Specification for Pumping Units

API SPECIFICATION 11E
NINETEENTH EDITION, NOVEMBER 2013

EFFECTIVE DATE: MAY 1, 2014

ERRATA, AUGUST 2015

AMERICAN PETROLEUM INSTITUTE

Special Notes

API publications necessarily address problems of a general nature. With respect to particular circumstances, local, state, and federal laws and regulations should be reviewed.

Neither API nor any of API's employees, subcontractors, consultants, committees, or other assignees make any warranty or representation, either express or implied, with respect to the accuracy, completeness, or usefulness of the information contained herein, or assume any liability or responsibility for any use, or the results of such use, of any information or process disclosed in this publication. Neither API nor any of API's employees, subcontractors, consultants, or other assignees represent that use of this publication would not infringe upon privately owned rights.

API publications may be used by anyone desiring to do so. Every effort has been made by the Institute to assure the accuracy and reliability of the data contained in them; however, the Institute makes no representation, warranty, or guarantee in connection with this publication and hereby expressly disclaims any liability or responsibility for loss or damage resulting from its use or for the violation of any authorities having jurisdiction with which this publication may conflict.

API publications are published to facilitate the broad availability of proven, sound engineering and operating practices. These publications are not intended to obviate the need for applying sound engineering judgment regarding when and where these publications should be utilized. The formulation and publication of API publications is not intended in any way to inhibit anyone from using any other practices.

Any manufacturer marking equipment or materials in conformance with the marking requirements of an API standard is solely responsible for complying with all the applicable requirements of that standard. API does not represent, warrant, or guarantee that such products do in fact conform to the applicable API standard.

Foreword

Nothing contained in any API publication is to be construed as granting any right, by implication or otherwise, for the manufacture, sale, or use of any method, apparatus, or product covered by letters patent. Neither should anything contained in the publication be construed as insuring anyone against liability for infringement of letters patent.

Shall: As used in a standard, “shall” denotes a minimum requirement in order to conform to the specification.

Should: As used in a standard, “should” denotes a recommendation or that which is advised but not required in order to conform to the specification.

This document was produced under API standardization procedures that ensure appropriate notification and participation in the developmental process and is designated as an API standard. Questions concerning the interpretation of the content of this publication or comments and questions concerning the procedures under which this publication was developed should be directed in writing to the Director of Standards, American Petroleum Institute, 1220 L Street, NW, Washington, DC 20005. Requests for permission to reproduce or translate all or any part of the material published herein should also be addressed to the director.

Generally, API standards are reviewed and revised, reaffirmed, or withdrawn at least every five years. A one-time extension of up to two years may be added to this review cycle. Status of the publication can be ascertained from the API Standards Department, telephone (202) 682-8000. A catalog of API publications and materials is published annually by API, 1220 L Street, NW, Washington, DC 20005.

Suggested revisions are invited and should be submitted to the Standards Department, API, 1220 L Street, NW, Washington, DC 20005, standards@api.org.

Contents

Contents

Contents

Page

Tables

Introduction

This specification under the jurisdiction of the API Executive Committee on Standardization and was developed with oversight from API Subcommittee 11 on Field Operating Equipment. This specification is intended to give requirements and information to both parties in the design, selection, and manufacture of beam pumping units. Furthermore, this specification addresses the minimum requirements with which the manufacturer is to comply so as to claim conformity with this specification.

Users of this specification should be aware that requirements above those outlined in this specification may be needed for individual applications. This specification is not intended to inhibit a manufacturer from offering, or the user/purchaser from accepting, alternative equipment or engineering solutions. This may be particularly applicable where there is innovative or developing technology. Where an alternative is offered, the manufacturer should identify any variations from this specification and provide details.

Annex A contains information on the application of the API Monogram for those organizations licensed to API Specification 11E. Forms are provided in Annex C for rating of crank counterbalances (Figure C.1) and for recording pumping unit stroke and torque factors (Figure C.2). Recommendations and examples for the calculation and application of torque factors are contained in Annex D to Annex G, and examples for calculating torque ratings for pumping unit reducers are contained in Annex H. Recommendations and considerations for conducting a system analysis are contained in Annex I. Annex J contains an illustration of a typical beam pumping unit and the nomenclature associated with it.

Specification for 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 speed reducer input shaft. This includes the following:

a) beam pump structures,

b) pumping unit gear reducer, and

c) pumping unit chain 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 speed 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.

See Annex A for product is supplied bearing the API Monogram and manufactured at a facility licensed by API.

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*

ASME B29.100 [3], *Precision Power Transmission, Double-Pitch Power Transmission, and Double-Pitch Conveyor Roller Chains, Attachments, and Sprockets*—Incorporating ASME B29.1, ASME B29.3, and ASME B29.4

[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 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 speed reducer output shaft.

3.1.2
beam pumping unit
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
Component of a pumping unit 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
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
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
Direction of rotation, either clockwise or counterclockwise as viewed from the side of the beam pumping unit with the horsehead to the right.

[3] ASME International, 3 Park Avenue, New York, New York 10016, www.asme.org.

3.1.9
cranks
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
hanger
See **carrier bar**.

3.1.13
horsehead
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.

3.1.14
pitmans
Connecting link in the pumping unit mechanism between the cranks and the equalizer.

3.1.15
saddle bearing
Primary fulcrum bearing supporting the walking beam of a conventional (Class 1 lever) pumping unit.

3.1.16
Samson post bearing
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.

3.1.17
speed reducer
Mechanism located between the belt drive and the cranks to transmit rotary power while reducing speed and increasing torque.

3.1.18
structural unbalance
B
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
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 speed 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.

A distance from the center of the saddle bearing to the centerline of the polished rod, in inches (in.) [millimeters (mm)]

A_t tensile area of fastener, in square inches (in.^{2}) [millimeters (mm^2)]

a area of cross section, in square inches (in.^{2}) [millimeters (mm^2)]

B structural unbalance, in pounds (lb) [newtons (N)]

C distance from the center of the saddle bearing to the center of the equalizer bearing, in inches (in.) [millimeters (mm)]

C_b bearing manufacturer's specific dynamic rating, in pounds (lb) [newtons (N)]

C_m load distribution factor for pitting resistance, unitless

C_p elastic coefficient, unitless

C_1 pitting velocity factor, unitless

C_2 pitting contact width factor, unitless

C_3 pitting stress for external helical gears, unitless

C_5 velocity factor for pitting resistance, unitless

CB counterbalance effect, in pounds (lb) [newtons (N)]

CD standard center distance between gear shafts, in inches (in.) [millimeters (mm)]

D operating pitch diameter of gear, in inches (in.) [millimeters (mm)]

D_m major diameter of fastener, in inches (in.) [millimeters (mm)]

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)

d_s shaft diameter (for tapered shaft use mean diameter), in inches (in.) [millimeters (mm)]

E modulus of elasticity, in pounds per square inch (psi) [megapascals (MPa)]

E_g modulus of elasticity for gears, in pounds per square inch (psi) [megapascals (MPa)]

E_p modulus of elasticity for pinions, in pounds per square inch (psi) [megapascals (MPa)]

F net face width, in inches (in.) [millimeters (mm)]

f_{cb} allowable compressive stress in bending, in pounds per square inch (psi) [megapascals (MPa)]

$f_{s,b}$ allowable stress due to bending, in pounds per square inch (psi) [megapascals (MPa)]

$f_{s,t}$ 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 or of large sprocket, in inches (in.) [millimeters (mm)]

R_1 bearing load ratio, unitless

r radius of gyration of section, in inches (in.) [millimeters (mm)]

S ultimate tensile strength of chain, in pounds (lb) [newtons (N)]

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)]

$T_{as,i}$ 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 compliance/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 15° 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 noncompliance with the stated performance.

4.2.5 Design Changes

The manufacturer shall, as 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 $0.2S_y$.

The following Equation (1) shall be used for all components acting as columns:

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

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} (mm^2);
- 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:

$$W = \frac{f_{cb}}{A} S_x \tag{2}$$

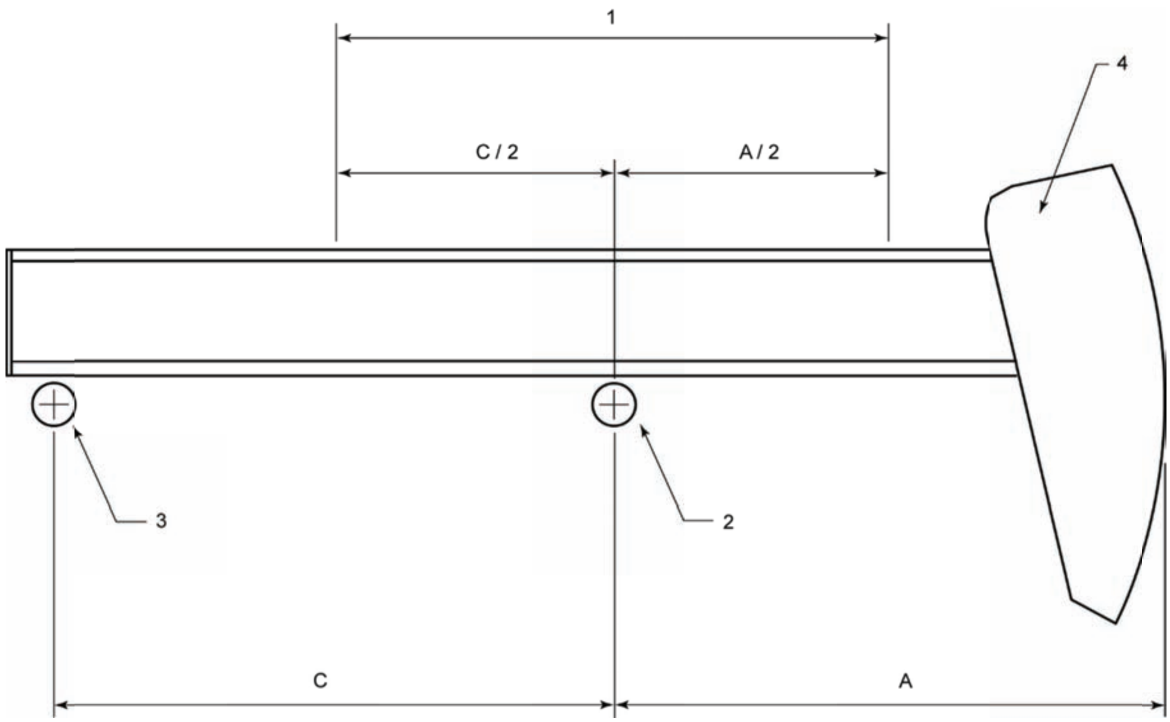
where

- 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} (mm^3);

A is the distance from centerline of saddle bearing to 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) is based on conventional beam construction using a single rolled section. The gross section of the rolled beam may 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).

With unconventional construction or built-up sections, consideration shall be given to changes in loading, to checking stresses at all critical sections and to the inclusion of stress concentration factors where applicable.

5.5 Maximum Allowable Stress for Walking Beams

The maximum allowable stress, f_{cb}, for the walking beam rating equation, Equation (2), shall be determined from Table 1. For standard rolled 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 Other Structural Components

5.6.1 Shafting

The limiting stresses for all bearing shafts as well as other structural shafting are given in 6.4.5.1.

5.6.2 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}	11,000 psi (75.84 MPa)
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}	11,000 psi (75.84 MPa)

where

J_t is the torsional constant of the beam section expressed in in.^{4} (mm^4);

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} (mm^4);

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

S_x is the section modulus expressed in in.^{3} (mm^3).

5.6.3 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.

5.6.4 Cranks

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

5.7 Structural Bearing Design

5.7.1 General

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

5.7.2 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:

$$R_1 = k\frac{C_b}{W_1} \tag{3}$$

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).

5.7.3 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.

5.8 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 but 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 Speed Reducer Requirements

6.1 General

Speed 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 speed reducers are specified in the following sections.

Included are the following types:

a) gear reducers,

b) chain reducers.

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.

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:

$$T_{ac} = C_1 C_2 C_3 \tag{4}$$

Table 2—Pumping Unit Reducer Sizes and Ratings

Size	**Peak Torque Rating** in.-lb (Nm)
6.4	6.400 (723)
10	10.000 (1.130)
16	16.000 (1.808)
25	25.000 (2.825)
40	40.000 (4.519)
57	57.000 (6.440)
80	80.000 (9.040)
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)
1.280	1.280.000 (144.600)
1.824	1.824.000 (206.000)
2.560	2.560.000 (289.000)
3.648	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

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:

$$C_1 = \frac{n_p d^2 C_5}{2 n_o} \tag{5}$$

In SI units:

$$C_1 = \frac{n_p d^2 C_5}{2000 n_o} \tag{5}$$

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:

$$C_5 = \frac{78}{78 + \sqrt{v_t}} \tag{6}$$

In SI units:

$$C_5 = \frac{78}{78 + \sqrt{200 v_t}} \tag{6}$$

where

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

In USC units:

$$v_t = 0.262 n_p d \tag{7}$$

In SI units:

$$v_t = \frac{\pi n_p d}{60{,}000} \tag{7}$$

where

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

The pitting contact width factor C_2 is given by:

$$C_2 = \frac{F}{C_m} k_h \tag{8}$$

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:

$$C_m = 1.24 + 0.0312F \qquad \text{for } F \le 16 \text{ in.} \tag{9}$$

In SI units:

$$C_m = 1.24 + 0.00123F \qquad \text{for } F \le 406 \text{ mm} \tag{9}$$

In USC units:

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

In SI units:

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

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, then it is recommended that the load distribution factor be calculated using AGMA 2001-D04 and AGMA 908-B89.

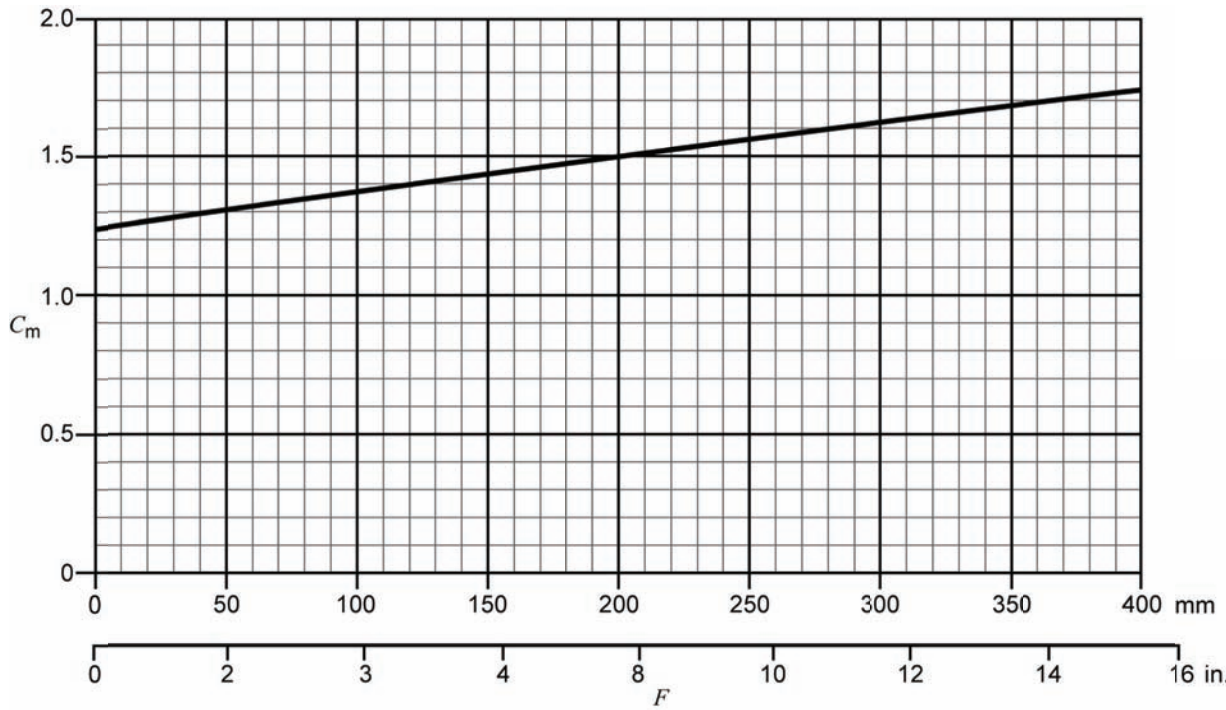


Figure 2—Helical Gear Load Distribution Factor, C_m, for Helical and Herringbone Gears and Well-controlled Heat-treating Processes

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:

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

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 5.2.6.

[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

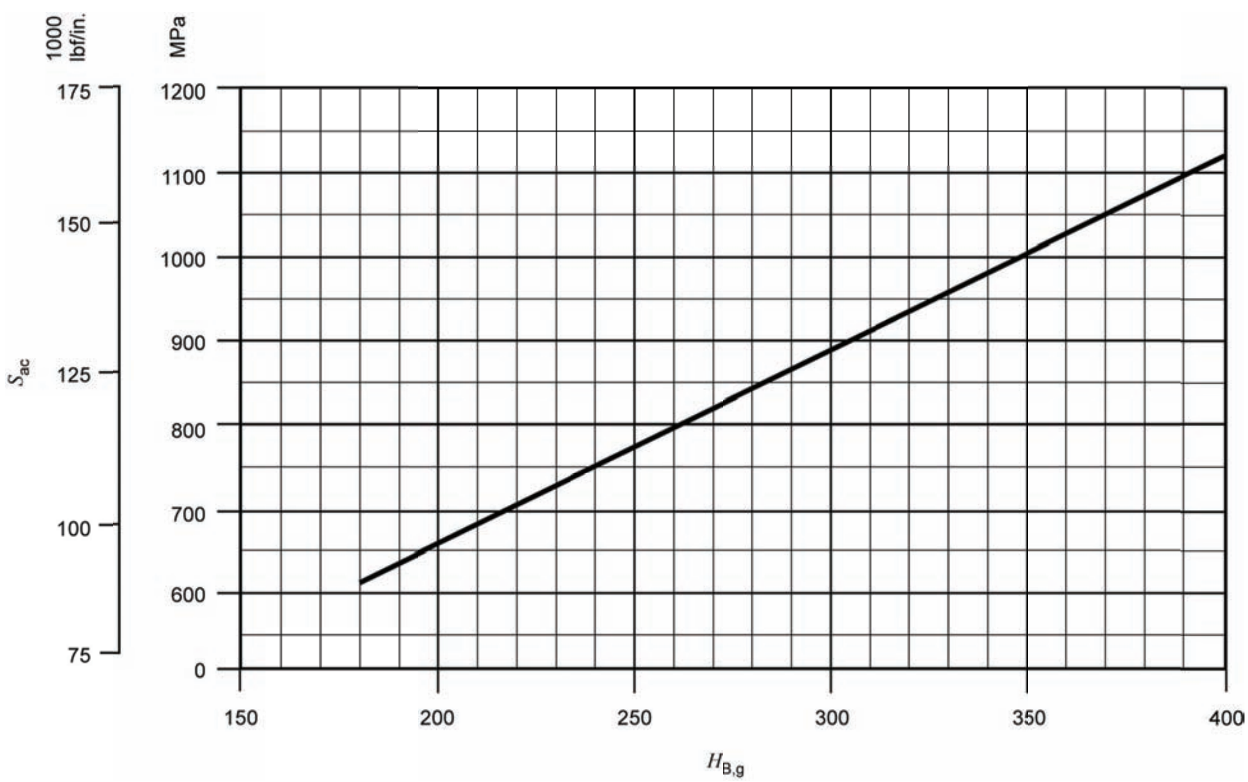


Figure 3—Allowable Contact Fatigue Stress for Through Hardened and Tempered Steel Gears S_{ac} for Helical and Herringbone Gears

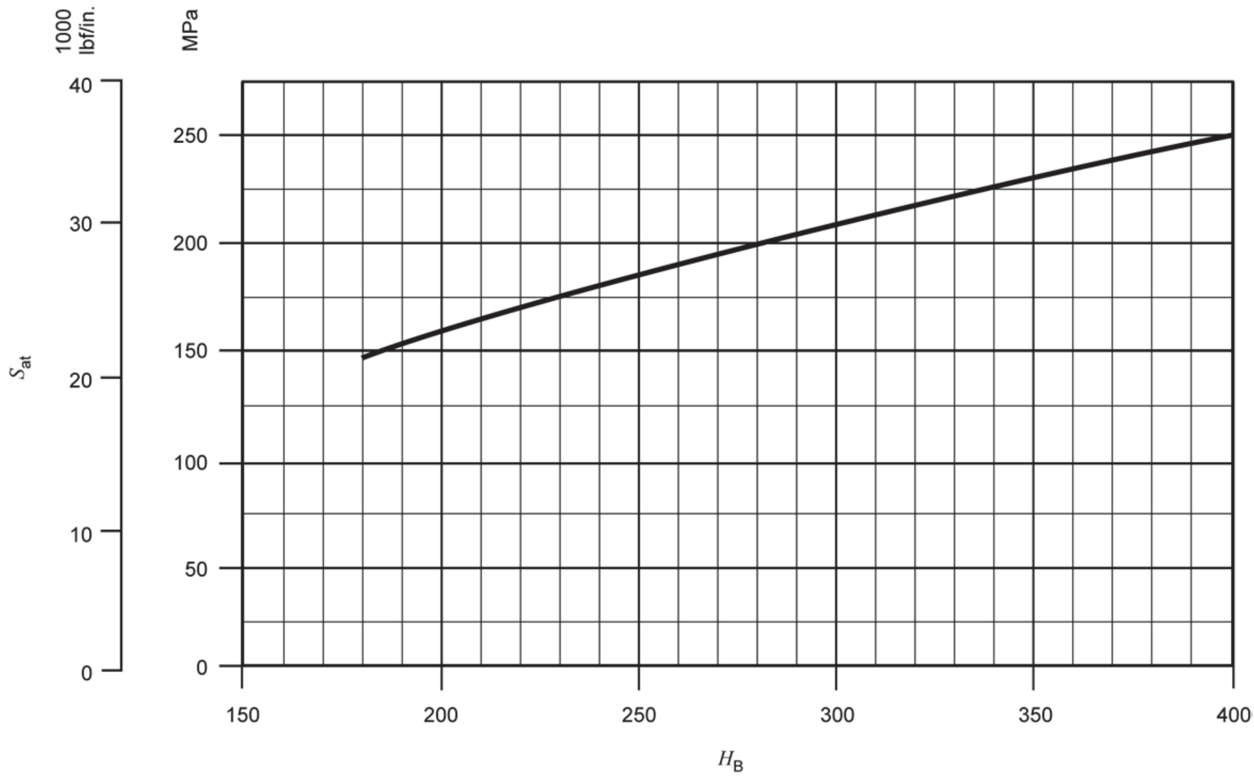


Figure 4—Allowable Bending Fatigue Stress for Through Hardened and Tempered Steel Gears S_{at}

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

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

where

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

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

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.

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

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

$$\frac{L_{min}}{F} = \frac{0.95Z}{p_N} \tag{15}$$

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:

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

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 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:

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

In SI units:

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

or

In USC units:

$$T_{\mathrm{ac}} = \left(\frac{n_{\mathrm{p}} d^2 C_5}{2 n_{\mathrm{o}}}\right)\left(\frac{F}{C_{\mathrm{m}}} k_{\mathrm{h}}\right)\left(\frac{\cos\phi_{\mathrm{t}} \sin\phi_{\mathrm{t}}}{2}\right)\left(\frac{m_{\mathrm{g}}}{m_{\mathrm{g}}+1}\right)\left(\frac{0.95 z}{p_{\mathrm{N}}}\right)\left(\frac{S_{\mathrm{ac}}}{C_{\mathrm{p}}}\right)^2$$

In SI units:

$$T_{\mathrm{ac}} = \left(\frac{n_{\mathrm{p}} d^2 C_5}{2000 n_{\mathrm{o}}}\right)\left(\frac{F}{C_{\mathrm{m}}} k_{\mathrm{h}}\right)\left(\frac{\cos\phi_{\mathrm{t}} \sin\phi_{\mathrm{t}}}{2}\right)\left(\frac{m_{\mathrm{g}}}{m_{\mathrm{g}}+1}\right)\left(\frac{0.95 z}{p_{\mathrm{N}}}\right)\left(\frac{S_{\mathrm{ac}}}{C_{\mathrm{p}}}\right)^2$$

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:

$$T_{\mathrm{at}} = K_1 K_2 S_{\mathrm{at}} K_4 \tag{18}$$

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);
- S_{at} is the allowable bending stress from below or Table 7;
- K_4 is the strength geometry number, see Equation (25).

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

In USC units:

$$K_1 = \frac{n_{\mathrm{p}} d K_5}{2 n_{\mathrm{o}}} \tag{19}$$

In SI units:

$$K_1 = \frac{n_{\mathrm{p}} d K_5}{2000 n_{\mathrm{o}}} \tag{19}$$

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	—	4,200 (30)
	30		As cast	175 BHN	7,200 (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	8,500 (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 (5mm) 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:

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{v_t}}} \tag{20}$$

In SI units:

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{200 v_t}}} \tag{20}$$

where

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

In USC units:

$$v_t = 0.262 n_p d \tag{21}$$

In SI units:

$$v_t = \frac{\pi n_p d}{60{,}000} \tag{21}$$

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

$$K_2 = \frac{F}{K_m} k_h \tag{22}$$

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).

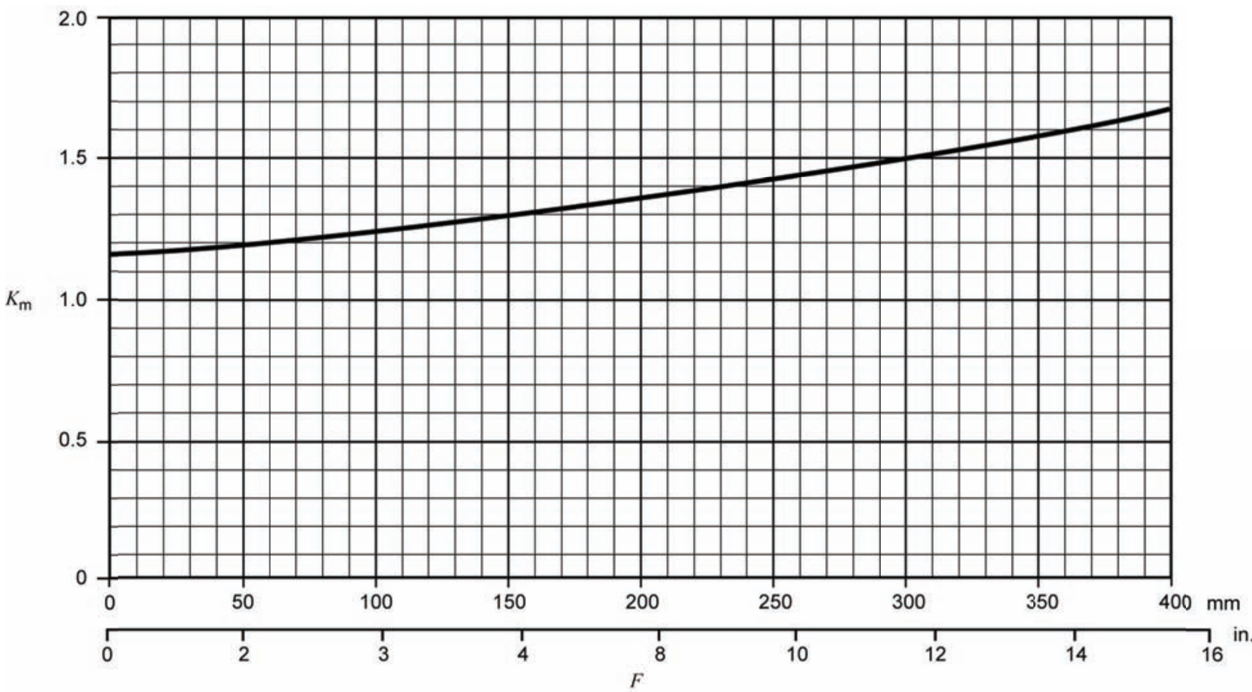


Figure 5—Helical Gear Load Distribution Factor, K_m, for Helical and Herringbone Gears

In USC units:

$$K_m = \frac{1}{0.872 - 0.0176F} \qquad \text{for } F \le 16 \text{ in.} \tag{23}$$

In SI units:

$$K_m = \frac{1}{0.872 - \left(6.93 \times 10^{-4}\right)F} \qquad \text{for } F \le 406 \text{ mm} \tag{23}$$

In USC units:

$$K_m = 1.7 \qquad \text{for } F > 16 \text{ in.} \tag{24}$$

In SI units:

$$K_m = 1.7 \qquad \text{for } F > 406 \text{ mm} \tag{24}$$

If deflection or other sources of misalignment are such that the values of K_m from Figure 5 do not represent the actual maldistribution of load across the face, then the load distribution factor should be calculated using AGMA 2001-D04 and AGMA 908-B89.

The K_2 values from Equation (22) can only be attained with well-controlled heat-treating processes. If the as-heat-treated accuracy is such that the required K_m values (for above K_2 values) cannot be attained, K_m should be calculated in accordance with AGMA 2001-D04 and AGMA 908-B89.

The strength geometry number K_4 is given by Equation (25):

In USC units:

$$K_4 = \frac{J_b}{P_d} \tag{25}$$

In SI units:

$$K_4 = J_b m \tag{25}$$

where

J_b is the geometry factor for bending strength in accordance with AGMA 908-B89;

P_d is the diametral pitch in plane of rotation (transverse);

m is the metric module in the plane of rotation (transverse), expressed in mm.

In USC units:

$$P_d = \frac{N_p}{d} = P_{nd} \cos \psi_g \tag{26}$$

In SI units:

$$m = \frac{d}{N_p} = \frac{m_n}{\cos \psi_g} \tag{26}$$

where

P_{nd} is the normal diametral pitch, nominal, expressed in in.$^{-1}$;

ψ_g is the operating helix angle, expressed in degrees;

m_n is the normal metric module, expressed in mm.

The bending strength rating shall be calculated for both pinion and gear. The lower of the two values is the bending strength rating of the gear set.

Incorporating the equations for K_1, K_2, and K_4 into Equation (4) gives the following Equation (27) for T_{at}:

In USC units:

$$T_{at} = \left(\frac{n_p d K_5}{2 n_o}\right)\left(\frac{F}{K_m} k_h\right)\left(S_{at}\right)\left(\frac{J_b}{P_d}\right) \qquad \text{(in.-lb)} \tag{27}$$

In SI units:

$$T_{at} = \left(\frac{n_p d K_5}{2000 n_o}\right)\left(\frac{F}{K_m} k_h\right)\left(S_{at}\right)(J_b m) \qquad \text{(Nm)} \tag{27}$$

where

In USC units:

$$S_{\text{at}} = 142H_{\text{B}} - 0.13H_{\text{B}}^2 \qquad \text{(psi) (see Figure 4)}$$

In SI units:

$$S_{\text{at}} = 0.979H_{\text{B}} - \left(8.96 \times 10^{-4}\right)H_{\text{B}}^2 \qquad \text{(MPa) (see Figure 4)}$$

6.2.3.2.4 Static Torque Rating

Static torque loads on the gear teeth are caused by resisting the torque exerted by the counterbalance or other nonoperating conditions. A description of the many conditions of installation, maintenance, and use of pumping unit reducers that can cause high static torques is not covered in this specification.

The static torque rating of the gear reducer to resist these loads shall be equal to or greater than 500 % of the reducer name plate rating. Certain pumping unit geometries may require a higher static torque rating. The system analysis (see Annex I) should be used to determine when a higher static torque rating is required.

The following Equation (28) shall be used to determine static torque rating of helical and herringbone gears:

In USC units:

$$T_{\text{as,i}} = \left(\frac{D}{2}\right)\left(\frac{J_{\text{b}}}{P_{\text{d}}}\right)\left(\frac{F}{K_{\text{ms}}}\right)S_{\text{ay}}K_{\text{y}} \tag{28}$$

In SI units:

$$T_{\text{as,i}} = \left(\frac{D}{2000}\right)J_{\text{b}}m\left(\frac{F}{K_{\text{ms}}}\right)S_{\text{ay}}K_{\text{y}} \tag{28}$$

where

$T_{\text{as,i}}$ is the allowable static torque at the gear or pinion being checked, expressed in in.-lb (Nm);

NOTE 1 $T_{\text{as,1}}$ = 1st reduction, $T_{\text{as,2}}$ = 2nd reduction, $T_{\text{as,n}}$ = nth reduction.

NOTE 2 Torque on output shaft may be calculated as $T_{\text{as,2}} = T_{\text{as,1}}(m_{\text{g2}})$, etc.

D is the operating pitch diameter of gear, expressed in inches (millimeters);

S_{ay} is the allowable yield strength of the gear or pinion material taken from Figure 6 for steel and nodular iron; for case hardened (flame, induction, nitrided, carburized) material, the core hardness from the gear manufacturer's datasheet (see Figure C.3) shall be used to determine yield strength, expressed in psi (MPa):

where

In USC units:

$$S_{ay} = 482 H_B - 32,800 \qquad \text{(see Figure 6)} \qquad (29)$$

In SI units:

$$S_{ay} = 3.323 H_B - 226 \qquad \text{(see Figure 6)} \qquad (29)$$

K_y is the yield strength factor from Table 8;

K_{ms} is the load distribution factor, static torque.

where

In USC units:

$$K_{ms} = 0.0144F + 1.07 \qquad \text{for } F \text{ measured in inches and } F \leq 16 \text{ in.} \qquad (30)$$

In SI units:

$$K_{ms} = \left(5.67 \times 10^{-4}\right) F + 1.07 \qquad \text{for } F \text{ measured in millimeters and } F \leq 406 \text{ mm} \qquad (30)$$

In USC units:

$$K_{ms} = 1.3 \qquad \text{for } F > 16 \text{ in.} \qquad (31)$$

In SI units:

$$K_{ms} = 1.3 \qquad \text{for } F > 406 \text{ mm} \qquad (31)$$

The allowable static torque rating determined using this formula is conservative since the geometry factor J_b includes a stress concentration factor for fatigue. It should be noted that some gear materials do not have a well-defined yield point and the ultimate strength is approximately equal to the yield. For these materials, a much lower value of K_y shall be selected. The user of this specification should satisfy themself that the yield values selected are appropriate for the materials used.

Table 8—Yield Strength Factor, K_y

Material	K_y
Steel (through hardened)	1.00
Nodular iron	1.00
Steel (flame or induction hardened)	0.85
Steel (case carburized)	1.20
Steel (nitrided)	0.85
Cast iron	0.75
Malleable iron	1.00

6.2.4 Metallurgy

The allowable stresses, S_{ac} and S_{at}, included in this specification are based on commercial ferrous material manufacturing practices. Hardness, tensile strength, and microstructure are the criteria for allowable stress values. Reasonable levels of cleanliness and metallurgical controls are required to permit the use of the allowable stress values contained in this specification.

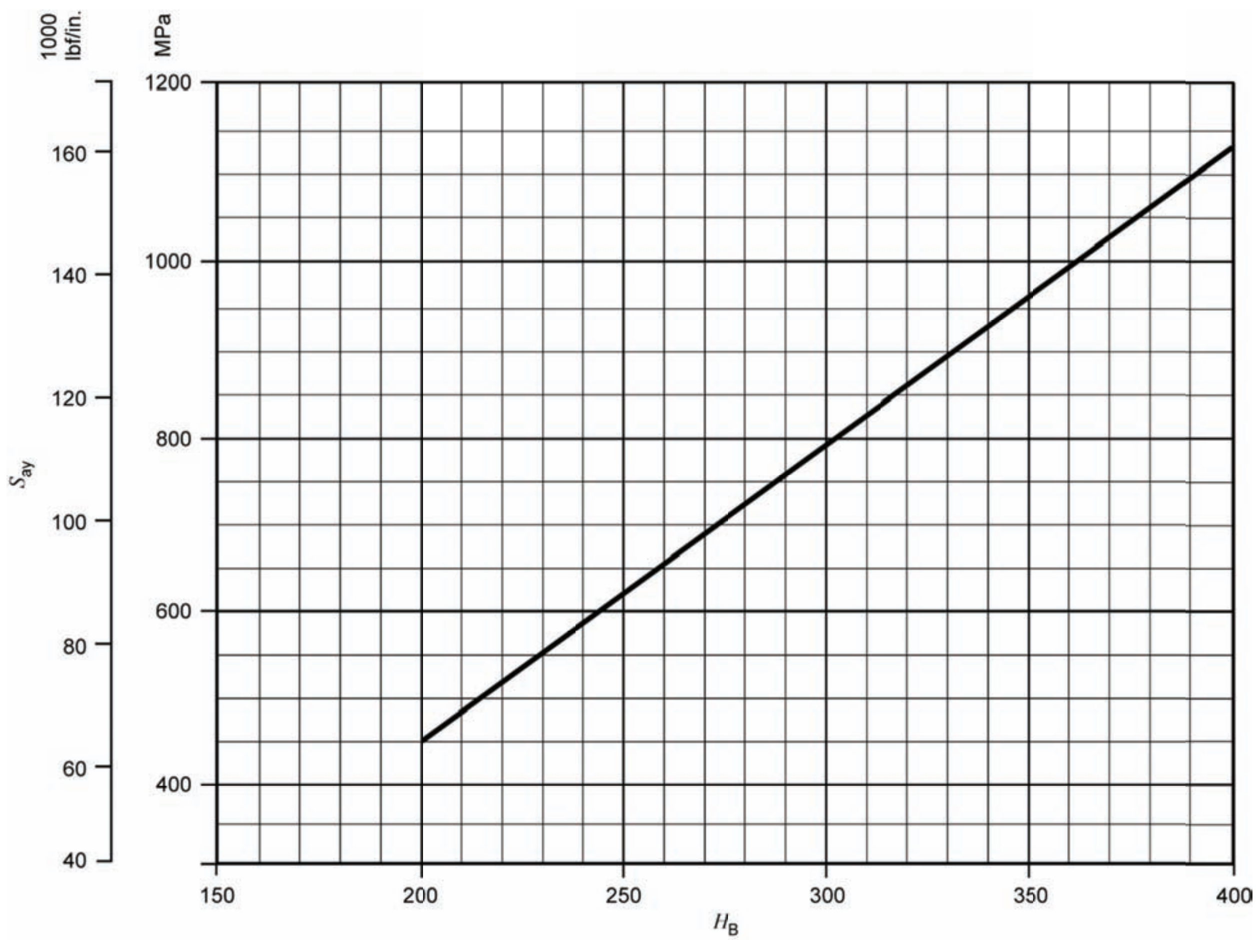


Figure 6—Allowable Yield Strength Number for Steel and Nodular Iron, S_{ay}

6.2.5 Residual Stress

Any material having a case-core relationship is likely to contain residual stresses. If properly managed, these residual stresses will be compressive and will enhance the bending strength performance of the gear teeth. Shot peening, case carburizing, nitriding, and induction hardening are common methods of inducing compressive prestress in the surface of the gear teeth.

Grinding the tooth surface after heat treatment can reduce the residual compressive stresses. Grinding the root fillet area can introduce tensile stresses in the root. Care shall be taken to avoid changes in microstructure during any grinding process. Shot peening may be performed after grinding to assure the presence of residual compressive stresses.

6.2.6 Minimum Effective Case Depths

6.2.6.1 General

The effective case depth is the depth of the case that has a minimum Rockwell C hardness of 50 Rc. The minimum effective case depth is a function of P_{nd}, the normal diametral pitch.

6.2.6.2 Flame and Induction Hardened Gears

The minimum effective case depth for flame or induction hardened gears shall be as defined in Equation (32), see Figure 7:

In USC units:

$$h_e = 0.264693 P_{nd}^{-1.12481} \quad \text{(in.)} \tag{32}$$

In SI units:

$$h_e = 0.17677 m_n^{1.12481} \quad \text{(mm)} \tag{32}$$

NOTE The minimum effective case depth is defined as the depth below the surface at which the Rockwell C hardness has dropped to 50 Rc or 5 points below the surface hardness, whichever is the lower hardness.

6.2.6.3 Carburized Gears

The minimum effective case depth in inches (millimeters) for carburized gears shall be in the ranges as defined in Equation (33) and Equation (34), see Figure 9:

In USC units:

$$h_{e,1} = 0.119935 P_{nd}^{-0.86105} \tag{33}$$

In SI units:

$$h_{e,1} = 0.188 m_n^{0.86105} \tag{33}$$

In USC units:

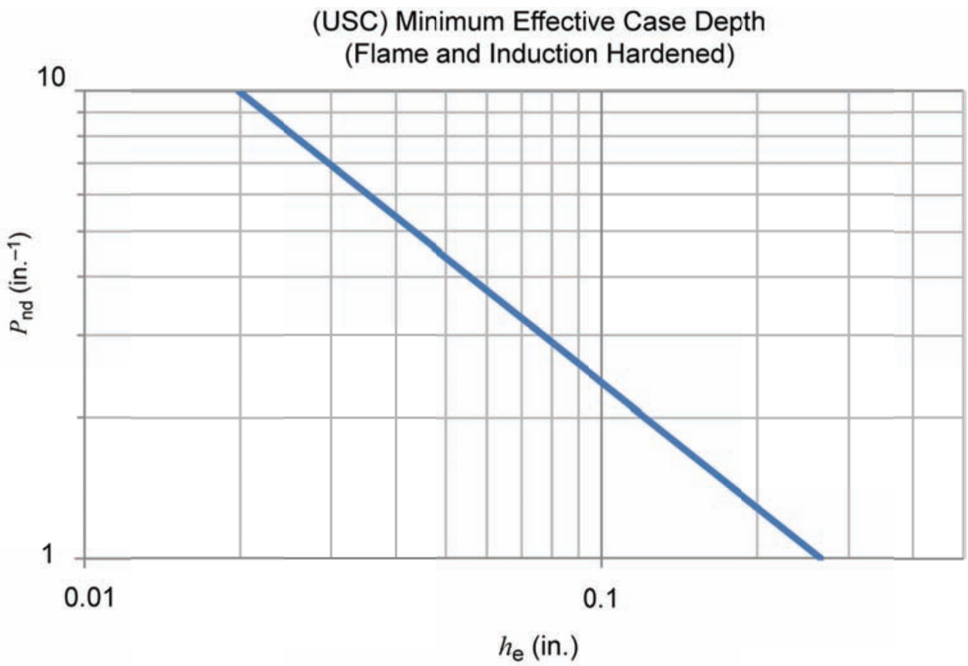
$$h_{e,2} = 0.264693 P_{nd}^{-1.12481} \tag{34}$$

In SI units:

$$h_{e,2} = 0.17677 m_n^{1.12481} \tag{34}$$

NOTE 1 The values and ranges shown on the case depth curves are to be used as guides. For gearing in which maximum performance is required, detailed studies must be made of the application, loading, and manufacturing procedures to obtain desirable gradients of both hardness and internal stress. Furthermore, the method of measuring the case as well as the allowable tolerance in case depth should be a matter of agreement between the customer and the manufacturer.

NOTE 2 The effective case depth is defined as the depth below the surface at which the Rockwell C hardness has dropped to 50 Rc. The total case depth to core carbon is approximately 1.5 × effective case depth.



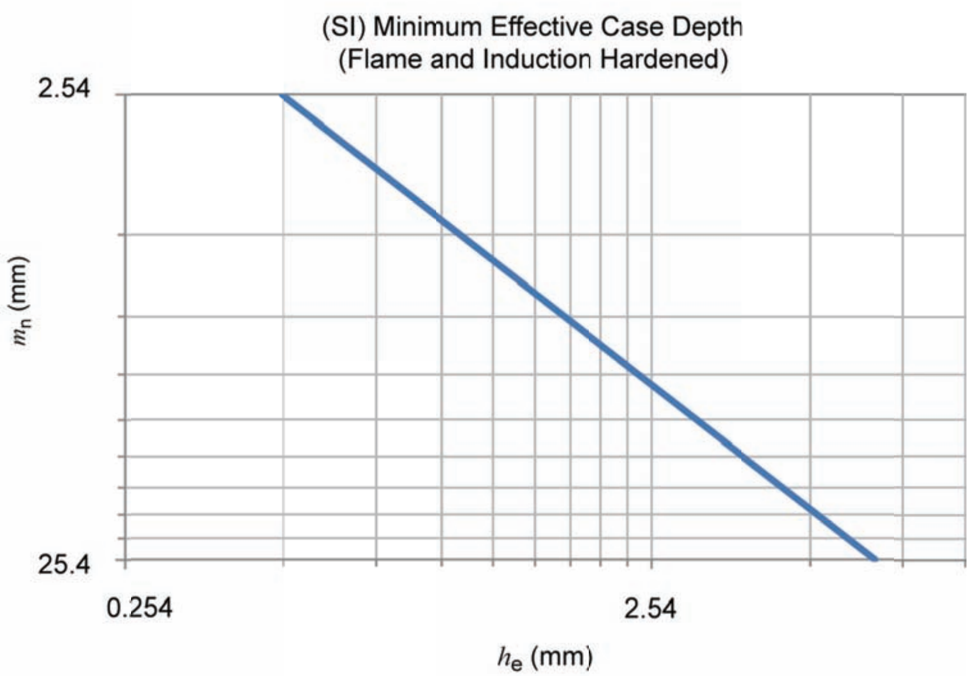


Figure 7—Minimum Effective Case Depth for Flame or Induction Hardened Gears, h_e

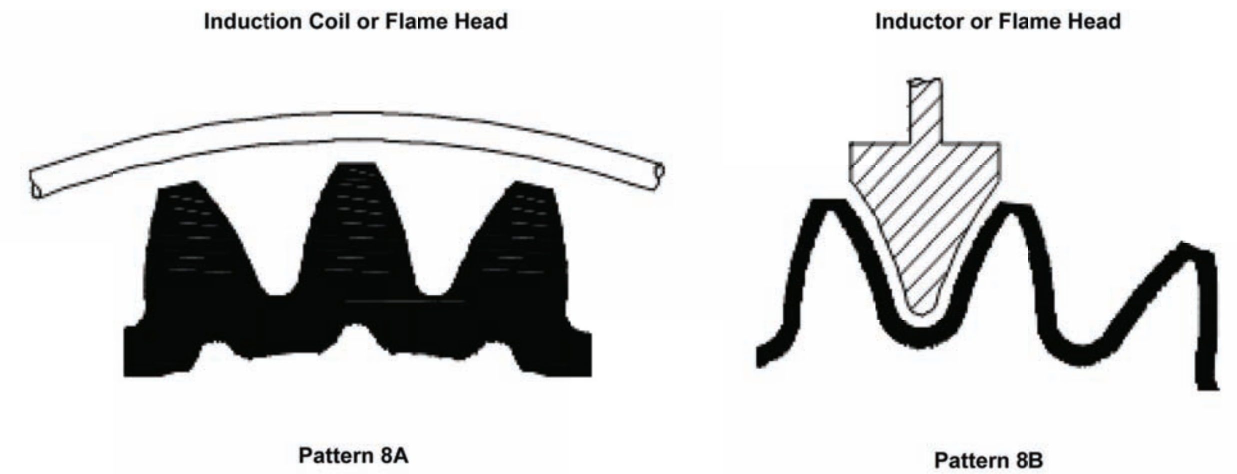


Figure 8—Acceptable Flame and Induction Hardening Patterns

6.2.6.4 Nitrided Gears

The minimum total case depth in inches (millimeters) for nitrided gears shall be in the ranges as defined in Equation (35) to Equation (36), see Figure 10:

In USC units:

$$h_{c,1} = \left(4.32896 \times 10^{-2}\right) - \left(9.68115 \times 10^{-3}\right) P_{nd} + \left(1.20185 \times 10^{-3}\right) P_{nd}{}^{-2} - \left(6.79721 \times 10^{-5}\right) P_{nd}{}^{3} + \left(1.37117 \times 10^{-6}\right) P_{nd}{}^{4} \quad (35)$$

In SI units:

$$h_{c,1} = 1.0995558 - 6.24589 m_n{}^{-1} + 19.6948 m_n{}^{-2} - 28.2921 m_n{}^{-3} + 14.4964 m_n{}^{-4} \quad (35)$$

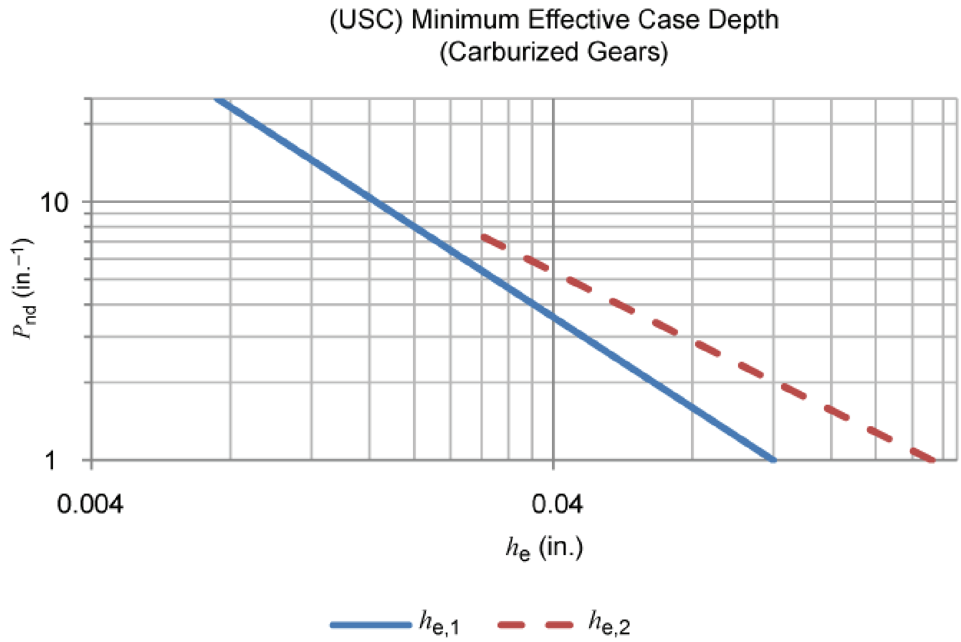
In USC units:

$$h_{c,2} = \left(6.60090 \times 10^{-2}\right) - \left(1.62224 \times 10^{-2}\right) P_{nd} + \left(2.09361 \times 10^{-3}\right) P_{nd}{}^{2} - \left(1.17755 \times 10^{-4}\right) P_{nd}{}^{3} + \left(2.33160 \times 10^{-6}\right) P_{nd}{}^{4} \quad (36)$$

In SI units:

$$h_{c,2} = 1.67663 - 10.4660 m_n{}^{-1} + 34.3081 m_n{}^{-2} - 49.0133 m_n{}^{-3} + 24.6503 m_n{}^{-4} \quad (36)$$

The values shown have been successfully used for nitrided gears and can be used as a guide. For gearing requiring maximum performance, especially large sizes, coarse pitches, and high working stresses, detailed studies must be made of application, loading, and manufacturing procedures to determine the desired gradients of hardness, strength, and internal residual stresses throughout the tooth.



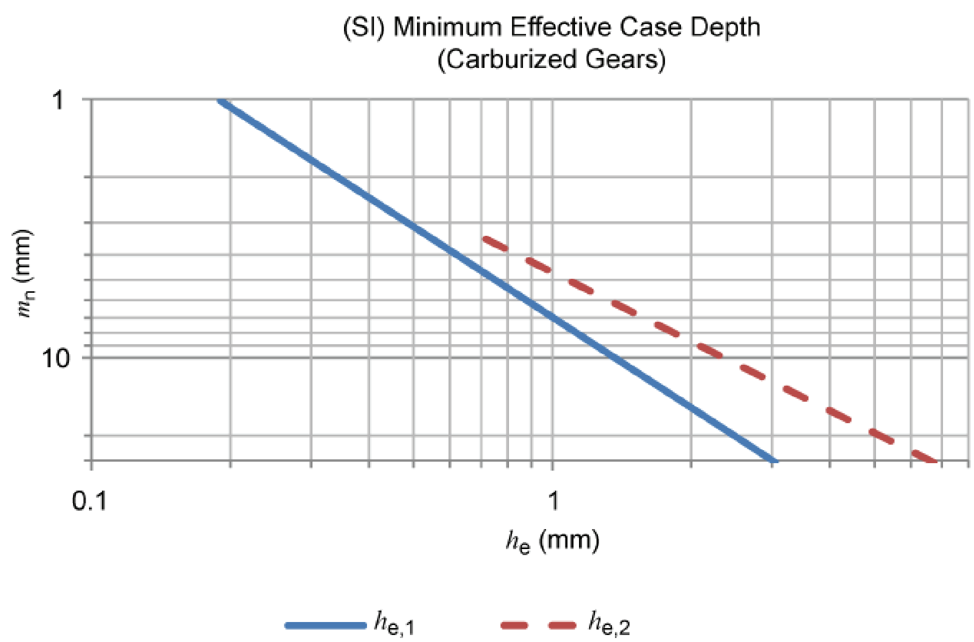


Figure 9—Effective Case Depth for Carburized Gears, h_e

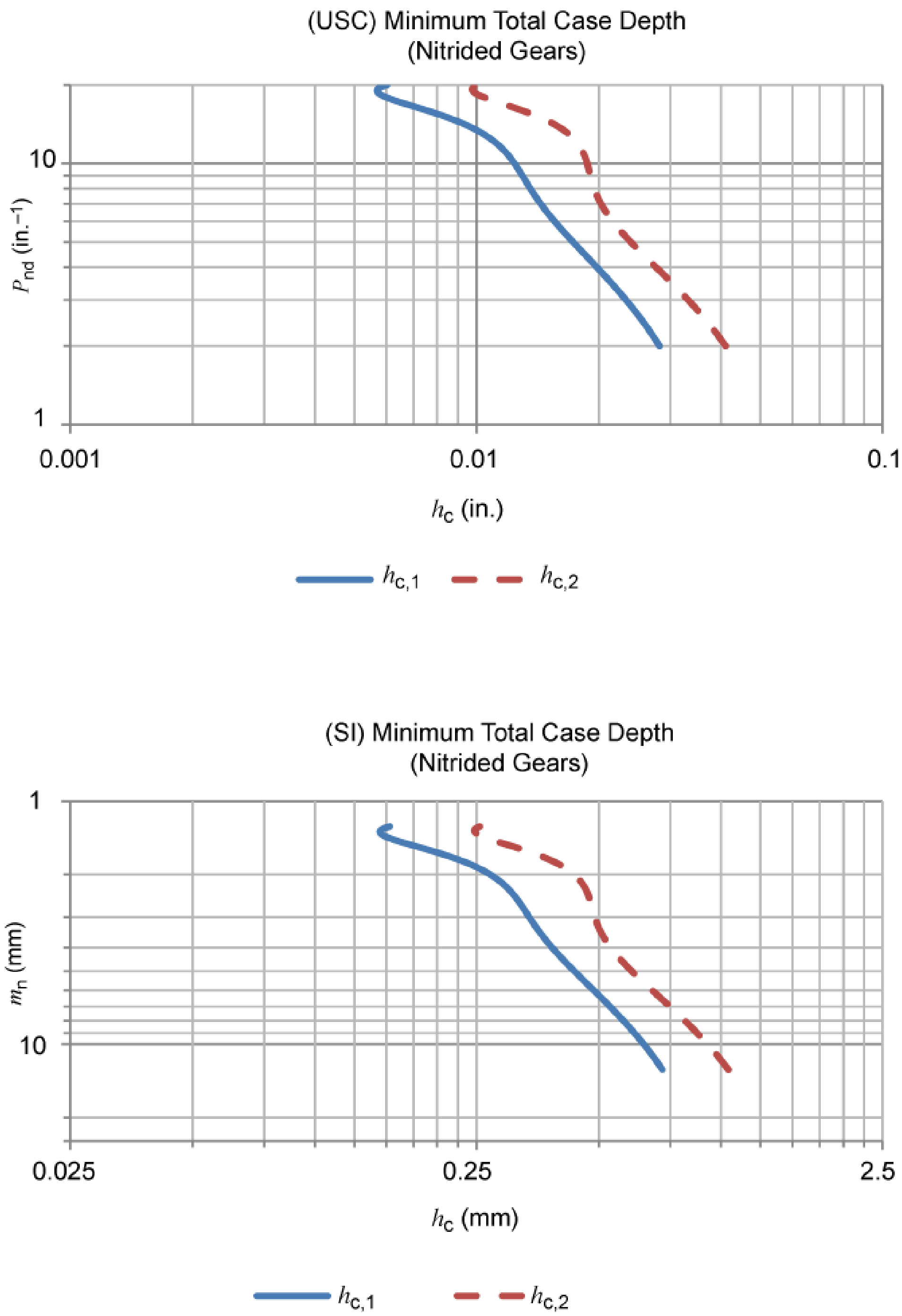


Figure 10—Minimum Total Case Depth for Nitrided Gears, h_c

6.3 Chain Reducers

6.3.1 Design

Chain drives shall be either single, double, or triple reduction.

Single or multiple strand roller chain, conforming to ASME B29.100 heavy series, shall be used. Link plates may be thicker than specified in ASME B29.100. Center link plates of multiple strand chains shall be press fitted on the pins.

Sprockets shall have the ASME tooth form. The small sprocket shall have not less than 11 teeth.

6.3.2 Rating Factors

Chain and sprocket ratings shall be based on a nominal pumping speed of 20 strokes per minute.

6.3.3 Metallurgy

The small sprocket shall be of steel and of a minimum of 225 Brinell hardness. The large sprocket shall be of steel or cast iron.

6.3.4 Dimensions

The distance between sprocket centerlines shall not be less than the sum of the pitch circle radius of the large sprocket plus the pitch circle diameter of the small sprocket. Chain length shall be selected to obtain an even number of pitches (no offset link).

A minimum take-up of two pitches, or 3 % of chain length, whichever is less, shall be provided.

6.3.5 Alignment

Shafts and sprockets shall be aligned to provide proper distribution of load across the width. Where a shaft is movable for take-up, reference marks shall be provided for checking parallelism.

6.3.6 Peak Torque Rating

The peak torque rating of the first reduction shall be calculated as follows.

a) For double reduction reducers, the peak torque rating of the first (high speed) reduction shall be related to the crankshaft peak torque rating by multiplying the high-speed reduction peak torque by the ratio of the second (low-speed) reduction.

b) For triple reduction reducers, the peak torque rating of the first (high-speed) reduction shall be related to the crankshaft peak torque rating by multiplying the high-speed reduction peak torque by the product of the ratios of the second (intermediate-speed) and third (low-speed) reductions.

The following Equation (37) shall be used for rating of chain:

$$T = \frac{SR}{12} \tag{37}$$

where

T is the peak torque rating expressed in in.-lb (Nm);

S is the ultimate tensile strength of chain in accordance with ASME B29.100 expressed in lb (N);

R is the pitch radius of large sprocket expressed in inches (millimeters).

6.4 Components

6.4.1 Housing

The housing may be of any design, provided it is sufficiently rigid to maintain shaft positions under the maximum gear and structural loads for which it is intended.

6.4.2 Bearings

Shafts may be supported in sleeve bearings or in antifriction bearings.

6.4.3 Sleeve Bearings

Sleeve bearings shall be designed for bearing pressures not in excess of 750 psi (5.17 MPa) on the bearing's projected area, based on actual loading (internal and external), at the rated peak torque.

6.4.4 Antifriction Bearings

Antifriction bearings shall be selected according to the bearing manufacturer's recommendations based on actual loads (internal and external) at rated peak torque and rated speed for not less than 15,000 hours L-10 life.

6.4.5 Shafts

6.4.5.1 Shaft Stresses

For steel shafts and for the torque rating of the unit, the maximum stress due to torsion $f_{\mathrm{s,t}}$ and the maximum stress due to bending $f_{\mathrm{s,b}}$ shall not exceed the values shown in Figure 11. These allowable stress limitations allow for stress concentrations arising from keyways, shoulders, and grooves, etc. not exceeding a factor of 3.0. More detailed analysis is required where stress concentrations (considering notch sensitivity) exceed a factor of 3.0, where press fit components are used or where there are unusual deflections.

6.4.5.2 Shaft Deflections

Shaft deflections causing tooth misalignment shall be analyzed regardless of stress levels to ensure satisfactory tooth contact as required to achieve the C_{m} (see 6.2.3.2.2) and K_{m} (see 6.2.3.2.3) values used to rate the gearing.

6.4.6 Key Stresses

The shear and compressive stress in a key shall be calculated using Equation (38) and Equation (39):

In USC units:

$$S_{\mathrm{s}} = \frac{2T_{\mathrm{t}}}{d_{\mathrm{s}} w L} \tag{38}$$

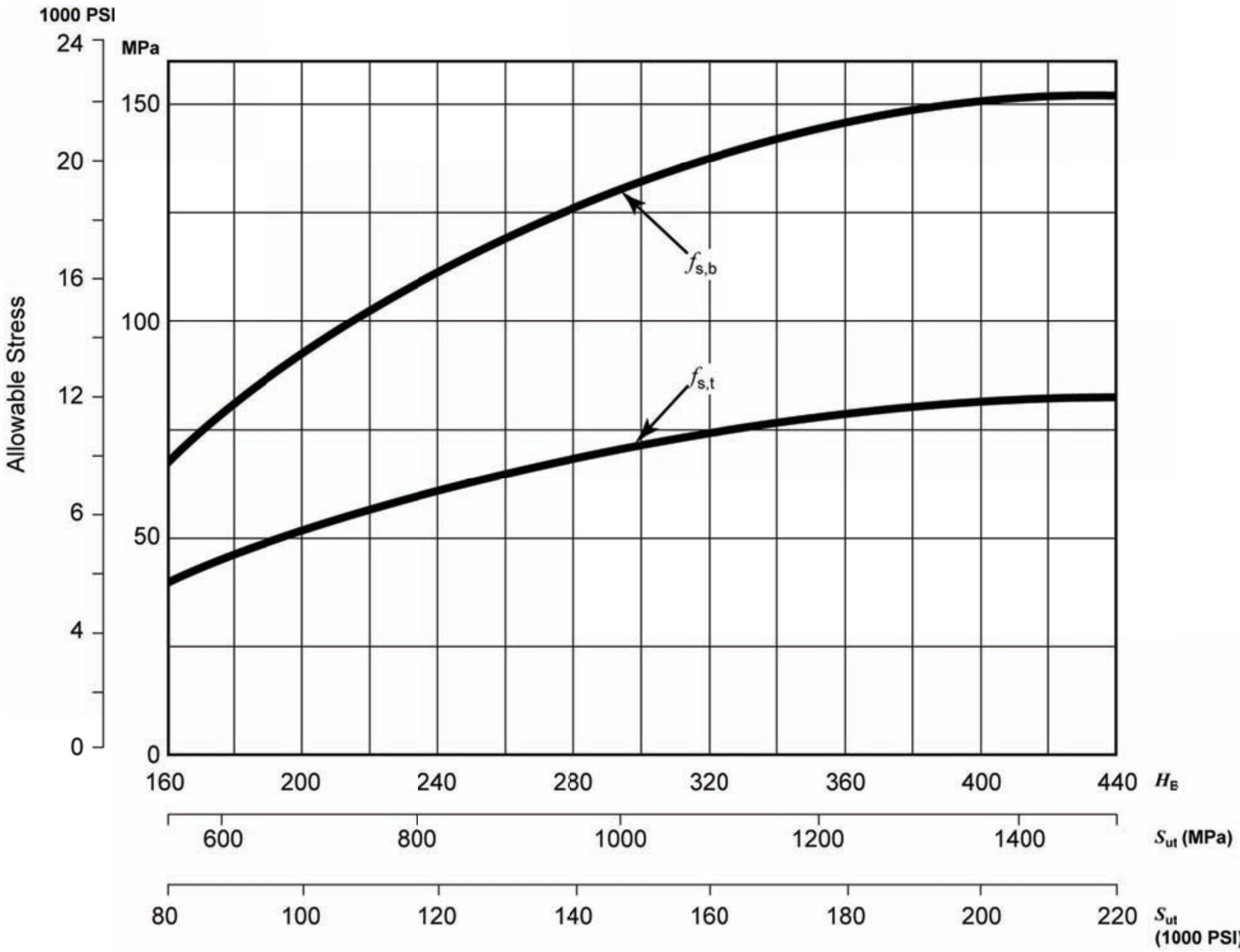


Figure 11—Allowable Stress—Shafting

In SI units:

$$S_s = \frac{2000T_t}{d_s w L} \tag{38}$$

In USC units:

$$S_c = \frac{2T_t}{d_s h_1 L} \tag{39}$$

In SI units:

$$S_c = \frac{2000T_t}{d_s h_1 L} \tag{39}$$

where

S_s is the shear stress of key, expressed in psi (MPa) (see Table 9);

S_c is the compressive stress of key, expressed in psi (MPa) (see Table 9);

T_t is the transmitted shaft torque carried by the key (excludes the minimum interference fit torque capacity between the hub and shaft), expressed in in.-lb (Nm);

d_s is the shaft diameter, expressed in inches (millimeters). For tapered shaft, use mean diameter;

w is the width of key, expressed in inches (millimeters);

L is the length of key, expressed in inches (millimeters);

h_1 is the height of key in the shaft or hub that bears against the keyway, expressed in inches (millimeters). For designs where unequal portions of the keyway are in the hub or shaft, h_1 is the minimum portion.

Maximum allowable key stresses based on peak torque rating are shown in Table 9 for AISI materials, appropriate allowable stresses should be developed if non-AISI materials are used. These stress limits are based on the assumption that an interference fit is used with a torque capability equal to or greater than the reducer rating at the shaft. Thus, typical running torque will not be carried by the key. Torque loads associated with counterbalance (decoupled from the well), braking, or installation and handling must be accommodated by the key design.

Table 9—Allowable Key Stresses

Key Material	Hardness (BHN)	Allowable Shear Stress [a] psi (MPa)	Allowable Compressive Stress [a] psi (MPa)
AISI 1018	None specified	10,000 (68.9)	20,000 (138)
AISI 1045	225 to 265	15,000 (103.4)	30,000 (207)
	265 to 305	20,000 (138)	40,000 (276)
AISI 4140	310 to 360	30,000 (207)	60,000 (345)

[a] The values tabulated assume an interference fit with a torque capacity equal to or greater than the reducer rating. When other methods of attachment are used, a detailed stress analysis shall be performed.

6.4.7 Peak Loading (Overloads)

The shaft to hub interface shall be capable of withstanding the manufacturer specified maximum anticipated loads associated with beam pumping units.

6.4.8 Fastener Stresses

Fastener stresses shall be determined from the forces developed at the torque rating of the gear reducer in addition to any external structure loading.

The maximum allowable stress at the tensile area of threaded fasteners (bolts, studs, or cap screws) shall not exceed the values given in Table 10. The tensile area (A_t) is calculated as follows:

In USC units:

$$A_t = 0.785\left(D_m - \frac{0.97}{N_t}\right)^2 \quad (40)$$

In SI units:

$$A_t = 0.785\left(D_m - 0.97p\right)^2 \quad (40)$$

where

A_t is the tensile area of fastener, expressed in in.^{2} (mm^2);

D_m is the major diameter of fastener, expressed in inches (millimeters);

N_t is the threads per inch of fastener;

p is the metric pitch of fastener threads, expressed in millimeters.

6.4.9 Tensile Preload

The tensile preload in the bolt, stud, or cap screw should be 70 % of the yield strength of the material as determined at the tensile area of the thread.

6.4.10 Special Seals and Breathers

Beam pumping units operate outdoors under potentially adverse atmospheric conditions and shall be equipped with seals and breathers designed for these conditions.

Table 10—Maximum Allowable Tensile Stress, Fasteners

SAE and/or ASTM Designation	**Threaded Fastener Diameter, D_m** in.	**Hardness (BHN)**	**Minimum Proof Strength** psi (MPa)	**Minimum Tensile Strength** psi (MPa)	**Maximum Allowable Tensile Stress** psi (MPa)
SAE 2	Over $^1/_4$ to $^3/_4$ incl.	149–241	55,000 (379)	74,000 (510)	11,000 (75.8)
	Over $^3/_4$ to 1 $^1/_2$ incl.	121–241	33,000 (228)	60,000 (414)	11,000 (75.8)
SAE 5 (ASTM A449)	Over $^1/_4$ to 1 incl.	241–302	85,000 (586)	120,000 (827)	20,000 (138)
	Over 1 to 1$^1/_2$ incl.	223–285	74,000 (510)	105,000 (724)	18,000 (124)
ASTM A449	Over 1$^1/_2$ to 3 incl.	183–235	55,000 (379)	90,000 (621)	13,000 (89.6)
ASTM A354	Over $^1/_4$ to 2$^1/_2$ incl.	217–285	80,000 (552)	105,000 (724)	17,000 (117)
Grade BB	Over 2$^1/_2$ to 4 incl.	217–285	75,000 (517)	100,000 (689)	17,000 (117)
ASTM A354	Over $^1/_4$ to 2$^1/_2$ incl.	255–321	109,000 (752)	125,000 (862)	22,000 (152)
Grade BC	Over 2$^1/_2$ to 4 incl.	255–321	99,000 (683)	115,000 (793)	22,000 (152)
SAE 7	Over $^1/_4$ to 1$^1/_2$ incl.	277–321	105,000 (724)	133,000 (917)	24,600 (170)
SAE 8 (ASTM A354 Grade BD)	Over $^1/_4$ to 1$^1/_2$ incl.	302–352	120,000 (827)	150,000 (1,034)	27,700 (186)
NOTE The basis for the values in Table 10 is to prevent joint opening at a peak-rated load.					

7 Product Identification

7.1 Beam Pump Structure Nameplate

Each beam pump structure shall be provided with a name plate similar to that shown in Figure 12. At the discretion of the manufacturer, the nameplate may contain other nonconflicting and appropriate information, such as model number or lubrication instructions. When structural unbalance is negative the minus (–) sign shall be stamped on the nameplate.

7.2 Speed Reducer Nameplate

Each pumping unit reducer shall be provided with a nameplate substantially as shown in Figure 13. The size (peak torque rating in 1000 in.-lb) shown on the nameplate shall be one of those listed in Table 2. No other rating marking shall be applied to the reducer. The nameplate may contain information such as model number, lubrication instructions, etc., provided such marking does not conflict with the rating marking.

7.3 Installation Markings

Clearly defined and readily usable markings shall be provided on the end cross members of the base to indicate the vertical projection of the walking beam centerline. The markings shall be applied with a chisel, punch, or other suitable tool.

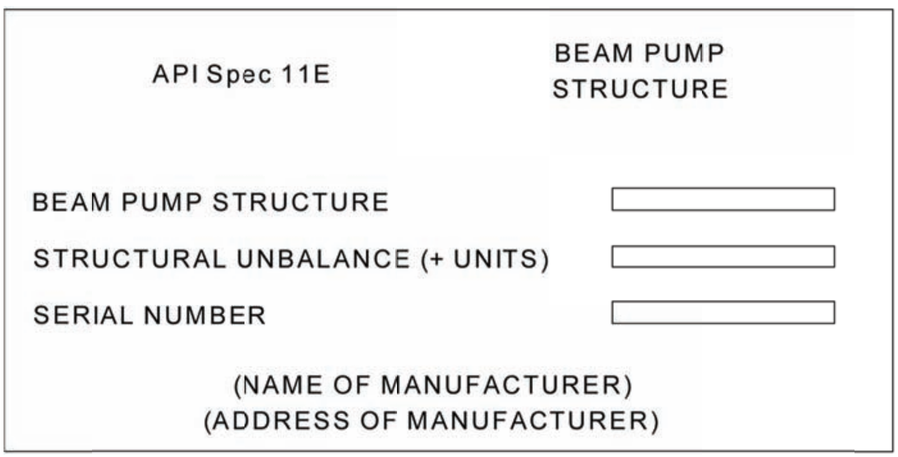


Figure 12—Beam Pump Structure Nameplate

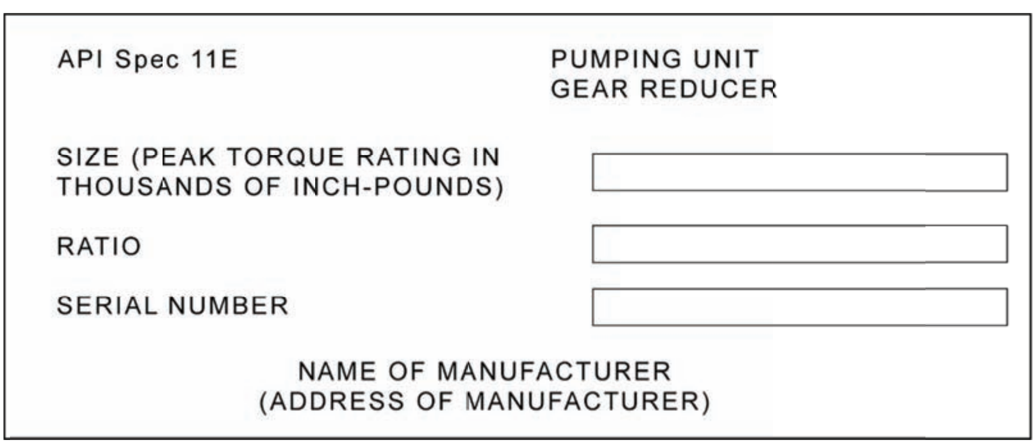


Figure 13—Pumping Unit Reducer Nameplate

7.4 Supplier/Manufacturing Requirements

7.4.1 Quality Control

The manufacturer is responsible for complying with all of the provisions of this specification. All quality control work shall be controlled by documented instructions that include acceptance criteria.

Only products in full compliance with this specification shall be considered as being in accordance with this specification.

Where a user/purchaser appoints an inspector to verify the manufacture, the inspector shall have unrestricted access to all works related to the manufacture of items for the purchaser. The manufacturer shall afford the inspector all reasonable facilities to satisfy him/her that the material is being furnished in accordance with this specification. Any inspection made at the place of manufacture shall be considered process inspection and shall be conducted so as not to interfere unnecessarily with the operation of the works.

7.4.2 Datasheet

The manufacturer shall retain in product files, and make available upon request, a completed Manufacturer's Gear Reducer Datasheet (see Figure C.3) for each gear reducer size manufactured.

8 Storage and Maintenance

8.1 Shipping and Handling

8.1.1 General

Products shall be packaged, stored and transported in such a manner as to preserve the full integrity of the equipment prior to installation in conformance with the manufacturer's written specifications.

8.1.2 Packaging

Products shall be packaged in such a way as to prevent physical damage during typical transport and deterioration during storage. Products shall be prevented from contact with contaminants.

8.1.3 Storage

Products shall be stored in conditions that meet the manufacturer's written specifications. Equipment shall be stored in an unstressed condition and shall be protected from the effects of abrasives and chemicals that may cause product damage. Where applicable the application of manufacturer approved anticorrosion fluids prolongs the products effective storage duration.

8.1.4 Handling and Transport

At each occasion the handling of the product shall meet the written requirements of the manufacturer/supplier to prevent any operational damage to the product. Transportation regulations governing size, weight, hazardous materials, etc. as set forth by state, regional, or national authorities and manufacturer recommendations shall be observed when shipping/transporting products.

8.2 Lubrication

Lubrication is recommended to be in accordance with API 11G.

Annex A
(informative)

API Monogram

A.1 Scope

A.1.1 Applicability

This annex is normative (mandatory) for product supplied bearing the API Monogram and manufactured at a facility licensed by API; for all other instances it is not applicable.

A.1.2 General

This annex is only normative for product supplied bearing the API Monogram and manufactured at a facility licensed by API; for all other instances it is not applicable.

The API Monogram® is a registered certification mark owned by the American Petroleum Institute (API) and authorized for licensing by the API Board of Directors. Through the API Monogram Program, API licenses product manufacturers to apply the API Monogram to products which comply with product specifications and have been manufactured under a quality management system that meets the requirements of API Q1. API maintains a complete, searchable list of all Monogram licensees on the API Composite List website (www.api.org/compositelist).

The application of the API Monogram and license number on products constitutes a representation and warranty by the licensee to API and to purchasers of the products that, as of the date indicated, the products were manufactured under a quality management system conforming to the requirements of API Q1 and that the product conforms in every detail with the applicable standard(s) or product specification(s). API Monogram program licenses are issued only after an on-site audit has verified that an organization has implemented and continually maintained a quality management system that meets the requirements of API Q1 and that the resulting products satisfy the requirements of the applicable API product specification(s) and/or standard(s). Although any manufacturer may claim that its products meet API product requirements without monogramming them, only manufacturers with a license from API can apply the API Monogram to their products.

Together with the requirements of the API Monogram license agreement, this annex establishes the requirements for those organizations who wish to voluntarily obtain an API license to provide API monogrammed products that satisfy the requirements of the applicable API product specification(s) and/or standard(s) and API Monogram Program requirements.

For information on becoming an API Monogram Licensee, please contact API, Certification Programs, 1220 L Street, NW, Washington, DC 20005 or call 202-682-8145 or by email at certification@api.org.

A.2 Normative References

In addition to the referenced standards listed earlier in this document, this annex references the following standard:

API Specification Q1, *Specification for Quality Management System Requirements for Manufacturing Organizations for the Petroleum and Natural Gas Industry*

For Licensees under the Monogram Program, the latest version of this document shall be used. The requirements identified therein are mandatory.

A.3 API Monogram Program: Licensee Responsibilities

A.3.1 Monogram Program Requirements

For all organizations desiring to acquire and maintain a license to use the API Monogram, conformance with the following shall be required at all times:

a) the quality management system requirements of API Q1;

b) the API Monogram Program requirements of API Q1, Annex A;

c) the requirements contained in the API product specification(s) to which the organization is licensed;

d) the requirements contained in the API Monogram Program License Agreement.

A.3.2 Control of the Application and Removal of the API Monogram

Each licensee shall control the application and removal of the API Monogram in accordance with the following:

a) Products that do not conform to API specified requirements shall not bear the API Monogram.

b) Each licensee shall develop and maintain an API Monogram marking procedure that documents the marking/monogramming requirements specified by this annex and any applicable API product specification(s) and/or standard(s). The marking procedure shall:

   1) define the authority responsible for application and removal of the API Monogram and license number;

   2) define the method(s) used to apply the Monogram and license number;

   3) identify the location on the product where the API Monogram and license number are to be applied;

   4) require the application of the date of manufacture of the product in conjunction with the use of the API Monogram and license number;

   5) require that the date of manufacture, at a minimum, be two digits representing the month and two digits representing the year (e.g. 05-12 for May 2012) unless otherwise stipulated in the applicable API product specification(s) or standard(s); and

   6) define the application of all other required API product specification(s) and/or standard(s) marking requirements.

c) Only an API licensee shall apply the API Monogram and its designated license number to API monogrammable products.

d) The API Monogram and license number, when issued, are site-specific and subsequently the API Monogram shall only be applied at that site-specific licensed facility location.

e) The API Monogram may be applied at any time appropriate during the production process but shall be removed in accordance with the licensee's API Monogram marking procedure if the product is subsequently found to be out of conformance with any of the requirements of the applicable API product specification(s) and/or standard(s) and API Monogram Program.

For certain manufacturing processes or types of products, alternative API Monogram marking procedures may be acceptable. Requirements for alternative API Monogram marking are detailed in the, *API Monogram Program Alternative Marking of Products License Agreement*, available on the API Monogram Program website at http://www.api.org/alternative-marking.

A.3.3 Design and Design Documentation

Each licensee and/or applicant for licensing shall maintain current design documentation as identified in API Q1 for all of the applicable products that fall under the scope of each Monogram license. The design document information shall provide objective evidence that the product design meets the requirements of the applicable and most current API product specification(s) and/or standard(s). The design documentation shall be made available during API audits of the facility.

In specific instances, the exclusion of design activities is allowed under the Monogram Program, as detailed in Advisory # 6, available on API Monogram Program website at http://www.api.org/advisories.

A.3.4 Manufacturing Capability

The API Monogram Program is designed to identify facilities that have demonstrated the ability to manufacture equipment that conforms to API specifications and/or standards. API may refuse initial licensing or suspend current licensing based on a facility's level of manufacturing capability. If API determines that an additional review is warranted, API may perform additional audits (at the organization's expense) of any subcontractors to ensure their conformance with the requirements of the applicable API product specification(s) and/or standard(s).

A.3.5 Use of the API Monogram in Advertising

An API Monogram licensee shall not use the API Monogram and/or license number on letterheads, buildings or other structures, websites or in any advertising without an express statement of fact describing the scope of Licensee's authorization (license number and product specification). The Licensee should contact API for guidance on the use of the API Monogram other than on products.

A.4 Product Marking Requirements

A.4.1 General

These marking requirements shall apply only to those API Licensees wishing to mark applicable products in conjunction with the requirements of the API Monogram Program.

A.4.2 Product Specification Identification

Manufacturers shall mark products as specified by the applicable API specifications or standards. Marking shall include reference to the applicable API specification and/or standard. Unless otherwise specified, reference to the API specifications and/or standards shall be, as a minimum, "API [Document Number]" (e.g. API 11E). Unless otherwise specified, when space allows, the marking may include use of "Spec" or "Std," as applicable (e.g. API Spec 11E).

A.4.3 Units

Products shall be marked with units as specified in the API specification and/or standard. If not specified, equipment shall be marked with U.S. customary (USC) units. Use of dual units [USC units and metric (SI) units] may be acceptable, if such units are allowed by the applicable product specification and/or standard.

A.4.4 Nameplates

Nameplates, when applicable, shall be made of a corrosion-resistant material unless otherwise specified by the API specification and/or standard. Nameplate shall be located as specified by the API specification and/or standard. If the location is not specified, then the licensee shall develop and maintain a procedure detailing the location to which the nameplate shall be applied. Nameplates may be attached at any time during the manufacturing process.

The API Monogram and license number shall be marked on the nameplate, in addition to the other product marking requirements specified by the applicable product specification and/or standard.

Each pumping unit and pumping unit reducer shall be provided with a nameplate substantially as shown in Section 7.

a) For pumping units, structural unbalance shall be identified as the force in pounds at the polished rods to hold the beam in a horizontal position with the pitmans disconnected from the crankpin. The structural unbalance shall be considered positive when the force required at the polished rod is downward, and negative when upward. The minus (–) sign shall be stamped on the nameplate when the value is negative.

b) For pumping unit reducers, the size (peak torque rating in 1000 in.-lb) shown on the nameplate shall be one of those listed in Table 2. No other rating marking shall be applied to the reducer.

Nameplates shall be made of a corrosion-resistant material and shall be located as indication in the marking section of this specification. If the location is not identified, then J.3.2 b) shall apply.

A.4.5 License Number

The API Monogram license number shall not be used unless it is marked in conjunction with the API Monogram. The license number shall be used in close proximity to the API Monogram.

A.5 API Monogram Program: Nonconformance Reporting

API solicits information on products that are found to be nonconforming with API specified requirements, as well as field failures (or malfunctions), which are judged to be caused by either specification and/or standard deficiencies or nonconformities against API specified requirements. Customers are requested to report to API all problems with API monogrammed products. A nonconformance may be reported using the API Nonconformance Reporting System available at http://compositelist.api.org/ncr.aspx.

Annex B
(normative)

Beam Pumping Unit Designations

The designation of a particular pumping unit is composed of a three-number sequence containing the reducer rating (expressed in 1000 in.-lb multiples), structure capacity (expressed in 100 lb multiples), and maximum stroke length (expressed in inches). It is recommended that pumping units furnished to this specification adhere to the gear reducer rating, structure capacity, and stroke length as given in Table B.1. The particular combinations in the table are typical, but combinations other than those listed are acceptable under this standard.

Table B.1—Pumping Unit Designation

Designation	Reducer Rating in.-lb (Nm)	Structure Capacity lb (N)	Max. Stroke Length in. (mm)
6.4-32-16	6,400 (723)	3,200 (14,234)	16 (406)
6.4-21-24	6,400 (723)	2,100 (9,341)	24 (610)
10-32-24	10,000 (1,130)	3,200 (14,234)	24 (610)
10-40-20	10,000 (1,130)	4,000 (17,793)	20 (508)
16-27-30	16,000 (1,808)	2,700 (12,010)	30 (762)
16-53-30	16,000 (1,808)	5,300 (23,576)	30 (762)
25-53-30	25,000 (2,825)	5,300 (23,576)	30 (762)
25-56-36	25,000 (2,825)	5,600 (24,910)	36 (914)
25-67-36	25,000 (2,825)	6,700 (29,803)	36 (914)
40-89-36	40,000 (4,519)	8,900 (39,589)	36 (914)
40-76-42	40,000 (4,519)	7,600 (33,806)	42 (1,067)
40-89-42	40,000 (4,519)	8,900 (39,589)	42 (1,067)
40-76-48	40,000 (4,519)	7,600 (33,806)	48 (1,219)
57-76-42	57,000 (6,440)	7,600 (33,806)	42 (1,067)
57-89-42	57,000 (6,440)	8,900 (39,589)	42 (1,067)
57-95-48	57,000 (6,440)	9,500 (42,258)	48 (1,219)
57-109-48	57,000 (6,440)	10,900 (48,486)	48 (1,219)
57-76-54	57,000 (6,440)	7,600 (33,806)	54 (1,372)
80-109-48	80,000 (9,039)	10,900 (48,486)	48 (1,219)
80-133-48	80,000 (9,039)	13,300 (59,161)	48 (1,219)
80-119-54	80,000 (9,039)	11,900 (52,934)	54 (1,372)
80-133-54	80,000 (9,039)	13,300 (59,161)	54 (1,372)
80-119-64	80,000 (9,039)	11,900 (52,934)	64 (1,626)
114-133-54	114,000 (12,880)	13,300 (59,161)	54 (1,372)
114-143-64	114,000 (12,880)	14,300 (63,610)	64 (1,626)
114-173-64	114,000 (12,880)	17,300 (76,954)	64 (1,626)
114-143-74	114,000 (12,880)	14,300 (63,610)	74 (1,880)
114-119-86	114,000 (12,880)	11,900 (52,934)	86 (2,184)
160-173-64	160,000 (18,078)	17,300 (76,954)	64 (1,626)
160-143-74	160,000 (18,078)	14,300 (63,610)	74 (1,880)
160-173-74	160,000 (18,078)	17,300 (76,954)	74 (1,880)
160-200-74	160,000 (18,078)	20,000 (88,964)	74 (1,880)
160-173-86	160,000 (18,078)	17,300 (76,954)	86 (2,184)
228-173-74	228,000 (25,761)	17,300 (76,954)	74 (1,880)
228-200-74	228,000 (25,761)	20,000 (88,964)	74 (1,880)
228-213-86	228,000 (25,761)	21,300 (94,747)	86 (2,184)
228-246-86	228,000 (25,761)	24,600 (109,426)	86 (2,184)
228-173-100	228,000 (25,761)	17,300 (76,954)	100 (2,540)
228-213-120	228,000 (25,761)	21,300 (94,747)	120 (3,048)

Table B.1—Pumping Unit Designation (Continued)

Designation	Reducer Rating in.-lb (Nm)	Structure Capacity lb (N)	Max. Stroke Length in. (mm)
320-213-86	320,000 (36,155)	21,300 (94,747)	86 (2,184)
320-256-100	320,000 (36,155)	25,600 (113,874)	100 (2,540)
320-305-100	320,000 (36,155)	30,500 (135,671)	100 (2,540)
320-213-120	320,000 (36,155)	21,300 (94,747)	120 (3,048)
320-256-120	320,000 (36,155)	25,600 (113,874)	120 (3,048)
320-256-144	320,000 (36,155)	25,600 (113,874)	144 (3,658)
456-256-120	456,000 (51,521)	25,600 (113,874)	120 (3,048)
456-305-120	456,000 (51,521)	30,500 (135,671)	120 (3,048)
456-365-120	456,000 (51,521)	36,500 (162,360)	120 (3,048)
456-256-144	456,000 (51,521)	25,600 (113,874)	144 (3,658)
456-305-144	456,000 (51,521)	30,500 (135,671)	144 (3,658)
456-305-168	456,000 (51,521)	30,500 (135,671)	168 (4,267)
640-305-120	640,000 (72,310)	30,500 (135,671)	120 (3,048)
640-256-144	640,000 (72,310)	25,600 (113,874)	144 (3,658)
640-305-144	640,000 (72,310)	30,500 (135,671)	144 (3,658)
640-365-144	640,000 (72,310)	36,500 (162,360)	144 (3,658)
640-305-168	640,000 (72,310)	30,500 (135,671)	168 (4,267)
640-305-192	640,000 (72,310)	30,500 (135,671)	192 (4,877)
912-427-144	912,000 (103,042)	42,700 (189,939)	144 (3,658)
912-305-168	912,000 (103,042)	30,500 (135,671)	168 (4,267)
912-365-168	912,000 (103,042)	36,500 (162,360)	168 (4,267)
912-305-192	912,000 (103,042)	30,500 (135,671)	192 (4,877)
912-427-192	912,000 (103,042)	42,700 (189,939)	192 (4,877)
912-470-240	912,000 (103,042)	47,000 (209,066)	240 (6,096)
912-427-216	912,000 (103,042)	42,700 (189,939)	216 (5,486)
1280-427-168	1,280,000 (144,621)	42,700 (189,939)	168 (4,267)
1280-427-192	1,280,000 (144,621)	42,700 (189,939)	192 (4,877)
1280-427-216	1,280,000 (144,621)	42,700 (189,939)	216 (5,486)
1280-470-240	1,280,000 (144,621)	47,000 (209,066)	240 (6,096)
1280-470-300	1,280,000 (144,621)	47,000 (209,066)	300 (7,620)
1824-427-192	1,824,000 (206,084)	42,700 (189,939)	192 (4,877)
1824-427-216	1,824,000 (206,084)	42,700 (189,939)	216 (5,486)
1824-470-240	1,824,000 (206,084)	47,000 (209,066)	240 (6,096)
1824-470-300	1,824,000 (206,084)	47,000 (209,066)	300 (7,620)
2560-470-240	2,560,000 (289,241)	47,000 (209,066)	240 (6,096)
2560-470-300	2,560,000 (289,241)	47,000 (209,066)	300 (7,620)
3648-470-240	3,648,000 (412,169)	47,000 (209,066)	240 (6,096)
3648-470-300	3,648,000 (412,169)	47,000 (209,066)	300 (7,620)

Annex C
(informative)

Recommended Data Forms [4]

C.1 General

Example data forms are provided for the manufacturer to communicate information on the crank counterbalances and stroke and torque factors.

C.2 Rating Form for Crank Counterbalances

Manufacturers are recommended to use the form in Figure C.1 when providing pumping unit crank counterbalances.

C.3 Stroke and Torque Factors

Manufacturers are recommended to use the form in Figure C.2 when providing pumping unit stroke and torque factors.

C.4 Gear Reducer Datasheet

Manufacturers are recommended to use this Figure C.3 when providing gear reducer information.

[4] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

Name of manufacturer ____________________ Date prepared ____________

Designation of unit

Description [a]	Total Weight [lb (N)]	Maximum Moment About Crankshaft [b] [in.-lb (Nm)]

[a] Describe parts in use accurately enough to avoid any possible misunderstanding, showing on separate lines a series of practical combinations from minimum to maximum.

[b] Equals total weight (column 2) times distance to center of gravity, with crank in horizontal position.

Figure C.1—Rating Form for Crank Counterbalances

Name of manufacturer ______________________ Date Prepared __________

Designation of unit ______________________

Pumping unit structural imbalance ______ Pounds

Position of Crank [a] (degrees)	**Position of Rods** [b]				**Torque Factor** [c]			
	Length of Stroke [in. (mm)]				**Length of Stroke** [in. (mm)]			
0°								
15°								
30°								
45°								
60°								
75°								
90°								
105°								
120°								
135°								
150°								
165°								
180°								
195°								
210°								
225°								
240°								
255°								
270°								
285°								
300°								
315°								
330°								
345°								

A ________ P ________

C ________ K ________

R_1 ________ H ________

R_2 ________ I ________

R_3 ________ G ________

[a] For crank counterbalance units with Class I geometry, the position of the crank is the angular displacement measured clockwise from the 12 o'clock position, viewed with the wellhead to the right. For crank counterbalanced units with Class III geometry, the position of the crank is the angular displacement measured counterclockwise from the 6 o'clock position, viewed with the wellhead to the right. For air counterbalanced units with Class III geometry, the position of the crank is the angular displacement measured clockwise from the 6 o'clock position, viewed with the wellhead to the right.

[b] Position is expressed as a fraction of stroke above lowermost position.

[c] Torque factor = $\frac{T}{P_R}$, where T = torque on pumping unit reducer due to polished rod load P_R.

NOTE See Annex D, Annex E, Annex F, or Annex G for symbol identification.

Figure C.2—Pumping Unit Stroke and Torque Factor Form

Manufactured by:		Date submitted	
Nominal reducer size			

Calculated Values

Pitting resistance torque					
First reduction		in.-lb (Nm)			
Second reduction		in.-lb (Nm)			
Third reduction		in.-lb (Nm)			
Bending strength torque	Units:				
First reduction:					
Gear		in.-lb (Nm)	Pinion		in.-lb (Nm)
Second reduction:					
Gear		in.-lb (Nm)	Pinion		in.-lb (Nm)
Third reduction:					
Gear		in.-lb (Nm)	Pinion		in.-lb (Nm)

Static torque					
First reduction:					
Gear		in.-lb (Nm)	Pinion		in.-lb (Nm)
Second reduction:					
Gear		in.-lb (Nm)	Pinion		in.-lb (Nm)
Third reduction:					
Gear		in.-lb (Nm)	Pinion		in.-lb (Nm)

Notes:

1. First reduction is high-speed reduction.

2. Second reduction is slow-speed reduction on double reduction gear reducers and the intermediate reduction on triple reduction gear reducers.

3. Third reduction is the slow-speed reduction on triple reduction reducers and is not applicable on double reduction reducers.

Construction Features

Type of reducer (Cross out if not applicable)

	(Single)	(Double)	(Triple) Reduction
	(Single)	(Double)	Helical gearing

Teeth

Number of teeth and normal diametral pitch or transverse diametral pitch:

First reduction	N_p		N_g		P_{nd} (m_n)		P_d (m)	
Second reduction	N_p		N_g		P_{nd} (m_n)		P_d (m)	
Third reduction	N_p		N_g		P_{nd} (m_n)		P_d (m)	

Center distance and net face width:

First reduction		CD		F
Second reduction		CD		F
Third reduction		CD		F

Figure C.3—Manufacturer's Gear Reducer Datasheet

Helix angle and normal pressure angle or transverse pressure angle (degrees):					
First reduction		ψ_g		ϕ_n		ϕ_t
Second reduction		ψ_g		ϕ_n		ϕ_t
Third reduction		ψ_g		ϕ_n		ϕ_t

Geometry factors, I and J (for pinion and gear)						
First reduction geometry factor	I		J_P		J_G	
Second reduction geometry factor	I		J_P		J_G	
Third reduction geometry factor	I		J_P		J_G	

Manufacturing methods					
Teeth generated by		process	Teeth finished by		process
Tooth hardening method					

Gear and pinion materials and hardness					
First reduction					
Gear material		Surface BHC/Rc		Core BNH [a]	
Pinion material		Surface BHC/Rc		Core BNH [a]	
Second reduction					
Gear material		Surface BHC/Rc		Core BNH [a]	
Pinion material		Surface BHC/Rc		Core BNH [a]	
Third reduction					
Gear material		Surface BHC/Rc		Core BNH [a]	
Pinion material		Surface BHC/Rc		Core BNH [a]	

Other components				
Crankshaft material		Hardness		
Housing material				
Housing type (Check):	Split		One piece	

Bearing sizes [b]		Bearing loading [d]	
High speed pinion		High speed pinion	
Intermediate speed pinion [c]		Intermediate speed pinion [c]	
Low speed pinion		Low speed pinion	
Low speed gear		Low speed gear	

[a] Core hardness required for surface hardened gears and pinions only.

[b] For journal bearings indicate projected area; for roller bearings indicate the American Bearing Manufacturer's Association (or equivalent) size. List all bearings on each shaft (state if bearings are mounted in carriers or directly in gear housing).

[c] Not applicable on double reduction reducers.

[d] For journal bearings, list loading on each bearing. For roller bearings, list L-10 life as calculated in 6.4.4.

Figure C.3—Manufacturer's Gear Reducer Datasheet (continued)

Annex D
(informative)

Torque Factor on Beam Pumping Units with Rear Mounted Geometry Class I Lever Systems with Crank Counterbalance

D.1 General

The following calculation technique is generally accepted. More precise results are dependent on the true stroke length, which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer. Determinations made with a dynamometer can determine more specific performance characteristics of the individual pumping unit.

D.2 Symbols

In addition to those listed in Section 2, the following system of nomenclature and symbols is used in this annex (see Figure D.1):

B structural unbalance, equal to the force at the polished rod required to hold the beam in a horizontal position with the pitmans disconnected from the crankpins, in pounds (lb) [newtons (N)]

NOTE This force is positive when acting downward and negative when acting upward.

G height from the center of the crankshaft to the bottom of the base beams, in inches (in.) [millimeters (mm)]

H height from the center of the saddle bearing to the bottom of the base beams, in inches (in.) [millimeters (mm)]

I horizontal distance between the centerline of the saddle bearing and the centerline of the crankshaft, in inches (in.) [millimeters (mm)]

J distance from the center of the crankpin bearing to the center of the saddle, in inches (in.) [millimeters (mm)]

K distance from the center of the crankshaft to the center of the saddle bearing, 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)]

P effective length of the pitman (from the center of the equalizer bearing to the center of the crankpin bearing), in inches (in.) [millimeters (mm)]

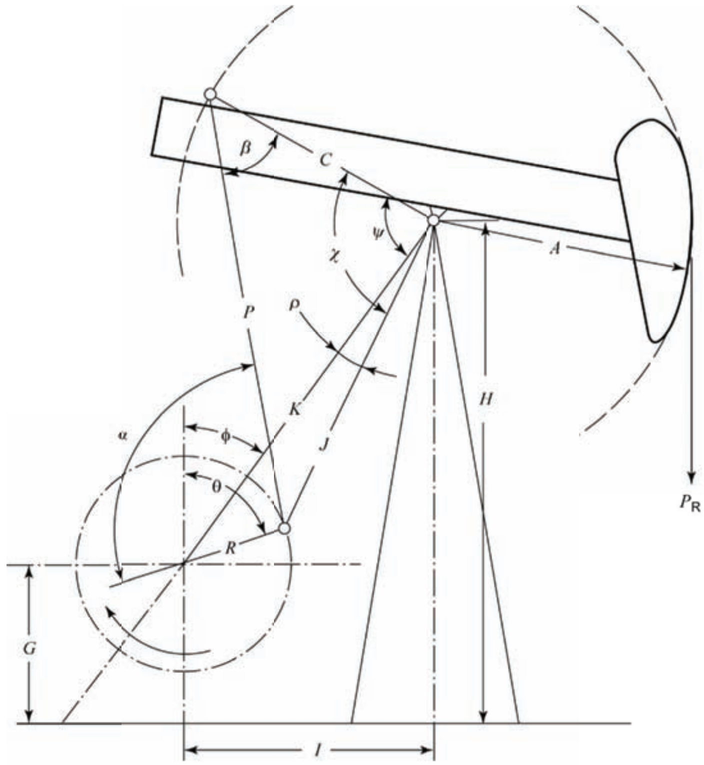
PRP polished rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle θ, unitless

T_n net torque at the crankshaft for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

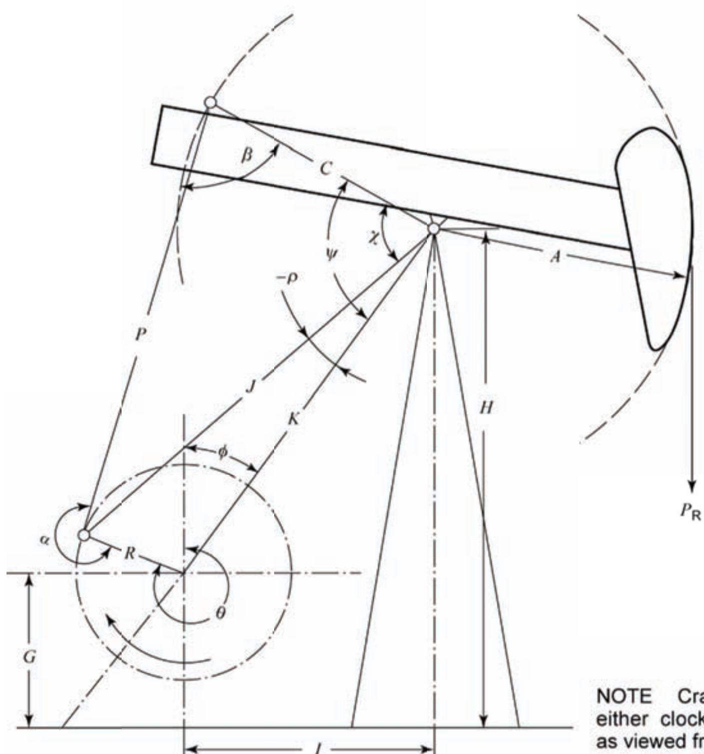
$$T_n = T_{wn} - T_r$$

T_r torque due to the rotary counterweights, cranks, and crankpins for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

$$T_r = M \sin\theta$$



a) Upstroke



b) Downstroke

NOTE Crank rotation is defined as either clockwise or counterclockwise as viewed from the side of the pumping unit with the wellhead to the right.

Figure D.1—Pumping Unit Geometry

T_{wn} torque due to the net polished rod load for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

$T_{wn} = \mathrm{TF} \times W_n$

TF torque factor for a given crank angle θ, in inches (in.) [meters (m)]

W_n net polished rod load, in pounds (lb) [newtons (N)]

$W_n = P_R - B$

α angle between P and R measured clockwise from R to P

β angle between C and P

χ angle between C and J

ρ angle between K and J

θ angle of crank rotation in a clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 12 o'clock

ϕ angle between the 12 o'clock position and K

ψ angle between C and K

$\psi = \chi - \rho$

ψ_b angle between C and K, at bottom (lowest) polished rod position

ψ_t angle between C and K, at top (highest) polished rod position

D.3 Method of Calculation

D.3.1 Torque Factors

Torque factors (as well as the polished rod position) may be determined by a scale layout of the unit geometry so that the various angles involved can be measured. They may alternatively be calculated from the dimensions of the pumping unit by mathematical treatment only.

D.3.2 Submission Form

A form for submission of torque factor and polished rod position data is shown in Figure C.2.

D.3.3 Data Submission

Torque factors and polished rod positions shall be furnished by pumping unit manufacturers for each 15° crank position with the zero position at 12 o'clock. Other crank positions shall be determined by the angular displacement in a clockwise direction viewed with the wellhead to the right. The polished rod position for each crank position should be expressed as a fraction of the stroke above the lowermost position.

D.3.4 Calculation Method

Application of the laws of trigonometric functions give the following expressions. All angles are calculated in terms of a given crank angle θ.

In USC units:

$$\mathrm{TF} = \frac{AR}{C}\left(\frac{\sin\alpha}{\sin\beta}\right) \quad \text{(expressed in inches)} \tag{D.1}$$

In SI units:

$$\mathrm{TF} = \frac{AR}{1000C}\left(\frac{\sin\alpha}{\sin\beta}\right) \quad \text{(expressed in meters)} \tag{D.1}$$

Sinα is positive when the angle α is between 0° and 180° and is negative when angle α is between 180° and 360°. Sinβ is always positive because the angle β is always between 0° and 180°. A negative torque factor (TF) only indicates a change in direction of torque on the crankshaft.

$$\phi = \tan^{-1}\left(\frac{I}{H-G}\right) \tag{D.2}$$

This is a constant angle for any given pumping unit.

$$\beta = \cos^{-1}\left[\frac{C^2 + P^2 - K^2 - R^2 + 2KR\cos(\theta-\phi)}{2CP}\right] \tag{D.3}$$

Cos($\theta-\phi$) is positive when ($\theta-\phi$) is between 270° and 90° moving clockwise and is negative from 90° to 270° moving clockwise. When the angle ($\theta-\phi$) is negative, it should be subtracted from 360°, and the following equations, Equations (D.4) and (D.5), apply.

$$\chi = \cos^{-1}\left(\frac{C^2 + J^2 - P^2}{2CJ}\right) \tag{D.4}$$

$$\rho = \sin^{-1} \pm\left[\frac{R\sin(\theta-\phi)}{J}\right] \tag{D.5}$$

The angle ρ should be taken as a positive angle when sinρ is positive. This occurs for crank positions between ($\theta-\phi$) = 0° and ($\theta-\phi$) = 180°. The angle ρ should be taken as a negative angle when sinρ is negative. This occurs for crank positions between ($\theta-\phi$) = 180° and ($\theta-\phi$) = 360°.

$$\psi = \chi - \rho \tag{D.6}$$

$$\alpha = \beta + \psi - (\theta - \phi) \tag{D.7}$$

$$\mathrm{PRP} = \frac{\psi_\mathrm{b} - \psi}{\psi_\mathrm{b} - \psi_\mathrm{t}} \tag{D.8}$$

$$\psi_\mathrm{b} = \cos^{-1}\left[\frac{C^2 + K^2 - (P+R)^2}{2CK}\right] \tag{D.9}$$

$$\psi_\mathrm{t} = \cos^{-1}\left[\frac{C^2 + K^2 - (P-R)^2}{2CK}\right] \tag{D.10}$$

D.4 Application of Torque Factors

D.4.1 General

Torque factors are used primarily for determining peak crankshaft torque on operating pumping units. The procedure is to take a dynamometer card and then use torque factors, polished rod position factors, and counterbalance information to plot the net torque curve. Example forms for recording calculations are provided in Figure D.4 and Figure D.5.

Points for plotting the net torque curve are calculated from the following equation (see Note):

$$T_n = \mathrm{TF}(P_R - B) - M\sin\theta \qquad \text{(D.11)}$$

NOTE This equation applies to pumping units where maximum counterbalance moment is obtained at θ equals 90° or 270°.

D.4.2 Changes Due to Structural Unbalance

The equation for net crankshaft torque, T_n, does not include the change in structural unbalance with change in crank angle; neglects the inertia effects of beam, beam weights, equalizer, pitman, crank, and crank counterweights; and neglects friction in the saddle, tail, and pitman bearings. For units having 100 % crank counterbalance and where crank-speed variation is not more than 15 % of average, these factors usually can be neglected without introducing errors greater than 10 %. When beam weights are used, the inertia effects of the weights should be included to determine peak torque with any degree of accuracy. The procedure for including the inertia effect of beam counterweights has been omitted because of the limited use of this type of counterbalance. Some nondynamic factors that can have an effect on the determination of instantaneous net torque loadings, and which accordingly should be recognized or considered, are outlined in D.4.7, D.4.8, and D.4.9.

D.4.3 Polish Rod Effects

Torque factors may be used to obtain the effect at the polished rod of the rotary counterbalance. This is done for a given crank angle by dividing the counterbalance moment, $M\sin\theta$, by the torque factor for the crank angle θ. The result is the rotary counterbalance effect at the polished rod.

D.4.4 Rotary Counterbalance Moment

Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the cranks in the 90° or 270° position and tying off the polished rod. Then, with a polished rod dynamometer, the counterbalance effect is measured at the polished rod. Using this method, the measured polished rod load (P_R) is the combined effect of the rotary counterbalance and the structural unbalance. The maximum rotary counterbalance moment can then be determined from the following equation:

$$M = \mathrm{TF}(P_R - B) \qquad \text{(D.12)}$$

To check measurements, the maximum moment, M, should be determined with the cranks in both the 90° and 270° positions. If there is a significant difference in the maximum moments calculated from measurements at 90° and 270°, a recheck of polished rod measurements and crank positions should be made. However, if there is only a slight difference, a satisfactory check is indicated and it is recommended that an arithmetic average of the two maximum moments be used.

EXAMPLE 1 [5]
To illustrate the use of torque factors, an example dynamometer card taken on a 4000 ft (1219 m) well is shown in Figure D.2. The first step in calculating the net crankshaft torque is to divide the dynamometer card so that the load may be determined for each 15° of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length, which is proportional to the length of the stroke. The length of the baseline or zero line is then divided into 10 equal parts and these parts are subdivided. This may easily be done with a suitable scale along a suitable diagonal line as shown (see Note).

NOTE Using the polished rod position data, vertical lines representing each 15° of crank angle θ are projected upward to intersect the dynamometer card. Then the polished rod load may be determined for each 15° of crank angle θ.

EXAMPLE 2
To further illustrate, a calculation is made considering the point where the crank angle θ equals 75°. From polished rod stroke and torque factor data for the particular 64 in. (1625 mm) stroke 160-D pumping unit used for this example, it is found that the position of the polished rod at 75° is 0.397 and that the torque factor TF is 34.38 in. (0.873 m). A vertical line is drawn from the 0.397 position on the scale up to the point of intersection with the load on the upstroke (Figure D.2). The dynamometer deflection at this point is read to be 1.16 in. (29.46 mm), which with a scale constant of 7,450 lb/in. (1305 N/mm), makes the load (P_R) at that point 8,650 lb (38,450 N).

EXAMPLE 3
In a similar manner, the polished rod load may be obtained for each 15° angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15° of crank angle. The structural unbalance, B, for the example unit equals +650 lb (+2891 N). Therefore, the net polished rod load, W_n, at θ = 75 is:

In USC units:

$$W_n = P_R - B = 8650 - (+650) = 8000 \text{ lb}$$

In SI units:

$$W_n = P_R - B = 38{,}450 - (+2{,}891) = 35{,}559 \text{ N}$$

The torque, T_{wn}, due to the net polished rod load is given by:

In USC units:

$$T_{wn} = \mathrm{TF}\left(P_R\right) = 34.38(8{,}000) = 275{,}000 \text{ in.-lb}$$

In SI units:

$$T_{wn} = \mathrm{TF}(P_R - B) = 0.873(35{,}559) = 31{,}043 \text{ Nm}$$

[5] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

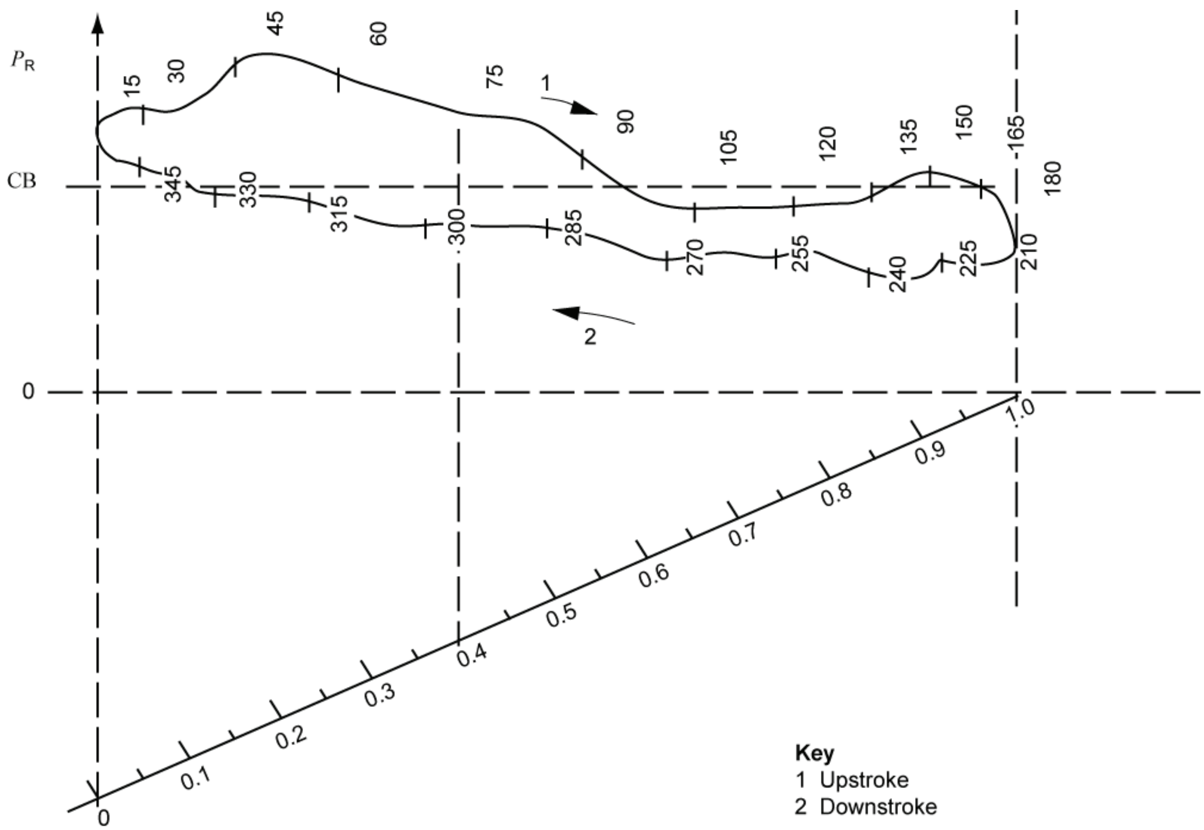


Figure D.2—Division of Dynamometer Card by Crank Angle Using Polished Rod Position Data

D.4.5 Torque Determination

To find the torque, T_r, due to the crank counterbalance, the maximum moment, M, has to be determined. This may be done either from manufacturers' counterbalance tables or curves, or as described in D.4.4. Because of the lack of manufacturers' counterbalance data in a majority of the cases, the polished rod measurement technique will be used more frequently in determining the maximum moment. Should the manufacturers' counterbalance data be used, it is suggested that a check be made using a polished rod measurement technique.

EXAMPLE [6] The horizontal dotted line drawn across the dynamometer card in Figure D.2 is the counterbalance effect measured with the dynamometer at the 90° crank angle and is 6,250 lb (27,800 N). The maximum moment can then be calculated as follows, using Equation (D.12):

In USC units:

$$M = \mathrm{TF}(P_\mathrm{R} - B) = 32.76(6{,}250 - 650) = 183{,}000 \text{ in.-lb}$$

(The torque factor of 32.76 in. is the value at the 90° crank position for the example unit.)

[6] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

In SI units:

$$M = \mathrm{TF}(P_\mathrm{R} - B) = 0.832(27{,}800 - 2{,}891) = 20{,}726 \text{ Nm}$$

(The torque factor of 0.832 m is the value at the 90° crank position for the example unit.)

Although not shown, the measured counterbalance effect for the 270° crank position was 6,410 lb (27,800 N). Using the torque factor of 32.04 in. (0.813 m) at the 270° crank position for the example unit, the maximum moment is: [7]

In USC units:

$$M = 32.04(6{,}410 - 650) = 185{,}000 \text{ in.-lb}$$

In SI units:

$$M = 0.813(28{,}513 - 2{,}891) = 20{,}831 \text{ Nm}$$

The maximum moments determined at the 90° and 270° crank positions are in good agreement, and the average maximum moment of 184,000 in.-lb (20,778 Nm) will be used.

The torque, T_r, due to the counterbalance at the 75° crank position would therefore be equal to:

In USC units:

$$T_\mathrm{r} = 184{,}000(\sin 75^\circ) = 184{,}000(0.966) = 178{,}000 \text{ in.-lb}$$

In SI units:

$$T_\mathrm{r} = 20{,}778(\sin 75^\circ) = 20{,}778(0.966) = 20{,}070 \text{ Nm}$$

The net torque at the crankshaft for the 75° crank position would then be calculated from Equation (D.11) as follows:

In USC units:

$$T_\mathrm{n} = \mathrm{TF}(P_\mathrm{R} - B) - M\sin\theta$$

$$= 34.88(8{,}650 - 650) - (184{,}000 \times 0.966)$$

$$= 275{,}000 - 178{,}000 = 97{,}000 \text{ in.-lb}$$

In SI units:

$$T_\mathrm{n} = \mathrm{TF}(P_\mathrm{R} - B) - M\sin\theta$$

$$= 0.873(38{,}450 - 2{,}891) - (20{,}778 \times 0.966)$$

$$= 31{,}040 - 20{,}070 = 10{,}970 \text{ Nm}$$

These values may be calculated for other crank angle positions in the same manner as outlined above. Shown in Figure D.3 is a plot of torque versus crank angle that includes the net polished rod load torque curve, the counterbalance torque curve, and the net crankshaft torque curve.

---

7 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

D.4.6 Alternative Crank Rotation

The foregoing example on the use of torque factors has been based on the pumping unit operating with the cranks rotating toward the well from top dead center. If the pumping unit is operating with the cranks rotating away from the well from top dead center, the calculation technique is changed only in the use of the torque factor in polished rod position data form (see Figure C.2). The angle of the crank (column 1) is reversed, starting from the bottom with 15° and counting up in 15° increments to 360°.

D.4.7 Alternative Techniques

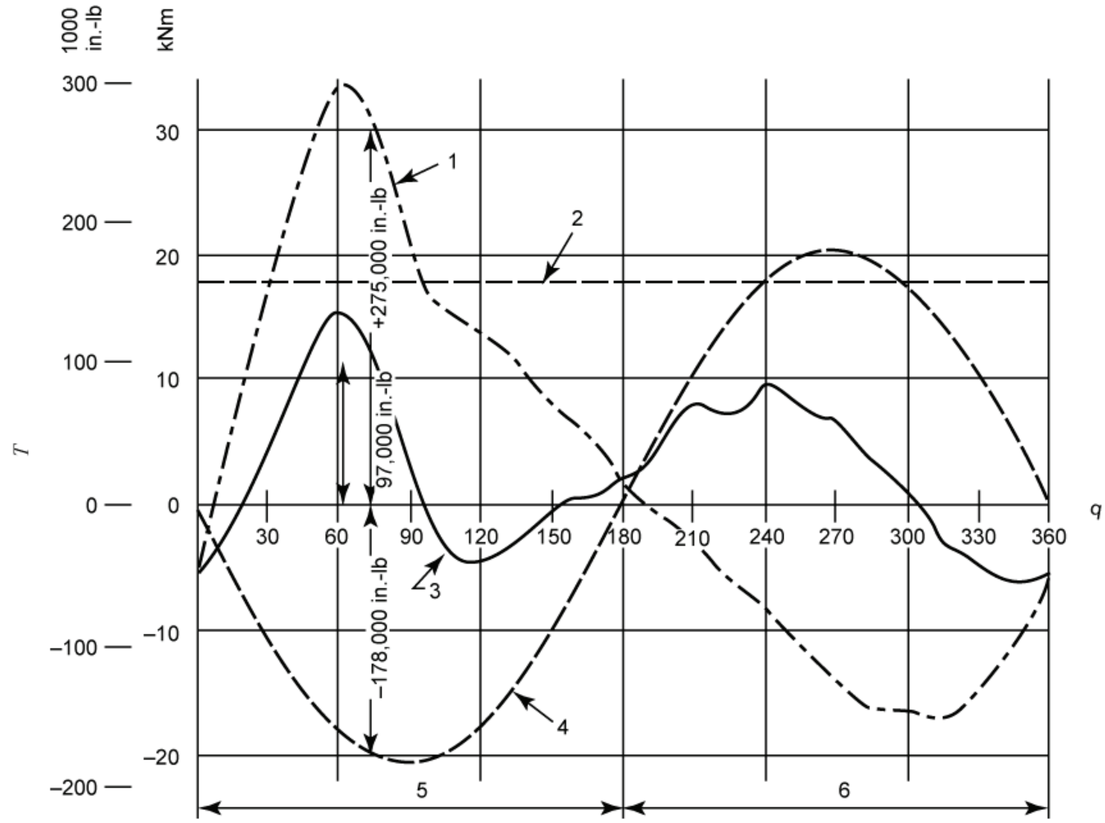
The foregoing technique is generally accepted. More precise results are dependent on the true stroke length, which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer.

D.4.8 Geometrical Influences

The geometry of the dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. The dynamometer manufacturer should be contacted for the performance characteristics of the particular dynamometer being used and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

D.4.9 Interpolation

Maximum and minimum loads will most frequently fall at points other than the 15° divisions for which torque factors are provided. Interpolation between 15° divisions is permissible without significant error.



Key

1	Well torque	2	160-D reducer limit	3	Net gear torque
4	Counterbalance torque	5	Upstroke	6	Downstroke

Figure D.3—Torque Curves Using Torque Factors

Net Reducer Torque Calculation Sheet (Conventional Crank Balanced Unit Only—Clockwise Rotation)

Well No.: ________________ Company: ________________

Unit Size: ________________ Location: ________________

θ	$\sin\theta$	P_R	B	$P_R - B$	TF	$TF(P_R - B)$	$-M(\sin\theta)$	T_n
0°	0						0	
15°	0.259						–	
30°	0.500						–	
45°	0.707						–	
60°	0.866						–	
75°	0.966						–	
90°	1.000						–	
105°	0.966						–	
120°	0.866						–	
135°	0.707						–	
150°	0.500						–	
165°	0.259						–	
180°	0						0	
195°	–0.259						+	
210°	–0.500						+	
225°	–0.707						+	
240°	–0.866						+	
255°	–0.966						+	
270°	–1.000						+	
285°	–0.966						+	
300°	–0.866						+	
315°	–0.707						+	
330°	–0.500						+	
345°	–0.259						+	
					Date Prepared			

NOTE $T_n = TF(P_R - B) - M\sin\theta$

where

T_n is the net reducer torque, in.-lb (Nm);

θ is the position of crank;

M is the maximum moment of counterbalance, in.-lb (Nm);

P_R is the measured polished rod load at θ, lb (N);

B is the unit structural unbalance, lb (N);

TF is the torque factor at θ, in. (m);

CB at 90° = ________________;

M = $TF_{at\ 90°}(CB_{at\ 90°} - B)$ = ______________.

Figure D.4—Calculation Sheet—Clockwise Rotation

Net Reducer Torque Calculation Sheet (Conventional Crank Balanced Unit Only—Counterclockwise Rotation)

Well No.: ______________________ Company: ______________________

Unit Size: ______________________ Location: ______________________

θ	$\sin\theta$	P_R	B	$P_R - B$	TF	$TF(P_R - B)$	$-M(\sin\theta)$	T_n
0°	0						0	
345°	–0.259						+	
330°	–0.500						+	
315°	–0.707						+	
300°	–0.866						+	
285°	–0.966						+	
270°	–1.000						+	
255°	–0.966						+	
240°	–0.866						+	
225°	–0.707						+	
210°	–0.500						+	
195°	–0.259						+	
180°	0						0	
165°	0.259						–	
150°	0.500						–	
135°	0.707						–	
120°	0.866						–	
105°	0.966						–	
90°	1.000						–	
75°	0.966						–	
60°	0.866						–	
45°	0.707						–	
30°	0.500						–	
15°	0.259						–	
					Date Prepared			

NOTE $T_n = TF(P_R - B) - M \sin\theta$

where

T_n is the net reducer torque, in.-lb (Nm);

θ is the position of crank;

M is the maximum moment of counterbalance, in.-lb (Nm);

P_R is the measured polished rod load at θ, lb (N);

B is the unit structural unbalance lb (N);

TF is the torque factor at θ, in. (m);

CB at 270° = ________________;

M = $TF_{\text{at } 270°}(CB_{\text{at } 270°} - B)$ = ______________.

Figure D.5—Calculation Sheet—Counterclockwise Rotation

Annex E
(informative)

Torque Factor on Beam Pumping Units with Front Mounted Geometry Class III Lever Systems with Crank Counterbalance

E.1 General

The following calculation technique is generally accepted. More precise results are dependent on the true stroke length, which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer. Determinations made with a dynamometer can determine more specific performance characteristics of the individual pumping unit.

E.2 Symbols

In addition to those listed in Section 2, the following system of nomenclature and symbols is used in this annex (see Figure E.1):

B structural unbalance, equal to the force at the polished rod required to hold the beam in a horizontal position with the pitmans disconnected from the crankpins, in pounds (lb) [newtons (N)]

NOTE This force acts upward on Class III geometry units and is negative.

G height from the center of the crankshaft to the bottom of the base beams, in inches (in.) [millimeters (mm)]

H height from the center of the Samson post bearing to the bottom of the base beams, in inches (in.) [millimeters (mm)]

I horizontal distance between the centerline of the Samson post bearing and the centerline of the crankshaft, in inches (in.) [millimeters (mm)]

J distance from the center of the crankpin bearing to the center of the Samson post bearing, in inches (in.) [millimeters (mm)]

K distance from the center of the crankshaft to the center of the Samson post bearing, 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)]

P effective length of the pitman (from the center of the equalizer, or cross yoke, bearing to the center of the crankpin bearing), in inches (in.) [millimeters (mm)]

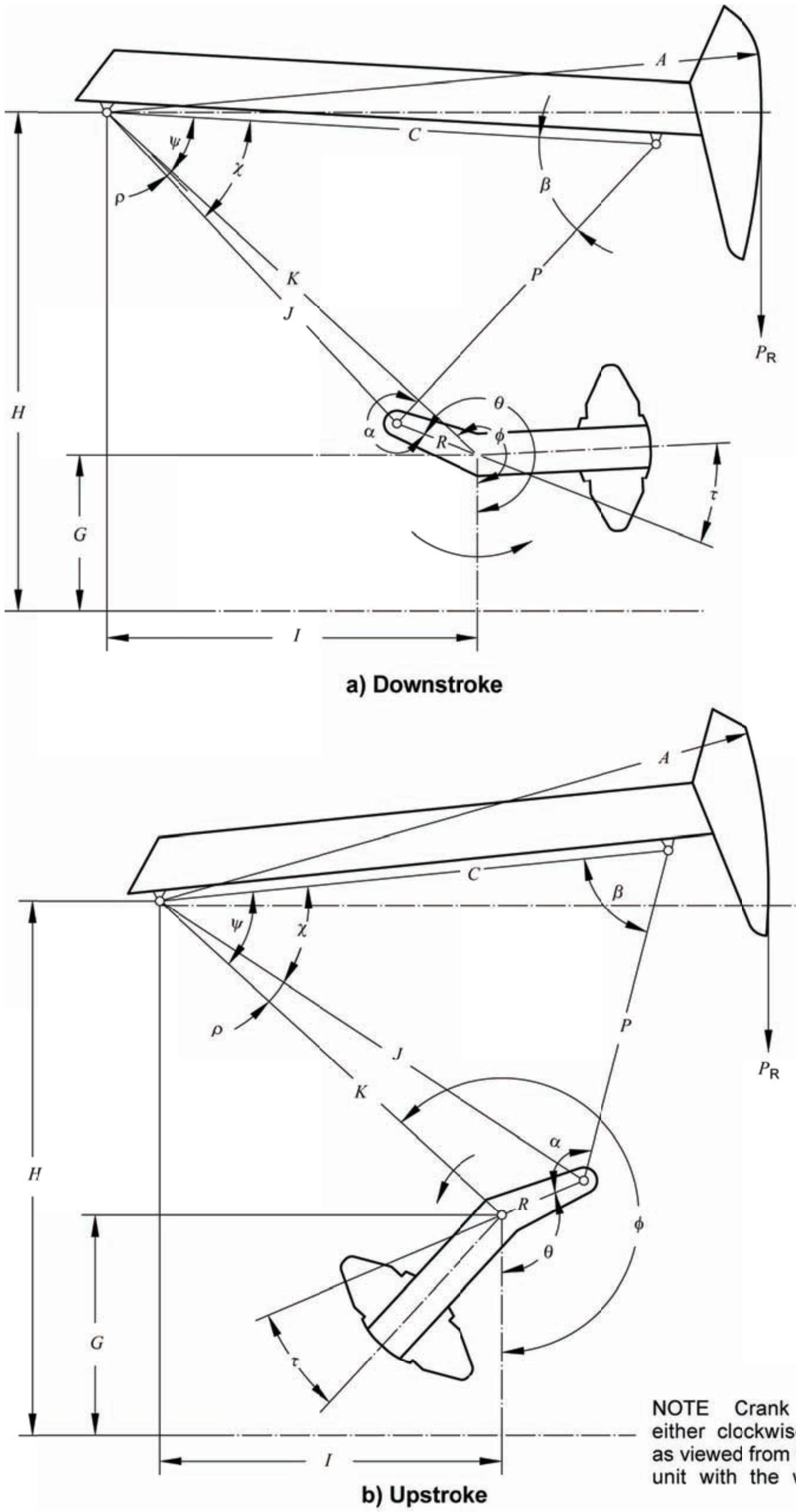
PRP polished rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle θ

T_n net torque at the crankshaft for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

$$T_n = T_{wn} - T_r$$

T_r torque due to the rotary counterweights, cranks, and crankpins for a given crank angle θ, (in.-lb) [newton-meters (Nm)]

$$T_r = M \sin(\theta + \tau)$$



a) Downstroke

b) Upstroke

NOTE Crank rotation is defined as either clockwise or counterclockwise as viewed from the side of the pumping unit with the wellhead to the right.

Figure E.1—Front Mounted Geometry, Class III Lever System

T_{wn} torque due to the net polished rod load for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

$$T_{wn} = \mathrm{TF} \times W_n$$

TF torque factor for a given crank angle θ in feet (ft) [meters (m)]

W_c counterbalance at the polished rod, determined using a dynamometer with crankpin at 90° position, in pounds (lb) [newtons (N)]

W_n net polished rod load, in pounds (lb) [newtons (N)]

$$W_n = P_R - B$$

α angle between P and R measured clockwise from R to P

β angle between C and P

χ angle between C and J

ρ angle between K and J

τ angle of crank counterweight arm offset for front mounted geometry (Class III lever systems)

θ angle of crank rotation in a counterclockwise direction viewed with the wellhead to the right and with zero degrees occurring at 6 o'clock

ϕ angle between the 6 o'clock position and K

ψ angle between C and K

$$\psi = \chi - \rho$$

ψ_b angle between C and K, at bottom (lowest) polished rod position

ψ_t angle between C and K, at top (highest) polished rod position

E.3 Method of Calculation

E.3.1 Torque Factors

Torque factors (as well as the polished rod position) may be determined by a scale layout of the unit geometry so that the various angles involved can be measured. They may alternatively be calculated from the dimensions of the pumping unit by mathematical treatment only.

E.3.2 Submission Form

A form for submission of torque factor and polished rod position data is shown in Figure C.2.

E.3.3 Data Submission

Torque factors and polished rod positions shall be furnished by pumping unit manufacturers for each 15° crank position with the zero position at 6 o'clock. Other crank positions shall be determined by the angular displacement in a counterclockwise direction viewed with the wellhead to the right. The polished rod position for each crank position should be expressed as a fraction of the stroke above the lowermost position.

E.3.4 Calculation Method

Application of the laws of trigonometric functions give the following expressions. All angles are calculated in terms of a given crank angle θ.

In USC units:

$$\mathrm{TF} = \frac{AR}{C}\left(\frac{\sin\alpha}{\sin\beta}\right) \qquad \text{(expressed in inches)} \qquad \text{(E.1)}$$

In SI units:

$$\mathrm{TF} = \frac{AR}{1000C}\left(\frac{\sin\alpha}{\sin\beta}\right) \qquad \text{(expressed in meters)} \qquad \text{(E.1)}$$

Sinα is positive when the angle α is between 0° and 180° and is negative when angle α is between 180° and 360°. Sinβ is always positive because the angle β is always between 0° and 180°. A negative torque factor (TF) only indicates a change in direction of torque on the crankshaft.

$$\phi = \tan^{-1}\left(\frac{I}{H - G_{\tau}}\right) + 180^{\circ} \qquad \text{(E.2)}$$

This is a constant angle for any given pumping unit.

$$\beta = \cos^{-1}\left[\frac{C^2 + P^2 - K^2 - R^2 + 2KR\cos(\theta - \phi)}{2CP}\right] \qquad \text{(E.3)}$$

It is important to get the sign of cos($\theta - \phi$) correct. Cos($\theta - \phi$) is negative when ($\theta - \phi$) is between 90° and 270° and positive for all other angles. When the angle ($\theta - \phi$) is negative, it should be subtracted from 360°, and the following equations, Equation (E.4) and Equation (E.5), apply.

$$\chi = \sin^{-1}\left(\frac{P\sin\beta}{J}\right) \qquad \text{(E.4)}$$

$$\rho = \sin^{-1}\left[\frac{R\sin(\theta - \phi)}{J}\right] \qquad \text{(E.5)}$$

In regards to the diagrams in Figure E.1, ρ shall be taken as positive in the upstroke diagram [Figure E.1 a)], and negative in the downstroke diagram [see Figure E.1 b)].

For Equation (E.6) to be correct it is necessary to use the correct sign for the angle ρ. The angle ρ should be taken as positive when sinρ is positive. This occurs for crank positions when 0° < ($\theta - \phi$) < 180°. The angle ρ is taken as negative when sin ρ is negative. This occurs for crank positions when 180° < ($\theta - \phi$) < 360°.When the angle ($\theta - \phi$) is negative it can be subtracted from 360°, and this new angle can be used to determine the proper sign.

$$\psi = \chi - \rho \qquad \text{(E.6)}$$

$$\sin\alpha = \sin\left[(\theta - \phi) - \psi - \beta\right] \qquad \text{(E.7)}$$

$$\mathrm{PRP} = \frac{\psi_{\mathrm{b}} - \psi}{\psi_{\mathrm{b}} - \psi_{\mathrm{t}}} \qquad \text{(E.8)}$$

$$\psi_{\mathrm{t}} = \cos^{-1}\left[\frac{C^2 + K^2 - (P+R)^2}{2CK}\right] \tag{E.9}$$

$$\psi_{\mathrm{b}} = \cos^{-1}\left[\frac{C^2 + K^2 - (P-R)^2}{2CK}\right] \tag{E.10}$$

E.4 Application of Torque Factors

E.4.1 General

Torque factors are used primarily for determining peak crankshaft torque on operating pumping units. The procedure is to take a dynamometer card and then use torque factors, polished rod position factors, and counterbalance information to plot the net torque curve. Points for plotting the net torque curve are calculated from the following equation:

$$T_{\mathrm{n}} = \mathrm{TF}(P_{\mathrm{R}} - B) - M\sin(\theta + \tau) \tag{E.11}$$

E.4.2 Changes Due to Structural Unbalance

The equation for net crankshaft torque, T_{n}, does not include the change in structural unbalance with change in crank angle; neglects the inertia effects of beam, beam weights, equalizer (or cross yoke), pitman, crank, and crank counterweights; and neglects friction in the bearings. For units having 100 % crank counterbalance and where crank-speed variation is not more than 15 % of average, these factors usually can be neglected without introducing errors greater than 10 %. Some nondynamic factors that can have an effect on the determination of instantaneous net torque loadings, and which accordingly should be recognized or considered, are outlined in E.4.6, E.4.7, and E.4.8.

E.4.3 Polished Rod Effects

Torque factors may be used to obtain the effect at the polished rod of the rotary counterbalance. This is done for a given crank angle by dividing the counterbalance moment, $M\sin(\theta + \tau)$, by the torque factor for the crank angle θ. The result is the rotary counterbalance effect at the polished rod.

E.4.4 Rotary Counterbalance Moment

Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the cranks in the 90° position and tying off the polished rod. Then, with a polished rod dynamometer, the counterbalance effect is measured at the polished rod. Using this method, the measured polished rod load (W_{c}) is the combined effect of the rotary counterbalance and the structural unbalance. The maximum rotary counterbalance moment can then be determined from the following equation:

$$M = \frac{\mathrm{TF}(W_{\mathrm{c}} - B)}{\sin(90^\circ + \tau)} \tag{E.12}$$

EXAMPLE 1 [8]
To illustrate the use of torque factors, an example dynamometer card taken on a 2872 ft (875 m) well is shown in Figure E.2. The first step in calculating the net crankshaft torque is to divide the dynamometer card so that the load may be determined for each 15° of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length, which is proportional to the length of the stroke. The length of the baseline or zero line is then divided into 10 equal parts and these parts are subdivided. This may easily be done with a suitable scale along a suitable diagonal line as shown (see Note).

NOTE Using the polished rod position data, vertical lines representing each 15° of crank angle θ are projected upward to intersect the dynamometer card. Then the polished rod load may be determined for each 15° of crank angle θ.

To further illustrate, a calculation is made considering the point where the crank angle θ equals 60°. From polished rod stroke and torque factor data for the particular 86 in. (2184 mm) stroke 160-D pumping unit used for this example, it is found that the position of the polished rod at 60° is 0.405 and that the torque factor TF is 35.45 in. (0.926 m). A vertical line is drawn from the 0.405 position on the scale up to the point of intersection with the load on the upstroke (Figure E.2). The dynamometer deflection at this point is read to be 0.99 in. (25.1 mm), which with a scale constant of 7500 lb/in. (1,313 N/mm), makes the load (P_R) at that point 7425 lb (32,970 N).

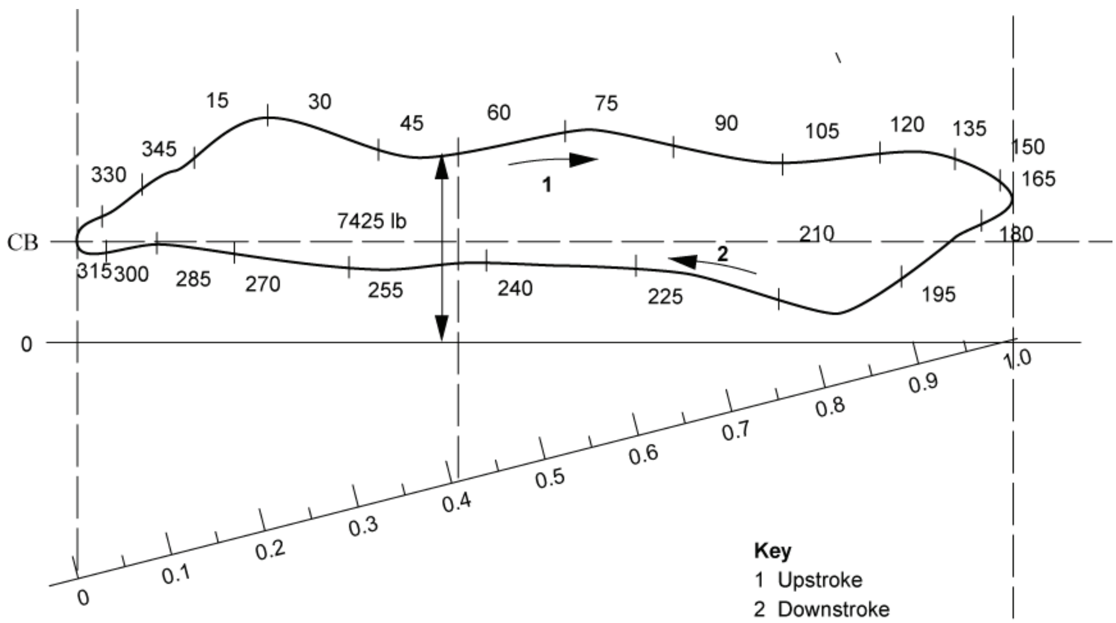


Figure E.2—Division of Dynamometer Card by Crank Angle Using Polished Rod Position Data

In a similar manner, the polished rod load may be obtained for each 15° angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15° of crank angle. The structural unbalance, B, for the example unit equals –1535 lb (–6830 N). Therefore, the net polished rod load, W_n, at $\theta = 60$ is:

In USC units:

$$W_n = P_R - B = 7425 - (-1535) = 8960 \text{ lb}$$

[8] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

In SI units:

$$W_n = P_R - B = 32{,}970 - (-6{,}830) = 39{,}800 \text{ N}$$

The torque, T_{wn}, due to the net polished rod load is given by:

In USC units:

$$T_{wn} = \mathrm{TF}(P_R) = 36.45(8{,}960) = 326{,}592 \text{ in.-lb}$$

In SI units:

$$T_{wn} = \mathrm{TF}(P_R - B) = 0.926(39{,}800) = 36{,}860 \text{ Nm}$$

E.4.5 Torque Determination

To find the torque, T_r, due to the crank counterbalance, the maximum moment, M, has to be determined. This may be done either from manufacturers' counterbalance tables or curves, or as described in E.4.4. Should the manufacturers' counterbalance data be used, it is suggested that a check be made using a polished rod measurement technique.

EXAMPLE [9] The horizontal dotted line drawn across the dynamometer card in Figure E.2 is the counterbalance effect measured with the dynamometer at the 90° crank angle and is 4,594 lb (20,435 N). The maximum moment can then be calculated as follows, using Equation (E.12):

In USC units:

$$M = \frac{\mathrm{TF}(W_c - B)}{\sin(90^\circ + \tau)} = 38.38\left[(4{,}594 + 1{,}535)/0.891\right] = 264{,}008 \text{ in.-lb}$$

(The torque factor of 38.38 in. is the value at the 90° crank position, and angle τ is 27° for the example unit.)

In SI units:

$$M = \frac{\mathrm{TF}(W_c - B)}{\sin(90^\circ + \tau)} = 0.975\left[(20{,}435 + 6{,}828)/0.891\right] = 29{,}830 \text{ Nm}$$

(The torque factor of 0.975 m is the value at the 90° crank position, and angle τ is 27° for the example unit.)

The torque, T_r, due to the counterbalance at the 60° crank position would therefore be equal to:

In USC units:

$$T_r = 264{,}008 \times \sin(60^\circ + 27^\circ) = 264{,}008 \times 0.999 = 263{,}744 \text{ in.-lb}$$

In SI units:

$$T_r = 29{,}380 \times \sin(60^\circ + 27^\circ) = 29{,}790 \text{ Nm}$$

---

9 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

The net torque at the crankshaft for the 60° crank position would then be calculated from Equation (E.11) as follows: [10]

In USC units:

$$T_n = \mathrm{TF}(P_R - B) - M\sin(\theta + \tau)$$

$$T_n = T_{wn} - T_r$$

$$= 362{,}592 - 263{,}744 = 62{,}848 \text{ in.-lb}$$

In SI units:

$$T_n = \mathrm{TF}(P_R - B) - M\sin(\theta + \tau)$$

$$T_n = T_{wn} - T_r$$

$$= 36{,}860 - 29{,}790 = 7{,}070 \text{ Nm}$$

These values may be calculated for other crank angle positions in the same manner as outlined above. Shown in Figure E.3 is a plot of torque versus crank angle that includes the net polished rod load torque curve, the counterbalance torque curve, and the net crankshaft torque curve.

E.4.6 Alternative Techniques

The foregoing technique is generally accepted. More precise results are dependent on the true stroke length, which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer.

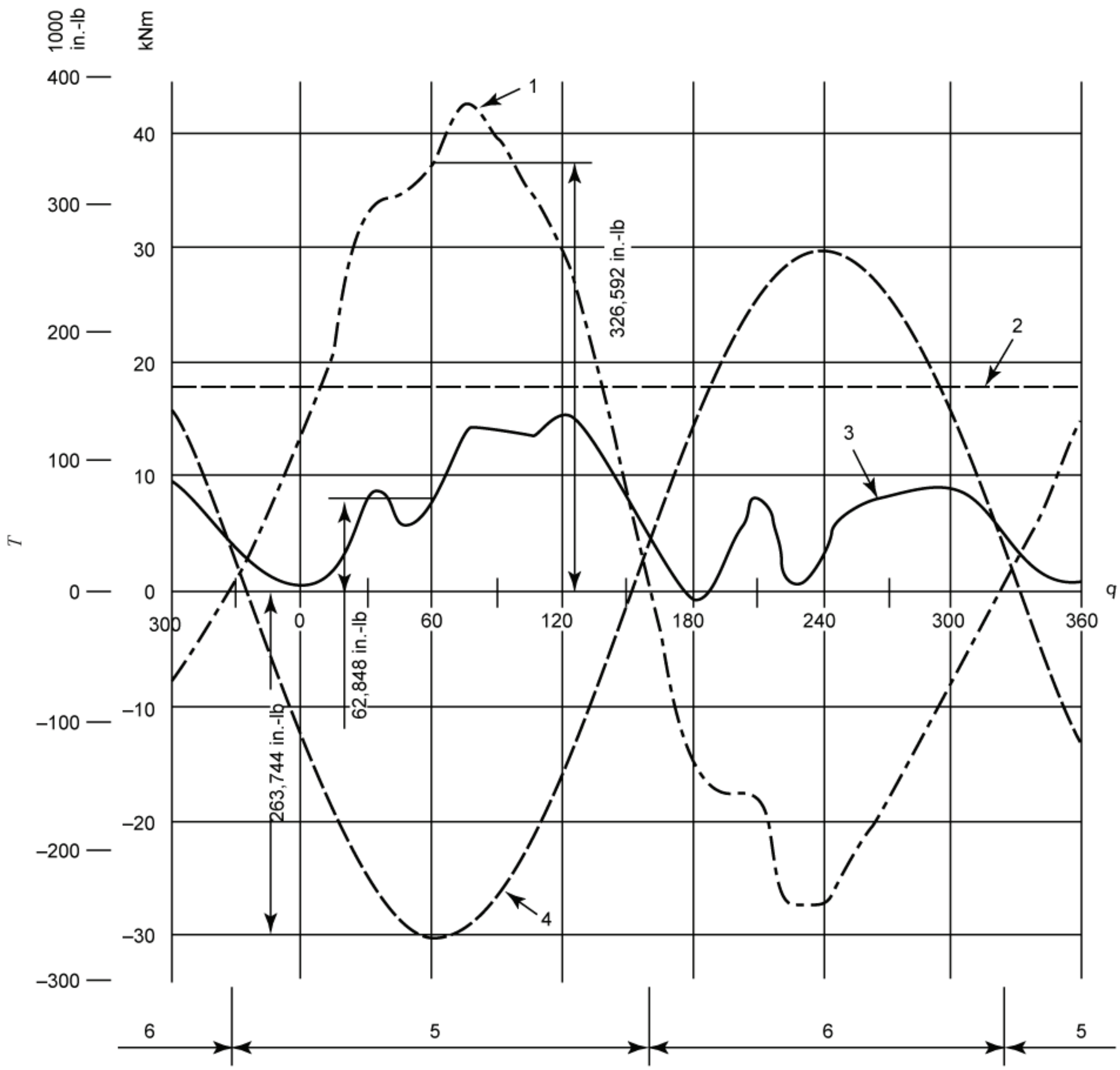
E.4.7 Geometrical Influences

The geometry of the dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. The dynamometer manufacturer should be contacted for the performance characteristics of the particular dynamometer being used and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

E.4.8 Interpolation

Maximum and minimum loads will most frequently fall at points other than the 15° divisions for which torque factors are provided. Interpolation between 15° divisions is permissible without significant error.

10 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.



Key

1	Well torque	2	160-D reducer limit	3	Net gear torque
4	Counterbalance torque	5	Upstroke	6	Downstroke

Figure E.3—Torque Curves Using Torque Factors

Annex F
(informative)

Torque Factor on Beam Pumping Units with Front Mounted Geometry Class III Lever System with Air Counterbalance

F.1 General

The following calculation technique is generally accepted. More precise results are dependent on the true stroke length which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer. Determinations made with a dynamometer can determine more specific performance characteristics of the individual pumping unit.

F.2 Symbols

In addition to those listed in Section 2, the following system of nomenclature and symbols is used in this annex (see Figure F.1):

G height from the center of the crankshaft to the bottom of the base beams, in inches (in.) [millimeters (mm)]

H height from the center of the Samson post bearing to the bottom of the base beams, in inches (in.) [millimeters (mm)]

I horizontal distance between the centerline of the Samson post bearing and the centerline of the crankshaft, in inches (in.) [millimeters (mm)]

J distance from the center of the crankpin bearing to the center of the Samson post bearing, in inches (in.) [millimeters (mm)]

where

$J = \sqrt{C^2 + P^2 - (2CP\cos\beta)}$

K distance from the center of the crankshaft to the center of the Samson post bearing, in inches (in.) [millimeters (mm)]

M_a geometry constant for a given unit (distance from Samson post bearing to air tank bearing multiplied by the area of the piston in the air cylinder divided by the distance from the Samson post bearing to the centerline of the polished rod), in square inches (in.^{2}) [square millimeters (mm^2)]

P effective length of the pitman (from the center of the equalizer bearing to the center of the crankpin bearing), in inches (in.) [millimeters (mm)]

P_a pressure in air counterbalance tank for a given crank position θ, in pounds per square inch (psi) [kilopascals (kPa)]

PRP polished rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle θ

S pressure in air counterbalance tank required to offset the weight of the walking beam, horsehead, equalizer, pitmans, etc., in pounds per square inch (psi) [kilopascals (kPa)]

T_n net torque at the crankshaft for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

TF torque factor for a given crank angle θ in feet (ft) [meters (m)]

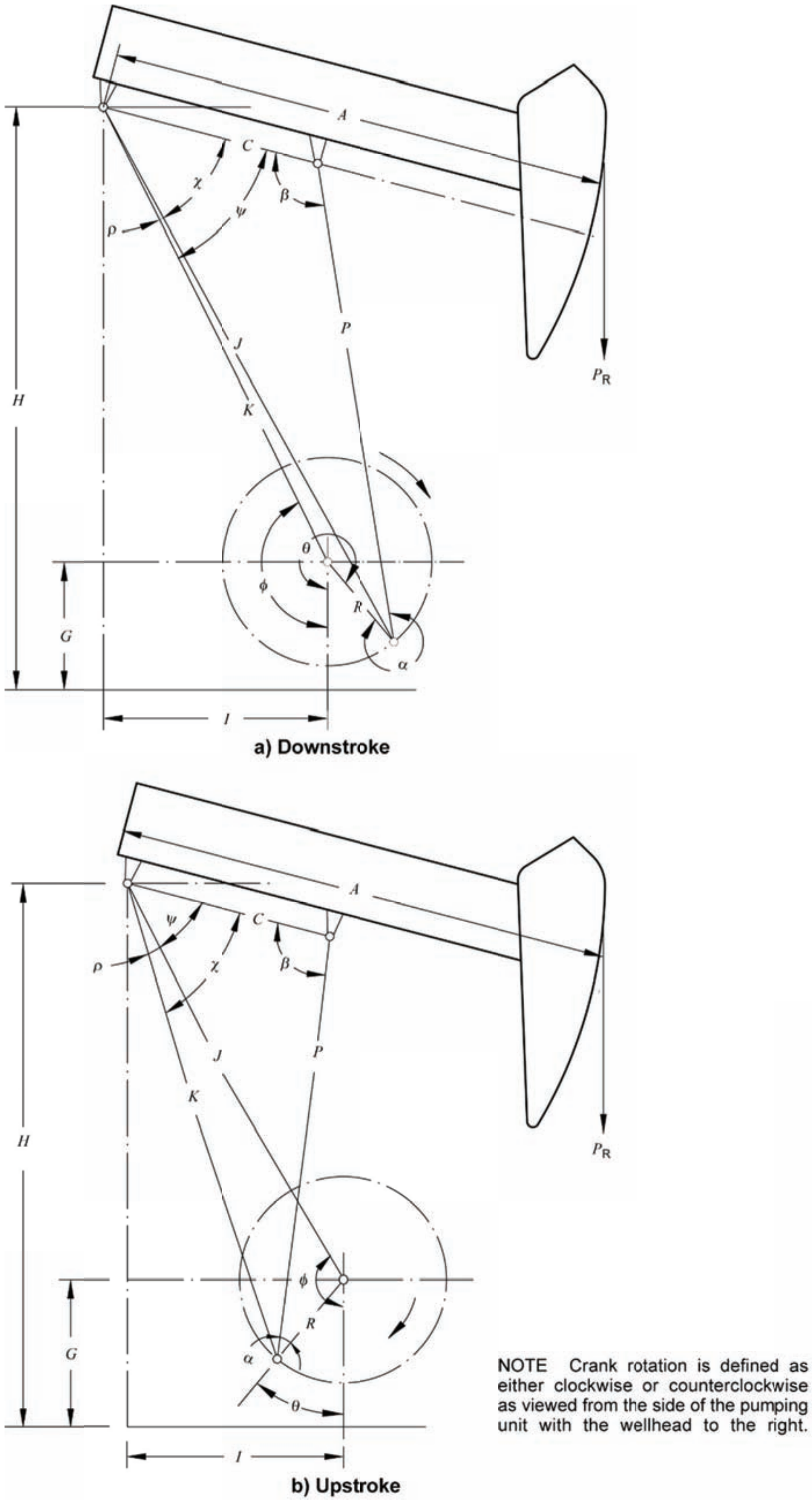


Figure F.1—Pumping Unit Geometry

W_c counterbalance effect at the polished rod at any specific crank angle θ, in pounds (lb) [newtons (N)]

$W_c = M_a(P_a - S)$

α angle between P and R measured clockwise from R to P

β angle between C and P

χ angle between C and J

ρ angle between K and J

θ angle of crank rotation in a clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 6 o'clock

ϕ angle between the 6 o'clock position and K

ψ angle between C and K

$\psi = \chi - \rho$

ψ_b angle between C and K, at bottom (lowest) polished rod position

ψ_t angle between C and K, at top (highest) polished rod position

F.3 Method of Calculation

F.3.1 Torque Factors

Torque factors (as well as the polished rod position) may be determined by a scale layout of the unit geometry so that the various angles involved can be measured. They may alternatively be calculated from the dimensions of the pumping unit by mathematical treatment only.

F.3.2 Submission form

A form for submission of torque factor and polished rod position data is shown in Figure C.2.

F.3.3 Data Submission

Torque factors and polished rod positions shall be furnished by pumping unit manufacturers for each 15° crank position with the zero position at 6 o'clock. Other crank positions shall be determined by the angular displacement in a clockwise direction viewed with the wellhead to the right. The polished rod position for each crank position should be expressed as a fraction of the stroke above the lowermost position.

F.3.4 Calculation Method

Application of the laws of trigonometric functions give the following expressions. All angles are calculated in terms of a given crank angle θ.

In USC units:

$$\mathrm{TF} = \frac{AR}{C}\left(\frac{\sin\alpha}{\sin\beta}\right) \quad \text{(expressed in inches)} \qquad \text{(F.1)}$$

In SI units:

$$\mathrm{TF} = \frac{AR}{1000C}\left(\frac{\sin\alpha}{\sin\beta}\right) \quad \text{(expressed in meters)} \qquad \text{(F.1)}$$

Sinα is positive when the angle α is between 0° and 180° and is negative when angle α is between 180° and 360°. Sinβ is always positive because the angle β is always between 0° and 180°. A negative torque factor (TF) only indicates a change in direction of torque on the crankshaft.

$$\phi = 180^\circ - \tan^{-1}\left(\frac{I}{H-G}\right) \tag{F.2}$$

This is a constant angle for any given pumping unit.

$$\beta = \cos^{-1}\left[\frac{C^2 + P^2 - K^2 - R^2 + 2KR\cos(\theta - \phi)}{2CP}\right] \tag{F.3}$$

Cos($\theta - \phi$) is positive when ($\theta - \phi$) is between 270° and 90° moving clockwise and is negative from 90° and 270° moving clockwise. When the angle ($\theta - \phi$) is negative, it should be subtracted from 360°, and the following equations, Equations (F.4) and (F.5), apply.

$$\chi = \sin^{-1}\left(\frac{P\sin\beta}{J}\right) \tag{F.4}$$

$$\rho = \sin^{-1}\left[\frac{R\sin(\theta - \phi)}{J}\right] \tag{F.5}$$

The angle ρ should be taken as positive when sinρ is positive. This occurs for crank positions when 0° < ($\theta - \phi$) < 180°.The angle ρ should be taken as negative when sinρ is negative. This occurs for crank positions when 180° < ($\theta - \phi$) < 360°.

$$\psi = \chi + \rho \tag{F.6}$$

$$\sin\alpha = \sin\left[\beta + \psi + (\theta - \phi)\right] \tag{F.7}$$

$$\mathrm{PRP} = \frac{\psi_\mathrm{b} - \psi}{\psi_\mathrm{b} - \psi_\mathrm{t}} \tag{F.8}$$

$$\psi_\mathrm{t} = \cos^{-1}\left[\frac{C^2 + K^2 - (P+R)^2}{2CK}\right] \tag{F.9}$$

$$\psi_\mathrm{b} = \cos^{-1}\left[\frac{C^2 + K^2 - (P-R)^2}{2CK}\right] \tag{F.10}$$

F.4 Application of Torque Factors

F.4.1 General

Torque factors are used primarily for determining peak crankshaft torque on operating pumping units. The procedure is to take a dynamometer card and then use torque factors, polished rod position factors, and counterbalance information to plot the net torque curve. Points for plotting the net torque curve are calculated from the following equation:

$$T_\mathrm{n} = \mathrm{TF}(P_\mathrm{R} - W_\mathrm{c}) \tag{F.11}$$

F.4.2 Changes Due to Structural Unbalance

The equation for net crankshaft torque, T_n, does not include the change in structural unbalance with change in crank angle; neglects the inertia effects of beam, beam weights, equalizer, pitman, and crank, and neglects friction in the Samson post, equalizer, and pitman bearings. For units where crank-speed variation is not more than 15 % of average, these factors usually can be neglected without introducing errors greater than 10 %. Some nondynamic factors that can have an effect on the determination of instantaneous net torque loadings, and which accordingly should be recognized or considered, are outlined in F.4.4, F.4.5, and F.4.6.

EXAMPLE 1 [11]
To illustrate the use of torque factors, an example dynamometer card taken on a 5560 ft (1695 m) well is shown in Figure F.2. The first step in calculating the net crankshaft torque is to divide the dynamometer card so that the load may be determined for each 15° of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length, which is proportional to the length of the stroke. The length of the baseline or zero line is then divided into 10 equal parts and these parts are subdivided. This may easily be done with a suitable scale along a suitable diagonal line as shown (see Note).

NOTE Using the polished rod position data, vertical lines representing each 15° of crank angle are projected upward to intersect the dynamometer card. Then the polished rod load may be determined for each 15° of crank angle.

The counterbalance line may then be drawn on the card. To avoid time-consuming geometrical considerations, it may be assumed that the counterbalance line is straight between the two end points of maximum and minimum counterbalance. The assumed counterbalance will be 3 % to 4 % lower than the actual counterbalance around the mid-point of the stroke, slightly higher at the bottom of the stroke, and nearly equal at the top of the stroke.

For the example calculation, the recorded maximum air counterbalance tank pressure at the bottom of the stroke, 0° crank position, was 328 psig (2261 kPa) and the minimum air pressure at the top of the stroke, 180° crank position, was 262 psig (1806 kPa). Using the equation, $W_c = M_a(P_a - S)$, where M_a = 52.5 in.^{2} (33,870 mm^2) and S is 73 psig (503 kPa) (as furnished by the pumping unit manufacturer), we calculate the following results.

a) Maximum counterbalance at the 0° crank position is W_c = 52.5 (328 – 73) = 13,388 lb [W_c = 33,870 (2261 – 503) = 59,543 N] counterbalance at the polished rod. 13,388 lb (59,543 N) divided by the scale constant, 11,300 lb/in. (1979 N/mm), gives 1.185 in. (30.1 mm).

b) Minimum counterbalance at the 180° crank position is W_c = 52.5 (262 – 73) = 9923 lb [W_c = 33,870 (1806 – 503) = 47,520 N]. 9923 lb (47,520 N) divided by 11,300 lb/in. (1979 N/mm) gives 0.878 in. (24.0 mm).

The counterbalance line can now be drawn on the dynamometer card as shown in Figure F.2.

EXAMPLE 2
To further illustrate, a calculation is made considering the point where the crank angle θ equals 75°. From polished rod stroke and torque factor data for the particular 86 in. (2184 mm) stroke 320-D pumping unit used for this example, it is found that the position of the polished rod at 75° is 0.332 and that the torque factor TF is 39.02 (0.991 m). A vertical line is drawn from the 0.332 position on the scale up to the point of intersection with the load on the upstroke (see Figure F.2). The dynamometer deflection at this point is read to be 1.45 in. (36.8 mm), which, with a scale constant of 11,300 lb/in. (1979 N/mm), makes the load (P_R) at that point 16,385 lb (72,886 N).

---

11 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

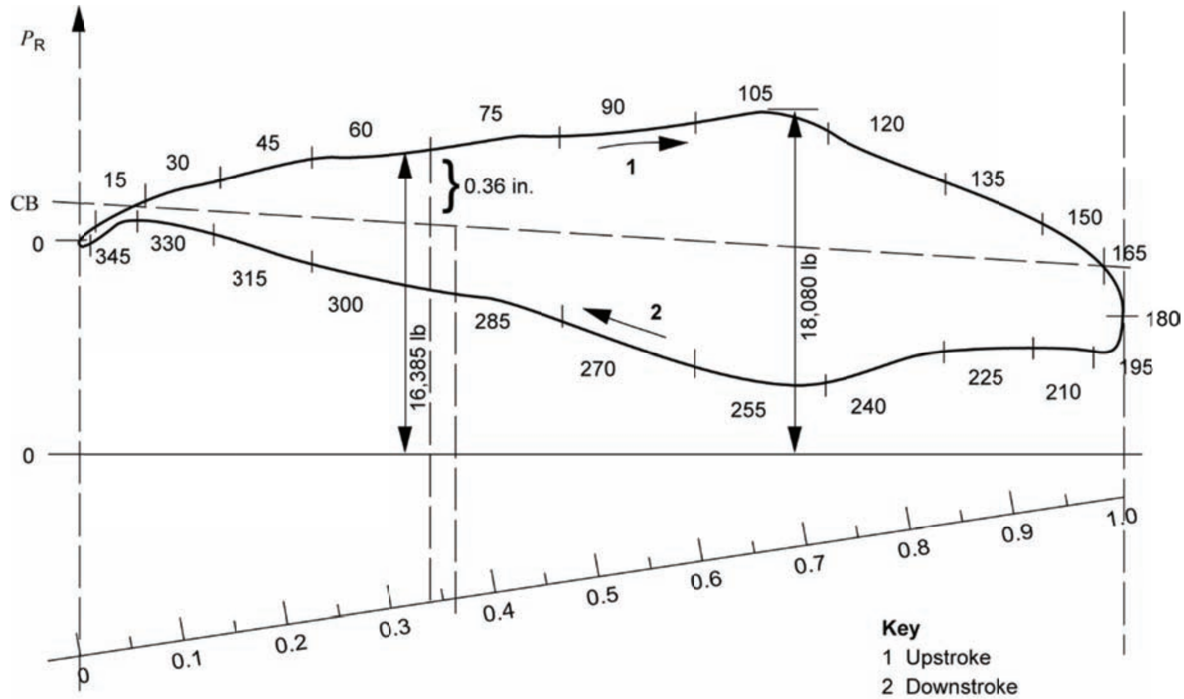


Figure F.2—Division of Dynamometer Card by Crank Angle Using Polished Rod Position Data

In a similar manner, the polished rod load may be obtained for each 15° angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15° of crank angle. However, it is usually only necessary to determine the maximum polished rod load, which in the example case occurs between the 105° and 120° crank position. The maximum dynamometer deflection at this point is 1.60 in. (40.6 mm) which when multiplied by the scale constant of 11,300 lb/in. (1979 N/mm) gives 18,080 lb (80,347 N) polished rod load.

The net torque, T_n, can now be determined. In the equation $T_n = \mathrm{TF}(P_R - W_c)$ the value $(P_R - W_c)$ is represented by the difference in the dynamometer deflection between the card and the counterbalance line. Referring to the card in Figure F.2, the difference in the dynamometer deflection between the counterbalance line and the well card is 0.36 in. (9.14 mm) at 75° crank position. This value multiplied by the scale constant of 11,300 lb/in. (1979 N/mm) and the torque factor of 39.25 in. (0.997 m) at 75° crank position gives 159,669 in.-lb (18,034 Nm) net torque. These values may be calculated for other crank positions in the same manner. Figure F.3 is a plot of the net torque curve.

F.4.3 Alternative Crank Rotation

The foregoing example on the use of torque factors has been based on the pumping unit operating with the cranks rotating toward the well from top dead center If the pumping unit is operating with the cranks rotating away from the well from top dead center, the calculation technique is changed only in the use of the torque factor in polished rod position data form (Figure C.2). The angle of the crank (column 1) is reversed, starting from the bottom with 15° and counting up in 15° increments to 360°.

F.4.4 Alternative Techniques

The foregoing technique is generally accepted. More precise results are dependent on the true stroke length which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer.

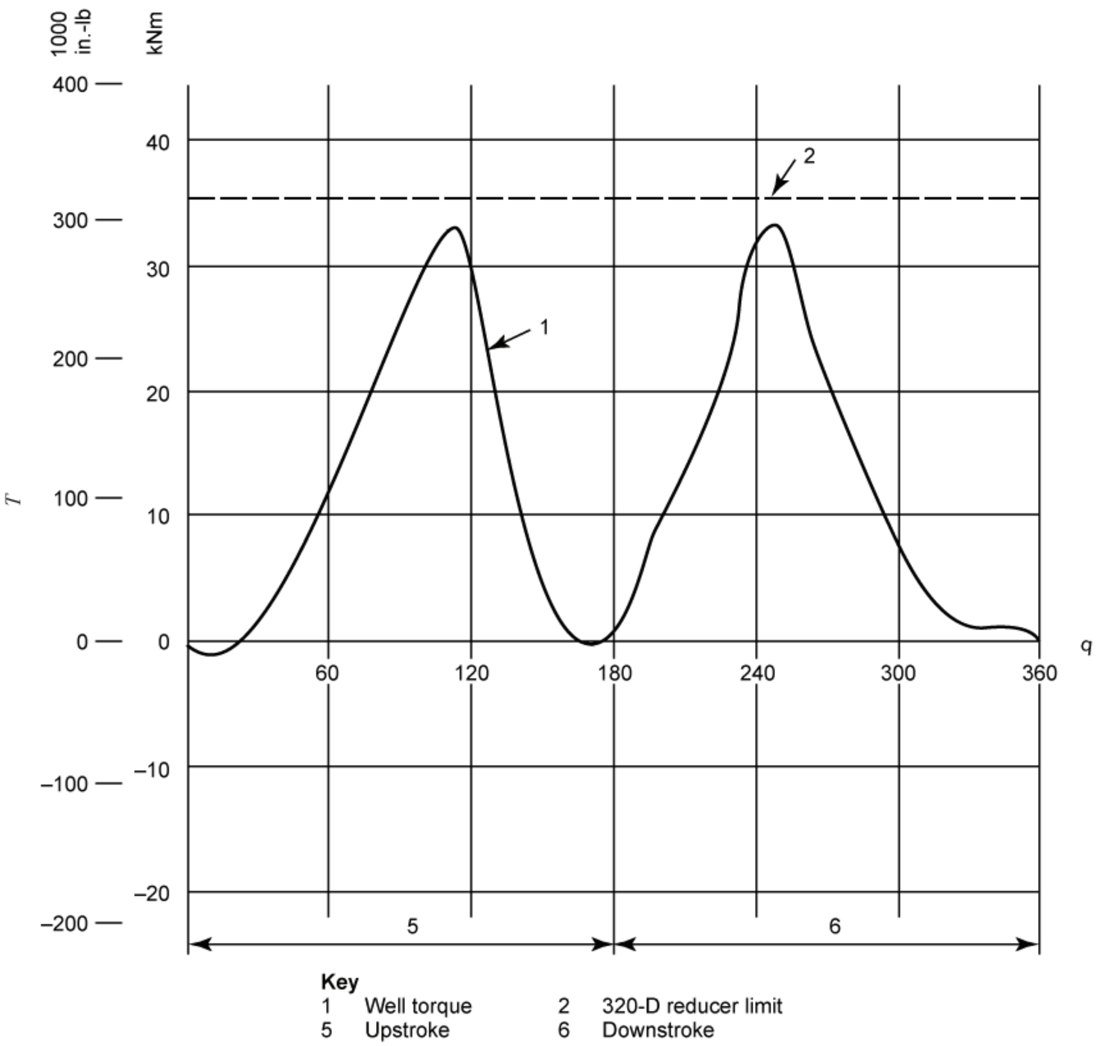


Figure F.3—Torque Curves Using Torque Factors

F.4.5 Geometrical Influences

The geometry of the dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. The dynamometer manufacturer should be contacted for the performance characteristics of the particular dynamometer being used and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

F.4.6 Interpolation

Maximum and minimum loads will most frequently fall at points other than the 15° divisions for which torque factors are provided. Interpolation between 15° divisions is permissible without significant error.

Annex G
(informative)

Torque Factor on Beam Pumping Units with Rear Mounted Geometry Class I Lever Systems with Phased Crank Counterbalance

G.1 General

The following calculation technique is generally accepted. More precise results are dependent on the true stroke length, which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer. Determinations made with a dynamometer can determine more specific performance characteristics of the individual pumping unit.

G.2 Symbols

In addition to those listed in Section 2, the following system of nomenclature and symbols is used in this annex (see Figure G.1):

B structural unbalance, equal to the force at the polished rod required to hold the beam in a horizontal position with the pitmans disconnected from the crankpins. This force is positive when acting downward and negative when acting upward, in pounds (lb) [newtons (N)]

G height from the center of the crankshaft to the bottom of the base beams, in inches (in.) [millimeters (mm)]

H height from the center of the saddle bearing to the bottom of the base, in inches (in.) [millimeters (mm)]

I horizontal distance between the centerline of the saddle bearing and the centerline of the crankshaft, in inches (in.) [millimeters (mm)]

J distance from the center of the crankpin bearing to the center of the saddle bearing, in inches (in.) [millimeters (mm)]

K distance from the center of the crankshaft to the center of the saddle bearing, 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)]

P effective length of the pitman (from the center of the equalizer bearing to the center of the crankpin bearing), in inches (in.) [millimeters (mm)]

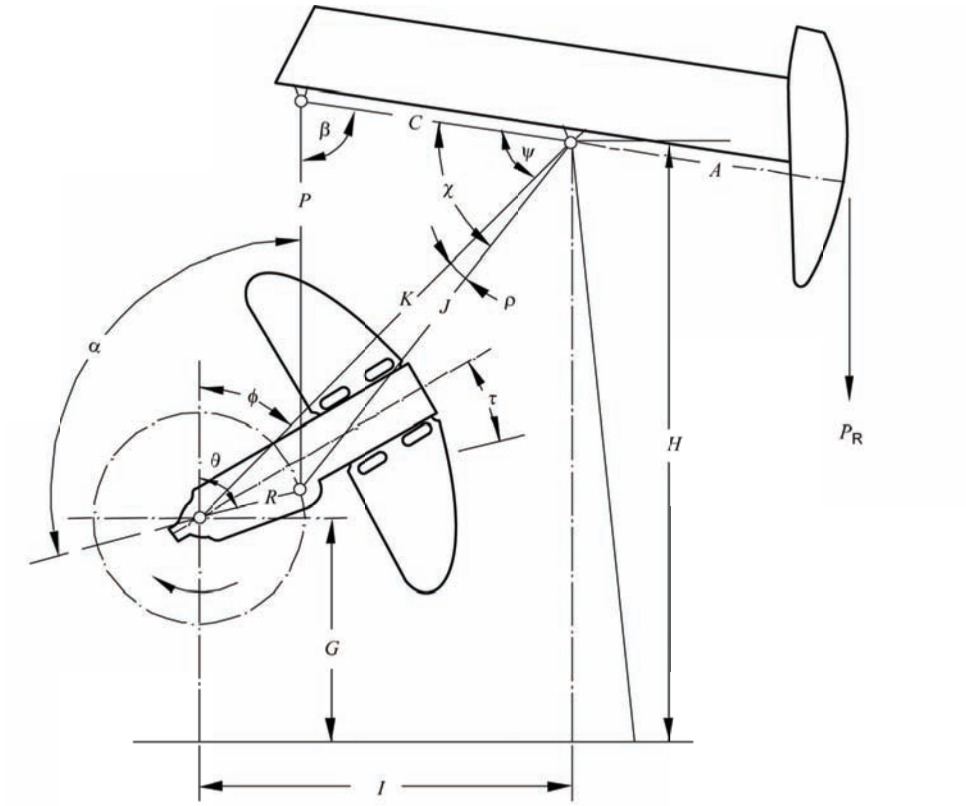
PRP polished rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle θ

T_n net torque at the crankshaft for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

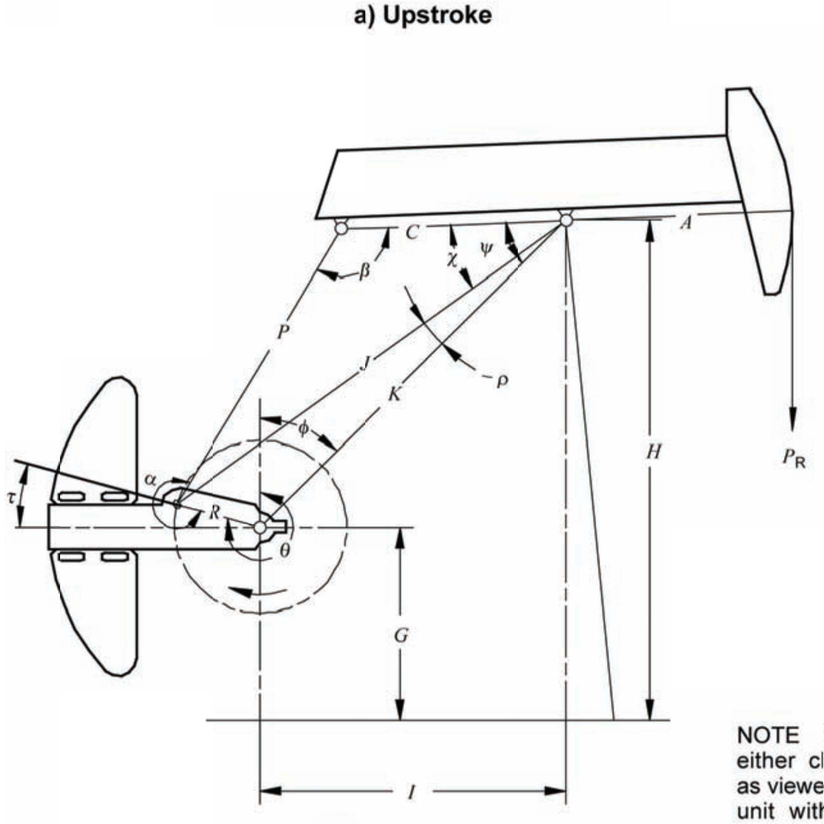
$$T_n = T_{wn} - T_r$$

T_r torque due to the rotary counterweights, cranks, and crankpins for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

$$T_r = M \sin(\theta + \tau)$$



a) Upstroke



b) Downstroke

NOTE Crank rotation is defined as either clockwise or counterclockwise as viewed from the side of the pumping unit with the wellhead to the right.

Figure G.1—Rear Mounted Geometry, Class I Lever System with Phased Crank Counterbalance

T_{wn} torque due to the net polished rod load for a given crank angle θ, in inch-pounds (in.-lb) [newton-meters (Nm)]

$T_{wn} = TF \times W_n$

TF torque factor for a given crank angle θ, in inches (in.) [meters (m)]

W_n net polished rod load, in pounds (lb) [newtons (N)]

$W_n = P_R - B$

α angle between P and R measured clockwise from R to P

β angle between C and P

χ angle between C and J

ρ angle between K and J

τ angle of crank counterweight arm offset (negative when weights are counterclockwise relative to crankpin bearings)

θ angle of crankpin rotation in a clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 12 o'clock

ϕ angle between the 12 o'clock position and K

ψ angle between C and K

$\psi = \chi - \rho$

ψ_b angle between C and K, at bottom (lowest) polished rod position

ψ_t angle between C and K, at top (highest) polished rod position

G.3 Method of Calculation

G.3.1 Torque Factors

Torque factors (as well as the polished rod position) may be determined by a scale layout of the unit geometry so that the various angles involved can be measured. They may alternatively be calculated from the dimensions of the pumping unit by mathematical treatment only. Example forms for recording calculations are provided in Figure G.4.

G.3.2 Submission Form

A form for submission of torque factor and polished rod position data is shown in Figure C.2.

G.3.3 Data Submission

Torque factors and polished rod positions shall be furnished by pumping unit manufacturers for each 15° crank position with the zero position at 12 o'clock. Other crank positions shall be determined by the angular displacement in a clockwise direction viewed with the wellhead to the right. The polished rod position for each crank position should be expressed as a fraction of the stroke above the lowermost position.

G.3.4 Calculation Method

Application of the laws of trigonometric functions give the following expressions. All angles are calculated in terms of a given crank angle θ.

In USC units:

$$\mathrm{TF} = \frac{AR}{C}\left(\frac{\sin\alpha}{\sin\beta}\right) \qquad \text{(expressed in inches)} \tag{G.1}$$

In SI units:

$$\mathrm{TF} = \frac{AR}{1000C}\left(\frac{\sin\alpha}{\sin\beta}\right) \qquad \text{(expressed in meters)} \tag{G.1}$$

Sinα is positive when the angle α is between 0° and 180° and is negative when angle α is between 180° and 360°. Sinβ is always positive because the angle β is always between 0° and 180°. A negative torque factor (TF) only indicates a change in direction of torque on the crankshaft.

$$\phi = \tan^{-1}\left(\frac{I}{H-G}\right) \tag{G.2}$$

This is a constant angle for any given pumping unit.

$$\beta = \cos^{-1}\left[\frac{C^2 + P^2 - K^2 - R^2 + 2KR\cos(\theta - \phi)}{2CP}\right] \tag{G.3}$$

Cos($\theta - \phi$) is positive when ($\theta - \phi$) is between 270° and 90° moving clockwise and is negative from 90° to 270° moving clockwise. When the angle ($\theta - \phi$) is negative, it should be subtracted from 360°, and the following equations apply.

$$\chi = \cos^{-1}\left(\frac{C^2 + J^2 - P^2}{2CJ}\right) \tag{G.4}$$

$$\rho = \sin^{-1} \pm\left[\frac{R\sin(\theta - \phi)}{J}\right] \tag{G.5}$$

The angle ρ should be taken as a positive angle when sin ρ is positive. This occurs for crank positions between ($\theta - \phi$) = 0° and ($\theta - \phi$) = 180°. The angle ρ should be taken as a negative angle when sin ρ is negative. This occurs for crank positions between ($\theta - \phi$) = 180° and ($\theta - \phi$) = 360°.

$$\psi = \chi - \rho \tag{G.6}$$

$$\sin\alpha = \sin\left[\beta + \psi - (\theta - \phi)\right] \tag{G.7}$$

$$\mathrm{PRP} = \frac{\psi_{\mathrm{b}} - \psi}{\psi_{\mathrm{b}} - \psi_{\mathrm{t}}} \tag{G.8}$$

$$\psi_{\mathrm{b}} = \cos^{-1}\left[\frac{C^2 + K^2 - (P+R)^2}{2CK}\right] \tag{G.9}$$

$$\psi_t = \cos^{-1}\left[\frac{C^2 + K^2 - (P - R)^2}{2CK}\right] \tag{G.10}$$

G.4 Application of Torque Factors

G.4.1 General

Torque factors are used primarily for determining peak crankshaft torque on operating pumping units. The procedure is to take a dynamometer card and then use torque factors, polished rod position factors, and counterbalance information to plot the net torque curve. Points for plotting the net torque curve are calculated from the following equation:

$$T_n = \mathrm{TF}(P_R - B) - M\sin(\theta - \tau) \tag{G.11}$$

G.4.2 Changes Due to Structural Unbalance

The equation for net crankshaft torque, T_n, does not include the change in structural unbalance with change in crankpin angle; neglects the inertia effects of beam, beam weights, equalizer, pitman, crank, and crank counterweights; and neglects friction in the saddle, tail, and pitman bearings. For units having 100 % crank counterbalance and where crank-speed variation is not more than 15 % of average, these factors usually can be neglected without introducing errors greater than 10 %. Some nondynamic factors that can have an effect on the determination of instantaneous net torque loadings, and which accordingly should be recognized or considered, are outlined in G.4.6, G.4.7, and G.4.8.

G.4.3 Polished Rod Effects

Torque factors may be used to obtain the effect at the polished rod of the rotary counterbalance. This is done for a given crankpin angle by dividing the counterbalance moment, $M\sin(\theta + \tau)$, by the torque factor for the crankpin angle θ. The result is the rotary counterbalance effect, in pounds (newtons), at the polished rod.

G.4.4 Rotary Counterbalance Moment

Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the cranks in the 90° position and tying off the polished rod. Then, with a polished rod dynamometer, the counterbalance effect is measured at the polished rod. Using this method, the measured polished rod load (P_R) is the combined effect of the rotary counterbalance and the structural unbalance. The maximum rotary counterbalance moment can then be determined from the following equation:

In USC units:

$$M = \frac{\mathrm{TF}(P_R - B)}{\sin(90^\circ + \tau)} \tag{G.12}$$

EXAMPLE 1 [12]
To illustrate the use of torque factors, an example dynamometer card taken on a 5954 ft (1814 m) well is shown in Figure G.2. The first step in calculating the net crankshaft torque is to divide the dynamometer card so that the load may be determined for each 15° of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length, which is proportional to the length of the stroke. The

[12] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

length of the baseline or zero line is then divided into 10 equal parts and these parts are subdivided. This may easily be done with a suitable scale along a suitable diagonal line as shown (see Note).

NOTE Using the polished rod position data, vertical lines representing each 15° of crank angle θ are projected upward to intersect the dynamometer card. Then the polished rod load may be determined for each 15° of crank angle θ.

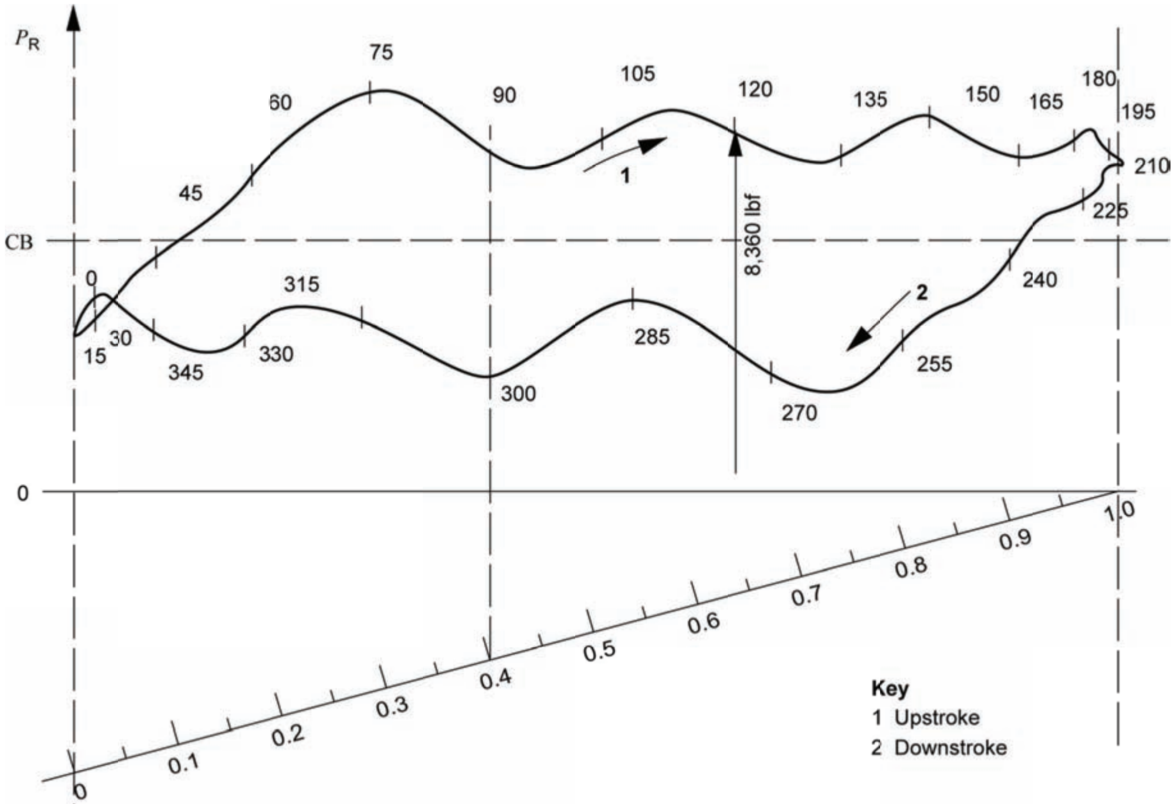


Figure G.2—Division of Dynamometer Card by Crank Angle Using Polished Rod Position Data

To further illustrate [13], a calculation is made considering the point where the crankpin angle θ equals 120°. From polished rod stroke and torque factor data for the particular 86-in. (2184-mm) stroke 114-D pumping unit used for this example, it is found that the position of the polished rod at 120° is 0.629 and that the torque factor TF is 35.446 in. (0.900 m). A vertical line is drawn from the 0.629 position on the scale up to the point of intersection with the load on the upstroke (Figure G.2). The dynamometer deflection at this point is read to be 1.672 in. (42.47 mm), which with a scale constant of 5,000 lb/in. (875 N/mm), makes the load (P_R) at that point 8360 lb (37,161 N).

In a similar manner, the polished rod load may be obtained for each 15° angle of crankpin rotation. The dynamometer card has been marked to show the load and position involved for each 15° of crank angle. The structural unbalance, P_B, for the example unit equals +231 lb (+1030 N). Therefore, the net polished rod load, W_n, at θ = 120° is:

In USC units:

$$W_n = P_R - B = 8360 - (+231) = 8129 \text{ lb}$$

[13] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

In SI units:

$$W_n = P_R - B = 37,190 - (+1,030) = 36,160 \text{ N}$$

The torque, T_{wn}, due to the net polished rod load is given by:

In USC units:

$$T_{wn} = \mathrm{TF}(W_n) = 35,446(8,129) = 288,140 \text{ in.-lb}$$

In SI units:

$$T_{wn} = \mathrm{TF}(W_n) = 0.900(36,160) = 32,544 \text{ Nm}$$

G.4.5 Torque Determination

To find the torque, T_r, due to the crank counterbalance, the maximum moment, M, has to be determined. This may be done either from manufacturers' counterbalance tables or curves, or as described in F.4.4. Because of the lack of manufacturers' counterbalance data in a majority of the cases, the polished rod measurement technique will be used more frequently in determining the maximum moment. Should the manufacturers' counterbalance data be used, it is suggested that a check be made using a polished rod measurement technique.

EXAMPLE [14]
The horizontal dotted line drawn across the dynamometer card in Figure G.2 is the counterbalance effect measured with the dynamometer at the 90° crank angle and is 7,000 lb (31,140 N). The maximum moment can then be calculated as follows, using Equation (F.12):

$$M = \frac{\mathrm{TF}(P_R - B)}{\sin(90^\circ + \tau)}$$

In USC units:

$$M = \frac{39.575[7,000 - (+231)]}{\sin[90^\circ + (-14^\circ)]} = 276,084 \text{ in.-lb}$$

In SI units:

$$M = \frac{1.005[31,137 - (+1,030)]}{\sin[90^\circ + (-14^\circ)]} = 31,184 \text{ Nm}$$

(The torque factor of 39.575 in. (1.005 m) is the value at the 90° crankpin position, and angle τ is –14° for the example unit.)

The torque, T_r, due to the counterbalance at the 120° crank position would therefore be equal to:

In USC units:

$$T_r = 276,084 \times \sin[120^\circ + (-14^\circ)] = 276,084 \times 0.961 = 265,389 \text{ in.-lb}$$

[14] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

In SI units:

$$T_r = 31{,}184 \times \sin[120^\circ + (-14^\circ)] = 29{,}975 \text{ Nm}$$

The net torque at the crankshaft for the 120° crankpin position would then be calculated from Equation (G.11) as follows: [15]

In USC units:

$$T_n = \mathrm{TF}(P_R - B) - M\sin(\theta - \tau)$$

$$T_n = T_{wn} - T_r$$

$$T_n = 288{,}140 - 265{,}389 = 22{,}751 \text{ in.-lb}$$

In SI units:

$$T_n = \mathrm{TF}(P_R - B) - M\sin(\theta - \tau)$$

$$T_n = T_{wn} - T_r$$

$$T_n = 32{,}544 - 29{,}975 = 2{,}569 \text{ Nm}$$

These values may be calculated for other crankpin angle positions in the same manner as outlined above. Shown in Figure G.3 is a plot of torque versus crankpin angle that includes the net polished rod load torque curve, the counterbalance torque curve, and the net crankshaft torque curve.

G.4.6 Alternative Techniques

The foregoing technique is generally accepted. More precise results are dependent on the true stroke length which can vary with changes in the beam position relative to the centerline of the saddle bearing. This can be due to an adjustment provided on most medium- to large-size units or due to manufacturing tolerances. Any dimensional deviation will produce some change in the angular relationships with a resultant minor change in the torque factors furnished by the manufacturer.

G.4.7 Geometrical Influences

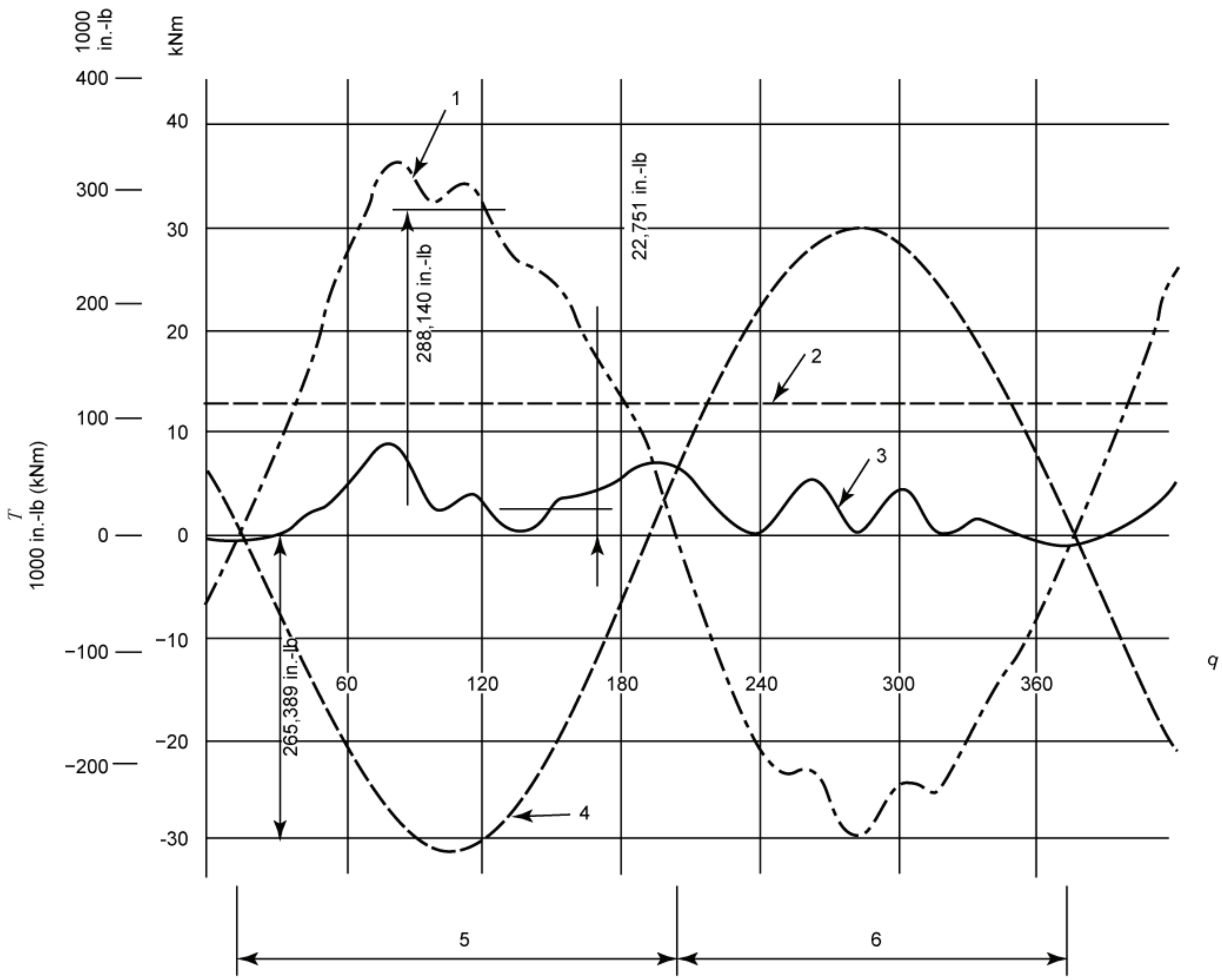
The geometry of the dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. The dynamometer manufacturer should be contacted for the performance characteristics of the particular dynamometer being used and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

G.4.8 Interpolation

Maximum and minimum loads will most frequently fall at points other than the 15° divisions for which torque factors are provided. Interpolation between 15° divisions is permissible without significant error.

---

[15] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.



Key

1	Well torque	2	114-D reducer limit	3	Net gear torque
4	Counterbalance torque	5	Upstroke	6	Downstroke

Figure G.3—Torque Curves Using Torque Factors

Net Reducer Torque Calculation Sheet
(Rear Mounted Geometry, Class I Lever System with
Phased Crank Counterbalance—Clockwise Rotation)

Well No.: ______________________ Company: ______________________

Unit Size: ______________________ Location: ______________________

θ	$\sin(\theta + \tau)$	P_R	B	$P_R - B$	TF	$TF(P_R - B)$	$M[\sin(\theta + \tau)]$	T_n
0°								
15°								
30°								
45°								
60°								
75°								
90°								
105°								
120°								
135°								
150°								
165°								
180°								
195°								
210°								
225°								
240°								
255°								
270°								
285°								
300°								
315°								
330°								
345°								
					Date Prepared			

NOTE $T_n = TF(P_R - B) - M \sin(\theta + \tau)$

where

T_n is the net reducer torque, in.-lb (Nm);

θ is the position of crank;

M is the maximum moment of counterbalance, in.-lb (Nm);

P_R is the measured polished rod load at θ, lb (N);

B is the unit structural unbalance, lb (N);

TF is the torque factor at θ, in. (m);

CB at 90° = ______________

$$M = \frac{(CB_{at\ 90^\circ} - B)(TF_{at\ 90^\circ})}{\sin(90^\circ + \tau)}$$ ______________

τ is the angle of crank counterweight arm offset (negative when weights are counterclockwise relative to crankpin bearings) = ______________

Figure G.4—Net Reducer Torque Calculation Sheet

Annex H
(informative)

Examples for Calculating Torque Ratings for Pumping Unit Reducers

H.1 Illustrative Example, Pitting Resistance [16]

This annex contains an example calculation of the allowable transmitted torque at the output shaft based on the pitting resistance for a first reduction helical gear set. For the example the pinion speed is 588 revolutions per minute (rpm), and the reducer output speed is 20 rpm.

EXAMPLE

Gear set data:

D = 16.833 in. (427.57 mm)

d = 3.167 in. (80.44 mm)

F = 3 in. (76.2 mm)

m = 4.233 mm

N_G = 101

N_P = 19

n_o = 20 rpm

n_p = 588 rpm

P_d = 6.0 in.$^{-1}$

ϕ_n = 17.4952°

ψ = 30°

Minimum pinion hardness = 340 BHN (steel)

Minimum gear hardness = 300 BHN (steel)

Determine pitting resistance torque rating as follows:

From Equation (7):

In USC units:

$$v_t = 0.262 n_p d = 0.262(3.167)(588) = 487.5 \text{ ft/min}$$

In SI units:

$$v_t = \frac{\pi n_p d}{60{,}000} = \frac{\pi(588)(80.44)}{60{,}000} = 2.476 \text{ m/s}$$

---

16 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

From Equation (6): [17]

In USC units:

$$C_5 = \frac{78}{78 + \sqrt{v_t}} = \frac{78}{78 + \sqrt{487.5}} = 0.779$$

In SI units:

$$C_5 = \frac{78}{78 + \sqrt{200 v_t}} = \frac{78}{78 + \sqrt{200 \times 2.476}} = 0.778$$

From Equation (5):

In USC units:

$$C_1 = \frac{n_p d^2 C_5}{2 n_o} = \frac{(588)(3.167)^2 (0.779)}{2(20)} = 114.9$$

C_m = 1.33 (from Figure 2)

In SI units:

$$C_1 = \frac{n_p d^2 C_5}{2000 n_o} = \frac{(588)(80.44)^2 (0.778)}{2000(20)} = 74.00$$

C_m = 1.33 (from Figure 2)

From Equation (8):

In USC units:

$$C_2 = \frac{F}{C_m} k_h = \frac{3}{1.33}(1) = 2.25$$

S_{ac} = 129,100 psi (see Figure 4)

In SI units:

$$C_2 = \frac{F}{C_m} k_h = \frac{76.2}{1.33}(1) = 57.13$$

S_{ac}=890.1 MPa (see Figure 3)

[17] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

From Equation (11):

In USC units:

$$C_3 = 0.225\left(\frac{m_g}{m_g+1}\right)\left(\frac{S_{ac}}{C_p}\right)^2 = 0.225\left(\frac{5.316}{5.316+1}\right)\left(\frac{129{,}100}{2{,}300}\right)^2 = 597$$

In SI units:

$$C_3 = 0.225\left(\frac{m_g}{m_g+1}\right)\left(\frac{S_{ac}}{C_p}\right)^2 = 0.225\left(\frac{5.316}{5.316+1}\right)\left(\frac{890.1}{191}\right)^2 = 4.11$$

From Equation (4):

In USC units:

$$T_{ac} = C_1C_2C_3 = (114.9)(2.25)(597) = 154{,}300 \text{ in.-lb}$$

In SI units:

$$T_{ac} = C_1C_2C_3 = (74.00)(57.13)(4.11) = 17{,}375 \text{ Nm}$$

NOTE The pitting resistance rating of this gear set is 154,300 in.-lb (17,375 Nm). The final rating is the lowest calculated value of pitting resistance rating and bending strength ratings as determined in Equation (4) and Equation (18) of this specification but not to exceed one of the standard pumping unit reducer sizes listed in Table 2.

H.2 Illustrative Example, Bending Strength [18]

H.2.1 General

This section contains an example calculation of the allowable transmitted torque at the output shaft based on bending strength for a first reduction helical (or double helical) gear set. For the example the pinion speed is 588 rpm, and the reducer output speed in 20 rpm.

NOTE This is the same gear set used in the pitting resistance calculation example.

H.2.2 Pinion

Determine strength numbers for pinion as follows:

EXAMPLE

From Equation (7):

In USC units:

$$v_t = 0.262n_pd = 0.262(3.167)(588) = 487.5 \text{ ft/min}$$

---

[18] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

In SI units: [19]

$$v_t = \frac{\pi n_p d}{60{,}000} = \frac{\pi(588)(80.44)}{60{,}000} = 2.476 \text{ m/s}$$

From Equation (20):

In USC units:

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{v_t}}} = \sqrt{\frac{78}{78 + \sqrt{487.5}}} = 0.883$$

In SI units:

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{200 v_t}}} = \sqrt{\frac{78}{78 + \sqrt{200 \times 2.476}}} = 0.882$$

From Equation (19):

In USC units:

$$K_1 = \frac{n_p d K_5}{2 n_o} = \frac{(588)(3.167)(0.883)}{2(20)} = 41.11$$

$$K_m = \frac{1}{0.872 - 0.0176 F} = 1.22 \text{ (see Figure 5)}$$

In SI units:

$$K_1 = \frac{n_p d K_5}{2000 n_o} = \frac{(588)(80.44)(0.882)}{2000(20)} = 1.043$$

$$K_m = \frac{1}{0.872 - \left(6.93 \times 10^{-4}\right) F} = 1.22 \text{ (see Figure 5)}$$

From Equation (22):

In USC units:

$$K_2 = \frac{F}{K_m} k_h = \frac{3}{1.22}(1) = 2.46$$

S_{at} = 33,250 psi (see Figure 4)

J_b = 0.437 calculated per AGMA 908-B89 and as recorded in Figure C.3

---

19 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

In SI units: [20]

$$K_2 = \frac{F}{K_m} k_h = \frac{76.2}{1.22}(1) = 62.46$$

S_{at} = 229.28 MPa (see Figure 4)

J_b = 0.437 calculated per AGMA 908-B89 and as recorded in Figure C.3

From Equation (25):

In USC units:

$$K_4 = \frac{J_b}{P_d} = \frac{0.437}{6.0} = 0.0728$$

In SI units:

$$K_4 = J_b m = (0.437)(4.233) = 1.850$$

From Equation (18):

In USC units:

$$T_{at} = K_1 K_2 S_{at} K_4 = (41.11)(2.46)(33{,}250)(0.0728) = 244{,}800 \text{ in.-lb } (\text{pinion})$$

In SI units:

$$T_{at} = K_1 K_2 S_{at} K_4 = (1.043)(62.46)(229.28)(1.85) = 27{,}633 \text{ Nm } (\text{pinion})$$

H.2.3 Gear

Determine bending strength torque rating for gear as follows:

From Equation (19):

In USC units:

K_1 = 41.11

In SI units:

K_1 = 1.043

[20] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

From Equation (22):

In USC units:

K_2 = 2.46

S_{at} = 30,900 psi (see Figure 4)

J_b = 0.387 calculated per AGMA 908-B89 and as recorded in Figure C.3

In SI units:

K_2 = 62.46

S_{at} = 213.06 MPa (see Figure 4)

J_b = 0.387 calculated per AGMA 908-B89 and as recorded in Figure C.3

From Equation (25):

In USC units:

$$K_4 = \frac{J_b}{P_d} = \frac{0.387}{6.0} = 0.0645$$

In SI units:

$$K_4 = J_b m = 0.387(4.233) = 1.638$$

From Equation (18):

In USC units:

$$T_{at} = K_1 K_2 S_{at} K_4 = (41.11)(2.46)(30{,}900)(0.0645) = 201{,}560 \text{ in.-lb (gear)}$$

In SI units:

$$T_{at} = K_1 K_2 S_{at} K_4 = (1.043)(62.46)(213.06)(1.638) = 22{,}739 \text{ Nm (gear)}$$

NOTE The calculated bending strength torque rating of this gear set is 201,560 in.-lb (22,739 Nm), the lower value of the bending strength ratings for the pinion and the gear. The calculated pitting resistance torque rating is 154,300 in.-lb (17,393 Nm) (see previous example). The next smaller torque rating shown in Table 2 is 114,000 in.-lb (12,880 Nm); therefore, 114,000 in.-lb (12,880 Nm); is the stated (nameplate) peak torque rating as far as this gear set is concerned.

H.3 Illustrative Example, Static Torque [21]

This section contains an example calculation of the allowable static torque rating based on bending strength for a first reduction helical (or double helical) gear set. For the example the pinion speed is 588 rpm.

[21] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

NOTE This is the same gear set used in the pitting resistance calculation example.

Gear set data: [22]

D = 16.833 in. (427.56 mm)

F = 3 in. (76.2 mm)

J = 0.387

K_{ms} = 0.0144 (3) + 1.07 = 1.113 [K_{ms} = (5.67 × 10^{-4})(75) + 1.07 = 1.113] [see Equation (30)]

K_y = 1.0 (see Table 8)

m = 4.233 mm

m_{g2} = 5.53

P_d = 6 in.$^{-1}$

S_{ay} = 112,000 psi (772 MPa) (see Figure 6)

From Equation (28):

In USC units:

$$T_{as,i} = \left(\frac{D}{2}\right)\left(\frac{J_b}{P_d}\right)\left(\frac{F}{K_{ms}}\right) S_{ay} K_y$$

$$= \left(\frac{16.833}{2}\right)\left(\frac{0.387}{6}\right)\left(\frac{3}{1.113}\right)(112{,}000)(1)$$

$$= 163{,}880 \text{ in.-lb at the high speed gear}$$

In SI units:

$$T_{as,i} = \left(\frac{D}{2}\right) J_b m \left(\frac{F}{K_{ms}}\right) S_{ay} K_y$$

$$= \frac{(427.56)(0.387)(4.233)(76.2)(772)(1)}{(2000)(1.113)}$$

$$= 18{,}510 \text{ Nm at the high speed gear}$$

The allowable static torque at the output shaft would be the value calculated above multiplied by the ratio to the output gear set.

From Note 2 following Equation (28):

In USC units:

22 These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

$$T_{\text{as},2} = T_{\text{as},1}\left(m_{\text{g}2}\right) = 163{,}800\left(5.53\right) = 906{,}260 \text{ in.-lb at output shaft}$$

In SI units: [23]

$$T_{\text{as},2} = T_{\text{as},1}\left(m_{\text{g}2}\right) = 18{,}510\left(5.53\right) = 102{,}360 \text{ Nm at output shaft}$$

The value is for the high speed gear only and shall be repeated for each gear and pinion in the reducer. The lowest value of T_{as} will be the maximum allowable imposed static torque but shall be equal to or greater than 500 % of the applicable nameplate rating recorded in Table 2. In this example the nameplate rating as far as the first reduction is concerned is 114,000 in.-lb (12,880 Nm) (see bending strength calculation above). The static torque rating shall therefore be equal to or greater than 5 × 114,000 = 570,000 in.-lb (5 × 12,880 = 64,400 Nm). The calculated static torque rating of 906,260 in.-lb (102,360 Nm) satisfies this condition and is the static torque rating as far as the first reduction set is concerned.

[23] These examples are merely for illustration purposes only (each company should develop its own approach). They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

Annex I
(informative)

System Analysis

The beam pumping system includes the prime mover (electric motor, multicylinder engine, or single cylinder engine), the beam pumping unit including gear reducer, the sucker rod string, the bottomhole pump, tubing, casing, and any other component or condition that influences the loading.

A beam pumping system analysis will indicate whether the calculated loading on the gear reducer is within the design limits for which it is offered. This analysis is the responsibility of the user/purchaser. A polished rod dynamometer can be used to determine the actual loading on the gear reducer.

Methods of computing or of measuring well loads are not within the scope of this specification; however, it is recommended to use API 11L to predict approximate polished rod loads and gear reducer torque values. The user should be cognizant of the possibility of actual loads exceeding apparent loads under one or more of the following conditions:

a) improper counterbalancing,

b) excessive fluctuation in engine power output,

c) serious critical vibrations of the reducer and engine system,

d) poor bottomhole pump operation, and

e) looseness in the beam pump structure.

Annex J
(informative)

Product Nomenclature

Many of the components of a beam pumping unit have common names which may or may not reflect their engineering purpose. See Figure J.1 for an illustration of a common pumping unit with these component names indicated.

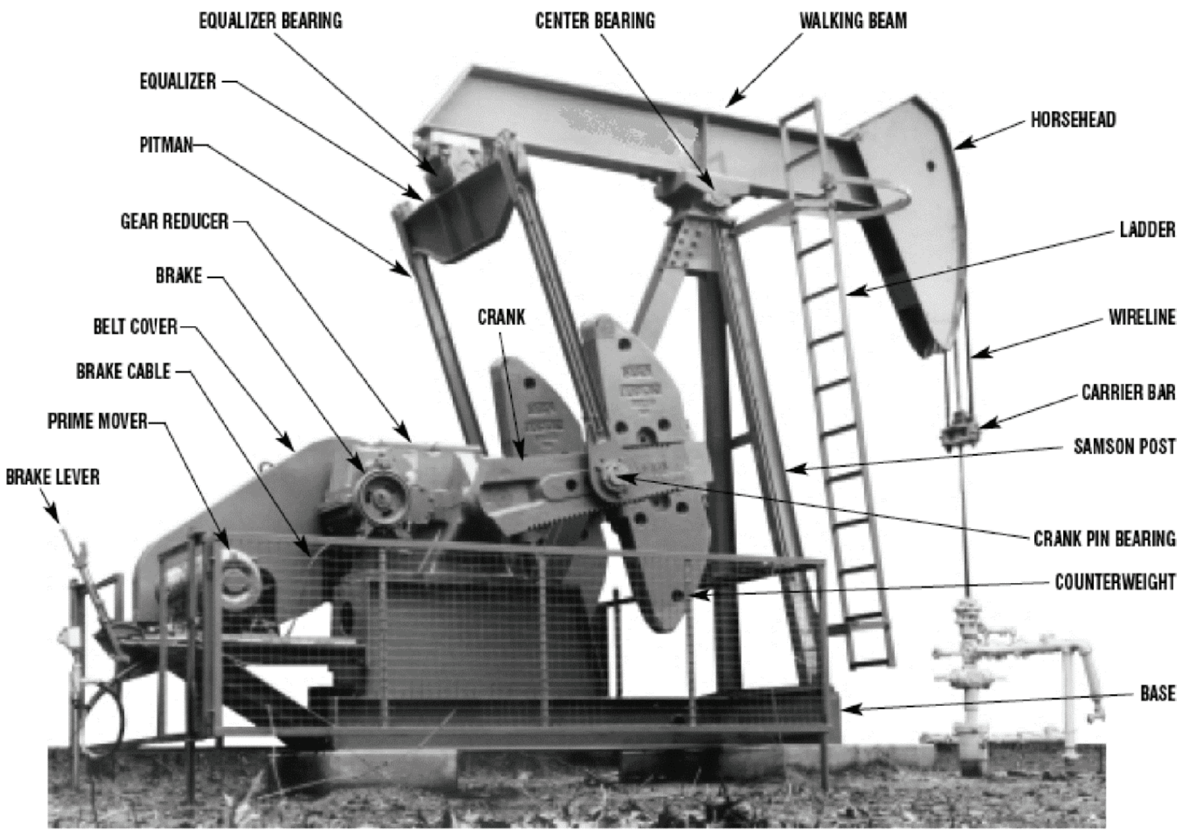


Figure J.1—Beam Pumping Unit Nomenclature

Bibliography

[1] AGMA 1010-E595 [24], *Appearance of Gear Teeth—Terminology of Wear and Failure*

[2] API Specification 11AX, *Specification for Subsurface Sucker Rod Pumps and Fittings*

[3] API Specification Q1, *Specification for Quality Management System Requirements for Manufacturing Organizations for the Petroleum and Natural Gas Industry*

[4] API Recommended Practice 11G, *Recommended Practice for Installation and Lubrication of Pumping Units*

[5] API Technical Report 11L, *Design Calculations for Sucker Rod Pumping Systems (Conventional Units)*

[6] ISO 10825 [25], *Gears—Wear and damage to gear teeth—Terminology*

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

[25] International Organization for Standardization, 1, ch. de la Voie-Creuse, Case postale 56, CH-1211 Geneva 20, Switzerland, www.iso.org.

AMERICAN PETROLEUM INSTITUTE

1220 L Street, NW
Washington, DC 20005-4070
USA

202-682-8000

Additional copies are available online at www.api.org/pubs

Phone Orders:	1-800-854-7179	(Toll-free in the U.S. and Canada)
	303-397-7956	(Local and International)
Fax Orders:	303-397-2740	

Information about API publications, programs and services is available on the web at www.api.org.

Product No. G11E019