Heavy Movable Structures, Inc.

SEVENTH BIENNIAL SYMPOSIUM

November 4 - 6, 1998

Grosvenor Resort
Walt Disney World Village
Lake Buena Vista, Florida

"An Overview of NCHRP 12-44-Proposed Recommended Specifications for Movable Highway Bridges"

by

Donald L. Miller, Modjeski and Masters, Inc.

An Overview of NCHRP PROJECT 12-44
RECOMMENDED SPECIFICATIONS FOR MOVABLE HIGHWAY BRIDGES
With Particular Focus on Mechanical Machinery Design

Donald L. Miller, P.E.
Modjeski and Masters, Inc.
Harrisburg, PA

INTRODUCTION:

On July 26, 1996, Modjeski and Masters, Inc. was awarded NCHRP Project 12-44- to initiate the complete re-write of the AASHTO Standard Specifications for Movable Highway Bridges. This may be the first complete re-write since the original publication of the movable bridge specification in 1938.

Modjeski and Masters has recently submitted the Second Draft of the new specifications on September 17, 1998, and this paper is based on that submittal. The final draft of the new specifications will be submitted on December 23, 1998, with the tentative date for approval of the new specifications by AASHTO during its Spring 1999 meeting.

Although this paper will briefly present the major changes to the entire specifications, the paper's main emphasis will be on the changes to the Mechanical section.

PROJECT OBJECTIVES:

The stated objective from the project statement was: "The recommended specifications will be based on current practice and technology, will be consistent with the format and philosophy of the AASHTO *LRFD Bridge Design Specifications*, and will be accompanied by a commentary."

KEY FEATURES OF NEW SPECIFICATIONS:

- A Recommended Design Specification (not a Contractor's Document)
- A totally Metric (S.I.) Document
- Follows the LRFD format- 50-50 Specification-Commentary
- Uses the LRFD design philosophy wherever applicable
- Contains totally new sections on Seismic Design and Vessel Collision
- Completely revised sections on Mechanical, Hydraulic, and Electrical Design

GENERAL OUTLINE OF THE NEW SPECIFICATIONS:

Section 1	General Provisions
Section 2	Structural Design
Section 3	Seismic Design
Section 4	Vessel Collision Considerations
Section 5	Mechanical Design
Section 6	Hydraulic Design
Section 7	Electrical Design

SIGNIFICANT TOPICS OF EACH SECTION:

Section 1-
- Design Philosophy
- Limit States- LRFD Methodology
    - Service
    - Fatigue
    - Strength
    - Extreme
- General Provisions

Section 2-
- Load Combinations for each Movable Bridge type
    - Revised and in Table Format

Section 3-
- Performance Criteria for Seismic Design
- Seismic Loads
- Seismic Analysis- Specific to Movable Bridge type
- Design Guidelines-Specific to Movable Bridge type
- Appendix listing Case Studies

Section 4-
- Performance Criteria for Vessel Collision
- Initial Planning
- Bridge Protection Systems
- Design Guidelines
- Appendix listing Collision Cases

Section 5-

- Design Loading Criteria- Specific to Movable Bridge type
  - Sizing the Prime Mover
  - Holding Requirements
  - Machinery Design Criteria (See Draft, p. 5-13)[1]
- Fatigue Limit State
  - Endurance Limit (See Draft, p. 5-22)[1]
  - Open Spur Gearing Design Stresses
    - Bending Fatigue
    - Surface Durability
    - Intermittent Overload
- Design of Machinery with Fatigue Limit State
  - Fatigue Stress Concentration Factors (See Draft, pp. 5-27, A5-3, A5-4)[1]
  - Design of Shafts and Trunnions (See Draft, p. 5-29)[1]
- Open Spur Gearing Design
  - Follows current AGMA Standards (See Draft, pp. 5-33,34,35, 36,37,38)[1]
    - Design to Prevent Fatigue Failure
    - Design to Prevent Wear and Pitting (Surface Durability)
    - Design to Prevent Failure by Yielding due to Intermittent Overload
- Plain Bearing Design
  - Design based on Pressure, Velocity, and the Product pV (See Draft, pp.5-41, 42, 43)[1]
    - Covers Self- Lubricating and Non-Metallic Bearing Materials
- Rolling Element Bearing Design
  - Follows current ABMA Standards (See Draft. p. 5-44)[1]
- Fits and Finishes
  - Follows Metric Standards (See Draft, p. 5-47)[1]
- Keys and Keyways
  - Follows Metric Standards (See Draft, p. 5-50)[1]
- Lubrication
  - Expanded Section- Lubrication Guidelines by Component
- Appendix with Stress Concentration Graphs and Equations

Section 6-

- Design Loading Criteria
- Design of Specific Components
- New Topics include:
  - Hydraulic Cylinder Buckling Equations
  - Guidelines for Working Pressures
  - Calculating Efficiencies
  - Guidelines for Maximum Flow Rates (Velocities)

Section 7-

- Electrical Supply and Power Distribution
- Electrical Control Systems
  - Relay Logic
  - Programmable Logic Controllers (PLC)
  - Position Indicator Systems
- Electric Motors and Motor Controls
  - Stepped Resistor
  - SCR Speed Control
  - Flux Vector Speed Control
- Improved Materials/Equipment Requirements
- New Section on Lightning Protection

REFERENCE:

1. RECOMMENDED SPECIFICATION FOR MOVABLE HIGHWAY BRIDGES; TASK 6, SECOND DRAFT SPECIFICATION SUBMISSION; NCHRP 12-44; MODJESKI AND MASTERS, INC.; SEPTEMBER 17, 1998

EXCERPTS FROM SECTION 5
OF THE REFERENCED DRAFT

ECIFICATIONS

COMMENTARY

Table 5.5.3.1-1 - Machinery Design Prime Mover Loads

Prime Mover	Normal Allowable Stresses (Normal Operating Time)	Overload Limit State Stress
.C. (Uncontrolled)	1.5 FLT	Greater of 1.5 ST or 1.5 BDT
.C. (Controlled)	1.5 FLT	Greater of 1.0 ST or 1.5 AT
).C. (Controlled)	1.5 FLT	3.0 FLT
lydraulic	1.3 FLT	1.0 PT @ Maximum Rated Pressure
.C. Engines	1.5 FLT	1.0 PT @ Full Throttle
Ianual Operation	See provisions of Article 5.5.2.3	

Where controlled A.C. Drivers are specified, design for eakdown torque shall not be considered.

The provisions of Article 5.5.4.5 shall apply when termining resistance to braking and holding loads.

An A.C. drive shall be considered "controlled" if it has an effective and proven mode of <u>TORQUE</u> limiting at any motor speed (0 RPM through full load speed). Commonly used current limiting may not produce effective torque limiting at all speeds.

5.3.2 ENGINE-GENERATOR DRIVES

The provisions of Article 5.9 shall apply.

5.3.3 EMERGENCY DRIVES

Machinery powered by Auxiliary Electric Motor Drive iall satisfy the requirements of Section 7.

Machinery powered by Auxiliary Hydraulic Motor/HPU ive shall satisfy the requirements of Section 6.

Machinery powered by Auxiliary Internal Combustion igine Drive shall be designed for 100 percent of aximum peak engine torque at full throttle.

Where the bridge or parts thereof are to be operated hand, the number of people and the time of operation iall be calculated based on the following assumptions:

The force which a person can exert continuously on a capstan lever is 180 N while walking at a speed of 0.8 m/s.

The force which a person can exert continuously on a crank at a radius of 380 mm is 130 N with rotation at 15 RPM.

For calculating the strength of the machinery parts, the force one person applies to a capstan lever shall be assumed as 670 N, and to a crank as 220 N. Under these forces, the allowable unit stresses may be increased 50 percent.

PECIFICATIONS

6.3.2 ENDURANCE LIMIT

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

$$= \alpha \, \sigma_{ut} \; (C_D \; C_S \; C_R \; C_T \; C_M) \qquad (5.6.3.2\text{-}1)$$

here:

$_t$ = specified minimum ultimate tensile strength (MPa)

= 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 .... $\alpha = 0.40$

= size factor based on shaft diameter, D (mm) taken as:

- For $D \le 8$ mm, $C_D = 1$ (5.6.3.2-2)
- For $D > 8$ mm, $C_D = (D / 7.6)^{-0.113}$ (5.6.3.2-3)

= surface roughness factor taken as:

$$C_S = a \; (\sigma_{ut})^b \qquad (5.6.3.2\text{-}4)$$

ble 5.6.3.2-1 - Variables for Determining C_s urface Roughness Factor)

Condition	a	b
or a ground surface	1.58	-0.085
or a cold finished or mooth machined surface ith $R_a \le 0.8$ μm	4.51	-0.265
or a hot rolled or rough achined with $R_a > 0.8$ m, or as a heat treated urface	57.7	-0.718
or an as cast or as rged surface	272.0	-0.995

= surface roughness factor taken as an arithmetic mean (μm)

COMMENTARY

C5.6.3.2

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 100 to 200 mm 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.

See National Standard ANSI/ASME B46.1

'ECIFICATIONS

COMMENTARY

= 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

These values 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.

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

= any miscellaneous factors applicable to the details of a particular design

Miscellaneous factors are associated with welding, shot peening, plating, corrosion, or other manufacturing or environmental factor 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.

;.4 Resistance of Open Spur Gearing Using owable Stresses

;.4.1 GENERAL

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

failure of the teeth at the fatigue limit state,

surface durability through pitting and wear resistance, and

resistance for overload conditions.

;.4.2 SPUR GEAR BENDING RESISTANCE AT THE TIGUE LIMIT STATE (σ_{FP})

C5.6.4.2

Resistance shall be based upon the allowable bending ess σ_{FP} (MPa) used in the equations specified in Article '.5.2.2.

For through hardened steel gear teeth having a Brinell 'dness between 180 and 400, and the use of AGMA ade 1 material, σ_{FP} may be taken as:

See AGMA 2101-C95 or latest revision for the definition of Grade 1 steel.

$$= 0.533\, H_B + 88.3 \qquad (5.6.4.2\text{-}1)$$

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

Equation 1 is for one way bending of the gear teeth during any revolution. For idler gears, use 70 percent of this value for σ_{FP}.

ere:

= Brinell hardness for the teeth

SPECIFICATIONS

5.7.3.2 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:

$$K_F = 1 + q\,(K_t - 1) \qquad (5.7.3.2\text{-}1)$$

$$K_{FS} = 1 + q\,(K_{ts} - 1) \qquad (5.7.3.2\text{-}2)$$

in which:

- For ductile materials,

$$q = \frac{1}{\left(1 + \frac{\sqrt{a}}{\sqrt{r}}\right)} \qquad (5.7.3.2\text{-}3)$$

where:

= radius of notch or fillet (mm)

$\sqrt{a}$ = Neuber constant corresponding to the minimum specified ultimate tensile stress as specified in Table 1 ($\sqrt{mm}$)

Table 5.7.3.2-1 - Value of Neuber Constant

σ_{ut} (MPa)	$\sqrt{a}$ $(mm)^{0.5}$
420	0.54
630	0.35
840	0.25
980	0.20
1260	0.12

COMMENTARY

C5.7.3.2

Stress concentrations are probably the most critical criteria to consider when designing to prevent fatigue failure caused by fluctuating stresses. 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 A5.

For ductile materials only, the theoretical stress concentration factors from the graphs, K_t or K_{ts} , 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.

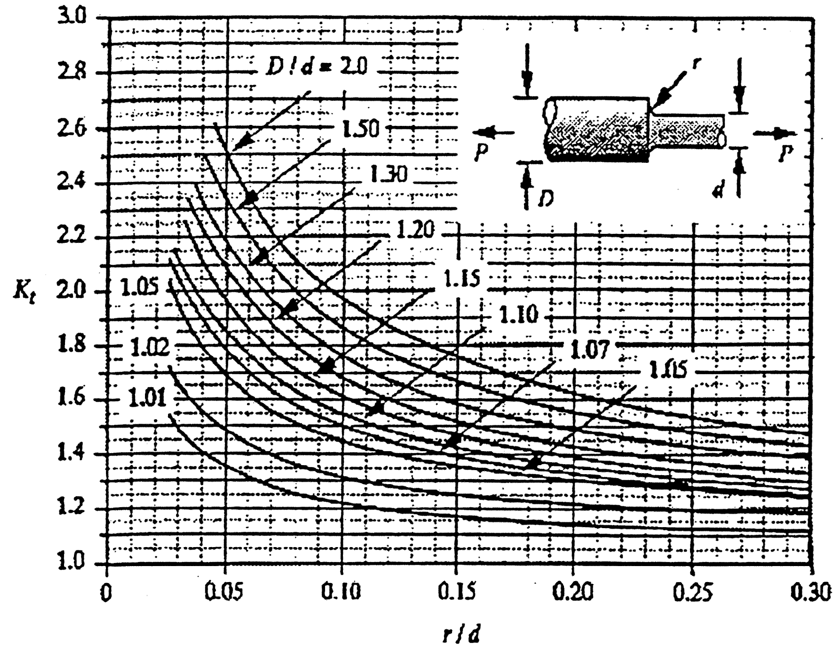
The tables in Appendix A5 provide values of K_F or K_{FS} for analyzing parts with threads and keyways. There is no modification required, using q, since these values are given as fatigue stress concentration factors.

For a fillet or notch radius equal to 10 mm. Equation 3 yields the following values:

Table C5.7.3.2-1 - Value of Notch Sensitivity Factor

σ_{ut} (MPa)	q
420	0.85
630	0.90
840	0.93
980	0. 94
1260	0.96

That is, $K_F = K_t$ and $K_{FS} = K_{ts}$

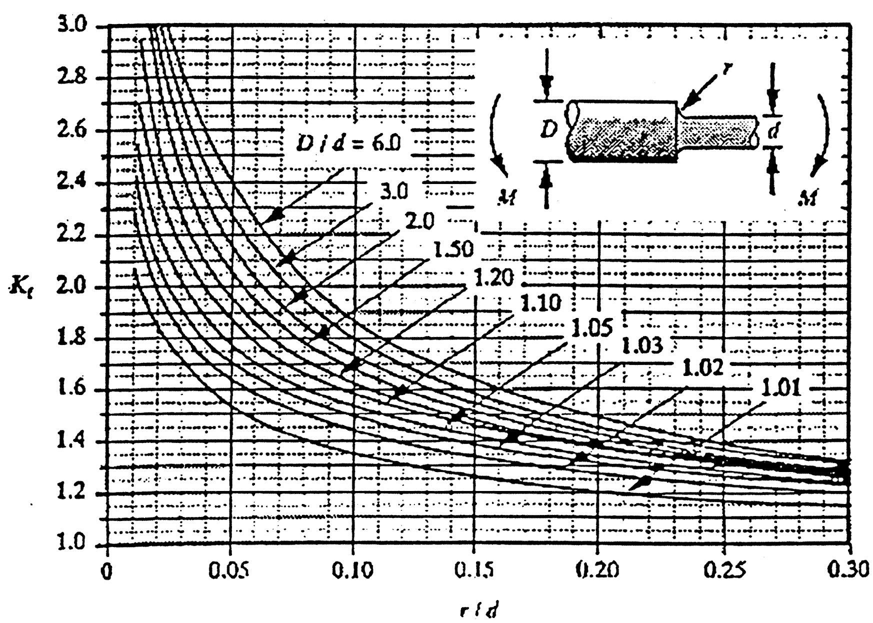


$K_t \cong A\left(\frac{r}{d}\right)^b$

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 A5.2-3: Geometric Stress Concentration Factor K_t for a Shaft with a Shoulder Fillet in Axial Tension. (Norton, 1998)

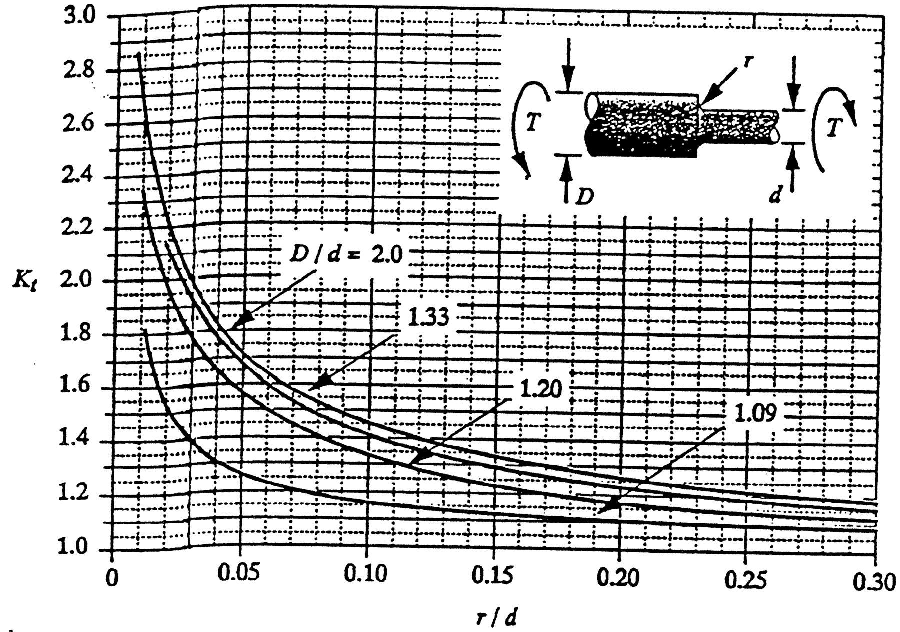


$K_t \cong A\left(\frac{r}{d}\right)^b$

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 A5.2-4: Geometric Stress Concentration Factor K_t for a Shaft with a Shoulder Fillet in Bending. (Norton, 1998)

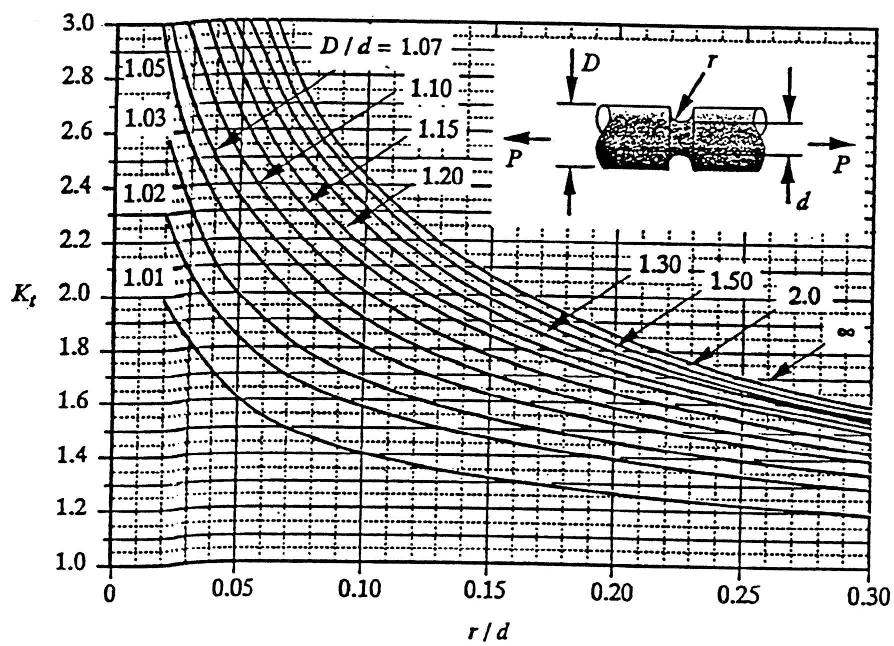


$$K_t \cong A\left(\frac{r}{d}\right)^b$$

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 A5.2-5: Geometric Stress Concentration Factor K_t 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 A5.2-6: Geometric Stress Concentration Factor K_t for a Grooved Shaft in Axial Tension. (Norton, 1998)

SPECIFICATIONS

where:

σ_a = amplitude normal stress specified in Article 5.7.3.3.1 (MPa)

σ_e = endurance limit of a steel shaft specified in Article 5.6.3.2 (MPa)

σ_m = mean normal stress specified in Article 5.7.3.3.1 (MPa)

τ_a = amplitude shear stress specified in Article 5.7.3.3.1 (MPa)

τ_m = mean shear stress specified in Article 5.7.3.3.1 (MPa)

K_F = stress concentration factor for fluctuating normal stress specified in Article 5.7.3.2 (DIM)

K_{FS} = stress concentration factor for fluctuating shear stress specified in Article 5.7.3.2 (DIM)

5.7.4 Shafts, Trunnions, Machine Elements Subjected to Cyclic Stresses

5.7.4.1 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:

$$\frac{32}{\pi d^3}\left[\frac{K_F M_a}{\sigma_e} + \frac{3K_{FS} T_m}{2\sigma_{yt}}\right] \leq 0.8 \qquad (5.7.4.1\text{-}1)$$

where:

K_F = fatigue stress concentration factor (bending)

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

M_a = amplitude bending moment (N-mm)

T_m = mean (steady) torsional moment (N-mm)

σ_e = endurance limit of the steel shaft specified in Article 5.6.3.2 (MPa)

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

COMMENTARY

C5.7.4.1

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.

SPECIFICATIONS

COMMENTARY

to the gear shaft, T (N-mm), and the power which the gear transmits, P (kW).

$$T = \frac{F_t d_{w1}}{2} \qquad \text{(C5.7.5.2.1-1)}$$

$$P = \frac{F_t d_{w1} \omega_1}{1.91 \times 10^7} \qquad \text{(C5.7.5.2.1-2)}$$

where:

d_{w1} = pinion pitch diameter (mm)

ω_1 = pinion RPM

AGMA presents design/analysis equations (ANSI/AGMA 2101-C95 - Metric Edition of ANSI/AGMA 2001-C95) to determine the safe power that can be transmitted by the gear teeth, either safe power based on fatigue strength P_{ay} (kW) or safe power based on pitting resistance/wear/ surface durability P_{az} (kW).

These equations have been modified into equations to find the factored flexural resistance, F_{tay}, and factored pitting resistance, F_{taz}.

.7.5.2.2 Design for the Fatigue Limit State

The factored flexural resistance, F_{tay}(N) of the spur ear teeth, based on fatigue, shall be determined as:

$$_{tay} = \frac{b m_t Y_J \sigma_{FP} Y_N}{K_o K_v K_s K_H K_B S_F Y_\theta Y_Z} \qquad \text{(5.7.5.2.2-1)}$$

which:

$$_v = \left[\frac{A + \sqrt{200 v_t}}{A}\right]^B \qquad \text{(5.7.5.2.2-2)}$$

$$= 50 + 56\,(1.0 - B) \qquad \text{(5.7.5.2.2-3)}$$

$$= 0.25\,(12 - Q_v)^{0.667} \qquad \text{(5.7.5.2.2-4)}$$

$$_t = \pi \omega_1 \frac{d_{w1}}{60\,000} \qquad \text{(5.7.5.2.2-5)}$$

here:

, = dynamic factor (DIM)

C5.7.5.2.2

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

The following must then be satisfied:

$$F_t \le F_{tay} \qquad \text{(C5.7.5.2.2-1)}$$

PECIFICATIONS

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

$_t$ = transverse tooth module (mm)
= d_{w1} / N_{w1}

$_J$ = geometry factor for bending strength

= pitch line velocity (m/s):

$_{v1}$ = pitch diameter of the pinion (mm)

$_{w1}$ = number of teeth on the pinion

$_{FP}$ = allowable bending stress specified in Article 5.6.3.2 (MPa)

$_N$ = life factor for bending resistance taken as:

- For $H_B \approx 250$ and $10^3 < n_L < 3 \times 10^6$

$$Y_N = 4.9404\ n_L^{-0.01045} \qquad (5.7.5.2.2\text{-}6)$$

- For $n_L > 3 \times 10^6$ load cycles, regardless of hardness

$$Y_N = 1.6831\ n_L^{-0.0323} \qquad (5.7.5.2.2\text{-}7)$$

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

$_B$ = Brinell hardness for the teeth

= number of load cycles

$_1$ = pinion RPM

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

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

$_H$ = load distribution factor taken as $K_H = 1.26 + 0.00102\,b$ for open gearing, adjusted at assembly, with $b < 700$ mm, and $b/d_{w1} < 1$ (DIM)

$_B$ = rim thickness factor taken as 1.0 if $m_B = t_R / h_t > 1.2$

$_B$ = the backup ratio (DIM)

COMMENTARY

Table 1 contains some standard tooth module values and equivalent diametral pitch values:

Table C5.7.5.2.2-1 - Standard Tooth Modules and Corresponding Diametral Pitches

m_t (mm)	P_d (in.$^{-1}$)
6	4.233
8	3.175
10	2.540
12	2.117
14	1.814
16	1.588
18	1.411
20	1.270
25	1.016
30	0.847
35	0.726
40	0.635
45	0.564
50	0.508

Tables C2 and C3 provide values of the geometry factor, Y_J for values of 18, 19, 20, and 21 tooth pinions, for 20° full depth, equal addendum teeth only. (Modified from AGMA 908-B89)

For a gear quality of 6 or 7 use Table C3 for Y_J and tooth loading at the tip; use the highest single tooth contact Table C2 for Y_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.

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.

SPECIFICATIONS

$_R$ = rim thickness (mm)

$_t$ = total tooth height (mm)

$_F$ = safety factor for bending strength (fatigue) $S_F > 1$

$_\theta$ = temperature factor taken as 1.0 for gear temperatures less than 120° C (DIM)

$_z$ = reliability factor (DIM):

- 1.0 for 99 percent reliability
- 1.25 for 99.9 percent reliability

COMMENTARY

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 $m_t > 10$). (Norton, 1997; Shigley, 1983)

This is an approximate equation, derived from AGMA Standard 2101-C95. See this or the latest AGMA standard for a more accurate calculation of K_H.

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

Table C5.7.5.2.2-2 - Y_J Factor for 20° Full Depth Spur Pinion/G (P, G)

Equal Addendum Loaded at Highest Point, Single Tooth Contact								
GEAR TEETH	PINION TEETH (N_{w1})							
	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	0.3
26	0.31	0.34	0.31	0.34	0.32	0.34	0.33	0.3
35	0.31	0.37	0.32	0.37	0.33	0.37	0.34	0.3
55	0.32	0.40	0.32	0.40	0.33	0.40	0.34	0.4
135	0.33	0.43	0.33	0.43	0.34	0.43	0.35	0.4

PECIFICATIONS

COMMENTARY

Table C5.7.5.2.2-3 - Y_J Factor for 20° Full Depth Spur Pinion/Ge (P, G)

Equal Addendum Loaded at Tooth Tip								
	PINION TEETH (N_{w1})							
GEAR TEETH	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

7.5.2.3 Surface Durability and Wear - Design Equations

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

$$_{az} = \frac{b\,d_{w1}Z_I}{K_oK_vK_sK_HK_R}\left(\frac{\sigma_{HP}Z_NZ_W}{Z_ES_HY_\theta Y_Z}\right)^2 \qquad (5.7.5.2.3\text{-}1)$$

here:

= tooth face width of the pinion or gear that has the narrowest face width (mm)

= geometry factor for pitting resistance (DIM)

= surface condition factor for pitting resistance taken as 1.0 for good tooth surface condition as specified in Article 5.4.8 for tooth surface finish depending on module (DIM)

$_{HP}$ = allowable contact stress for the lower Brinell hardness number of the pinion/gear pair as specified in Article 5.6.3.3 (MPa)

C5.7.5.2.3

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

The following must then be satisfied:

$$F_t \le F_{taz} \qquad (C5.7.5.2.3\text{-}$$

Table C1 gives values of Z_I modified from AGMA 90 B89, Z_I geometry factor tables - values for 18, 19, 20, a 21 tooth pinions, for 20° full depth, equal addendum tee only.

PECIFICATIONS

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

$_L$ = number of load cycles

$_W$ = hardness ratio factor for pitting resistance, taken as 1.0 if $H_{BP}/H_{BG} < 1.2$

$_H$ = safety factor for pitting resistance, i.e., surface durability taken as >1

$_E$ = elastic coefficient taken as 190 for steel pinion-steel gear $(MPa)^{0.5}$

7.5.2.4 Yield Failure at Intermittent Overload

Spur gear teeth shall be investigated for an infrequent erload condition, for which yield failure due to bending ight occur. In the absence of more design-specific quirements, the overload may be taken as that which uld have fewer than 100 cycles during the design life of e machinery.

The maximum factored resistance, F_{max} (N), based on yield failure of the gear teeth shall be taken as:

$$_{nax} = \frac{K_y b m_t K_f Y_J \sigma_s}{K_{HS}} \qquad (5.7.5.2.4\text{-}1)$$

ere:

COMMENTARY

Table C5.7.5.2.3-1 - Z_I Factors for 20° Full Depth Spur Pinions/Gears

Equal Addendum Factors same for both Pinion and Gear								
GEAR TEETH	PINION TEETH (N_{w1})							
	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	

n_L for surface durability is not necessarily the same as used for the fatigue life factor Y_N.

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

For other material combinations, refer to the AGM Standards. (If steel-cast iron or steel-bronze or plastic)

C5.7.5.2.4

The equation given in the AGMA Standards is modifi to solve for F_{max} which is the maximum peak transmitt tangential tooth load, based on yielding.

The following must then be satisfied:

$$F_t\,(max) \le F_{max} \qquad (C5.7.5.2.4\text{-}$$

based on the overload condition.

PECIFICATIONS

$_y$ = yield strength factor taken as 0.50 (DIM)

$_f$ = stress correction factor = 1 (DIM)

$_s$ = allowable yield stress number specified in Article 5.6.3.4 (MPa)

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

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

$_t$ = transverse tooth module (mm)

$_J$ = geometry factor for bending strength (DIM)

7.6 Enclosed Speed Reducers

7.6.1 GENERAL

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

Enclosed reducers shall be rated for a service factor of 0 as defined by the AGMA based on the power quirements specified in Article 5.5.4.1.

For electric motor drives, the peak or breakdown rque of the reducer shall not be less than the peak or eakdown torque of the motor.

Enclosed reducer bearings shall be of the rolling ement type and shall have a L-10 life of 40,000 hours as ecified in Article 5.7.7.2.1.

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

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

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

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

COMMENTARY

AGMA suggests using K_f = 1, since this is a yie criteria for failure of a ductile material. However, this K_f not the same term as defined in Article 5.4.3.1. This term is defined by AGMA 908-B89 (Equation 5.72) and a value less than 1.

AGMA gives an equation only for an enclosed drive

See Tables C5.7.5.2.2-2 and C5.7.5.2.2-3 fo suggested values of Y_J.

C5.7.6

C5.7.6.1

It is recommended that all gearing, except final driv gearing (e.g., rack and pinion), be designed using enclose speed reducers wherever feasible.

See the provisions of Article 5.10.4.2.

SPECIFICATIONS

Large bearings shall be provided with effective means 'or 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.

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

5.7.7.1.2 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.

$$p = \frac{F_r}{(DL)} \quad (5.7.7.1.2\text{-}1)$$

$$V = \frac{\pi D \omega}{60\,000} \quad (5.7.7.1.2\text{-}2)$$

where:

F_r = radial load (N)

p = pressure (MPa)

V = surface velocity (m/s)

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

L = length of the bearing (mm)

ω = journal rotational speed (RPM)

5.7.7.1.3 Lubricated Plain Bearings

Journal bearings normally shall have bronze bushings. For lightly loaded bearings, the bushings may be bronze or nonmetal 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 6 mm shall exist between the bushing of the cap and the bushing of the base into which laminated liners

COMMENTARY

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

C5.7.7.1.2

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

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

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

Table C1 lists maximum values for p, V and pV various commonly used bronze bearing alloys:

Table C5.7.7.1.2-1 - Performance Parameters for Cast Bronze Bearings

UNS ALLOY	p (MPa)	V (m/s)	pV (MPa·m/s)	COMMON NAME
C 86300	55.2	0.12	2.45	Mang. Bronze
C 91100	17.2	0.25	1.05	Phos. Bronze
C 91300	20.7	0.25	1.05	Phos. Bronze
C 93700	6.9	1.25	1.05	Tin Bronze
C 85400	24.1	0.50	1.75	Alum. Bronze

;PECIFICATIONS

COMMENTARY

hall be placed. The inside longitudinal corners of both alves shall be rounded or chamfered, except for a istance of 10 mm from each end or from the shaft houlder fillet tangent point.

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

.7.7.1.4 Self-Lubricating; Low Maintenance Plain ;earings

.7.7.1.4a Metallic Bearings

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

C5.7.7.1.4a

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

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

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

Table C5.7.7.1.4a-1 - Performance Parameters for Oi Impregnated Metals

MATERIAL	ASTM NO.	p (MPa)	V (m/s)	pV (MPa·m/s
Oilite Bronze	B-438-73 Gr 1 Type II	13.8	6.10	1.75
Super Oilite	B-439-70 Gr 4	27.6	1.14	1.23
Super Oilite - 16	B-426 Gr 4 Type II	55.2	0.18	2.63

.7.7.1.4b NonMetallic Bearings

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

C5.7.7.1.4b

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.

SPECIFICATIONS

COMMENTARY

Table C5.7.7.1.4b-1 - Performance Parameters for Nonmetal Bearings

MATERIAL	p (MPa)	V (m/s)	pV (MPa·m/s)
Acetal ("Delrin")	6.9	5.08	0.105
Nylon	6.9	5.08	0.105
Phenolics	41.0	12.70	0.525
TFE ("Teflon")	3.5	0.25	0.035
PTFE Composite	69.0	0.76	0.875

;.7.7.2 ROLLING ELEMENT BEARINGS

;.7.7.2.1 General

Roller elements shall be designed at the overload limit ;tate and shall satisfy:

$$P \geq F \quad (5.7.7.2.1\text{-}1)$$

/here:

= factored radial resistance specified in Articles 5.7.7.2.2 and 5.7.7.2.4 (N)

= factored load applied specified in Article 5.7.7.2.2 and 5.7.7.2.4 (N)

Rolling element bearings shall also be sized so that nder loads generated from 150% of the full load torque of ıe prime mover, the L-10 life shall be 40,000 hours at the verage running speed of the bearing.

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

Foot-mounted pillow block bases for units with small ore, usually taken to mean under 75 mm, should have lotted holes at the mounting feet to permit easy erection, djustment and replacement. If the mounting feet have lotted holes, the feet shall have machined ends to permit ıe use of end chocks or the unit shall be doweled in place fter installation and alignment.

C5.7.7.2.1

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).

PECIFICATIONS

COMMENTARY

7.7.2.2 Rolling Element Bearing Design

Where a radial roller bearing will have 40,000 hours of e with a reliability of 90 percent, i.e., L-10 life, the juivalent dynamic factored radial resistance, P_r (N) shall determined as:

$$. = 0.77C_r\left(\frac{1}{\omega}\right)^{0.3} \qquad (5.7.7.2.2\text{-}1)$$

here:

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

= rotational speed of the bearing inner race (RPM)

For radial roller bearings, the factored dynamic radial ad F_{xy} (N) may be determined as:

$$_y = X F_r + Y F_a \geq F_r \qquad (5.7.7.2.2\text{-}2)$$

iere:

= applied radial load (N)

= applied axial thrust (N)

= radial factor (DIM)

= thrust factor (DIM)

C5.7.7.2.2

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 5.7.7.2.3 and 5.7.7.2.4)

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.

At less than 5 RPM, the value of P_r calculated by Equation 1 may exceed 0.5 C_r. This will cause plastic deformation in the contact area. For low RPM, the conditions of Article 5.7.7.2.4 must also be satisfied.

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

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.

7.7.2.3 Roller Bearings for Heavy Loads

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

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

be relatively short for their diameter,

be closely spaced in separator cages,

C5.7.7.2.3

Heavy loads are generally defined as being greater than 3×10^6 N.

SPECIFICATIONS

Table 5.7.8-1 - Fits and Finishes

PART	FIT	FINISH R_a(µm)
Machinery base on steel	–	6.3
Machinery base on masonry	–	12.5
Shaft journals	H8/f7	0.2
Journal bushing	H8/f7	0.4
Split bushing in base	H7/h6	3.2
Solid bushing in base (to 6 mm wall)	H7/p6	1.6
Solid bushing in base (over 6 mm wall)	H7/s6	1.6
Hubs on shafts (to 50 mm bore)	H7/s6	0.8
Hubs on shafts (over 50 mm bore)	H7/s6	1.6
Hubs on main trunnions	H7/u6	1.6
Turned bolts in finished holes	H7/h6	1.6
Sliding bearings	H8/f7	0.8
Center discs	–	0.8
Keys and keyways		
Top and Bottom	H7/h6	1.6
Sides	H7/s6	1.6
Machinery parts in fixed contact	–	3.2
Teeth of open spur gears		
Under 8 mm module	–	0.8
8 to 16 mm module	–	1.6
Over 16 mm module	–	3.2

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

COMMENTARY

Table C5.7.8-1 - Equivalence for Surface Finish (µm vs. µin)

R_a(µm)	R_a(µin)
12.5	500
6.3	250
3.2	125
1.6	63
0.8	32
0.4	16
0.2	8

Using the hole basis system, the following ISO metric fits are recommended. The closest ANSI Inch Series Fit is listed in Table C2 to the right of reference.

Table C5.7.8-2 - ISO Preferred Fits (Hole Basis)

ISO PREFERRED FITS		ANSI-INCH
Clearance Fits		
H11/c11	Loose Running	RC9
H9/d9	Free Running	RC7
H8/f7	Close Running	RC4
H7/g6	Sliding Fit	RC2
H7/h6	Locational Clearance	LC2
Transitional Fits		
H7/k6	Locational Transition	LT3
H7/n6	Locational Transition	LT5
Interference Fits		
H7/p6	Locational Interference	LN2
H7/s6	Medium Drive Fit	FN2
H7/u6	Force Fit	FN4

Fits other than those listed in Table C2 may be used at the discretion of the Engineer.

PECIFICATIONS

COMMENTARY

Table C5.7.10.1-1 - Key Sizes

DIAMETER (mm) OVER-TO-INCLUDE		KEY SIZE (mm) (b x h)
12	17	5 x 5
17	22	6 x 6
22	30	8 x 7
30	38	10 x 8
38	44	12 x 8
44	50	14 x 9
50	58	16 x 10
58	65	18 x 11
65	75	20 x 12
75	85	22 x 14
85	95	25 x 14
95	110	28 x 16
110	130	32 x 18
130	150	36 x 20
150	170	40 x 22
170	200	45 x 25
200	230	50 x 28
230	260	56 x 32
260	290	63 x 32
290	330	70 x 36
330	380	80 x 40
380	440	90 x 45

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