# Specification for Pumping Units

API SPECIFICATION 11E
SEVENTEENTH EDITION, NOVEMBER 1, 1994

| Contains ISO 10431:1993 |
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| Petroleum and natural gas industries—Pumping units—Specification |

**American Petroleum Institute**
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# Specification for Pumping Units

**Exploration and Production Department**

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# FOREWORD

Note: This section is not part of ISO 10431·1993

API Specification 11E serves as the basis for ISO 10431:1993. The complete text of both the API and ISO standards is contained in this document. Some differences exist between the API version and the ISO version of this standard; for example:

- The Special Notes and Foreword are not part of ISO 10431:1993.
- Appendix G is not part of Specification 11E.
- References in the text to the API monogram program are not part of ISO 10431:1993.

*This standard shall become effective on the date printed on the cover but may be used voluntarily from the date of distribution.*

# Specification for Pumping Units

## 1 Scope

**1.1** This specification covers designs and ratings of beam type pumping unit components. It does not cover chemical properties of materials, nor the use of the equipment.

**1.2** The following nomenclature is standard:

a. Pumping unit.
b. Pumping unit structure.
c. Pumping unit gear reducer.
d. Pumping unit chain reducer.
e. Pumping unit beam counterbalance.
f. Pumping unit crank counterbalance.

**1.3** Approved forms are given in Appendix A for rating of crank counterbalances and for recording pumping unit stroke and torque factors.

**1.4** This specification covers the design and rating of the following:

a. Pumping unit structure.
b. Pumping unit gear reducer.
c. Pumping unit chain reducer.

## 2 Definitions and Symbols

### 2.1 DEFINITIONS

The terms used, wherever applicable, conform to AGMA 1012-F90, *Gear Nomenclature, Definition of Terms With Symbols.*[1]

### 2.2 SYMBOLS

The symbols used in the pitting resistance and bending strength equations are shown in Table 3.

Note: 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.

## 3 Pumping Unit Structures

This section covers the following:

a. Standardization of specific structure sizes in combination with established reducer sizes as given in section 4.
b. Walking beam design, with specific rating equation.
c. Design loads and limiting working stresses on other structural components are also included.

[1]American Gear Manufacturers Association, 1500 King Street, Alexandria, Virginia 22314

Note: Only loads imposed on the structure and/or gear reducer by the polished rod load are considered in this specification. Additional loads on the pumping unit imposed by add-on devices attached to the reducer, walking beam, or other structural components are not part of this specification. These would include such devices as compressors, stroke increasers, etc.

No dimensional requirements, other than stroke length, are established. Rating methods are given only for polished rod capacities; however, allowable working stresses of other structural components for a given polished rod capacity are defined. Other design criteria such as bearing design, braking capacity, etc., are also established.

### 3.1 STANDARD PUMPING UNIT SERIES

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 2, although the combinations of these items that make up the pumping unit designation need not be identical to those in the table. The particular combinations in the table are typical, but combinations other than those listed are acceptable under this standard (see note).

Note: It is the spirit and intent of above provision that any manufacturer having authority to use the API monogram on pumping unit structures may not represent a structure carrying the monogram or for which the letters API or the words "American Petroleum Institute" are used in its description as having a rating of any kind or size other than provided above. This applies to sales information as well as to structure markings.

### 3.2 WALKING BEAM

The following equation shall be used for rating conventional walking beams as shown in Figure 1:

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

*Where:*

$W$ = walking beam rating in pounds of polished rod load.
$f_{cb}$ = compressive stress in bending in pounds per square inch (see Table 1 for maximum allowable stress).
$S_x$ = section modulus in cubic inches. The gross section of the rolled beam may be used except that holes or welds are not permissible on the tension flange in the critical zone (see Figure 1).
$A$ = distance from centerline of saddle bearing to centerline of well in inches (see Figure 1).
$C$ = distance from centerline of saddle bearing to centerline of equalizer bearing in inches (see Figure 1).

Note: The equation given in 3.2 is based on the conventional beam construction using a single rolled section. With unconventional construction or built-up sections, due regard shall be given to change in loading, to checking stresses at all critical sections, and to the existence of stress concentrating factors.

### 3.3 LIMITING WORKING STRESS

The working stress, $f_{cb}$, for the beam rating equation given in 3.2 shall be determined from Table 1. For standard rolled beams having cross sections symmetrical with the horizontal neutral axis, the critical stress will be compression in the lower flange. The maximum value of this stress, $f_{cb}$, is the smaller of the values determined from lines 2 and 3 of Table 1.

### 3.4 UNIT ROTATION

Viewed from the side of the pumping unit with the well head to the right, crank rotation is defined as either clockwise or counterclockwise.

### 3.5 DESIGN LOADS FOR ALL STRUCTURAL MEMBERS EXCEPT WALKING BEAMS

For all pumping unit geometries, and unless otherwise specified, use the maximum load exerted on the component in question by examining the loads on the component at each 15 degree crank position on the upstroke of the pumping unit. Use polished rod load, $W$, for all upstroke crack positions.

For units with bi-directional rotation and nonsymmetrical torque factors, the direction of rotation for design calculations shall be that which results in the highest loading on structural components.

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

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

### 3.6 DESIGN STRESSES FOR ALL STRUCTURAL MEMBERS EXCEPT WALKING BEAMS, BEARING SHAFTS, AND CRANKS

**3.6.1** Design stresses for all structural components shall be a function of the yield strength of the material, $Sy$, pounds per square inch.

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

**3.6.3** Components subjected to reverse bending shall have a limiting stress of $0.2Sy$.

**3.6.4** The following equation shall be used for all components acting as columns:

$$W_2 = \frac{aSy}{4}\left[1 - \frac{Sy}{4n\,\pi^2 E}\left(\frac{l}{r}\right)^2\right]$$

*Where:*

$W_2$ = maximum applied load on column, pounds.
$a$ = area of cross section, square inches.
$Sy$ = yield strength of material, pounds per square inch.
$n$ = end restraint constant (assume = 1).
$E$ = modulus of elasticity, pounds per square inch.
$l$ = unbraced length of column, inches.
$r$ = radius of gyration of section, inches.
$\frac{l}{r}$ = shall be limited to a maximum of 90. For $\frac{l}{r}$ values of 30 or less, columns may be assumed to be acting in simple compression (see 3.6.2).

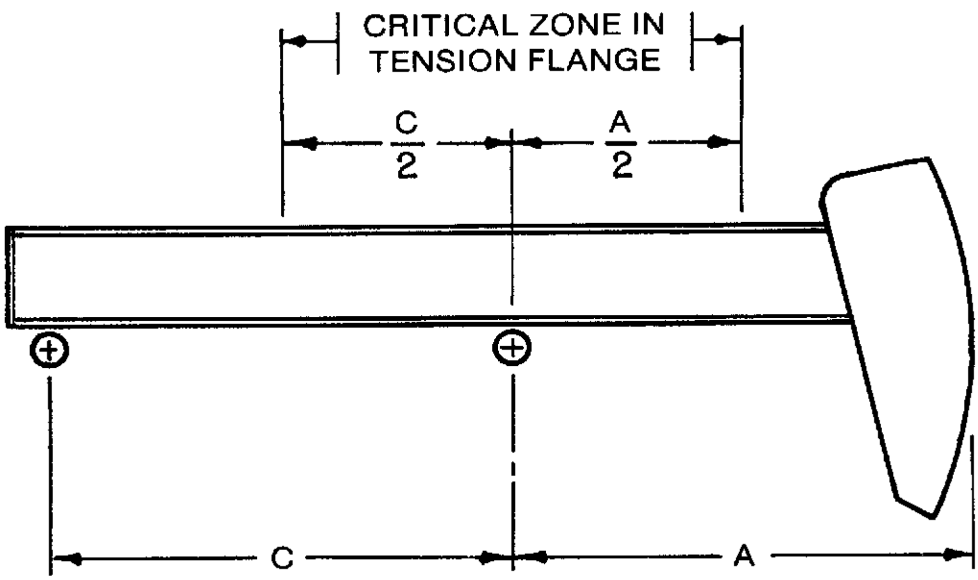



Figure 1—Walking Beam Elements

Table 1—Maximum Allowable Stresses in Pumping Unit Walking Beams (See Figure 1)

| (1) Line | (2) Stress | (3) Symbol | (4) ASTM A36 Structural Steel |
|---|---|---|---|
| 1 | Tensile stress in extreme fibers in bending, psi. | $f_{tb}$ | 11,000 |
| 2 | Compressive stress in extreme fibers in bending, psi (may not exceed values on line 3). | $f_{cb}$ | $\sqrt{\frac{EI_yGJ}{SJ}}$ [a] |
| 3 | Maximum compressive stress in bending, except as limited by equation on line 2, psi. | $f_{cb}$ | 11,000 |
| 4 | Minimum yield strength of material, psi. | | 36,000 |

[a]*Where:*
$J$ = Torsional constant, in.$^4$
$l$ = Longest laterally, unbraced length of beam, inches [longer of A or C (see Figure 1)].
$E$ = Modulus of elasticity; 29,000,000 psi.
$I_y$ = Weak axis moment of inertia, in.$^4$
$G$ = Shear modulus; 11,200,000 psi.

## 3.7 SHAFTING

All bearing shafts as well as other structural shafting shall have limiting stresses as outlined in 4.8 in the reducer section of this specification.

## 3.8 HANGER

Wireline for horseheads shall have a minimum factor of safety of five when applied to the breaking strength of the wireline.

For allowable stresses on carrier bar, end fittings, etc., see 3.6.2 and 3.6.3.

## 3.9 BRAKES

Pumping unit brakes shall have sufficient braking capacity to withstand a torque exerted by the cranks at any crank position with a maximum amount of counterbalance torque designed by the manufacturer for the particular unit involved. This braking torque is to be effective with the pumping unit at rest under normal operating conditions with the well disconnected.

Note: The pumping unit brake is not intended as a safety stop but is intended for operational stops only. When operations or maintenance are to be conducted on or around a pumping unit, the position of the crank arms and counter weights should be securely fixed in a stationary position by chaining or other acceptable means.

## 3.10 HORSEHEADS

**3.10.1** Horseheads shall be either hinged or removable to provide access for well servicing.

**3.10.2** Horseheads shall be attached to the walking beam in such a manner as to prevent falling off due to a high rod part or other sudden load changes.

**3.10.3** 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. $\pm{}^1/_2$ inch for stroke lengths to 100 inches.
b. $\pm{}^5/_8$ inch for stroke lengths 100 inches to 200 inches.
c. $\pm{}^3/_4$ inch for stroke lengths of 200 inches and longer.

## 3.11 CRANKS

All combined stresses in cranks shall be limited to a maximum value of $0.15Sy$.

## 3.12 STRUCTURAL BEARING DESIGN

Structural bearing shafts may be supported in sleeve or anti-friction bearings.

### 3.12.1 Anti-Friction Bearings

For bearings subject to oscillation or rotation, use the bearing load ratio equation:

$$R_1 = k\frac{C_1}{W_1}$$

*Where:*
$R_1$ = bearing load ratio.
$k$ = 1.0 for bearings rated at $33^1/_3$ rpm and 500 hours
or
$k$ = 3.86 for bearings rated at 500 rpm and 3000 hours.
$C_1$ = bearing manufacturer's specific dynamic rating in-lbs.
$W_1$ = maximum load on bearing in-lbs.

For bearings subject to oscillation, only use an $R_1$ value of 2.0 or greater.

For bearings subject to rotation, use an $R_1$ value of 2.25 or greater.

### 3.12.2 Sleeve Bearings

The design of sleeve bearings is beyond the scope of this specification. It shall be the responsibility of the pumping unit manufacturer to design sleeve bearings, based on available test data and field experience, that are comparable in performance to anti-friction bearings designed for the same operating loads and speeds.

### 3.13 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.

### 3.14 MARKING (SEE NOTE)

Each pumping unit structure shall be provided with a name plate substantially as shown in Figure 2. At the discretion of the manufacturer, the name plate may contain other nonconflicting and appropriate information, such as model number or lubrication instructions.

Note: Users of this specification should note that there is no longer a requirement for marking a product with the API monogram. The American Petroleum Institute continues to license use of the monogram on products covered by this specification, but it is administered by the staff of the Institute separately from the specification. The policy describing licensing and use of the monogram is contained in Appendix H. No other use of the monogram is permitted.

### 3.15 STROKE AND TORQUE FACTORS

In order that the torque on a reducer may be determined conveniently and accurately from dynamometer test data, manufacturers of pumping units shall provide, on request of the purchaser, stroke and torque factors for each 15 degree position of the crank. An approved form for the submission of these data is shown in Appendix A.

## 4 Pumping Unit Reducers

### 4.1 APPLICABILITY

This specification is applicable to enclosed speed reduc-

Table 2—Pumping Unit Size Ratings

| (1) Pumping Unit Size | (2) Reducer Rating (in.-lb) | (3) Structure Capacity (lb) | (4) Max. Stroke Length (in.) |
|---|---|---|---|
| 6.4-32-16 | 6,400 | 3,200 | 16 |
| 6.4-21-24 | 6,400 | 2,100 | 24 |
| 10-32-24 | 10,000 | 3,200 | 24 |
| 10-40-20 | 10,000 | 4,000 | 20 |
| 16-27-30 | 16,000 | 2,700 | 30 |
| 16-53-30 | 16,000 | 5,300 | 30 |
| 25-53-30 | 25,000 | 5,300 | 30 |
| 25-56-36 | 25,000 | 5,600 | 36 |
| 25-67-36 | 25,000 | 6,700 | 36 |
| 40-89-36 | 40,000 | 8,900 | 36 |
| 40-76-42 | 40,000 | 7,600 | 42 |
| 40-89-42 | 40,000 | 8,900 | 42 |
| 40-76-48 | 40,000 | 7,600 | 48 |
| 57-76-42 | 57,000 | 7,600 | 42 |
| 57-89-42 | 57,000 | 8,900 | 42 |
| 57-95-48 | 57,000 | 9,500 | 48 |
| 57-109-48 | 57,000 | 10,900 | 48 |
| 57-76-54 | 57,000 | 7,600 | 54 |
| 80-109-48 | 80,000 | 10,900 | 48 |
| 80-133-48 | 80,000 | 13,300 | 48 |
| 80-119-54 | 80,000 | 11,900 | 54 |
| 80-133-54 | 80,000 | 13,300 | 54 |
| 80-119-64 | 80,000 | 11,900 | 64 |
| 114-133-54 | 114,000 | 13,300 | 54 |
| 114-143-64 | 114,000 | 14,300 | 64 |
| 114-173-64 | 114,000 | 17,300 | 64 |
| 114-143-74 | 114,000 | 14,300 | 74 |
| 114-119-86 | 114,000 | 11,900 | 86 |
| 160-173-64 | 160,000 | 17,300 | 64 |
| 160-143-74 | 160,000 | 14,300 | 74 |
| 160-173-74 | 160,000 | 17,300 | 74 |
| 160-200-74 | 160,000 | 20,000 | 74 |
| 160-173-86 | 160,000 | 17,300 | 86 |
| 228-173-74 | 228,000 | 17,300 | 74 |
| 228-200-74 | 228,000 | 20,000 | 74 |
| 228-213-86 | 228,000 | 21,300 | 86 |
| 228-246-86 | 228,000 | 24,600 | 86 |
| 228-173-100 | 228,000 | 17,300 | 100 |
| 228-213-120 | 228,000 | 21,300 | 120 |
| 320-213-86 | 320,000 | 21,300 | 86 |
| 320-256-100 | 320,000 | 25,600 | 100 |
| 320-305-100 | 320,000 | 30,500 | 100 |
| 320-213-120 | 320,000 | 21,300 | 120 |
| 320-256-120 | 320,000 | 25,600 | 120 |
| 320-256-144 | 320,000 | 25,600 | 144 |
| 456-256-120 | 456,000 | 25,600 | 120 |
| 456-305-120 | 456,000 | 30,500 | 120 |
| 456-365-120 | 456,000 | 36,500 | 120 |
| 456-256-144 | 456,000 | 25,600 | 144 |
| 456-305-144 | 456,000 | 30,500 | 144 |
| 456-305-168 | 456,000 | 30,500 | 168 |
| 640-305-120 | 640,000 | 30,500 | 120 |
| 640-256-144 | 640,000 | 25,600 | 144 |
| 640-305-144 | 640,000 | 30,500 | 144 |
| 640-365-144 | 640,000 | 36,500 | 144 |
| 640-305-168 | 640,000 | 30,500 | 168 |
| 640-305-192 | 640,000 | 30,500 | 192 |
| 912-427-144 | 912,000 | 42,700 | 144 |
| 912-305-168 | 912,000 | 30,500 | 168 |
| 912-365-168 | 912,000 | 36,500 | 168 |
| 912-305-192 | 912,000 | 30,500 | 192 |
| 912-427-192 | 912,000 | 42,700 | 192 |
| 912-470-240 | 912,000 | 47,000 | 240 |
| 912-427-216 | 912,000 | 42,700 | 216 |
| 1280-427-168 | 1,280,000 | 42,700 | 168 |
| 1280-427-192 | 1,280,000 | 42,700 | 192 |
| 1280-427-216 | 1,280,000 | 42,700 | 216 |
| 1280-470-240 | 1,280,000 | 47,000 | 240 |
| 1280-470-300 | 1,280,000 | 47,000 | 300 |
| 1824-427-192 | 1,824,000 | 42,700 | 192 |
| 1824-427-216 | 1,824,000 | 42,700 | 216 |
| 1824-470-240 | 1,824,000 | 47,000 | 240 |
| 1824-470-300 | 1,824,000 | 47,000 | 300 |
| 2560-470-240 | 2,560,000 | 47,000 | 240 |
| 2560-470-300 | 2,560,000 | 47,000 | 300 |
| 3648-470-240 | 3,648,000 | 47,000 | 240 |
| 3648-470-300 | 3,648,000 | 47,000 | 300 |

ers wherein the involute gear tooth designs include helical and herringbone gearing. This specification is intended primarily for beam-type pumping units.

## 4.2 LIMITATIONS

The rating methods and influences identified in this specification are limited to single and multiple stage designs applied to oil field pumping units in which the pitch-line velocity of any stage does not exceed 5000 feet per minute and the speed of any shaft does not exceed 3600 revolutions per minute.

## 4.3 RESPONSIBILITY

### 4.3.1 Gear Reducer Designers

Professionals using this specification should realize that it is quite difficult to identify and offer solutions to all the influences affecting a gear reducer. For this reason, it is recommended that this specification be used by engineers with significant experience in mechanical systems.

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.

### 4.3.2 Rating Factors

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

### 4.3.3 Metallurgy

The allowable stress numbers, $s_{ac}$ 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 numbers. Reasonable levels of cleanliness and metallurgical controls are required to permit the use of the allowable stress numbers contained in this specification.

### 4.3.4 Residual Stress

Any material having a case-core relationship is likely to have residual stresses. If properly managed, these 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 may reduce the residual compressive stresses. Grinding the root fillet area may introduce tensile stresses in the root. Care must be taken to avoid changes in microstructure during the grinding process. Shot peening is often performed after grinding to assure the presence of residual compressive stresses.

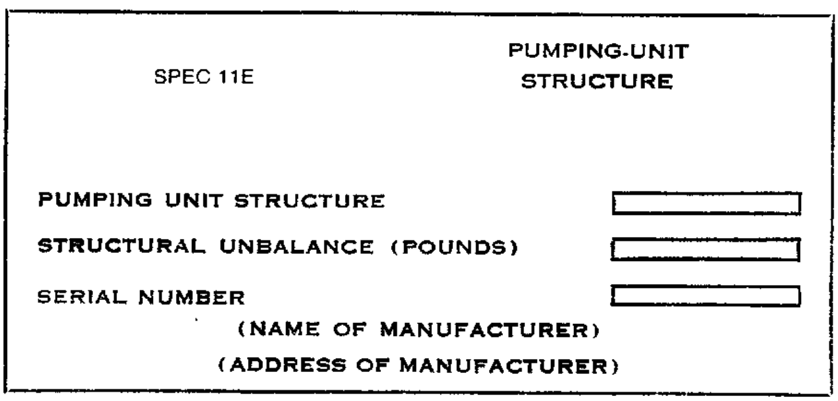
### 4.3.5 System Analysis

A pumping system analysis is the responsibility of the user. This analysis will indicate whether the calculated loading on the gear reducer is within the design limits for which it is offered. 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, API Recommended Practice 11L can be used 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 bottom hole pump operation.
e. Looseness in the pumping unit structure.

The pumping system includes the prime mover (electric motor, multi-cylinder engine, or single cylinder engine), the pumping unit structure including gear reducer, the sucker rod




Note: Structural unbalance is that force in pounds required at the polished rod to hold the beam in a horizontal position with the pitmans disconnected from the crank pins. This structural unbalance is 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 this value is negative.

Figure 2–Pumping Unit Structure Nameplate

string, the bottom hole pump, tubing, casing, and any other component or condition that influences the loading.

## 4.4 GEAR RATING TERMINOLOGY

### 4.4.1 Peak Torque Rating

The peak torque rating of the gear reducer will be the lower of the pitting resistance torque rating, bending strength torque rating, or static torque ratings as determined by the use of the applicable formulas listed.

Gear ratings as given in the equations listed are extracted from AGMA 422.03, *Practice for Helical and Herringbone Speed Reducers for Oil Field Pumping Units*, with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Alexandria, Virginia 22314.

### 4.4.2 Standard Sizes

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 4.

### 4.4.3 Rating Speeds

Gear ratings shall be based on a nominal pumping speed of 20 strokes per minute up to and including the 320 API gear reducer size (peak torque rating 320,000 in.-lbs). On gear reducers with ratings in excess of 320,000 in.-lbs, the ratings shall be based on the following nominal pumping speeds:

| Strokes Per Minute ($n_o$) | Peak Torque Rating (in.-lbs) |
|---|---|
| 16 | 456,000 |
| 16 | 640,000 |
| 15 | 912,000 |
| 14 | 1,280,000 |
| 13 | 1,842,000 |
| 11 | 2,560,000 and larger |

## 4.5 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 110.04, *Nomenclature of Gear Tooth Failure Modes.*

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.

The following equation shall be used for rating the pitting resistance of gears:

$$T_{ac} = \frac{n_p d^2 C_5}{2n_o} \times \frac{F}{C_m} \times I \left( \frac{s_{ac}}{C_p} \right)^2 \quad (1)$$

or

$$T_{ac} = C_1 \times C_2 \times C_3 \quad (2)$$

*Where:*

$T_{ac}$ = Allowable transmitted torque at output shaft, based on pitting resistance, in.-lb.

$$C_1 = \frac{n_p d^2 C_5}{2n_o} \text{ pitting velocity number.} \quad (3)$$

$n_p$ = pinion speed, rpm.

$d$ = operating pitch diameter of pinion, inches. In the equations for $C_1$, and $T_{ac}$ above, the value of $d$ may be taken as the outside diameter minus two standard addendums for enlarged pinions.

$C_5$ = velocity factor for pitting resistance.

$$C_5 = \frac{78}{78 + \sqrt{v_t}} \quad (4)$$

$v_t$ = $(d)(n_p)(.262)$, pitch line velocity, feet per minute. (Do not use enlarged pinion pitch diameter.) (5)

$n_o$ = speed of output shaft, rpm (pumping speed, strokes per minute).

$$C_2 = \frac{F}{C_m} \text{ pitting contact number.} \quad (6)$$

$F$ = net face width in inches of the narrowest of the mating gears. For herringbone or double helical gearing, the net face width is the sum of the face widths of each helix.

$C_m$ = load-distribution factor for pitting resistance from Figure 4. If deflections or other sources of misalignment are such that the values of $C_m$ from Figure 4 do not represent the actual maldistribution of load across the face, then calculate the load distribution factor using AGMA 2001-B88, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth,* and AGMA 908-B89, *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth.*

Note: When gears are hardened after cutting and the profiles and leads are not corrected or otherwise processed to insure high accuracy, the tooth distortion will affect load distribution. This makes it necessary to apply a distortion factor to the $C_2$ value. The following shall be used:
a. Multiply $C_2$ by 0.95 when one element is hardened after cutting.
b. Multiply $C_2$ by 0.90 when both elements are hardened after cutting.

The above $C_2$ factors 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) can not be attained, calculate $C_m$ per AGMA 908-B89.

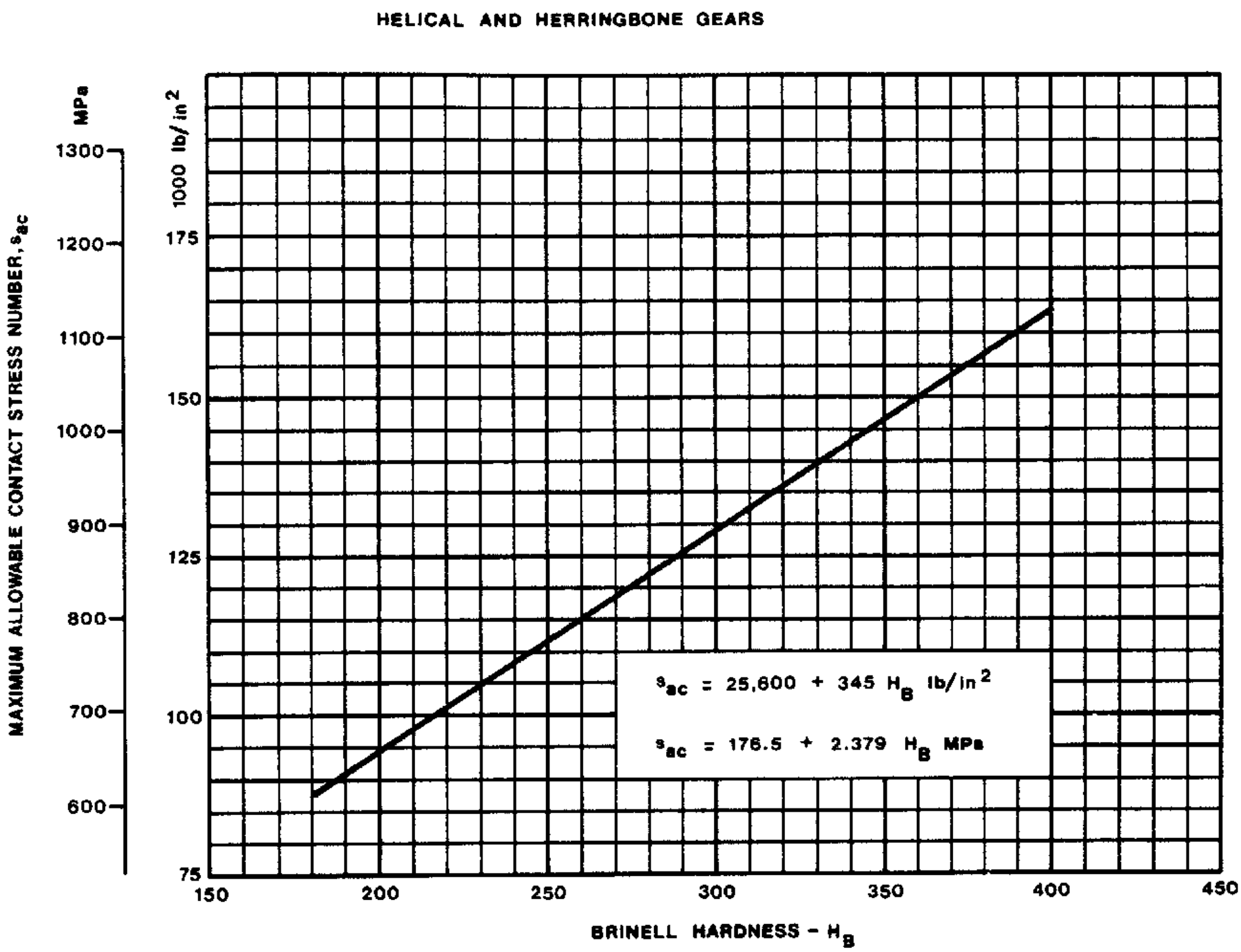
$$C_3 = 0.225 \times \frac{m_G}{m_G+1} \left( \frac{s_{ac}}{C_p} \right)^2 \text{ pitting stress number for external helical gears.} \quad (7)$$

$S_{ac}$ = allowable contact stress number from Figure 3 or Table 3.

$C_p$ = elastic coefficient,
= 2300 for mating steel elements. (See Table 6 for $C_p$ values of materials other than steel.)

## Table 3—Symbols Used in Gear Rating Equations

| Symbol | Description | Reference |
|---|---|---|
| $A$ | Tensile area of fastener, in.$^2$ | 4.8 |
| $C_m$ | Load distribution factor for pitting resistance | 4.5 |
| $C_p$ | Elastic coefficient | 4.5 |
| $C_1$ | Pitting velocity number | 4.5 |
| $C_2$ | Pitting contract number | 4.5 |
| $C_3$ | Pitting stress number | 4.5 |
| $C_5$ | Velocity factor for pitting resistance | 4.5 |
| $D$ | Operating pitch diameter of gear, in. | 4.7 |
| $D_m$ | Mean diameter of fastener, in. | 4.8 |
| $d$ | Operating pitch diameter of pinion, in. | 4.5 |
| $d_s$ | Shaft diameter, in. | 4.8 |
| $E_G$ | Modulus of elasticity for gear, psi | Table 6 |
| $E_p$ | Modulus of elasticity for pinion, psi | Table 6 |
| $F$ | Net face width of narrowest member, in. | 4.5 |
| $H_B$ | Brinell hardness | Figure 3 |
| $h_1$ | Key height in shaft or hub, in. | 4.8 |
| $I$ | Geometry factor for pitting resistance | 4.5 |
| $J$ | Geometry factor for bending strength | 4.6 |
| $K_m$ | Load distribution factor for bending strength | 4.6 |
| $K_{ms}$ | Load distribution factor, static torque | 4.7 |
| $K_y$ | Yield strength factor | 4.7 |
| $K_1$ | Strength velocity number | 4.6 |
| $K_2$ | Strength contact number | 4.6 |
| $K_3$ | Strength stress number | 4.6 |
| $K_4$ | Strength geometry number | 4.6 |
| $K_5$ | Velocity factor for bending strength | 4.6 |
| $L$ | Length of key, in. | 4.8 |
| $L_{min}$ | Minimum total length of lines of contact in contact zone | 4.5 |
| $m_G$ | Gear ratio | 4.5 |
| $N_G$ | Number of teeth in gear | 4.5 |
| $N_T$ | Threads per inch of fastener | 4.8 |
| $N_p$ | Number of teeth in pinion | 4.5 |
| $n_o$ | Speed of output shaft, rpm | 4.5 |
| $n_p$ | Pinion speed, rpm | 4.5 |
| $P_d$ | Diametral pitch, nominal, in the plane rotation (transverse), in.$^{-1}$ | 4.6 |
| $P_{nd}$ | Normal diametral pitch, in.$^{-1}$ | 4.6 |
| $P_N$ | Normal base pitch, in. | 4.5 |
| $s_s$ | Key shear stress, psi | Figure 9 |
| $s_c$ | Key compressive stress, psi | Figure 9 |
| $s_{ac}$ | Allowable contact stress number, psi | 4.5 |
| $s_{at}$ | Allowable bending stress number, psi | 4.6 |
| $s_{ay}$ | Allowable yield strength number, psi | 4.7 |
| $T_t$ | Shaft torque transmitted, in.-lbs | 4.8 |
| $T_{ac}$ | Allowable transmitted torque at output shaft based on pitting resistance, in.-lbs | 4.5 |
| $T_{at}$ | Allowable transmitted torque at output shaft based on bending strength, in.-lbs | 4.6 |
| $T_{as}$ | Allowable static torque, in.-lbs | 4.7 |
| $T_{as1}$ | Allowable static torque, 1st reduction, in.-lbs | 4.7 |
| $T_{as2}$ | Allowable static torque, 2nd reduction, in.-lbs | 4.7 |
| $T_{asn}$ | Allowable static torque, nth reduction, in.-lbs | 4.7 |
| $v_t$ | Pitchline velocity, ft/min | 4.5 |
| $w$ | Width of key, in. | 4.8 |
| $Z$ | Length of line action in transverse plane, in. | 4.5 |
| $\phi_n$ | Normal operating pressure angle, degrees | 4.5 |
| $\phi_t$ | Operating transverse pressure angle, degrees | 4.5 |
| $\psi$ | Helix angle at operating pitch diameter | 4.5 |




Note: Values are to be taken from the curve above for the minimum hardness specified for the gear. Suggested gear and pinion hardness combinations are tabulated below for convenience.

**SUGGESTED MINIMUM GEAR AND PINION BRINELL HARDNESS COMBINATIONS FOR THROUGH HARDENED AND TEMPERED STEEL GEARS**

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Gear | 180 | 210 | 225 | 245 | 255 | 270 | 285 | 300 | 335 | 350 | 375 |
| Pinion | 210 | 245 | 265 | 285 | 295 | 310 | 325 | 340 | 375 | 390 | 415 |

Figure 3—Allowable Contact Fatigue Stress for Through Hardened and Tempered Steel Gears–$s_{ac}$ (From AGMA 422.03)

Table 4—Pumping Unit Reducer Sizes and Ratings

| Size | Peak Torque Rating (in.-lbs) |
|---|---|
| 6.4 | 6,400 |
| 10 | 10,000 |
| 16 | 16,000 |
| 25 | 25,000 |
| 40 | 40,000 |
| 57 | 57,000 |
| 80 | 80,000 |
| 114 | 114,000 |
| 160 | 160,000 |
| 228 | 228,000 |
| 320 | 320,000 |
| 456 | 456,000 |
| 640 | 640,000 |
| 912 | 912,000 |
| 1280 | 1,280,000 |
| 1824 | 1,824,000 |
| 2560 | 2,560,000 |
| 3648 | 3,648,000 |

$$m_G = \text{gear ration} = \frac{N_g}{N_p} \quad (8)$$

The values of $C_3$ determined from this equation are minimums for good gear design. $C_3$ may be determined more precisely as follows:

$$C_3 = I\left(\frac{s_{ac}}{C_p}\right)^2 \quad (9)$$

$$I = \frac{\cos\phi_t \sin\phi_t}{2} \times \frac{m_G}{m_G+1} \times \frac{L_{min}}{F} \quad (10)$$

$I$ = geometry factor for pitting resistance (wear).

$\phi_t$ = operating transverse pressure angle, degrees.

$$\phi_t = \tan^{-1}\left(\frac{\tan\phi_n}{\cos\psi}\right) \quad (11)$$

$\phi_n$ = normal operating pressure angle, degrees.

$\psi$ = operating helix angle.

$L_{min}$ = minimum total length of lines of contact in contact zone. For most helical gears having a face contact ratio of 2 or more; a conservative estimate is

$$\frac{L_{min}}{F} = \frac{.95Z}{p_N} \quad (12)$$

With good gear design, the above value of $\frac{L_{min}}{F}$ is acceptable for a face contact ratio of 1.0 to 2.0 but a less conservative estimate.

$Z$ = length of line of action in the transverse plane, inches.

$p_N$ = normal base pitch, inches.

$$C_3 = \frac{\cos\phi_t \sin\phi_t}{2} \times \frac{m_G}{m_G+1} \times \frac{.95Z}{p_N}\left(\frac{s_{ac}}{C_p}\right)^2 \quad (13)$$

The method used in this specification for determining the geometry factors for pitting resistance "$I$" is simplified. A more precise and detailed analysis can be made using the method in AGMA 2001-B88 and AGMA 908-B89. The more precise method in AGMA must be used for face contact ratios less than 1.0. When "$I$" is determined in accordance with AGMA 2001-B88 and AGMA 908-B89 and if $2C/(m_G + 1)$ is not equal to outside diameter minus two standard addendums, the operating pitch diameter of the pinion in all of the preceding rating equations must be defined in AGMA 2001-B88 and AGMA 908-B89.

### 4.6 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 110.04.

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 design life of the teeth.

The following equation shall be used for rating the bending strength of helical and herringbone gears:

$$T_{at} = \frac{n_p\, d\, K_5}{2n_o} \times \frac{F}{K_m} \times s_{at} \times \frac{J}{P_d} \quad (14)$$

or

$$T_{at} = K_1 \times K_2 \times K_3 \times K_4 \quad (15)$$

*Where:*

$T_{at}$ = allowable transmitted torque at output shaft based on bending strength, in.-lb.

$K_1 = \frac{n_p\, d\, K_5}{2n_o}$ strength velocity number. (16)

$n_p$ = pinion speed, rpm.

$d$ = operating pitch-diameter of pinion, inch.

$K_5$ = velocity factor for bending strength.

$$K_5 = \sqrt{\frac{78}{78+\sqrt{v_t}}} \quad (17)$$

$n_o$ = speed of output shaft, rpm (pumping speed, strokes per minute).

$v_t$ = $(d)(n_p)(.262)$ pitch-line velocity, feet per min. (18)

## Table 5—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) |
|---|---|---|---|---|---|
| Steel | | | Flame or induction hardened[a] | 50 HRC<br>54 HRC | 170,000<br>175,000 |
| | | | Carburized and case hardened[b] | 55 HRC<br>60 HRC | 180,000<br>200,000 |
| | | AISI 4140 | Nitrided[c] | 48 HRC | 155,000 |
| | | AISI 4340 | Nitrided | 46 HRC | 155,000 |
| Cast iron | 20 | | As cast | — | 57,000 |
| | 30 | | As cast | 175 BHN | 70.000 |
| | 40 | | As cast | 200 BHN | 80,000 |
| 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 3) |
| | A-7-c | 80-55-06 | Quenched & tempered | 180 BHN | |
| | A-7-d | 100-70-03 | Quenched & tempered | 230 BHN | |
| | A-7-e | 120-90-02 | Quenched & tempered | 270 BHN | |
| | — | 120-90-02 mod. | Quenched & tempered | 300 BHN | |
| Malleable iron (pearlitic) | A-8-c | 45007 | — | 165 BHN | 68,000 |
| | A-8-e | 50005 | — | 180 BHN | 74,000 |
| | A-8-f | 53007 | — | 195 BHN | 79,000 |
| | A-8-i | 80002 | — | 240 BHN | 89,000 |

[a]For minimum flame or induction hardened case depths and hardening pattern, see Figure 9.
[b]For minimum carburized case depth per Figure 7.
[c]For minimum nitrided case depth per Figure 8.
[d]The higher allowable stress for nodular iron is determined by metallurgical controls.

## Table 6—Elastic Coefficient—$C_p$

| Pinion Materials and Modulus of Elasticity $E_p$ | | Steel $30 \times 10^6$ | Gear Materials & Modulus of Elasticity $E_g$, psi: Malleable Iron $25 \times 10^6$ | Nodular Iron $24 \times 10^6$ | Cast Iron $22 \times 10^6$ |
|---|---|---|---|---|---|
| Steel | $30 \times 10^6$ | 2300 | 2180 | 2160 | 2100 |
| Mall. Iron | $25 \times 10^6$ | 2180 | 2090 | 2070 | 2020 |
| Nod. Iron | $24 \times 10^6$ | 2160 | 2070 | 2050 | 2000 |
| Cast Iron | $22 \times 10^6$ | 2100 | 2020 | 2000 | 1960 |

Note: Poisson's ratio = 0.30.

Table 7—Allowable Bending Fatigue Stress Number, $s_{at}$
(For Other Than Through Hardened and Tempered Steel Gears)

| Material | AGMA Class | Commercial Designation | Heat Treatment | Minimum Hardness at Surface[a] | $s_{at}$ (psi) |
|---|---|---|---|---|---|
| Steel | | | Flame or induction | 50–54 RC<br>Hardened[b] | 38,300 |
| | | | Carburized and case hardened[c] | 55 HRC<br>60 HRC | 47,000<br>47,000 |
| | | AISI 4140 | Nitrided[d] | 48 RC | 29,000 |
| | | AISI 4340 | Nitrided | 46 RC | 31,000 |
| Cast iron | 20 | | As cast | — | 4,200 |
| | 30 | | As cast | 175 BHN | 7,200 |
| | 40 | | As cast | 200 BHN | 11,000 |
| 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 & tempered | 180 BHN | |
| | A-7-d | 100-70-03 | Quenched & tempered | 230 BHN | |
| | A-7-e | 120-90-02 | Quenched & tempered | 270 BHN | |
| | — | 120-90-02 Mod. | Quenched & tempered | 300 BHN | |
| Malleable iron (pearlitic) | A-8-c | 45007 | — | 165 BHN | 8,500 |
| | A-8-e | 50005 | — | 180 BHN | 11,000 |
| | A-8-f | 53007 | — | 195 BHN | 13,600 |
| | A-8-i | 80002 | — | 240 BHN | 17,900 |

[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 Data Sheet (Figure 14).
[b] For minimum flame or induction hardened case depths and hardening pattern, see Figure 9 and Figure 10. Pattern 3.8A is limited to approximately 5DP 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 3.8B. Tooth distortion and lack of ductility may necessitate a reduction of allowable stress numbers.
[c] For minimum carburized case depths per Figure 7.
[d] For minimum nitrided case depths per Figure 8.
[e] The higher allowable stress for nodular iron is determined by metallurgical controls.

$$K_2 = \frac{F}{K_m} \text{ strength contact numbers.} \tag{19}$$

Note: When gears are hardened after cutting and the profiles and leads are not corrected or otherwise processed to insure high accuracy, the tooth distortion will affect load distribution. This makes it necessary to apply a distortion factor to the $K_2$ value. The following shall be used:
a. Multiply $K_2$ by 0.95 if one element is hardened after cutting.
b. Multiply $K_2$ by 0.90 if both elements are hardened after cutting.

The above $K_2$ factor 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 the above $K_2$ values) cannot be attained, calculate $K_m$ per AGMA 2001-B88 and AGMA 908-B89.

$F$ = face width in inches 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.

$K_m$ = load distribution factor from Figure 6. If deflection or other sources of misalignment are such that the values of $K_m$ from Figure 6 do not represent the actual maldistribution of load across the face, then calculate the load distribution factor using AGMA 2001-B88 and AGMA 908-B89.

$$K_3 = s_{at} \text{ strength stress number.} \tag{20}$$

$s_{at}$ = allowable bending stress number, pounds per square inch, from Figure 5 or Table 7.

$$K_4 = \frac{J}{P_d} \text{ strength geometry number.} \tag{21}$$

$J$ = geometry factor for bending strength per AGMA 908-B89. For reference, see Appendix A in AGMA 422.03.

$P_d$ = diametral pitch in plane of rotation (transverse).

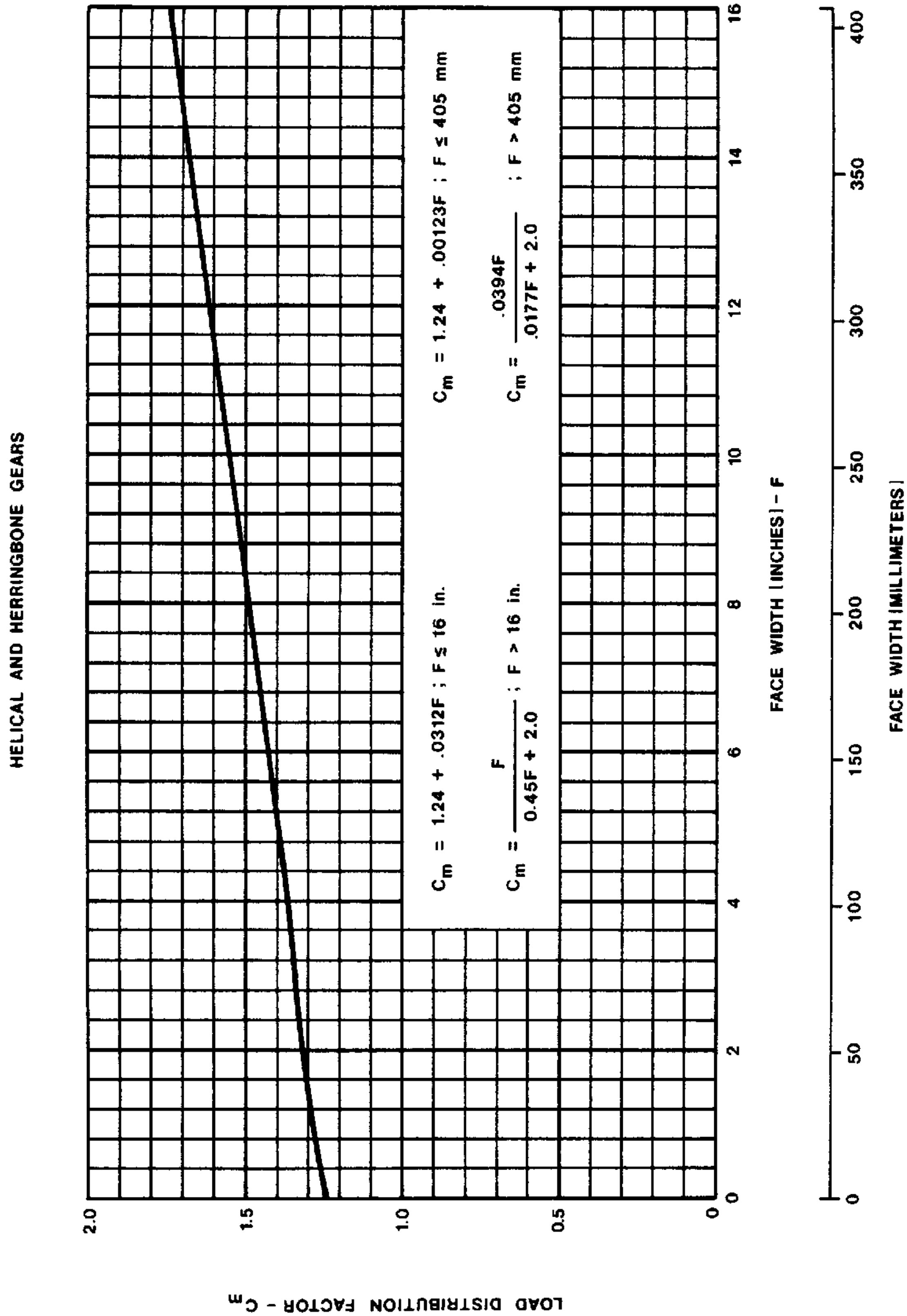
$$P_d = (P_{nd})(\cos \psi). \tag{22}$$

$P_{nd}$ = normal diametral pitch, nominal, in.$^{-1}$

Note: The bending strength rating must be calculated for both pinion and gear. The lower value is the bending strength rating of the gear set.

## 4.7 STATIC TORQUE RATING

The static torque loads on the gear teeth can be 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 to be applied is not within the scope of this specification.




Note: From AGMA 422.03.

Figure 4—Helical Gear Load Distribution Factor—$C_m$

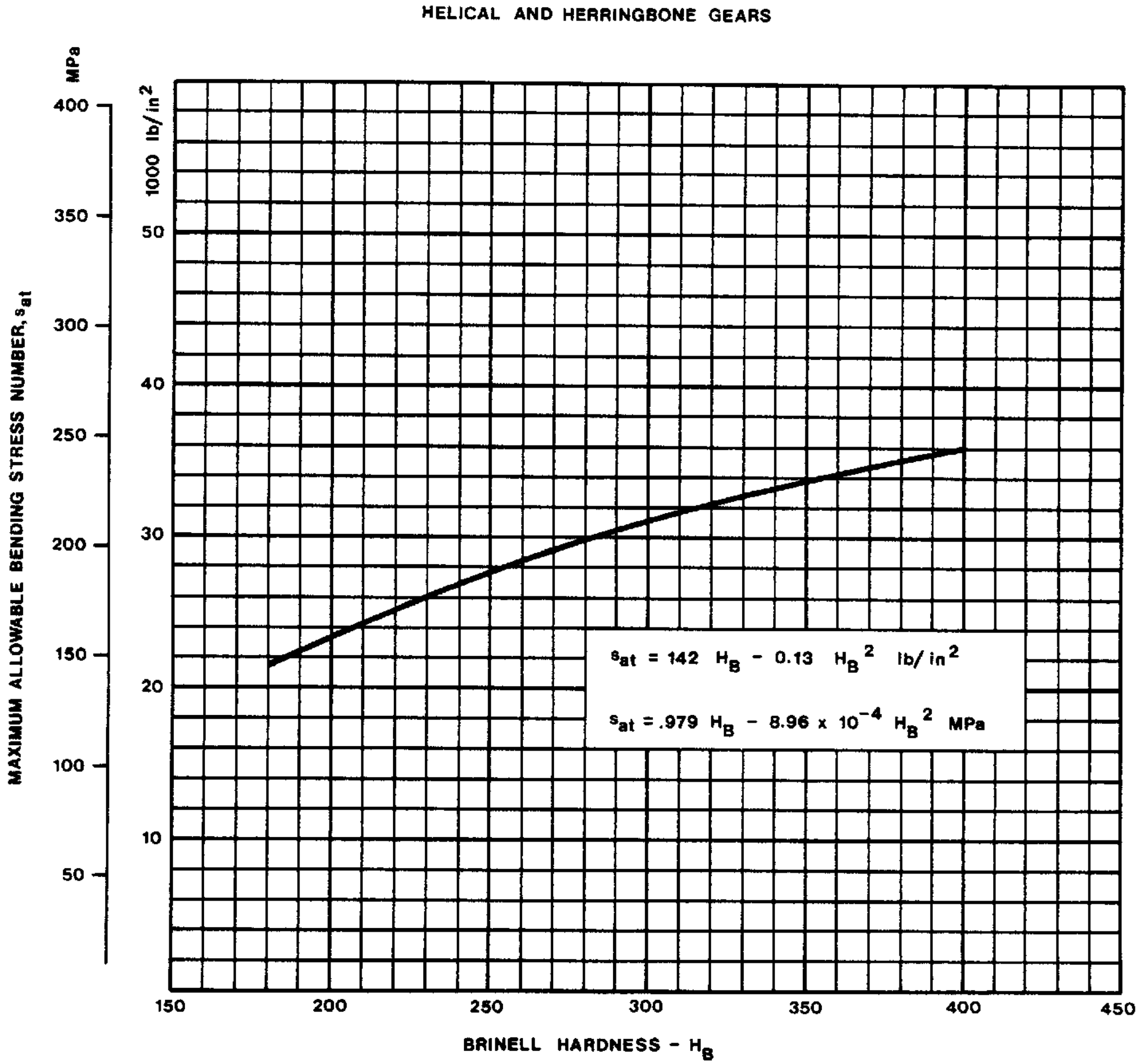



Figure 5—Allowable Bending Fatigue Stress for Through Hardened and Tempered Steel Gears—$s_{at}$
(From AGMA 422.03)

The static torque rating of the gear reducer to resist these loads must be equal to or greater than 500 percent of the reducer name plate rating. Certain pumping unit geometries may require a higher static torque rating. The system analysis required by 4.3 will be used to determine when the higher static torque rating is required.

The following equation shall be used to determine static torque rating of helical and herringbone gears:

$$T_{as} = \frac{D}{2} \times \frac{J}{P_d} \times \frac{F}{K_{ms}} \times s_{ay} \times K_y \qquad (23)$$

*Where:*

$D$ = operating pitch diameter of gear, inches.

$T_{as}$ = allowable static torque at the gear or pinion being checked; $T_{as1}$ = 1st reduction, $T_{as2}$ = 2nd reduction, $T_{asn}$ = nth reduction.

(Torque on output shaft, $T_{as2} = T_{as1} \times m_{G2}$, etc.) (24)

$s_{ay}$ = allowable yield strength number of the gear or pinion material. See Figure 11 for steel and nodular iron; for case hardened (flame, induction, nitrided, carburized) material, use core hardness from Figure 14 to determine yield strength number.

$K_y$ = yield strength factor (see Table 8).

$K_{ms}$ = $0.0144F + 1.07$ for $F \leq 16$. (25)

$K_{ms}$ = 1.3 for $F > 16$ inches.

$K_{ms}$ = load distribution factor, static torque.

Allowable static torque rating determined using this formula will be conservative since the geometry factor $J$ includes a stress concentration factor for fatigue. It should be pointed out 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 $K_y$ must be selected. The user of this specification should satisfy himself that the yield values selected are appropriate for the materials used.

## 4.8 COMPONENTS

### 4.8.1 Component Design

Gear reducers for oil field pumping units must 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.

Table 8—Yield Strength Factor, $K_y$

| Material | $K_y$ |
|---|---|
| Steel (through hardened) | 1.0 |
| Nodular iron | 1.0 |
| Steel (flame or induction hardened) | 0.85 |
| Steel (case carburized) | 1.20 |
| Steel (nitrided) | 0.85 |
| Cast iron | 0.75 |
| Mallable iron | 1.0 |

### 4.8.2 Housing

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

### 4.8.3 Bearings

Shafts may be supported in sleeve or anti-friction bearings.

### 4.8.4 Sleeve Bearings

Sleeve bearings shall be designed for bearing pressures not in excess of 750 pounds per square inch of projected area, based on actual loading (internal and external), at the rated peak torque.

### 4.8.5 Anti-Friction Bearings

Anti-friction 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.

### 4.8.6 Shaft Stresses

For steel shafts, the maximum stress due to torsion and the maximum stress due to bending shall not exceed the values shown in Figure 12 for the torque rating of the unit. These allowable stress limitations provide for effective stress concentrations arising from keyways, shoulders, and grooves, etc., not exceeding a value of 3.0. Effective stress concentration (considering notch sensitivity) exceeding a value of 3.0, press fits, or unusual deflections require detailed analysis.

### 4.8.7 Shaft Deflections

Shaft deflections causing tooth misalignment must be analyzed regardless of stress levels to insure satisfactory tooth contact as required to achieve the $C_m$ and $K_m$ values used to rate the gearing.

### 4.8.8 Key Stresses

The shear and compressive stress in a key is calculated as follows:

$$s_s = \frac{2T_t}{(d_s)(w)(L)} \qquad (26)$$

$$s_c = \frac{2T_t}{(d_s)(h_1)(L)} \qquad (27)$$

*Where:*

$s_s$ = shear stress of key, pounds per square inch (see Table 9).

$s_c$ = compressive stress of key, pounds per square inch (see Table 9).

$T_t$ = transmitted shaft torque, in.-lbs.

$d_s$ = shaft diameter, inches (for tapered shaft use mean diameter).

$w$ = width of key, inches.
$L$ = length of key, inches.
$h_1$ = height of key in the shaft or hub that bears against the keyway, inches. (For designs where unequal portions of the keyway are in the hub or shaft, $h_1$ must be the minimum portion.)

### 4.8.9 Allowable Stresses

Maximum allowable key stresses based on peak torque rating are shown in Table 9. 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.

### 4.8.10 Overloads

The shaft to hub interface must be capable of withstanding the overloads associated with oil field pumping units.

### 4.8.11 Fastener Stresses

Fastener stresses are to 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 capscrews) shall not exceed the values given in Table 10. The tensile area ($A$) is calculated as follows:

$$A = 0.785\left(D_m - \frac{0.97}{N_T}\right)^2 \qquad (28)$$

*Where:*

$A$ = tensile area of fastener, square inches.
$D_m$ = major diameter of fastener, inches.
$N_T$ = threads per inch of fastener.

### 4.8.12 Tensile Preload

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

### 4.8.13 Special Seals and Breathers

It is recognized that oil field pumping units operate outdoors under adverse atmospheric conditions and must be equipped with seals and breathers designed for these conditions.

Table 9—Allowable Key Stresses[a]

| Key Material | Hardness (BHN) | Allowable Stress (psi) Shear | Allowable Stress (psi) Comp. |
|---|---|---|---|
| AISI 1018 | None specified | 10,000 | 20,000 |
| AISI 1045 | 225-265 | 15,000 | 30,000 |
| | 265-305 | 20,000 | 40,000 |
| AISI 4140 | 310-360 | 30,000 | 60,000 |

[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 must be performed.

## 4.9 LUBRICATION

See API Recommended Practice 11G.

## 4.10 DATA SHEET

The manufacturer shall retain in his files, and make available to an API surveyor upon request, a completed Manufacturer's Gear Reducer Data Sheet as shown in Figure 14 for each gear reducer size manufactured.

## 4.11 MARKING

Note: Users of this specification should note that there is no longer a requirement for marking a product with the API monogram. The American Petroleum Institute continues to license use of the monogram on products covered by this specification, but it is administered by the staff of the Institute separately from the specification. The policy describing licensing and use of the monogram is contained in Appendix H. No other use of the monogram is permitted.

Each pumping unit reducer shall be provided with a nameplate substantially as shown in Figure 13. The size (peak torque rating in 1,000 in.-lbs.) shown on the nameplate shall be one of those listed in Table 4. No other rating marking shall be applied to the reducer. The nameplate may, at the option of the manufacturer, contain information such as model number, lubrication instructions, etc., provided such marking does not conflict with the API rating marking.

Note: It is the spirit and intent of the above provision that any manufacturer having authority to use the API monogram on pumping unit reducers may not represent a reducer carrying the monogram for which the letters API or the words "American Petroleum Institute" are used in its description as having a rating of any kind or size other than provided above. This applies to sales information as well as to reducer markings.

## 4.12 CHAIN REDUCERS

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

**4.12.2** Single, or multiple strand roller chain, conforming to ANSI[2] B29.1 heavy series, shall be used. Link plates may be thicker than specified. Center link plates of multiple strand chains shall be press fitted on the pins.

**4.12.3** Sprockets shall have ANSI tooth form.

**4.12.4** The small sprocket shall have not less than eleven teeth.

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

**4.12.6** 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).

[2]American National Standards Institute, 11 West 42nd Street, New York, New York 10036.

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

**4.12.8** 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.

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

**4.12.10** 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.

**4.12.11** The following equation shall be used for rating of chain:

$$T = \frac{S \times R}{12}$$

*Where:*

$T$ = peak torque rating in.-lbs.

$S$ = ANSI ultimate tensile strength of chain in pounds.

$R$ = pitch radius of large sprocket in inches.

## 5 Inspection and Rejection

**5.1** The inspector representing the purchaser shall have free entry at all times while work on the contract of the purchaser is being performed to all parts of the manufacturer's works that concern the manufacturer of the material specified hereinbefore. The manufacturer shall afford the inspector, free of charge, all reasonable facilities to satisfy him 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. The manufacturer shall furnish the inspector with gauges or other necessary measuring instruments, the accuracy of which shall be proved to the satisfaction of the inspector.

**5.2** Material manufactured and rated under this specification that proves to be defective subsequent to acceptance may be rejected, and the manufacturer shall be notified.

**5.3** No rejections under this or any other specification are to be stamped with the API monogram or sold as API material.

**5.4** Compliance. The manufacturer is responsible for complying with all of the provisions of this specification. The purchaser may make any investigation necessary to satisfy himself of compliance by the manufacturer and may reject any material that does not comply with this specification.

Table 10—Maximum Allowable Tensile Stress, Fasteners

| SAE and/or ASTM Designation | Threaded Fastener Diameter (Inches) | Hardness (BHN) | Ultimate Yield Strength (psi min.) | Allowable Applied Tensile Strength (psi min.) | Tensile Stress (psi max.) |
|---|---|---|---|---|---|
| SAE 2 | Over 1/4 to 3/4 incl. | 149-241 | 55,000 | 74,000 | 11,000 |
| | Over 3/4 to 1 1/2 incl. | 121-241 | 33,000 | 60,000 | 11,000 |
| SAE 5 | Over 1/4 to 1 incl. | 241-302 | 85,000 | 120,000 | 20,000 |
| (ASTM A-449) | Over 1 to 1 1/2 incl. | 223-285 | 74,000 | 105,000 | 18,000 |
| ASTM A-449 | Over 1 1/2 to 3 incl. | 183-235 | 55,000 | 90,000 | 13,000 |
| ASTM A-354 | Over 1/4 to 2 1/2 incl. | 217-285 | 80,000 | 105,000 | 17,000 |
| Grade BB | Over 2 1/2 to 4 incl. | 217-285 | 75,000 | 100,000 | 17,000 |
| ASTM A-354 | Over 1/4 to 2 1/2 incl. | 255-321 | 109,000 | 125,000 | 22,000 |
| Grade BC | Over 2 1/2 to 40 incl. | 255-321 | 99,000 | 115,000 | 22,000 |
| SAE 7 | Over 1/4 to 1 1/2 incl. | 277-321 | 105,000 | 133,000 | 24,600 |
| SAE 8 (ASTM A-354 Grade BD) | Over 1/4 to 1 1/2 incl. | 302-352 | 120,000 | 150,000 | 27,700 |

Note: The basis for the values in Table 10 is to prevent joint opening at a peak-rated load.

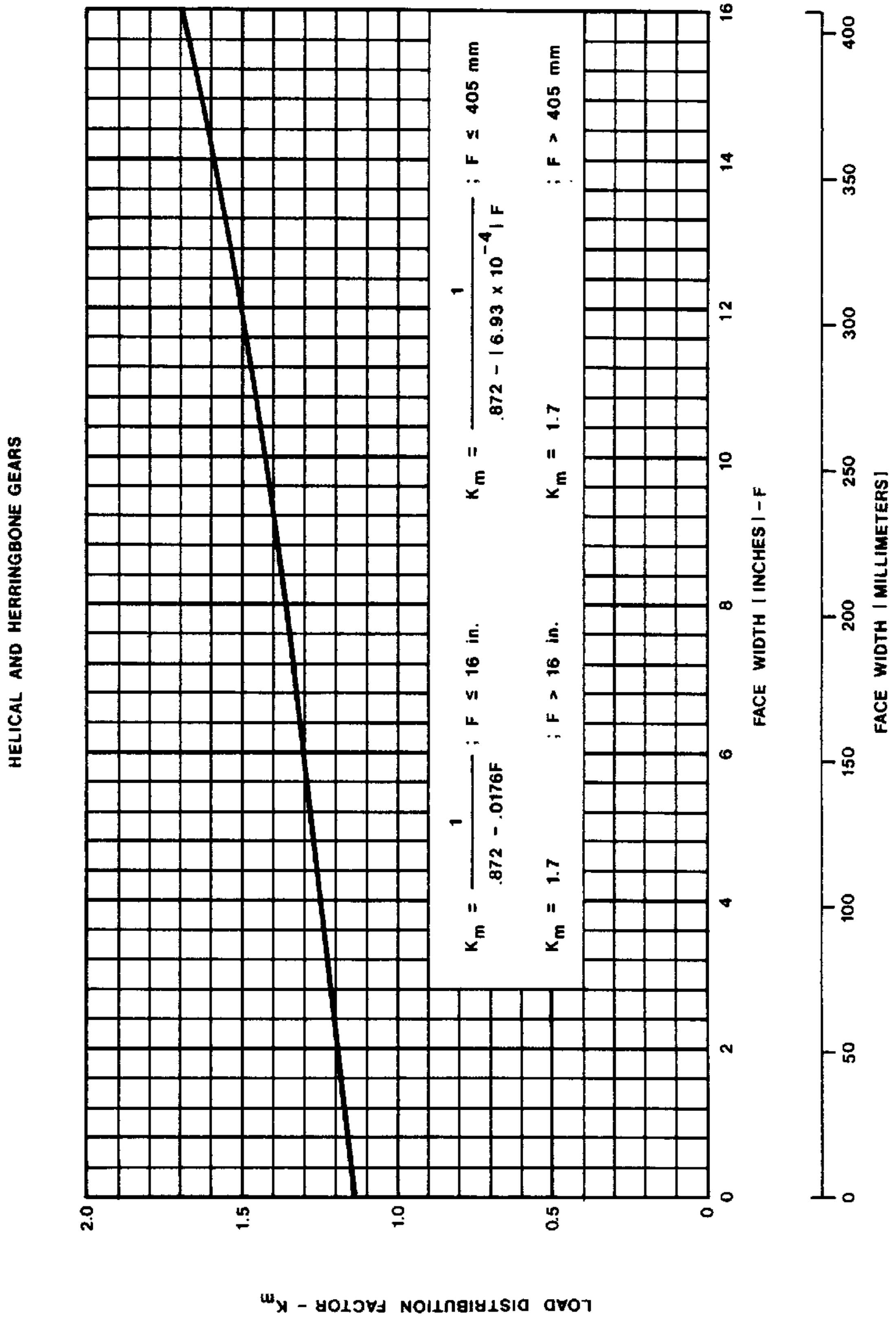



Figure 6—Helical Gear Load Distribution Factor—$K_m$
(From AGMA 422.03)

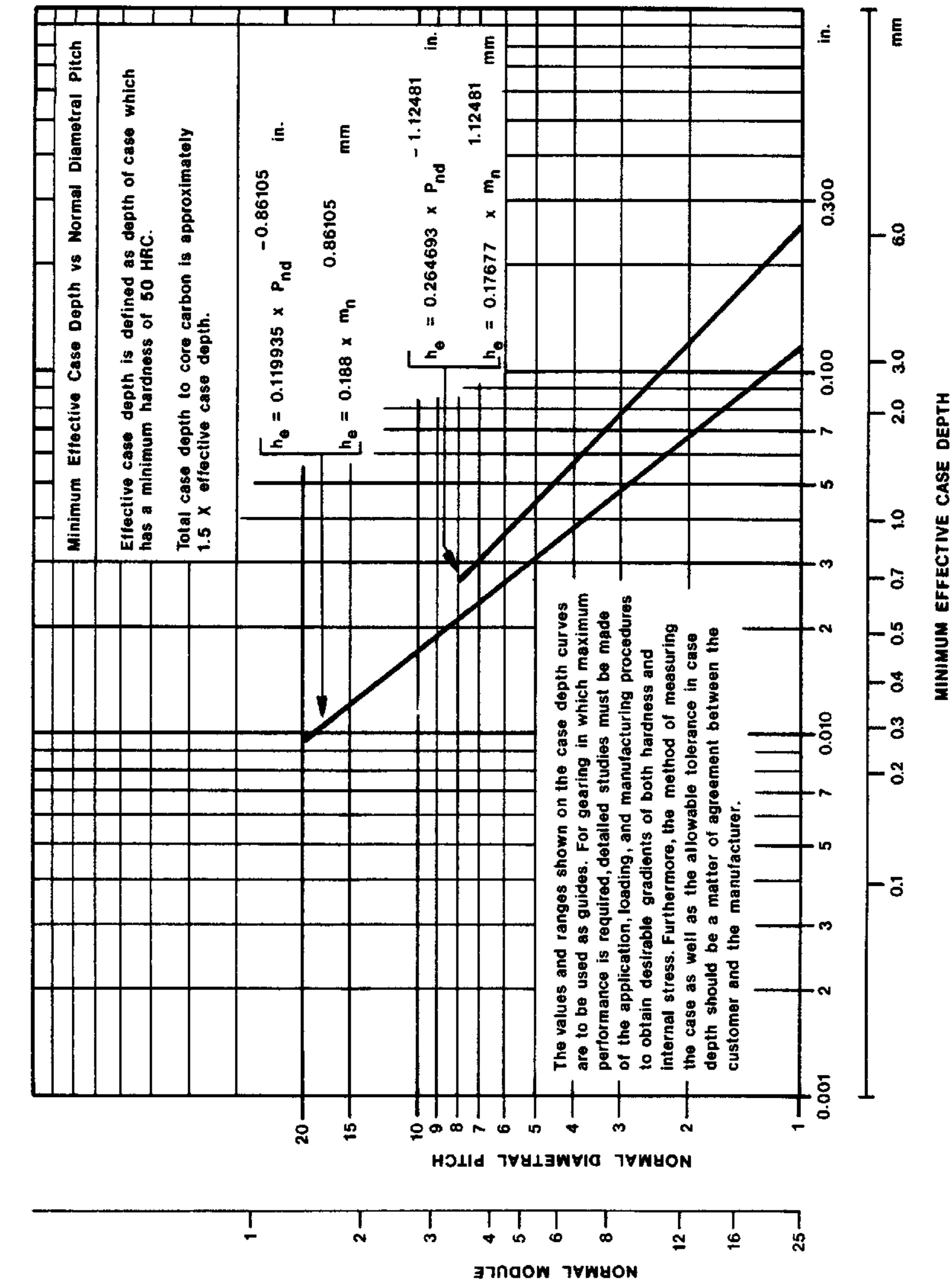



Figure 7—Effective Case Depth for Carburized Gears, $h_e$
(From AGMA 422.03)

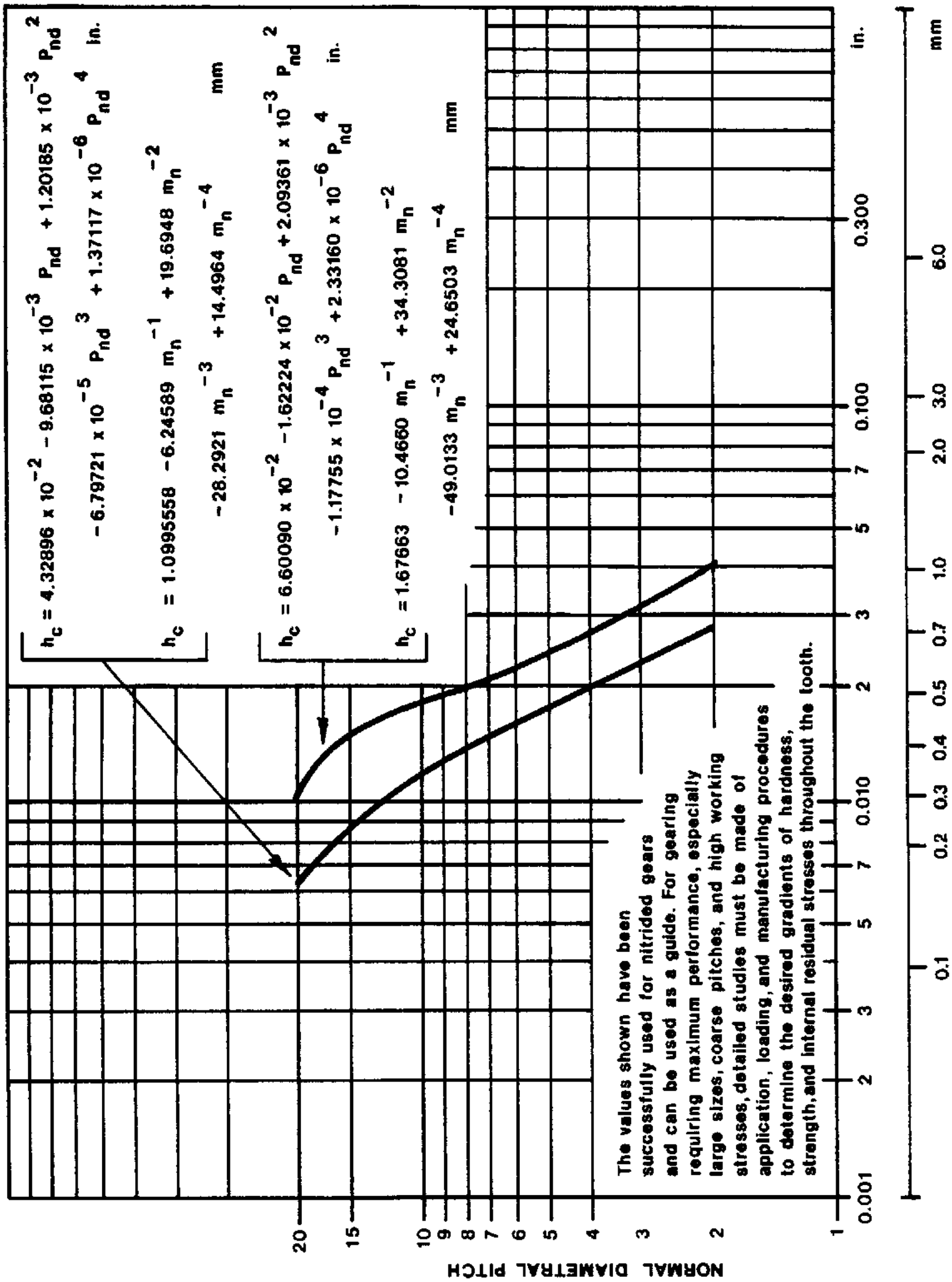



Figure 8—Minimum Total Case Depth for Nitrided Gears, $h_c$
(From AGMA 422.03)

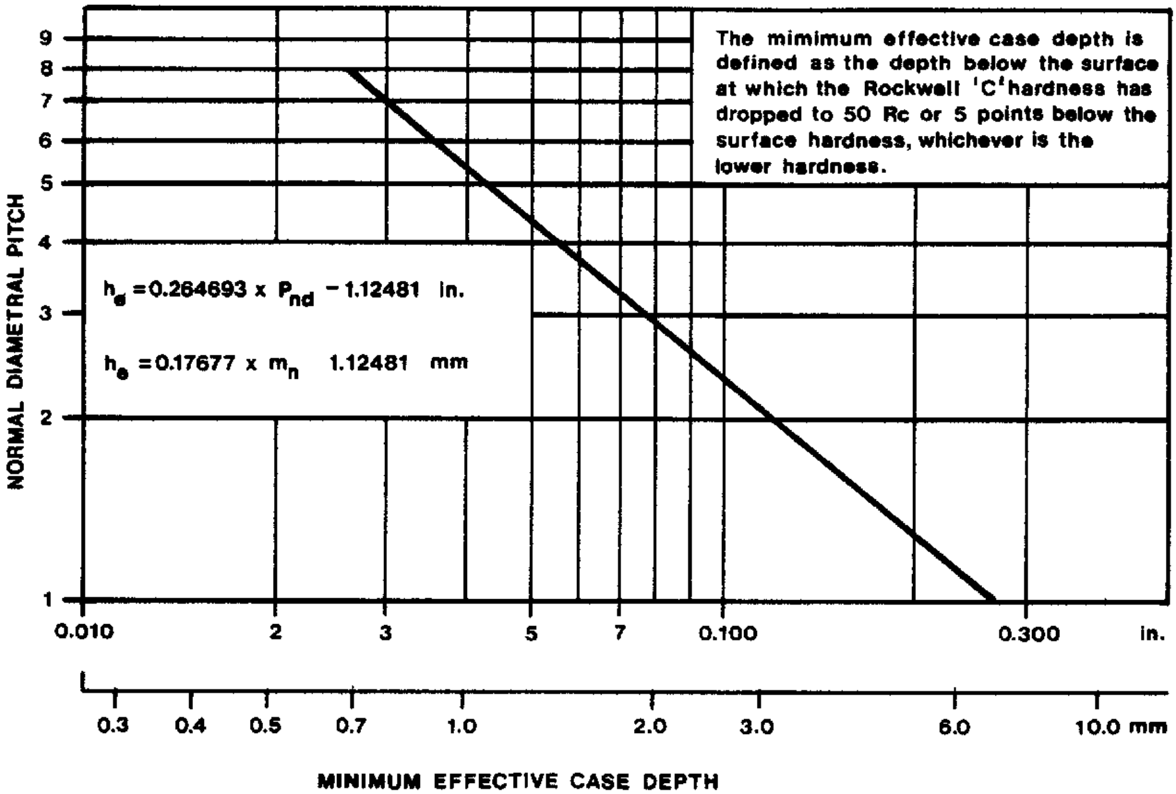



Figure 9—Minimum Effective Case Depth for Flame or Induction Hardened Gears, $h_e$
(From AGMA 422.03)

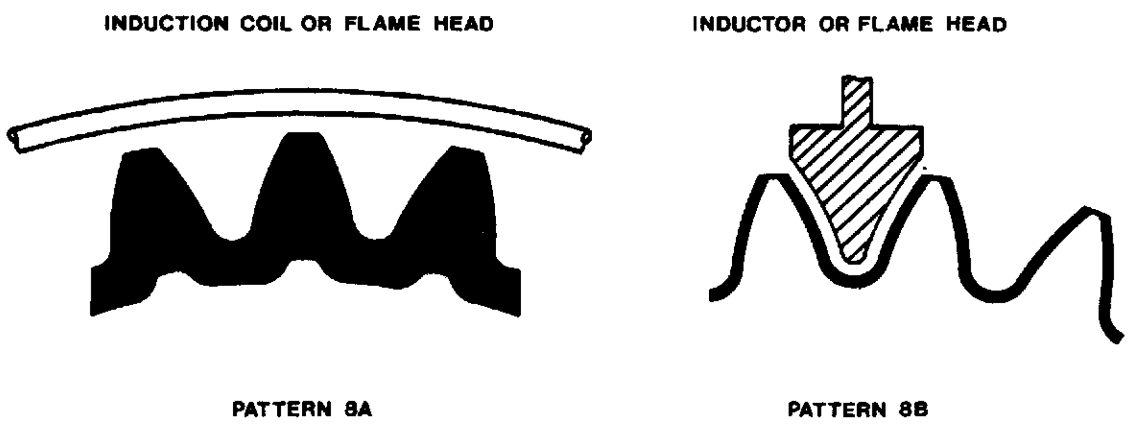



Figure 10—Acceptable Flame and Induction Hardening Patterns
(From AGMA 422.03)

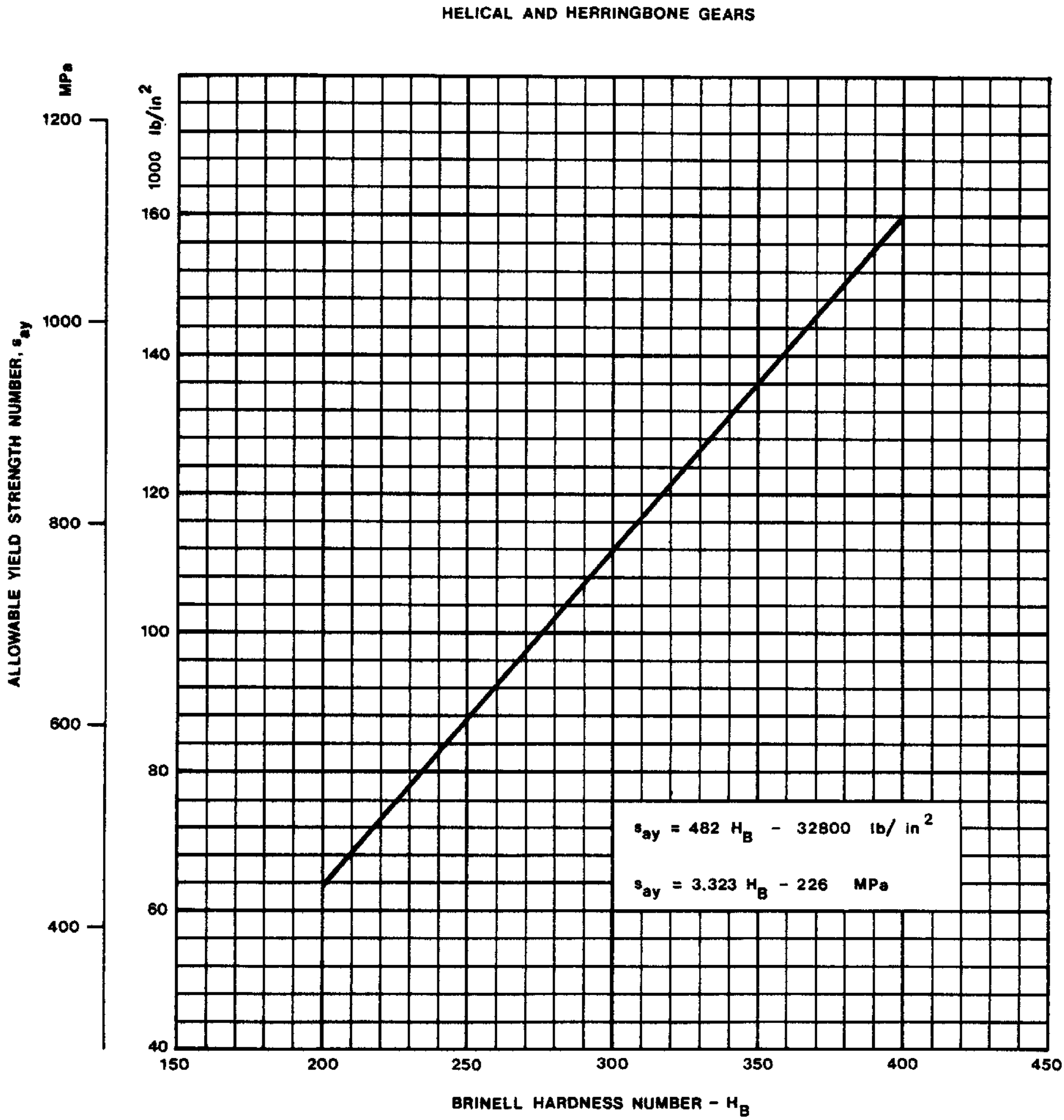



Figure 11—Allowable Yield Strength Number for Steel and Nodular Iron, $s_{ay}$
(From AGMA 422.03)

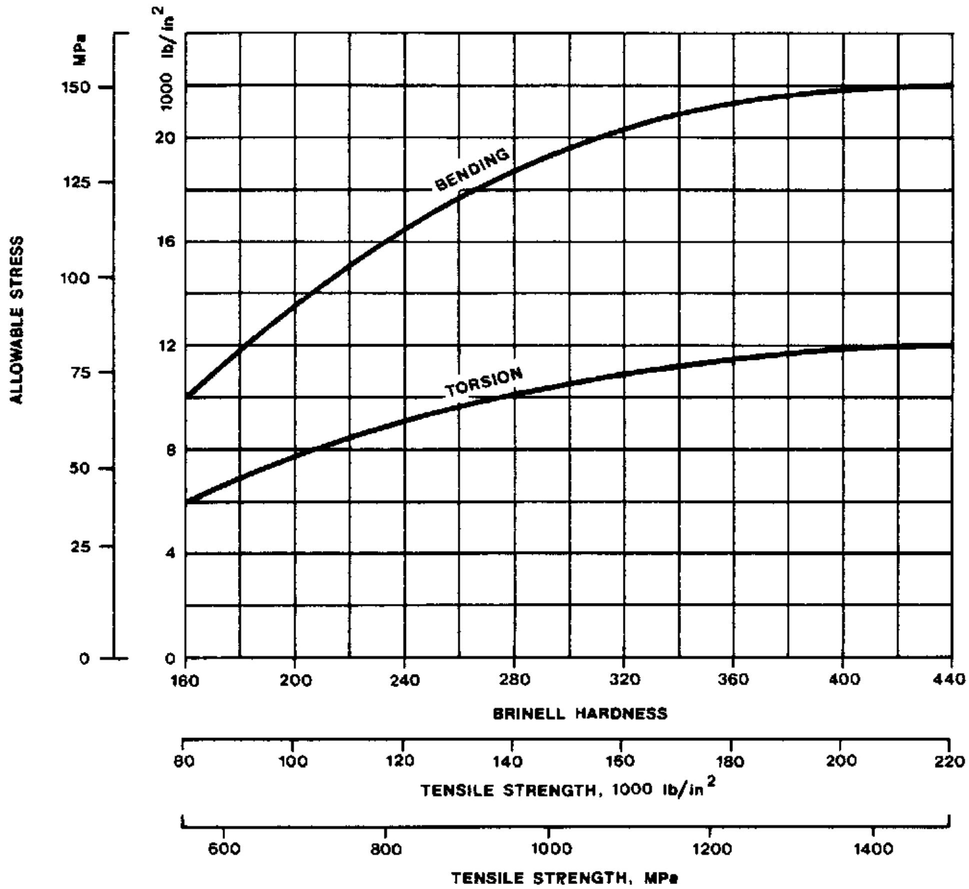
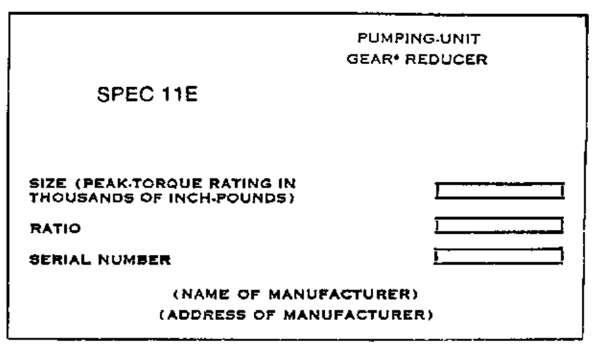



Figure 12—Allowable Stress—Shafting
(From AGMA 422.03)




*Substitute "CHAIN" when appropriate.

Figure 13—Pumping Unit Reducer Nameplate

MANUFACTURED BY: ______________________ DATE SUBMITTED ______________________

NOMINAL API REDUCER SIZE ______________

CALCULATED VALUES

PITTING RESISTANCE TORQUE

First reduction ______________ in.-lb

Second reduction ______________ in.-lb

Third reduction ______________ in.-lb

BENDING STRENGTH TORQUE

First reduction:

Gear ______________ in.-lb, Pinion ________ in.-lb

Second reduction:

Gear ______________ in.-lb, Pinion ________ in.-lb

Third reduction:

Gear ______________ in.-lb, Pinion ________ in.-lb

STATIC TORQUE

First reduction:

Gear ______________ in.-lb, Pinion ________ in.-lb

Second reduction:

Gear ______________ in.-lb, Pinion ________ in.-lb

Third reduction:

Gear ______________ in.-lb, Pinion ________ in.-lb

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, $N_P$ ____________, $N_G$ __________, $P_{nd}$ ________ $P_d$ __________

Second reduction, $N_P$ ____________, $N_G$ __________, $P_{nd}$ ________ $P_d$ __________

Third reduction, $N_P$ ____________, $N_G$ __________, $P_{nd}$ ________ $P_d$ __________

Center distance and net face width:

First reduction, __________C.D., ________F.W.

Second reduction, __________C.D., ________F.W.

Third reduction, __________C.D., ________F.W.

Helix angle and normal pressure angle or transverse pressure angle (degrees):

First reduction, __________H.A., ________NPA, ________ TPA

Second reduction, __________H.A., ________NPA, ________ TPA

Third reduction, __________H.A., ________HPA, ________ TPA

Figure 14—Manufacturer's Gear Reducer Data Sheet

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 BHN/Rc ________, Core BHN[a] ________

Pinion mtl. ________, Surface BHN/Rc ________, Core BHN[a] ________

Second reduction:

Gear material ________, Surface BHN/Rc ________, Core BHN[a] ________

Pinion mtl. ________, Surface BHN/Rc ________, Core BHN[a] ________

Third reduction:

Gear material ________, Surface BHN/Rc ________, Core BHN[a] ________

Pinion mtl. ________, Surface BHN/Rc ________, Core BHN[a] ________

OTHER COMPONENTS

Crankshaft material ________, Hardness ________

Housing material ________

Housing type (Check✓ ): Split ______, One Piece ______

BEARING SIZES[b]

High speed pinion ________

Intermediate speed pinion[c] ________

Low speed pinion ________

Low speed gear ________

BEARING LOADING[d]

High speed pinion ________

Intermediate speed pinion[c] ________

Low speed pinion ________

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 AFBMA (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 psi loading on each bearing. For roller bearings, list L-10 life as calculated in 4.8.

Figure 14—Manufacturer's Gear Reducer Data Sheet (Continued)

# APPENDIX A—APPROVED DATA FORMS

## A.1 Rating Form for Crank Counterbalances

An approved form for rating of pumping unit crank counterbalances is shown in Figure A-1. Manufacturers are urged to use this form when providing the information indicated.

## A.2 Stroke and Torque Factors

An approved form for submission of pumping unit stroke and torque factors is shown in Figure A-2. Manufacturers are urged to use this form when supplying such information.

Name of manufacturer ______________________ Designation of unit ______________________

| (1) | (2) | (3) |
|---|---|---|
| Description[a] | Total Weight (lb) | Maximum[b] Moment About Crankshaft (in.-lb) |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |

[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 in inches, with crank in horizontal position.

Figure A-1—API Rating Form for Crank Counterbalances

Name of manufacturer ____________________

Designation of unit ____________________

Pumping unit structural unbalance ____________________ pounds

| (1) | (2) | (3) | (4) | (5) | (6) | (7) | (8) | (9) |
|---|---|---|---|---|---|---|---|---|
| Position of Crank[a] (degrees) | Position of Rods[b] Length of Stroke, inches | | | | Torque Factor[c] Length of Stroke, inches | | | |
| | | | | | | | | |
| 0 | | | | | | | | |
| 15 | | | | | | | | |
| 30 | | | | | | | | |
| 45 | | | | | | | | |
| 60 | | | | | | | | |
| 75 | | | | | | | | |
| 90 | | | | | | | | |
| 105 | | | | | | | | |
| 120 | | | | | | | | |
| 135 | | | | | | | | |
| 150 | | | | | | | | |
| 165 | | | | | | | | |
| 180 | | | | | | | | |
| 195 | | | | | | | | |
| 210 | | | | | | | | |
| 225 | | | | | | | | |
| 240 | | | | | | | | |
| 255 | | | | | | | | |
| 270 | | | | | | | | |
| 285 | | | | | | | | |
| 300 | | | | | | | | |
| 315 | | | | | | | | |
| 330 | | | | | | | | |
| 345 | | | | | | | | |

$A$ ____________________ $P$ ____________________

$C$ ____________________ $K$ ____________________

$R_1$ ____________________ $H$ ____________________

$R_2$ ____________________ $I$ ____________________

$R_3$ ____________________ $G$ ____________________

Note: See Appendixes B, C, D, or E for symbol identification.

[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 = $\frac{T}{W}$, where $T$ = torque on pumping unit reducer due to polished-rod load $W$.

Figure A-2—Pumping Unit Stroke and Torque Factors

# APPENDIX B—RECOMMENDED PRACTICE FOR THE CALCULATION AND APPLICATION OF TORQUE FACTOR ON PUMPING UNITS (REAR MOUNTED GEOMETRY CLASS I LEVER SYSTEMS WITH CRANK COUNTERBALANCE)

## B.1 Definition

The torque factor for any given crank angle is the factor that, when multiplied by the load in pounds at the polished rod, gives the torque in inch-pounds at the crankshaft of the pumping unit reducer.

## B.2 Method of Calculation

**B.2.1** 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 may be measured. They may also be calculated from the dimensions of the pumping unit by mathematical treatment only. The approved form for submission of torque factor and polished-rod position data is given in Figure A-2.

**B.2.2** Torque factors and polished-rod positions are to be furnished by pumping unit manufacturers for each 15-degree crank position with the zero position at 12 o'clock. Other crank positions are determined by the angular displacement in a clockwise direction viewed with the wellhead to the right. The polished-rod position for each crank position is expressed as a fraction of the stroke above the lowermost position.

**B.2.3** Referring to Figure B-1, the following system of nomenclature and symbols is adopted:

$A$ = Distance from the center of the saddle bearing to the centerline of the polished rod, inches.
$C$ = Distance from the center of the saddle bearing to the center of the equalizer bearing, inches.
$P$ = Effective length of the pitman, inches (from the center of the equalizer bearing to the center of the crankpin bearing).
$R$ = Radius of the crank, inches.
$K$ = Distance from the center of the crankshaft to the center of the saddle bearing, inches.
$H$ = Height from the center of the saddle bearing to the bottom of the base beams, inches.
$I$ = Horizontal distance between the centerline of the saddle bearing and the centerline of the crankshaft, inches.
$G$ = Height from the center of the crankshaft to the bottom of the base beams, inches.
$J$ = Distance from the center of the crankpin bearing to the center of the saddle bearing, inches.
$\phi$ = Angle between the 12 o'clock position and $K$, degrees;

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

$\theta$ = Angle of crank rotation in a clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 12 o'clock, degrees.
$\beta$ = Angle between $C$ and $P$, degrees.
$\alpha$ = Angle between $P$ and $R$, degrees, measured clockwise from $R$ to $P$.
$\psi$ = Angle between $C$ and $K$, degrees (equals angle $\chi$ − angle $\rho$).
$\psi_t$ = Angle between $C$ and $K$, degrees, at top (highest) polished-rod position.
$\psi_b$ = Angle between $C$ and $K$, degrees, at bottom (lowest) polished-rod position.
$\chi$ = Angle between $C$ and $J$, degrees.
$\rho$ = Angle between $K$ and $J$, degrees.
$\overline{TF}$ = Torque factor for a given crank angle $\theta$, inches.
$W$ = Polished-rod load at any specific crank angle $\theta$, pounds.
$B$ = Structural unbalance, pounds; 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.
$W_n$ = Net polished-rod load, pounds; equal to $W - B$.
$T_{wn}$ = Torque, in.-lbs, due to the net polished-rod load for a given crank angle $\theta$ (equals $\overline{TF} \times W_n$).
$M$ = Maximum moment of the rotary counterweights, cranks, and crankpins about the crankshaft, in.-lbs.
$T_r$ = Torque, in.-lbs, due to the rotary counterweights, cranks, and crankpins for a given crank angle $\theta$ (equals $M \sin\theta$).
$T_n$ = Net torque, in.-lbs, at the crankshaft for a given crank angle $\theta$ (equals $T_{wn} - T_r$).
$\overline{PR}$ = Polished-rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle $\theta$.

**B.2.4** By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crank angle $\theta$.

$$\overline{TF} = \frac{AR}{C}\,\frac{\sin\alpha}{\sin\beta} \qquad \text{(B-1)}$$

Sin $\alpha$ is positive when the angle $\alpha$ is between 0 degrees and 180 degrees and is negative when angle $\alpha$ is between 180 degrees and 360 degrees. Sin $\beta$ is always positive because the angle $\beta$ is always between 0 degrees and 180 degrees. A negative torque factor ($\overline{TF}$) only indicates a change in direction of torque on the crankshaft.

$$\phi = \tan^{-1}\left(\frac{I}{H-G}\right) \qquad \text{(B-2)}$$

This is a constant angle for any given pumping unit.

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

The cos of $(\theta - \phi)$ is positive when this angle is between 270 degrees and 90 degrees moving clockwise and is negative from 90 degrees to 270 degrees moving clockwise. When the angle $(\theta - \phi)$ is negative, it should be subtracted from 360 degrees, and the foregoing rules apply.

$$\chi = \cos^{-1}\left(\frac{C^2+J^2-P^2}{2CJ}\right) \qquad \text{(B-4)}$$

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

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

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

$$\alpha = \beta + \psi - (\theta - \phi) \qquad \text{(B-7)}$$

$$\overline{PR} = \frac{\psi_b - \psi}{\psi_b - \psi_t} \qquad \text{(B-8)}$$

$$\psi_b = \cos^{-1}\left[\frac{C^2+K^2-(P+R)^2}{2CK}\right] \qquad \text{(B-9)}$$

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

## B.3 Application of Torque Factors

**B.3.1** 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 (see note):

$$T_n = \overline{TF}(W - B) - M\sin\theta \qquad \text{(B-11)}$$

Note: This equation applies to pumping units where maximum counterbalance moment is obtained at $\theta$ equals 90 degrees or 270 degrees.

**B.3.2** 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-percent crank counterbalance and where crank-speed variation is not more than 15 percent of average, these factors usually can be neglected without introducing errors greater than 10 percent. When beam weights are used, the inertia effects of the weights must 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 balance. 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 B.3.12, B.3.13, and B.3.14.

**B.3.3** 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 $\theta$. The result is the rotary counterbalance effect, in pounds, at the polished rod.

**B.3.4** Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the cranks in the 90-degree or 270-degree 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 $(W)$ 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:

$$M = \overline{TF}(W - B) \qquad \text{(B-12)}$$

To check measurements, the maximum moment, $M$, should be determined with the cranks in both the 90-degree and 270-degree positions. Should there be a significant difference in the maximum moments calculated from measurements at 90-degrees and 270-degrees, 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 suggested that an arithmetic average of the two maximum moments be used.

**B.3.5** To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 4000-foot well is shown in Figure B-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 degrees of crank angle $\theta$. 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 degrees of crank angle $\theta$ are projected upward to intersect the dynamometer card. Then the polished-rod load may be determined for each 15 degrees of crank angle $\theta$.

**B.3.6** To further illustrate, a calculation will be made considering the point where the crank angle $\theta$ equals 75 degrees. From polished-rod stroke and torque factor data for the particular 64-inch stroke 160-D pumping unit used for this example, it is found that the position of the polished-rod at 75 degrees is 0.397 and that the torque factor $\overline{TF}$ is 34.38. 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 B-2). The dynamometer deflection at this point is read to be 1.16 inches, which, with a scale constant of 7450 pounds per inch, makes the load $(W)$ at that point 8650 pounds.

**B.3.7** In a similar manner, the polished-rod load may be obtained for each 15-degree angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15 degrees of crank angle. The structural unbalance, $B$, for the example unit equals +650 pounds. Therefore, the net polished-rod load, $W_n$, at $\theta = 75$ degrees = $W - B$ = 8650 – (+650) = 8000 lbs. The torque, $T_{wn}$, due to the net polished-rod load = $\overline{TF} \times W_n$ = 34.38 × 8000 = 275,000 in.-lb.

**B.3.8** To find the torque, $T_r$, due to the crank counterbalance, the maximum moment, $M$, must be determined. This may be done either from manufacturers' counterbalance tables or curves, or as described in B.3.4. 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.

**B.3.9** The horizontal dotted line drawn across the dynamometer card in Figure B-2 is the counterbalance effect measured with the dynamometer at the 90-degree crank angle and is 6250 pounds. The maximum moment can then be calculated as follows, using Equation B-12:

$$\begin{aligned} M &= \overline{TF}\,(W - B) \\ &= 32.76 \times (6250 - 650) \\ &= 183{,}000 \text{ in.-lb} \end{aligned}$$

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

Although not shown, the measured counterbalance effect for the 270-degree crank position was 6410 lbs. Using the torque factor of 32.04 at the 270-degree crank position for the example unit, the maximum moment is

$$\begin{aligned} M &= 32.04 \times (6410 - 650) \\ &= 185{,}000 \text{ in.-lb} \end{aligned}$$

The maximum moments determined at the 90-degree and 270-degree crank positions are in good agreement, and the average maximum moment of 184,000 in.-lb will be used.

**B.3.10** The torque, $T_r$, due to the counterbalance at the 75-degree crank position would therefore be equal to 184,000 × sin 75-degree = 184,000 × 0.966 = 178,000 in.-lb. The net torque at the crankshaft for the 75-degree crank position would then be calculated from Equation B-11 as follows:

$$\begin{aligned} T_n &= TF\,(W - B)\;\; M \sin\theta \\ &= 34.88 \times (8650 - 650) - 184{,}000 \times 0.966 \\ &= 275{,}000 - 178{,}000 = 97{,}000 \text{ in.-lb} \end{aligned}$$

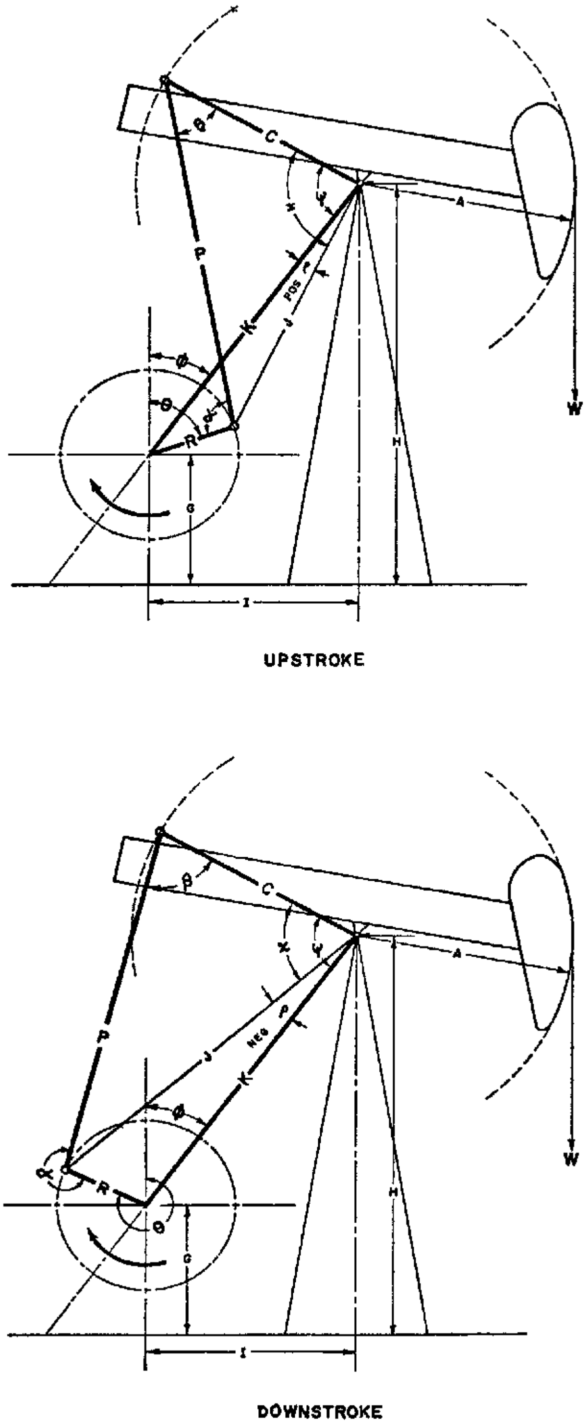
These values may be calculated for other crank angle positions in the same manner as outlined above. Shown in Figure B-3 is a plot of torque versus crank angle that includes the net polished-rod load torque curve, the counterbalance torque curve, and the net crankshaft torque curve.

**B.3.11** The foregoing sample illustration 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 A-2). The position of crank, degrees (column 1) is reversed, starting from the bottom with 15 degrees and counting up in 15-degree increments to 360 degrees.

**B.3.12** The foregoing technique is generally accepted. Those wanting more precise results must realize the true stroke length can vary somewhat with a change in beam position in relation to the centerline of the saddle bearing due to an adjustable feature 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.

**B.3.13** The geometry of the utilized dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. When critical calculations are to be made, the dynamometer manufacturer should be contacted for information on the involved performance characteristics of his dynamometer and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

**B.3.14** It must be recognized that the maximum and minimum loads will most frequently fall at points other than the 15-degree divisions for which torque factors are provided. Interpolation between 15-degree divisions is permissible without significant error.




Note: See B.2.3 for definition of symbols.

Figure B-1—Pumping Unit Geometry

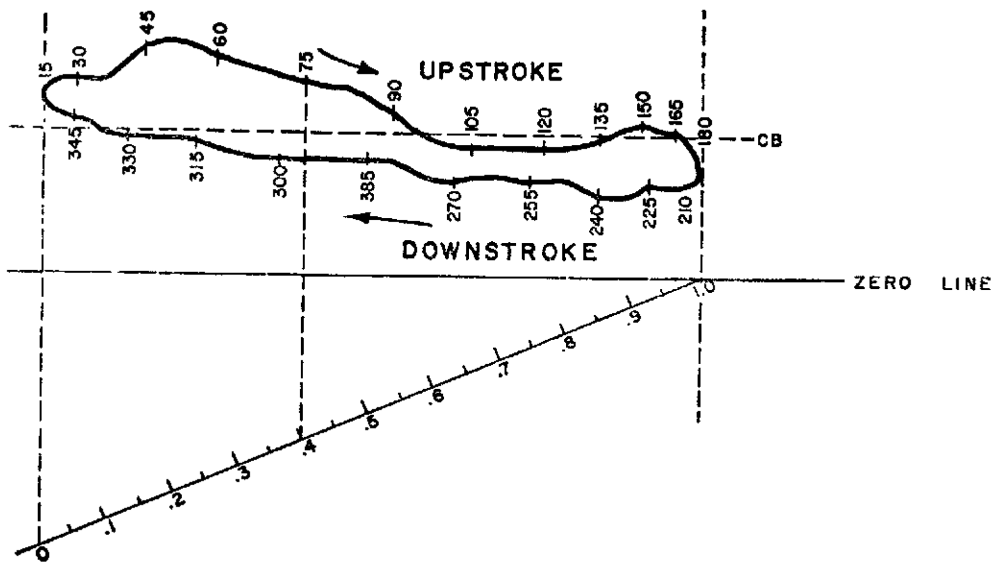



Figure B-2—Division of Dynamometer Card by Crank Angle Using API Polished-Rod Position Data

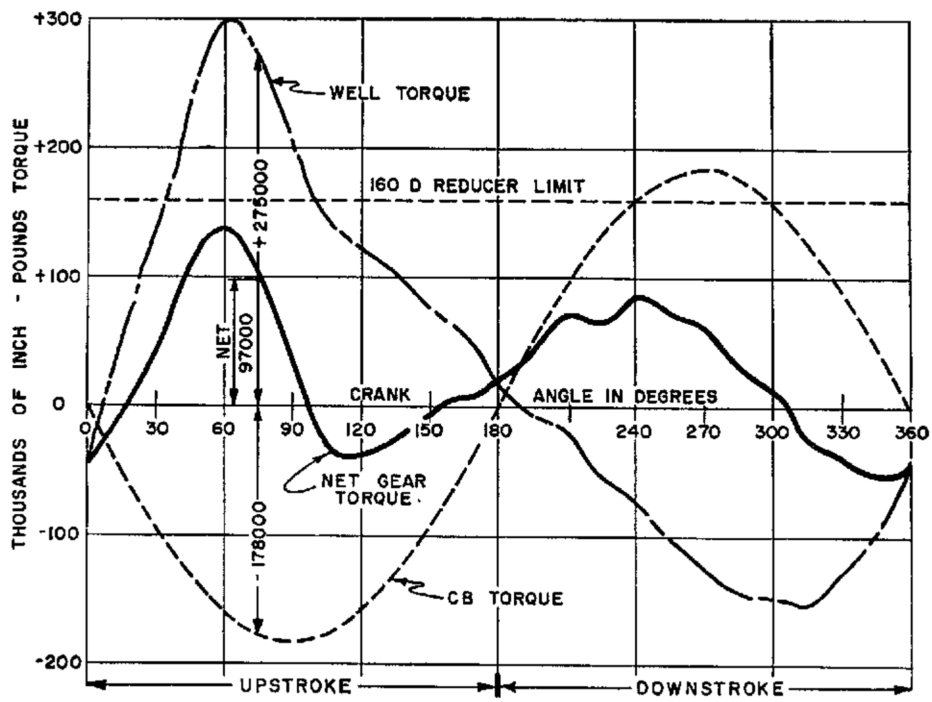



Figure B-3—Torque Curves Using API Torque Factors

## NET REDUCER TORQUE CALCULATION SHEET
## (CONVENTIONAL CRANK BALANCED UNIT ONLY—CLOCKWISE ROTATION)

Well No.: ______________________ Company: ______________________

Unit Size: ______________________ Location: ______________________

| $\theta$ | Sin $\theta$ | $W$ | $B$ | $W-B$ | $\overline{TF}$ | $\overline{TF}(W-B)$ | $-M$ (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 | | | | | | + | |

Note:

$$T_n = \overline{TF}(W-B) - M \sin\theta$$

*Where:*

$T_n$ = Net reducer torque, in.-lbs.
$\theta$ = Position of crank
$M$ = Maximum moment of counterbalance, in.-lbs
$W$ = Measured polished rod load at $\theta$, lbs.
$B$ = Unit structural unbalance, lbs.

$\overline{TF}$ = Torque factor at $\theta$, in.
$CB$ at 90 degrees = ____________
$M$ = ($CB$ at 90 degrees – $B$ ($\overline{TF}$ at 90 degrees)) = ____________

## NET REDUCER TORQUE CALCULATION SHEET
## (CONVENTIONAL CRANK BALANCED UNIT ONLY—COUNTERCLOCKWISE ROTATION)

Well No.: ____________________ Company: ____________________

Unit Size: ____________________ Location: ____________________

| $\theta$ | Sin $\theta$ | $W$ | $B$ | $W-B$ | $\overline{TF}$ | $\overline{TF}(W-B)$ | $-M$ (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 | | | | | | – | |

Note:

$$T_n = \overline{TF}(W-B) - M \sin\theta$$

*Where:*

$T_n$ = Net reducer torque, in.-lbs.
$\theta$ = Position of crank
$M$ = Maximum moment of counterbalance, in.-lbs
$W$ = Measured polished rod load at $\theta$, lbs.
$B$ = Unit structural unbalance, lbs.

$\overline{TF}$ = Torque factor at $\theta$, in.
$CB$ at 270 degrees = ____________
$M$ = ($CB$ at 270 degrees – $B$ ($\overline{TF}$ at 270 degrees)) = ____________

# APPENDIX C—RECOMMENDED PRACTICE FOR THE CALCULATION AND APPLICATION OF TORQUE FACTOR ON PUMPING UNITS (FRONT MOUNTED GEOMETRY CLASS III LEVER SYSTEMS WITH CRANK COUNTERBALANCE)

## C.1 Definition

The torque factor for any given crank angle is the factor that, when multiplied by the load in pounds at the polished rod, gives the torque in inch-pounds at the crankshaft of the pumping unit reducer.

## C.2 Method of Calculation

**C.2.1** 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 may be measured. They may also be calculated from the dimensions of the pumping unit by mathematical treatment only. The approved form for submission of torque factor and polished-rod position data is given in Figure A-2.

**C.2.2** Torque factors and polished-rod positions are to be furnished by pumping unit manufacturers for each 15-degree crank position with the zero position at 6 o'clock. Other crank positions are determined by the angular displacement in a counterclockwise direction viewed with the wellhead to the right. The polished-rod position for each crank position is expressed as a fraction of the stroke above the lowermost position.

**C.2.3** Referring to Figure C-1, the following system of nomenclature and symbols is adopted:

$A$ = Distance from the center of the Samson Post bearing to the centerline of the polished rod, inches.

$C$ = Distance from the center of the Samson Post bearing to the center of the equalizer (or cross yoke) bearing, inches.

$P$ = Effective length of the pitman, inches, (from the center of the equalizer, or cross yoke, bearing to the center of the crankpin bearing).

$R$ = Radius of the crank, inches.

$K$ = Distance from the center of the crankshaft to the center of the Samson Post bearing, inches.

$H$ = Height from the center of the Samson Post bearing to the bottom of the base beams, inches.

$I$ = Horizontal distance between the centerline of the Samson Post bearing and the centerline of the crankshaft, inches.

$G$ = Height from the center of the crankshaft to the bottom of the base beams, inches.

$J$ = Distance from the center of the crankpin bearing to the center of the Samson Post bearing, inches.

$\phi$ = Angle between the 6 o'clock position and $K$, degrees;

$$= \tan^{-1}\left[\frac{I}{H-G}\right] + 180 \text{ degrees}$$

$\theta$ = Angle of crankpin rotation in a counter-clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 6 o'clock, degrees.

$\beta$ = Angle between $C$ and $P$, degrees.

$\alpha$ = Angle between $P$ and $R$, degrees, measured clockwise from $R$ to $P$.

$\psi$ = Angle between $C$ and $K$, degrees, (equals angle $\chi$ – angle $\rho$).

$\psi_t$ = Angle between $C$ and $K$, degrees, at top (highest) polished-rod position.

$\psi_b$ = Angle between $C$ and $K$, degrees, at bottom (lowest) polished-rod position.

$\chi$ = Angle between $C$ and $J$, degrees.

$\rho$ = Angle between $K$ and $J$, degrees.

$\overline{TF}$ = Torque factor for a given crank angle $\theta$, inches.

$W$ = polished-rod load at any specific crank angle $\theta$, pounds.

$W_c$ = Counterbalance in pounds at the polished rod; determined using dynamometer with crankpin at 90-degree position.

$B$ = Structural unbalance, pounds; 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 acts upward on Class III Geometry Units and is negative.

$W_n$ = Net polished-rod load, pounds; equal to $W-B$.

$T_{wn}$ = Torque, in.-lbs, due to the net polished-rod load for a given crank angle $\theta$ (equals $\overline{TF} \times W_n$).

$M$ = Maximum moment of the rotary counterweights, cranks, and crankpins about the crankshaft, in.-lbs.

$\tau$ = Angle of crank counterweight arm offset for front mounted geometry (Class III Lever System).

$T_r$ = Torque, in.-lbs, due to the rotary counterweights, cranks, and crankpins for a given crank angle $\theta$ [equals $M \sin(\theta + \tau)$].

$T_n$ = Net torque, in.-lbs, at the crankshaft for a given crank angle $\theta$ (equals $T_{wn}-T_r$).

$\overline{PR}$ = Polished-rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle $\theta$.

**C.2.4** By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crank angle $\theta$.

$$\overline{TF} = \frac{AR}{C}\ \frac{\sin\alpha}{\sin\beta} \qquad \text{(C-1)}$$

Sin $\alpha$ is positive when the angle $\alpha$ is between 0 degrees and 180 degrees and is negative when angle $\alpha$ is between 180 degrees and 360 degrees. Sin $\beta$ is always positive because the angle $\beta$ is always between 0 degrees and 180 degrees. A negative torque factor ($\overline{TF}$) only indicates a change in direction of torque on the crankshaft.

$$\phi = \tan^{-1}\left[\frac{I}{H-G}\right] + 180 \text{ degrees} \qquad \text{(C-2)}$$

This is a constant angle for any given pumping unit.

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

The sign of cos $(\theta-\phi)$ must be correct. Cos $(\theta-\phi)$ is negative when $(\theta-\phi)$ is 90 degrees through 270 degrees. It is positive for all other angles between 0 degrees and 360 degrees. When the angle $(\theta-\phi)$ is a negative number, it can be subtracted from 360 degrees and this new angle can be used to determine the proper sign.

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

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

For Equation C-6 to be correct, it is necessary to use the proper sign for the angle $\rho$. The angle $\rho$ is taken as positive when sin $\rho$ is positive. This occurs for crank positions where $(\theta-\phi)$ = 0 degrees to $(\theta-\phi)$ = 180 degrees. The angle $\rho$ is taken as negative when sin $\rho$ is negative. This occurs for crank positions $(\theta-\phi)$ = 180 degrees through $(\theta-\phi)$ = 360 degrees. When the angle $(\theta-\phi)$ is negative, it can be subtracted from 360 degrees, and this new angle can be used to determine the proper sign.

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

$$\sin\alpha = \sin[(\theta-\phi) - \psi - \beta] \qquad \text{(C-7)}$$

$$\overline{PR} = \frac{\psi_b - \psi}{\psi_b - \psi_t} \qquad \text{(C-8)}$$

$$\psi_t = \cos^{-1}\left[\frac{C^2+K^2-(P+R)^2}{2CK}\right] \qquad \text{(C-9)}$$

$$\psi_b = \cos^{-1}\left[\frac{C^2+K^2-(P-R)^2}{2CK}\right] \qquad \text{(C-10)}$$

## C.3 Application of Torque Factors

**C.3.1** 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 equation:

$$T_n = \overline{TF}(W-B) - M\sin(\theta+\tau) \qquad \text{(C-11)}$$

**C.3.2** 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, equalizer (or cross yoke), pitman, crank, and crank counterweights; and neglects friction in the bearings. For units having 100-percent crank counterbalance and where crank-speed variation is not more than 15 percent of average, these factors usually can be neglected without introducing errors greater than 10 percent. 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 C.3.11, C.3.12, and C.3.13.

**C.3.3** 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 $\theta$. The result is the rotary counterbalance effect, in pounds, at the polished rod.

**C.3.4** Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the crankpins in the 90-degree 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 counterbalance effect in pounds ($W_c$) is the combined effect of the rotary counterbalance and the structural unbalance. The maximum rotary counterbalance moment can then be determined from the equation:

$$M = \frac{\overline{TF}(W_c - B)}{\sin(90 \text{ degrees} + \tau)} \qquad \text{(C-12)}$$

**C.3.5** To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 2872-foot well is shown in Figure C-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 degrees of crank angle $\theta$. 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 degrees of crank angle $\theta$ are projected upward to intersect the dynamometer card. Then the polished-rod load may be determined for each 15 degrees of crank angle $\theta$.

**C.3.6** To further illustrate, a calculation will be made considering the point where the crank angle $\theta$ equals 60 degrees.

From polished-rod stroke and torque factor data for the particular 86-inch stroke 160-D pumping unit used for this example, it is found that the position of the polished rod at 60 degrees is 0.405, and that the torque factor $\overline{TF}$ is 35.45. 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 C-2). The dynamometer deflection at this point is read to be 0.99 inches, which, with a scale constant of 7500 pounds per inch, makes the load ($W$) at that point 7425 pounds.

**C.3.7** In a similar manner, the polished-rod load may be obtained for each 15-degree angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15 degree of crank angle. The structural unbalance, $B$, for the example unit equals $-1535$ lbs. Therefore, the net polished-rod load, $W_n$ at $\theta = 60$ degrees $= W - B = 7425 - (-1535) = 8960$ pounds. The torque, $T_{wn}$, due to the net polished-rod load $= \overline{TF} \times W_n = 36.45 \times 8960 = 326{,}592$ in.-lb.

**C.3.8** To find the torque, $T_r$, due to the crank counterbalance, the maximum moment, $M$, must be determined. This may be done either from manufacturers' counterbalance tables or curves, or as described in C.3.4. Should the manufacturers' counterbalance data be used, it is suggested that a check be made using a polished-rod measurement technique.

**C.3.9** The horizontal dotted line drawn across the dynamometer card in Figure C-2 is the counterbalance effect measured with the dynamometer at the 90-degree crank angle and is 4594 pounds. The maximum moment can then be calculated as follows, using Equation C-12:

$$M = \frac{\overline{TF}\,(W_c - B)}{\sin\,(90 \text{ degrees} + \tau)}$$

$$= 38.38 \times (4594 + 1535)/.891$$

$$= 264{,}008 \text{ in.-lb}$$

(The torque factor of 38.38 is the value at the 90-degree crank position, and angle $\tau$ is 27 degrees for the example unit.)

**C.3.10** The torque, $T_r$, due to the counterbalance at the 60 degree crank position would therefore be equal to 264,008 $\times$ sin (60 degrees + 27 degrees) = 264,008 $\times$ 0.999 = 263,744 in.-lb. The net torque at the crankshaft for the 60-degree crank position would then be calculated from Equation C-11 as follows:

$$T_n = \overline{TF}\,(W - B) - M \sin\,(\theta + \tau)$$

$$= T_{wn} - T_r$$

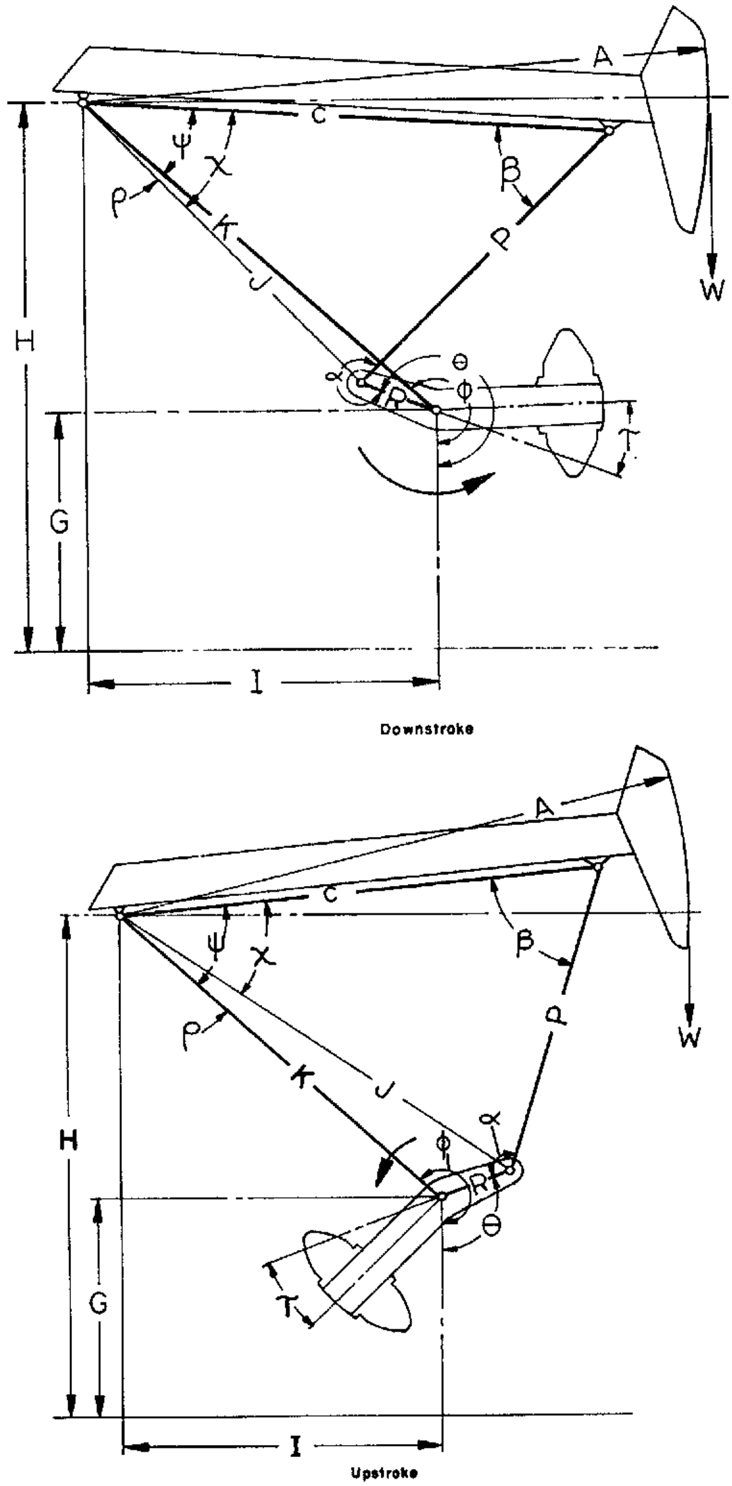
$$= 326{,}592 - 263{,}744 = 62{,}848 \text{ in.-lb}$$

These values may be calculated for other crank angle positions in the same manner at outlined above. Shown in Figure C-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.

**C.3.11** The foregoing technique is generally accepted. Those wanting more precise results must realize the true stroke length can vary somewhat with a change in beam position in relation to the centerline of the saddle bearing due to an adjustable feature 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.

**C.3.12** The geometry of the utilized dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. When critical calculations are to be made, the dynamometer manufacturer should be contacted for information on the involved performance characteristics of his dynamometer and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

**C.3.13** It must be recognized that the maximum and minimum loads will most frequently fall at points other than the 15-degree divisions for which torque factors are provided. Interpolation between 15-degree divisions is permissible without significant error.




Note: See C.2.3 for definition of symbols.

**Figure C-1—Front Mounted Geometry, Class III Lever System**

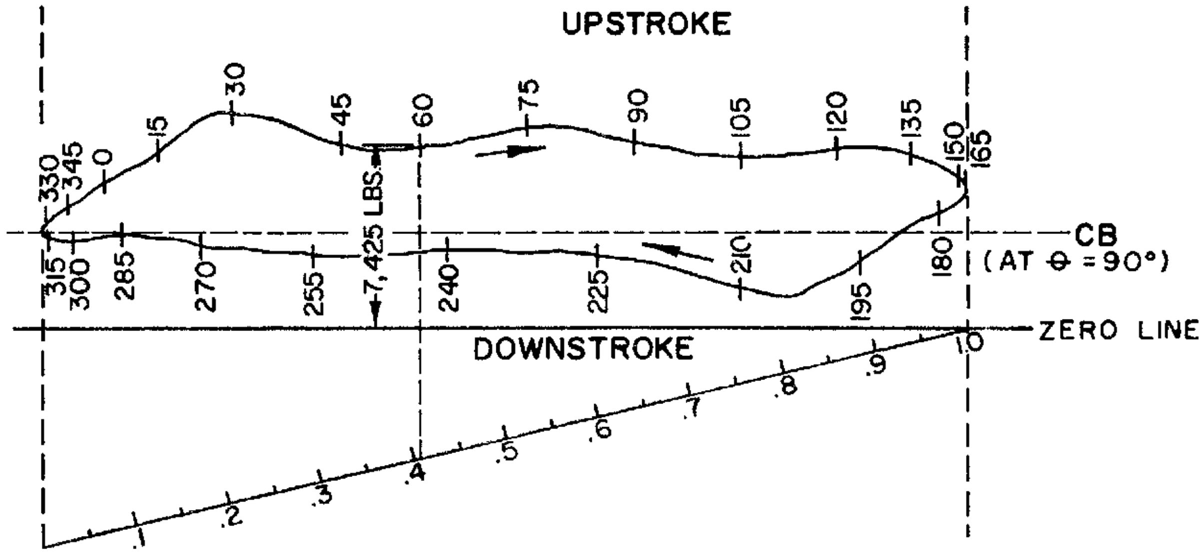



Figure C-2—Division of Dynamometer Card by Crank Angle Using API Polished-Rod Position Data

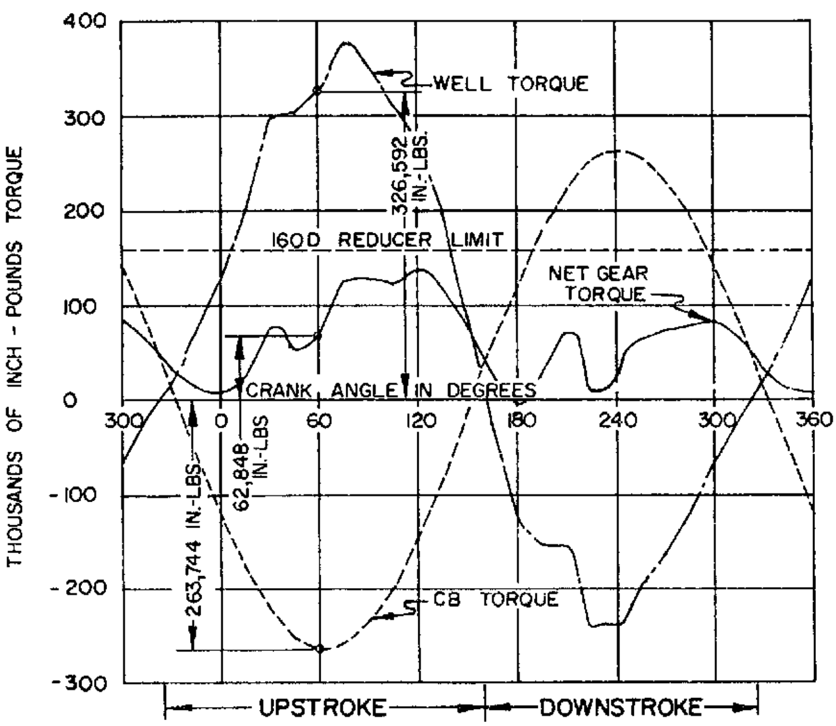



Figure C-3—Torque Curves Using API Torque Factors

# APPENDIX D—RECOMMENDED PRACTICE FOR THE CALCULATION AND APPLICATION OF TORQUE FACTOR ON PUMPING UNITS (FRONT MOUNTED GEOMETRY CLASS III LEVER SYSTEM AIR COUNTERBALANCE)

## D.1 Definition

The torque factor for any given crank angle is the factor that, when multiplied by the load in pounds at the polished rod, gives the torque in inch-pounds at the crankshaft of the pumping unit reducer.

## D.2 Method of Calculation

**D.2.1** 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 may be measured. They may also be calculated from the dimensions of the pumping unit by mathematical treatment only. The approved form for submission of torque factor and polished-rod position data is given in Figure A-2.

**D.2.2** Torque factors and polished-rod positions are to be furnished by pumping unit manufacturers for each 15-degree crank position with the zero position at 6 o'clock. Other crank positions are determined by the angular displacement in a clockwise direction viewed with the wellhead to the right. The polished-rod position for each crank position is expressed as a fraction of the stroke above the lowermost position.

**D.2.3** Referring to Figure D-1, the following system of nomenclature and symbols is adopted:

$A$ = Distance from the center of the Samson Post bearing to the centerline of the polished-rod, inches.
$C$ = Distance from the center of the Samson Post bearing to the center of the equalizer bearing, inches.
$P$ = Effective length of the pitman, inches (from the center of the equalizer bearing to the center of the crankpin bearing).
$R$ = Radius of the crank, inches.
$K$ = Distance from the center of the crankshaft to the center of the Samson Post bearing, inches.
$H$ = Height from the center of the Samson Post bearing to the bottom of the base beams, inches.
$I$ = Horizontal distance between the centerline of the Samson Post bearing and the centerline of the crankshaft, inches.
$G$ = Height from the center of the crankshaft to the bottom of the base beams, inches.
$J$ = Distance from the center of the crankpin bearing to the center of the Samson Post bearing, inches;
$= \sqrt{C^2 + P^2 - (2CP\cos\beta)}$
$\phi$ = Angle between the 6 o'clock position and $K$, degrees;
$= 180 \text{ degrees} - \tan^{-1}\left[\dfrac{I}{H-G}\right]$
$\theta$ = Angle of crank rotation in a clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 6 o'clock, degrees.
$\beta$ = Angle between $C$ and $P$, degrees.
$\alpha$ = Angle between $P$ and $R$, degrees, measured clockwise from $R$ to $P$.
$\psi$ = Angle between $C$ and $K$, degrees (equals angle $\chi$ − angle $\rho$).
$\psi_t$ = Angle between $C$ and $K$, degrees, at top (highest) polished-rod position.
$\psi_b$ = Angle between $C$ and $K$, degrees, at bottom (lowest) polished-rod position.
$\chi$ = Angle between $C$ and $J$, degrees.
$\rho$ = Angle between $K$ and $J$, degrees.
$\overline{TF}$ = Torque factor for a given crank angle $\theta$, inches.
$W$ = polished-rod load at any specific crank angle $\theta$, pounds.
$W_c$ = Counterbalance effect at the polished rod at any specific crank angle $\theta$, pounds [equals $M(P_a - S)$].
$T_n$ = Net torque, in.-lbs, at the crankshaft for a given crank angle $\theta$.
$M$ = Geometry constant for a given unit, square inches (distance from Samson Post bearing to air tank bearing multiplied by the area of piston in the air cylinder divided by the distance from the Samson Post bearing to the centerline of the polished rod).
$P_a$ = Pressure, psig, in air counterbalance tank for a given crank position $\theta$.
$S$ = Pressure, psig, in air counterbalance tank required to offset the weight of the walking beam, horsehead, equalizer, pitmans, etc.
$\overline{PR}$ = Polished-rod position expressed as a fraction of the stroke length above the lowermost position for a given crank angle $\theta$.

**D.2.4** By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crank angle $\theta$.

$$\overline{TF} = \frac{AR}{C}\frac{\sin\alpha}{\sin\beta} \tag{D-1}$$

Sin $\alpha$ is positive when the angle $\alpha$ is between 0 degrees and 180 degrees and is negative when angle $\alpha$ is between 180 degrees and 360 degrees. Sin $\beta$ is always positive because the angle $\beta$ is always between 0 degrees and 180 degrees. A negative torque factor ($\overline{TF}$) only indicates a change in direction of torque on the crankshaft.

$$\phi = 180 \text{ degrees} - \tan^{-1}\left[\frac{I}{H-G}\right] \tag{D-2}$$

This is a constant angle for any given pumping unit.

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

The cos of $(\theta - \phi)$ is positive when this angle is between 270 degrees and 90 degrees moving clockwise and is negative from 90 degrees to 270 degrees moving clockwise. When the angle $(\theta - \phi)$ is negative, it should be subtracted from 360 degrees, and the foregoing rules apply.

$$\chi = \sin^{-1}\left[\frac{P\sin\beta}{J}\right] \quad \text{(D-4)}$$

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

The angle $\rho$ is taken as a positive angle when $\sin\rho$ is positive. This occurs for crank positions between $(\theta - \phi) = 0$ degrees to $(\theta - \phi) = 180$ degrees. The angle $\rho$ is taken as a negative angle when $\sin\rho$ is negative. This occurs for crank positions between $(\theta - \phi) = 180$ degrees to $(\theta - \phi) = 360$ degrees.

$$\psi = \chi + \rho \quad \text{(D-6)}$$

$$\sin\alpha = \sin[\beta + \psi + (\theta - \phi)] \quad \text{(D-7)}$$

$$\overline{PR} = \frac{\psi_b - \psi}{\psi_b - \psi_t} \quad \text{(D-8)}$$

$$\psi_t = \cos^{-1}\left[\frac{C^2+K^2-(P+R)^2}{2CK}\right] \quad \text{(D-9)}$$

$$\psi_b = \cos^{-1}\left[\frac{C^2+K^2-(P-R)^2}{2CK}\right] \quad \text{(D-10)}$$

## D.3 Application of Torque Factors

**D.3.1** 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:

$$T_n = \overline{TF}(W - W_c) \quad \text{(D-11)}$$

**D.3.2** 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, equalizer, pitman, and crank; and neglects friction in the Sampson Post, equalizer, and pitman bearings. For units where crank-speed variation is not more than 15 percent of average, these factors usually can be neglected without introducing errors greater than 10 percent. Some nondynamic factors that can have an effect on the determination of instantaneous net torque loadings, and accordingly should be recognized or considered, are outlined in D.3.9, D.3.10, and D.3.11.

**D.3.3** To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 5560-foot 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 degrees of crank angle $\theta$. 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 degrees of crank angle are projected upward to intersect the dynamometer card. Then the polished-rod load may be determined for each 15 degrees of crank angle.

**D.3.4** The counterbalance line may then be drawn on the card. To avoid time-consuming geometrical considerations, it can 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 percent lower than the actual counterbalance around the midpoint of the stroke, slightly higher at the bottom of the stroke, and nearly equal at the top of the stroke.

For the sample calculation, the recorded maximum air counterbalance tank pressure at the bottom of the stroke, 0-degree crank position, was 328 pounds per square inch guage and the minimum air pressure at the top of the stroke, 180-degree crank position, was 262 pounds per square inch guage. Using the equation, $W_c = M(P_a - S)$, where $M = 52.5$ square inches and $S$ is 73 pounds per square inch gauge (as furnished by the pumping unit manufacturer), we calculate the following results:

a. Maximum counterbalance at the 0-degree crank position is $W_c = 52.5\ (328 - 73) = 13{,}388$ pounds counterbalance at the polished rod. 13,388 divided by the scale constant, 11,300 pounds per inch, gives 1.185 inches.

b. Minimum counterbalance at the 180-degree crank position is $W_c = 52.5\ (262 - 73) - 9923$ pounds. 9923 divided by 11,300 pounds per inch gives 0.878 inch.

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

**D.3.5** To further illustrate, a calculation will be made considering the point where the crank angle $\theta$ equals 75 degrees. From polished-rod stroke and torque factor data for the particular 86-inch stroke 320-D pumping unit used for this example, it is found that the position of the polished-rod at 75 degrees is 0.332 and that the torque factor $\overline{TF}$ is 39.02. 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 (Figure D-2). The dynamometer deflection at this point is read to be 1.45 inches, which, with a scale constant of 11,300 pounds per inch, makes the load ($W$) at that point 16,385 pounds.

**D.3.6** In a similar manner, the polished-rod load may be obtained for each 15-degree angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15 degrees 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-degree crank position. The maximum dynamometer deflection at this point is 1.60 inches, which when multiplied by the scale constant of 11,300 pounds per inch gives 18,080 pounds polished-rod load.

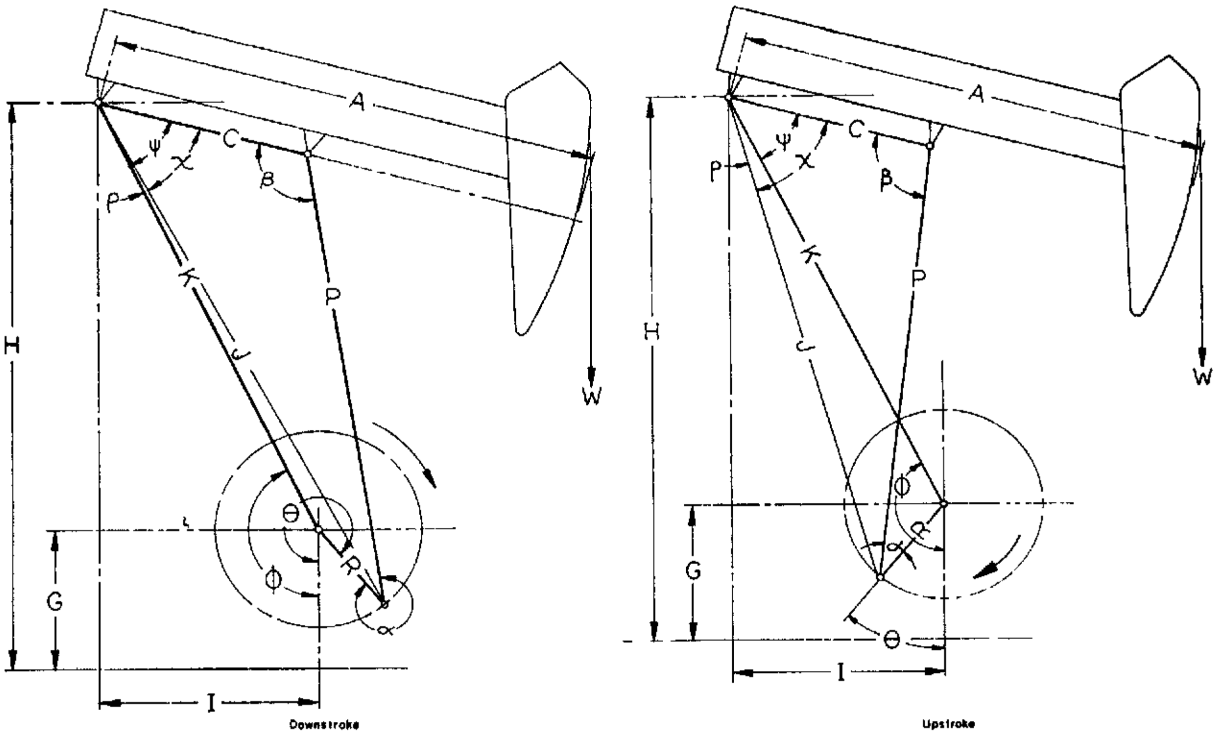
**D.3.7** The net torque, $T_n$, can now be determined. In the equation $T_n = \overline{TF}\,(W - W_c)$, the value $(W - W_c)$ is represented by the difference in the dynamometer deflection between the card and the counterbalance line. Referring to the card in Figure D-2, we read the difference in the dynamometer deflection between the counterbalance line and the well card as 0.36 inch at 75-degree crank position. This value multiplied by the scale constant of 11,300 and the torque factor of 39.25 at 75-degree crank position gives 159,669 in.-lbs net torque. These values may be calculated for other crank positions in the same manner. Figure D-3 is a plot of the net torque curve.

**D.3.8** The foregoing sample illustration 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 the polished-rod position data form (Figure A-2). The position of crank, degrees (column 1), is reversed, starting from the bottom with 15 degrees and counting up in 15-degree increments to 360 degrees.

**D.3.9** The foregoing technique is generally accepted. Those wanting more precise results must realize the true stroke length can vary somewhat with a change in beam position in relation to the centerline of the saddle bearing due to an adjustable feature 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.3.10** The geometry of the utilized dynamometer can influence the determination of instantaneous load values for the various specified or selected crank angles. When critical calculations are to be made, the dynamometer manufacturer should be contacted for information on the involved performance characteristics of his dynamometer and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

**D.3.11** It must be recognized that the maximum and minimum loads will most frequently fall at points other than the 15-degree divisions for which torque factors are provided. Interpolation between 15-degree divisions is permissible without significant error.




Note: See D.2.3 for definition of symbols.

Figure D-1—Pumping Unit Geometry

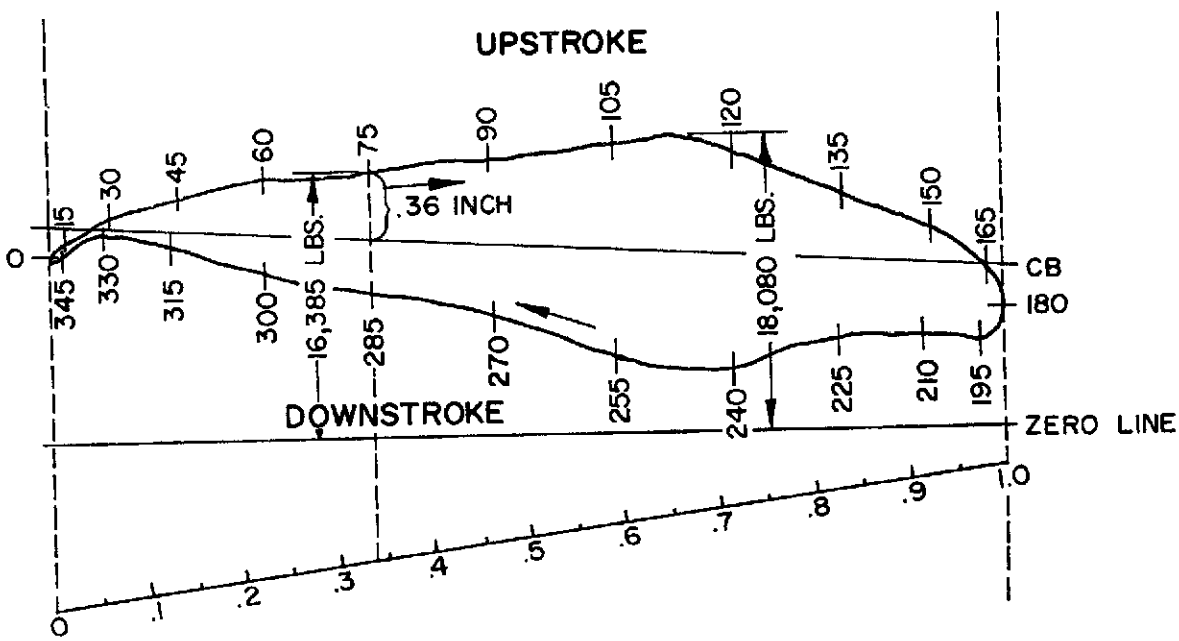



Figure D-2—Division of Dynamometer Card by Crank Angle Using API Polished-Rod Position Data

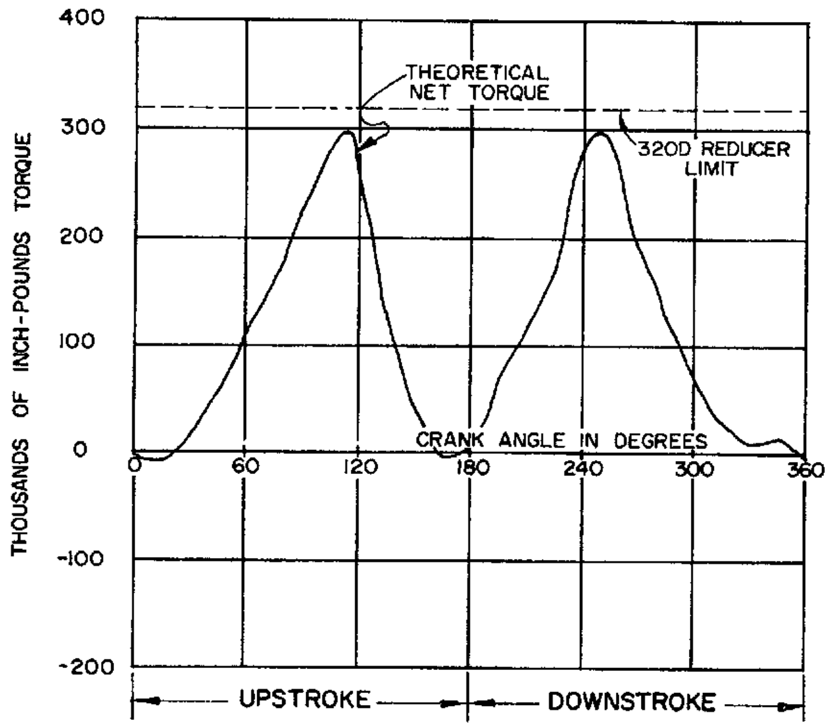



Figure D-3—Torque Curves Using API Torque Factors

# APPENDIX E—RECOMMENDED PRACTICE FOR THE CALCULATION AND APPLICATION OF TORQUE FACTOR ON PUMPING UNITS (REAR MOUNTED GEOMETRY CLASS I LEVER SYSTEMS WITH PHASED CRANK COUNTERBALANCE)

## E.1 Definition

The torque factor for any given crankpin angle is the factor that, when multiplied by the load in pounds at the polished rod, gives the torque in inch-pounds at the crankshaft of the pumping unit reducer.

## E.2 Method of Calculation

**E.2.1** 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 may be measured. They may also be calculated from the dimensions of the pumping unit by mathematical treatment only. The approved form for submission of torque factor and polished-rod position data is given in Figure A-2.

**E.2.2** Torque factors and polished-rod positions are to be furnished by pumping unit manufacturers for each 15-degree crankpin position with the zero position at 12 o'clock. Other crankpin positions are determined by the angular displacement in a clockwise direction viewed with the wellhead to the right. The polished-rod position for each crankpin position is expressed as a fraction of the stroke above the lowermost position.

**E.2.3** Referring to Figure E-1, the following system of nomenclature and symbols is adopted:

$A$ = distance from the center of the saddle bearing to the centerline of the polished rod, inches.

$C$ = distance from the center of the saddle bearing to the center of the equalizer bearing, inches.

$P$ = effective length of the pitman, inches (from the center of the equalizer bearing to the center of the crankpin bearing).

$R$ = radius of the crankpin, inches.

$K$ = distance from the center of the crankshaft to the center of the saddle bearing, inches.

$H$ = height from the center of the saddle bearing to the bottom of the base beams, inches.

$I$ = horizontal distance between the centerline of the saddle bearing and the centerline of the crankshaft, inches.

$G$ = height from the center of the crankshaft to the bottom of the base beams, inches.

$J$ = distance from the center of the crankpin bearing to the center of the saddle bearing, inches.

$\phi$ = angle between the 12 o'clock position and $K$, degrees;

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

$\theta$ = angle of crankpin rotation in a clockwise direction viewed with the wellhead to the right and with zero degrees occurring at 12 o'clock, degrees.

$\beta$ = angle between $C$ and $P$, degrees.

$\alpha$ = angle between $P$ and $R$, degrees, measured clockwise from $R$ to $P$.

$\psi$ = angle between $C$ and K, degrees, (equals angle $\chi$ − angle $\rho$).

$\psi_t$ = angle between $C$ and $K$, degrees, at top (highest) polished-rod position.

$\psi_b$ = angle between $C$ and $K$, degrees, at bottom (lowest) polished-rod position.

$\chi$ = angle between $C$ and $J$, degrees.

$\rho$ = angle between $K$ and $J$, degrees.

$\overline{TF}$ = torque factor for a given crankpin angle $\theta$, inches.

$W$ = polished-rod load at any specific crankpin angle $\theta$, pounds.

$B$ = structural unbalance, pounds; 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.

$W_n$ = net polished-rod load, pounds; equal to $W$–$B$.

$T_{wn}$ = torque, inch-pounds; due to the net polished-rod load for a given crankpin angle $\theta$ (equals $\overline{TF} \times W_n$).

$M$ = maximum moment of the rotary counterweights, cranks, and crankpins about the crankshaft, inch-pounds.

$\tau$ = angle of crank counterweight arm offset (negative when weights are counterclockwise relative to crankpin bearings).

$T_r$ = torque, inch-pounds, due to the rotary counterweights, cranks and crankpins for a given crankpin angle $\theta$ [equals $M \sin(\theta + \tau)$].

$T_n$ = net torque, inch-pounds, at the crankshaft for a given pin angle $\theta$ (equals $T_{wn} - T_r$).

$\overline{PR}$ = polished-rod position expressed as a fraction of the stroke length above the lowermost position for a given crankpin angle $\theta$.

**E.2.4** By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crankpin angle $\theta$.

$$\overline{TF} = \frac{AR}{C} \frac{\sin \alpha}{\sin \beta} \quad \text{(E-1)}$$

Sin $\alpha$ is positive when the angle $\alpha$ is between 0 degrees and 180 degrees and is negative when angle $\alpha$ is between 180 degrees and 360 degrees. Sin $\beta$ is always positive because the angle $\beta$ is always between 0 degrees and 180 degrees. A negative torque factor ($\overline{TF}$) only indicates a change in direction of torque on the crankshaft.

$$\phi = \tan^{-1}\left(\frac{I}{H\text{–}G}\right) \quad \text{(E-2)}$$

This is a constant angle for any given pumping unit.

$$\beta = \cos^{-1} \frac{C^2+P^2-K^2-R^2+2KR\cos(\theta-\phi)}{2CP} \quad \text{(E-3)}$$

The cos of $(\theta-\phi)$ is positive when this angle is between 270 degrees and 90 degrees moving clockwise and is negative from 90 degrees to 270 degrees moving clockwise. When the angle $(\theta-\phi)$ is negative, it should be subtracted from 360 degrees and the foregoing rules apply.

$$\chi = \cos^{-1}\left(\frac{C^2+J^2-P^2}{2CJ}\right) \quad \text{(E-4)}$$

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

The angle $\rho$ is taken as a positive angle when sin $\rho$ is positive. This occurs for crankpin positions between $(\theta-\phi) = 0$ degrees and $(\theta-\phi) = 180$ degrees. The angle $\rho$ is taken as a negative angle when sin $\rho$ is negative. This occurs from crankpin positions between $(\theta-\phi) = 180$ degrees and $(\theta-\phi) = 360$ degrees.

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

$$\sin a = \sin[\beta+\psi-(\theta-\phi)] \quad \text{(E-7)}$$

$$\overline{PR} = \frac{\psi_b-\psi}{\psi_b-\psi_t} \quad \text{(E-8)}$$

$$\psi_b = \cos^{-1} \frac{C^2+K^2-(P+R)^2}{2CK} \quad \text{(E-9)}$$

$$\psi_t = \cos^{-1} \frac{C^2+K^2-(P-R)^2}{2CK} \quad \text{(E-10)}$$

## E.3 Application of Torque Factors

**E.3.1** 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:

$$T_n = \overline{TF}(W-B) - M\sin(\theta-\tau) \quad \text{(E-11)}$$

**E.3.2** The formula 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, equalizer, pitman, crank, and crank counterweights; and neglects friction in the saddle, tail, and pitman bearings. For units having 100-percent crank counterbalance and where crank-speed variation is not more than 15 percent of average, these factors usually can be neglected without introducing errors greater than 10 percent. 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.3.11, E.3.12, and E.3.13.

**E.3.3** Torque factors may be used to obtain the affect 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 $\theta$. The result is the rotary counterbalance effect, in pounds, at the polished rod.

**E.3.4** Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the crankpins in the 90-degree 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 ($W$) 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:

$$M = \frac{\overline{TF}(W-B)}{\sin(90 \text{ degrees}+\tau)} \quad \text{(E-12)}$$

**E.3.5** To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 5954-foot 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 degrees of crankpin angle $\theta$. 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 date, vertical lines representing each 15 degrees of crankpin angle $\theta$ are projected upward to intersect the dynamometer card. Then the polished-rod load may be determined for each 15 degrees of crankpin angle $\theta$.

**E.3.6** To further illustrate, a calculation will be made considering the point where the crankpin angle $\theta$ equals 120 degrees. From polished-rod stroke and torque factor data for the particular 86-inch stroke 114-D pumping unit used for this example, it is found that the position of the polished rod at 120 degrees is 0.629 and that the torque factor $\overline{TF}$ is 35.446. 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 E-2). The dynamometer deflection at this point is read to be 1.672 inches, which, with a scale constant of 5000 pounds per inch, makes the load ($W$) at that point 8360 pounds.

**E.3.7** In a similar manner, the polished-rod load may be obtained for each 15-degree angle of crankpin rotation. The dynamometer card has been marked to show the load and position for each 15 degrees of crankpin angle. The structural unbalance, $B$, for the example unit equals +231 pounds. Therefore, the net polished-rod load, $W_n$ at $\theta = 120$ degrees = W−B = 8360 − (+231) = 8129 pounds. The torque, $T_{wn}$, due to the net polished-rod load = $TF \times W_n$ = 35.446 × 8129 = 288,140 in.-lb.

**E.3.8** To find the torque $T_r$, due to the crank counterbalance, the maximum moment, $M$, must be determined. This may be done either from manufacturers' counterbalance tables or curves, or as described in E.3.4. 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.

**E.3.9** The horizontal dotted line drawn across the dynamometer card in Figure E-2 is the counterbalance effect measured with the dynamometer at the 90-degree crankpin angle and is 7000 pounds. The maximum moment can then be calculated as follows, using Equation E-12:

$$M = \frac{\overline{TF}(W-B)}{\sin\,(90 \text{ degrees} + \tau)}$$

$$= \frac{39.575 \times [7000-(+231)]}{\sin\,[90 \text{ degrees} + (-14 \text{ degrees})]} = 276{,}084 \text{ in.-lbs}$$

(The torque factor of 39.575 is the value at the 90-degree crankpin position and angle $\tau$ is −14 degrees for the example unit.)

**E.3.10** The torque, $T_r$, due to the counterbalance at the 120-degree crankpin position would therefore be equal to 276,084 × sin [120 degrees +(−14 degrees)] = 276,084 × 0.961 = 265,389 in.-lbs. The net torque at the crankshaft for the 120-degree crankpin position would then be calculated from Equation E-11 as follows:

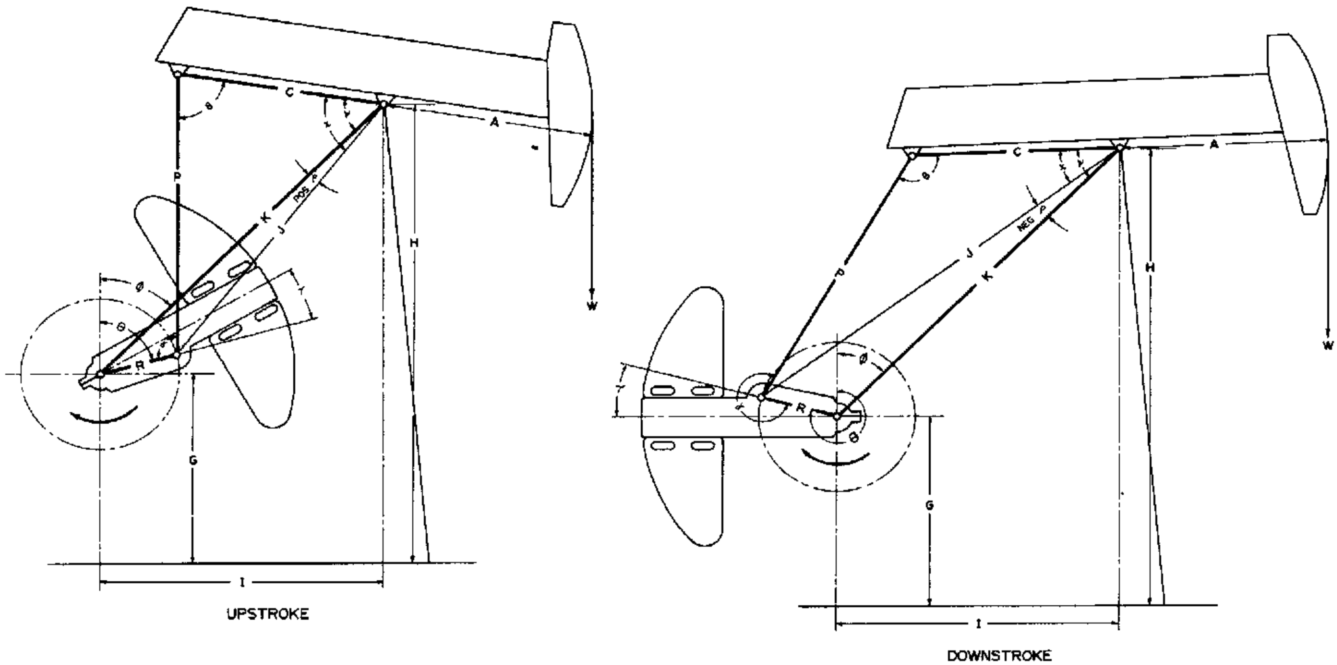
$$\begin{aligned} T_n &= \overline{TF}\,(W-B) - M \sin\,(\theta + \tau) \\ &= T_{wn} - T_r \\ &= 288{,}140 - 265{,}389 = 22{,}751 \text{ in.-lb} \end{aligned}$$

These values may be calculated for other crankpin angle positions in the same manner as outlined above. Shown in Figure E-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.

**E.3.11** The foregoing technique is generally accepted. Those wanting more precise results must realize the true stroke length can vary somewhat with a change in beam position in relation to the centerline of the saddle bearing due to an adjustable feature 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.3.12** The geometry of the utilized dynamometer can influence the determination of instantaneous load values for the various specified or selected crankpin angles. When critical calculations are to be made, the dynamometer manufacturer should be contacted for information on the involved performance characteristics of his dynamometer and the procedures that should be followed to adjust the recorded card when completely accurate data are required.

**E.3.13** It must be recognized that the maximum and minimum loads will most frequently fall at points other than the 15-degree divisions for which torque factors are provided. Interpolation between 15-degree divisions is permissible without significant error.




Note: See E.2.3 for definition of symbols.

Figure E-1—Rear Mounted Geometry Class I Lever System With Phased Crank Counterbalance

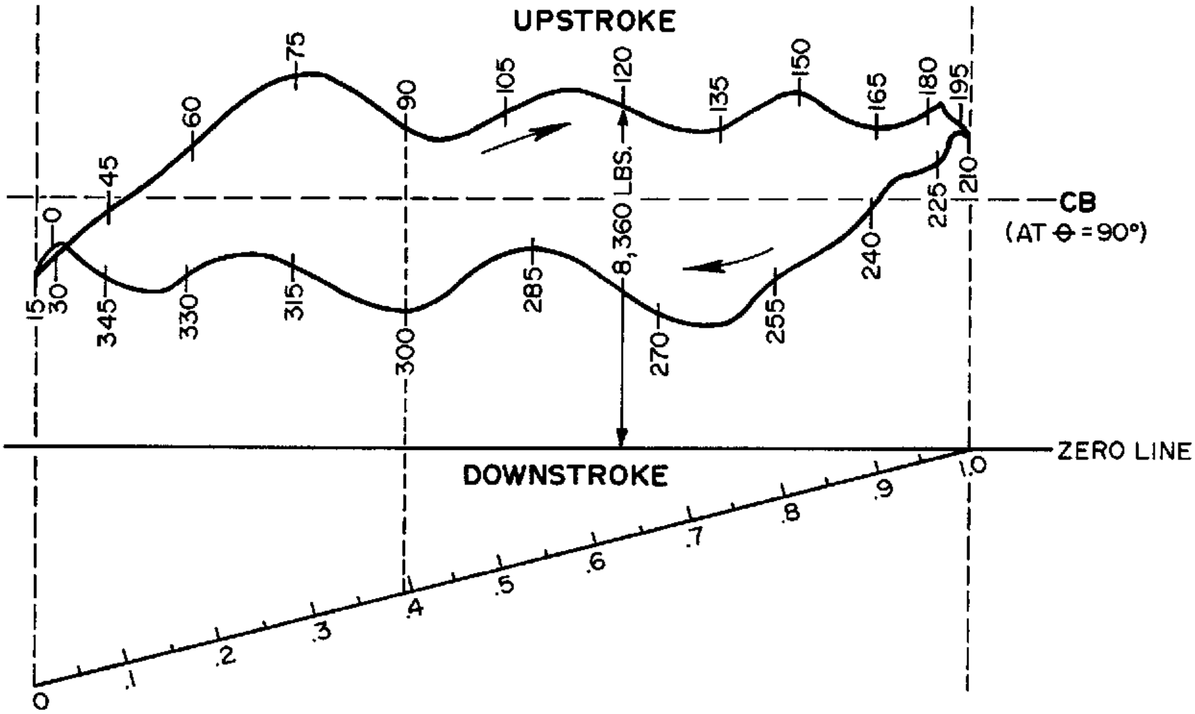



Figure E-2—Division of Dynamometer Card by Crankpin Angle Using API Polished-Rod Position Data

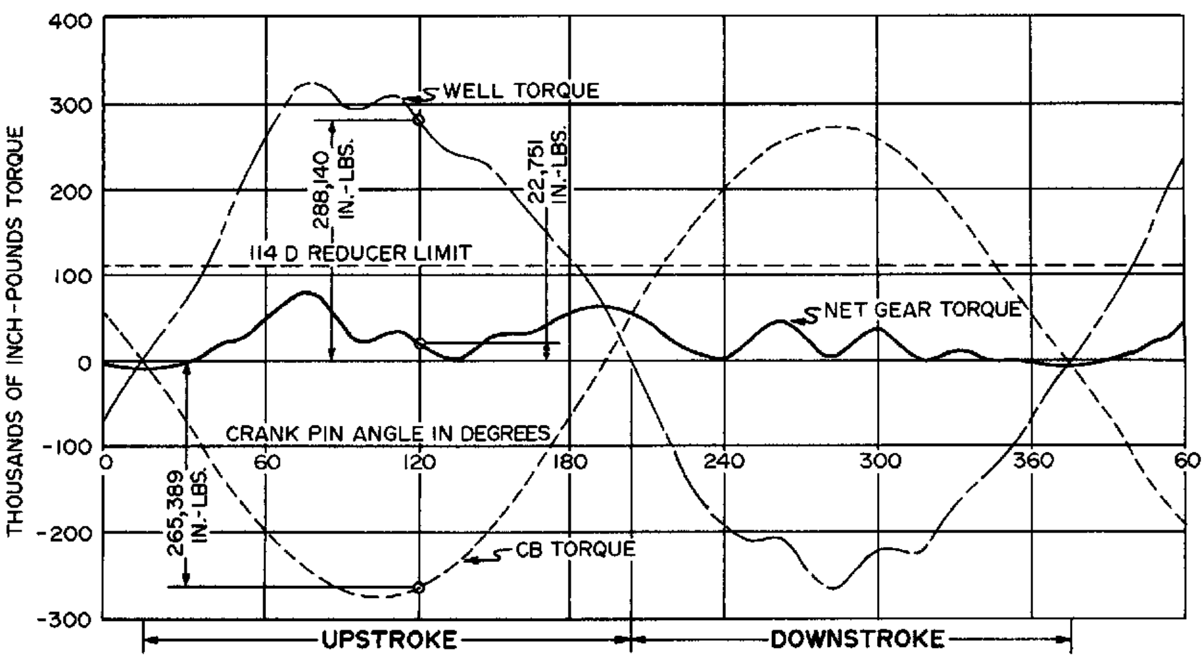



Figure E-3—Torque Curves Using API Torque Factors

# NET REDUCER TORQUE CALCULATION SHEET
# (REAR MOUNTED GEOMETRY CLASS I LEVER SYSTEMS WITH PHASED CRANK COUNTERBALANCE–CLOCKWISE ROTATION ONLY)

Well No.: ____________________ Company: ____________________

Unit Size: ____________________ Location: ____________________

| $\theta$ | Sin $(\theta+\tau)$ | $W$ | $B$ | $W-B$ | $\overline{TF}$ | $\overline{TF}\,(W-B)$ | $M$ [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 | | | | | | | | |

Note:

$T_n = \overline{TF}\,(W-B) - M \sin(\theta+\tau)$

*Where:*

$T_n$ = Net reducer torque, in.-lbs.
$\theta$ = Position of crankpin.
$M$ = Maximum moment of counterbalance, in.-lbs.
$W$ = Measured polished rod load at $\theta$, lbs.
$B$ = Unit structural unbalance, lbs.
$\overline{TF}$ = Torque factor at $\theta$, in.

$CB$ at 90-degree crankpin angle = ____________

$$M = \frac{(CB \text{ at 90 degrees} - B)\,(\overline{TF} \text{ at 90 degrees})}{\sin(90 \text{ degrees} + \tau)} = \_\_\_\_\_\_\_\_$$

$\tau$ = Angle of crank counterweight arm offset (negative when weights are counterclockwise relative to crankpin bearings) = ____________

# APPENDIX F—RECOMMENDED PRACTICE FOR CALCULATING TORQUE RATINGS FOR PUMPING UNIT GEAR REDUCERS

## F.1 Illustrative Example, Pitting Resistance

Calculate the allowable transmitted torque at the output shaft based on the pitting resistance for the following first reduction helical gear set. The pinion speed is 588 revolutions per minute (rpm), and the reducer output speed is 20 rpm.

*Gear set data:*

$d$ = 3.167 inches
$D$ = 16.833 inches
$N_P$ = 19
$N_G$ = 101
$P_d$ = 6.0
$F$ = 3 inches
$\phi_n$ = 17.4952 degrees
$\psi$ = 30 degrees
$n_p$ = 588 rpm
$N_o$ = 20 rpm
Minimum pinion hardness = 340 BHN (steel)
Minimum gear hardness = 300 BHN (steel)

Determine pitting resistance torque rating as follows:

a. From Equation 5:

$$v_t = \frac{\pi(3.167)(588)}{12} = 487.5 \text{ ft/min}$$

b. From Equation 4:

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

c. From Equation 3:

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

$$C_m = 1.33 \text{ (see Figure 3)}$$

d. From Equation 6:

$$C_2 = \frac{3}{1.33} = 2.25$$

$s_{ac}$ = 129,100 pounds per square inch (see Figure 3)

e. From Equation 7:

$$C_3 = (0.225)\frac{5.316}{5.316+1}\left(\frac{129,100}{2300}\right)^2 = 597$$

f. From Equation 2:

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

Note: The pitting resistance rating of this gear set is 154,300 in.-lb. The final rating will be the lowest calculated value of pitting resistance rating and bending strength ratings as determined in Equations 2 and 15 of this specification, but not to exceed one of the standard pumping unit reducer sizes listed in Table 4.

## F.2 Illustrative Example, Bending Strength

Calculate the allowable transmitted torque at the output shaft based on bending strength for the following first reduction helical (or double helical) gear set. 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.

### F.2.1 PINION

Determine strength numbers for pinion as follows:

a. From Equation 5:

$$v_t = \frac{\pi(3.167)(588)}{12} = 487.5 \text{ ft/min}$$

b. From Equation 17:

$$K_5 = \sqrt{\frac{78}{78+\sqrt{487.9}}} = 0.883$$

c. From Equation 16:

$$K_1 = \frac{(588)(3.167)(0.883)}{2(20)} = 41.11$$

$$K_m = 1.22 \text{ (see Figure 6)}$$

d. From Equation 19:

$$K_2 = \frac{3}{1.22} = 2.46$$

$K_3$ = 33,250 (see Figure 5)

$J$ = 0.437 calculated per Appendix A in AGMA 422.03 and as recorded on Figure 14

e. From Equation 21:

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

f. From Equation 15:

$$T_{at} = (41.11)(2.46)(33,250)(0.0728) = 244,800 \text{ in.-lbs (pinion)}$$

### F.2.2 GEAR

Determine bending strength torque rating for gear as follows:

a. From Equation 21:

$$K_I = 41.11$$

b. From Equation 22:

$K_2$ = 2.46
$K_3$ = 30,900 (see Figure 5)
$J$ = 0.387 calculated per Appendix A in AGMA 422.03 and as recorded on Figure 14

c. From Equation 21:

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

d. From Equation 15:

$$T_{at} = (41.11)(2.46)(30{,}900)(.0645)$$
$$= 201{,}560 \text{ in.-lbs (gear)}$$

Note: The calculated bending strength torque rating of this gear set is 201,560 in.-lbs, the lower value of the bending strength ratings for the pinion and the gear. The calculated pitting resistance torque rating is 154,300 in.-lbs (see previous example). The next smaller torque rating shown in Table 4 is 114,000; therefore, 114,000 is the stated (nameplate) peak torque rating as far as this gear set is concerned.

## F.3 Illustrative Example, Static Torque

Calculate the allowable static torque rating based on bending strength for the first reduction helical (or double helical) gear set. The pinion speed is 588 rpm.

Note: This is the same gear set used in the pitting resistance calculation example and the bending strength calculation example.

*Gear set data:*

$D$ = 16.833 inches
$J$ = 0.387
$m_{G2}$ = 5.53
$P_d$ = 6
$F$ = 3 inches
$s_{ay}$ = 112,000 psi (see Figure 11)
$K_y$ = 1.0 (see Table 8)
$K_{ms}$ = .0144 (3) + 1.07 = 1.113 (see Equation 25)

a. From Equation 23:

$$T_{as1} = \frac{16.833}{2} \times \frac{0.387}{6} \times \frac{3}{1.113}(112{,}000)(1.0)$$
$$= 163{,}880 \text{ in.-lbs at the high speed gear}$$

The allowable static torque at the output shaft would be the value calculated above (163,880 in.-lbs) multiplied by the ratio to the output gear set.

b. From Equation 24:

$$T_{as2} = (T_{as1})(M_{G2})$$
$$= (163{,}800)(5.53)$$
$$= 906{,}260 \text{ in.-lbs at output shaft}$$

The value is for the high speed gear only and must 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 must be equal to or greater than 500 percent of the applicable nameplate rating recorded in Table 4. In this example the nameplate rating as far as the first reduction is concerned is 114,000 in.-lbs (see bending strength calculation above). The static torque rating must therefore be equal to or greater than 5 × 114,000 = 570,000 in.-lbs. The calculated static torque rating of 906,260 in.-lbs satisfies this condition and is the static torque rating as far as the first reduction set is concerned.

# APPENDIX G—SI UNITS

Note: This appendix is not part of API Specification 11E.

The conversion of English units shall be made in accordance with ISO 31-3.

Table G-1—SI Units

| Quantity | U.S. Customary Unit | SI Unit |
|---|---|---|
| Area | 1 square inch (in.$^2$) | 645.16 square millimeters (mm$^2$) (exactly) |
| Flow rate | 1 barrel per day (bbl/d) | 0.158987 cubic meters per day (m$^3$/d) |
| | 1 cubic foot per minute (ft$^3$/min) | 0.02831685 cubic meters per minute (m$^3$/min) or 40.776192 cubic meters per day (m$^3$/d) |
| Force | 1 pound-force (lbf) | 4.448222 newtons (N) |
| Impact energy | 1 foot pound-force (ft•lbf) | 1.355818 Joules (J) |
| Length | 1 inch (in.) | 25.4 millimeters (mm) (exactly) |
| | 1 foot (ft) | 304.8 millimeters (mm) (exactly) |
| Mass | 1 pound (lb) | 0.45359237 kilograms (kg) (exactly) |
| Pressure | 1 pound-force per square inch (lbf/in.$^2$) or 1 pound per square inch (psi) (Note: 1 bar = $10^5$ Pa) | 6894.757 pascals (Pa) |
| Strength or stress | 1 pound-force per square inch (lbf/in.$^2$) | 6894.757 pascals (Pa) |
| Temperature | The following formula was used to convert degrees Fahrenheit (°F) to degrees Celsius (°C): | °C = 5/9 (°F – 32) |
| Torque | 1 inch pound-force (in•lbf) | 0.112985 newton meters (N•m) |
| | 1 foot pound-force (ft•lbf) | 1.355818 newton meters (N•m) |
| Velocity | 1 foot per second (ft/s) | 0.3048 meters per second (m/s) (exactly) |
| Volume | 1 cubic inch (in.$^3$) | 16.387064•$10^{-3}$ cubic decimeters (dm$^3$) (exactly) |
| | 1 cubic foot (ft$^3$) | 0.0283168 cubic meters (m$^3$) or 28.3168 cubic decimeters (dm$^3$) |
| | 1 gallon (U.S.) | 0.0037854 cubic meters (m$^3$) or 3.7854 cubic decimeters (dm$^3$) |
| | 1 barrel (U.S.) | 0.158987 cubic meters (m$^3$) or 158.987 cubic decimeters (dm$^3$) |

# APPENDIX H—USE OF API MONOGRAM

(Note: This appendix is not part of ISO 10431:1993.)

The API monogram is a registered trademark of the American Petroleum Institute.

Manufacturers desiring to warrant that articles manufactured or sold by them conform with this specification shall obtain the license to use the Official API Monogram.

The original resolutions adopted by the Board of Directors of the American Petroleum Institute on October 20, 1924, embodied the purpose and conditions under which such official monogram may be used.

The following restatement of the resolution was adopted by the Board of Directors on November 14, 1977:

WHEREAS, The Board of Directors of the American Petroleum Institute has caused a review of the Institute's program for licensing the use of the API monogram and

WHEREAS, It now appears desirable to restate and clarify such licensing policy and to confirm and make explicitly clear that it is the licensees, not API, who make the representation and warranty that the equipment or material on which they have affixed the API monogram meets the applicable standards and specifications prescribed by the Institute;

NOW, THEREFORE, BE IT RESOLVED, That the purpose of the voluntary Standardization Program and the Monogram Program of the American Petroleum Institute is to establish a procedure by which purchasers of petroleum equipment and materials as are represented and warranted by the manufacturers thereof to conform to applicable standards and specifications of the American Petroleum Institute; and be it further

RESOLVED, That the previous action under which the following monogram was adopted as the official monogram of the American Petroleum Institute is reaffirmed;

BE IT FURTHER RESOLVED, That the American Petroleum Institute's monogram and standardization programs have been beneficial to the general public as well as the petroleum industry and should be continued and the Secretary is hereby authorized to license the use of the monogram to anyone desiring to do so under such terms and conditions as may be authorized by the Board of Directors of the American Petroleum Institute, provided that the licensee shall agree that the use of the monogram by such licensee shall constitute the licensee's representation and warranty that equipment and materials bearing such monogram complies with the applicable standards and specifications of the American Petroleum Institute; and that licensee shall affix the monogram in the following manner:

BE IT FURTHER RESOLVED, That the words "Official Publication" shall be incorporated with said monogram on all such standards and specifications that may hereafter be adopted and published by the American Petroleum Institute, as follows:

**OFFICIAL PUBLICATION**

**REG. U.S. PATENT OFFICE**

## H.1 API Monogram

The API monogram – – is a registered trademark/servicemark of the American Petroleum Institute. Authorization to use the monogram is granted by the Institute to qualified licensees for use as a warranty that they have obtained a valid license to use the monogram and that each individual item which bears the monogram conformed, in every detail, with the API specification applicable at the time of manufacture. However, the American Petroleum Institute does not represent, warrant or guarantee that products bearing the API monogram do in fact conform to the applicable API standard or specification. Such authorization does not include use of the monogram on letterheads or in advertising without the express statement of fact describing the scope of licensee's authorization and further does not include use of the monogram, the name AMERICAN PETROLEUM INSTITUTE or the description "API" in any advertising or otherwise to indicate API approval or endorsement of products.

The formulation and publication of API specifications and the API monogram program is not intended in any way to inhibit the purchase of products from companies not licensed to use the API monogram.

## H.2 Application for Authority to Use Monogram

Manufacturers desiring to warrant that products manufactured by them comply with the requirements of a given API specification may apply for a license to use the monogram with forms provided in an appendix to each specification.

The "Agreement" form must be submitted in duplicate for each specification under which monogram rights are desired. One "Statement of Manufacturer's Qualifications" is required for each facility.

A manufacturer desiring to apply the monogram at more than one facility (a facility is any manufacturing location) must submit a separate application for each facility.

Applicants shall have an approved functioning quality program in conformance with API Specification Q1 prior to being issued a license to use the API monogram.

## H.3 Authorization to Use the Monogram

A decision to award or withhold monogram rights will be made by the staff of the Institute. A survey of the applicant's facilities will be made by an approved Institute surveyor prior to a decision to approve or withhold the license. The basis of the survey shall be the appropriate product specification and all applicable portions of API Specification Q1.

For a manufacturer having more than one facility (plant), each facility will be judged separately and if determined to be eligible for authorization to use the monogram will be granted a separate license for each specification, or part thereof, under which authorization is granted. The application of the monogram may not be subcontracted.

## H.4 Fee for Use of Monogram

### H.4.1 INITIAL AUTHORIZATION FEE

The applicant will be invoiced an initial authorization fee for the first specification included in the application, and a separate fee for each additional specification included in the application. The applicant will also be invoiced for the surveyor's fee.

### H.4.2 ANNUAL RENEWAL FEE

In addition to the initial authorization fee, licensees will be assessed an annual renewal fee for each specification under which he is authorized to use the monogram. Applicants issued monogram certificates dated November 1 through December 31 shall not be required to pay a renewal fee for the following year.

The fees assessed are to defray the cost of the Monogram Program.

## H.5 Periodic Surveys

Existing licensees must be periodically surveyed by an approved Institute surveyor to determine whether or not they continue to qualify for authorization to use the monogram. The frequency of the periodic surveys will be at the discretion of the staff of the Institute. The surveyor's fee and expenses for making a periodic survey will be paid by the Institute.

## H.6 Cancellation of Monogram Rights

The right to use the monogram is subject to cancellation for the following causes:

a. Applying the monogram on any product that does not meet the specification.
b. Failure to maintain reference master gauges in accordance with the specifications.
c. Failure to meet the requirements of any resurvey.
d. Failure to pay the annual renewal fee for use of the monogram.
e. For any other reason satisfactory to the Executive Committee on Standardization of Oil Field Equipment and Materials.

## H.7 Reinstatement of Monogram Rights

Manufacturers whose authorization to use the monogram has been cancelled may request reinstatement at any time. If a request for reinstatement is made within sixty (60) days after cancellation, and if the reason for cancellation has been corrected, no new application is necessary. A resurvey of the manufacturer's facilities will be made by an approved Institute surveyor prior to a decision to reinstate monogram rights. The manufacturer will be invoiced for this resurvey regardless of the Institute's decision on reinstatement. If the resurvey indicates that the manufacturer is qualified, the license will be reissued.

Request for reinstatement made more than sixty (60) days after cancellation shall be treated as a new application unless the circumstances dictate an extension of this time period as agreed upon by the API staff.

## H.8 Appeals

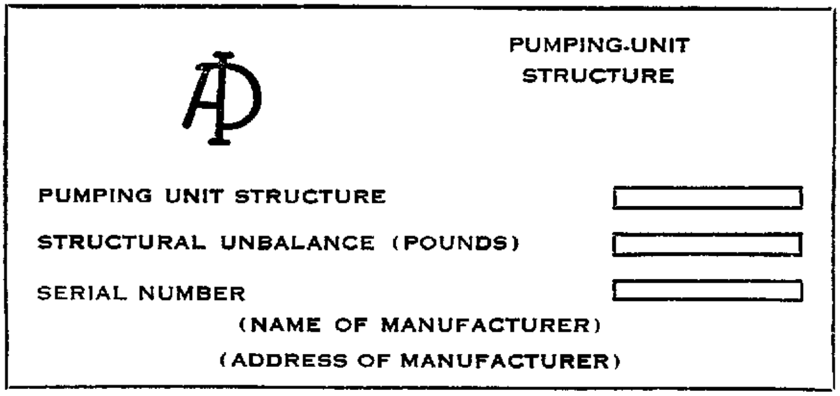
An interested party may appeal a decision by the API staff to withhold monogram rights. Appeals shall be directed to the Director, API Exploration and Production Department and handled by the General Committee of the Exploration and Production Department with a further right of appeal to the API Management Committee. Competing suppliers or manufacturers of the product or service to which the standard applies or might apply may not be involved in appeals. The General Committee and the Management Committee may convene appeals boards to hear and act on appeals.

## H.9 Marking

The following marking requirements apply to licensed manufacturers using the API monogram on products covered by this specification.

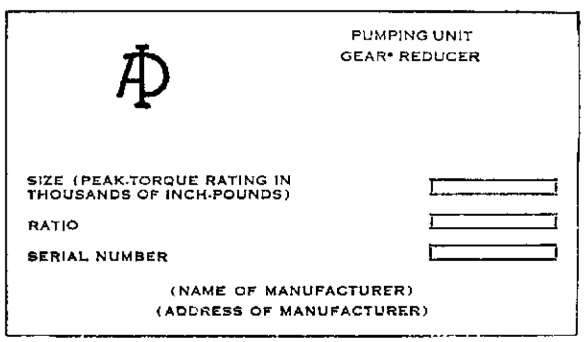
**H.9.1** Each pumping unit structure shall be provided with a nameplate substantially as shown in Figure H-1, except that the API monogram may be applied only by authorized manufacturers. At the discretion of the manufacturer, the nameplate may contain other nonconflicting and appropriate information, such as model number or lubrication instructions.

**H.9.2** Each pumping unit reducer shall be provided with a nameplate substantially as shown in Figure H-2, except that the API monogram may be applied only by authorized manufacturers. The size (peak torque rating in 1000 in.-lbs) shown on the nameplate shall be one of those listed in Table 4. No other rating marking shall be applied to the reducer. The nameplate may, at the option of the manufacturer, contain information such as model number, lubrication instructions, etc., provided such marking does not conflict with the API rating marking.




Note: Structural unbalance is that force in pounds required at the polished-rod to hold the beam in a horizontal position with the pitmans disconnected from the crankpins. This structural unbalance is 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 this value is negative.

Figure H-1—Pumping Unit Structure Nameplate




*Substitute "CHAIN" when appropriate.

Figure H-2—Pumping Unit Reducer Nameplate

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