to be safely under the shaft critical speed.

C6.7.4.4

SPECIFICATIONS

COMMENTARY

- if the minimum gear hub thickness is less than 40 percent of the shaft diameter.

Minimum gear hub thickness for gears with keyways shall be taken as the minimum length between the keyway and root of the teeth.

6.7.5 Design of Open Spur Gearing

6.7.5.1 GENERAL

The equations specified herein apply only to full depth spur gear teeth.

Unless specified otherwise, gear teeth:

- shall be machine cut,
- shall be of the involute type,
- shall have a pressure angle of 20 degrees.

For spur gear pitch diameter tooth speeds over 600 fpm and where quiet operation is desired, an enclosed helical gear speed reducer should be considered.

Unless otherwise specified, all gear teeth shall be cut from solid rims. For open spur gears, the AGMA gear quality shall be Class 7 or higher and the backlash shall be as established by AGMA based on center distance and diametral pitch (Pd).

For full depth spur gear teeth, the addendum shall be the inverse of the diametral pitch (equal to the tooth module), the dedendum shall be 1.250 divided by the diametral pitch (1.157 times the module), and the circular pitch shall be π divided by the diametral pitch (π times the tooth module).

The face width of a spur gear should be not less than 8/Pd, nor more than 14 Pd (not less than 8, nor more than 14, times the tooth module).

The diametral pitch of spur gears shall not be less than:

- for pinions other than motor pinions, transmitting power for moving the span .... 3.14 in.
- for motor pinions .... 4.19 in.
- for main rack teeth .... 2.09 in.

C6.7.5.1

The use of stub teeth is not recommended, however, AGMA does cover unequal addendum tooth systems. The tooth geometry factors, used for both fatigue and surface durability, for unequal addendum tooth systems are different than those presented below, under Articles C6.7.5.2.2 and C6.7.5.2.3. See AGMA 908-B89.

Spur gear design is based on diametral pitch, which is defined by the number of teeth on the gear divided by pitch diameter (in.). For large teeth (Pd<1), circular pitch is commonly used instead of diametral pitch.

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Pinions, including rack pinions and motor pinions, should have not less than 18 teeth.

Use of less than 18 teeth, with a 20 degree pressure angle, may cause undercutting of the teeth, thereby weakening the critical cross-section at the root of the teeth.

If less than 18 teeth is required for the pinion, consider using the AGMA unequal addendum tooth system.

6.7.5.2 AGMA SPUR GEAR DESIGN EQUATIONS

6.7.5.2.1 General

The provisions of this article shall be taken to apply only to the design of open spur gears.

$$W_t = \frac{2T}{d} \qquad (6.7.5.2.1\text{-}1)$$

where:

T = FLT (lb.-in.) applied at gear shaft

d = pinion pitch diameter (in.)

C6.7.5.2.1

The following equations give the relationship between pitch line tangential tooth force, W_t (lb.), the torque applied to the gear shaft, T (lb.-in.), and the power which the gear transmits, P (hp).

$$P = \frac{W_t d n_p}{126{,}000} \qquad (C6.7.5.2.1\text{-}1)$$

where:

n_p = pinion rotational velocity (RPM)

SPECIFICATIONS

COMMENTARY

AGMA presents design/analysis equations ANSI/AGMA 2001-C95) to determine the safe power that can be transmitted by the gear teeth, either safe power based on fatigue strength, $\underline{P_{at}}$ (hp), or safe power based on pitting resistance/wear/ surface durability, $\underline{P_{ac}}$ (hp).

These equations have been modified into equations to find the factored flexural resistance, $\underline{W_{tat}}$, and factored pitting resistance, $\underline{W_{tac}}$.

6.7.5.2.2 Design for the Fatigue Limit State

Compliance with Equation 1 is intended to prevent fatigue failure of the gear teeth.

The following must then be satisfied:

$$W_t \leq W_{tat} \qquad (6.7.5.2.2\text{-}1)$$

The factored flexural resistance, $\underline{W_{tat}}$ (lb.) of the spur gear teeth, based on fatigue, shall be determined as:

$$W_{tat} = \frac{F J_{Sat} Y_N}{P_d K_o K_v K_s K_m K_B S_F K_T K_R} \qquad (6.7.5.2.2\text{-}2)$$

in which:

$$K_v = \left[\frac{A + \sqrt{V_t}}{A}\right]^B \qquad (6.7.5.2.2\text{-}3)$$

$$A = 50 + 56\,(1.0 - B) \qquad (6.7.5.2.2\text{-}4)$$

$$B = 0.25\,(12 - Q_v)^{0.667} \qquad (6.7.5.2.2\text{-}5)$$

$$v_t = \pi n_p \frac{d}{12} \qquad (6.7.5.2.2\text{-}6)$$

where:

K_v = dynamic factor (DIM)

$\underline{F}$ = tooth face width of the spur pinion or gear that is being analyzed/designed (in.)

$\underline{P_d}$ = diametral pitch taken as N_p/d (in.$^{-1}$)

v_t = pitch line velocity (fpm)

$\underline{d}$ = pitch diameter of the pinion (in.)

$\underline{N_p}$ = number of teeth on the pinion

C6.7.5.2.2

Tables C1 and C2 provide values of the geometry factor, $\underline{J}$ for values of 18, 19, 20, and 21 tooth pinions, for 20^{o} full depth, equal addendum teeth only. (Modified from AGMA 908-B89)

For a gear quality of 6 or 7 use Table C2 for $\underline{J}$ and tooth loading at the tip; use the highest single tooth contact Table C1 for $\underline{J}$ only if the gears are of high quality and accurately aligned at assembly.

Note that the factor K_V is now greater than 1. Previous editions of the AGMA Standards used K_V as less than 1.

SPECIFICATIONS

$\underline{S_{at}}$ = allowable bending stress specified in Article 6.6.4.2 (psi)

K_o = overload factor taken as > 1.0 where momentary overloads up to 200 percent exceed 4 in 8 hours, and exceed one second duration (DIM)

H_B = Brinell hardness for the teeth (DIM)

$\underline{N}$ = number of load cycles

$\underline{n_p}$ = pinion (RPM)

Q_V = gear quality number taken as an integer between 7 and 12 (DIM)

K_S = tooth size factor to reflect nonuniformity of tooth material properties, due to large tooth size, gear diameter, and face width taken as >1 (DIM)

$\underline{K_m}$ = load distribution factor taken as $\underline{K_m = 1.21 + 0.0259F}$ for open gearing, adjusted at assembly, with $\underline{F < 28 \text{ in.}}$, and $\underline{F/d < 1}$ (DIM)

K_B = rim thickness factor taken as 1.0 if $m_B = t_R / h_t >$ 1.2 (DIM)

m_B = the backup ratio (DIM)

t_R = rim thickness (in.)

h_t = total tooth height (in.)

S_F = safety factor for bending strength (fatigue) $S_F \geq 1.2$ (DIM)

$\underline{J}$ = geometry factor for bending strength (DIM)

Y_N = life factor for bending resistance taken as (DIM):

- For $H_B \approx 250$ and $10^3 < \underline{N} < 3 \times 10^6$

$$Y_N = 4.9404\ \underline{N}^{-0.1045} \qquad (6.7.5.2.2\text{-}6)$$

- For $\underline{N} > 3\text{x}10^6$ load cycles, regardless of hardness

$$Y_N = 1.6831\ \underline{N}^{-0.0323} \qquad (6.7.5.2.2\text{-}7)$$

COMMENTARY

Refer to the AGMA Standards for a definition of the Gear Quality Number. The accuracy of a gear increases with an increase of the quality number. Therefore, tighter manufacturing tolerances must be met and thus may increase cost. However, the production methods and tooling of many gearing manufacturers is such that the minimum quality number they will produce may be $Q_v = 9$ or higher. In such cases, the designer may have little or no cost savings in specifying a lower quality number.

AGMA gives no further guidance on K_S, however other references recommend using K_S of 1.2 to 1.5 for large tooth size (say $\underline{P_d < 2.5}$). (Norton, 1998; Shigley, 1983)

This is an approximate equation, derived from AGMA Standard 2001-C95. See this or the latest AGMA standard for a more accurate calculation of $\underline{K_m}$.

A good design guideline is to have $m_B > 1.2$.

See Tables C6.7.5.2.2-1 and C6.7.5.2.2-2 for suggested values of $\underline{J}$.

SPECIFICATIONS

K_t = temperature factor taken as 1.0 for gear temperatures less than 250° F (DIM)

K_R = reliability factor (DIM):

- for 99 percent reliability
- 1.25 for 99.9 percent reliability

COMMENTARY

Table C6.7.5.2.2-1 - J Factor for 20°Full Depth Spur Pinion/Gear (P, G)

Equal Addendum Loaded at Highest Point, Single Tooth Contact								
GEAR TEETH	PINION TEETH (N_p)							
	18		19		20		21	
	P	G	P	G	P	G	P	G
18	0.30	0.30	-	-	-	-	-	-
19	0.30	0.30	0.31	0.31	-	-	-	-
20	0.30	0.31	0.31	0.31	0.31	0.31	-	-
21	0.30	0.32	0.31	0.32	0.32	0.32	0.33	0.33
26	0.31	0.34	0.31	0.34	0.32	0.34	0.33	0.35
35	0.31	0.37	0.32	0.37	0.33	0.37	0.34	0.37
55	0.32	0.40	0.32	0.40	0.33	0.40	0.34	0.40
135	0.33	0.43	0.33	0.43	0.34	0.43	0.35	0.43

SPECIFICATIONS | COMMENTARY

Table C6.7.5.2.2-2 - J Factor for 20° Full Depth Spur Pinion/Gear (P, G)

Equal Addendum Loaded at Tooth Tip								
GEAR TEETH	PINION TEETH (N_p)							
	18		19		20		21	
	P	G	P	G	P	G	P	G
18	0.23	0.23	-	-	-	-	-	-
19	0.23	0.23	0.23	0.23	-	-	-	-
20	0.23	0.24	0.23	0.24	0.24	0.24	-	-
21	0.23	0.24	0.23	0.24	0.24	0.24	0.24	0.24
26	0.23	0.25	0.23	0.25	0.24	0.25	0.24	0.25
35	0.23	0.26	0.23	0.26	0.24	0.26	0.24	0.26
55	0.23	0.28	0.23	0.28	0.24	0.28	0.24	0.28
135	0.23	0.29	0.23	0.29	0.24	0.29	0.24	0.29

6.7.5.2.3 Surface Durability and Wear - Design Equations

C6.7.5.2.3

Compliance with Equation 1 is intended to promote surface durability and pitting resistance.

The following must then be satisfied:

$$W_t \le W_{tac} \qquad (6.7.5.2.3\text{-}1)$$

The factored surface durability resistance, F_{taz} (N), of the spur gear teeth based on pitting resistance shall be determined as:

$$W_{tac} = \frac{F d I}{K_o K_v K_s K_m C_F} \left(\frac{S_{ac} Z_N C_H}{C_p S_H K_T K_R} \right)^2 \qquad (6.7.5.2.3\text{-}2)$$

where:

F = tooth face width of the pinion or gear that has the narrowest face width (in.)

S_{ac} = allowable contact stress for the lower Brinell hardness number of the pinion/gear pair as specified in Article 6.6.4.3 (psi)

N = number of load cycles (DIM)

SPECIFICATIONS

S_H = safety factor for pitting resistance, i.e., surface durability taken as >1 (DIM)

$\underline{C_p}$ = elastic coefficient taken as 2,300 for steel pinion-steel gear (psi)$^{0.5}$

Z_N = stress cycle factor for pitting resistance taken as $2.466\ \underline{N}^{-0.056}$ for $10^4 < \underline{N} < 10^{10}$ (DIM)

$\underline{C_H}$ = hardness ratio factor for pitting resistance, taken as 1.0 if $H_{BP}/H_{BG} < 1.2$ (DIM)

$\underline{I}$ = geometry factor for pitting resistance (DIM)

$\underline{C_f}$ = surface condition factor for pitting resistance taken as 1.0 for good tooth surface condition as specified in Article 6.7.8 for tooth surface finish depending on module (DIM)

COMMENTARY

Table C1 gives values of $\underline{I}$ modified from AGMA 908-B89, $\underline{I}$ geometry factor tables - values for 18, 19, 20, and 21 tooth pinions, for 20° full depth, equal addendum teeth only.

Table C6.7.5.2.3-1 - $\underline{I}$ Factors for 20° Full Depth Spur Pinions/Gears

Equal Addendum Factors same for both Pinion and Gear								
GEAR TEETH	PINION TEETH ($\underline{N_p}$)							
	18		19		20		21	
	P	G	P	G	P	G	P	G
18	0.075		-		-		-	
19	0.077		0.076		-		-	
20	0.079		0.078		0.076		-	
21	0.080		0.080		0.078		0.078	
26	0.084		0.084		0.084		0.084	
35	0.091		0.091		0.091		0.091	
55	0.100		0.101		0.102		0.102	
135	0.112		0.114		0.116		0.118	

Usually the hardness of the gear is lower than that of the pinion. H_{BP} and H_{BG} are the Brinell hardness of the pinion and gear, respectively. The H_{BP}/H_{BG} ratio will usually be less than or equal to 1.2. For example, it is common to have the pinion hardness equal to 350 BHN and the gear equal to 300 BHN, for a ratio of 1.17.

For other material combinations, i.e., steel-cast iron or steel-bronze, refer to the AGMA Standards.

SPECIFICATIONS

6.7.5.2.4 Yield Failure at Intermittent Overload

Spur gear teeth shall be investigated for an infrequent overload condition at the overload limit state for which yield failure due to bending might occur.

The following must then be satisfied:

$$W_t\ (\max) \leq W_{max} \quad (6.7.5.2.4\text{-}1)$$

based on the overload condition.

The maximum factored resistance, W_{max} (lb.), based upon yield failure of the gear teeth shall be taken as:

$$W_{max} = \frac{K_y F K_f J S_{ay}}{P_d K_{my}} \quad (6.7.5.2.4\text{-}2)$$

where:

K_y = yield strength factor taken as 0.50 (DIM)

K_f = stress correction factor = 1 (DIM)

S_{ay} = allowable yield stress number specified in Article 6.6.4.4 (psi)

K_{my} = load distribution factor for overload conditions, taken as K_{my} > 1.1 for straddle-mounted gear (DIM)

F = tooth face width of the spur pinion or gear that is being analyzed/designed (in.)

P_d = diametral pitch (in.$^{-1}$)

J = geometry factor for bending strength (DIM)

COMMENTARY

C6.7.5.2.4

The equation given in the AGMA Standards is modified to solve for W_{max} which is the maximum allowable peak tangential tooth load that can be transmitted, based on yielding.

AGMA suggests using K_f = 1, since this is a yield criteria for failure of a ductile material. K_f is defined by AGMA 908-B89 (Equation 5.72).

AGMA gives an equation only for an enclosed drive.

See Tables C6.7.5.2.2-1 and C6.7.5.2.2-2 for suggested values of J.

6.7.6 Enclosed Speed Reducers

SPECIFICATIONS

6.7.6.1 GENERAL

Whenever possible, enclosed speed reducers should be used instead of open gearing.

Enclosed speed reducers shall be specified on the basis of torque at the service limit state at an AGMA service factor of 1.0 and shall resist torque at the overload limit state without exceeding 75 percent of the yield strength of any component.

Enclosed reducer bearings shall be of the rolling element type and shall have a L-10 life of 40,000 hours.

COMMENTARY

C6.7.6.1

It is recommended that all gearing, except final drive gearing, e.g., rack and pinion, be designed using enclosed speed reducers wherever feasible.

SPECIFICATIONS

Gear quality for enclosed reducers shall be AGMA Class 9 or higher, and backlash shall be in accordance with AGMA standards.

Lubrication of the gears and bearings shall be automatic and continuous while the unit is being operated.

Provisions shall be made for filling, draining, and ventilating the housings and a sight gage or dip stick shall be mounted on the unit to facilitate monitoring the lubricant level.

The design of machinery shall accommodate a ±4 percent variation in the reducer exact ratio from the design ratio in the specifications.

6.7.6.2 PARALLEL SPUR, HELICAL, AND BEVEL GEAR REDUCERS

The contract documents shall specify that spur, helical, herringbone, and bevel enclosed gear speed reducers and increasers be manufactured in accordance with the requirements of the applicable AGMA standards and shall carry the AGMA symbol on the nameplate. The nameplate shall be specified to contain the AGMA horsepower rating, the thermal rating, input and output speeds, and the exact ratio.

6.7.6.3 WORM GEAR REDUCERS

Except for the end lifts and center wedges of swing bridges, worm gearing should not be used for transmitting power to move the span. Where used, worm gear reducers should be commercial units which shall be selected on the basis of their rating under AGMA recommended practice.

The contract documents shall specify that commercial worm gear reducers shall provide that, or custom designs shall provide that:

- the worms be heat-treated alloy steel and the worm gear shall be typically phosphor, tin, or manganese alloys of bronze,

- the thread of the worm be ground and polished, and the teeth of the gear shall be accurately cut to the correct profile,

- the worm and gear thrust loads be taken by rolling element bearings, mounted in water and oil-tight housings, and

- the unit shall be mounted in a cast-iron or steel/cast steel housing and the lubrication shall be continuous while in operation.

COMMENTARY

See the provisions of Article 6.10.4.2.

C6.7.6.3

It is recommended that use of worm gearing should be limited to enclosed gear reducers.

SPECIFICATIONS

Worm gear units that are used for end and center lifts or wedges of swing bridges shall be self-locking.

The contract documents shall specify that worm gear speed reducers and worm gear motors be manufactured in accordance with the requirements of applicable AGMA standards and shall carry the AGMA symbol on the nameplate. The nameplate shall be specified to contain the AGMA horsepower rating, the thermal rating, input and output speeds, and the exact ratio.

6.7.6.4 PLANETARY GEAR REDUCERS

The contract documents shall specify planetary gear reducers be manufactured in accordance with the requirements of applicable AGMA standards and shall carry the AGMA symbol on the nameplate. The nameplate shall be specified to contain the AGMA horsepower rating, the thermal rating, input and output speeds, and the exact ratio.

6.7.6.5 CYCLOIDAL SPEED REDUCERS

The contract documents shall specify that the nameplate contain the horsepower rating, the thermal rating, input and output speeds, and the exact ratio.

6.7.6.6 MECHANICAL ACTUATORS

The contract documents shall specify that mechanical actuators using ball screws, with recirculating balls, or using the Acme screw and nut, shall be standard manufactured enclosed units.

The ball screw actuators shall be specified to have a brake to lock the actuator in position.

The Acme screw actuators shall be specified self-locking, depending on the friction and the pitch of the screw.

6.7.7 Bearing Design

6.7.7.1 PLAIN BEARINGS

6.7.7.1.1 General

Bearings shall be placed close to the points of loading and located so that the applied unit bearing pressure will be as nearly uniform as possible.

Large journal bearings shall be of the split type with one half recessed into the other half. The length of a bearing shall be not less than its diameter. The base half of bearings for gear trains and for mating gears and pinions shall be in one piece. The caps of bearings shall be secured to the bases with turned bolts with square heads

COMMENTARY

C6.7.6.5

Cycloidal speed reducers do not use gears to produce a speed reduction, but cycloidal discs and pin rollers.

C6.7.6.6

These units are typically driven by electric motors through use of helical gearing or worm gear drives.

Ball screw actuators are typically nonlocking, because of low friction.

A brake unit is recommended as a precautionary measure. Also, brake units, usually motor brakes, aid in accurate positioning.

C6.7.7.1.1

Large journal pillow block bearings are generally greater than <u>4 in.</u> bore diameter.

recessed into the base or threaded dowels and with double hexagonal nuts. The nuts shall bear on finished bosses or spot-faced seats.

Where it is obvious that aligning and adjustment will be necessary during erection, provisions shall be made for the aligning of bearings by means of shims, and for the adjustment of the caps by means of laminated liners or other effective devices.

Large bearings shall be provided with effective means for cleaning lubrication passages without dismantling parts. Jacking holes shall be provided between machinery bearing caps and bases to facilitate maintenance.

The shaft (journal) should be specified to be at least 100 BHN points harder than the metallic bearing material.

This requirement is specified because of the variability of the hardness of metallic bearing materials.

Thrust loads shall be absorbed by using thrust flanges on the bearing, or by thrust collars or thrust washers.

6.7.7.1.2 Plain Bearing Design Equations

Plain cylindrical bearings, i.e., sleeve bearings, that are boundary lubricated shall be sized based on three main parameters: pressure, surface velocity of journal, determined as indicated below, and the product of the two.

$$p = \frac{F_{ur}}{(DL)} \qquad (6.7.7.1.2\text{-}1)$$

$$V = \frac{p\,D\,n}{12} \qquad (6.7.7.1.2\text{-}2)$$

where:

F_{ur} = applied radial load (lb.)

p = pressure (psi)

V = surface velocity (fpm)

D = diameter of the journal (bearing I.D.) (in.)

L = length of the bearing (in.)

n = journal rotational speed (RPM)

Where better data is not available, the maximum values for p, V and pV for various commonly used bronze bearing alloys may be taken from Table 1.

C6.7.7.1.2

It is common practice to reduce the projected area, D x L, by about 5 percent if grease grooves are present, unless a more accurate projected area is known.

Radial bearing wear is directly related to the product of pV whereas bearing life is indirectly related to pV. Refer to bearing manufacturers as the factors used to determine bearing life vary significantly with material, whether the material is metallic or nonmetallic, the type and method of lubrication, and contamination of the lubricant.

The relationship between D and L is generally that the length, L, should usually be between 100 percent and 150 percent of the diameter, D.

SPECIFICATIONS

Table 6.7.7.1.2-1 - Performance Parameters for Cast Bronze Bearings

UNS ALLOY	p (psi)	V (fpm)	pV (psi·fpm)	COMMON NAME
C 86300	8,000	25	70,000	Mang. Bronze
C 91100	2,500	50	30,000	Phos. Bronze
C 91300	3,000	50	30,000	Phos. Bronze
C 93700	1,000	250	30,000	Tin Bronze
C 95400	3,500	100	50,000	Alum. Bronze

6.7.7.1.3 Lubricated Plain Bearings

Journal bearings should have bronze bushings. For lightly loaded bearings, the bushings may be bronze or a nonmetallic substance as specified herein. For split bearings, the bushing shall be in halves and shall be provided with an effective device to prevent its rotation under load. The force tending to cause rotation shall be taken as 6 percent of the maximum load on the bearing and as acting at the outer circumference of the bushing. A clearance of approximately ¼ in. shall exist between the bushing of the cap and the bushing of the base into which laminated liners shall be placed. The inside longitudinal corners of both halves shall be rounded or chamfered, except for a distance of 0.4 in. from each end or from the shaft shoulder fillet tangent point.

Bushings for solid bearings shall be in one piece and shall be pressed into the bearing bore and effectively held against rotation.

6.7.7.1.4 Self-Lubricating; Low Maintenance Plain Bearings

6.7.7.1.4a Metallic Bearings

The oil-impregnated powdered metal bearings shall comply with the provisions of ASTM Standards B 438, oil-impregnated sintered bronze, B 439, oil impregnated iron-base sintered, and B 783, ferrous powdered metal.

COMMENTARY

C6.7.7.1.4a

The most common of this type of bearing is the oil-impregnated or graphite impregnated copper alloy (bronze) or iron alloy powdered metal bearings.

Caution should be used when specifying a stainless steel shaft (journal) with oil-impregnated bearings. The "300 Series" Austenitic Stainless Steel may not be as satisfactory as a "400 Series" Ferritic Stainless Steel because of the high nickel content in the 300 Series reacting with the normally used oil-impregnated lubricant.

Table C1 lists maximum values for p, V and pV for various commonly used oil-impregnated bearing materials:

SPECIFICATIONS

COMMENTARY

Table C6.7.7.1.4a-1 - Performance Parameters for Oil-Impregnated Metals

MATERIAL	ASTM NO.	p (psi)	V (fpm)	pV (psi·fpm)
Oilite Bronze	B-438-73 Gr 1 Type II	2,000	1,200	50,000
Super Oilite	B-439-70 Gr 4	4,000	225	35,000
Super Oilite - 16	B-426 Gr 4 Type II	8,000	35	75,000

6.7.7.1.4b NonMetallic Bearings

Plastic bearing materials, such as nylons, acetal resins (Delrin), TFE fluorocarbons (Teflon), PTFE, and fiber reinforced variations of these materials may be used where conditions permit.

C6.7.7.1.4b

As a general guide to the important properties of “plastic” bearings and other plastic parts, refer to ASTM D 5592 “Standard Guide for Material Properties Needed in Engineering Design Using Plastics.”

Refer to manufacturers of “plastic” bearings for detailed information on the allowable p, V, and pV values, and any particular design methods. The properties of some nonmetallic bearing materials are given in Table C1.

Table C6.7.7.1.4b-1 - Performance Parameters for Nonmetal Bearings

MATERIAL	p (psi)	V (fpm)	pV (psi·fpm)
Acetal (“Delrin”)	1,000	1,000	3,000
Nylon	1,000	1,000	3,000
Phenolics	6,000	2,500	15,000
TFE (“Teflon”)	500	50	1,000
PTFE Composite	10,000	150	25,000

6.7.7.2 ROLLING ELEMENT BEARINGS

6.7.7.2.1 General

Rolling element bearings shall be designed at the service limit state so that the L-10 life shall be 40,000 hours at the average running speed of the bearing and shall also be designed at the overload limit state and shall satisfy:

$$P \geq F \qquad (6.7.7.2.1\text{-}1)$$

C6.7.7.2.1

The life of rolling element bearings is based on the L-10 life which is defined by the American Bearing Manufacturers Association as the life for which 90 percent of a group of identical bearings will survive under a given equivalent radial load (i.e.,10 percent failures).

SPECIFICATIONS

where:

P = factored radial resistance specified in Articles 6.7.7.2.2 and 6.7.7.2.4 (lb.)

F = factored radial load applied specified in Article 6.7.7.2.2 and 6.7.7.2.4 (lb.)

Where separately mounted in pillow blocks, bearings shall be self-aligning. Housings should be of cast steel and shall be split on the centerline. Seals shall be designed to retain lubricants and to keep dirt or moisture out of the bearing.

Foot-mounted pillow block bases for units with small bore, usually taken to mean under 3 in., should have slotted holes at the mounting feet to permit easy erection, adjustment and replacement. If the mounting feet have slotted holes, the feet shall have machined ends to permit the use of end chocks or the unit shall be doweled in place after installation and alignment.

6.7.7.2.2 Rolling Element Bearing Design

Where a radial roller bearing is intended to have an L-10 life greater than 40,000 hours of life with a minimum reliability of 90 percent, the equivalent dynamic factored radial resistance, P_r (lb.), shall be determined as:

$$P_r = 0.77\,C_r \left(\frac{1}{n}\right)^{0.3} \qquad (6.7.7.2.2\text{-}1)$$

where:

C_r = basic dynamic radial load rating of the bearing (lb.)

n = rotational speed of the bearing inner race (RPM)

For radial roller bearings, the factored dynamic radial load F_{xy} (lb.) may be determined as:

$$F_{xy} = X\,F_{ur} + Y\,F_{ua} \geq F_{ur} \qquad (6.7.7.2.2\text{-}2)$$

where:

F_{ur} = applied radial load (lb.)

F_{ua} = applied axial load (lb.)

X = dynamic radial load factor (DIM)

Y = dynamic thrust load factor (DIM)

COMMENTARY

C6.7.7.2.2

Design of Rolling Element Bearings is covered by American Bearing Manufactures Association (ABMA) standards. Where rolling element bearings are used for trunnion bearings on bascule spans or sheave/counterweight trunnion bearings on vertical lift spans, refer to Articles 6.7.7.2.3 and 6.7.7.2.4)

For the design/sizing of single row roller bearings, the equations given in the standards (ANSI/ABMA Standard 11-1990 or Latest Edition) have been modified into a resistance equation format used in Equation 1.

Nomenclature varies slightly from ABMA standard practice in order to maintain continuity within this document.

The factored dynamic radial load F_{xy} shall not be less than the applied radial load F_{ur}.

Values for X and Y are obtained from manufacturer's catalogues, for the particular size and type of bearing under consideration.

SPECIFICATIONS

Where P_r exceeds 0.5 C_r, the provisions of Article 6.7.7.2.4 shall also apply.

6.7.7.2.3 Roller Bearings for Heavy Loads

Where specified in the contract documents, rolling element bearings shall be used to support the counterweight sheave shafts of vertical lift bridges, fixed trunnions on bascule spans, and similar shafts/trunnions carrying heavy loads.

Each roller bearing shall be of a type, or shall be so mounted, such that the deflection of the trunnion/shaft will produce no overloading of any part of the bearing or housing. The bearing rollers shall:

- be relatively short for their diameter,
- be closely spaced in separator cages,
- run between hardened-steel races mounted in the housing and on the shaft.

The bearing mountings on each shaft shall be such that the trunnion or shaft will be restrained from axial movement by one mounting, and be free to move in the other mounting.

The ratio of length to diameter of any cylindrical roller or roller segment shall not exceed 3.25. For segmented rollers, the ratio of the total length of roller to diameter ratio shall not exceed 6.5.

Cylindrical roller bearings shall be provided with rolling element thrust bearings capable of restraining an axial thrust equal to 15 percent of the total radial load on the shaft or trunnion. Spherical or tapered roller bearings shall be proportioned for a minimum design axial load equal to the greater of the applied axial load or 15 percent of the total radial load on the bearing.

Each roller bearing shall be mounted in an oil- and water-tight steel housing, which shall be provided with means for replenishing the lubricant and arranged for convenient access for thorough cleaning of the operating parts.

The rolling element bearing shall have a means for ease of removal from the trunnion/shaft by hydraulic or other acceptable process.

Rollers and races shall be of bearing quality steel, as specified in ASTM A 295 and A 485 for through hardened steels and ASTM A 534 for carburized steels. The typical hardness level ranges shall be Rockwell C hardness 60 to 65 for the rollers and 58 to 64 for the races. Bearings specified shall be made by a manufacturer of established

COMMENTARY

When the outer race of the bearing rotates and the inner race is stationary, some bearing manufacturers require an extra rotation multiplication factor in addition to X.

C6.7.7.2.3

Heavy loads are generally defined as being greater than 675,000 lb.

SPECIFICATIONS

reputation who has had bearings of comparable size of the same materials and type in successful service for at least ten years.

6.7.7.2.4 Sizing of Large Rolling Element Bearings

For slow rotation large rolling element bearings, or those that do not complete one full revolution during its loading cycle, sizing should be based on C_{or}, the basic static radial load rating of the bearing.

The factored radial design resistance, P_{or} (lb.) for a bearing subjected to static loads and low speeds shall be determined as:

$$P_{or} = \frac{C_{or}}{n_s} \tag{6.7.7.2.4-1}$$

where:

- For counterweight sheave trunnion bearings of vertical lift bridges, the static design factor should be taken as .... $n_S = 5$
- For bascule spans with a fixed trunnion, the static design factor should be taken as..................$n_S = 8.5$

For a combination of the applied radial and axial loads acting on the bearing, the factored bearing load, F_{oxy} (lb.), shall be determined as:

$$F_{oxy} = X_o F_{ur} + Y_o F_{ua} \tag{6.7.7.2.4-2}$$

where:

F_{ur} = applied radial load (lb.)

F_{ua} = applied axial load taken as not less than 15 percent of F_{ur} (lb.)

X_o = a static axial load factor (DIM)

Y_o = a static radial load factor (DIM)

COMMENTARY

C6.7.7.2.4

Large rolling element bearings generally have a bore larger than 4 in., and a rotational speed less than 5 RPM.

It is necessary to work closely with bearing manufacturers on the large rolling element bearings. The specific operating parameters may necessitate special lubricating or other requirements, especially in applications such as a bascule trunnion bearing whose inner race operates at less than one quarter revolution each cycle.

As an example, one manufacturer of large spherical roller bearings, for the 232 Series, uses $X_o = 1$ and $Y_o \approx 2$. (Y_o varies from about 1.75 for a 14 in. to about 2.06 for a 36 in. bore bearing in this series.)

Therefore, if the axial force is 15 percent of the radial force, this gives $F_{oxy} \approx 1.3\ F_{ur}$. For a vertical lift span, the required static radial load rating would be:

$$P_{or} = \frac{C_{or}}{5} \tag{C6.7.7.2.4-1}$$

then:

$$\frac{C_{or}}{5} \geq 1.3F_{ur} \quad \text{or} \quad C_{or} \geq 6.5F_{ur}$$

The values of X_o and Y_o are values that depend on bearing size and type, and manufacturer.

SPECIFICATIONS

6.7.8 Fits and Finishes

The fits and surface finishes for machinery parts, specified in Table 1, are in accordance with ANSI B4.1, Preferred Limits and Fits for Cylindrical Parts, and ASME B46.1, Surface Texture.

Fits other than the preferred fits listed in this ANSI Standard may be used.

Surface finishes are given as the arithmetic average roughness height (R_a) in microinches; if additional limits are required for waviness and lay, they shall be specified.

The fits for cylindrical parts, specified in Table 1, shall also apply to the major dimensions of noncylindrical parts.

Table 6.7.8-1 - Fits and Finishes

PART	FIT	FINISH R_a(μin.)
Machinery base on steel	--	250
Machinery base on masonry	--	500
Shaft journals	RC6	8
Journal bushing	RC6	16
Split bushing in base	LC1	125
Solid bushing in base (to ¼ in. wall)	FN1	63
Solid bushing in base (over ¼ in. wall)	FN2	63
Hubs on shafts(to 2 in. bore)	FN2	32
Hubs on shafts (over 2 in. bore)	FN2	63
Hubs on main trunnions	FN3	63

COMMENTARY

C6.7.8

Roughness height does not define waviness or flatness, which is a separate criteria. A surface may be "smooth", but be wavy or warped, therefore, not a proper flat surface to mount machinery to.

Fits other than those listed in Table 6.7.8-1 may be used at the discretion of the Engineer.

The range of fits given for hubs on main trunnions allows the designer some flexibility to provide the most appropriate fit for the particular detail.

SPECIFICATIONS

COMMENTARY

Turned bolts in finished holes	LC6	63
Sliding bearings	RC6	32
Center discs	--	32
Keys and keyways		
Top and Bottom	LC4	63
Sides	FN2	63
Machinery parts in fixed contact	--	125
Teeth of open spur gears		
Over 3 diametral pitch	--	32
3 to 1.75 diametral pitch	--	63
Under 1.75 diametral pitch	--	125

6.7.9 Hubs, Collars, and Couplings

6.7.9.1 HUBS

If practical, the length of all wheel hubs should be not less than the diameter of the bore, and for gear wheels not less than 1.25 times the width of the teeth. The minimum thickness at any place on the hub, especially at keyways, should not be less than 0.4 of the gross section diameter of the bore.

All hubs shall be provided with keys, splines or mechanical shrink fit assemblies designed to carry the total torque to be transmitted to the shaft.

For bascule trunnion hubs, the provisions of Article 6.8.1.3.1 shall apply for required fit with structural members.

C6.7.9.1

See also the provisions of Article 6.7.4.4 which covers when to make pinions integral with their shafts.

6.7.9.2 COLLARS

Collars shall be provided wherever necessary to prevent the shaft from moving axially. Where the collar is held in position by set screws, there shall be at least two set screws, 120 degrees apart. The set screws shall have cone points, and the shafts shall be counterbored for the set screws. The edges of the holes shall be peened over the set screws after the collars are adjusted. If a shaft or

C6.7.9.2

There are other alternatives to this type of set-screwed collar, e.g., split clamp-type one or two piece collars, shrink disk friction locking hub, shrink disk friction locking assemblies, and keyless friction locking bushings. These types of assemblies usually require no shaft drilling, key slots, or other shaft preparation, which may also cause stress risers in the shaft. Also these assemblies will

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trunnion receives an axial force, there shall be a thrust bearing to prevent axial movement.

6.7.9.3 COUPLINGS

When practical, all couplings used in connection with the machinery should be standard manufactured flexible couplings placed close to the bearings.

Couplings between machinery units should be of the gear type, providing for angular misalignment or for both angular and offset misalignment.

Couplings connecting machinery shafts to electric motor or internal combustion engine shafts shall be flexible couplings, transmitting the torque through metal parts and providing for both misalignment and shock.

The use of shock resistant couplings, with nonmetal torque transmitting components, may be permitted only where the coupling design is such that normal operating torques can be transmitted by the coupling in the event of nonmetallic component failure.

All coupling and shaft fits and finishes shall comply with the provisions of Article 6.7.8 for hubs on shafts.

All couplings should be keyed to the shafts. Keyed couplings shall be fitted to their shafts in the shop, the couplings after manufacture being shipped to the manufacturers of the shafts as necessary to accomplish this result.

Where necessary, couplings utilizing mechanical shrink fit assemblies or some other means of indexing shall be provided to permit infinite and repeatable axial and angular machinery alignment.

6.7.10 Keys and Keyways

6.7.10.1 GENERAL

Keys for securing machinery parts to shafts should be parallel-faced, with square or rectangular cross-section.

All keys shall be fitted into keyways machined into the hub and shaft. Preferably, the keyways in the shaft should have closed ends, which shall be milled to a semi-circle equal to the width of the key. Keyways shall not extend into any bearing or shaft shoulder fillet. Keyways shall have a fillet in the bottom in each corner according to ANSI B17.1.

Keys shall have a width, b (in.), and a height, h (in.), conforming to ANSI Standards for keys. The total length of the key or keys shall be such that the allowable stresses in shear and bearing specified in Article 6.6.1 are not exceeded.

Keys that are not set into the closed-end keyways shall be held by safety set screws, or other effective means; in

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transmit a torque while preventing axial motion. See also the provisions of Article 6.7.12.

C6.7.9.3

Angular alignment couplings are known as single engagement type; angular and offset alignment couplings are known as double engagement type.

This criteria rules out almost all shock resistant couplings with nonmetallic elastomeric flexing elements usually made of neoprene. One type of coupling uses square jaws on each metal hub, with a nonmetallic or soft metal “spider” between the metal jaws of each hub. In the event of a spider failure, the square metal jaws would still transmit the torque, although poorly and with much noise, and with a lot of backlash between the jaws.

C6.7.10.1

Tapered keys may be used to meet special requirements.

A table of ANSI preferred square and rectangular key sizes, and the shaft size range they are to be used on, is given in Table C1.

Bearing stresses are usually the controlling factor when the cross-section of the key is rectangular, $h < b$.

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vertical shafts, collars clamped about the shafts, or similar devices shall be used.

In hubs of spoked wheels, the keyways shall be located in the centers of the spokes.

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Table C6.7.10.1-1 - Key Sizes

DIAMETER (in.)		KEY SIZE (in.) (b x h)
OVER-TO-INCLUDE		
5⁄16	7⁄16	3⁄32 ´ 3⁄32
7⁄16	9⁄16	1⁄8 ´ 1⁄8
9⁄16	7⁄8	3⁄16 ´ 3⁄16
7⁄8	1¼	¼ × ¼
1¼	1⅜	5⁄16 ´ 5⁄16
1⅜	1¾	⅜ × ⅜
1¾	2¼	½ × ½
2¼	2¾	⅝ × ⅝
2¾	3¼	¾ × ¾
3¼	3¾	⅞ × ⅞
3¾	4½	1 × 1
4½	5½	1¼ × 1¼
5½	6½	32 × 18
6½	7½	1¾ × 20
7½	9	2 × 1½
9	11	2½ × 1¾
11	13	3 × 2
13	15	3½ × 2½
15	18	4 × 3
18	22	5 × 3½
22	26	6 × 4
26	30	7 × 5

6.7.10.2 CAPACITY OF KEYS

Keys used to transmit loads generated by the prime mover from a shaft to another component, i.e., gears, couplings, etc. shall have sufficient resistance to develop the full torsional strength of the shaft.

If two keys are required, they shall be placed 120° apart. When using two keys, each key shall be capable of

C6.7.10.2

It is desired to have the shaft fail before the key(s) since signs of distress in the shaft will normally be more visible. Key failure may more easily lead to an uncontrolled situation, but yield failure of the shaft may have a higher tendency to misalign components and jamb in place.

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carrying 60 percent of the full torsional strength of the shaft.

The resistance of keys used to connect components that are minor parts shall be greater than the force effect corresponding to the torsional requirement of the component.

For trunnions and similar parts which are designed chiefly for bending and bearing, the keys shall have sufficient resistance to hold the trunnion from rotating. The force tending to cause rotation shall be as specified in Article 6.8.1.3.1.

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A shaft that is part of the main drive train may also be used to drive an electrical component such as a selsyn or limit switch unit. The key is sized based on the torsional requirement of this component.

Keyslots cause a stress concentration which must be addressed in the fatigue analysis of the shaft.

6.7.11 Splines

Splined connections for securing machinery parts to shafts shall use standard involute splines complying with ANSI B 92.1, Involute Splines, providing either a major diameter fit or a side fit.

Where cut splines are used, the capacity of the splined shaft may be approximated as that of an equivalent shaft of a size equal to the root diameter of the spline. Stress concentration should be considered using a factor analogous to a "sled-runner" keyslot.

C6.7.11

(For guidance on design of metric involute splines, see ANSI Standard B 92.2M which is derived from ISO 4156 Standard. This standard is not a "soft metric" conversion of the inch-based standard, and, therefore, components would not be compatible. See also Machinery's Handbook., 25th ed. Pg 2064.)

See Appendix A6, Table A6.1-2 for K_f values.

6.7.12 Mechanical Shrink / Friction Fit Assemblies

Mechanical shrink fit assemblies for securing machinery parts to shafts shall be a frictional, keyless, shaft/hub locking device. Each assembly shall be furnished as a complete unit from one manufacturer. The assemblies shall provide an adjustable shrink fit between hubs and shaft for infinite axial positioning and rotational indexing without loss of transmissible torque capacity.

C6.7.12

Caution should be used when using a "frictional" device to transmit a torque. Careful sizing of the device and specifying of the correct assembly tightening torque is extremely important.

6.7.13 Motor and Machinery Brake Design

6.7.13.1 GENERAL

For all brakes and spans where practical, the pressure on the rubbing surface of the brake should not exceed 30 psi, and the product of the pressure on the rubbing surface times the velocity of the brake wheel surface in feet per minute (fpm) should not exceed 90,000 psi-fpm.

Brakes shall be provided with adjustable electrical, or preferably, mechanical means for delaying brake setting so that all brakes do not set at the same time, thereby inducing excessively high torques in the machinery.

C6.7.13.1

Velocity is usually taken at the rim surface for drum brakes, or disc speed at the disc-pad radius for disc brakes.

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6.7.13.2 REQUIREMENTS FOR ELECTRICALLY RELEASED MOTOR BRAKES

The brakes for the span-driving motors, designated as motor brakes, shall be fail-safe type disc or shoe (drum) brakes which are held in the set position by springs with such force as to provide the retarding torques specified herein. Disc hubs or brake wheels for the motor brakes shall be mounted on the main motor shaft, or on a rear motor shaft extension.

Brakes shall be designed for intermittent duty. The brakes shall be designed to release when the current is on and to set automatically when the current is cut off. Brakes for the span operation shall be provided with mechanical or electrical escapements, such that the brakes will not be applied simultaneously.

The brakes shall be equipped with a means for adjusting the torque and shall be set in the shop for the specified torque. Each brake shall be provided with a nameplate which shall be specified to state the rated torque range of the brake and the actual torque setting. Shoe-type brakes shall be designed so that it is possible to adjust the brakes or replace the shoe linings without changing the torque settings.

Brakes released by direct-current shall be released by hydraulic power units, thrustor units or shunt-coil solenoids. Shunt coils shall have discharge resistors so as to avoid high transient voltage upon opening of the shunt-coil circuit.

Brakes released by alternating-current shall be released by hydraulic power units, thrustor units, or motor operators. Hydraulic power units and thrustor motors exposed to the atmosphere shall be totally enclosed, nonventilated with special weatherproof insulation and conduit box.

For shoe-type (drum) brakes, the releasing mechanism shall be capable of exerting a force of not less than 130 percent of the force actually required to release the brake when set at the specified torque setting, and at the lowest ambient temperature expected at the site of the bridge.

The brakes for motors other than main drive motors shall be solenoid-released, spring-applied, shoe-type brakes or dry-type disc brakes. Brakes shall have an intermittent rating not less than the full load torque of the motors with which they are used.

All brakes shall be of a construction which ensures uniform wear, and shall be provided with independent adjustments for adjusting lining wear, equalizing clearance between friction surfaces, and adjusting the retarding torque. The brake linings shall be of materials which are not affected by moisture and are a nonasbestos material. The solenoids, hydraulic power units, thrustor units, and motor operators shall be moisture proof. All fittings shall be

COMMENTARY

C6.7.13.2

The term “fail-safe” shall mean that the brake will set when electrical power is discontinued.

The designer will normally be selecting brake units as a standard manufactured item with special modifications. The requirements set forth will affect layout on the contract drawings and will likely need to be included in the contract specifications.

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corrosion resisting. Hydraulic power units and thrustors shall be provided with all-weather oil.

Shoe-type brakes shall be provided with a low force hand release lever permanently attached to the brake mechanism and arranged so that one person can operate the releases easily and rapidly. Means shall be provided for latching the lever in the set and released positions.

Disc-type brakes shall have provisions for hand release and arranged so one person can operate them easily and rapidly and which can be latched in the released position.

When brakes are located outside of the machinery house, they shall be of weatherproof construction or shall be provided with a weatherproof housing. The housing shall be arranged to permit operation of the hand release from outside the housing.

If installed on the moving portion of the span, the brakes shall be designed to operate satisfactorily in any position of the span.

Consideration should be given to specifying that brakes be provided with:

- heating elements to prevent the accumulation of moisture and frost,
- limit switches for control, and
- lights on the operator's control panel to indicate the position of the brakes and their hand release levers.

6.7.13.3 ELECTRICALLY RELEASED MACHINERY BRAKES

The brakes for the span-operating machinery which are designated as machinery brakes shall meet the requirements for the motor brakes specified in Article 6.7.13.2, except as otherwise specified herein.

6.7.13.4 HYDRAULICALLY RELEASED MACHINERY BRAKES

The brakes for the span-operating machinery which are designated as machinery brakes shall meet the requirements for the motor brakes specified in Article 6.7.13.2, except as otherwise specified herein.

6.7.13.5 HAND OR FOOT RELEASED BRAKES

For calculating the strength of the machinery parts under the action of manually-operated brakes, the force applied at the extreme end of a hand lever shall be assumed at 150 lb. and the force applied on a foot pedal shall be assumed at 200 lb. Under this condition, the

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normal allowable unit stresses for any linkage or mechanism may be increased 50 percent.

Hand brakes and foot brakes should be arranged so that the brake is applied by means of a weight or spring and released manually.

6.7.14 Machinery Support Members and Anchorage

6.7.14.1 MACHINERY SUPPORTS

In the design of structural parts subject to loads from machinery or from forces applied for moving or stopping the span, due consideration shall be given to securing adequate stiffness and rigidity and the avoidance of resonance. Beams subject to such stresses should have a depth not less than 1/8 of their span. If shallower beams are used, the section shall be increased so that the deflection will not be greater than if the above limiting depth had not been exceeded. Deflections shall be investigated sufficiently to insure that they will not interfere with proper machinery operation.

6.7.14.2 ANCHORAGE

Anchor bolts or other anchorages that resist uplift shall be designed to provide at least 1½ times the uplift and to support that force at the allowable stress.

6.7.15 Fasteners, Turned Bolts, and Nuts

All bolts for connecting machinery parts to each other or to supporting steelwork should be high-strength bolts conforming to AASHTO M 164 (ASTM A 325), or ASTM A 490.

SAE grade bolts may be used in place of AASHTO or ASTM grades at the discretion of the Engineer. In no case shall less than SAE Grade 5 be used.

Where specified, all turned bolts shall have turned shanks, semi-finished, washer-faced, hexagonal heads, and rolled, cut, or ground threads. The finished shanks shall be $\frac{1}{16}$ in. larger in diameter than the major diameter of the thread. The bolt blank size, usually ⅛ in. larger than the thread size, shall determine the head dimensions.

The dimensions of all bolt heads shall be in accordance with the ANSI heavy hex structural bolt series specified in ANSI B18.2.6, and threads shall be in accordance with the ANSI coarse thread thread series specified in ANSI B1.1.

Turned bolts shall be fitted in reamed holes, to an LC6 fit.

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C6.7.15

The use of ASTM A 325, Type 2, or ASTM A 490, Type 2, low carbon martensitic steel, is not recommended in any applications where the connection is subjected to fluctuating loads or the possibility of impact loadings.

Refer to ASTM F 568 for a description of the ASTM property classes.

For details of SAE fasteners, refer to SAE J1199.

Table C1 lists standard bolt sizes.

Rolled threads are preferred because this threading process provides the lowest stress concentrations in the thread roots.

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The dimensions of all hex nuts shall be in accordance with the ANSI heavy hex nut series specified in ANSI B18.2.6, and threads shall be in accordance with the ANSI coarse thread series specified in ANSI B1.1. Nuts shall conform to the requirements of ASTM A563.

Hardened steel washers shall be in accordance with ASTM F 436.

All bolt heads and nuts shall bear on seats square with the axis of the bolt. On castings, except where recessed, the bearing shall be on finished bosses or spot-faced seats. Bolt heads which are recessed in castings shall be square. All nuts shall be secured by effective locks. If double nuts are used, both nuts shall be of standard thickness.

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Table C6.7.15-1 - Standard Bolt Sizes, Thread Pitch, Tensile Stress Areas

MAJOR DIA. D (in.)	THREADS PER INCH n ($in.^{-1}$)	MINOR DIA. (in.)	TENSILE STRESS AREA A_t ($in.^2$)
⅝	11	0.5135	0.226
¾	10	0.6273	0.334
⅞	9	0.7387	0.462
1	8	0.8466	0.606
1⅛	7	0.9497	0.763
1¼	7	1.0747	0.969
1⅜	6	1.7105	1.155
1½	8	1.2955	1.405

The values listed in Table C2 for proof and tensile loads are from ASTM. For ASTM A 325 bolts, the proof loads are based on a strength of 85,000 psi (½ to 1, incl.) and 74,000 psi (1⅛ to 1½ incl.); the tensile loads are based on a minimum tensile strength of 120,000 psi (½ to 1, incl.) and 105,000 psi (1⅛ to 1½, incl.). For ASTM A 490 bolts, the proof loads are based on a strength of 120,000 psi; the tensile loads are based on a minimum tensile strength of 150,000 psi.

Table C6.7.15-2 - Proof and Tensile Loads

DIA×TPI	A 325		A 490	
	PROOF (lb.)	TENSILE (lb.)	PROOF (lb.)	TENSILE (lb.)
⅝-11	19,200	27,100	27,100	33,900
¾-10	28,400	40,100	40,100	50,100
⅞-9	39,250	55,450	55,450	69,300
1-8	51,500	72,700	72,700	90,900
1⅛-7	56,450	80,100	91,550	114,450
1¼-7	71,700	101,700	116,300	145,350
1⅜-6	85,450	121,300	138,600	173,250
1½-6	104,000	147,500	168,600	210,750

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6.7.16 Miscellaneous Machinery Requirements

6.7.16.1 SAFETY COVERS AND GUARDS

Safety guards for protection of persons shall be specified. The contract documents shall specify that all safety regulations shall be observed.

Safety gear guards shall be provided for all gears in the machinery houses. If gears or sheaves are located where falling objects may foul them, they shall be protected by metal covers that are easily removable.

6.7.16.2 DUST AND WATER PROTECTION COVERS

Dust covers shall be shown in the contract documents wherever necessary to protect the sliding and rotating surfaces and prevent dust from mixing with the lubricant.

Counterweight sheave rims shall be covered to protect them from the weather.

Covers shall be designed such that access is provided for maintenance and inspection.

6.7.16.3 DRAIN HOLES

There shall be drain holes not less than 1 in. in diameter at places where water is likely to collect.

Semicircular holes may be used in sheave or gear web to rim interfaces. The semicircular holes shall be $\frac{9}{16}$ in. minimum radius.

6.7.16.4 DRIP PANS

Drip pans shall be provided where open gearing is used and located so as to prevent excess lubrication from falling on the machinery room floor, or prevent contamination of waterways where appropriate.

6.7.16.5 COMPRESSED AIR DEVICES

Mechanical devices using power transmitted by compressed air may be used for the operation of auxiliary span drives, span locks, and end lifting and centering devices. Operating air pressure shall be no more than 100 psi.

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6.8 BRIDGE TYPE SPECIFIC MECHANICAL MACHINERY DESIGN

6.8.1 Bascule Spans

6.8.1.1 DRIVE MACHINERY

Drive machinery for bascule spans shall normally include drive motor(s), main reducer, output shafts and two pinions driving two racks mounted to the two main girders.

6.8.1.2 RACKS AND PINIONS

6.8.1.2.1 General

Where a multiple rack and pinion drive is used, either there shall be mechanical devices, usually a differential gear reducer, on the bridge to equalize the torques at the main pinions, or another such equalization method.

6.8.1.2.2 Racks

Racks shall be made in segments, the number of which depends on the size of the bridge. If feasible, the number of teeth in each segment should be the same. The rack segments should normally be made of high-strength steel castings, or equivalent strength steel weldments. The teeth shall be machine cut in a fixture using the true pitch radius for curved racks with fixed pitch radius, and the joints between the segments shall not vary in circular pitch from the rest of the teeth. The joint shall be machined accurately and located at the middle of a tooth space. The rack segments shall be machined on all surfaces in contact with the structure, or mounting hardware.

6.8.1.2.3 Pinions

Each pinion shall be supported by two bearings, preferably rolling element self-aligning pillow blocks.

The pinion may be forged integral with its shaft, or be separate and press fitted and keyed to its shaft.

The pinion shall have no fewer than 18 machine cut teeth, and normally be 20 degree full depth involute. The pinions should be made from an alloy steel, heat treated with through-hardened teeth with a minimum hardness of 180 BHN.

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C6.8.1.1

There are exceptions, e.g., machinery to drive one pinion/rack centrally located on the span.

C6.8.1.2.2

The tolerance at the joint is the same as the circular pitch tolerance for the other teeth, as this is covered by the gear quality number.

C6.8.1.2.3

See Article 6.7.4.4 for guidelines on when to make the shaft integral with the pinion.

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6.8.1.3 TRUNNIONS AND BEARINGS

6.8.1.3.1 Trunnions

Trunnions shall be designed to transfer span loads to the trunnion bearings which shall include loads from the span drive machinery during operation. Torsional loading shall be resisted by keys and/or turned bolts. The provisions of Article 6.7.4.1 shall apply to fatigue design.

Trunnion design may include distinct hubs that increase the bearing area of the trunnion girders and have hub flanges that bolt to the girder webs to transfer torsional and axial loads. These trunnion hubs shall fit tightly into structural parts with an ANSI FN3 fit and the fit between the hub and trunnion shall be as specified in Table 6.7.8-1.

Trunnion designs that do not require hubs shall have an ANSI FN3 fit between the trunnion and mating structural parts. Trunnion collars and/or retaining rings shall be used to transmit torsional and axial loads. The trunnion shall have an integrally forged collar for axial positioning.

Transmitters, resolvers or encoders should be geared to the trunnion shafts if suitable for the particular installation. If synchronous position indicators are used, the receivers in the control desk shall be geared to the indicators. The gearing shall be arranged so as to give the greatest practical accuracy in indication. Use of low backlash enclosed gearing should be considered for all position indication systems.

6.8.1.3.2 Trunnion Bearings

Trunnion bearings should be self-aligning spherical plain or rolling element bearings retained in a split steel housing.

The bearing assembly shall be designed to support:

- the dead load and ice load where applicable, live load and impact loads of the bascule span when closed,
- the dead load and wind loads when open, and
- a thrust (axial) load equal to approximately 15 percent of the maximum radial load.

The bearing housings shall be adjustable to proper elevation, alignment and position on the supporting pedestals in the field by the use of full length shims.

The holes through the supporting steel housing for the anchor bolts shall be oversized holes previously drilled in the shop.

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C6.8.1.3.1

When the span is closed, trunnions are subject to dead load, live load, and impact.

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6.8.1.4 BUFFERS

To aid in seating the moving span smoothly, spans may be equipped with:

- specially designed air buffers,
- industrial shock absorbers, or
- a control system capable of performing the smooth seating in a positive manner.

If the buffer is a specially designed air buffer, the following recommendations shall be considered:

- Provide a bore of the cylinder of the air buffer not less than 10 in. and the stroke not less than 24 in.
- Provide three cast iron or PTFE packing rings for each piston.
- Provide each air buffer with a needle valve and a check valve. The system shall be suitable for sustaining short intervals of air pressure of 1,000 psi and a temperature of 400°F.

6.8.1.5 SPAN AND TAIL LOCKS, CENTERING DEVICES

6.8.1.5.1 Locking Devices

Single leaf spans shall either have a locking device at the toe end for each outside girder or truss to force down and hold down the toe end to its seats or live load shoe locks.

Double leaf spans shall be provided with center locks to lock together the toe ends of the spans and tail locks or latches. Center locks shall transfer live load and impact from one leaf to the other.

The locking devices of single leaf bascules and tail locks of double leaf bascules shall resist the greater of:

- any uplift force that may result from live loads, and
- maximum uplift created by the drive machinery at a stalled condition.

6.8.1.5.2 Centering Devices

The bridges shall be equipped with self-centering devices at the toe end. Transverse centering shall be accomplished by a device preferably located on the centerline of the bridge, as near the roadway level as practicable, with a total clearance not to exceed ⅛ in.

6.8.2 Swing Spans

6.8.2.1 DRIVE MACHINERY

Drive machinery for spans shall normally include drive motor(s), main reducer, output shafts, and pinions/gears driving the operating rack. There shall be a minimum of two pinions, diametrically opposite, providing equal torque to rotate the span. Either the main gear reducer shall be of the differential type, or equalization of torque shall be provided by another method acceptable to the Engineer.

Where four pinions are used, they shall be placed in pairs which shall be diametrically opposite.

Motor brakes shall be provided at each motor, and two machinery brakes should be placed on the reducer output shafts if practical.

6.8.2.2 RACKS AND PINIONS

Racks shall be made in sections not less than 72 in. long. The joints in the rack shall be accurately finished and at the center of a tooth space, the space at the joint having the same dimension as the other tooth spacings.

If a cast track is used and loads are light, as in some pivot-bearing bridges, the rack and balance wheel track segments should be cast in one piece. In rim bearing bridges, the rack shall be separate from the track, so that the parts may be easily removed for repairs.

Each main pinion shaft shall be supported in double bearings, which shall be provided with bolted caps to permit easy removal of the pinion shaft, and to provide adjustment for wear. A thrust locking means shall be provided at the top bearing to carry the weight of the pinion and shaft. Means shall be provided for holding the pinion against movement along the shaft. The double bearing shall be proportioned for the maximum pinion load and shall be adequately braced and attached to the rim girder or superstructure.

Sufficient shims shall be provided between the bearing base and the steelwork to accommodate any necessary adjustment in position of the bearing. Where feasible, the bearings should be shipped assembled to the support steelwork with the shims in place.

C6.8.2.2

The design may make use of rolling element bearings, mounted in pillow blocks, to support the vertical pinion shafts. A spherical roller bearing unit will support the thrust load, whereas a cylindrical roller bearing unit may require a separate thrust bearing unit to take the pinion/shaft weight and any misalignment loads.

6.8.2.3 PIVOT BEARING

Pivot bearing bridges shall be designed so that the entire weight of the moving span is carried on the pivot bearing when the bridge is swinging.

Pivot bearings shall consist of disk bearings or rolling element thrust bearings upon which the span rotates together with supporting pedestal.

C6.8.2.3

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Disk pivot bearings shall consist of two disks, one of phosphor bronze and one of hardened steel. The disks shall be so anchored that the sliding will take place only at the surface of contact of the disks.

Rolling element thrust pivot bearings shall be designed to support the bridge weight as a thrust load and any wind loads or other horizontal forces as radial loads.

Where possible, pivots shall be designed so that the bearings may be removed and replaced while the bridge span is closed, without stopping vehicular traffic over the bridge.

Adjustment for height shall be provided.

6.8.2.4 END LIFTS

End wedges, roller lifts, or equivalent end lifting devices shall lift the ends of the bridge an amount sufficient to produce a positive reaction at each end as defined in Article 2.4.2.2.

End lifting machinery shall be proportioned to exert lifting force(s) equal to the greater of:

- the lifting force(s) specified above plus the reaction due to temperature difference specified in Article 2.5.1.2.6, or,

- the lifting force(s) required to raise each end of the span 1 in.

End lifts and supports shall be designed at allowable stresses defined in Articles 6.6.2.3 and 6.6.2.6 for the maximum positive end reaction including impact and temperature differential.

6.8.2.5 CENTER WEDGES

Center wedges and supports shall be designed at allowable bearing stresses defined in Article 6.6.2.3 for the reaction of the maximum vehicular live load and impact.

6.8.2.6 BALANCE WHEELS

Where possible, no fewer than eight wheels, moving on a circular track, should be provided to resist the tilting of the bridge while the bridge is swinging. The maximum overturning moment shall be determined using ice and/or wind loading as defined in Article 5.4.3. The balance wheel clearance with the track shall be adjustable for height, preferably by shims between the superstructure and the seats of the bearings. For short, narrow bridges, four wheels may be used.

Unless there is compelling analytic evidence to the contrary, the full overturning moment shall be resisted by

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The type of rolling element bearing recommended for this application is the spherical roller thrust bearing.

C6.8.2.4

Swings spans with unequal arms will have different end lift forces from one end to the other.

C6.8.2.5

Center wedges shall not lift the bridge, but shall be designed to drive firmly to resist only the vehicular live load and impact.

A different distribution of loads to balance wheels can be justified by a suitable three-dimensional analysis which

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a single balance wheel where there are only four, and shared, 60 percent to 40 percent, by two balance wheels when there are eight or more.

When wheels are not cast integral with their axles, they shall have pressed fits thereon; either the axles should rotate in bronze-bushed bearings or the shaft should be held from rotation at the ends, and bronze-bushed bearings pressed into the balance wheels.

Balance wheel bearings shall be provided with a means for lubrication.

Balance wheels and their bearings shall be designed for twice the allowable stresses at the service limit state.

6.8.2.7 RIM BEARING WHEELS

Rollers of rim bearing or combined rim and center bearing bridges shall be proportioned for:

- the dead load when the bridge is swinging, and
- the dead load plus vehicular live, and impact loads when the bridge is closed.

In computing the load on the rollers, the rim girder shall be considered as distributing the load uniformly over a distance equal to twice the depth of the girder, out-to-out of flanges. This distance shall be symmetrical about the vertical centerline of the concentrated load.

6.8.2.8 TRACKS

Tracks should be specified to be made in sections, preferably not less than 72 in. long. The track shall be deep enough to insure good distribution of the balance wheel or roller loads to the pier for rim bearing bridges, tracks shall not be less than 4 in. deep.

The joints in the track shall be detailed to be staggered. The track shall be anchored to the pier by bolts not less than 1.5 in. in diameter, extending at least 12 in. into the pier cap, and set in mortar or grout. The track of hand-operated, center bearing bridges shall have a sufficient number of anchor bolts so that the mortar or grout in which they are set will not be crushed by the tractive force developed when turning the bridge. When center bearing bridges are operated by mechanical power and a curved rack is attached or integral with the track, the track shall be anchored down by bolts, and the reactive force developed when turning the bridge shall be resisted by lugs extending from the bottom of the track downward into the pier cap and set in cement mortar, grout or concrete.

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includes allowances for installation inaccuracies and wheel and tread irregularities.

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6.8.2.9 CENTERING DEVICES

Bridges shall be equipped with self-centering devices at one or both ends of the swing span. The centering device(s) shall preferably be located on the centerline of the bridge, as near the roadway level as practicable, with a total clearance not to exceed $\frac{5}{16}$ in.

6.8.2.10 SPAN LOCKS

When the centering devices are not designed to hold the span in the closed position, bridges shall be equipped with span locks. If the swing span is left normally in the open position, it shall also be locked in this position. The span locks shall resist the greatest turning moment created by the span operating machinery while stalled.

6.8.3 Vertical Lift Spans

6.8.3.1 SPAN DRIVE VERTICAL LIFTS

6.8.3.1.1 Drive Machinery

The drive machinery for span drives should include drive motor(s), main reducer, output shafts, and pinions/gears driving the operating drums or ring gears. Motor brakes shall be provided at each motor, and two machinery brakes should be placed on the reducer output shafts where practical. The main gear reducer shall not be of the differential type.

6.8.3.1.2 Operating Ropes

The transverse deviation of a rope from a plane through the groove of a drum or sheave at right angles to the axis of the shaft of the drum or sheave shall not exceed 1 in 30, and where practical should not exceed 1 in 40.

There shall be at least two full turns of the rope on the operating drum when the span is in the fully open or closed position and, in addition, the end of the rope shall be rigidly clamped to the drum, the attachment being such as to avoid sharp bends in the wires.

6.8.3.1.3 Operating Drums and Deflector Sheaves

For operating ropes, the diameter of the drums and deflector sheaves shall be not less than 45 times the diameter of the rope, and preferably should not be less than 48 times, except for deflector sheaves with small angles of contact (less than 45 degrees) between rope and sheave. The provisions of Article 6.8.3.3.5 shall apply.

C6.8.3.1.3

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Operating drums shall be press fitted on their shafts, and in addition shall have keys of sufficient resistance to carry the total torque to be transmitted to the shafts.

The shape of the groove on operating drums shall conform as closely as feasible to the rope section. The center-to-center distance of the grooves shall be not less than ⅛ in. more than the diameter of the wire rope.

Deflector sheaves should have the same diameter as the drums. Intermediate deflector sheaves shall be provided as necessary to prevent rubbing of the ropes on other parts of the machinery or the bridge, and to avoid excessive sag of the ropes.

Where operating ropes have small angles of contact with deflector sheaves, the sheaves shall be supported on roller or ball bearings and shall be designed as light as practicable to insure easy turning and minimum rope slippage in starting and stopping.

All deflector sheaves shall have deep grooves to prevent displacement of the ropes.

6.8.3.1.4 Take-Up Assemblies

There shall be take-ups for controlling slack in the operating ropes consisting of turnbuckles or other devices, such as manually-operated take-up reels. The take-up assemblies shall be readily accessible and capable of being operated by one person.

6.8.3.2 TOWER DRIVE VERTICAL LIFTS

6.8.3.2.1 Drive Machinery

The drive machinery for tower drives should include drive motor(s), main reducer, output shafts and pinions driving the sheave ring gears. Motor brakes shall be provided at each motor, and, where feasible, two machinery brakes should be placed on the pinion/output shafts, as close to the pinion as practical.

6.8.3.2.2 Ring Gears and Pinions

Two pinions engaging ring gears on the two counterweight sheaves in each tower should be provided for tower drives.

The counterweight sheave ring gears should be specified to be made in segments. The segments shall be accurately fit together, with the joint at the center of a tooth space, and with no variation in the circular pitch. The segments shall be machined to accurately bolt to the counterweight sheave.

COMMENTARY

Generally sheaves with a groove depth at least equal to the wire rope diameter are considered deep grooved.

C6.8.3.1.4

The take-ups shall be such as to prevent any rotation of the ropes about their axes during adjustment.

Refer to the provisions of Article 5.7.2.1 for the maximum manual effort one person can apply to any manual device.

C6.8.3.2.2

Occasionally, four sheaves may be required per tower. In such cases, four pinions engaging ring gears on each sheave should be provided.

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6.8.3.2.3 Equalizing Devices

On tower drive vertical lift spans operated through pinions engaging ring gears on the counterweight sheaves, devices shall be specified to equalize the forces at the ring gear pinions when two counterweight sheaves and two pinions are used at each corner of the span, i.e., four sheaves per tower. Equalizing devices should not be used between pinions at opposite sides of the span, but adjusting devices shall be provided between such pinions to permit leveling of the span.

6.8.3.3 WIRE ROPES AND SOCKETS

6.8.3.3.1 Diameter of Wire Ropes

The diameter of counterweight ropes shall be not less than 1 in. nor more than 2.5 in. The diameter of operating ropes shall be not less than ¾ in. The actual diameter of a wire rope, taken as the diameter of the circumscribed circle, shall be measured when the rope is unstressed. The amount by which the actual diameter of a rope may differ from the nominal diameter shall be not greater than the tolerances specified in Table 1.

Table 6.8.3.3.1-1 - Rope Diameter Tolerance (in.)

NOMINAL DIA	UNDERSIZE	OVERSIZE
⅝ to ¾	0	1/32
13/16 to 1⅛	0	3/64
1 3/16 to 1½	0	1/16
1 9/16 to 2¼	0	3/32
2 5/16 to 2½	0	⅛

6.8.3.3.2 Construction

Wire ropes shall be specified to be made of improved plow steel (IPS) or extra improved plow steel (EIPS) wire. All operating ropes shall be preformed wire rope.

Wire rope shall be specified to be 6 x 19 class wire rope of 6 x 25 filler wire construction with a hard fiber core. Each strand shall consist of 19 main wires and 6 filler wires fabricated in one operation, with all wires interlocking. There shall be four sizes of wires in each strand; 12 outer wires of one size, 6 filler wires of one size, 6 inner wires of one size and a core wire. The hard fiber core should be polypropylene. Use of independent wire rope core (IWRC) may be permitted.

COMMENTARY

C6.8.3.3.1

Most common nominal diameters of wire rope in the 6 x 19 class, fiber core are in Table 6.8.3.3.6-1.

C6.8.3.3.2

Double extra improved plow steel (EEIPS) grade wire rope is also available in some sizes.

The use of IWRC has not been expressly permitted until this time, however, its use has wide acceptance in general industry and growing use and acceptance in movable railroad bridges.

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6.8.3.3.3 Lay

Unless otherwise specified, all wire ropes shall be right regular lay, and the maximum length of rope lay shall be taken as:

- Operating ropes: 6.75 times nominal rope diameter.
- Counterweight ropes: 7.5 times nominal rope diameter

The lay of the wires in the strands shall be such as to make the wires approximately parallel to the axis of the rope where they come in contact with a circular cylinder circumscribed on the rope.

6.8.3.3.4 Wire Rope Stresses

Where a wire rope is bent over a sheave, the maximum bending stress, σ_b in psi, on the rope may be conservatively determined as:

$$s_b = E_w \frac{d_w}{D} \quad (6.8.3.3.4\text{-}1)$$

where:

d_W = diameter of the outer wires in the wire rope (in.)

D = tread diameter of sheave rope grooves (in.)

E_W = tensile modulus of elasticity of the steel wire = 30×10^6 psi

The maximum total stress in the rope, σ_t in psi, may be determined as:

$$s_t = \frac{P}{A} + s_b + \frac{P_o}{A} \quad (6.8.3.3.4\text{-}2)$$

where:

P = direct load on the ropes (lb.)

A = effective cross-sectional area of the ropes (in.^{2})

P_o = operating loads, e.g., the larger of starting or inertial loads (lb.)

COMMENTARY

C6.8.3.3.4

The maximum bending stress occurs on the outer-most wires at the least bending radius.

For rope of the 6 x 19 class - 6 strands of 19 main wires each, $d_W \approx d / 16$

therefore:

$$s_b = \frac{1.875 \times 10^6}{D/d} \quad (C6.8.3.3.4\text{-}1)$$

where:

d = diameter of the wire rope (in.)

When determining P_o for counterweight ropes, only inertial loads are effective.

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The maximum total stress shall not exceed the allowable tensile stresses specified in Article 6.6.5.

6.8.3.3.5 Short Arc of Contact

When operating ropes have less than a 45 degree arc of contact with a deflector sheave, the minimum sheave diameter shall be at least 26 times the wire rope diameter.

6.8.3.3.6 Wire Rope Tensile Strengths

The minimum tensile breaking resistance, P_{ut} , for the 6x19 class (6x25 FW), fiber core wire ropes, both improved plow steel (IPS) and extra improved plow steel (EIPS), based on WRTB and manufacturers' specifications shall be taken as specified in Table 1.

Table 6.8.3.3.6-1 - Physical Properties of Rope

		P_{ut} (lb.)	
D (in.)	WEIGHT / LENGTH (lb./ft.)	IPS	EIPS
¾	0.095	47,600	52,400
⅞	1.29	64,400	70,800
1	1.68	83,600	92,000
1⅛	2.13	105,200	115,600
1¼	2.63	129,200	142,200
1⅜	3.18	155,400	171,000
1½	3.78	184,000	202,000
1⅝	4.44	214,000	236,000
1¾	5.15	248,000	274,000
1⅞	5.91	282,000	312,000
2	6.72	320,000	352,000
2⅛	7.59	358,000	394,000
2¼	8.51	400,000	440,000
2⅜	9.48	–	488,000
2½	10.5	–	538,000

COMMENTARY

C6.8.3.3.5

Where a rope is in contact with a small deflector sheave over a short arc, taken as 45 degrees or less, the actual radius of curvature of the rope is usually larger than the deflector sheave radius.

C6.8.3.3.6

FW = Filler wire construction, total of 25 wires per strand, 19 main wires, and 6 filler wires.

WRTB Wire Rope Users Manual, 1993.

EIPS with fiber core is not listed in WRTB manual.

Values are based on a 9 percent increase in tensile strength over the IPS fiber core.

Table 1 is for bright, uncoated wire rope. For galvanized wire rope, refer to specific manufacturer's specifications. Galvanized wire rope typically has about a 10 percent lower breaking strength than the above values.

To find the ultimate tensile strength, σ_{ut} in psi, divide the values of the tensile breaking resistance, P_{ut} in lb., in the table by the area of the wire rope in in.^{2}:

$$s_{ut} = \frac{P_{ut}}{A} \qquad \text{(C6.8.3.3.6-1)}$$

in which:

$A = 0.417\ d^2$

for 6 x 25 FW wire rope with a fiber core.

The elongation of wire rope under load may be determined using the following E_R values :

E_R (psi)	Percent of Ultimate Load
10.8×10^6 (74 500)	0-20
12×10^6 (83 000)	21-65

and the equation:

$$d = \frac{PL}{AE_R} \qquad \text{(C6.8.3.3.6-2)}$$

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COMMENTARY

The contract documents shall require testing to destruction of all wire ropes to verify minimum tensile breaking resistances.

6.8.3.3.7 Wire Rope Sockets

All sockets used with wire ropes, except those for 2½ in. diameter ropes, shall be specified to conform to the requirements of Federal Specification RR-S-550, latest revision. Sockets for 2½ in. diameter ropes may be cast steel conforming to ASTM A148 (A 148M), Grade 80-50.

Sockets shall be specified to be attached to the ropes by using zinc of a quality not less than that defined for high grade in the current specifications for slab zinc (Spelter) of AASHTO M 120, and using a method that will not permit the rope to slip more than 16.7 percent of the nominal diameter of the rope when stressed to 80 percent of its specified ultimate strength. The contract documents shall specify that if a greater movement should occur, the method of attachment shall be changed until a satisfactory one is found.

The sockets shall be stronger than the rope with which they are used. The contract documents shall require testing of sockets and shall specify that:

- If a socket should break during testing, two others shall be selected and attached to another piece of rope and the test repeated, and this process shall be continued until the inspector is satisfied with their reliability, in which case the lot shall be accepted, and

- If 10 percent or more of all the sockets tested break at a load less than the specified minimum ultimate strength, the entire lot shall be rejected, and new ones, of greater resistance, shall be furnished.

Pin and socket fits different from those specified by the Federal Specification may be specified by the Engineer.

Sockets shall be specified to be painted in the shop as specified for structural steel.

6.8.3.4 SHEAVES

6.8.3.4.1 General

Sheaves shall be designed based on internal stresses determined using stress analysis methods, including finite element analysis (FEA) method whenever possible. Sheaves shall be designed so that deflection of the rim under action of the ropes is within allowable tolerances for the pitch or tread diameter.

Sheaves fabricated by welding shall be specified to be made of structural steel, AASHTO M 183 (M 183M),

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M 222 (M 222M), or M 223 (M 223M) or forged carbon steel, AASHTO M 102, Class D (ASTM A 668, Class D, S4).

The resistance of the sheave shall be such that the stresses under dead load with the sheave rotating and with impact do not exceed:

- tension or compression in base metal or filler metal: 10,000 psi with stress range not to exceed 10,000 psi,
- shear in base metal or filler metal: 5,000 psi with stress range not to exceed 5,000 psi,

The contract documents shall specify that:

- the rim be fabricated from not more than three pieces of plate and stiffened by transverse ribs if necessary to carry the load,
- the rim be welded into a complete ring and the welds ground flush on all four sides before being welded into the sheave assembly,
- each web be fabricated from not more than two pieces of plate,
- web welds, if used, shall be ground flush on both sides,
- the hub shall be made from a one-piece forging,
- all welds shall be full penetration welds made with low hydrogen procedures,
- automatic submerged-arc welding shall be used to the greatest extent practicable,
- after completion of the weldment and before final machining, the sheave be stress relieved, and
- unless otherwise specified, the sheave assembly shall be stress relieved by heat treatment prior to final machining.

The contract documents shall specify that:

- the grooves be accurately machined to insure uniformity of the pitch diameter for all of the grooves, and
- the pitch diameter variation shall not exceed plus or minus 0.01 in.

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Operating rope drums shall conform to all general requirements of this article.

6.8.3.4.2 Counterweight Sheaves

For main counterweight ropes, the pitch diameter of the counterweight sheave, center-to-center of ropes, shall be not less than 72 times the diameter of the rope, and preferably not less than 80 times. For auxiliary counterweight ropes, the pitch diameter of the sheave shall be not less than 60 times the diameter of the rope.

The shape of the grooves shall be detailed to conform as closely as feasible to the rope section so that the ropes run freely in the grooves. The sides of the grooves shall prevent the ropes from flattening under static loads. The distance center-to-center of grooves shall be at least 6.5 mm more than the diameter of the rope.

6.8.3.4.3 Sheave Trunnions and Bearings

Counterweight sheaves shall be detailed to be shrink fitted on their trunnions, and then secured by driving-fit dowels set in holes drilled into the sheave hub and the trunnion.

Counterweight sheave bearings shall be designed so that they can be aligned in the field at proper elevation, alignment and position on the supporting steel parts by the use of full length shims, with due allowance for movements of the bearings. The holes through the supporting steel parts for the connecting bolts shall be drilled through the holes in the bearings, which are previously drilled in the shop.

6.8.3.5 COUNTERWEIGHTS AND ROPE ANCHORAGES

6.8.3.5.1 Counterweights

The balance-block pockets shall be detailed in the contract documents and shall be placed as near the ends of the counterweights as practical.

6.8.3.5.2 Counterweight Rope Anchorages

The connections of the counterweight ropes to the lift span and counterweights shall be detailed in the contract documents to permit replacement of any one rope without disturbing the other ropes. Provision shall also be made for replacement of all the ropes simultaneously.

On the lift span side, the counterweight ropes shall be detailed to be separated sufficiently to prevent wind induced slapping of the ropes against each other while the span is in the closed position.

COMMENTARY

C6.8.3.4.3

The dead load to be placed on the bearings may result in structural deflection which must be accounted for in design and alignment requirements.

The use of tapered full-length shims may be required to achieve a horizontal trunnion installation and maintain full bearing.

C6.8.3.5.1

This provision is required in order to aid in securing the required balance between the lift span and the counterweights at each of the four corners of the span.

C6.8.3.5.2

Replacement of all ropes simultaneously will usually require details for supporting the counterweights from the towers.

This may be accomplished either by use of widely spaced grooves on the sheaves, or by using deviations of the ropes from a vertical plane.

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The transverse deviation of a counterweight rope from a vertical plane through the center of the groove on the sheaves should not exceed one-half the spacing of the grooves, and shall be the same for all the ropes on a sheave. In no case shall transverse deviation of slope exceed 1 in 40. The longitudinal deviation of a counterweight rope leading from the sheave, measured from a vertical plane tangent to the pitch diameter of the sheave, shall not exceed 1 in 30, and shall be the same for all the ropes on a sheave. These deviations shall not be exceeded on the span side for the lift span in its highest possible position, and on the counterweight side for the span in the closed position.

The connections of all ropes shall be made in such manner as to prevent rope rotation and give equal loads on several ropes of a group.

The connections of all ropes shall be so made that the centerline of the rope above the socket is at all times at right angles to the axis of the socket pin for pin sockets and to the bearing face of the socket for block sockets. Rope deflector castings or equivalent devices shall be provided near the sockets, as necessary, to accomplish this.

6.8.3.5.3 Clearance Below Counterweights

The counterweights should be detailed to clear the roadway by not less than 60 in. when the span is fully open. In computing this clearance, the counterweight ropes shall be assumed to stretch 1 percent of their length.

6.8.3.6 BUFFERS

The contract documents should require provisions of either:

- air buffers or industrial shock absorbers to aid in seating the moving span smoothly, or
- a control system capable of performing the smooth seating in a positive manner.

Vertical lift spans should also be equipped with air buffers, industrial shock absorbers, or other type of bumper to aid in stopping the movable span at maximum lift height, without damage to the structure.

The provisions of Article 6.8.1.4 shall apply to the design of a custom air buffer.

COMMENTARY

This may be accomplished either by adjustment of the tension in the ropes using jacks and shims, turnbuckles, or by use of equalizers.

C6.8.3.5.3

The 1 percent stretch shall be applied in addition to calculated elastic stretch of the wire ropes from the dead load.

C6.8.3.6

The purpose of this provision is to prevent damage in the event that the span is accidentally moved beyond the prescribed limits of lift.

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6.8.3.7 SPAN LOCKS AND CENTERING DEVICES

6.8.3.7.1 Locking Devices

Bridge design should include locking devices at each end of the lift span to prevent the span from rising after it has been seated by the operating machinery. The locking device may be self-locking, i.e., spring loaded, with power release, or a power driven/release locking bar.

Locking devices shall be designed at two times the overload limit state to resist the maximum load produced by the prime mover at a stalled condition.

6.8.3.7.2 Centering Devices

Bridges shall be equipped with self-centering devices at each end. Transverse centering shall be accomplished by devices located on the center line of bridge, as near the roadway level as practicable, with a total clearance not to exceed ⅛ in. For truss bridges these centering devices shall be supplemented by close transverse centering of the unloaded chords, to be accomplished by special centering devices or by the span guides.

6.8.3.8 SPAN AND COUNTERWEIGHT GUIDES

The span and its counterweights shall be detailed so that they are held in position transversely and longitudinally during their movement by means of guides engaging guide flanges on the towers. Truss spans shall have transverse guides at both top and bottom chord. The guides may be of either the sliding or the rolling type. The ends of guide flanges shall be planed smooth. The guides shall be adjustable, and shall be set to provide a normal running clearance of ⅜ in., except for the transverse span guides for the seated position of the span where the clearance shall not exceed ⅛ in.

COMMENTARY

C6.8.3.8

Roller guides are strongly recommended to reduce friction and wear.

6.9 EMERGENCY DRIVES

6.9.1 Engines - For Driving Generators, Hydraulic Power Units, and for Span Drive

These requirements apply to separately mounted engines and to engines forming part of an engine-generator set or an engine driven hydraulic power unit; the provisions of Article 8.3.9 apply to electric generators.

For determining the required engine size, allowable prime mover torque overloads shall be taken as specified in Article 5.4.1.

C6.9.1

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COMMENTARY

The rated engine torque, as referred to above, shall be taken to mean the lesser of:

- the rated torque of the engine at the speed to be used for operation, measured at the flywheel, with all metal housing in place, and with radiator, fan, and all other power-consuming accessories in place, or
- 85 percent of the rated torque of the stripped engine.

Engines shall be of the industrial duty truck or marine type. The engines should be diesel and operate at a speed of not more than 1,800 rpm and preferably not more than 1,400 rpm, and shall be equipped with a governor to limit the maximum speed to the designated value. Unless otherwise specified, the engine shall have not less than 4 cylinders. If the engine is used for span operation, the contract documents shall specify that the engine be tested by the manufacturer prior to shipment to demonstrate that it will develop the rated torque specified herein.

The contract documents shall specify that the engine shall be equipped with reversing gears, preferably of the helical type, and preferably in a separate gear unit, with a gear ratio of not less than 2 to 1. Reversing shall be by means of approved friction clutch or clutches on the countershaft operated by a lever or other approved device. The arrangement shall be such that the machinery may be operated in either direction without stopping the engine. The reversing-gear unit shall satisfy the requirements given in Article 6.7.6.

All engines having a rating of 20 hp or more shall be equipped with an electric starter with generator and storage battery. Where electric current is available at the bridge, a battery charging unit shall also be provided. All engines having a rating of 60 hp or less shall also be provided with a hand cranking device, if feasible.

Provision shall be made for effectively cooling the engine. Smaller engines may be air cooled, whereas larger engines are typically liquid cooled. There shall also be provided a corrosion resisting metallic exhaust pipe and muffler discharging outside the engine room. Air inlets, including louvers, shall be arranged to assure an adequate air supply to the engines at all times.

The fuel tank shall be located outside the engine room, below the level of the intake. The tank shall be made of corrosion-resisting metal and shall be large enough to hold fuel for 30 days of normal operation. It shall be protected from the sun. It shall be equipped with an automatic gage to show the quantity of fuel in the tank. The fuel pipe and fittings connecting the fuel tank and the engine shall be of copper or brass, so arranged and supported as to effectively provide for temperature and vibration movements tending to produce fracture and leakage at

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connections. Protective fill and vent seal units shall be included to prevent accidental vapor ignition. A day-tank, including pumps, shall be provided for engines over 60 hp. The installation shall be in accordance with the requirements of the National Fire Protection Association.

A small control board containing throttle and choke controls, ignition switch, starter button, and oil and temperature gages shall be provided at the engine.

The engine shall be enclosed in readily removable metal housing unless located in a protected space, and together with reversing gears and all other engine accessories, shall be mounted in the shop on a rigid steel frame so as to form a complete engine unit ready for installation.

Indicators shall be provided in the engine room to show the position of the moving span and, if so specified, of the lifting and locking apparatus.

If cold ambient temperatures may affect starting reliability, a water jacket heater or other suitable means to warm the fuel shall be provided. Protective features shall include low oil pressure cut-out, high water temperature cut-out, engine overspeed shutdown, and overcranking protection if applicable.

The contract documents shall specify the type and quantity of spare parts for engines to be furnished.

On all bridges operated by engines, means shall be provided for interlocking the span movement with operation of the locks and wedges so that power cannot be applied to the span until locks or wedges are released. For swing spans, such interlocking between span and lock mechanisms can generally be accomplished by means of mechanical trips which will allow the gears to be engaged only in proper sequence.

When engines for span or lock operation are used in conjunction with electrically operated lights, gates or other safety devices, interlocking shall be provided which will not permit the locks to be retracted until the safety devices are in operation, nor permit the safety devices to go out of operation until the span is seated and the locks reseated.

Means shall be provided for by-passing the interlocking system in an emergency.

6.9.2 Manual Operation

6.9.2.1 GENERAL

Provision for manual operation in case of power failure shall be provided on all brakes, locks, gates and barriers. Interlock with the span mechanisms shall be provided to prevent bridge openings before gate closure.

COMMENTARY

Such interlocking can generally be accomplished by magnetically actuated trips and relays which will allow the switches and gear to be engaged only in proper sequence. Interlocking by connection to the ignition system of the engine will generally be unsatisfactory.

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6.9.2.2 HAND OR FOOT POWER

On any machinery component having a manual, i.e., hand or foot, override or emergency operation arrangement, the maximum forces to cause the action shall be no more than the values listed in Articles 5.7.2.1 and 6.7.13.5.

6.9.2.3 HAND OR FOOT BRAKES

Brakes for bridges operated by power other than electricity shall be operated manually, or where so specified, they shall be electrically operated from an auxiliary electric generator.

The provisions of Article 6.7.13.5 shall apply as applicable.

6.9.2.4 MANUAL OPERATION OF SPAN LOCKS AND LIFTS

Gear motors and other electrical actuators should be provided with an extension of the high-speed shaft to allow emergency hand operation of the mechanism, by attaching a wheel or crank. Interlocking shall be provided such that electrical operation is disabled during hand operation.

If the actuators are hydraulic, driven by an electric HPU, emergency operation shall be available, either with a hand operated pump in small installations, or emergency diesel engine driven HPU.

6.10 LUBRICATION

6.10.1 General

Provision shall be made for effective lubrication of all sliding surfaces, gearing, and of roller and ball bearings. Lubricating devices shall be easily accessible.

The contract documents shall specify that two hand-operated grease guns shall be provided for each type of grease used to service all lubrication fittings, and that all necessary adapters shall be provided for the equipment.

6.10.2 Lubrication Fittings

Lubrication fittings should be of the giant button head pressure type, with built in check valve.

If feasible, all lube fittings should be standardized to one size and type, for ease of maintenance.

COMMENTARY

C6.10.1

Consideration should be given by the designer to verification of lubrication when designing the lubrication provisions. Maintenance experience exists demonstrating that even though lubrication is being faithfully attempted, lubrication lines and grooves blocked by hardened grease sometimes create lubrication starvation, unbeknown to maintenance personnel. A significant amount of premature bearing and trunnion wear, including scoring and galling, has been attributed to this type of lubrication failure.

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6.10.3 Lubrication of Bearings

6.10.3.1 PLAIN JOURNAL BEARINGS

The contract documents shall specify that each sliding bearing requiring lubrication shall have a high-pressure grease fitting, containing a small receiving ball or cone check valve, made of steel, that will receive the grease and close against back pressure. These fittings shall be connected to the bushings of bearings by means of corrosion resisting pipe, which shall be screwed into the bushing through a hole in the cap. If the bearings are not readily accessible, the fittings shall be placed where they will be accessible, and shall be connected to the bearings by means of corrosion resisting pipe.

Grease ducts shall be so located that the lubricant will tend to flow, by gravity, toward the bearing surface. Grooves shall be provided, wherever necessary, for proper distribution of the lubricant.

The grooves for plain trunnion bearings shall be cut in the bushing. Such grooves shall be straight, parallel to the axis of the shaft, and for large bearings no fewer than three; they shall be so located that the entire bearing surface will be swept by lubricant in the lesser of:

- one movement of opening or closing the bridge, or
- 90 degrees rotation of the shaft.

Each such groove shall be served with lubricant by a separate pressure fitting. The grooves shall be of such size that an $^5/_{16}$ .in. diameter wire will lie wholly within the groove; their bottoms shall be rounded to a ¼ in. radius. The grooves shall be accessible for cleaning with a wire.

The grooves for the counterweight sheave bearings may either be in accordance with the requirements of the foregoing paragraph, or they may be spiral grooves cut in the bushing and served with pressure fittings. A clean-out hole shall be provided in the bearing base and connected to the lowest point of the spiral grooves so that the journal surface can be cleaned and the grooves flushed out. Lubricants and greases used for journal bearings shall be of a composition which will not solidify in lubrication passages during the bridge life.

In disk bearings, grooves emanating at the center and extending to the outer edge shall be specified in the upper of the two rubbing surfaces in contact. The grooves shall not be less than ¼ in. wide and deep, and the corners shall be rounded to a radius not less than half the width of the groove. The corners at the bottom of the grooves shall be filleted so there shall be no sharp corners.

Small bearings with bearing pressures less than 1,000 psi and slow or intermittent motion, and not readily accessible,

SPECIFICATIONS

may be self-lubricating bushings. Bushings shall be of a character which will not be injured by the application of oil, and the bearings shall be provided with oil holes for emergency lubrication, and oil holes shall be fitted with readily removable screw plugs.

6.10.3.2 ROLLING ELEMENT BEARINGS

Each roller bearing shall be specified to be mounted in a steel housing, which shall be provided with means for replenishing the lubricant and arranged for convenient access for thorough cleaning of the operating parts. The housing shall contain a sealing system that will prevent water and contaminant from entering while retaining the grease or oil lubricant.

6.10.4 Lubrication of Gears

6.10.4.1 OPEN SPUR GEARING

To insure that open gearing receive proper lubrication, the contract documents shall specify that the gear teeth first be clean and properly aligned for the required distribution of the lubricant on the surface of the gear teeth.

Intermittent methods of lubrication shall be permitted to open gears having a pitch line velocity of less than about 1,000 fpm.

6.10.4.2 ENCLOSED GEARING

Lubrication of the gears and bearings in enclosed gearboxes shall be automatic and continuous while in operation. Provisions shall be made for filling, draining and ventilating the housings and a sight gage or dip stick shall be mounted on the unit to read the lubricant level.

For worm gear enclosed gear boxes, the anti-friction bearings shall be mounted in water and oil-tight housings. The unit shall be mounted in a cast-iron or steel housing and the lubrication shall be continuous while in operation.

COMMENTARY

C6.10.4.1

Viscosity of the gear lubricant varies with temperature, therefore it is necessary to give the selection of the lubricants careful attention, especially in regions where temperature changes are considerable between winter and summer.

For most open gearing applications, an extreme pressure (EP) lubricant designed for open gearing is recommended. These lubricants have the necessary properties to carry heavy tooth loads, while giving the necessary lubrication to the teeth surfaces.

Heavy grades of EP lubricants have the necessary properties to allow for intermittent application of the lubricant. Many of these grades contain an asphaltic base to provide extra film adhesion to the tooth surface. The lubricant may usually be hand brushed onto the gear teeth.

C6.10.4.2

Automatic and continuous lubrication refers to the reducer gears being lubricated during every operation of the movable span. The gears may be lubricated by use of a mechanically or electrically driven pressurized pump, or they may be splash lubricated by use of slingers, or the reducer may be filled with sufficient lubricating oil to bath the gears.

For reducers subjected to long periods of inactivity, care must be taken so that the exposed portions of the gears do not suffer from rusting and corrosion.

Some reducers are filled with oil so that the gears are completely covered, thereby offering protection from rusting. However, the efficiency of this reducer may suffer due to power loss from oil churning.

SPECIFICATIONS

6.10.5 Lubrication of Couplings and Miscellaneous Mechanical Components

The contract documents shall specify that couplings be re-lubricated at approximately six month intervals, unless subjected to excessive misalignment, shock loads, sudden reversals, axial movement, or extreme variations in temperature or humidity.

6.10.6 Lubrication of Wire Ropes

The contract documents shall require that during fabrication, each wire shall be coated with a rust-inhibiting lubricant as it is laid in the strand, and preferably receive an additional application of lubricant during the rope closing operation. In ropes with a fiber core, the fibers shall be prelubricated before the rope closing operation.

6.10.7 Lubrication of Wedges and Strike Plates

Wedge plates, strike plates and other sliding flat surfaces should be detailed to have the capability of being lubricated while the span is closed, through the use of lube fittings and grease grooves to distribute and retain the lubricant.

COMMENTARY

C6.10.5

Gear or grid couplings require adequate lubrication for satisfactory operation and longevity.

Extreme pressure (EP) greases are normally recommended, with NLGI #0 or #1 being the normal grades, depending on coupling RPM. NLGI #0 is recommended for low speed applications.

Refer to manufacturer's catalogues for specific guidelines with regard to lubrication procedure, type to use, and interval requirements.

C6.10.6

This factory lubrication treatment protects against corrosion during shipping and storage, as well as providing lubrication during the initial service period.

Based on manufacturer's guidelines, the wire rope must be periodically re-lubricated at intervals during its lifetime. The lubricant shall be specifically manufactured as a wire rope dressing.

A good wire rope lubricant in the Wire Rope Users Manual (1993) should:

- be free from acids and alkalis,
- have sufficient adhesive strength to remain on the ropes,
- be of a viscosity capable of penetrating the interstices between wires and strands,
- not be soluble in the medium surrounding it under the actual operating conditions,
- have a high film strength, and
- resist oxidation.

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APPENDIX A6

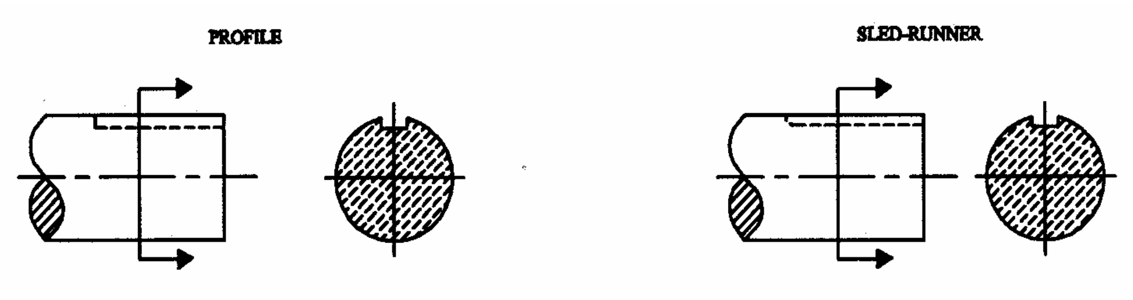
A6.1 STRESS CONCENTRATION FACTORS FOR KEYWAYS AND THREADS

The following tables give the values for K_F and K_{FS} directly. Use these when analyzing keyways and threads. There are no further calculations required using q.

Table A6.1-1 - Fatigue Stress Concentration Factors for Threads

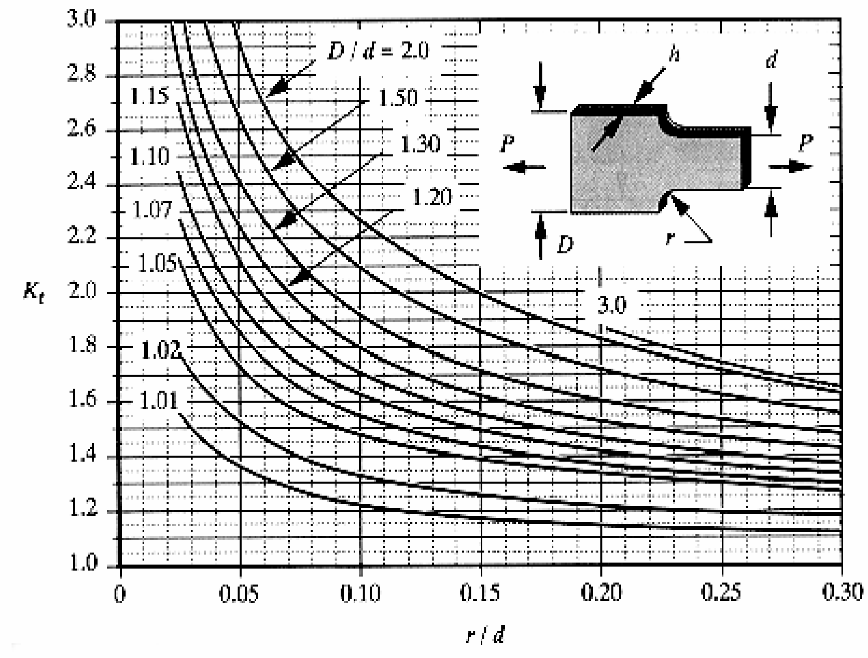
		BENDING OR TENSION	
		ROLLED	CUT
K_F or K_{FS}	ANNEALED	2.2	2.3
	QUENCHED & DRAWN	3.0	3.8

Table A6.1-2 - Fatigue Stress Concentration Factors for Keyways



	PROFILE		SLED-RUNNER	
	BENDING K_F	TORSION K_{FS}	BENDING K_F	TORSION K_{FS}
ANNEALED	1.6	1.3	1.3	1.3
QUENCHED & DRAWN	2.0	1.6	1.6	1.6

A6.2 CHARTS OF THEORETICAL STRESS CONCENTRATION FACTORS

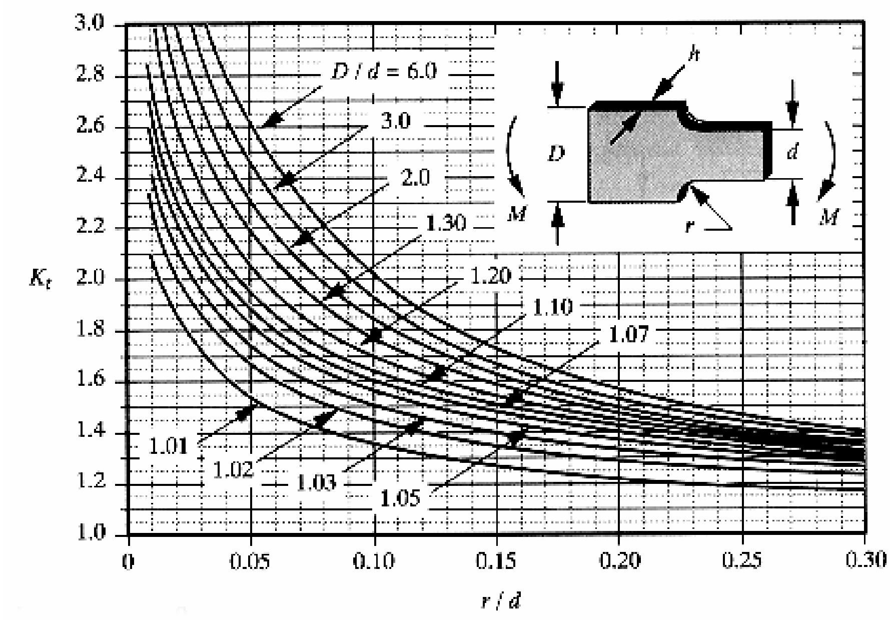


$$K_t \cong A\left(\frac{r}{d}\right)^b$$

where :

D / d	A	b
2.00	1.099 60	−0.320 77
1.50	1.076 90	−0.295 58
1.30	1.054 40	−0.270 21
1.20	1.035 10	−0.250 84
1.15	1.014 20	−0.239 35
1.10	1.013 00	−0.215 35
1.07	1.014 50	−0.193 66
1.05	0.987 97	−0.138 48
1.02	1.025 90	−0.169 78
1.01	0.976 62	−0.106 56

Figure A6.2-1: Geometric Stress Concentration Factor K_t for a Filleted Flat Bar in Axial Tension (Norton 1998).

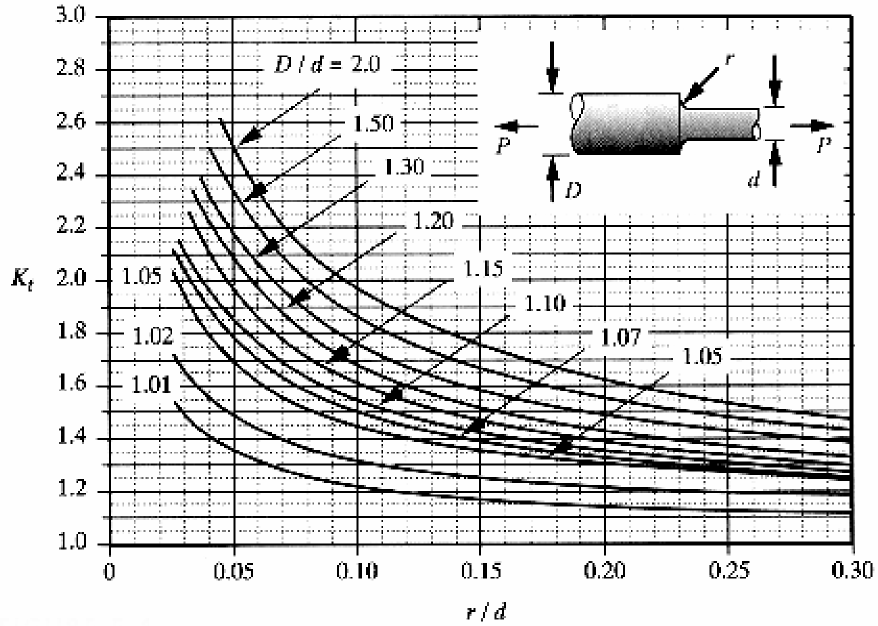


$$K_t \cong A\left(\frac{r}{d}\right)^b$$

where :

D / d	A	b
6.00	0.895 79	−0.358 47
3.00	0.907 20	−0.333 33
2.00	0.932 32	−0.303 04
1.30	0.958 80	−0.272 69
1.20	0.995 90	−0.238 29
1.10	1.016 50	−0.215 48
1.07	1.019 90	−0.203 33
1.05	1.022 60	−0.191 56
1.03	1.016 60	−0.178 02
1.02	0.995 28	−0.170 13
1.01	0.966 89	−0.154 17

Figure A6.2-2: Geometric Stress Concentration Factor K_t for a Filleted Flat Bar in Bending (Norton 1998).

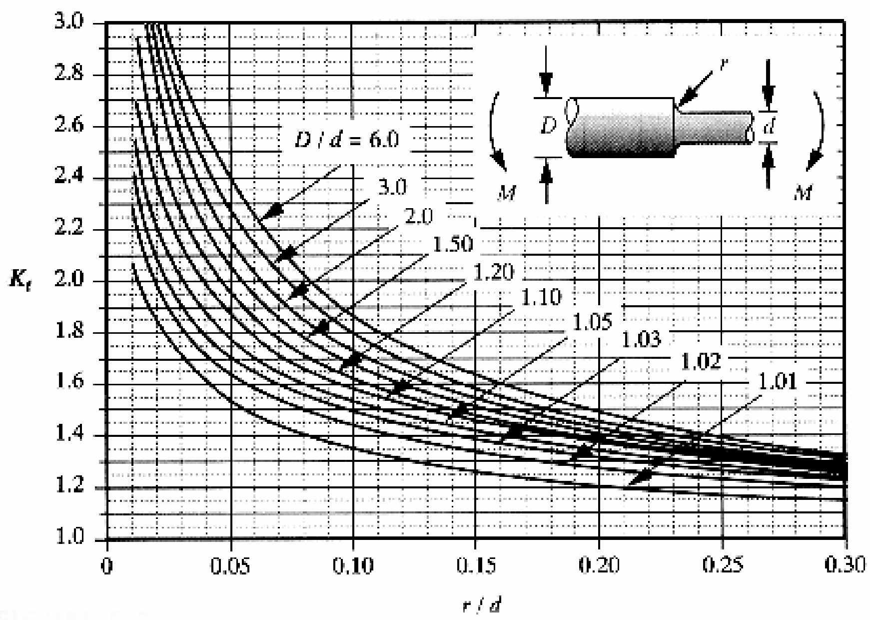


$$K_t \cong A\left(\frac{r}{d}\right)^b$$

where :

D/d	A	b
2.00	1.014 70	–0.300 35
1.50	0.999 57	–0.282 21
1.30	0.996 82	–0.257 51
1.20	0.962 72	–0.255 27
1.15	0.980 84	–0.224 85
1.10	0.984 50	–0.208 18
1.07	0.984 98	–0.195 48
1.05	1.004 80	–0.170 76
1.02	1.012 20	–0.124 74
1.01	0.984 13	–0.104 74

Figure A6.2-3: Geometric Stress Concentration Factor K_t for a Shaft with a Shoulder Fillet in Axial Tension (Norton 1998).

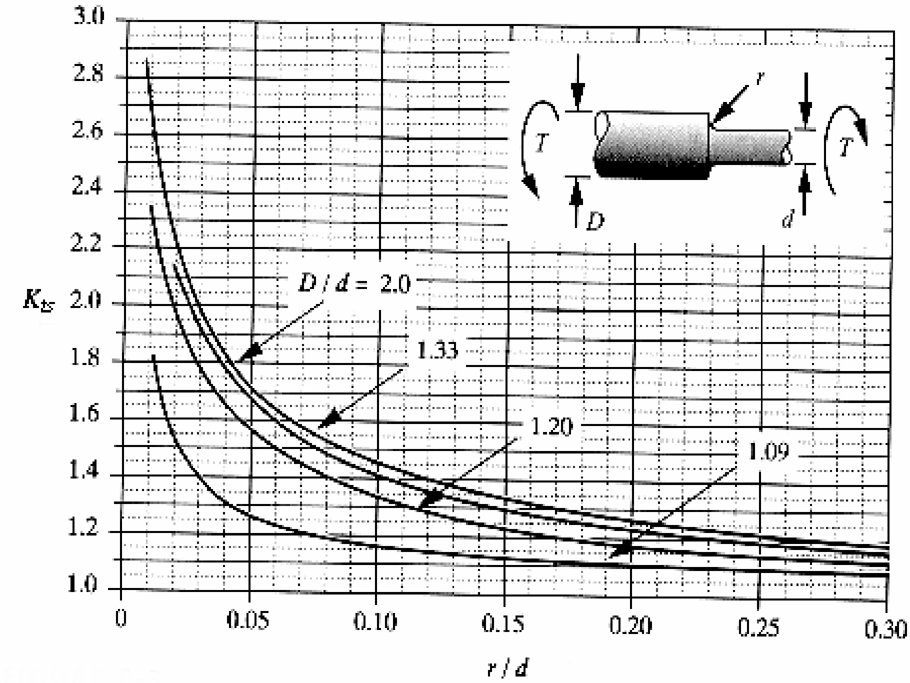


$$K_t \cong A\left(\frac{r}{d}\right)^b$$

where :

D/d	A	b
6.00	0.878 68	–0.332 43
3.00	0.893 34	–0.308 60
2.00	0.908 79	–0.285 98
1.50	0.938 36	–0.257 59
1.20	0.970 98	–0.217 96
1.10	0.951 20	–0.237 57
1.07	0.975 27	–0.209 58
1.05	0.981 37	–0.196 53
1.03	0.980 61	–0.183 81
1.02	0.960 48	–0.177 11
1.01	0.919 38	–0.170 32

Figure A6.2-4: Geometric Stress Concentration Factor K_t for a Shaft with a Shoulder Fillet in Bending (Norton 1998).

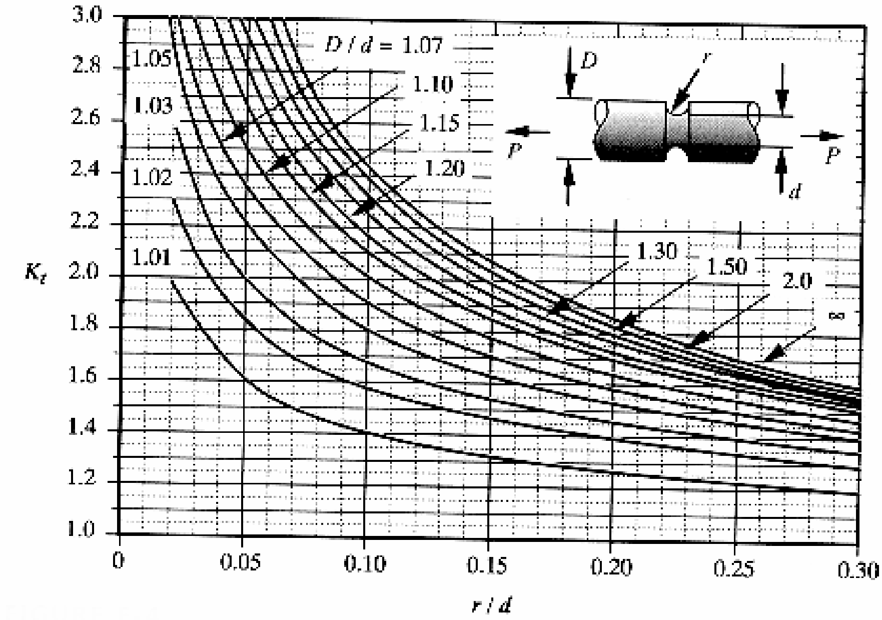


$$K_{ts} \cong A\left(\frac{r}{d}\right)^{b}$$

where :

D/d	A	b
2.00	0.863 31	−0.238 65
1.33	0.848 97	−0.231 61
1.20	0.834 25	−0.216 49
1.09	0.903 37	−0.126 92

Figure A6.2-5: Geometric Stress Concentration Factor K_{ts} for a Shaft with a shoulder Fillet in Torsion (Norton 1998).

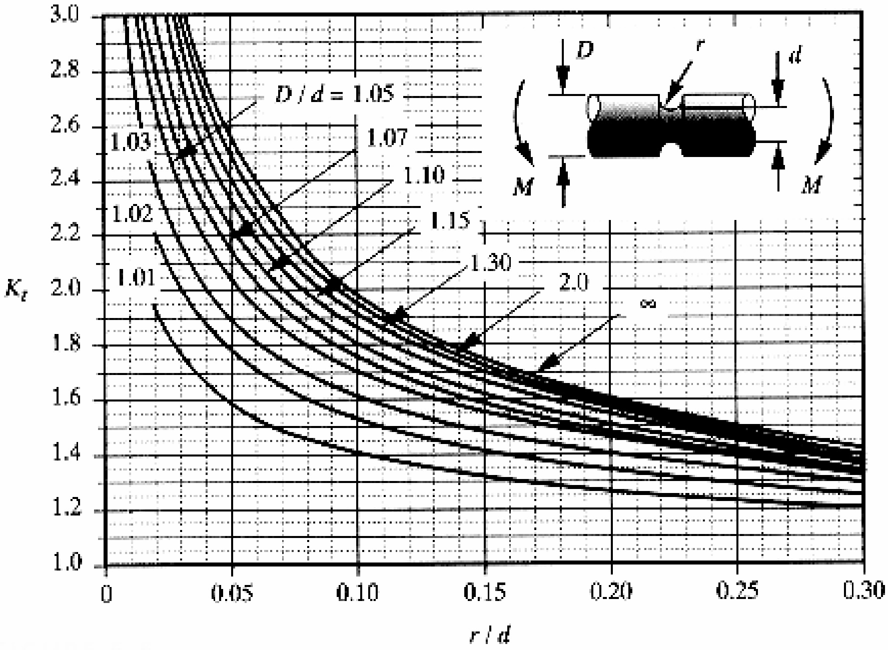


$$K_t \cong A\left(\frac{r}{d}\right)^{b}$$

where :

D/d	A	b
∞	0.993 72	−0.393 52
2.00	0.993 83	−0.382 31
1.50	0.998 08	−0.369 55
1.30	1.004 90	−0.355 45
1.20	1.010 70	−0.337 65
1.15	1.026 30	−0.316 73
1.10	1.027 20	−0.294 84
1.07	1.023 80	−0.276 18
1.05	1.027 20	−0.252 56
1.03	1.036 70	−0.216 03
1.02	1.037 90	−0.187 55
1.01	1.000 30	−0.156 09

Figure A6.2-6: Geometric Stress Concentration Factor K_t for a Grooved Shaft in Axial Tension (Norton 1998).

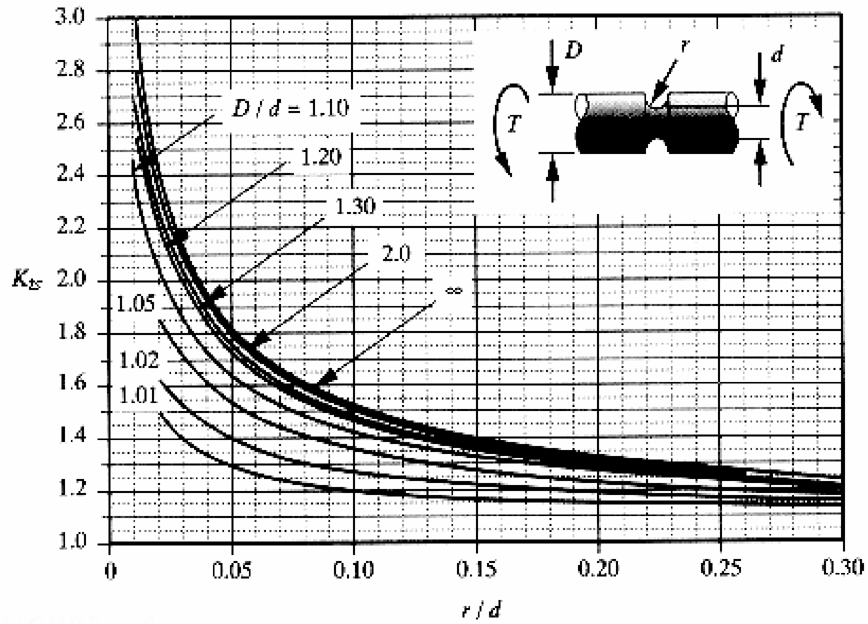


$$K_t \cong A\left(\frac{r}{d}\right)^b$$

where :

D/d	A	b
∞	0.948 01	−0.333 02
2.00	0.936 19	−0.330 66
1.50	0.938 94	−0.323 80
1.30	0.942 99	−0.315 04
1.20	0.946 81	−0.305 82
1.15	0.953 11	−0.297 39
1.12	0.955 73	−0.288 86
1.10	0.954 54	−0.282 68
1.07	0.967 74	−0.264 52
1.05	0.987 55	−0.241 34
1.03	0.990 33	−0.215 17
1.02	0.977 53	−0.197 93
1.01	0.993 93	−0.152 38

Figure A6.2-7: Geometric Stress Concentration Factor K_t for a Grooved Shaft in Bending (Norton 1998).

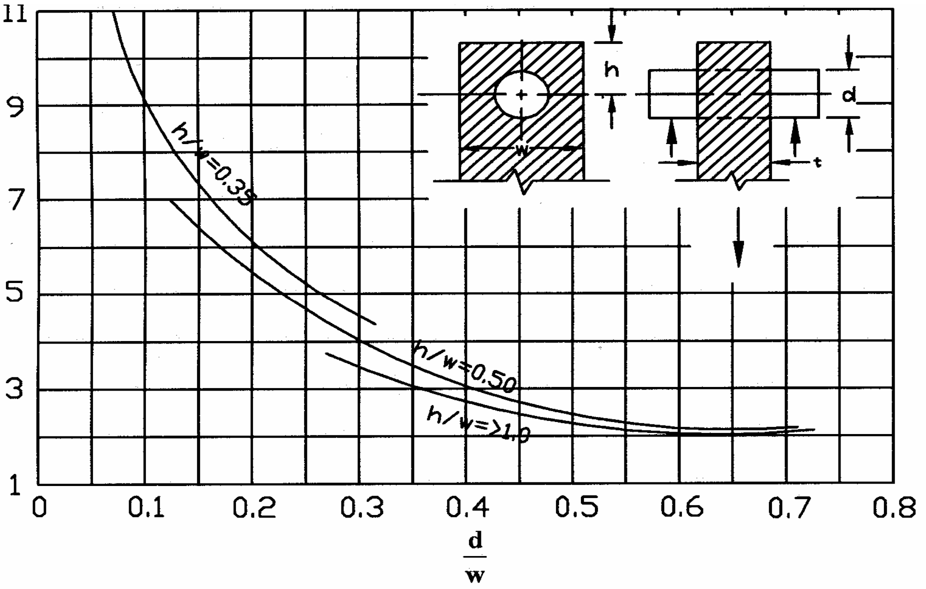


$$K_{ts} \cong A\left(\frac{r}{d}\right)^b$$

where :

D/d	A	b
∞	0.881 26	−0.252 04
2.00	0.890 35	−0.240 75
1.30	0.894 60	−0.232 67
1.20	0.901 82	−0.223 34
1.10	0.923 11	−0.197 40
1.05	0.938 53	−0.169 41
1.02	0.968 77	−0.126 05
1.01	0.972 45	−0.101 62

Figure A6.2-8: Geometric Stress Concentration Factor K_{ts} for a Grooved Shaft in Torsion (Norton 1998).



$$K_t \approx A\left(\frac{d}{w}\right)^b$$

where:

$\frac{h}{w}$	A	b
≥ 1.00	1.15	–0.9723
0.50	1.537	–0.7923
0.35	2.071	–0.6447

Figure A6.2-9: Geometric Stress Concentration Factor K_t for a Plate Loaded in Tension by a Pin Through a Hole (Shigley, 1989).