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

in which:

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

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

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

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

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

where:

K_v = dynamic factor (dim)

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

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

v_t = pitch line velocity (fpm)

d = pitch diameter of the pinion (in.)

N_p = number of teeth on the pinion

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

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

H_B = Brinell hardness for the teeth (dim)

N = number of load cycles

n_p = pinion (rpm)

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

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

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

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

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

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

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

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

m_B = the backup ratio (dim)

t_R = rim thickness (in.)

h_t = total tooth height (in.)

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

J = geometry factor for bending strength (dim)

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

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

- For $H_B \approx 250$ and $10^3 < N < 3 \times 10^6$
- $Y_N = 4.9404\ N^{-0.1045}$ (6.7.5.2.2-7)
- For $N > 3 \times 10^6$ load cycles, regardless of hardness
- $Y_N = 1.6831\ N^{-0.0323}$ (6.7.5.2.2-8)

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

K_R = reliability factor (dim):

- for 99 percent reliability
- 1.25 for 99.9 percent reliability

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

Equal Addendum Loaded at Highest Point, Single Tooth Contact								
Gear Teeth	Pinion Teeth, N_p							
	18		19		20		21	
	P	G	P	G	P	G	P	G
18	0.30	0.30	—	—	—	—	—	—
19	0.30	0.30	0.31	0.31	—	—	—	—
20	0.30	0.31	0.31	0.31	0.31	0.31	—	—
21	0.30	0.32	0.31	0.32	0.32	0.32	0.33	0.33
26	0.31	0.34	0.31	0.34	0.32	0.34	0.33	0.35
35	0.31	0.37	0.32	0.37	0.33	0.37	0.34	0.37
55	0.32	0.40	0.32	0.40	0.33	0.40	0.34	0.40
135	0.33	0.43	0.33	0.43	0.34	0.43	0.35	0.43

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

Equal Addendum Loaded at Tooth Tip								
Gear Teeth	Pinion Teeth, N_p							
	18		19		20		21	
	P	G	P	G	P	G	P	G
18	0.23	0.23	—	—	—	—	—	—
19	0.23	0.23	0.23	0.23	—	—	—	—
20	0.23	0.24	0.23	0.24	0.24	0.24	—	—
21	0.23	0.24	0.23	0.24	0.24	0.24	0.24	0.24
26	0.23	0.25	0.23	0.25	0.24	0.25	0.24	0.25
35	0.23	0.26	0.23	0.26	0.24	0.26	0.24	0.26
55	0.23	0.28	0.23	0.28	0.24	0.28	0.24	0.28
135	0.23	0.29	0.23	0.29	0.24	0.29	0.24	0.29

6.7.5.2.3 Surface Durability and Wear—Design Equations

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

The following must then be satisfied:

$$W_t \leq W_{tac} \quad (6.7.5.2.3\text{-}1)$$

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

$$W_{tac} = \frac{F\,dI}{K_o K_v K_s K_m C_f}\left(\frac{S_{ac} Z_N C_H}{C_p S_H K_T K_R}\right)^2 \quad (6.7.5.2.3\text{-}2)$$

where:

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

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

N = number of load cycles (dim)

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

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

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

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

I = geometry factor for pitting resistance (dim)

C_f = surface condition factor for pitting resistance taken as 1.0 for good tooth surface condition as specified in Article 6.7.8 for tooth surface finish depending on diametral pitch (dim)

C6.7.5.2.3

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

Table C6.7.5.2.3-1—I Factors for 20° Full Depth Spur Pinions/Gears

Equal Addendum Factors same for both Pinion and Gear				
Gear Teeth	Pinion Teeth, N_p			
	18	19	20	21
	P G	P G	P G	P G
18	0.075	—	—	—
19	0.077	0.076	—	—
20	0.079	0.078	0.076	—
21	0.080	0.080	0.078	0.078
26	0.084	0.084	0.084	0.084
35	0.091	0.091	0.091	0.091
55	0.100	0.101	0.102	0.102
135	0.112	0.114	0.116	0.118

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

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

6.7.5.2.4 Yield Failure at Intermittent Overload

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

The following must then be satisfied:

$$W_t\,(max) \leq W_{max} \tag{6.7.5.2.4-1}$$

based on the overload condition.

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

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

where:

K_y = yield strength factor taken as 0.50 (dim)

f = stress correction factor = 1 (dim)

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

C6.7.5.2.4

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

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

AGMA gives an equation only for an enclosed drive.

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

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

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

J = geometry factor for bending strength (dim)

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

6.7.6 Enclosed Speed Reducers

6.7.6.1 General

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

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

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

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

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

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

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

C6.7.6.1

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

See the provisions of Article 6.10.4.2.

6.7.6.2 Parallel Spur, Helical, and Bevel Gear Reducers

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

6.7.6.3 Worm Gear Reducers

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

C6.7.6.3

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

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

- the worms be heat-treated alloy steel and the worm gear shall be typically phosphor, tin, or manganese alloys of bronze,
- the thread of the worm be ground and polished, and the teeth of the gear shall be accurately cut to the correct profile,
- the worm and gear thrust loads be taken by rolling element bearings, mounted in water and oil-tight housings, and
- the unit shall be mounted in a cast-iron or steel/cast steel housing and the lubrication shall be continuous while in operation.

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

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

6.7.6.4 Planetary Gear Reducers

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

6.7.6.5 Cycloidal Speed Reducers

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

C6.7.6.5

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

6.7.6.6 Mechanical Actuators

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

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

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

C6.7.6.6

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

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

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

6.7.7 Bearing Design

6.7.7.1 Plain Bearings

6.7.7.1.1 General

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

Large journal bearings shall be of the split type with one half recessed into the other half. The length of a bearing shall be not less than its diameter. The base half of bearings for gear trains and for mating gears and pinions shall be in one piece. The caps of bearings shall be secured to the bases with turned bolts with square heads recessed into the base or threaded dowels and with double hexagonal nuts. The nuts shall bear on finished bosses or spot-faced seats.

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

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

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

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

C6.7.7.1.1

Large journal pillow block bearings are generally greater than 4 in. bore diameter.

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

6.7.7.1.2 Plain Bearing Design Equations

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

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

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

where:

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

p = pressure (psi)

V = surface velocity (fpm)

C6.7.7.1.2

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

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

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

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

L = length of the bearing (in.)

n = journal rotational speed (rpm)

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

Table 6.7.7.1.2-1—Performance Parameters for Cast Bronze Bearings

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

6.7.7.1.3 Lubricated Plain Bearings

Journal bearings should have bronze bushings. For lightly loaded bearings, the bushings may be bronze or a nonmetallic substance as specified herein. For split bearings, the bushing shall be in halves and shall be provided with an effective device to prevent its rotation under load. The force tending to cause rotation shall be taken as six percent of the maximum load on the bearing and as acting at the outer circumference of the bushing. A clearance of approximately $^1/_4$ in. shall exist between the bushing of the cap and the bushing of the base into which laminated liners shall be placed. The inside longitudinal corners of both halves shall be rounded or chamfered, except for a distance of 0.4 in. from each end or from the shaft shoulder fillet tangent point.

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

6.7.7.1.4 Self-Lubricating; Low Maintenance Plain Bearings

6.7.7.1.4a Metallic Bearings

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

C6.7.7.1.4a

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

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

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

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

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

6.7.7.1.4b NonMetallic Bearings

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

C6.7.7.1.4b

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

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

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

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