



American
Gear Manufacturers
Association

ANSI/AGMA 6001-F19
Revision of ANSI/AGMA 6001-E08

American National Standard

Design and Selection of Components for Enclosed Gear Drives

ANSI/AGMA 6001-F19

**American
National
Standard*****Design and Selection of Components for Enclosed Gear Drives***

ANSI/AGMA 6001-F19

(Revision of ANSI/AGMA 6001-E08)

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ABSTRACT

This standard outlines the basic practices for the design and selection of components, other than gearing, for use in commercial and industrial enclosed gear drives.

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Contents

Foreword	v
1 Scope.....	1
1.1 Exceptions.....	1
1.2 Intended use.....	1
2 Normative references	1
3 Symbols and definitions.....	2
3.1 Definitions.....	2
3.2 Symbols.....	2
4 Design conditions	5
4.1 Load spectrum analysis	5
4.2 Momentary peak loads.....	6
4.3 System analysis	6
5 Shafts.....	6
5.1 Design criteria	6
5.2 Fatigue safety factor.....	7
5.3 Peak load safety factor.....	7
5.3.1 Allowable stress to yield strength factor, F_{ya}	8
5.3.2 Peak load factor, F_p	8
5.4 Calculated stresses	8
5.4.1 The general case.....	10
5.4.2 The simplified case.....	12
5.5 Material requirements	14
5.5.1 Estimated properties of steel.....	14
5.6 Modified fatigue strength.....	15
5.6.1 Surface finish factor, k_a	16
5.6.2 Size factor, k_b	16
5.6.3 Reliability factor, k_c	17
5.6.4 Temperature factor, k_d	17
5.6.5 Life factor, k_e	18
5.6.6 Modifying factor for stress concentration, k_f	18
5.6.7 Miscellaneous effects factor, k_g	22
5.6.8 Permissible number of peak load cycles.....	23
5.7 Deflection	23
5.7.1 Permissible deflection.....	24
6 Keys and interference fits	25
6.1 Size and tolerances.....	25
6.2 Limitations	25
6.3 Keyed interference fit	26
6.4 Allowable compressive stress.....	26
6.5 Allowable shear stress	27
6.6 Allowable torque based on compressive stress calculation.....	27
6.7 Allowable torque based on shear stress calculation	27
6.8 Allowable torque.....	27
6.9 Keyless interference fit.....	27

6.10	Reversing loads	28
7	Bearings.....	29
7.1	Roller and ball bearing selection criteria	29
7.1.1	Reliability	29
7.1.2	Life adjustment factors	29
7.2	Sleeve bearing selection criteria	29
7.2.1	Boundary and mixed film regimes	30
7.2.2	Hydrodynamic regime.....	30
8	Housings.....	31
8.1	Housing construction and styles	31
8.2	Housing materials	32
8.3	Housing rigidity and strength.....	32
8.4	Housing accessory provisions.....	33
8.4.1	Inspection covers.....	33
8.4.2	Fill, vent, and drain provisions	33
8.4.3	Lifting provisions.....	33
9	Threaded fasteners	33
9.1	Fastener preload tensile stress, σ_M	34
9.2	Fastener torque, T_f	35
9.3	Allowable fastener tensile stress, σ_{fa}	35
9.3.1	Allowable tensile load, F_A	36
9.4	Engagement length	36
9.5	Locking devices for fasteners.....	36
10	Miscellaneous components	36
10.1	Shims	36
10.2	Gaskets	37
10.3	Oil seals.....	37
10.4	Breathers.....	37
10.5	Expansion chambers.....	38
10.6	Oil level indicators	38
10.7	Bearing retainers	38
10.8	Grease retainers	38
10.9	Dowels and pins.....	38
10.9.1	Dowels and pins used for positive location	38
10.9.2	Dowels and pins used to prevent movement	38
10.10	Spacers	39
10.11	Seal retainers	39
10.12	Special tools	39
10.13	Monitoring.....	39
10.13.1	Temperatures; sump, inlet, bearing race(s)	39
10.13.2	Pressures; inlet.....	39
10.13.3	Oil level	39
10.13.4	Chip or ferrous debris monitor	39
10.13.5	Flow	40
10.13.6	Vibration.....	40

Annexes

Annex A (informative) Allowable stresses for typical key materials	41
Annex B (informative) Material properties for typical threaded fasteners	43
Annex C (informative) Previous method – shaft design	45
Annex D (informative) Sample problems – transmission shaft design	48
Annex E (informative) Shaft deflection	55
Annex F (informative) Sample problems – keys	60
Annex G Bibliography	66

Figures

Figure 1 – Design criteria	7
Figure 2 – Cyclic loading	10
Figure 3 – Stress convention showing orbiting element	11
Figure 4 – Surface finish factor, k_a [2]	16
Figure 5 – Size factor, k_b [2]	16
Figure 6 – Reliability factor, k_c [1] [2]	17
Figure 7 – Notch sensitivity – through hardened steel, q [2]	19
Figure 8 – Theoretical stress concentration factor in bending for a circular shaft with a square shoulder, K_t (nominal stress is calculated at diameter d_{she}) [7]	20
Figure 9 – Theoretical stress concentration factor in bending for a circular shaft with a u-notch, K_t (nominal stress is calculated at diameter d_{she}) [7]	21
Figure 10 – Theoretical stress concentration factor in bending for a circular shaft with a radial hole, K_t (based on full section without considering hole) [7]	22
Figure 11 – Key nomenclature	26
Figure 12 – Variation of coefficient of friction versus the bearing parameter	30
Figure 13 – Cast housings	31
Figure 14 – Fabricated housings	32
Figure 15 – Fastener grip requirement	34

Tables

Table 1 – Symbols used in equations	2
Table 2 – Modifying factor for stress concentration, k_t – typical values for keyways in solid round steel shafts ¹⁾ [8]	22
Table 3 – Typical allowable misalignment per bearing type	24
Table 4 – Joint stiffness factor, K_j	36

Foreword

[The foreword, footnotes and annexes, if any, in this document are provided for informational purposes only and are not to be construed as a part of AGMA Standard 6001-F19, *Design and Selection of Components for Enclosed Gear Drives*.]

AGMA 260.02 was approved by the AGMA membership on February 1, 1973 and issued in January of 1974. It consolidated with minor revision, information contained in the following superseded AGMA Standards:

AGMA 255.02 (November 1964), *Bolting (Allowable Tensile Stress) for Gear Drives*;

AGMA 260.01 (March 1953), *Shafting – Allowable Torsional and Bending Stresses*;

AGMA 260.02, also incorporated allowable stresses for keys;

AGMA 265.01, *Bearings – Allowable Loads and Speeds*.

The purpose of ANSI/AGMA 6001-C88, as a replacement for AGMA 260.02, was to establish a common base for the design and selection of components for the different types of commercial and industrial gear drives.

ANSI/AGMA 6001-C88 was expanded to include a generalized shaft stress equation which included hollow shafting, miscellaneous components, housings, and keyway stress calculations. All design considerations were revised to allow for 200 percent peak load for helical, spiral bevel, spur and herringbone gearing, and 300 percent peak load for wormgearing. The bearing section was updated to include consideration of life adjustment factors, bearing lives other than 5000 hours and reliability levels other than L_{10} .

During the preparation of ANSI/AGMA 6001-C88, a considerable amount of time was spent on the shaft design section in an effort to include the most recent theories on shaft stresses and material characteristics. The standard included the existing practice for shaft design, and for reference purposes, appendix C included a description of, and excerpts from, ANSI/ASME B106.1M, *Design of Transmission Shafting*, published in 1985 [1].

ANSI/AGMA 6001-C88 was approved by the membership in May 1988 and approved as an American National Standard on June 24, 1988.

ANSI/AGMA 6001-D97 was expanded to include more recent theories on shaft design and analysis. Also, equations for shaft deformation were added. ANSI/AGMA 6001-D97 was approved by the membership in October 1996 and approved as an American National Standard on August 7, 1997.

ANSI/AGMA 6001-E08 was updated as required by ANSI practices. In the process, several improvements and simplifications were included. Minimum material requirements were added for shaft material and the shaft deflection clause was moved to an annex. Also, the clauses on keys and fasteners were revised to reflect current practices with higher allowable stresses in some cases.

The first draft of ANSI/AGMA 6001-E08 was made in August 2002. It was approved by the AGMA membership in September 2008. It was approved as an American National Standard on December 19, 2008.

ANSI/AGMA 6001-F19 was reviewed and updated in accordance with ANSI practices, with general updates and corrections that accumulated since publication of E08. In particular, Clause 8, Housings, was expanded to include graphics and discussion of stress analysis. Also, Clause 10, Miscellaneous components, was significantly upgraded in content.

The first draft of ANSI/AGMA 6001-F19 was made in April 2018. It was approved by the AGMA membership in December 2018. It was approved as an American National Standard on January 18, 2019.

Suggestions for improvement of this standard will be welcome. They should be sent to tech@agma.org.

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American National Standard –

Design and Selection of Components for Enclosed Gear Drives

1 Scope

This standard provides an acceptable practice for the design and selection of components for enclosed gear drives. Fundamental equations provide for the proper sizing of shafts, keys, and fasteners based on stated allowable stresses. Other components are discussed in a manner to provide an awareness of their function or specific requirements. This standard applies to the following types of commercial and industrial enclosed gear drives, individually or in combination: spur, helical, herringbone, double helical, or bevel gearing in single or multiple stages. Bevel gear drives may include shaft angles other than 90 degrees.

1.1 Exceptions

The equations in this standard are not applicable when gear drives are subjected to conditions that introduce vibrations of a sufficient level to affect the performance of the gear drive and may result in unpredictable fatigue failure.

The procedure for design or selection of the specific gear components is varied and complex and is beyond the scope of this standard. Designers must refer to the specific rating or enclosed drive standards for this aspect of drive design.

1.2 Intended use

The equations and values presented provide a general approach to design. Deviations from the methods and values stated in this standard may be made when justified by experience, testing, or more specific analysis. This standard is intended for use by experienced gear designers capable of selecting reasonable values based on their knowledge of the performance of similar designs and the effect of such items as lubrication, deflection, manufacturing tolerances, metallurgy, residual stresses, and system dynamics. It is not intended for use by the engineering public at large.

2 Normative references

The following documents contain provisions which, through reference in this text, constitute provisions of this American National Standard. At the time of publication, the editions were valid. All standards are subject to revision, and parties to agreements based on this standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below.

ANSI B17.1, *Keys and Keyseats*

ANSI B17.2, *Woodruff Keys and Keyseats*

AGMA 927-A01, *Load Distribution Factors – Analytical Methods for Cylindrical Gears*

AGMA 938-A05, *Shot Peening of Gears*

ANSI/AGMA 1012-G05, *Gear Nomenclature, Definitions of Terms with Symbols*

ANSI/AGMA 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*

ANSI/AGMA 9003-C17, *Flexible Couplings – Keyless Fits*

ANSI/AGMA 9005-F16, *Industrial Gear Lubrication*

ANSI/AGMA ISO 6336-6-A08, *Calculation of Load Capacity of Spur and Helical Gears – Part 6: Calculation of Service Life Under Variable Load*

ANSI/AGMA 9002-C14, *Bores and Keyways for Flexible Couplings (Inch Series)*

DIN 6885-1:1968, *Drive Type Fastenings without Taper Action; Parallel Keys, Keyways, Deep Pattern*

3 Symbols and definitions

The symbols and definitions used in this standard may differ from those in other AGMA standards. The user should not assume that familiar symbols can be used without a careful study of the applicable section(s) and equation(s).

3.1 Definitions

The terms used, wherever applicable, conform to the following standard:

ANSI/AGMA 1012-G05, *Gear Nomenclature, Definitions of Terms with Symbols*

3.2 Symbols

The symbols used in this standard are shown in Table 1.

NOTE: The symbols and terms contained in this document may vary from those used in other AGMA standards. Users of this standard should assure themselves that they are using these symbols and terms in the manner indicated herein.

Table 1 – Symbols used in equations

Symbol	Term	Units	First referenced
A	Coefficient	--	Fig 4
A_c	Minimum compressive area of key in shaft or hub	in ²	Eq 49
A_R	Fit holding capacity	lb	Eq 52
A_s	Shear area	in ²	Eq 50
A_{ts}	Tensile strength area of fastener	in ²	Eq 60
B	Coefficient	--	Fig 4
b_k	Width of the key	in	Eq 41
c	Coefficient	--	Eq 37
D_f	Fastener nominal diameter	in	Eq 57
D_{fm}	Fastener nominal diameter	mm	Eq 58
d_{he}	Outside diameter of hub	in	Eq 55
d_{hi}	Inside diameter of hub	in	Eq 55
d_{she}	Outside diameter of the shaft section being analyzed	in	Eq 6
d_{shi}	Shaft inside diameter	in	Eq 6
d_{sho}	Outside diameter adjacent to the shaft section being analyzed	in	Fig 8
E_H	Modulus of elasticity for hub material	lb/in ²	Eq 53
E_S	Modulus of elasticity for shaft material	lb/in ²	Eq 53
F_A	Allowable tensile load	lb	Eq 60
F_M	Fastener tensile preload	lb	Eq 57

(continued)

Table 1 (continued)

Symbol	Term	Units	First referenced
F_p	Peak load factor	--	Eq 5
F_{Sf}	Fatigue safety factor	--	Eq 1
F_{Sp}	Peak load safety factor	--	Eq 2
F_{ya}	Allowable stress to yield strength factor	--	Eq 5
H_B	Brinell hardness number	HBW	Eq 30
h	Radial step	in	Fig 8
h_k	Height of the key	in	Eq 42
I	Actual or minimum possible interference fit	in	Eq 53
i	Number of keys	--	Eq 45
K	Calculation constant for shear stress	--	Eq 8
K_J	Joint stiffness factor	--	Eq 59
K_t	Theoretical stress concentration factor in bending	--	Eq 38
K_{tc}	Fastener torque coefficient	--	Eq 58
k	Fatigue strength modification factor	--	Eq 34
k_a	Surface finish factor	--	Eq 35
k_b	Size factor	--	Eq 35
k_c	Reliability factor	--	Eq 35
k_d	Temperature factor	--	Eq 35
k_e	Life factor	--	Eq 35
k_f	Modifying factor for stress concentration	--	Eq 35
k_g	Miscellaneous effects factor	--	Eq 35
L	Length of engagement	in	Eq 52
l_g	Length of fastener	in	Clause 9
l_{tr}	Bearing length of the key	in	Eq 44
M	Bending moment	lb-in	Eq 7
m	Coefficient	--	Eq 37
N	Number of stress cycles	--	Eq 37
N_{fo}	Permissible number of momentary peak load cycles	--	Eq 39
n_f	Fastener thread pitch	in	Eq 57
P_H	Pressure at common shaft/hub interference	lb/in ²	Eq 52
P_x	Axial force	lb	Eq 10
q	Notch sensitivity	--	Eq 38
R	Reliability (survival rate)	--	Eq 36
Ra	Surface finish	μin	Fig 4
R_e	Tensile yield strength of the material	lb/in ²	Eq 46
r	Notch radius, fillet radius, hole radius	in	Fig 7
T	Torque	lb-in	Eq 6

(continued)

Table 1 (continued)

Symbol	Term	Units	First referenced
T_a	Allowable torque based on the lesser of T_C and T_s	lb-in	6.8
T_C	Allowable torque based on allowable compressive stress	lb-in	Eq 49
T_f	Fastener torque	lb-in	Eq 58
T_R	Torque transmitted due to interference fit	lb-in	Eq 51
T_s	Allowable torque based on allowable shear stress	lb-in	Eq 50
t_k	Shaft keyway depth	in	Eq 42
V	Transverse shear force	lb	Eq 8
θ	Angular position of shaft element	radians	Fig 3
θ_c	Critical stress angle	radians	Eq 28
μ	Coefficient of friction	- -	Eq 52
ρ_H	Poisson's ratio for hub material	- -	Eq 53
ρ_S	Poisson's ratio for shaft material	- -	Eq 9
σ_a	Alternating component of stress	lb/in ²	Eq 12
σ_{ax}	Alternating axial normal stress	lb/in ²	Eq 13
σ_{ay}	Alternating radial normal stress	lb/in ²	Eq 13
σ_{az}	Alternating tangential normal stress	lb/in ²	Eq 13
σ_b	Axial normal stress due to bending	lb/in ²	Eq 7
σ_e	σ_f at 10^6 stress cycles	lb/in ²	Eq 37
σ_f	Modified fatigue strength (endurance limit)	lb/in ²	Eq 3
σ_{fa}	Allowable fastener tensile stress	lb/in ²	Eq 59
σ_{fe}	Basic fatigue strength (endurance limit) of polished, unnotched test specimen in reverse bending	lb/in ²	Eq 32
σ_M	Fastener preload tensile stress	lb/in ²	Eq 56
σ_m	Mean component of stress	lb/in ²	Eq 11
σ_{max}	Maximum stress	lb/in ²	Fig 2
σ_{min}	Minimum stress	lb/in ²	Fig 2
σ_{mx}	Mean axial normal stress	lb/in ²	Eq 14
σ_{my}	Mean radial normal stress	lb/in ²	Eq 14
σ_{mz}	Mean tangential normal stress	lb/in ²	Eq 14
σ_p	Axial normal stress due to tension or compression	lb/in ²	Eq 10
σ_{sc}	Allowable compressive stress	lb/in ²	Eq 46
σ_{tx}	Total axial normal stress	lb/in ²	Eq 15
σ_{ty}	Total radial normal stress	lb/in ²	Eq 15
σ_{tz}	Total tangential normal stress	lb/in ²	Eq 15
σ_u	Ultimate tensile strength	lb/in ²	Eq 30

(continued)

Table 1 (concluded)

Symbol	Term	Units	First referenced
σ_y	Tensile yield strength	lb/in ²	Eq 3
$\bar{\sigma}_a$	Von Mises alternating stress	lb/in ²	Eq 3
$\bar{\sigma}_m$	Von Mises mean stress	lb/in ²	Eq 3
$\bar{\sigma}_o$	Equivalent uniaxial stress under peak loading	lb/in ²	Eq 39
$\bar{\sigma}_{total}$	Von Mises total stress	lb/in ²	Eq 5
τ_{at}	Alternating torsional shear stress	lb/in ²	Eq 20
τ_{axy}	Alternating radial shear stress	lb/in ²	Eq 13
τ_{ayz}	Alternating axial shear stress	lb/in ²	Eq 13
τ_{azx}	Alternating tangential shear stress	lb/in ²	Eq 13
τ_{mt}	Mean torsional shear stress	lb/in ²	Eq 19
τ_{mxy}	Mean radial shear stress	lb/in ²	Eq 14
τ_{myz}	Mean axial shear stress	lb/in ²	Eq 14
τ_{mzx}	Mean tangential shear stress	lb/in ²	Eq 14
τ_{ps}	Allowable shear stress	lb/in ²	Eq 48
τ_t	Torsional shear stress	lb/in ²	Eq 6
$\tau_{t \max}$	Maximum torsional shear stress	lb/in ²	Eq 19
$\tau_{t \min}$	Minimum torsional shear stress	lb/in ²	Eq 19
τ_{test}	Test shear stress to find θ_c	lb/in ²	Eq 27
τ_{txy}	Total radial shear stress	lb/in ²	Eq 15
τ_{tyz}	Total axial shear stress	lb/in ²	Eq 15
τ_{tzx}	Total tangential shear stress	lb/in ²	Eq 15
τ_v	Shear stress due to shear force	lb/in ²	Eq 8
ϕ	Key load share factor	- -	Eq 49

4 Design conditions

This standard should be used in conjunction with appropriate current AGMA standards. When the operating conditions are known, each component of the drive shall be designed to meet or exceed those conditions. When operating conditions are not known, all load carrying components of the drive shall be designed to support the stated mechanical rating of the drive for continuous duty based on a unity service factor (1.0). External loads shall be considered as acting in directions and rotations producing the most unfavorable stresses unless more specific information is available. Peak loads shall be accounted for in the design.

For enclosed drives designed to operate under specific conditions such as load, speed, duty cycle and life, components shall be selected accordingly.

4.1 Load spectrum analysis

It is recommended that the cumulative fatigue damage criteria proposed by Miner's Rule similar to the method in ANSI/AGMA ISO 6336-6-A08 may be employed to evaluate the effects of variable loading on the life of components.

4.2 Momentary peak loads

This standard is based on an allowable momentary mechanical peak load rating of 200 percent (mechanical peak load rating multiplied by 2.0) for spur, helical, herringbone and bevel gear drives and an allowable momentary mechanical peak load rating of 300 percent (mechanical peak load rating multiplied by 3.0) for wormgear drives. Allowable momentary peak loads for a combination of gear types within one gear drive should be consistent through all the components of the gear drive, whether it is 200% or 300%. A limited number of peak stress cycles are allowed, typically less than 10 000 for spur, helical, herringbone, and bevel gear drives and 25 000 for wormgear drives.

Frequency and duration of overloads shall be considered when designing and selecting components. If the overload exceeds the allowable peak load rating and the frequency or duration of the overload becomes significant (greater than 100 cycles during the design life), the designer should consider a cumulative fatigue analysis such as Miner's Rule.

If the overload exceeds the allowable peak load rating and the frequency or duration of the overload is less than 100 cycles during the design life, the conditions should be evaluated to assure that the yield strength of any component is not exceeded.

4.3 System analysis

This standard assumes that within the operating speed range, the system of connected rotating parts is compatible and free from critical speeds and torsional or other types of vibrations, no matter how induced.

The gear drive designer or manufacturer is not responsible for the system analysis unless this provision is clearly identified by contractual agreement.

5 Shafts

This clause covers a stress analysis procedure applicable to cylindrical steel shafts used in conjunction with, and as a part of, enclosed gear drives. This analysis may or may not be applicable to other materials or other types of machinery.

Shaft stress is but one consideration in the design of shafting. A shaft must have proper radial, axial, and torsional stiffness to limit deflections to acceptable levels and to avoid unwanted vibratory motion.

The equations and allowable stresses shown in Annex C have been accepted practice for shaft design that have been included in AGMA standards and used successfully for many years.

5.1 Design criteria

Shafts must pass two stress analysis tests to be considered adequately designed. First, they must be designed to resist fatigue failure due to cyclic loading over their intended life. All operating loads, including momentary peak loads, shall be considered and a Miner's Rule analysis similar to the method in ANSI/AGMA ISO 6336-6-A08 may be required to properly account for the different stress states. See 5.6.8.

$$F_{Sf} \geq 1.0 \quad (1)$$

where

F_{Sf} is fatigue safety factor.

Second, they shall be designed to withstand momentary peak loads without distress or permanent deformation.

$$F_{Sp} \geq 1.0 \quad (2)$$

where

F_{Sp} is peak load safety factor.

The requirement for both of these criteria is to compare the stress condition for the given shaft geometry and loading to properties of the shaft material. Each stress condition applied to the shaft shall be checked

with Equation 1 and Equation 2 such that $F_{Sf} \geq 1.0$ and $F_{Sp} \geq 1.0$. See Figure 1. Where nominal or estimated material properties are used, a factor of safety, F_{Sf} , greater than 1.0 is recommended.

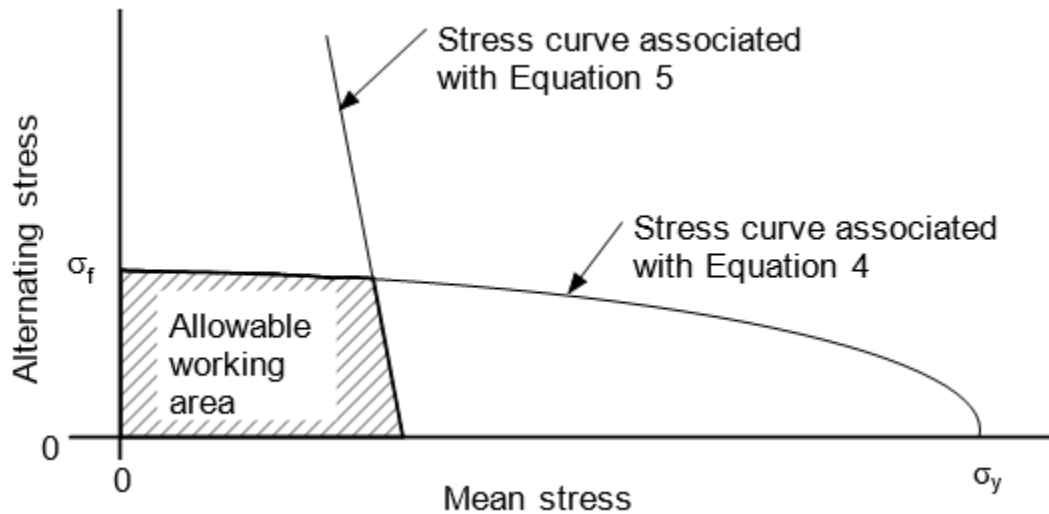


Figure 1 – Design criteria

In selecting a value for safety factor, the consequence of failure shall be considered. If the consequence of failure is high, a safety factor considerably greater than unity shall be used. If the consequence of failure is minimal, a safety factor closer to unity may be used.

5.2 Fatigue safety factor

The elliptic equation has been selected for analysis of the fatigue failure mode. It is found in references [1], [2] and [3]. This equation is:

$$\left(\frac{\bar{\sigma}_a}{\sigma_f}\right)^2 + \left(\frac{\bar{\sigma}_m}{\sigma_y}\right)^2 = \left(\frac{1}{F_{Sf}}\right)^2 \quad (3)$$

where

- $\bar{\sigma}_a$ is Von Mises alternating stress, lb/in²;
- σ_f is modified fatigue strength, lb/in²;
- $\bar{\sigma}_m$ is Von Mises mean stress, lb/in²;
- σ_y is tensile yield strength, lb/in².

This equation can be rewritten to solve for the fatigue safety factor.

$$F_{Sf} = \frac{1}{\left[\left(\frac{\bar{\sigma}_a}{\sigma_f}\right)^2 + \left(\frac{\bar{\sigma}_m}{\sigma_y}\right)^2\right]^{0.5}} \quad (4)$$

For the design to be considered acceptable for fatigue condition, the resulting fatigue safety factor, F_{Sf} , must be equal to or greater than 1.0.

5.3 Peak load safety factor

The following peak load analysis equation is used to solve for the peak load safety factor:

$$F_{Sp} = \frac{F_{ya} \sigma_y}{F_p \bar{\sigma}_{total}} \quad (5)$$

where

F_{ya} is allowable stress to yield strength factor, see 5.3.1;

F_p is peak load factor, see 5.3.2;

$\bar{\sigma}_{total}$ is Von Mises total stress, lb/in².

CAUTION: Equation 5 is based on a ductile material. For purposes of this standard, a material is considered ductile if the tensile elongation of the core material is at least 10%. For nonductile materials, the effects of stress concentration should be considered. See 5.6.6.

If $\bar{\sigma}_{total}$ includes stresses which are not a function of load, such as stress resulting from the weight of components or stress resulting from shrink fit of components, F_{Sp} may be conservative. Considerations may be given to only applying F_p to those stresses of $\bar{\sigma}_{total}$ that are load related.

For the design to be considered acceptable for peak load condition, the resulting peak load safety factor, F_{Sp} , must be equal to or greater than 1.0.

The safety factors are to be chosen based on experience and engineering judgement.

5.3.1 Allowable stress to yield strength factor, F_{ya}

The allowable stress to yield strength factor provides conservatism over the stress resulting from expected peak load conditions and variations in the tensile yield strength. Values between 0.66 and 0.80 have traditionally been employed for this variable. Unless otherwise agreed upon, a value of 0.75 is recommended.

5.3.2 Peak load factor, F_p

The peak load factor accounts for momentary peak loads over the unity service factor load. In the absence of other known conditions, the following values for the peak load factor are to be used:

- For spur, helical, herringbone, and bevel gear drives, $F_p = 2.0$;
- For wormgear drives, $F_p = 3.0$.

If a multistage reducer incorporates either cylindrical or globoidal wormgearing in conjunction with other types of gearing, then one of two approaches may be used:

- $F_p = 3.0$, and the components and non-wormgearing (spur, helical, or bevel gearing) designed/selected to withstand 300% momentary load;
- $F_p = 2.0$, and the components designed/selected to withstand 200% momentary load. The manufacturer needs to indicate that the momentary load is limited to 200%.

5.4 Calculated stresses

There are four types of loading applied to shafting for the simplified case. These are torque (T), bending moments (M), shear forces (V), and axial tension or compression (P_x). The equations converting these loads to stresses are given in Equations 6 through 10. Tensile stresses shall be represented as positive values while compressive stresses shall be represented as negative values. All forces and stresses shall be those which would be present when the drive is loaded to its unity (1.0) service factor rating.

For torque:

$$\tau_t = \frac{16 T d_{she}}{\pi (d_{she}^4 - d_{shi}^4)} \quad (6)$$

where

τ_t is torsional shear stress, lb/in²;

T is torque, lb-in;

d_{she} is shaft outside diameter, in;

d_{shi} is shaft inside diameter, in.

For bending moment:

$$\sigma_b = \frac{32Md_{she}}{\pi(d_{she}^4 - d_{shi}^4)} \quad (7)$$

where

σ_b is axial normal stress due to bending, lb/in²;

M is bending moment, lb-in.

For shear force [4]:

$$\tau_v = \frac{4VK}{\pi(d_{she}^2 - d_{shi}^2)} \quad (8)$$

where

τ_v is shear stress due to shear force, lb/in²;

V is transverse shear force, lb.

and

$$K = \frac{(1 + 2\rho_s)(d_{she}^2 + 2.25d_{shi}^2)}{(1 + \rho_s)(d_{she}^2 + d_{shi}^2)} \quad (9)$$

where

ρ_s is the material's Poisson's ratio.

For a solid steel shaft, where $\rho_s = 0.3$, $K = 1.23$ and for a thin walled hollow steel shaft, K approaches 2.0.

For axial tension or compression:

$$\sigma_p = \frac{4P_x}{\pi(d_{she}^2 - d_{shi}^2)} \quad (10)$$

where

σ_p is axial normal stress due to tension or compression, lb/in²;

P_x is axial force, lb.

All of these stresses can have alternating and mean components. See Figure 2.

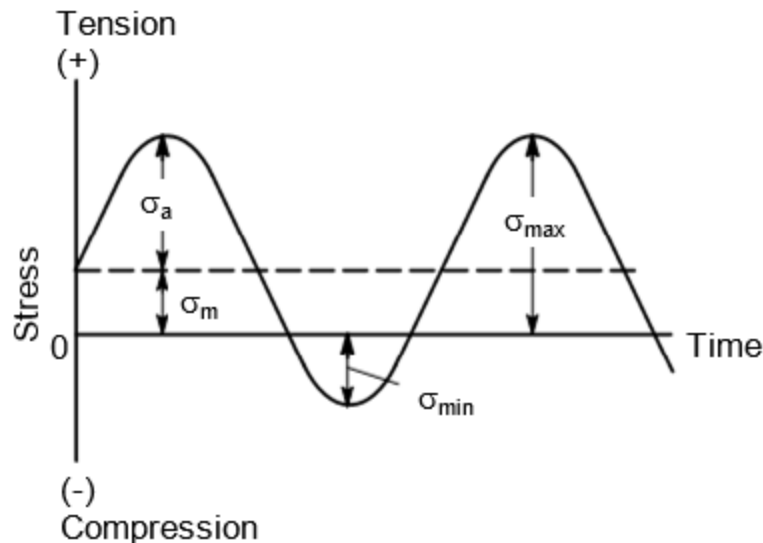


Figure 2 – Cyclic loading

Therefore:

$$\sigma_m = 0.5 (\sigma_{\max} + \sigma_{\min}) = \text{mean component of stress} \quad (11)$$

$$\sigma_a = 0.5 (\sigma_{\max} - \sigma_{\min}) = \text{alternating component of stress} \quad (12)$$

Equations 11 and 12 are generalized for a normal stress, but also apply to shear stresses.

The case of completely reversed stress, where $\sigma_a = \sigma_{\max}$ and $\sigma_m = 0$, occurs for the axial normal bending stress, σ_b , and the shear stress, τ_v , (due to transverse shear force) when the shaft is rotating and is subjected to a constant direction load. This is a common loading condition.

The location of the maximum and minimum stress intensities due to these loads varies as the stress element being analyzed orbits with the rotating shaft. See Figure 3. In this figure, the loads are fixed and only the shaft element under analysis is rotating with the shaft. If all loads are treated as positive values and are therefore additive, the critical locations to be examined are at positions A ($\theta = 0$) and

B ($\theta = \pi / 2$) and at positions θ between them. Depending on the relationships between the four stresses, any position of θ between 0 and $\pi / 2$ may be the location of maximum stress.

Where stress is not uniformly distributed around the periphery of the shaft, the stress intensity of an element orbiting between A and B will vary approximately in a sine or cosine pattern as appropriate.

Most shaft stress analyses are concerned only with position A, as the bending stress, σ_b , is typically much larger than the transverse shear stress, τ_v .

Stress at position B or θ , between A and B, can become important in a short shaft section with a high shear force as may be found adjacent to a bearing.

5.4.1 The general case

For a complete 3-dimensional fatigue analysis, the Von Mises stresses will be used. See Figure 3. These are given in Equations 13 through 15, where the total or maximum stress is the sum of the alternating and mean stresses ($\sigma_{tx} = \sigma_{ax} + \sigma_{mx} + \dots$).

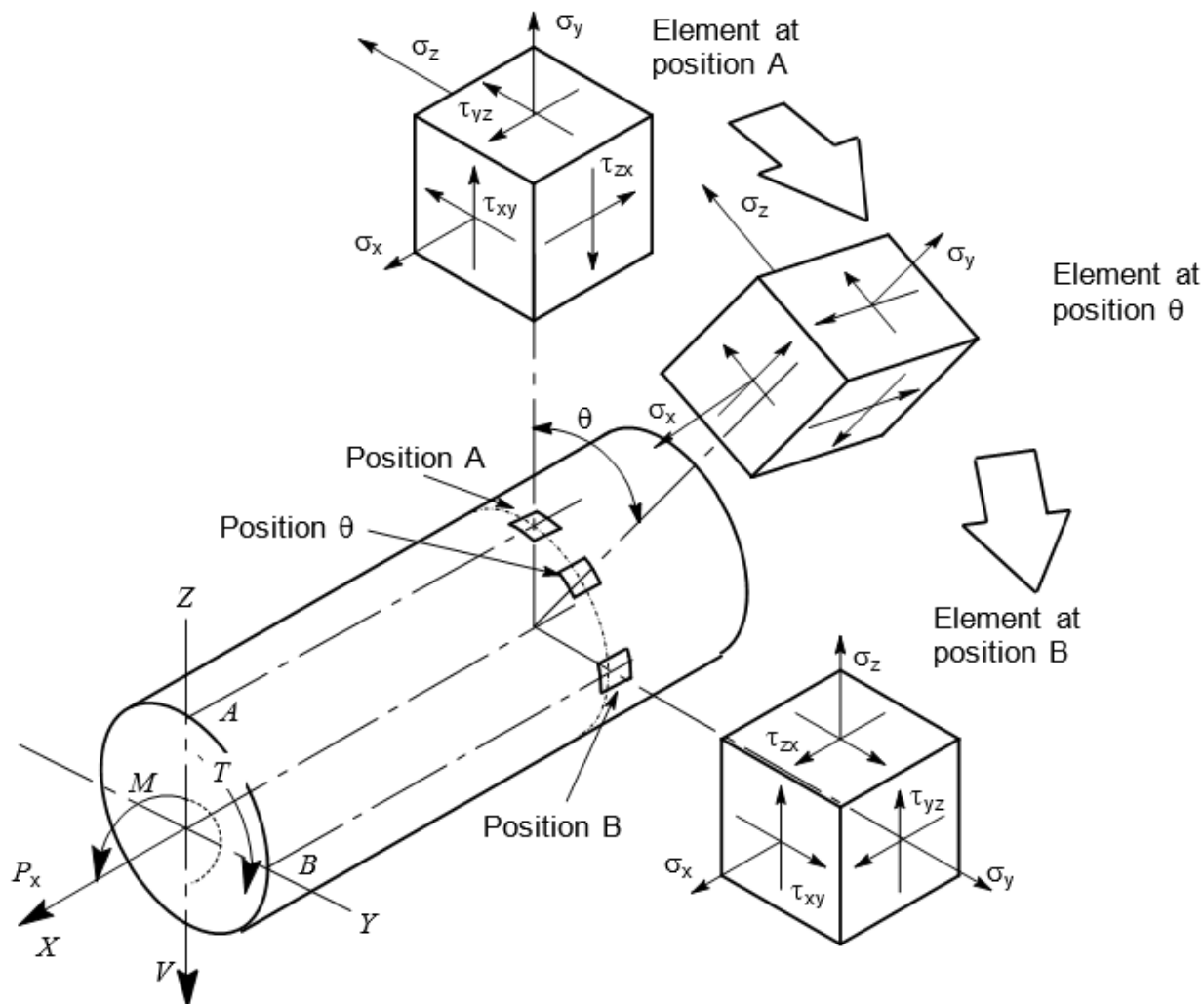


Figure 3 – Stress convention showing orbiting element

Von Mises alternating stress:

$$\bar{\sigma}_a = \left\{ 0.5 \left[(\sigma_{ax} - \sigma_{ay})^2 + (\sigma_{ay} - \sigma_{az})^2 + (\sigma_{az} - \sigma_{ax})^2 \right] + 3 \left[\tau_{axy}^2 + \tau_{ayz}^2 + \tau_{azx}^2 \right] \right\}^{0.5} \quad (13)$$

where

σ_a	is Von Mises alternating stress, lb/in ² ;
σ_{ax}	is alternating axial normal stress, lb/in ² ;
σ_{ay}	is alternating radial normal stress, lb/in ² ;
σ_{az}	is alternating tangential normal stress, lb/in ² ;
τ_{axy}	is alternating radial shear stress, lb/in ² ;
τ_{ayz}	is alternating axial shear stress, lb/in ² ;
τ_{azx}	is alternating tangential shear stress, lb/in ² .

Von Mises mean stress:

$$\bar{\sigma}_{ma} = \left\{ 0.5 \left[(\sigma_{mx} - \sigma_{my})^2 + (\sigma_{my} - \sigma_{mz})^2 + (\sigma_{mz} - \sigma_{mx})^2 \right] + 3 \left[\tau_{mxy}^2 + \tau_{myz}^2 + \tau_{mzx}^2 \right] \right\}^{0.5} \quad (14)$$

where

- $\bar{\sigma}_m$ is Von Mises mean stress, lb/in²;
- σ_{mx} is mean axial normal stress, lb/in²;
- σ_{my} is mean radial normal stress, lb/in²;
- σ_{mz} is mean tangential normal stress, lb/in²;
- τ_{mxy} is mean radial shear stress, lb/in²;
- τ_{myz} is mean axial shear stress, lb/in²;
- τ_{mzx} is mean tangential shear stress, lb/in².

Von Mises total stress:

$$\bar{\sigma}_{total} = \left\{ 0.5 \left[(\sigma_{tx} - \sigma_{ty})^2 + (\sigma_{ty} - \sigma_{tz})^2 + (\sigma_{tz} - \sigma_{tx})^2 \right] + 3 \left[\tau_{txy}^2 + \tau_{tyz}^2 + \tau_{tzx}^2 \right] \right\}^{0.5} \quad (15)$$

where

- $\bar{\sigma}_{total}$ is Von Mises total stress, lb/in²;
- σ_{tx} is total axial normal stress, lb/in²;
- σ_{ty} is total radial normal stress, lb/in²;
- σ_{tz} is total tangential normal stress, lb/in²;
- τ_{txy} is total radial shear stress, lb/in²;
- τ_{tyz} is total axial shear stress, lb/in²;
- τ_{tzx} is total tangential shear stress, lb/in².

All unused terms in the above equations are set equal to zero. The results of Equations 13 through 15 are used in Equations 3 through 5.

5.4.2 The simplified case

There are stresses that are not included in these simplified case formulas but which the designer shall appropriately address if they are encountered. These include, but are not limited to, hoop stresses due to press fitted parts, pressure on hollow shafts, thermal stresses, stresses due to unbalance and centrifugal forces, and residual stresses, any of which can become significant.

For the simplified case, all stresses on the free (outer) surface of the element are zero (i.e., no interference fitted elements), as are shear stress in the radial direction and the tangential stress. However, experience has shown that the effect of the interference fit can be modeled using the simplified case and the modifying factor for stress concentration, k_f , as discussed in 5.6.6. Keeping in mind sign convention,

At position θ :

$$\sigma_x = \sigma_b \cos \theta + \sigma_p \quad (16)$$

$$\tau_{zx} = \tau_v \sin \theta + \tau_t \quad (17)$$

$$\sigma_y = \sigma_z = \tau_{xy} = \tau_{yz} = 0 \quad (18)$$

After determining the forces and stresses on the shaft at the critical sections, determine the amount of each stress that is alternating and the amount that is mean, per Equations 11 and 12. For many cases σ_b

and τ_v are completely alternating, whereas σ_p is entirely mean. The torsional shear stress, τ_t , often has both a mean (τ_{mt}) and an alternating component (τ_{at}).

$$\tau_{mt} = 0.5(\tau_{t \max} + \tau_{t \min}) \quad (19)$$

$$\tau_{at} = 0.5(\tau_{t \max} - \tau_{t \min}) \quad (20)$$

If such is the case, at position θ ,

Mean stresses:

$$\sigma_{mx} = \sigma_p \quad (21)$$

$$\tau_{mzx} = \tau_{mt} \quad (22)$$

Alternating stresses:

$$\sigma_{ax} = \sigma_b \cos \theta \quad (\text{max at } \theta = 0) \quad (23)$$

$$\tau_{azx} = \tau_v \sin \theta + \tau_{at} \quad \left(\text{max at } \theta = \frac{\pi}{2} \right) \quad (24)$$

The mean components of stress have m as the first subscript (σ_{mx} , τ_{mzx}) while the alternating components have a as the first subscript (σ_{ax} , τ_{azx}).

For standard catalogued gear drives that are not subjected to torque reversals, alternating torques in the range of 25% to 50% are suggested. If no data exists, a worst case 50% alternating torque (that is between zero and the maximum torque, where $\tau_{mt} = \tau_{at}$) should be used. A value lower than 25% may be used if experience shows that reduced torque fluctuations exist.

If frequent torque reversals are present, consider setting $\tau_{at} = \tau_{max}$ and $\tau_{mt} = 0$ (100% alternating torque).

Von Mises alternating stress for the simplified case:

$$\bar{\sigma}_a = \left[\sigma_{ax}^2 + 3 \tau_{azx}^2 \right]^{0.5} \quad (25)$$

Von Mises mean stress for the simplified case:

$$\bar{\sigma}_m = \left[\sigma_{mx}^2 + 3 \tau_{mzx}^2 \right]^{0.5} \quad (26)$$

Calculate Equations 23 through 25 for θ_c depending on τ_{test} and τ_{at} :

$$\tau_{test} = \frac{\sigma_b^2 - 3 \tau_v^2}{3 \tau_v} \quad (27)$$

If $\tau_{at} < \tau_{test}$,

$$\theta_c = \sin^{-1} \left(\frac{\tau_{at}}{\tau_{test}} \right) \quad (28)$$

If $\tau_{at} \geq \tau_{test}$, $\theta_c = \frac{\pi}{2}$ radians

where

θ_c is the critical stress angle, radians.

and use the components σ_{ax} and τ_{azx} for the maximum value of $\bar{\sigma}_a$ to calculate Von Mises total stress for the simplified case:

$$\bar{\sigma}_{\text{total}} = \left[(\sigma_{\text{mx}} + \sigma_{\text{ax}})^2 + 3(\tau_{\text{mzx}} + \tau_{\text{azx}})^2 \right]^{0.5} \quad (29)$$

This represents the maximum value of stress at any point of the shaft surface at that location for the general case. The results of Equations 25, 26 and 29 are used in Equations 3 through 5.

5.5 Material requirements

Mechanical property values, determined from test, of the specified grade of steel in its processed and heat treated condition should be used whenever available. Mechanical properties of iron shafts are beyond the scope of this standard.

For through hardened materials, the basis for defining allowable stress is the minimum surface hardness at the critically stressed section. The minimum hardenability shall be such that the minimum hardness at a depth from the surface of 1/4 the radius of the critically stressed section shall be 75% of the minimum hardness at the surface.

For case hardened materials, the basis for defining allowable stress is the minimum core hardness at a distance of three times the effective case depth below the surface in the critically stressed section. For case hardened materials where the effective case depth of the critical section has been removed or masked, the allowable stress should be evaluated as if it were a through hardened material.

For both through hardened and case hardened materials, the hardness will be converted to tensile strength as defined in 5.5.1.

The hardness at the specified radius may be determined by measuring the hardness at the same radius on a representative test bar coupon of the same alloy which has been heat treated with the product shaft. The coupon shall have the same diameter as the shaft when it is heat treated.

The material for shafts shall meet the requirements of grade 1 of ANSI/AGMA 2001-D04 with particular attention to grain size, decarburization and stock removal and heat treatment for through hardened material. For carburized and hardened shafting, emphasis is placed on grain size, decarburization in case, surface hardness, limit of carbides in case, tempering, surface carbon in case and core hardness.

Where gear teeth are cut integral to the shaft, a grade 2 material should be considered and the appropriate grade 2 material specification for the properties specified in the previous paragraph applied.

Ground surfaces shall be free of grinding temper in the critically stressed areas.

5.5.1 Estimated properties of steel

It is important to note that the estimated material properties may deviate significantly from actual test values. Thus, an appropriate factor of safety should be used.

In the absence of test data, the following properties of forged or rolled steel shafting material may be used. For through hardened steel shafts, the ultimate tensile strength can be based on the Brinell hardness of the shaft, near the outer surface, at the diameter under consideration. For steel shafts that are case hardened by processes such as carburizing and nitriding, the ultimate tensile strength is based on the Brinell hardness of the shaft underneath the hardened case unless a detailed analysis or experience indicates that a different tensile strength should be used. See sample problem 3 in Annex D.

The yield strength and modified fatigue strength can be calculated based on the ultimate tensile strength. The following equations have been developed by curve fitting representative test results from references [1], [5] and [6].

Ultimate tensile strength:

$$\sigma_u = 500 H_B \quad (30)$$

where

σ_u is ultimate tensile strength, lb/in²;

H_B is Brinell hardness number, HBW.

NOTE: The value of 500 in Equation 30 is a typical value. In reference [8], 450 is mentioned as a minimum value.

For quench and tempered alloy steel, the tensile yield strength is:

$$\sigma_y = 0.94\sigma_u - 12\,500 \quad (31)$$

where

σ_y is tensile yield strength, lb/in².

NOTE: For other steels, contact the material supplier for tensile yield strength.

Basic fatigue strength:

$$\sigma_{fe} = 0.5 \sigma_u \text{ if } \sigma_u \leq 200\,000 \text{ lb/in}^2 \quad (32)$$

NOTE: The value of 0.5 in Equation 32 is an average value. The values can range from 0.4 to 0.6 depending upon heat treatment. See [3] for detailed information.

$$\sigma_{fe} = 100\,000 \text{ lb/in}^2 \text{ if } \sigma_u > 200\,000 \text{ lb/in}^2 \quad (33)$$

where

σ_{fe} is basic fatigue strength of polished, unnotched test specimen in reverse bending, lb/in².

5.6 Modified fatigue strength

Since the fatigue strength is largely influenced by physical conditions, environmental conditions, and application conditions, as well as material conditions, the basic fatigue strength must be modified.

$$\sigma_f = k \sigma_{fe} \quad (34)$$

where

σ_f is modified fatigue strength, lb/in²;

k is fatigue strength modification factor.

The fatigue strength modification factor, k , is the product of the fatigue factors (k_a through k_g). The fatigue safety factor, F_{St} , is greatly affected by the fatigue strength modification factor. Therefore, care should be exercised in determining the values of the fatigue factors [1].

$$k = k_a k_b k_c k_d k_e k_f k_g \quad (35)$$

5.6.1 Surface finish factor, k_a

The surface finish factor accounts for the difference between the actual shaft and a highly polished test specimen. Values for k_a are given in Figure 4.

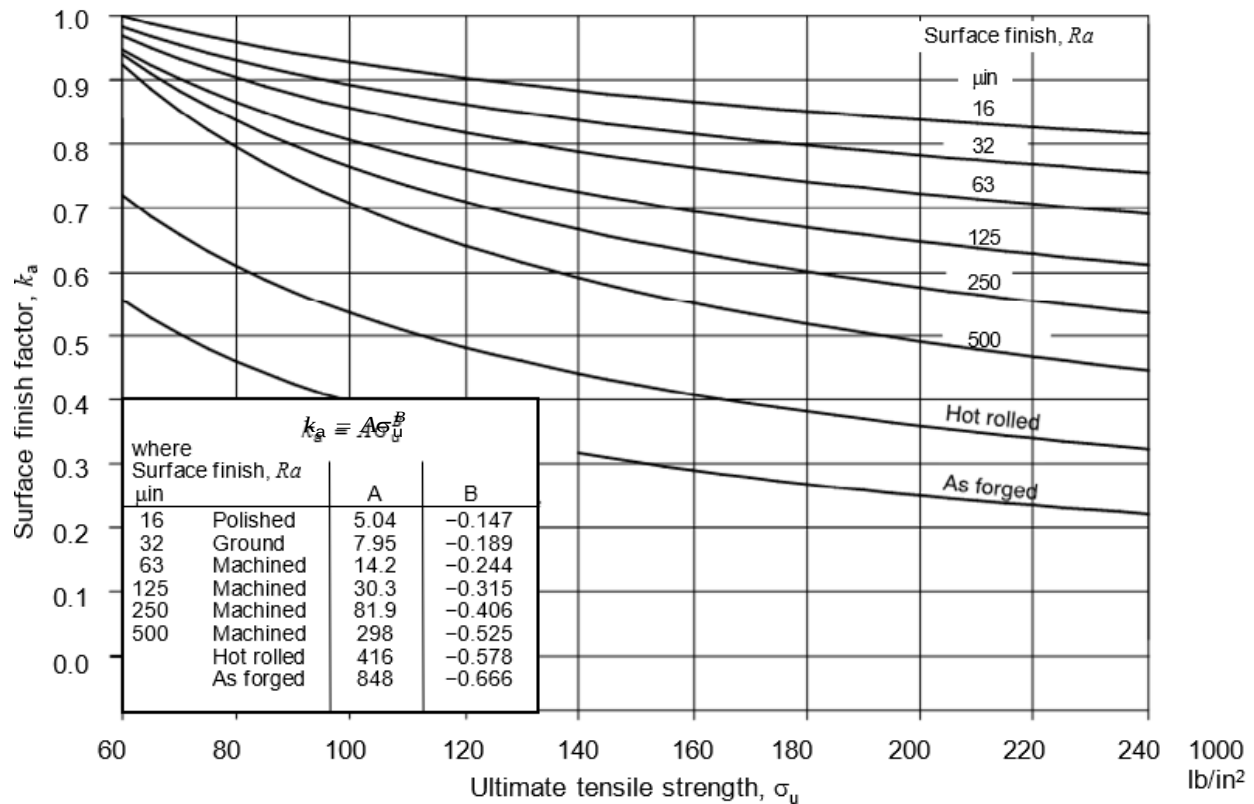


Figure 4 – Surface finish factor, k_a [2]

5.6.2 Size factor, k_b

The size factor accounts for the increased likelihood of encountering a fatigue initiating defect as the shaft diameter increases. Values for k_b are given in Figure 5.

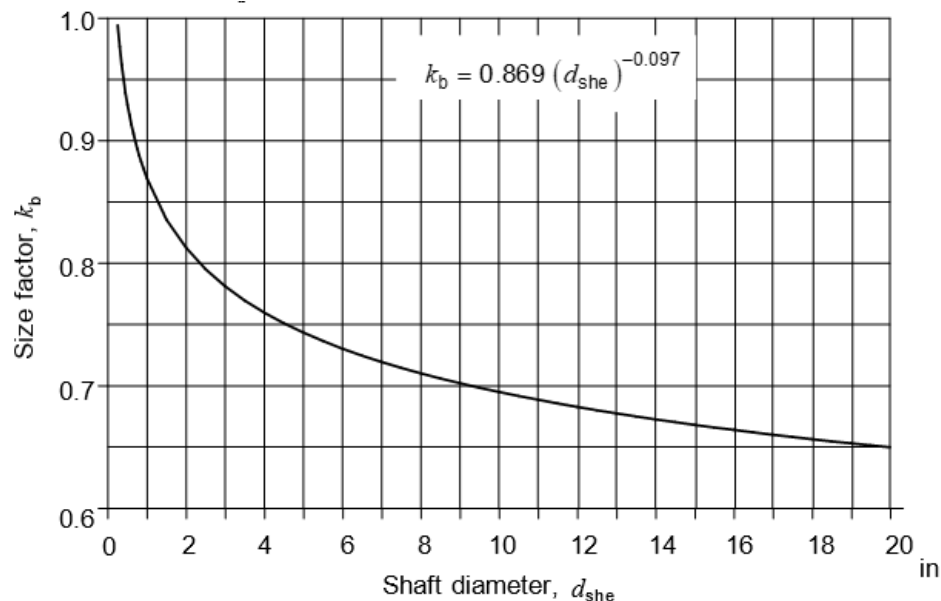


Figure 5 – Size factor, k_b [2]