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L.Borden, Modjeski & Masters, Pa.

DESIGN CRITERIA FOR SHAFTS AND OPEN GEARING ON MOVABLE BRIDGES

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INTRODUCTION

Whether you are rehabilitating an old movable bridge, or designing a new replacement movable bridge, correct design procedures are necessary to insure the long life of shafts, trunnions and open gearing.

AREA and AASHTO are typically noted for being conservative with their design equations and allowable stresses, but there are situations involving shafts, trunnions and open gearing where the resulting design could be flawed because fatigue, wear and stress concentrations were not adequately considered.

AREA/AASHTO equations for shafts and gearing are generally based on static stress conditions. There is no or little allowance for fatigue, wear, and stress concentrations in the design equations of these mechanical components. Even with liberal factors of safety based on yield strength, a static analysis/design will, in some cases, produce an improper design.

The design of shafts and trunnions will be treated separately from the design of open gearing, the latter using AGMA recommended equations.

SHAFT/TRUNNION DESIGN

Most shafts rotate, and when they rotate the bending stresses fluctuate or reverse, producing the probability of a fatigue failure. A fatigue failure, unlike a ductile static yielding failure, occurs suddenly and usually without prior warning. The failure resembles a brittle fracture, and occurs as the result of a circumferencial crack which propagates radially inward until the remaining material can no longer support the loading, hence a sudden unexpected fatigue failure.

AREA/AASHTO equations[1,2]* for shafts and trunnions are based on static stresses only, with a K factor intended to account for rotation of the shaft.

For instance:

Bending Stress:

$$\sigma = f_b = \frac{16K}{\pi d^3}\left(M + \sqrt{M^2 + T^2}\right) \qquad \text{(AREA 6.4.6)}$$

*See reference at end of paper.

Shear Stress:

$$\tau = S_s = \frac{16K}{\pi d^3}\left(\sqrt{M^2 + T^2}\right) \qquad \text{(AREA 6.4.6)}$$

where M = bending moment (lb-in), T = torque (lb-in), d = shaft diameter (inch).

For trunnions and counterweight shafts, K = 1.

For Rotating Shafts:

$$K = 1 + 0.03\sqrt{n} \qquad \text{(AREA 6.4.2)}$$

where n = shaft speed in RPM.

Allowable stresses given by AREA 6.4.2 provide for stress concentrations of 140% of computed stress. A 1.4 factor is relatively small and also, no reference is made as to how to determine the stress concentration factors, or when and how to use them in the equations. The reason for ignoring stress concentrations is that with static stresses, they do not effect failure. However, since most shafts are subjected to fluctuating stresses, fatigue failure is possible, and stress concentrations have a definite effect on fatigue life.

DESIGNING TO PREVENT FATIGUE FAILURE OF SHAFTS AND TRUNNIONS

The important quantity to consider in fatigue design is the endurance limit of the part (S_e). Steel has an endurance limit, aluminum does not. Knowing S_e, the part can be designed so that it will have millions of cycles of stress without fatigue failure.

The endurance limit depends on many factors, the most important of these being: ultimate tensile strength, size, surface finish, reliability and temperature.

By equation[3] (for steel subject to bending and torsion):

$S_e = 0.5 \times S_{ut} \times C_D \times C_S \times C_R \times C_T \times C_M$

S_{ut} = ultimate tensile strength (psi)

C_D = size (diameter) factor; for 1/2" or greater diameter, $C_D = 0.872(d)^{-0.1133}$

C_s = surface finish factor; for a typically machined surface (8-16 μin.):

$$C_s = 0.875 - \frac{S_{ut}(psi)}{800{,}000}$$

(for $60{,}000 \le S_{ut} \le 200{,}000$ psi)

C_R = reliability factor based on an 8% standard deviation on endurance limit. Use this, especially when the ultimate tensile stress S_{ut} is a "typical" value. R = reliability.

for: R = 50%, $C_R = 1.0$
R = 90%, $C_R = 0.90$
R = 99%, $C_R = 0.81$

C_T = temperature factor

C_T = 1 for steel up to 400° F
C_T = 0.7 for steel at 1,000° F

C_M = any miscellaneous factor

eg: welding, plating, shot peening, corrosion, etc.

Some design references include stress concentration fatigue factors with the endurance limit equation. Since most applications in design consist of combined bending and torsional stresses, and the stress concentration factors are different for each, it is better to include these factors with the respective stresses (or bending and torsional moments).

Stress concentration factors depend on the shaft configuration and on the type of loading. The most common stress concentration occurs at filleted shoulders on shafts. However, other common factors are for keyways, threaded shaft portions, laterally drilled holes, or circumferencial grooves.

A foremost reference of theoretical stress concentration factors (K_t for bending, K_{ts} for torsion) is by R. E. Peterson[4]. Some figures for bending and torsion of filleted shafts are included with this paper (Figures 1 and 2). The K_t and K_{ts} values depend on the ratio of diameters (D/d) and the ratio of fillet size to smaller diameter (r/d).

These theoretical factors are modified for fatigue by a notch sensitivity factor q or q_s to give K_f for bending and K_{fs} for torsion.

$K_f = 1 + q\ (K_t - 1)$
$K_{fs} = 1 + q_s\ (K_{ts} - 1)$

For fillet radii greater than 1/8-inch (r > 0.12") approximate q values are:

for S_{ut} = 80 ksi: q = 0.85, q_s = 0.87
for S_{ut} = 140 ksi: q = 0.90, q_s = 0.92

When in doubt, using q or q_s = 1 is conservative.
Then, $K_f = K_t$ and $K_{fs} = K_{ts}$.

The fatigue stress concentration factors and the endurance limit are then used in a design equation to solve for required shaft size.

$$d = \left[\frac{32n}{\pi}\left\{\left(\frac{K_f M}{S_e}\right)^2 + \left(\frac{K_{fs} T}{S_y}\right)^2\right\}^{\frac{1}{2}}\right]^{\frac{1}{3}}$$

The factor of safety (n) typically used for fatigue design is $n \geq 1.25$. (Note: n in this equation is safety factor, not shaft speed RPM.)

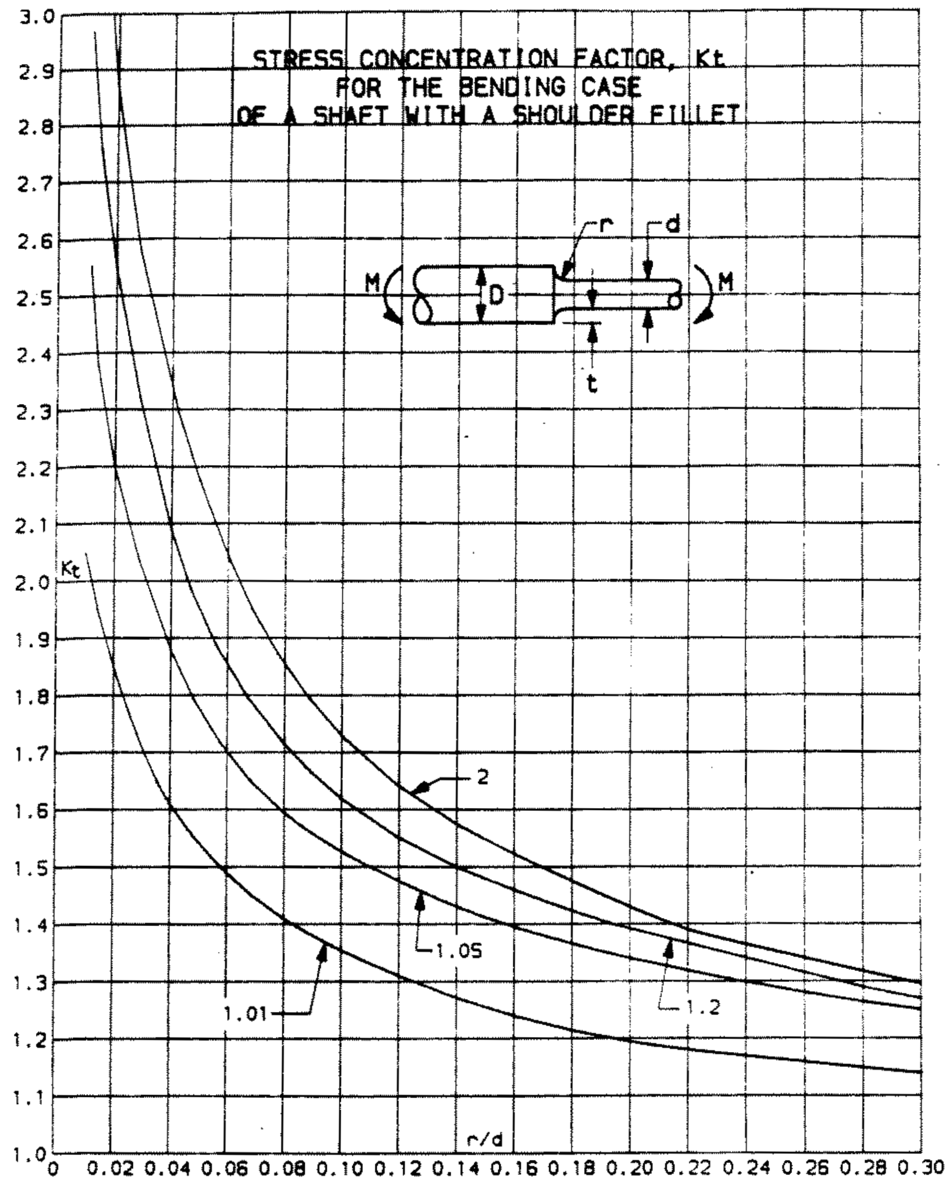

FIGURE 1

(Extracted from: "Stress Concentration Factors" by R. E. Peterson)[4]

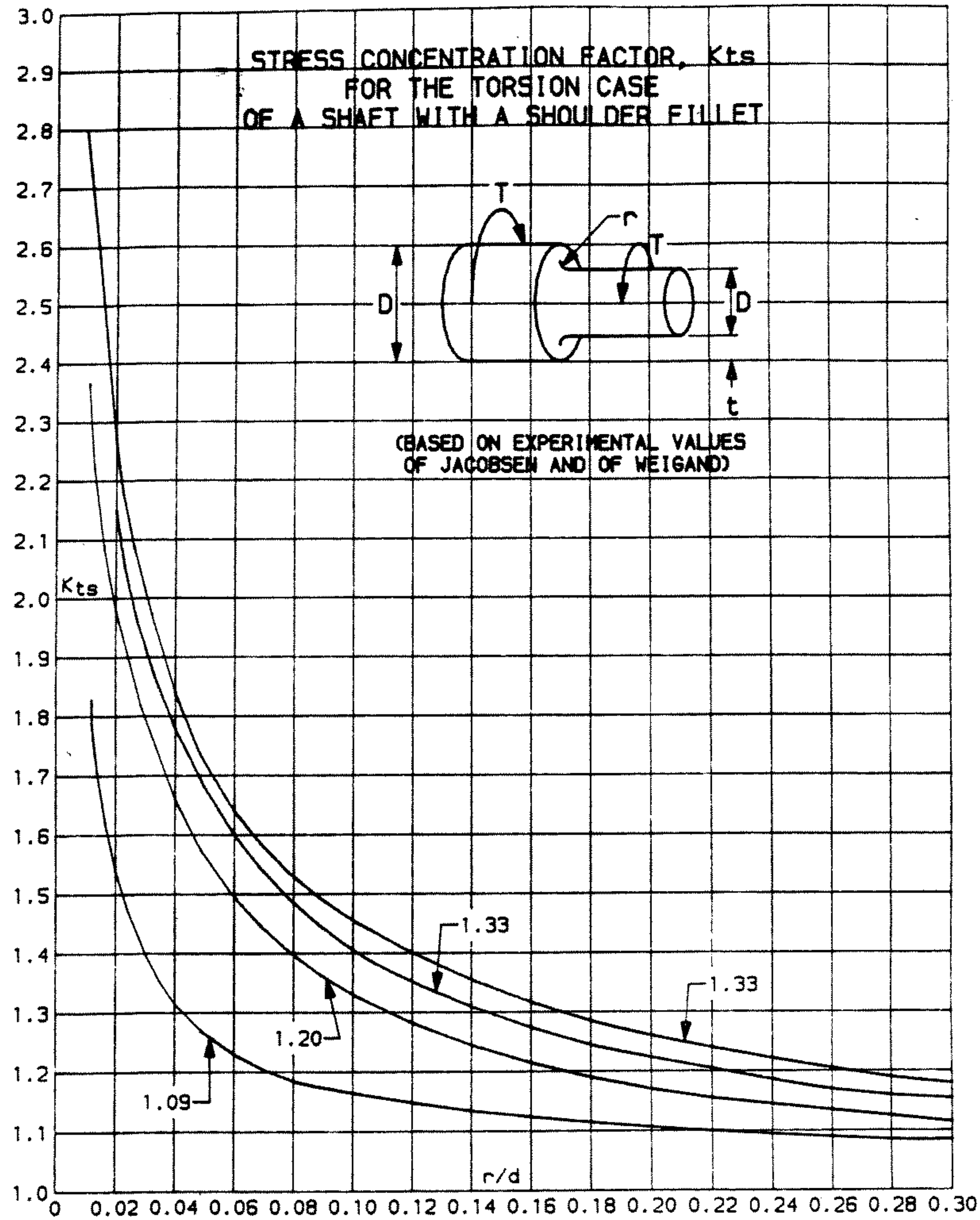

FIGURE 2

(Extracted from: "Stress Concentration Factors" by R. E. Peterson)[4]

The above design equation is based on the Soderberg diagram fatigue failure curve and the maximum shear stress theory for combined reversing bending and steady torsion stresses. For cases of non-reversing bending stresses (trunnions which rotate $\leq 90°$) this equation is conservative.

EXAMPLE DESIGN CALCULATIONS

The following example will demonstrate the comparison between AREA/AASHTO static equations and the design based on fatigue equations.

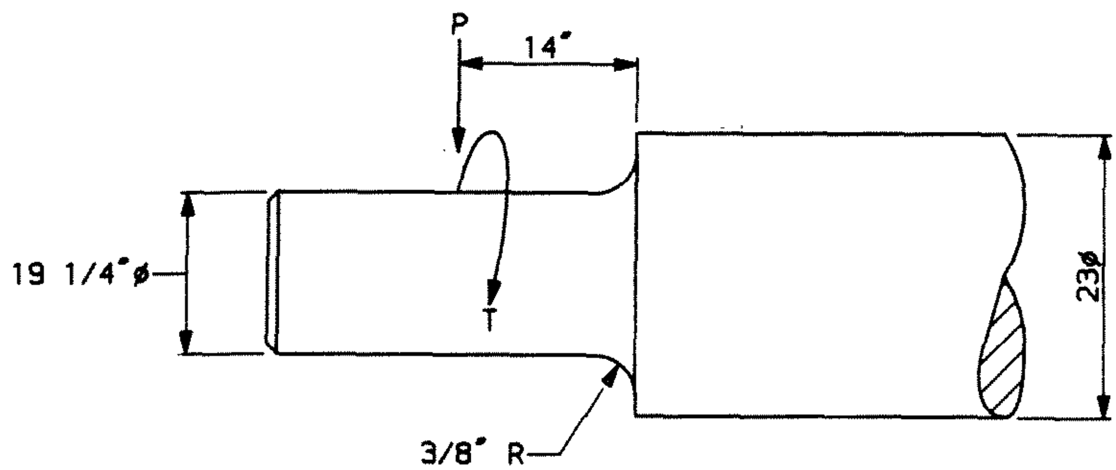

FIGURE 3

The above figure shows the shaft used in the following calculations. The example comes from the analysis of an existing design of a counterweight sheave shaft, using ASTM A668, Class D steel.

The load P was equal to 725 kips and the torque (due to starting friction) was equal to 1.3×10^6 lb-in. At the fillet,
$M = 725 \times 10^3$ lb $\times$ 14" $= 10.15 \times 10^6$ lb-in.; $T = 1.3 \times 10^6$ lb-in.

Using the AREA/AASHTO equations and K = 1:

$$\sigma = f_b = \frac{16 \times 1}{\pi(19.25)^3}\left(10.15 \times 10^6 + \sqrt{(10.15 \times 10^6)^2 + (1.3 \times 10^6)^2}\right)$$

$\sigma = 14{,}550\ psi < 15{,}000\ psi\ allowable;\ ok$

$$\tau = S_s = \frac{16 \times 1}{\pi\,(19.25)^3}\sqrt{(10.15 \times 10^6)^2 + (1.3 \times 10^6)^2}$$

τ = 7,300 psi < 7,500 psi allowable; ok

The conclusion would be the design is ok, based on AREA/AASHTO equations.

Using the previous fatigue design equation and first solving for the endurance limit.

S_{ut} = 75,000 psi; S_y = 37,500 psi (ASTM, A668, Class D)
$C_D = 0.872 \times (19.25)^{-0.1133} = 0.624$
$C_S = 0.875 - \frac{75{,}000}{800{,}000} = 0.781$
C_R = 1 (S_{ut} is a minimum strength and, therefore, do not adjust for reliability)
$S_e = 0.5 \times 75{,}000 \times 0.624 \times 0.781 \times 1$
S_e = 18,300 psi

The stress concentration factors for bending and torsion are found from the appropriate figure knowing first r/d and D/d.

$$\frac{r}{d} = \frac{0.375}{19.25} = 0.0195 \sim 0.02$$

$$\frac{D}{d} = \frac{23.0}{19.25} = 1.195 \sim 1.2$$

For Bending, Figure 1, K_t = 2.6

For Torsion, Figure 2, K_{ts} = 2.0

Modifying for Fatigue (q = 0.85, q_s = 0.87):

K_f = 1+0.85 (2.6-1) = 2.36

K_{fs} = 1+0.87 (2.0-1) = 1.87

Since we are analyzing a present design, rearrange the fatigue design equation to solve for n, the factor of safety based on "indefinite" life.

$$n = \frac{\pi d^3}{32} \times \frac{1}{\sqrt{\left(\frac{K_f M}{S_e}\right)^2 + \left(\frac{K_{fs} T}{S_y}\right)^2}}$$

$$n = \frac{\pi \, (19.25)^3}{32} \times \frac{1}{\sqrt{\left(\frac{2.36 \times 10.15 \times 10^6}{18{,}300}\right)^2 + \left(\frac{1.87 \times 1.3 \times 10^6}{37{,}500}\right)^2}}$$

n = 0.533 < 1.25

Redesigning for an n = 1.25, the required shaft size is d = 25.6"!

If we were doing a redesign, the first thing to change is the fillet radius, making it much larger so that K_f and K_{fs} would be less. Using a one-inch fillet radius, $K_t = 2.0$, $K_{ts} = 1.57$, $K_f = 1.85$, $K_{fs} = 1.50$.

Required diameter is:

$$d = \left\{\frac{32 \times 1.25}{\pi}\left[\left(\frac{1.85 \times 10.15 \times 10^6}{18{,}300}\right)^2 + \left(\frac{1.5 \times 1.3 \times 10^6}{37{,}500}\right)^2\right]^{\frac{1}{2}}\right\}^{\frac{1}{3}}$$

d = 23.6"

As you can see, the original design was inadequate for long fatigue life, although AREA/AASHTO equations say the design is satisfactory.

An estimate of the expected fatigue life of the original shaft was approximately 35,000 cycles of stress. Depending on the frequency of bridge opening, this present design could have prematurely failed by fatigue. [For 20 years of life, no more than five openings per day could be allowed!]

In this application, the final counterweight sheave shaft will be 25 inches small diameter, 26.5 inches larger diameter and a one-inch fillet radius at the shoulder. Part of the reason for the larger shaft was the change to roller bearings from the original plain journal bronze bearings.

A couple more points should be made here with regard to the design to prevent fatigue failure. 1) Even if there was no filleted shoulder where the shaft gets pressed into the counterweight sheave hub, there will still be a stress concentration in the shaft at the edge of the hub. For a hub length approximately equal to the shaft diameter, the K_t factor for bending can approach a value of 2, when the pressure due to the press fit is about equal to the nominal bending stress in the shaft outer surface[6]. 2) Older bridge designs made extensive use of plain journal bronze sleeve bearings. In non-self aligning plain sleeve bearings, the force distribution along the length is not uniform and tends to be concentrated nearer the inner edge of the bearing, thereby reducing the shaft bending moment and reducing the chance of fatigue failure. However, on many rehabilitations, the plain bearings are being replaced with self-aligning roller bearings. This type of bearing will cause the load to be concentrated at a larger moment arm and may, therefore, <u>increase</u> the chance of fatigue failure. Always check the design of the shaft carefully to be sure you are <u>not</u> worsening an already underdesigned situation.

<u>OPEN-GEARING DESIGN</u>

[The following will be limited to the design of open spur gearing, but procedures are similar for other types (Helical, bevel).]

AREA/AASHTO for gear tooth strength (as in shaft design), use equations which are based primarily on static bending stresses of the teeth. The equations are well founded, being based on the original Lewis equation (1892), but they have long since lost their usefulness, since AGMA equations take so many more factors into account.

In the latest AASHTO Standard Specification for Movable Highway Bridges (1988)[2], a sentence was added to Specification 2.6.12 (Strength of Gear Teeth): "Gear tooth design shall meet AGMA standards for surface durability (pitting resistance) and bending strength." The specification offers no further design guidance.

It is, therefore, the purpose of this paper to compare AREA/AASHTO equations to AGMA and briefly review the appropriate AGMA standard to the reader for open spur gearing design.

The AREA/AASHTO equation for 20° full-depth involute spur gears (AREA 6.5.19) is:

$$W - fsp\left(0.154 - \frac{0.912}{n}\right)\left(\frac{600}{600 + v}\right)$$

where W = allowable tooth load (lbs.)
f = effective face width (in.)
s = allowable unit stress (psi)
p = circular pitch (in.)
n = number of teeth
v = pitch line velocity (fpm)

The term in the first parentheses is the Lewis form factor and is given the symbol y by many references. The term in the second parentheses is the Barth velocity factor (also originated in the nineteenth century). This factor was based on tests of cast iron gears with cast teeth, probably with cycloidal profile (not involute).

The Lewis equation forms the basis for the AGMA equations in bending strength and fatigue. AGMA Standard 908[7] gives the equations for both gear *tooth bending strength (fatigue) and for surface durability (pitting and wear).*

Each is treated separately, and then the final gear design is based on the more critical of the two procedures.

Bending Strength (Fatigue):

$$S_t = \frac{W_t K_a}{K_v} \times \frac{P_d}{F} \times \frac{K_s K_m}{J}$$

Allowable Bending Stress Number:

$$S_t \leq \frac{S_{at} K_L}{K_T K_R}$$

These can be combined into an AGMA power rating equation:

$$P_{at} = \frac{n_p d K_v}{126{,}000 K_a} \times \frac{F}{P_d} \times \frac{J}{K_s K_m} \times \frac{S_{at} K_L}{K_R K_T}$$

From the previous equations:

S_t = bending stress number (psi)
W_t = transmitted tangential tooth load (lbs)
P_d = diametral Pitch (1/in)
F = net face width of narrowest gear (in)
J = geometry factor for bending
S_{at} = allowable bending stress number (psi)
n_p = pinion speed (RPM)
d = pitch diameter of pinion (in.)
K_a = application factor for bending
K_L = life factor for bending
K_m = load distribution factor for bending
K_R = reliability factor for bending
K_S = size factor for bending
K_T = temperature factor for bending
K_v = velocity factor for bending

The following equations or values for the above factors are appropriate for open gearing in movable bridges. Refer to AGMA Standard 908 for more complete figures and tables.

K_a = 1.25 (uniform power source, moderate shock load)
K_L = 1.0 (10^7 cycles of stress)
K_m = 1.156 + 0.0271 x F + 0.11 x F/d - 0.0000612 x F^2 (based on open gearing, adjusted at assembly)
K_R = 1.0 (for 99% reliability)
K_S = 1.1 (for large teeth, P_d < 6)
K_T = 1.0 (for gear temperatures < 250° F)

$$K_v = \left(\frac{60}{60+\sqrt{v}}\right)^{0.826}$$ (gear quality, $Q_v = 6$)

where v = pitch line velocity of gear (ft/min.)

$J = 0.312 \times e^{(-5.24/N)}$ (for tip loadings) or
$J = 0.461 \times e^{(-7.77/N)}$ (for highest single tooth contact)
where N = number of teeth

Tip loading gives a more conservative design and is recommended for open gearing with low gear quality ($Q_v = 6$).

$S_{at} = -274 + 167 \times H_B - 0.152 \times H_B^2$
for the range $160 \leq H_B = \leq 400$
(for steel gears; H_B = Brinell Hardness of teeth)
(For idler gears, use an S_{at} equal to 70% of this.)

Some other equations and relations are:

$v = \pi d\ n_p/12$ (ft/min)
where d = pitch diameter of pinion (in)
n_p = speed of pinion (RPM)

$$P_d = \pi/p_c = \frac{N_p}{d} = \frac{N_G}{D}$$

p = circular pitch of teeth (inch)
N = number of teeth on pinion/gear
d or D = pitch diameter of pinion or gear (inch)

$F \leq 3p$ (this is an AREA/AASHTO limit)

(AGMA gives no relation, but other references (3,5) use a range:

$$\frac{8}{P_d} \leq F \leq \frac{16}{P_d};$$

which would be approximately $2.5p_c \leq F \leq 5p_c$

Surface Durability (Pitting, Wear):

$$S_c = C_p \sqrt{\frac{W_t C_a}{C_v} \times \frac{C_s}{dF} \times \frac{C_m C_f}{I}}$$

Allowable Contact Stress Number:

$$S_c \leq S_{ac} \times \frac{C_L C_H}{C_T C_R}$$

These can be combined into an AGMA pitting resistance power rating equation:

$$P_{ac} = \frac{n_p F}{126{,}000} \times \frac{I C_v}{C_s C_m C_f C_a} \times \left(\frac{d S_{ac}}{C_p} \frac{C_L C_H}{C_T C_R} \right)^2$$

Where from the previous equations:

S_c = contact stress number (psi)
S_{ac} = allowable contact stress number (psi)
W_t, d, F, n_p as before
C_p = elastic coefficient $\sqrt{psi}$
C_a = application factor for pitting
C_f = surface condition factor
C_H = hardness ratio factor
C_L = life factor for pitting
C_m = load distribution factor for pitting
C_R = reliability factor for pitting
C_s = size factor for pitting
C_T = temperature factor for pitting
C_v = dynamic velocity factor for pitting
I = geometry factor for pitting

The following equations and values are appropriate for open gearing in movable bridges. Refer to AGMA 908 for more complete information.

$C_a \geq 1.0$ (AGMA gives no specific values)
$C_f \geq 1.0$ (Use greater than unity when there is poor tooth surface conditions)
$C_m = K_m$
C_R = 1.0 for 99% reliability
C_s = 1.0 (no values given by AGMA)
C_T = 1.0 (for gear temperatures < 250° F)
$C_v = K_v$
C_H = 1.0 if $\frac{H_B(Pinion)}{H_B(Gear)} \leq 1.2$ for a high hardness ratio, $C_H > 1$
C_L = 1.0 FOR 10^7 stress cycles
C_p = 2,300 for steel pinion - steel gear
C_p = 2,000 for steel pinion - cast iron gear
I = $\frac{M_G}{(8.65 \times M_G + 4.87)}$; ($for\ N_p \approx 18\text{-}20\ teeth$)
For $M_G > 10$, use the I value for $M_G = 10$.
$S_{ac} = 26{,}000 + 327 \times H_B$ (psi) in the range $160 \leq H_B \leq 400$

Yielding Criteria

AGMA also presents equations for cases of infrequent momentary overload (less than 100 stress cycles). For this situation, allowable yield strength properties are the determining criteria, rather than the fatigue strength of the gear material.

$$S_{ay} \times K_y \geq \frac{W_{max} K_a}{K_v} \times \frac{P_d}{F} \times \frac{K_s K_m}{J K_f}$$

W_{max} = maximum peak tangential tooth load (lbs)
K_y = yield strength factor
use K_y = 0.75 (Industrial Practice)
or K_y = 0.50 (more conservative)
S_{ay} = 482 x H_B - 32,800 (psi) in the range $160 \leq H_B \leq 410$
K_f = stress correction factor for 17-20 teeth pinion, $K_f \approx 1.3$

Sample Calculations

To demonstrate the disparity in using the AREA/AASHTO equation for spur gear design/analysis, rather than the AGMA equations, the following example is presented:

In a particular vertical lift bridge rehabilitation project, the open hoist gearing was showing signs of excessive wear, after a relatively short life (about 20 years).

An analysis revealed that the gearing design using the AREA/AASHTO equation allowed for a much higher horsepower rating than using appropriate AGMA equations for fatigue and wear.

The input pinion driving the hoisting rope drums had the following specifications:

16 teeth, P_c = 2", d = 10.186", F = 5.5", 20° full-depth involute teeth, Material: ASTM A235, Class C1 (new specification: A668, Class C) forged steel. RPM ≈ 28.4 meshing with a 44-tooth cast steel gear.

Using the AREA/AASHTO equation and allowable stress, S (allowable) = 20,000 psi (AREA 6.5.19)

$$v = \frac{\pi d n}{12} = \pi \times 10.186 \times \frac{28.4}{12} = 76\, fpm$$

$$W = 5.5'' \times 20{,}000\, psi \times 2'' \left(0.154 - \frac{0.912}{16}\right)\left(\frac{600}{676}\right)$$

W = 18,940 lbs. (allowable tooth load)

Allowable horsepower would then be:

$$hp = \frac{W \times \frac{d}{2} \times n\,(RPM)}{63{,}000} = \frac{18{,}940 \times \frac{10.186}{2} \times 28.4}{63{,}000}$$

hp = **43.5** (AREA/AASHTO Power Rating)

For this same pinion, the AGMA power rating equation for bending fatigue gives the following result.

For A668, Class C, an average Brinell Hardness is H_B = 160.

Therefore, $S_{at} = -274 + 167 \times 160 - 0.152 \times 160^2 = 22{,}550$ psi

The other factors are:

$$K_v = \left(\frac{60}{60 + \sqrt{v}}\right)^{0.826} = 0.894,\ for\ v = 76\,fpm$$

$$J = 0.461e^{(-7.77/16)} = 0.284$$

$$K_m = 1.156 + 0.0271 \times 5.5 + 0.11 \times \frac{5.5}{10.186} - 6.12 \times 10^{-5}\,(5.5)^2 = 1.36$$

K_a = 1.25
K_s = 1.1
$K_L = K_R = K_T$ = 1.0

$$P_{at} = \frac{28.4 \times 10.186 \times 0.894}{126{,}000 \times 1.25} \times \frac{5.5}{\pi/2} \times \frac{0.284}{1.1 \times 1.36} \times \frac{22550 \times 1}{1 \times 1}$$

P_{at} = **24.6 hp** (based on bending fatigue)

This value is approximately 56% of the AREA/AASHTO allowable horsepower.

The AGMA power rating equation based on pitting resistance (surface durability and wear gives the following result:

$$M_G = \frac{N_G}{N_p} = \frac{44}{16} = 2.75$$

$$I = \frac{2.75}{(8.65 \times 2.75 + 4.87)} = 0.096$$

S_{ac} = 26,000 x 327 x 160 = 78,300 psi
Cp = 2,300 $\sqrt{psi}$
$C_m = K_m = 1.36$; $C_v = K_v = 0.894$
$C_a = C_f = C_R = C_s = C_H = C_L = C_T = 1.0$

$$P_{ac} = \frac{28.4 \times 5.5}{126{,}000} \times \frac{0.096 \times 0.894}{1 \times 1.36 \times 1 \times 1} \left(\frac{10.186 \times 78{,}300}{2{,}300} \times \frac{1 \times 1}{1 \times 1} \right)^2$$

P_{ac} = 9.41 hp

This is about 22% of the AREA/AASHTO allowable horsepower.

The AGMA equation for overload yielding gives the following results:

S_{ay} = 482 x 160 - 32,800 = 44,300 psi
$K_y = 0.75$; $K_f = 1.3$

Solving from W_{max}:

$$W_{max} = \frac{S_{ay} \times K_y \times K_u \times F \times J \times K_f}{K_a \times P_d \times K_s \times K_m}$$

Using the previous K values:

$$W_{max} = \frac{44{,}300 \times 0.75 \times 0.894 \times 5.5 \times 0.284 \times 1.3}{1.25 \times \frac{\pi}{2} \times 1.1 \times 1.36}$$

W_{max} = 20,500 lbs.

This value compares more closely with the AREA/AASHTO value of W = 18,940 lbs. However, AGMA says that this high a force should only be infrequent.

The previous calculations should show that the AGMA equations give a much more conservative allowable horsepower rating, both for fatigue and surface durability.

The original gearing on this job was designed using AREA equations and a 1.5 overload factor, the pinion tooth load being based on 30 hp. The input shaft to the pinion, however, transmitted 60 hp, since the pinion drove two gears (one through an idler pinion). Therefore, the pinion teeth were loaded twice each revolution contributing to premature wear. Also, AREA equations do not treat an idler any different, even though the teeth are subjected to reversed bending stresses every revolution.

For the rehabilitation, the gearing on the hoisting rope drum drive were redesigned using AGMA equations for wear and fatigue strength. Two double pinion shafts were used so that there was no idler and no pinion which was loaded twice per revolution. Material strength and face widths were increased in order to retain original circular pitches and center distances.

SUMMARY

This paper has presented recommended design equations to be used when designing rotating shafts, trunnions, and open spur gearing for movable bridges. Using only AREA/AASHTO equations, which account for static stresses alone, may lead to premature fatigue failure for shafts and premature wear or fatigue failure for spur gearing. Critical bridge machinery parts, such as these, must be carefully designed, using appropriate fatigue design equations, endurance limits, stress concentration factors, and AGMA gear equations and geometry factors.

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