

Pumping Units

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Pumping Units

1 Scope

This specification provides the requirements and guidelines for the design and rating of beam pumping units for use in the petroleum and natural gas industry. Included are all components between the carrier bar and the gear reducer input shaft. This includes the following:

- a) beam pump structures;
- b) pumping unit gear reducer.

Only loads imposed on the structure and/or gear reducer by the polished rod load are considered in this specification.

Also included are the requirements for the design and rating of enclosed gear reducers wherein the involute gear tooth designs include helical and herringbone gearing. The rating methods and influences identified in this specification are limited to single- and multiple-stage designs applied to beam pumping units in which the pitch-line velocity of any stage does not exceed 5000 ft/min and the speed of any shaft does not exceed 3600 rpm.

This standard does not cover chemical properties of materials, installation, and maintenance of the equipment, beam type counterbalance units, prime movers, and power transmission devices outside the gear reducer or control systems.

2 Normative References

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

API Specification 11B, *Specification for Sucker Rods*

AGMA 908-B89 ¹, *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth*

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

ANSI ²/AGMA 1012-G05, *Gear Nomenclature, Definitions of Terms with Symbols*

AWS ³ D1.1/D.1M, *Structural Welding Code—Steel*

¹ American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314, www.agma.org.

² American National Standards Institute, 25 West 43rd Street, 4th floor, New York, New York 10036, www.ansi.org.

³ American Welding Society, 8669 NW 36th Street, #130, Miami, Florida 33166, www.awsnow.org.

3 Terms, Definitions, Abbreviations, and Symbols

3.1 Terms and Definitions

For the purposes of this document, the following terms and definitions apply. Additionally, the terms provided in ANSI/AGMA 1012-G05 also apply. See Figure J.1 for an illustration of a beam pumping unit.

3.1.1

beam pump structure

All components between the carrier bar and the gear reducer output shaft.

3.1.2

beam pumping unit

A machine for translating rotary motion from a crankshaft to linear reciprocating motion for the purpose of transferring mechanical power to a downhole pump.

3.1.3

brake

A component of a pumping unit that is designed to restrain motion in all rotary joints.

NOTE It is often composed of a disk or drum mounted on the reducer input shaft combined with a mechanism to impart a restraining friction torque.

3.1.4

carrier bar

Part of the pumping unit wireline assembly that supports the load of the sucker rod string through its interface with the polished rod clamp.

3.1.5

Class I lever system

A lever system in which the fulcrum is located between the load and the applied force or effort.

NOTE An example of this is a beam pumping unit with the fixed saddle bearing located along the walking beam between the equalizer and the well.

3.1.6

Class III lever system

A lever system in which the applied force (effort) is located between the load and fulcrum.

NOTE An example of this is a beam pumping unit with the equalizer located between the fixed Samson post bearing and the well.

3.1.7

counterbalance effect

CB

The lift-assisting force as measured at the polished rod arising from the moment of the counterweights, where in the case of crank mounted counterbalance, maximum *CB* is typically reported with the cranks oriented at 90° (horizontal).

3.1.8

crank rotation

The direction of rotation, either clockwise or counterclockwise, as viewed from the side of the beam pumping unit with the horsehead to the right.

3.1.9**cranks**

A driving link in the four-bar linkage of a beam pumping unit that is located between the output shaft of the gear reducer and the pitman link.

3.1.10**diametral**

Along a diameter.

3.1.11**equalizer**

Connects the pitman links to the walking beam.

3.1.12**gear reducer**

A mechanism located between the prime mover and the cranks to transmit rotary power while reducing unit speed and increasing torque.

3.1.13**hanger**

See **carrier bar**.

3.1.14**horsehead**

A component of a beam pumping unit designed to transmit force and motion from the walking beam to the flexible wireline.

NOTE Its shape is such that the imparted motion is directed vertically above the wellhead, allowing the polished rod to move without undue side loads.

3.1.15**pitmans**

The connecting link in the pumping unit mechanism between the cranks and the equalizer.

3.1.16**saddle bearing**

The primary fulcrum bearing supporting the walking beam of a conventional (Class 1 lever) pumping unit.

3.1.17**Samson post bearing**

A bearing mounted to a fixed location atop the Samson post of the pumping unit that is attached to and provides the fulcrum location for the walking beam.

3.1.18**structural unbalance**

B

The force required at the polished rod to balance the beam in a horizontal position with the pitmans disconnected from the crank pin and no applied well load.

NOTE The structural unbalance is considered positive when the force required at the polished rod is downward, and negative when upward.

3.1.19**torque factor***TF*

The factor for any given crank angle that, when multiplied by the load at the polished rod, gives the torque at the crankshaft of the beam pumping unit gear reducer.

NOTE The torque factor has units of length.

3.2 Abbreviations and Symbols

The symbols and definitions used in this specification may differ from other specifications. Users should assure themselves that they are using these symbols and definitions in the manner indicated herein. See Annex D, Annex E, Annex F, and Annex G for additional symbol definitions that are exclusive to those annexes.

<i>A</i>	distance from the center of the saddle bearing to the centerline of the polished rod, in inches (in.) [millimeters (mm)]
<i>A_t</i>	tensile area of fastener, in square inches (in. ²) [millimeters (mm ²)]
<i>a</i>	area of cross section, in square inches (in. ²) [millimeters (mm ²)]
<i>B</i>	structural unbalance, in pounds (lb) [newtons (N)]
<i>C</i>	distance from the center of the saddle bearing to the center of the equalizer bearing, in inches (in.) [millimeters (mm)]
<i>C_b</i>	bearing manufacturer's specific dynamic rating, in pounds (lb) [newtons (N)]
<i>C_m</i>	load distribution factor for pitting resistance, unitless
<i>C_p</i>	elastic coefficient, unitless
<i>C₁</i>	pitting velocity factor, unitless
<i>C₂</i>	pitting contact width factor, unitless
<i>C₃</i>	pitting stress for external helical gears, unitless
<i>C₅</i>	velocity factor for pitting resistance, unitless
<i>CB</i>	counterbalance effect, in pounds (lb) [newtons (N)]
<i>CD</i>	standard center distance between gear shafts, in inches (in.) [millimeters (mm)]
<i>D</i>	operating pitch diameter of gear, in inches (in.) [millimeters (mm)]
<i>D_m</i>	major diameter of fastener, in inches (in.) [millimeters (mm)]
<i>d</i>	the operating pitch diameter of the pinion. If the pinion is enlarged, <i>d</i> can be taken to be the outside diameter minus two standard addendums, expressed in inches (millimeters)
<i>d_s</i>	shaft diameter (for tapered shaft use mean diameter), in inches (in.) [millimeters (mm)]
<i>E</i>	modulus of elasticity, in pounds per square inch (psi) [megapascals (Mpa)]
<i>E_g</i>	modulus of elasticity for gears, in pounds per square inch (psi) [megapascals (Mpa)]
<i>E_p</i>	modulus of elasticity for pinions, in pounds per square inch (psi) [megapascals (Mpa)]
<i>F</i>	net face width, in inches (in.) [millimeters (mm)]
<i>f_{cb}</i>	allowable compressive stress in bending, in pounds per square inch (psi) [megapascals (Mpa)]
<i>f_{s,b}</i>	allowable stress due to bending, in pounds per square inch (psi) [megapascals (Mpa)]
<i>f_{s,t}</i>	allowable stress due to torsion, in pounds per square inch (psi) [megapascals (Mpa)]

f_{tb}	tensile stress in extreme fibers in bending, in pounds per square inch (psi) [megapascals (Mpa)]
G	height from the center of the crankshaft to the bottom of the base beams, in inches (in.) [millimeters (mm)]
G_{τ}	shear modulus, in pounds per square inch (psi) [megapascals (Mpa)]
H	height from the center of the saddle bearing to the bottom of the base beams, in inches (in.) [millimeters (mm)]
H_B	Brinell hardness
$H_{B,g}$	Brinell hardness for gears, unitless
$H_{B,p}$	Brinell hardness for pinions, unitless
$h_c, h_{c,1}, h_{c,2}$	minimum total case depth, in inches (in.) [millimeters (mm)]; NOTE The subscript labels 1 and 2 define a range in which the total case depth (h_c) must lie.
h_e, h_{e1}, h_{e2}	minimum effective case depth, in inches (in.) [millimeters (mm)]; NOTE The subscript labels 1 and 2 define a range in which the total case depth (h_e) must lie.
h_1	height of key in the shaft or hub that bears against the keyway, in inches (in.) [millimeters (mm)]
I	horizontal distance between the centerline of the saddle bearing and the centerline of the crank shaft, in inches (in.) [millimeters (mm)]
I_p	geometry factor for pitting resistance, unitless
I_y	weak axis second moment of inertia, in inches to the power four (in. ⁴) [millimeters to the power of four (mm ⁴)]
J	distance from the center of the crankpin bearing to the center of the saddle bearing, in inches (in.) [millimeters (mm)]
J_b	geometry factor for bending strength, unitless
$J_{b,g}$	geometry factor for bending gears, unitless
$J_{b,p}$	geometry factor for bending pinions, unitless
J_t	torsional constant, in inches to the power four (in. ⁴) [millimeters to the power of four (mm ⁴)]
K	distance from the center of the crankshaft to the center of the saddle bearing, in inches (in.) [millimeters (mm)]
K_m	helical gear load distribution factor, unitless
K_{ms}	load distribution factor, static torque, unitless
K_y	yield strength factor, unitless
K_1	strength velocity factor, unitless
K_2	strength contact number, unitless
K_4	strength geometry number, unitless
K_5	velocity factor for bending strength, unitless
k	bearing rating factor, unitless
k_h	factor applied to account for any uncorrected distortion due to hardening the gears, unitless
L	length of key, in inches (in.) [millimeters (mm)]

L_{min}	minimum total length of lines of contact in contact zone, in inches (in.) [millimeters (mm)]
l	unbraced length of column, in inches (in.) [millimeters (mm)]
M	maximum moment of the rotary counterweights, cranks, and crankpins about the crankshaft, in inch-pounds (in.-lb) [newton-meters (Nm)]
M_a	geometry constant for a given unit, in square inches (in. ²) [millimeters (mm ²)]
m	metric module in plane of rotation (transverse), expressed in millimeters (mm)
m_g	gear ratio, unitless
m_n	normal metric module, expressed in millimeters (mm)
n	end restraint constant, unitless
n_o	rotational speed of output shaft, equal to the pumping speed, in revolutions per minute (rpm)
n_p	pinion rotational speed, in revolutions per minute (rpm)
N_g	number of teeth on gear, unitless
N_p	number of teeth on pinion, unitless
N_t	threads per inch of fastener
P	effective length of the pitman (between bearing axes), in inches (in.) [millimeters (mm)]
P_R	polished rod load, in pounds (lb) [newtons (N)]
P_a	pressure in air counterbalance tank for a given crank position θ , in pounds per square inch (psi) [kilopascals (kPa)]
P_d	diametral pitch in plane of rotation (transverse), in inverse inches (in. ⁻¹)
P_{nd}	the normal diametral pitch (the number of teeth per inch of diameter of the gear), in inverse inches (in. ⁻¹)
PRP	polished rod position for each crank position expressed as a fraction of the stroke above the lowermost position, unitless
p	thread pitch of metric fastener, in millimeters (mm)
p_N	normal base pitch, in inches (in.) [millimeters (mm)]
R	radius of the crank in inches (in.) [millimeters (mm)]
R_1	bearing load ratio, unitless
r	radius of gyration of section, in inches (in.) [millimeters (mm)]
S_{ac}	allowable contact stress, in pounds per square inch (psi) [megapascals (Mpa)]
S_{at}	allowable bending stress, in pounds per square inch (psi) [megapascals (Mpa)]
S_{ay}	allowable yield strength of the gear or pinion material, in pounds per square inch (psi) [megapascals (Mpa)]
S_c	compressive stress of key, in pounds per square inch (psi) [megapascals (Mpa)]
S_s	shear stress of key, in pounds per square inch (psi) [megapascals (Mpa)]
S_{ut}	tensile strength of material, in pounds per square inch (psi) [megapascals (Mpa)]
S_x	section modulus of walking beam, in inches cubed (in. ³) [millimeters cubed (mm ³)]
S_y	yield strength of material, in pounds per square inch (psi) [megapascals (Mpa)]
T	peak torque rating, in inch-pounds (in.-lb) [newton-meters (Nm)]

T_{ac}	allowable transmitted torque at output shaft, based on pitting resistance, in inch-pounds (in.-lb) [newton-meters (Nm)]
$T_{as,l}$	allowable static torque at the gear or pinion being checked, in inch-pounds (in.-lb) [newton-meters (Nm)]
T_{at}	allowable transmitted torque at output shaft, based on bending strength, in inch-pounds (in.-lb) [newton-meters (Nm)]
T_n	net torque at the crankshaft, in inch-pounds (in.-lb) [newton-meters (Nm)]
T_r	torque due to the rotary counterweights, cranks, and crank pins for a given crank angle θ , in inch-pounds (in.-lb) [newton-meters (Nm)]
T_t	transmitted shaft torque, in inch-pounds (in.-lb) [newton-meters (Nm)]
T_{wn}	torque due to the net polished rod load for a given crank angle θ , in inch-pounds (in.-lb) [newton-meters (Nm)]
TF	torque factor, in inches (in.) [meters (m)]
v_t	pitch-line velocity, in feet per minute (f/min) [meters per second (m/s)]
W	walking beam rating, in pounds (lb) [newtons (N)]
W_c	counterbalance at the polished rod, determined using a dynamometer with crankpin at 90°, in pounds (lb) [newtons (N)]
W_n	net polished rod load, in pounds (lb) [newtons (N)]
W_1	maximum load on bearing, in pounds (lb) [newtons (N)]
W_2	maximum applied load on column, in pounds (lb) [newtons (N)]
w	width of key, in inches (in.) [millimeters (mm)]
Z	length of line of action in the transverse plane, in inches (in.) [millimeters (mm)]
α	angle between P and R measured clockwise from R to P , in degrees
β	angle between C and P , in degrees
χ	angle between C and J , in degrees
ρ	angle between K and J , in degrees
τ	angle of crank counterweight arm offset for front mounted geometry (Class III lever systems), in degrees
θ	angle of crank rotation viewed with the wellhead to the right NOTE The zero-degree origin for θ changes with respect to the class of pumping unit.
ϕ	angle between K and the upward vertical
ϕ_n	normal operating pressure angle, in degrees
ϕ_t	operating transverse pressure angle, in degrees
ψ	operating helix angle, in degrees
ψ_b	angle between C and K , at bottom (lowest) polished rod position, in degrees
ψ_t	angle between C and K , at top (highest) polished rod position, in degrees

4 Product Requirements

4.1 Functional Requirements

The user/purchaser shall determine the applicable well and environmental operational conditions to order products that conform to this specification, and specify the requirements and/or identify the manufacturer's specific products. These requirements may be conveyed by means of dimensional drawing, datasheet, or other suitable documentation.

To ensure proper interfaces with the other elements of the beam pumping system, such as the complete sucker rod string and the downhole reciprocating pump, the following requirements shall be specified:

- a) required well lifting capacity by identification of the applicable downhole pump;
- b) required sucker rod size in alignment with well depth, rod design, or other mechanical well parameters;
- c) the total sucker rod string mass (weight) in the well;
- d) potential extra loads due to the well configuration, friction, and dynamic loading;
- e) required gear configuration and resulting gear loading expressed as gear reducing rating, defining the required lifting energy input;
- f) required load capability of the beam pump structure to accommodate the sucker rod string weight and additional loads; and
- g) the required maximum stroke length.

The combined requirements of gear reduction rating, structure loading capacity, and maximum stroke length shall be used to identify the specific beam pumping unit to be ordered as indicated by the designation number provided in Annex B, Table B.1.

It is recommended that beam pumping units furnished to this specification adhere to the gear reducer rating, structure capacity, and stroke length as given in Table B.1, although the combinations of these items that make up the pumping unit designation need not be identical to those in the table.

Recommended forms are provided in Annex C for rating of crank counterbalances (see Figure C.1) and for recording pumping unit stroke and torque factors (see Figure C.2). Recommendations and examples for the calculation and application of torque factor on pumping units are contained in Annex D to Annex G, and examples for calculating torque ratings for pumping unit reducers are contained in Annex H.

A recommendation for conducting a system analysis is contained in Annex I.

Annex J identifies common names of components in a typical beam pump configuration that are used in this specification.

4.2 Technical Requirements

4.2.1 General

Designs developed after the publication of this specification shall be conducted according to the methods and assumptions as defined in Section 5 and Section 6.

Beam pumping unit designs developed prior to this specification for which the manufacturer can document satisfactory conformance/performance to the requirements included in this standard shall be considered as meeting this standard.

4.2.2 Stroke and Torque Factors

For the torque on a reducer to be determined conveniently and accurately from dynamometer test data, manufacturers of beam pumping units shall, if requested by the purchaser, provide stroke and torque factors for each 1° position of the crank. Figure C.2 is an example form for recording this data.

4.2.3 Design Requirements

Design requirements shall include those criteria defined in Section 5 and Section 6 and other pertinent requirements upon which the design is based. Additive dimensional tolerances of components shall be such that proper operation of the beam pumping unit is assured. This requirement applies to manufacturer-assembled equipment and to replacement components or subassemblies.

4.2.4 Design Documentation

Documentation of designs shall include methods, assumptions, calculations, and design requirements. Design documentation shall be reviewed and verified by a qualified individual other than the individual who created the original design. Design documentation according to the list below shall be maintained for 10 years after date of last manufacture.

- a) One complete set of drawings and written specifications/standards, including material type and yield strength as designated in Section 5 and Section 6.
- b) Instructions providing methods for the safe assembly and disassembly of the beam pumping unit and stating the operations that are permitted in order to preclude failure and/or nonconformance with the stated performance.

4.2.5 Design Changes

The manufacturer shall, at a minimum, consider the following when making design changes: stress levels of the modified or changed components; material changes; and functional changes. All design changes and modifications shall be identified, documented, reviewed, and approved before their implementation. Design changes and changes to design documents shall require the same control features as the original design.

5 Beam Pump Structure Requirements

5.1 General

Requirements for beam pump structures are specified in the following sections. Only loads imposed on the structure and/or gear reducer by the polished rod load are considered in this specification. Polished rod load ratings are specified in API 11B.

Additional loads on the beam pumping unit imposed by add-on devices (such as compressors and stroke increasers) attached to the reducer, walking beam, or other structural components are not covered by this specification.

No dimensional requirements, other than stroke length, are given.

5.2 Design Loads for All Structural Members Except Walking Beams

For all pumping unit geometries, and unless otherwise specified, the maximum load exerted on the component being considered shall be determined by examining the loads on the component at each 15° crank position on the upstroke of the pumping unit.

The polished rod load, P_R , shall be used for all upstroke crank positions.

For units with bidirectional rotation and nonsymmetrical torque factors, the direction of rotation used for design calculations shall be that which results in the highest forces in structural components.

Due consideration shall be given to the direction of loading on all structural bearings and on the structural members supporting these bearings.

5.3 Design Stresses for All Structural Members Except Walking Beams, Bearing Shafts, and Cranks

Allowable stress levels are based on simple stresses without consideration of stress risers. Adequate stress concentration factors shall be used when stress risers occur.

Design stresses for all structural components shall be a function of the yield strength of the material, S_y .

Components subjected to simple tension or compression and nonreversing bending shall have a limiting stress of $0.3S_y$. If stress risers occur in critical zones of tension members, the limiting stress shall be $0.25S_y$.

Components subjected to reverse bending shall have a limiting stress of $0.2S_y$.

Equation (1) shall be used for all components acting as columns.

$$W_2 = \frac{aS_y}{4} \left[1 - \frac{S_y}{4n\pi^2 E} \left(\frac{l}{r} \right)^2 \right] \quad (1)$$

where

W_2 is the maximum applied load on column expressed in lb (N);

a is the area of cross section expressed in in.² (mm²);

S_y is the yield strength of material expressed in psi (mPa);

n is the end restraint constant, assumed to be 1.0;

E is the modulus of elasticity expressed in psi (mPa);

l is the unbraced length of column expressed in inches (millimeters);

r is the radius of gyration of section expressed in inches (millimeters).

The value for $\left(\frac{l}{r} \right)$ shall not exceed 90. For $\left(\frac{l}{r} \right)$ values of 30 or less, columns may be assumed to be acting in simple compression.

5.4 Design Loads for Walking Beam

Equation (2) shall be used for rating conventional walking beams as shown in Figure 1.

$$W = \frac{f_{cb}}{A} S_x \quad (2)$$

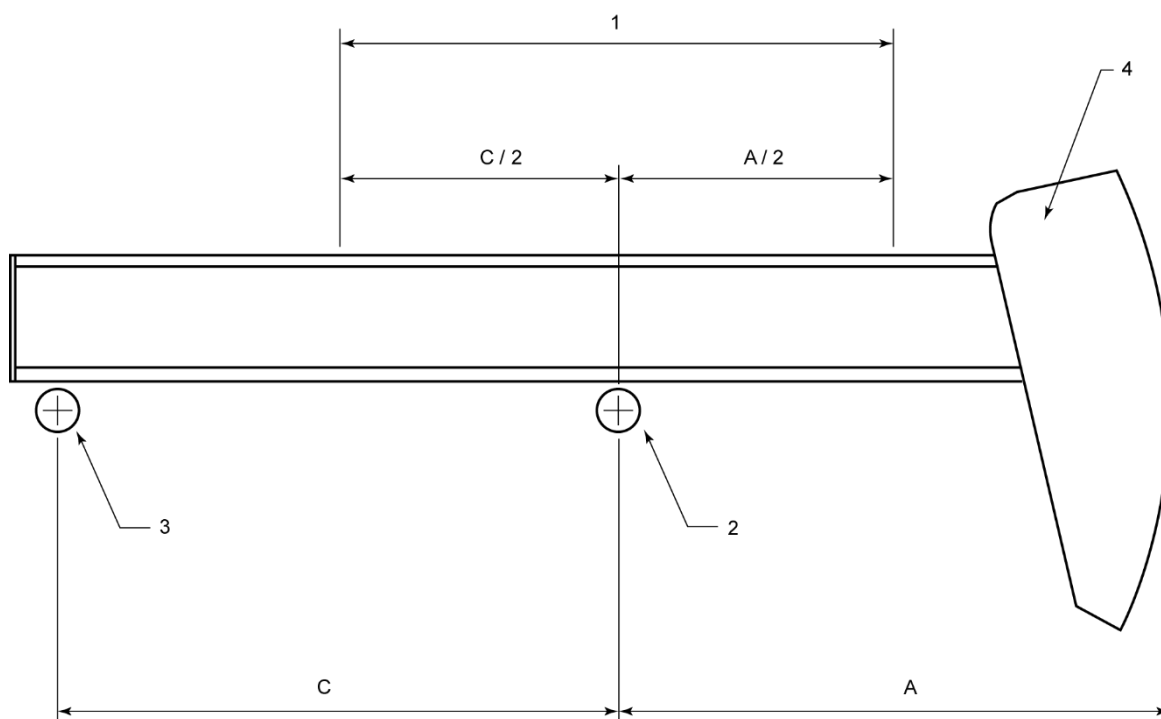
where

W is the walking beam rating, equal to the design polished rod load expressed in lb (N);

f_{cb} is the allowable compressive stress in bending expressed in psi (mPa) (see Table 1 for maximum allowable stresses);

S_x is the section modulus of the walking beam expressed in in.³ (mm³);

A is the distance from the centerline of the saddle bearing to the centerline of the polished rod expressed in inches (millimeters) (see Figure 1).



Key

- 1 Critical zone in tension flange
- 2 Saddle bearing
- 3 Equalizer bearing
- 4 Horsehead

Figure 1—Walking Beam Elements

Equation (2) shall be used to calculate the design loads for the following structural type walking beams:

- a) I beam (metric equivalent);
- b) H beam (metric equivalent);

- c) wide flange beam (metric equivalent);
- d) S beam (metric equivalent).

For walking beam construction using a single rolled section, the gross section of the rolled beam shall be used to determine the section modulus; however, holes or welds are not permissible on the tension flange in the critical zone (see Figure 1) for either rolled beams or fabricated beams.

Designs for welded walking beam construction, and for the addition of built-up sections, shall include changes in loading, checks on stresses at all critical sections, and the calculations for stress concentration. However, holes or welds are not permissible on the tension flange in the critical zone (see Figure 1).

The design of welded beams shall conform to AWS D1.1/D1.1M.

5.5 Maximum Allowable Stress for Walking Beams

The maximum allowable stress, f_{cb} , for both the conventional rolled walking beam and unconventional construction or built-up sections, shall be determined from Table 1. For standard rolled beams, maximum allowable tensile/compressive stress in line 2 and line 4 shall be $0.3S_y$. For unconventional construction and built-up sections, the maximum allowable tensile/compressive stress in line 2 and line 4 shall be $0.25S_y$.

For beams having cross sections symmetrical with the horizontal neutral axis, the critical stress is compression in the lower flange. The maximum value of this stress, f_{cb} , is the smaller of the values determined from lines 3 and 4 in Table 1.

5.6 Weld Workmanship Requirements for All Structural Members

5.6.1 General

These requirements apply to load-bearing members. Accessories such as crank guards are excluded.

5.6.2 Procedure and Welder Qualification

All welding procedure specifications (WPS's) shall conform with AWS D1.1. Where permitted by AWS D1.1, radiographic testing may be substituted for guided bend tests.

WPS and performance qualification report (PQR) documentation shall be maintained for 10 years after the date of last manufacture.

5.6.3 Fabrication

Fabrication of all structural members shall conform with AWS D1.1.

5.6.4 Inspection

Inspection of all welds shall conform with AWS D1.1. All welds shall be inspected against cyclically loaded inspection criteria.

5.7 Other Structural Components

5.7.1 Shafting

The limiting stresses for all bearing shafts, as well as other structural shafting, are given in 6.3.5.1.

5.7.2 Hanger

Wire lines for horseheads shall have a minimum factor of safety of five with respect to breaking strength.

For allowable stresses on carrier bar, end fittings, etc., see 5.3.

Table 1—Maximum Allowable Stresses in Pumping Unit Walking Beams of Structural Steel (see Figure 1)

Line	Stress	Symbol	Values
1	Specified minimum yield strength of material	S_y	36,000 psi (248.21 MPa)
2	Maximum allowable tensile stress in extreme fibers in bending	f_{tb}	$(0.25, 0.3) S_y$
3	Maximum allowable compressive stress in extreme fibers in bending, not to exceed value in line 4	f_{cb}	$\frac{\sqrt{EI_y G_\tau J_t}}{S_x l}$
4	Maximum allowable compressive stress in extreme fibers in bending, except if limited by line 3	f_{cb}	$(0.25, 0.3) S_y$

where

J_t is the torsional constant of the beam section expressed in in.⁴ (mm⁴);

l is the longest laterally, unbraced length of beam expressed in inches (mm) [longer of C or A (see Figure 1)];

E is the modulus of elasticity [29,000,000 psi (200,000 MPa)];

I_y is the weak axis second moment of inertia expressed in in.⁴ (mm⁴);

G_τ is the shear modulus [11,200,000 psi (77,200 MPa)];

S_x is the section modulus expressed in in.³ (mm³).

0.3 S_y is for a single rolled section.

0.25 S_y is for unconventional construction or built-up sections.

It is advised that the equation for lateral torsional buckling only applies to I-shaped beams. If boxed or double-webbed beams are used, other detailed analyses may be necessary.

5.7.3 Horseheads

Horseheads shall be either hinged or removable to provide access for well servicing and shall be attached to the walking beam in such a manner as to prevent detachment in event of a high rod failure or other sudden load changes.

The distance from the pivot point of the horsehead to the tangent point of the wire line on the horsehead shall have a maximum dimensional tolerance at any position of the stroke of the following values:

- a) $\pm 1/2$ in. (± 12.7 mm) for stroke lengths up to 100 in. (2540 mm);
- b) $\pm 5/8$ in. (± 15.9 mm) for stroke lengths from 100 in. (2540 mm) up to 200 in. (5080 mm);
- c) $\pm 3/4$ in. (± 19 mm) for stroke lengths of 200 in. (5080 mm) and longer.

5.7.4 Cranks

All combined stresses in cranks resulting from operational loads shall be limited to a maximum value of $0.15S_y$.

5.8 Structural Bearing Design

5.8.1 General

Structural bearing shafts shall be supported in sleeve or antifriction bearings.

5.8.2 Antifriction Bearings

For bearings subject to oscillation or rotation, the bearing load ratio R_1 shall be determined using Equation (3) but shall not be less than the minimum values given below. For bearings subject to only oscillation, R_1 shall be 2.0 or greater. For bearings subject to full rotation, R_1 shall be 2.25 or greater:

$$R_1 = k \frac{C_b}{W_1} \quad (3)$$

where

R_1 is the bearing load ratio;

k is a bearing rating factor:

$k = 1.0$ for bearings rated at $33^{1/3}$ rpm and 500 h;

$k = 3.86$ for bearings rated at 500 rpm and 3000 h;

C_b is the bearing manufacturer's specific dynamic rating expressed in lb (N);

W_1 is the maximum load on bearing expressed in lb (N).

5.8.3 Sleeve Bearings

The design of sleeve bearings is outside the scope of this specification. The pumping unit manufacturer shall design sleeve bearings based on available test data and field experience that are comparable in performance to antifriction bearings designed for the same operating loads and speeds.

5.9 Brakes

Pumping unit brakes shall have sufficient braking capacity to withstand the torque exerted by the cranks at any crank position with the maximum amount of counterbalance torque designed by the manufacturer for the particular unit involved. This braking torque shall be effective with the pumping unit at rest under normal operating conditions with the well disconnected. The pumping unit brake is not intended as a safety stop; it is intended for operational stops only.

NOTE When operations or maintenance are to be conducted on or around a pumping unit, it is recommended that the position of the crank arms and counter weights be securely fixed in a stationary position by chains or by other acceptable means.

6 Gear Reducer Requirements

6.1 General

Gear reducers for beam pumping units shall be designed for the unusual external loads encountered in this service. All components are subject to loading determined by the structural geometry and the load rating of the pumping unit. The data in this section are general in nature and should only be used after careful consideration of all factors that influence the loading.

Reducers rated under this specification and properly applied, installed, lubricated, and maintained shall be capable of safely carrying the rated peak torque under normal oil field conditions. Requirements for beam pump gear reducers are specified in the following sections.

6.2 Gear Reducers

6.2.1 General

Gear reducers typically consist of a set of gears enclosed in a housing located between the prime mover and the cranks to transmit rotary power while reducing speed and increasing torque.

The gear rating equations contained in this specification apply only to gear elements possessing involute tooth form geometry.

6.2.2 Standard Sizes, Peak Torque Ratings, and Speed

The pumping unit reducer of a given size shall have a capacity, calculated as provided herein, as near as practical to but not less than the corresponding peak torque rating in Table 2. Gear peak torque ratings shall be based on a nominal pumping speed (strokes per minute)—see Table 3.

Table 2—Pumping Unit Reducer Sizes and Ratings

Size	Peak Torque Rating in.-lb (Nm)
6.4	6400 (723)
10	10,000 (1130)
16	16,000 (1808)
25	25,000 (2825)
40	40,000 (4519)
57	57,000 (6440)
80	80,000 (9040)
114	114,000 (12,880)
160	160,000 (18,080)
228	228,000 (25,800)
320	320,000 (36,200)
456	456,000 (51,500)
640	640,000 (72,300)
912	912,000 (103,000)
1280	1,280,000 (144,600)
1824	1,824,000 (206,000)
2560	2,560,000 (289,000)
3648	3,648,000 (412,000)

Table 3—Speeds for Peak Torque Rating for Gear Reducers

Strokes per Minute	Peak Torque Rating in.-lb (Nm)
20	320,000 (36,200) and smaller
16	456,000 (51,500)
16	640,000 (72,300)
15	912,000 (103,00)
14	1,280,000 (144,600)
13	1,824,000 (206,000)
11	2,560,000 (289,000) and larger

6.2.3 Rating Factors

6.2.3.1 General

The allowable stresses in this specification are maximum allowed values. Less conservative values for other rating factors in the document shall not be used.

6.2.3.2 Peak Torque Rating

6.2.3.2.1 General

The peak torque rating of the gear reducer is the lower of the pitting resistance torque rating, bending strength torque rating, or static torque ratings as determined by the use of the equations in this section.

6.2.3.2.2 Pitting Resistance Torque Rating

Pitting is considered to be a fatigue phenomenon and is a function of the stresses at the tooth surface.

The two kinds of pitting, initial pitting and destructive pitting, are illustrated in AGMA 1010-E95.

The aim of the pitting resistance equation is to determine a load rating at which destructive pitting of the teeth does not occur during their design life.

Equation (4) or the equivalent Equation (17) shall be used for rating the pitting resistance of gears:

$$T_{ac} = C_1 C_2 C_3 \quad (4)$$

where

T_{ac} is the allowable transmitted torque at output shaft, based on pitting resistance expressed in in.-lb (Nm);

C_1 is the pitting velocity factor, Equation (5);

C_2 is the pitting contact width factor, Equation (8);

C_3 is the pitting stress for external helical gears, Equation (11).

The pitting velocity factor is given by:

In USC units:

$$C_1 = \frac{n_p d^2 C_5}{2n_o} \quad (5)$$

In SI units:

$$C_1 = \frac{n_p d^2 C_5}{2000n_o} \quad (5)$$

where

n_p is the pinion rotational speed, expressed in revolutions per minute;

n_o is the rotational speed of output shaft expressed in revolutions per minute and equal to the pumping speed expressed in strokes per minute;

d is the operating pitch diameter of the pinion. If the pinion is enlarged, d can be taken to be the outside diameter minus two standard addendums, expressed in inches (millimeters);

C_5 is the velocity factor for pitting resistance.

In USC units:

$$C_5 = \frac{78}{78 + \sqrt{v_t}} \quad (6)$$

In SI units:

$$C_5 = \frac{78}{78 + \sqrt{200v_t}} \quad (6)$$

where

v_t is the pitch-line velocity (do not use enlarged pinion pitch diameter) expressed in ft/min (m/s).

In USC units:

$$v_t = 0.262n_p d \quad (7)$$

In SI units:

$$v_t = \frac{\pi n_p d}{60,000} \quad (7)$$

where

d is the operating pitch diameter of pinion expressed in inches (millimeters).

The pitting contact width factor C_2 is given by:

$$C_2 = \frac{F}{C_m} k_h \quad (8)$$

where

F is the net face width of the narrowest of the mating gears, expressed in inches (millimeters). For herringbone or double helical gearing, the net face width is the sum of the face widths of each helix;

k_h is a factor applied to account for any uncorrected distortion due to hardening the gears. When gears are hardened after cutting and the profiles and leads are not corrected or otherwise processed to ensure high accuracy, the tooth distortion will affect load distribution. This makes it necessary to apply the distortion factor k_h ; $k_h = 1.0$ if no hardening has been undertaken, $k_h = 0.95$ when one element is hardened after cutting, and $k_h = 0.90$ when both elements are hardened after cutting;

C_m is a load distribution factor for pitting resistance given by Equation (9) and Equation (10), which for F up to 16 in. (406 mm) may be read from Figure 2.

In USC units:

$$C_m = 1.24 + 0.0312F \quad \text{for } F \leq 16 \text{ in.} \quad (9)$$

In SI units:

$$C_m = 1.24 + 0.00123F \quad \text{for } F \leq 406 \text{ mm} \quad (9)$$

In USC units:

$$C_m = F / (0.45F + 2.0) \quad \text{for } F > 16 \text{ in.} \quad (10)$$

In SI units:

$$C_m = 0.394F / (0.0177F + 2.0) \quad \text{for } F > 406 \text{ mm} \quad (10)$$

If deflections or other sources of misalignment are such that the values of C_m from Figure 2 do not represent the actual maldistribution of load across the face, it is recommended that the load distribution factor be calculated using AGMA 2001-D04 and AGMA 908-B89.

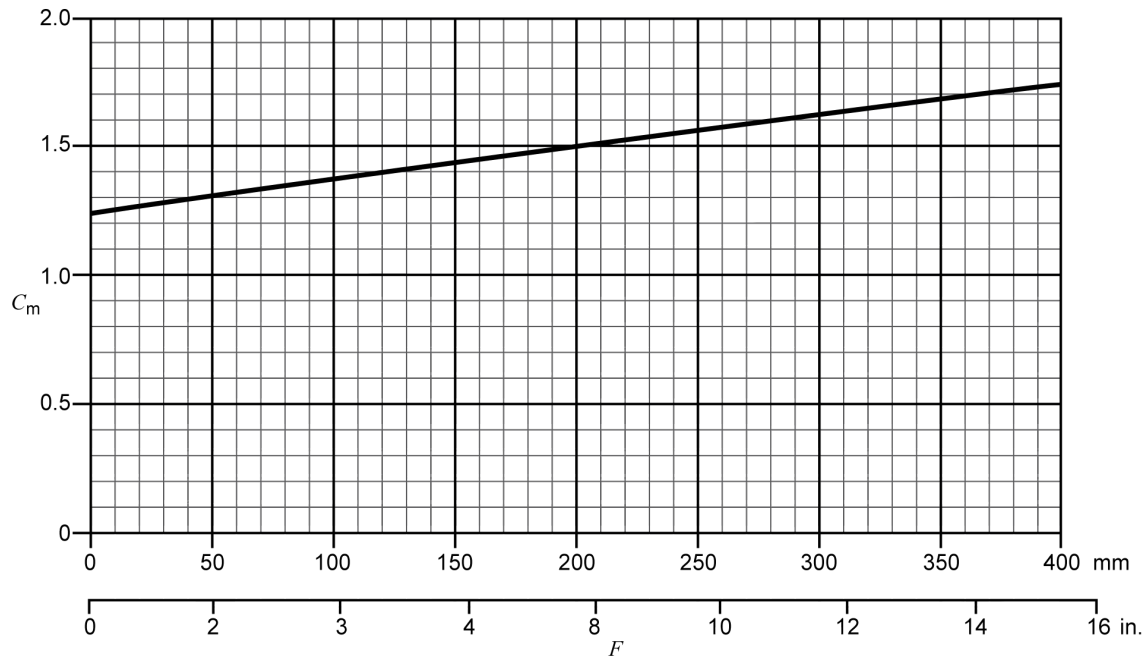


Figure 2—Helical Gear Load Distribution Factor, C_m , for Helical and Herringbone Gears and Well-controlled Heat-treating Processes

The C_2 values from Equation (8) can only be attained with well-controlled heat-treating processes. If the as-heat-treated accuracy is such that the required C_m values (for above C_2 values) cannot be attained, it is recommended that C_m be calculated in accordance with AGMA 908-B89.

The pitting stress for external helical gears C_3 is given by:

$$C_3 = 0.225 \left(\frac{m_g}{m_g + 1} \right) \left(\frac{S_{ac}}{C_p} \right)^2 \quad (11)$$

where

S_{ac} is the allowable contact stress expressed in psi (MPa) from Figure 3 or Table 4;

NOTE Recommended gear and pinion hardness combinations are given in Table 6.

C_p is the elastic coefficient (see Table 5);

m_g is the gear ratio $m_g = N_g/N_p$, where N_g is the number of teeth in gear and N_p is the number of teeth in pinion.

Table 4—Maximum Allowable Contact Stress Number— S_{ac} for Other Than Through Hardened and Tempered Steel Gears

Material	AGMA Class	Commercial Designation	Heat Treatment	Minimum Hardness at Surface	S_{ac} psi (MPa)
Steel			Flame or induction hardened ^a	50 HRC	170,000 (1,170)
				54 HRC	175,000 (1,205)
			Carburized and case-hardened ^b	55 HRC	180,000 (1,240)
				60 HRC	200,000 (1,380)
		AISI 4140	Nitrided ^c	48 HRC	155,000 (1,070)
		AISI 4340	Nitrided ^c	46 HRC	155,000 (1,070)
Cast iron	20		As cast	—	57,000 (395)
	30		As cast	175 BHN	70,000 (480)
	40		As cast	200 BHN	80,000 (550)
Nodular (ductile) iron	A-7-a	60-40-18	Annealed	140 BHN	90 % to 100 % ^d of S_{ac} value of steel with same hardness (see Figure 4)
	A-7-c	80-55-06	Quenched and tempered	180 BHN	
	A-7-d	100-70-03	Quenched and tempered	230 BHN	
	A-7-e	120-90-02	Quenched and tempered	270 BHN	
		120-90-02 mod	Quenched and tempered	300 BHN	
Malleable Iron (pearlitic)	A-8-c	45007		165 BHN	68,000 (470)
	A-8-e	50005	—	180 BHN	74,000 (510)
	A-8-f	53007	—	195 BHN	79,000 (545)
	A-8-i	80002	—	240 BHN	89,000 (615)

^a Minimum effective case depth requirements are given in 6.2.6.

^b Minimum effective case depth requirements are given in Figure 9.

^c Minimum effective case depth requirements are given in Figure 10.

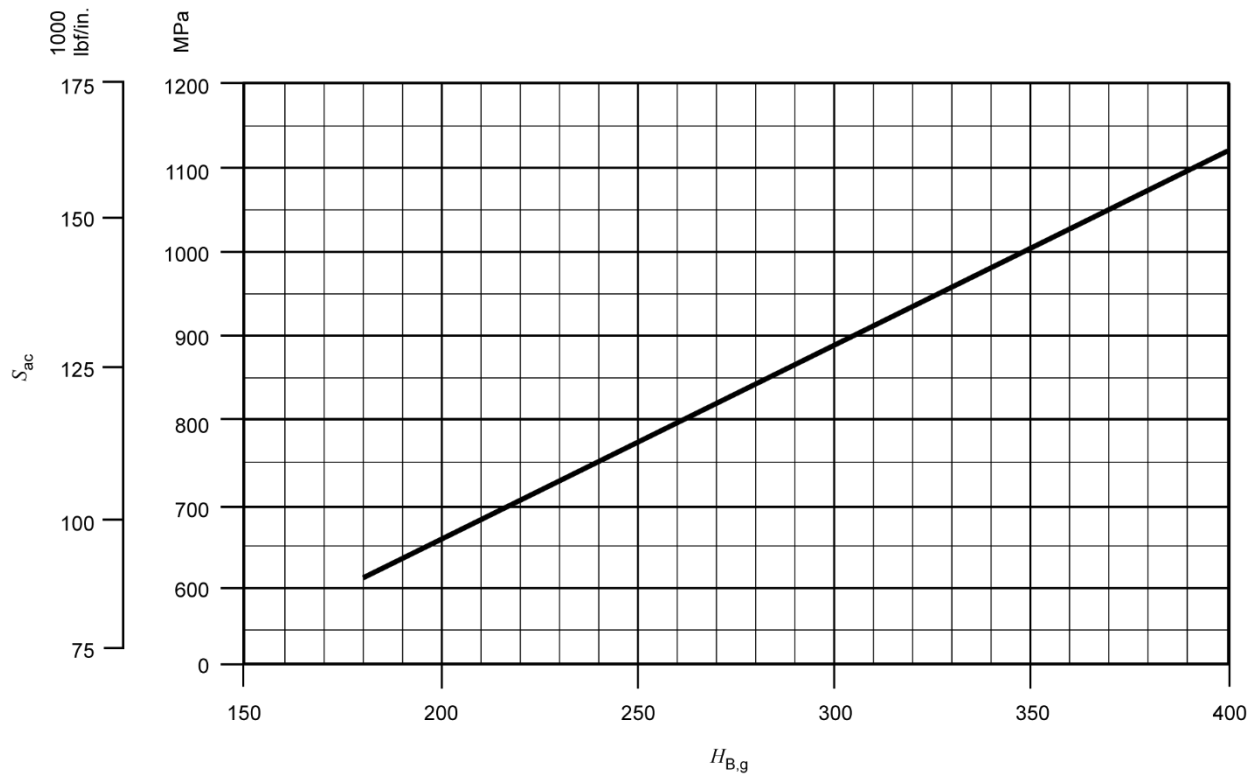
^d The higher allowable stress for nodular iron is determined by metallurgical controls as defined by the manufacturer.

Table 5—Elastic Coefficient C_p for Gear/Pinion Material Combinations

Pinion		Gear Materials and Modulus of Elasticity, E_g , psi			
Materials	Modulus of Elasticity E_p	Steel 30×10^6 psi (20×10^4 MPa)	Malleable Iron 25×10^6 psi (17×10^4 MPa)	Nodular Iron 24×10^6 psi (17×10^4 MPa)	Cast Iron 22×10^6 psi (15×10^4 MPa)
Steel	30×10^6 psi (20×10^4 MPa)	2300	2180	2160	2100
Malleable iron	25×10^6 psi (17×10^4 MPa)	2180	2090	2070	2020
Nodular iron	24×10^6 psi (17×10^4 MPa)	2160	2070	2050	2000
Cast iron	22×10^6 psi (15×10^4 MPa)	2100	2020	2000	1960

Table 6—Minimum Gear and Pinion Brinell Hardness Combinations for Through Hardened and Tempered Steel Gears

Gear $H_{B,g}$	Pinion $H_{B,p}$
180	210
210	245
225	265
245	285
255	295
270	310
285	325
300	340
335	375
350	390
375	415

**Figure 3—Allowable Contact Fatigue Stress for Through Hardened and Tempered Steel Gears S_{ac} for Helical and Herringbone Gears**

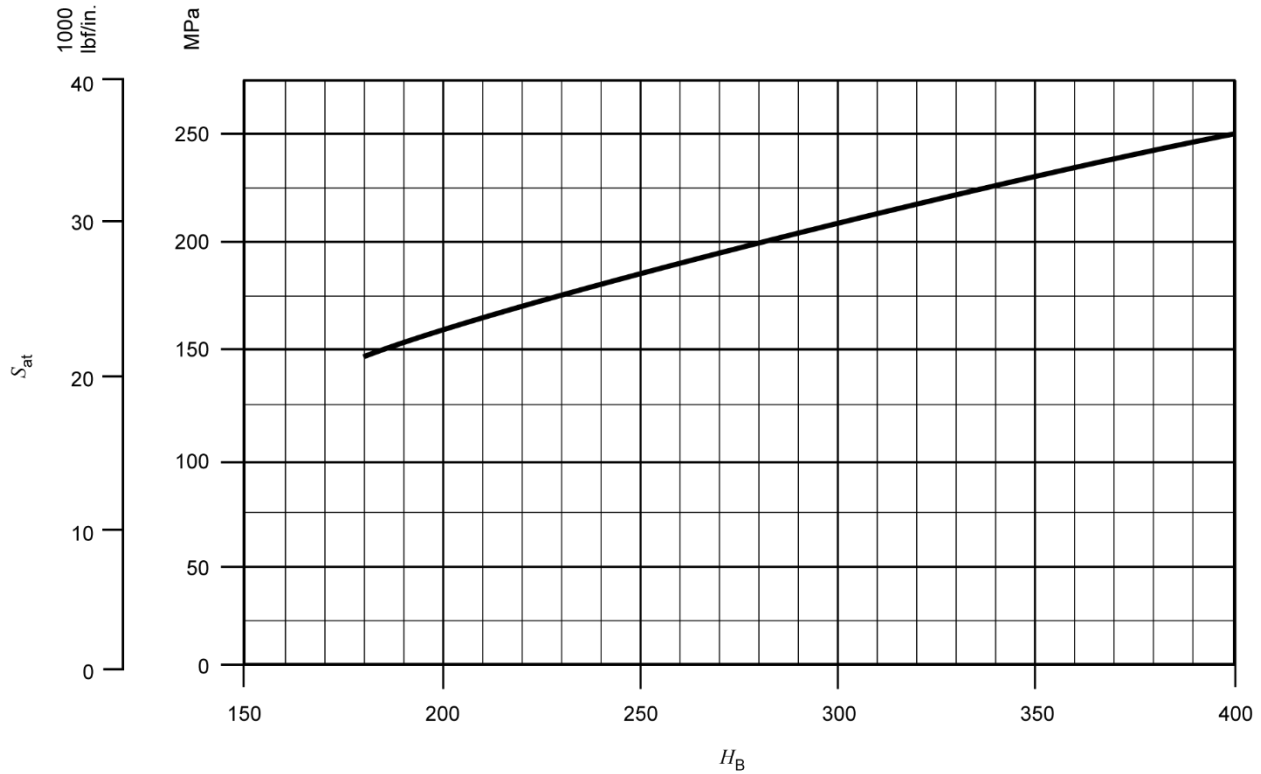


Figure 4—Allowable Bending Fatigue Stress for Through Hardened and Tempered Steel Gears S_{at}

The values of C_3 determined from Equation (11) are minimums for acceptable gear design. C_3 may be determined more precisely as follows:

$$C_3 = I_p \left(\frac{S_{ac}}{C_p} \right)^2 \quad (12)$$

where

I_p is a geometry factor for pitting resistance (wear) given by Equation (13):

$$I_p = \left(\frac{\cos \phi_t \sin \phi_t}{2} \right) \left(\frac{m_g}{m_g + 1} \right) \left(\frac{L_{min}}{F} \right) \quad (13)$$

where

L_{min} is the minimum total length of lines of contact in contact zone, expressed in inches (millimeters);

F is the net face width of the narrowest element, expressed in inches (millimeters);

ϕ_t is the operating transverse pressure angle, expressed in degrees.

$$\phi_t = \tan^{-1} \left(\frac{\tan \phi_n}{\cos \psi_g} \right) \quad (14)$$

where

ϕ_n is the normal operating pressure angle, expressed in degrees;

ψ_g is the operating helix angle, expressed in degrees.

For most helical gears having a face contact ratio of 2 or more, a conservative estimate of $L_{\min}/F$ is

$$\frac{L_{\min}}{F} = \frac{0.95Z}{p_N} \quad (15)$$

where

Z is the length of line of action in the transverse plane, expressed in inches (millimeters);

p_N is the normal base pitch, expressed in inches (millimeters).

With acceptable gear design, the above value of $L_{\min}/F$ is acceptable for a face contact ratio of 1.0 to 2.0. Equation (16) incorporates the expansion of I_p into a more precise equation for C_3 as:

$$C_3 = \left(\frac{\cos \phi_i \sin \phi_t}{2} \right) \left(\frac{m_g}{m_g + 1} \right) \left(\frac{0.95z}{p_N} \right) \left(\frac{S_{ac}}{C_p} \right)^2 \quad (16)$$

The method used in this specification for determining the geometry factors for pitting resistance I_p is simplified. A more precise and detailed analysis may be made using the method in AGMA 2001-D04 and AGMA 908-B89. The more precise method mentioned previously shall be used for face contact ratios less than 1.0. When I is determined in accordance with AGMA 2001-D04 and AGMA 908-B89 and if $2C_s/(m_g + 1)$ is not equal to the outside diameter minus two standard addendums, the operating pitch diameter of the pinion in all of the preceding rating equations shall be defined in accordance with AGMA 2001-D04 and AGMA 908-B89.

Incorporating the equations for C_1 , C_2 , and C_3 into Equation (4) gives the following Equation (17) for T_{ac} :

In USC units:

$$T_{ac} = \left(\frac{n_p d^2 C_5}{2n_o} \right) \left(\frac{F}{C_m} k_h \right) I_p \left(\frac{S_{ac}}{C_p} \right)^2 \quad (17)$$

In SI units:

$$T_{ac} = \left(\frac{n_p d^2 C_5}{2000n_o} \right) \left(\frac{F}{C_m} k_h \right) I_p \left(\frac{S_{ac}}{C_p} \right)^2 \quad (17)$$

or

In USC units:

$$T_{ac} = \left(\frac{n_p d^2 C_5}{2n_o} \right) \left(\frac{F}{C_m} k_h \right) \left(\frac{\cos \phi_t \sin \phi_t}{2} \right) \left(\frac{m_g}{m_g + 1} \right) \left(\frac{0.95z}{p_N} \right) \left(\frac{S_{ac}}{C_p} \right)^2$$

In SI units:

$$T_{ac} = \left(\frac{n_p d^2 C_5}{2000n_o} \right) \left(\frac{F}{C_m} k_h \right) \left(\frac{\cos \phi_t \sin \phi_t}{2} \right) \left(\frac{m_g}{m_g + 1} \right) \left(\frac{0.95z}{p_N} \right) \left(\frac{S_{ac}}{C_p} \right)^2$$

6.2.3.2.3 Bending Strength Torque Rating

Bending strength rating is related to fracture at the gear tooth root fillet. Fracture in this area is considered to be a fatigue phenomenon and is a function of the bending stress in the tooth as a cantilever plate.

Typical fractures are illustrated in AGMA 1010-E95.

The aim of the bending strength rating equation is to determine a load rating at which tooth root fillet fracture does not occur during the anticipated design life of the teeth.

The following Equation (18) or the expanded Equation (27) shall be used for rating the bending strength of helical and herringbone gears:

$$T_{at} = K_1 K_2 K_4 S_{at} \quad (18)$$

where

T_{at} is the allowable transmitted torque at output shaft based on bending strength expressed in in.-lb (Nm);

K_1 is the strength velocity factor; see Equation (19);

K_2 is the strength contact number; see Equation (22);

K_4 is the strength geometry number; see Equation (25).

S_{at} is the allowable bending stress from below or Table 7;

The strength velocity factor K_1 is given by Equation (19):

In USC units:

$$K_1 = \frac{n_p d K_5}{2n_o} \quad (19)$$

In SI units:

$$K_1 = \frac{n_p d K_5}{2000n_o} \quad (19)$$

Table 7—Allowable Bending Fatigue Stress, S_{at} (for Other Than Through Hardened and Tempered Steel Gears)

Material	AGMA Class	Commercial Designation	Heat Treatment	Minimum Hardness at Surface ^a	S_{at} psi (MPa)
Steel			Flame or induction hardened ^b	50–54 HRC	38,300 (264)
			Carburized and case-hardened ^c	55 HRC	47,000 (325)
				60 HRC	47,000 (325)
		AISI 4140	Nitrided ^d	48 HRC	29,000 (200)
		AISI 4340	Nitrided	46 HRC	31,000 (215)
Cast iron	20		As cast	—	4200 (30)
	30		As cast	175 BHN	7200 (50)
	40		As cast	200 BHN	11,000 (75)
Nodular (ductile) iron	A-7-a	60-40-18	Annealed	140 BHN	90 % to 100 % ^e of S_{at} value of steel with same hardness
	A-7-c	80-55-06	Quenched and tempered	180 BHN	
	A-7-d	100-70-03	Quenched and tempered	230 BHN	
	A-7-e	120-90-02	Quenched and tempered	270 BHN	
		120-90-02 mod	Quenched and tempered	300 BHN	
Malleable Iron (pearlitic)	A-8-c	45007	—	165 BHN	8500 (60)
	A-8-e	50005	—	180 BHN	11,000 (75)
	A-8-f	53007	—	195 BHN	13,600 (95)
	A-8-i	80002	—	240 BHN	17,900 (125)

^a Core hardness for nitrided gears to be a minimum of 300 BHN. Core hardness for case-hardened and ground gears and pinions to be shown in Manufacturer's Gear Reducer Datasheet (Figure C.3).

^b For minimum flame or induction-hardened hardening pattern, see Figure 8. Pattern 8A is limited to approximately 5DP (5 mm) and finer. Process control is important to the achievement of correct hardening pattern. Parts of this type should be carefully reviewed since residual compressive stresses are less than with pattern 8B. Tooth distortion and lack of ductility may necessitate a reduction of allowable stress numbers.

^c Minimum effective case depth requirements are given in Figure 9.

^d Minimum effective case depth requirements are given in Figure 10.

^e The higher allowable stress for nodular iron is determined by metallurgical controls.

where

- n_p is the pinion speed, expressed in rotations per minute;
- d is the operating pitch-diameter of pinion, expressed in inches (millimeters);
- n_o is the speed of output shaft, pumping speed, expressed in strokes per minute;
- K_5 is the velocity factor for bending strength.

In USC units:

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{v_t}}} \quad (20)$$

In SI units:

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{200v_t}}} \quad (20)$$

where

- v_t is the pitch-line velocity, expressed in ft/min (m/s).

In USC units:

$$v_t = 0.262n_p d \quad (21)$$

In SI units:

$$v_t = \frac{\pi n_p d}{60,000} \quad (21)$$

The strength contact number K_2 is given by Equation (22):

$$K_2 = \frac{F}{K_m} k_h \quad (22)$$

where

- F is the face width of the narrowest of the mating gears. For herringbone or double helical gearing, the net face width is the sum of the face width of each helix, expressed in inches (millimeters);
- k_h is a factor applied to account for any uncorrected distortion due to hardening the gears. When gears are hardened after cutting and the profiles and leads are not corrected or otherwise processed to ensure high accuracy, the tooth distortion will affect load distribution; $k_h = 1.0$ if no hardening has been undertaken, $k_h = 0.95$ when one element is hardened after cutting, and $k_h = 0.90$ when both elements are hardened after cutting;
- K_m is the load distribution factor from Equation (23) to Equation (24) and shown in Figure 5 for $F \leq 16$ in. ($F \leq 406$ mm).