INTERNATIONAL STANDARD

ISO 10431

First edition
1993-12-15

Petroleum and natural gas industries — Pumping units — Specification

Industries du pétrole et du gaz naturel — Unités de pompage — Spécifications

Reference number
ISO 10431:1993(E)

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

International Standard ISO 10431 was prepared by the American Petroleum Institute (API) (as Spec 11E, 16th edition) and was adopted, under a special "fast-track procedure", by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, in parallel with its approval by the ISO member bodies.

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Introduction

International Standard ISO 10431:1993 reproduces the content of API Spec 11E, 16th edition, 1989 and its supplement 1 (July 1, 1991). ISO, in endorsing this API document, recognizes that in certain respects the latter does not comply with all current ISO rules on the presentation and content of an International Standard. Therefore, the relevant technical body, within ISO/TC 67, will review ISO 10431:1993 and reissue it, when practicable, in a form complying with these rules.

This standard is not intended to obviate the need for sound engineering judgement as to when and where this standard should be utilized and users of this standard should be aware that additional or differing requirements may be needed to meet the needs for the particular service intended.

Standards referenced herein may be replaced by other international or national standards that can be shown to meet or exceed the requirements of the referenced standards.

Appendix G to this document shall not be considered as requirements.

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Petroleum and natural gas industries — Pumping units — Specification

1 Scope

This International Standard lays down specification covering the design and rating of pumping units.

2 Requirements

Requirements are specified in:

"API Specification 11E (Spec 11E), Sixteenth Edition, October 1, 1989 — *Specification for Pumping Units*",

which is adopted as ISO 10431.

For the purposes of internationl standardization, however, modifications shall apply to specific clauses and paragraphs of publication API Spec 11E. These modifications are outlined below.

Throughout publication API Spec 11E, the conversion of English units shall be made in accordance with ISO 31, parts 1 and 3. In particular,

LENGTH	1 inch (in) 1 foot (ft)	= 25,4 mm (exactly) = 304,8 mm (exactly)
MASS	1 pound (lb)	= 0,453 592 37 kg (exactly)
PRESSURE	1 pound-force per square inch (lbf/in^2) or 1 psi	= 6 894,76 Pa
VOLUME	1 cubic inch (in^3)	= 16,387 064·10^{-3} dm^3 (exactly)
AREA	1 square inch (in^2)	= 645,16 mm^2 (exactly)
VELOCITY	1 foot per second (ft/s)	= 0,304 8 m/s (exactly)
TORQUE	1 inch pound-force (in·lbf)	= 0,112 985 N·m

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Information given in the POLICY is relevant to the API publication only.

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Appendix G

Information relating to the use of API monogram is relevant to the API publication only.

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Specification for Pumping Units

API SPECIFICATION 11E (SPEC 11E)
SIXTEENTH EDITION, OCTOBER 1, 1989

American Petroleum Institute
1220 L Street, Northwest
Washington, DC 20005

Supplement 1
(July 1, 1991)

Specification for Pumping Units

API SPECIFICATION 11E (SPEC 11E)
SIXTEENTH EDITION, OCTOBER 1, 1989

American Petroleum Institute
1220 L Street, Northwest
Washington, DC 20005

Issued by
AMERICAN PETROLEUM INSTITUTE
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Users of this publication should become completely familiar with its scope and content, including any provisions it may have regarding marking of manufactured products. This document is intended to supplement rather than replace individual engineering judgment.

OFFICIAL PUBLICATION

REG. U.S. PATENT OFFICE

Foreword

This supplement contains revisions authorized at the 1990 Standardization Conference as reported in Circ PS-1920 and approved by letter ballot.

Page 5. *Replace Par. 2.4 and Fig. 2.1 with the following:*

2.4 Walking Beam. The following formula shall be used for rating conventional walking beams as shown in Fig. 2.1.

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

Wherein:

W = walking-beam rating in pounds of polished-rod load.

f_{cb} = compressive stress in bending in pounds per square inch. See Table 2.1 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 Fig. 2.1.

A = distance from centerline of saddle bearing to centerline of well in inches. See Fig. 2.1.

C = distance from centerline of saddle bearing to centerline of equalizer bearing in inches. See Fig. 2.1.

Page 6. *Replace the equation in Column 4, Row 2 of Table 2.1 with:*

$$* \quad \frac{\sqrt{E\, I_y\, G\, J}}{S_x l}$$

Delete all the nomenclature at the end of Table 2.1 and replace it with:

* Where:

J = Torsional constant, in^4

l = Longest laterally, unbraced length of beam, inches (longer of A or C (See Fig. 2.1)).

E =Modulus of elasticity; 29,000,000 psi.

I_y =Weak axis moment of inertia, in^4.

G =Shear modulus; 11,200,000 psi.

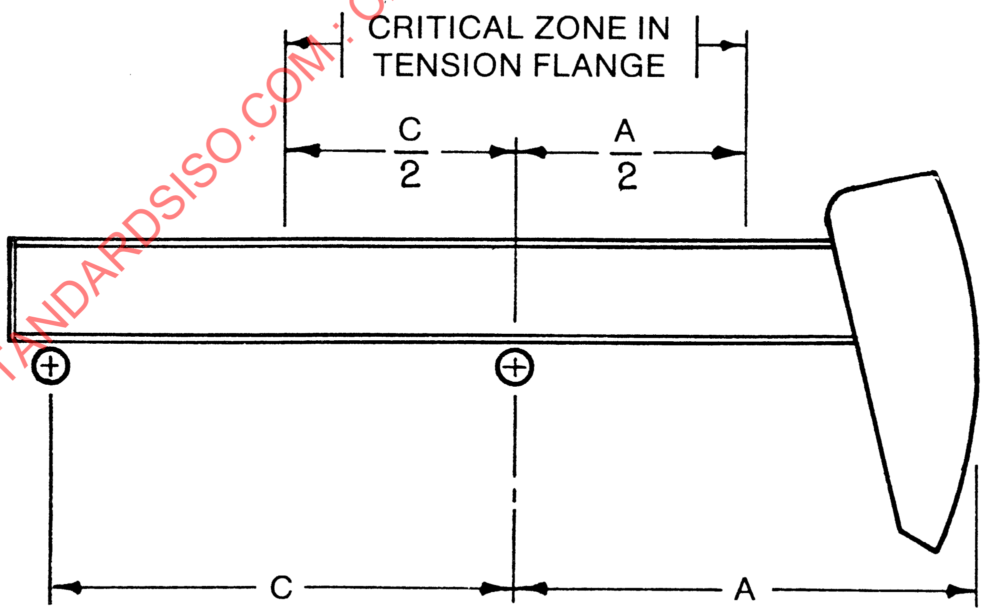

FIG. 2.1
WALKING-BEAM ELEMENTS

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Note

This edition supersedes the fifteenth edition of Spec 11E. It includes changes adopted at the 1988 Standardization Conference as reported in Circ PS-1858 and subsequently passed by letter ballot.

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API SPECIFICATION FOR PUMPING UNITS

Foreword

a. This specification is under the jurisdiction of the Committee on Standardization of Production Equipment.

b. 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.

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

d. The following nomenclature is standard:

1. **Pumping unit.**
2. **Pumping-unit structure.**
3. **Pumping-unit gear reducer.**
4. **Pumping-unit chain reducer.**
5. **Pumping-unit beam counterbalance.**
6. **Pumping-unit crank counterbalance.**

e. **Attention Users of this Publication:** Portions of this publication have been changed from the previous edition. The location of changes has been marked with a bar in the margin. In some cases the changes are significant, while in other cases the changes reflect minor editorial adjustments. The bar notations in the margins are provided as an aid to users to identify those parts of this publication that have been changed from the previous edition, but API makes no warranty as to the accuracy of such bar notations.

SECTION 1
SCOPE

1.1 This specification covers the design and rating of the following:

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

1.2 American Petroleum Institute (API) Specifications are published as aids to the procurement of standardized equipment and materials, as well as instructions to manufacturers of equipment or materials covered by an API Specification. These Specifications are not intended to obviate the need for sound engineering, nor to inhibit in any way anyone from purchasing or producing products to other specifications.

1.3 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.

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1.5 Any manufacturer producing equipment or materials represented as conforming with an API Specification is responsible for complying with all the provisions of that Specification. The American Petroleum Institute does not represent, warrant or guarantee that such products do in fact conform to the applicable API standard or specification.

SECTION 2
PUMPING-UNIT STRUCTURES

2.1 Scope. This section covers:

a. Standardization of specific structure sizes in combination with established reducer sizes as given in Section 3.

b. Walking beam design, with specific rating formula.

c. Design loads and limiting working stresses on other structural components are also included.

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.*

2.2 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.

2.3 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 2.2, 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.

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.***

2.4 Walking Beam. **The following formula shall be used for rating conventional walking beams as shown in Fig. 2.1.**

$$W = \frac{f_{cb}S}{L}$$

Wherein:

W = walking-beam rating in pounds of polished-rod load.

L = greater of l_f or l_r

f_{cb} = compressive stress in bending in pounds per square inch. See Table 2.1 for maximum allowable stress.

S = 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 Fig. 2.1.

l_f = distance from centerline of saddle bearing to centerline of well in inches. See Fig. 2.1.

l_r = distance from centerline of saddle bearing to centerline of equalizer bearing in inches. See Fig. 2.1.

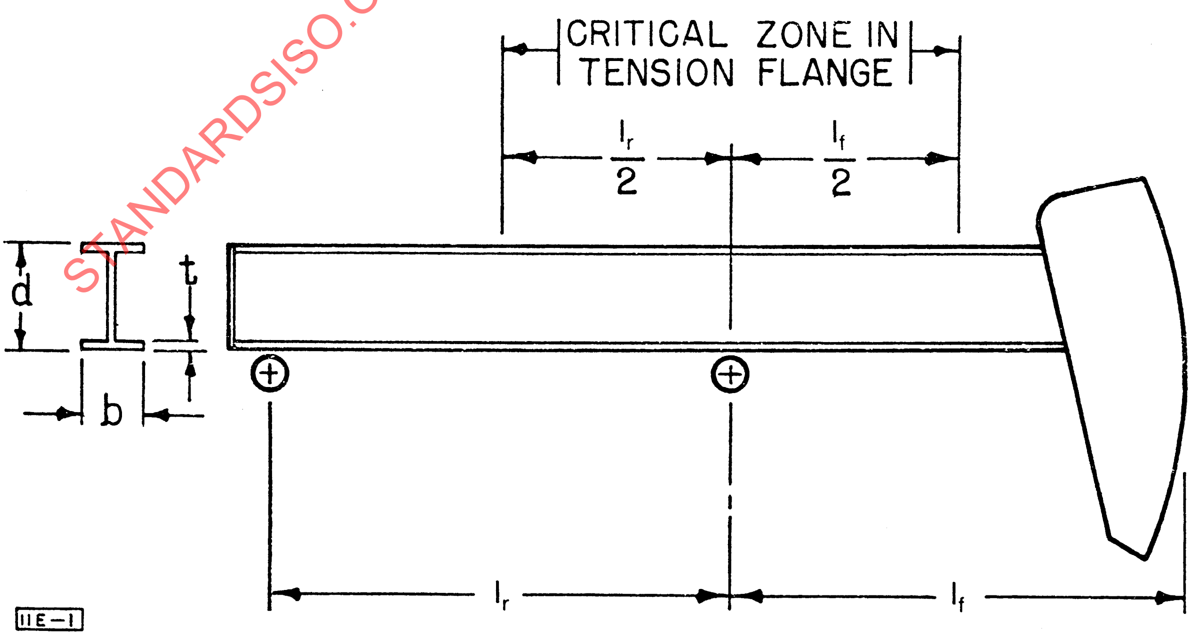


FIG. 2.1
WALKING-BEAM ELEMENTS

TABLE 2.1
MAXIMUM ALLOWABLE STRESSES IN PUMPING-UNIT WALKING BEAMS
(See Fig. 2.1)

1	2	3	4
	Stress	Symbol	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}	$^{*}\dfrac{6{,}000{,}000}{\frac{ld}{bt}}$
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

*In the quantity $\dfrac{l\ d}{b\ t}$:

l = longest laterally, unbraced length of beam, inches (longer of l_r or l_f, see Fig. 2.1).
d = depth of beam section, inches.
b = width of compression flange, inches.
t = thickness of compression flange, inches.

NOTE: *The formula given in Par. 2.4 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.*

2.5 The working stress, f_{cb} for the beam rating formula given in Par. 2.4, shall be determined from Table 2.1. 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 2.1.

2.6 **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 counter-clockwise.

2.7 **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° crank position on the upstroke of the pumping unit. Use polished rod load W, for all upstroke crank positions. (See Par. 2.1c)

For units with bi-directional rotation and non-symmetrical torque factors, the direction of rotation for

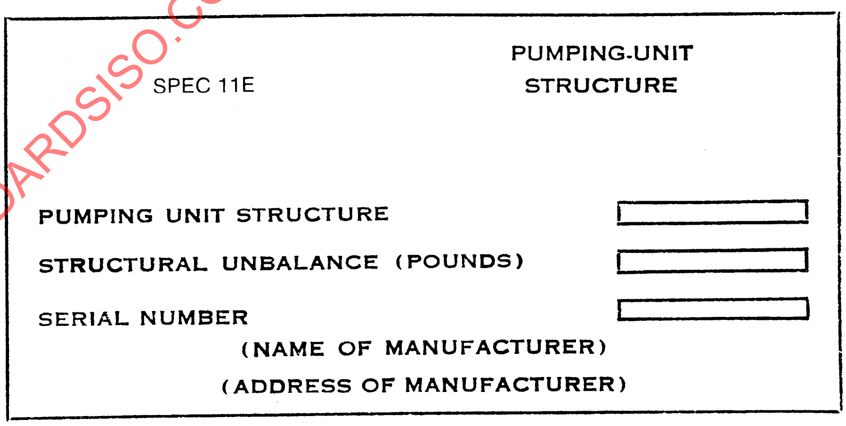


FIG. 2.2
PUMPING-UNIT STRUCTURE NAME PLATE

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 name plate when this value is negative.*

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.*

2.8 Design Stresses for All Structural Members Except Walking Beam, Bearing Shafts and Cranks.

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

b. Components subjected to simple tension or compression and non-reversing bending shall have a limiting stress of .3 Sy. If stress risers occur in critical zones of tension members, the limiting stress shall be .25 Sy.

c. Components subjected to reverse bending shall have a limiting stress of .2 Sy.

d. The following formula shall be used for all components acting as columns:

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

Wherein:

W_2 = maximum applied load on column, lbs.

a = area of cross section, sq. in.

Sy = yield strength of material, psi

n = end restraint constant (assume = 1)

E = modulus of elasticity, psi

l = unbraced length of column, in.

r = radius of gyration of section, in.

$\frac{l}{r}$ 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 Par. 2.8b).

2.9 Shafting. All bearing shafts as well as other structural shafting shall have limiting stresses as outlined in Par. 3.8 in the reducer section of this specification.

2.10 Hanger. Wirelines 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 Par. 2.8b and 2.8c.

2.11 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 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 counterweights should be securely fixed in a stationary position by chaining or other acceptable means.

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

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.

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

±½ in. for stroke lengths to 100 in.
±⅝ in. for stroke lengths 100 in. to 200 in.
±¾ in. for stroke lengths of 200 in. and longer

2.13 Cranks. All combined stresses in cranks shall be limited to a maximum value of .15 Sy.

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

a. **Anti-Friction Bearings.**

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

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

Where:

R_1 = bearing load ratio
k = 1 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.

TABLE 2.2
PUMPING UNIT SIZE RATINGS

1	2	3	4
Pumping Unit Size	Reducer Rating, in.-lb	Structure Capacity, lb	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

b. 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, which are comparable in performance to anti-friction bearings designed for the same operating loads and speeds.

2.15 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.

2.16 Marking.* Each pumping-unit structure shall be provided with a name plate substantially as shown in Fig. 2.2. At the discretion of the manufacturer, the name plate may contain other non-conflicting and appropriate information, such as model number or lubrication instructions.

2.17 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-deg position of the crank. An approved form for the submission of these data is shown in Appendix A.

*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, herein. No other use of the monogram is permitted.

SECTION 3
PUMPING-UNIT REDUCERS

3.1 SCOPE

Applicability. This Specification is applicable to enclosed speed reducers wherein the involute gear tooth designs include helical and herringbone gearing. This Specification is intended primarily for beam-type pumping units.

Limitations. The rating methods and influences identified in this Specification are limited to single and multiple stage designs applied to oilfield 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.

3.2 RESPONSIBILITY

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 oilfield conditions.

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

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.

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.

TABLE 3.1
SYMBOLS USED IN GEAR RATING EQUATIONS

Symbol	Description	Reference
A	Tensile Area of Fastener, in.^{2}	3.8
C_m	Load Distribution Factor for Pitting Resistance	3.5
C_p	Elastic Coefficient	3.5
C_1	Pitting Velocity Number	3.5
C_2	Pitting Contact Number	3.5
C_3	Pitting Stress Number	3.5
C_5	Velocity Factor for Pitting Resistance	3.5
D	Operating Pitch Diameter of Gear, in.	3.7
D_m	Mean Diameter of Fastener, in.	3.8
d	Operating Pitch Diameter of Pinion, in.	3.5
d_s	Shaft Diameter, in.	3.8
E_G	Modulus of Elasticity for Gear, psi	Table 3.4
E_p	Modulus of Elasticity for Pinion, psi	Table 3.4
F	Net Face Width of Narrowest Member, in.	3.5
H_B	Brinell Hardness	Fig. 3.1
h_1	Key Height in Shaft or Hub, in.	3.8
I	Geometry Factor for Pitting Resistance	3.5
J	Geometry Factor for Bending Strength	3.6
K_m	Load Distribution Factor for Bending Strength	3.6
K_{ms}	Load Distribution Factor, Static Torque	3.7
K_y	Yield Strength Factor	3.7
K_1	Strength Velocity Number	3.6
K_2	Strength Contact Number	3.6
K_3	Strength Stress Number	3.6
K_4	Strength Geometry Number	3.6
K_5	Velocity Factor for Bending Strength	3.6
L	Length of Key, in.	3.8
L_{min}	Minimum Total Length of Lines of Contact in Contact Zone	3.5
m_G	Gear Ratio	3.5
N_G	Number of Teeth in Gear	3.5
N_Γ	Threads Per Inch of Fastener	3.8
N_p	Number of Teeth in Pinion	3.5
n_o	Speed of Output Shaft, rpm	3.5
n_p	Pinion Speed, rpm	3.5
P_d	Diametral Pitch, Nominal, in the Plane Rotation (Transverse), in.$^{-1}$	3.6
P_{nd}	Normal Diametral Pitch, in.$^{-1}$	3.6
p_N	Normal Base Pitch, in.	3.5
s_s	Key Shear Stress, psi	Fig. 3.7
s_c	Key Compressive Stress, psi	Fig. 3.7
s_{ac}	Allowable Contact Stress Number, psi	3.5
s_{at}	Allowable Bending Stress Number, psi	3.6
s_{ay}	Allowable Yield Strength Number, psi	3.7
T_t	Shaft Torque Transmitted, lb. in.	3.8
T_{ac}	Allowable Transmitted Torque at Output Shaft Based on Pitting Resistance, lb. in.	3.5
T_{at}	Allowable Transmitted Torque at Output Shaft Based on Bending Strength, lb. in.	3.6
T_{as}	Allowable Static Torque, lb. in.	3.7
T_{as1}	Allowable Static Torque, 1st Reduction, lb. in.	3.7
T_{as2}	Allowable Static Torque, 2nd Reduction, lb. in.	3.7
T_{asn}	Allowable Static Torque, nth Reduction, lb. in.	3.7
v_t	Pitchline Velocity, ft/min	3.5
w	Width of Key, in.	3.8
Z	Length of Line Action in Transverse Plane, in.	3.5
Φ_n	Normal Operating Pressure Angle, degrees	3.5
Φ_t	Operating Transverse Pressure Angle, degrees	3.5
Ψ	Helix Angle at Operating Pitch Diameter	3.5

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.

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 API RP11L can be used to predict approximate polished rod loads and gear-reducer torque values. Cognizance should be taken by the user of the possibility of actual loads exceeding apparent loads under one or more of the following conditions:

(1) Improper counterbalancing.

(2) Excessive fluctuation in engine power output.

(3) Serious critical vibrations of the reducer and engine system.

(4) Poor bottom-hole pump operation.

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

3.3 DEFINITIONS AND SYMBOLS

Definitions. The terms used, wherever applicable, conform to the following standards:

(1) ANSI Y10.3-1968 "Letter Symbols for Quantities Used in Mechanics of Solids."

(2) AGMA 112, "Gear Nomenclature, Terms, Definitions, Symbols, and Abbreviations."

Symbols. The symbols used in the pitting resistance and bending strength formulas are shown in Table 3.1.

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.*

3.4 GEAR RATING TERMINOLOGY

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 formulas listed are extracted from "AGMA Application Standard for Helical and Herringbone Speed Reducers for Oilfield Pumping Units" (AGMA 422.03), with the permission of the publisher, the American Gear Manufacturers Association, 1901 North Fort Myer Drive, Suite 1000, Arlington, Virginia 22209.

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 3.2.

TABLE 3.2
PUMPING UNIT REDUCER SIZES AND RATINGS

Size	Peak Torque Rating, lb. in.
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

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 pound inches). On gear reducers with ratings in excess of 320,000 pound inches, the ratings shall be based on the following nominal pumping speeds:

Strokes Per Minute, n_0	Peak Torque Rating Pounds Inches
16	456,000
16	640,000
15	912,000
14	1,280,000
13	1,824,000
11	2,560,000 and larger

3.5 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 Standard 110, "Nomenclature of Gear Tooth Wear and Failure."

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

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

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

or

$$T_{ac} = C_1 \cdot C_2 \cdot C_3 \quad \text{(Eq. 2)}$$

where:

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

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

n_p = pinion speed, rpm

d = operating pitch diameter of pinion, in. 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

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

v_t = (d) (n_p) (.262), pitch line velocity, ft/min. (Do not use enlarged pinion pitch diameter) (Eq. 5)

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

$C_2 = \frac{F}{C_m}$ pitting contact number (Eq. 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 Fig. 3.2. If deflections or other sources of misalignment are such that the values of C_m from Fig. 3.2 do not represent the actual maldistribution of load across the face, then calculate the load distribution factor using AGMA 218, Load Distribution Factor, Analytical Method.

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:*

(1) Multiply C_2 by 0.95 when one element is hardened after cutting

(2) 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 218, Load Distribution Factor, Analytical Method.

$$C_3 = 0.225 \cdot \frac{m_G}{m_G + 1} \cdot \left(\frac{s_{ac}}{C_p}\right)^2 \quad \text{(Eq. 7)}$$

pitting stress number for external helical gears

s_{ac} = allowable contact stress number from Fig. 3.1 or Table 3.3

C_p = elastic coefficient

= 2300 for mating steel elements. Consult Table 3.4 for C_p values of materials other than steel

$$m_G = \text{gear ratio} = \frac{N_g}{N_p} \quad \text{(Eq. 8)}$$

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

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

$$I = \frac{\cos \Phi_t \sin \Phi_t}{2} \cdot \frac{m_G}{m_G + 1} \cdot \frac{L_{min}}{F} \quad \text{(Eq. 10)}$$

I = geometry factor for pitting resistance (wear)

Φ_t = operating transverse pressure angle, degrees

$$\Phi_t = \tan^{-1}\left(\frac{\tan \Phi_n}{\cos \Psi}\right) \quad \text{(Eq. 11)}$$

Φ_n = normal operating pressure angle, degrees

Ψ = 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:

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

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

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

p_N = normal base pitch, inches

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

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 218. The more precise method in 218 must be used for face contact ratios less than 1.0. When "I" is determined in accordance with AGMA 218 and if $2C \div (m_G + 1)$ 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 218.

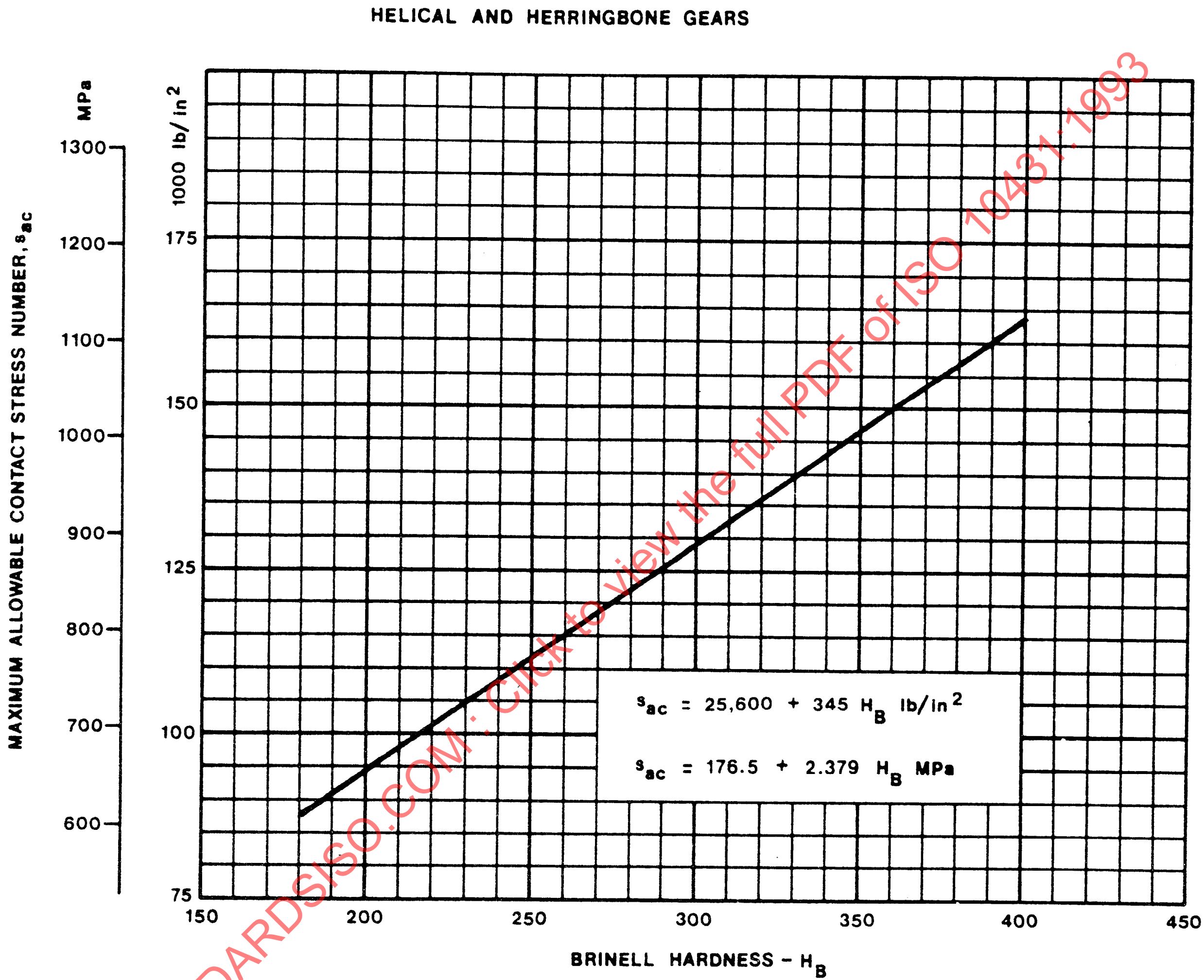


FIG. 3.1
ALLOWABLE CONTACT FATIGUE STRESS FOR THROUGH HARDENED AND TEMPERED STEEL GEARS — s_{ac}
FROM AGMA 422.03

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

TABLE 3.3
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 2*	50 HRC 54 HRC	170,000 175,000
			Carburized and Case Hardened*	55 HRC 60 HRC	180,000 200,000
		AISI 4140	Nitrided**	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	1* 90 to 100% of s_{ac} value of steel with same hardness (see Fig. 3.1)
	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 (Pearl-itic)	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

*For minimum carburized case depth Per Fig. 3.5
**For minimum nitrided case depth Per Fig. 3.6
1*The higher allowable stress for nodular iron is determined by metallurgical controls.
2*For minimum flame or induction hardened case depths and hardening pattern, see Fig. 3.7

TABLE 3.4
ELASTIC COEFFICIENT—C_p

Pinion Material and Modulus of Elasticity E_p		Gear Material & Modulus of Elasticity E_g — psi: Steel $30x10^6$	Malleable Iron $25x10^6$	Nodular Iron $24x10^6$	Cast Iron $22x10^6$
Steel	$30x10^6$	2300	2180	2160	2100
Mall. Iron	$25x10^6$	2180	2090	2070	2020
Nod. Iron	$24x10^6$	2160	2070	2050	2000
Cast Iron	$22x10^6$	2100	2020	2000	1960

Poisson's ratio = 0.30

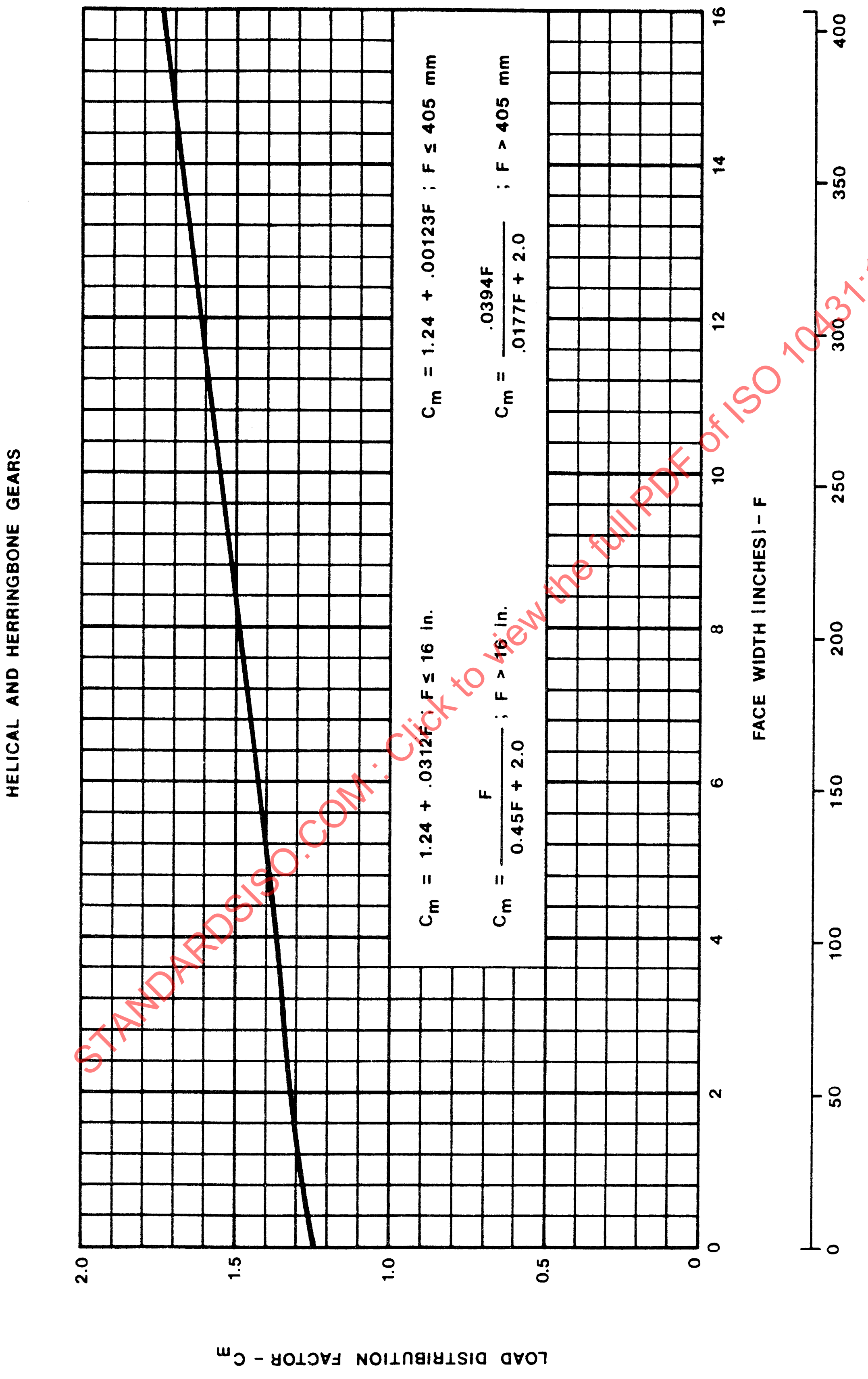


FIG 3.2
HELICAL GEAR LOAD DISTRIBUTION FACTOR — C_m
FROM AGMA 422.03

3.6 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 Standard 110, "Nomenclature of Gear Tooth Wear and Failure."

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

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

$$T_{at} = \frac{n_p\, d\, K_5}{2n_0} \cdot \frac{F}{K_m} \cdot s_{at} \cdot \frac{J}{P_d} \qquad \text{(Eq. 14)}$$

or

$$T_{at} = K_1 \cdot K_2 \cdot K_3 \cdot K_4 \qquad \text{(Eq. 15)}$$

where:

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

$K_1 = \frac{n_p\, d\, K_5}{2n_0}$ strength velocity number (Eq. 16)

n_p = pinion speed, rpm

d = operating pitch diameter of pinion, in.

K_5 = velocity factor for bending strength

$$K_5 = \sqrt{\frac{78}{78 + \sqrt{v_t}}} \qquad \text{(Eq. 17)}$$

n_0 = speed of output shaft, rpm (pumping speed, strokes per minute)

$v_t = (d)\,(n_p)\,(.262)$ pitchline velocity, ft/min. (Eq. 18)

$K_2 = \frac{F}{K_m}$ strength contact number (Eq. 19)

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 Fig. 3.4. If deflection or other sources of misalignment are such that the values of K_m from Fig. 3.4 do not represent the actual maldistribution of load across the face, then calculate the load distribution factor using AGMA 218, Load Distribution Factor, Analytical Method.

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:*

(1) Multiply K_2 by 0.95 if one element is hardened after cutting.

(2) 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) can not be attained, calculate K_m per AGMA 218, Load Distribution Factor, Analytical Method.

$K_3 = s_{at}$ strength stress number (Eq. 20)

s_{at} = allowable bending stress number, psi, from Fig. 3.3 or Table 3.5

$K_4 = \frac{J}{P_d}$ strength geometry number (Eq. 21)

J = geometry factor for bending strength per AGMA 226. For reference, see Appendix A in AGMA 422.03.

P_d = diametral pitch in plane of rotation, (transverse)

$P_d = (P_{nd})\,(\cos \Psi)$ (Eq. 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.*

3.7 STATIC TORQUE RATING

The static torque loads on the gear teeth can be caused by resisting the torque exerted by the counterbalance or other non-operating conditions. A description of the many conditions of installation, maintenance, and use of pumping unit reducers which can cause high static torques to be applied is not within the scope of this Specification.

The static torque rating of the gear reducer to resist these loads must be equal to or greater than 500% of the reducer name plate rating. Certain pumping unit geometries may require a higher static torque rating. The system analysis required by 3.2 will be used to determine when the higher static torque rating is required.

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

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

where:

D = operating pitch diameter of gear, inches

HELICAL AND HERRINGBONE GEARS

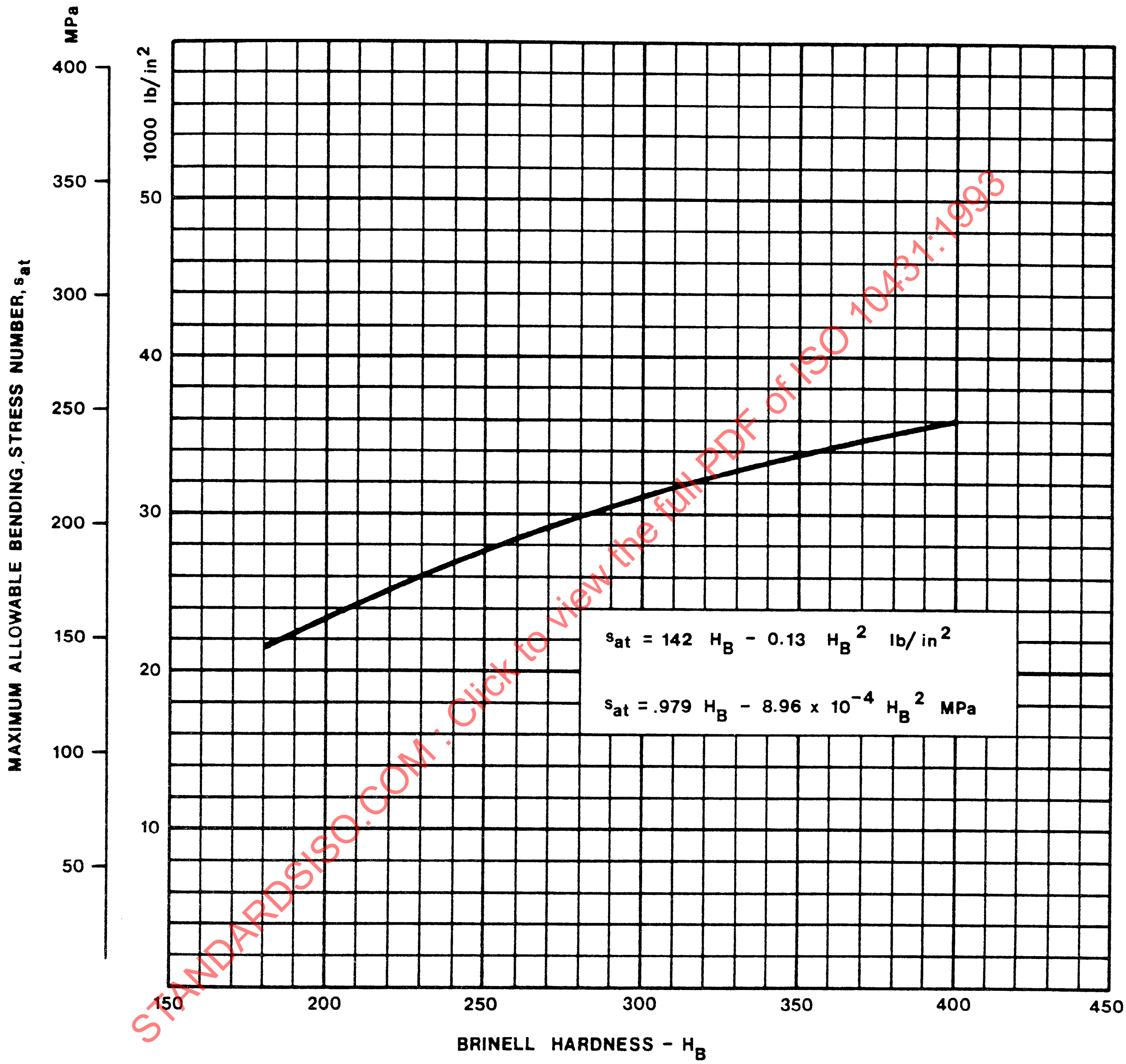


FIG. 3.3
ALLOWABLE BENDING FATIGUE STRESS FOR THROUGH HARDENED AND TEMPERED STEEL GEARS — s_{at}
FROM AGMA 422.03

HELICAL AND HERRINGBONE GEARS

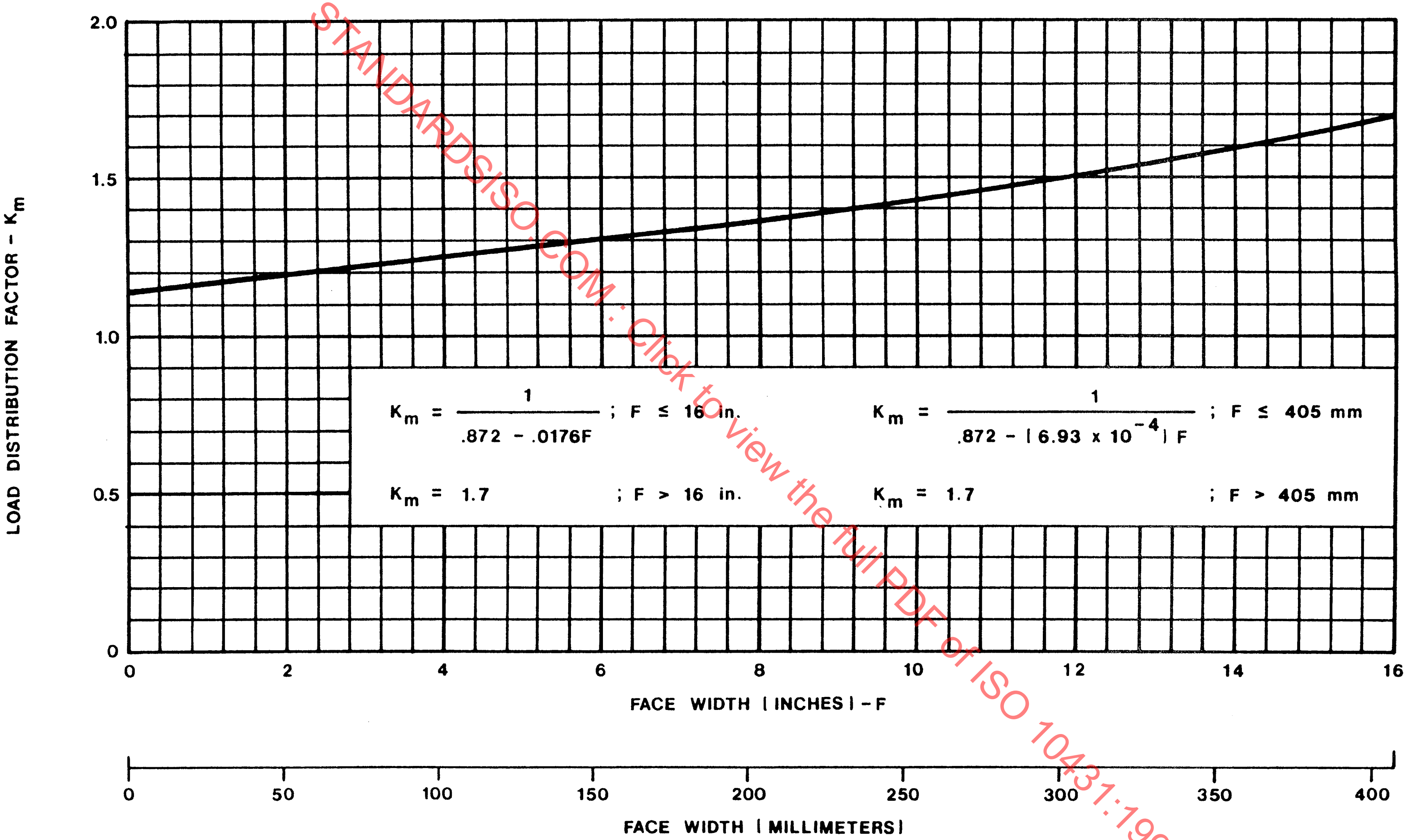


FIG. 3.4
HELICAL GEAR LOAD DISTRIBUTION FACTOR — K_m
FROM AGMA 422.03

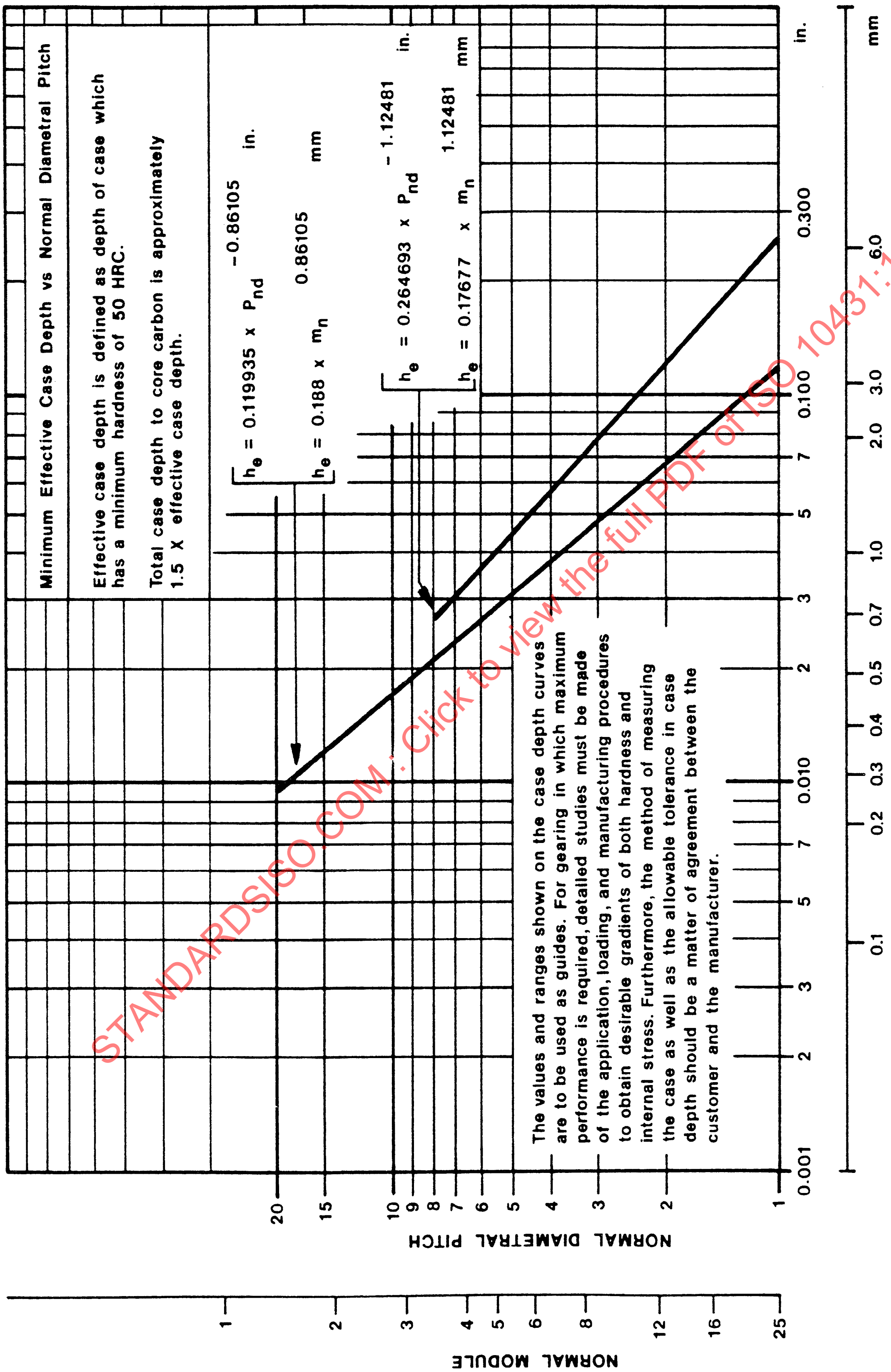


FIG. 3.5
EFFECTIVE CASE DEPTH FOR CARBURIZED GEARS, h_e
FROM AGMA 422.03

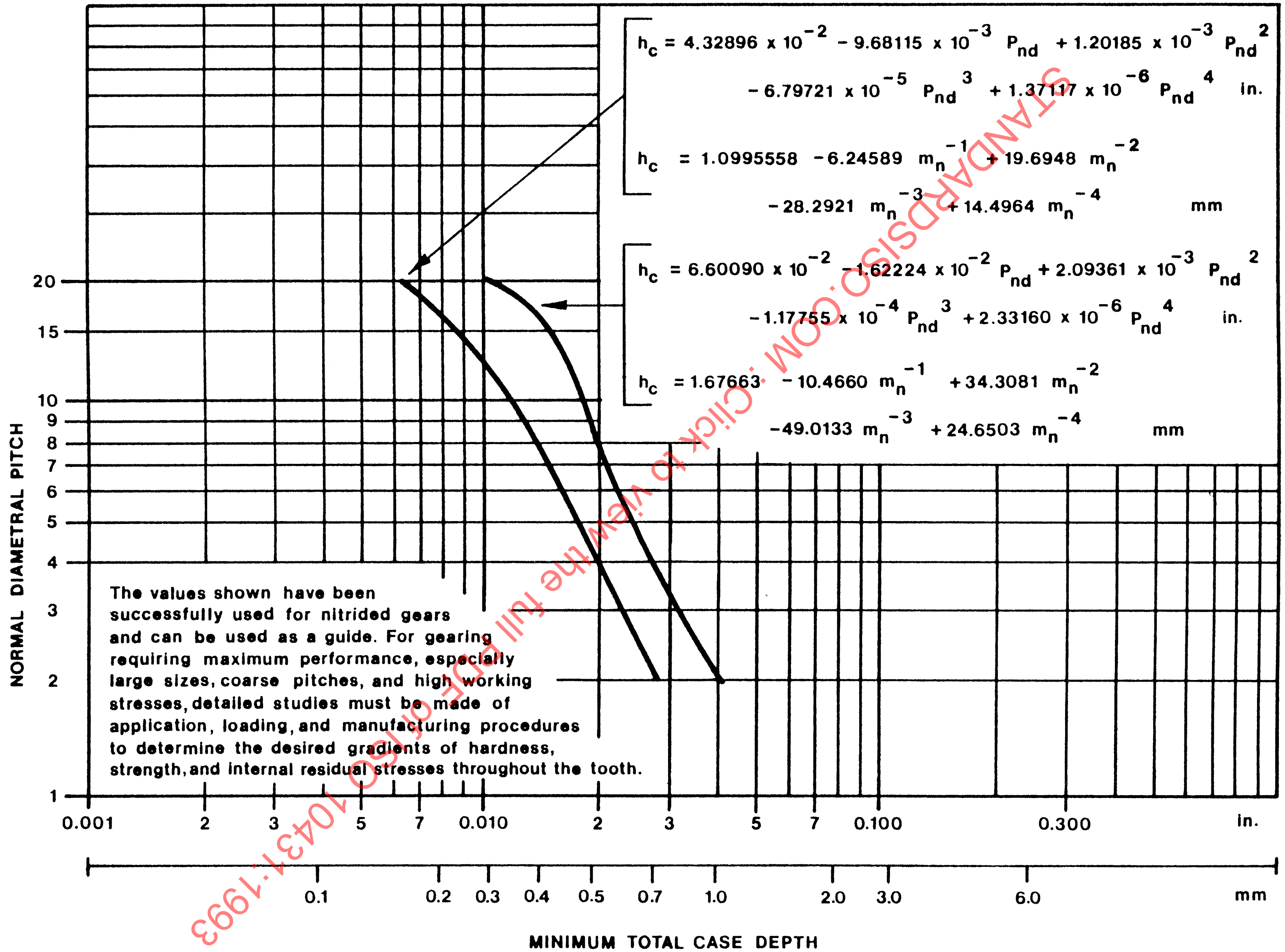


FIG. 3.6
MINIMUM TOTAL CASE DEPTH FOR NITRIDED GEARS, h_c
FROM AGMA 422.03

TABLE 3.5
ALLOWABLE BENDING FATIGUE STRESS NUMBER — s_{at}

(For Other Than Through Hardened and Tempered Steel Gears)

Material	AGMA Class	Commercial Designation	Heat Treatment	Min. Surface Hardness §	s_{at}, psi
Steel			Flame or Induction	50-54 RC	38,300
				Hardened 2*	
			Carburized & Case Hardened*	55 RC	47,000
				60 RC	47,000
		AISI 4140	Nitrided**	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	1* 90 to 100%
	A-7-c	80-55-06	Quenched & Tempered	180 BHN	of s_{at} value of steel with
	A-7-d	100-70-03	Quenched & Tempered	230 BHN	same hardness
	A-7-e	120-90-02	Quenched & Tempered	270 BHN	
	—	120-90-02 Mod.	Quenched & Tempered	300 BHN	
Malleable Iron (Pearl-itic)	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

*For minimum carburized case depths Per Fig. 3.5
**For minimum nitrided case depths Per Fig. 3.6
1*The higher allowable stress for nodular iron is determined by metallurgical controls.
2*For minimum flame or induction hardened case depths and hardening pattern, see Fig. 3.7 and Fig. 3.8. 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.
§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 Data Sheet 4.5.

T_{as} = allowable static torque at the gear or pinion being checked:

T_{as1} = 1st reduction,

T_{as2} = 2nd reduction,

T_{asn} = nth reduction

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

s_{ay} = allowable yield strength number of the gear or pinion material; Fig. 3.9 for steel and nodular iron. For case hardened (flame, induction, nitrided, carburized) material, use core hardness from Manufacturers Data Sheet to determine yield strength number

K_y = yield strength factor. See Table 3.6.

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

$K_{ms} = 1.3$ for $F > 16''$

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.

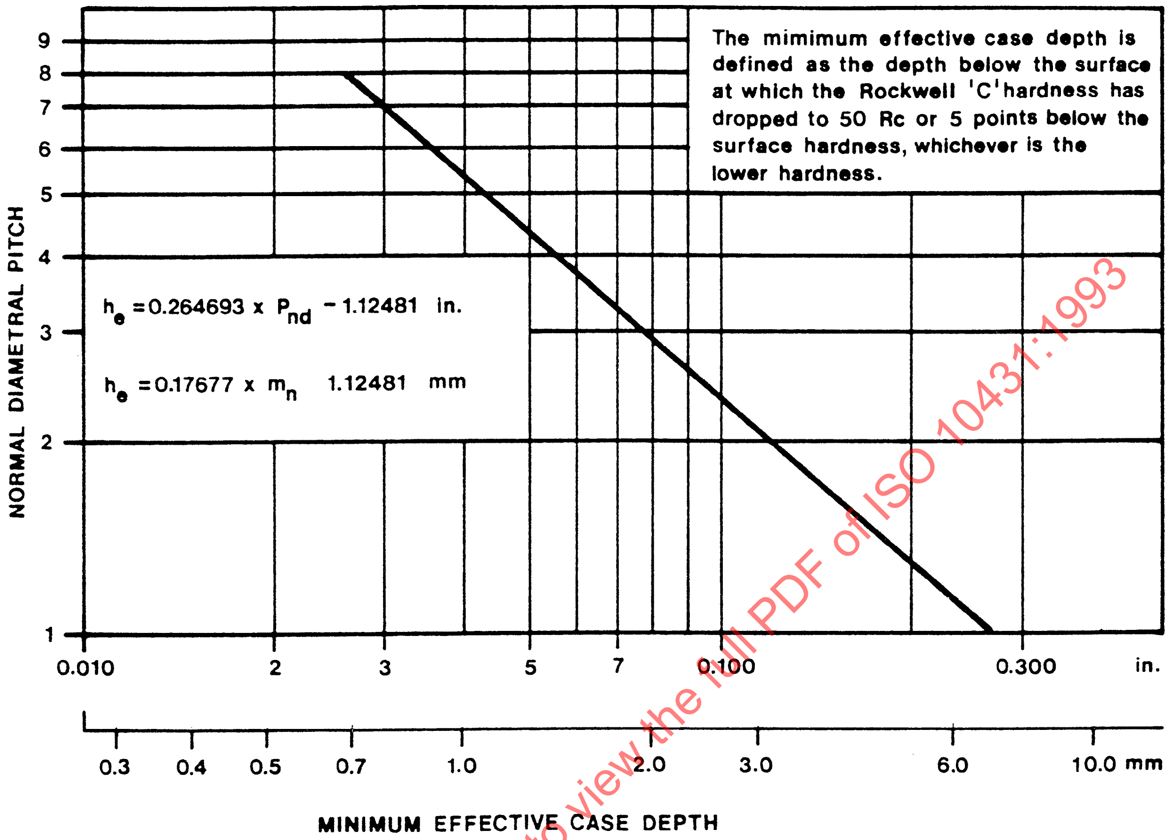


FIG. 3.7
MINIMUM EFFECTIVE CASE DEPTH FOR FLAME OR INDUCTION HARDENED GEARS, h_e
FROM AGMA 422.03

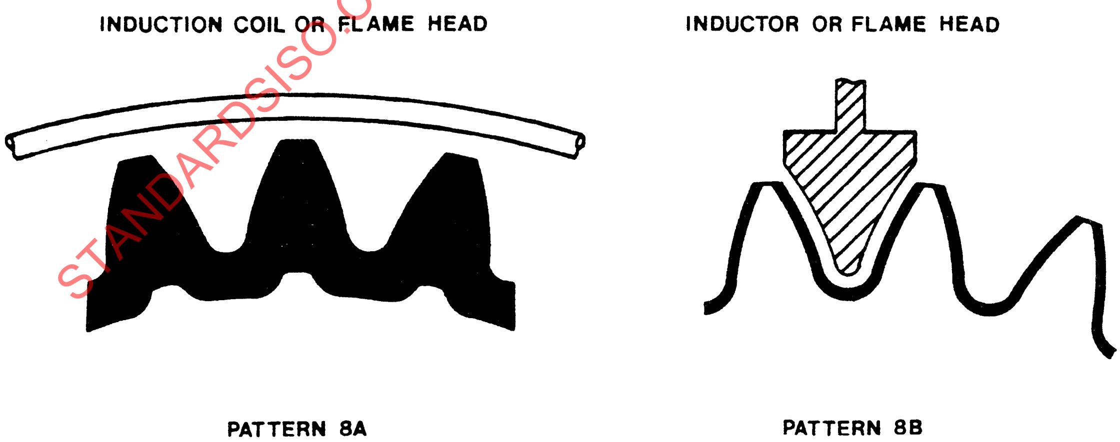


FIG. 3.8
ACCEPTABLE FLAME AND INDUCTION HARDENING PATTERNS
FROM AGMA 422.03

HELICAL AND HERRINGBONE GEARS

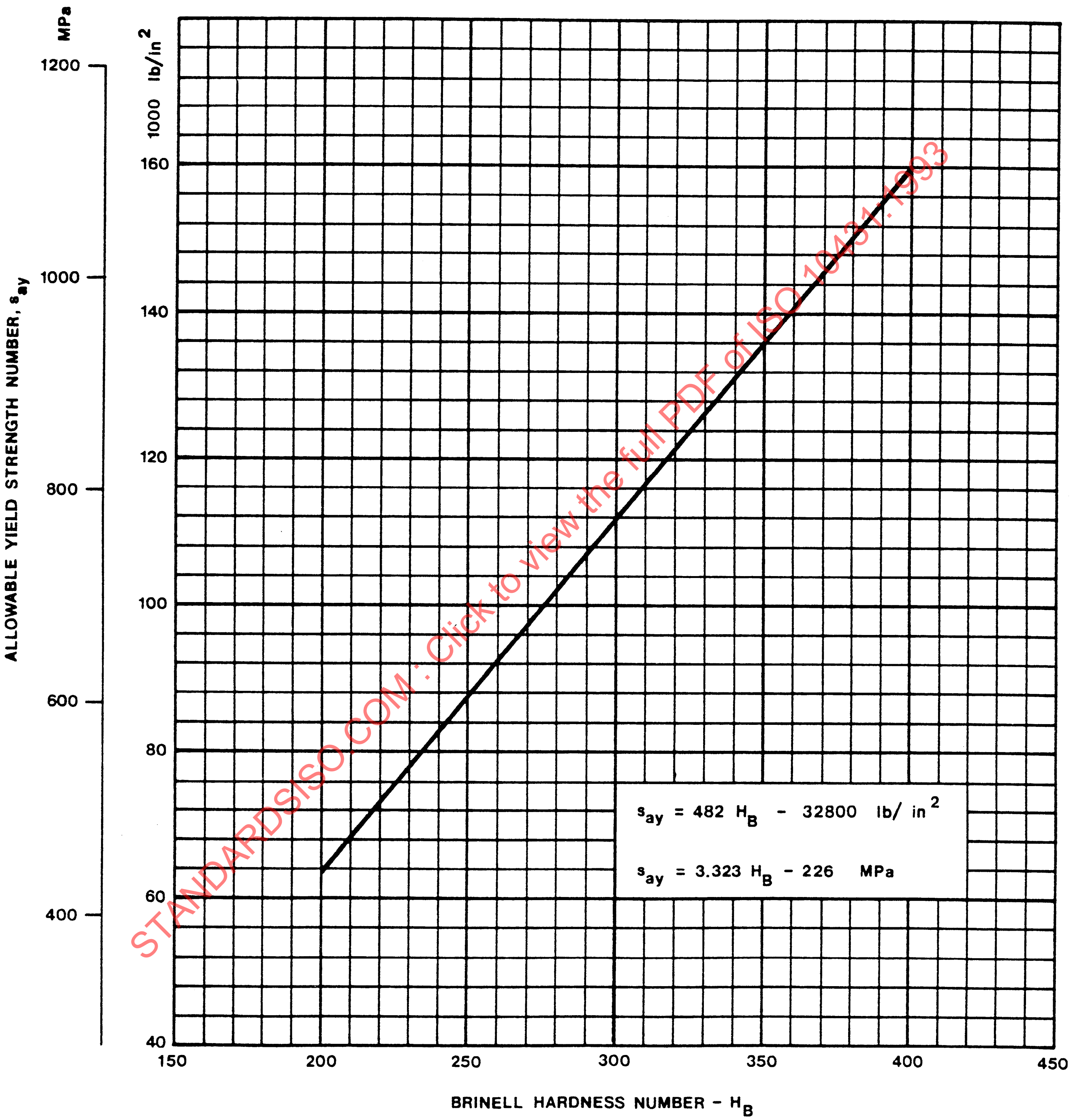


FIG. 3.9
ALLOWABLE YIELD STRENGTH NUMBER — s_{ay}
STEEL AND NODULAR IRON
FROM AGMA 422.03

TABLE 3.6
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
Malleable Iron	1.0

3.8 COMPONENTS

Component Design. Gear reducers for oilfield 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 which influence the loading.

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.

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

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.

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

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 Fig. 3.10 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.

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.

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

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

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

where:

s_s = shear stress of key, psi (see Table 3.7)

s_c = compressive stress of key, psi (see Table 3.7)

T_t = transmitted shaft torque, lb. in.

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

w = width of key, in.

L = length of key, in.

h_1 = height of key in the shaft or hub that bears against the keyway, in.

For designs where unequal portions of the keyway are in the hub or shaft, h_1 must be the minimum portion.

Allowable Stresses. Maximum allowable key stresses based on peak torque rating are shown in Table 3.7. 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 that shaft.

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

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 structural loading.

The maximum allowable stress at the tensile area of threaded fasteners (bolts, studs, or capscrews) shall not exceed the values given in Table 3.8. The tensile area (A) is calculated as follows:

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

where:

A = tensile area of fastener, in.²

D_m = major diameter of fastener, inches

N_T = threads per inch of fastener

TABLE 3.7
ALLOWABLE KEY STRESSES*

Key Material	Hardness BHN	Allowable Stress, psi Shear	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

*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.

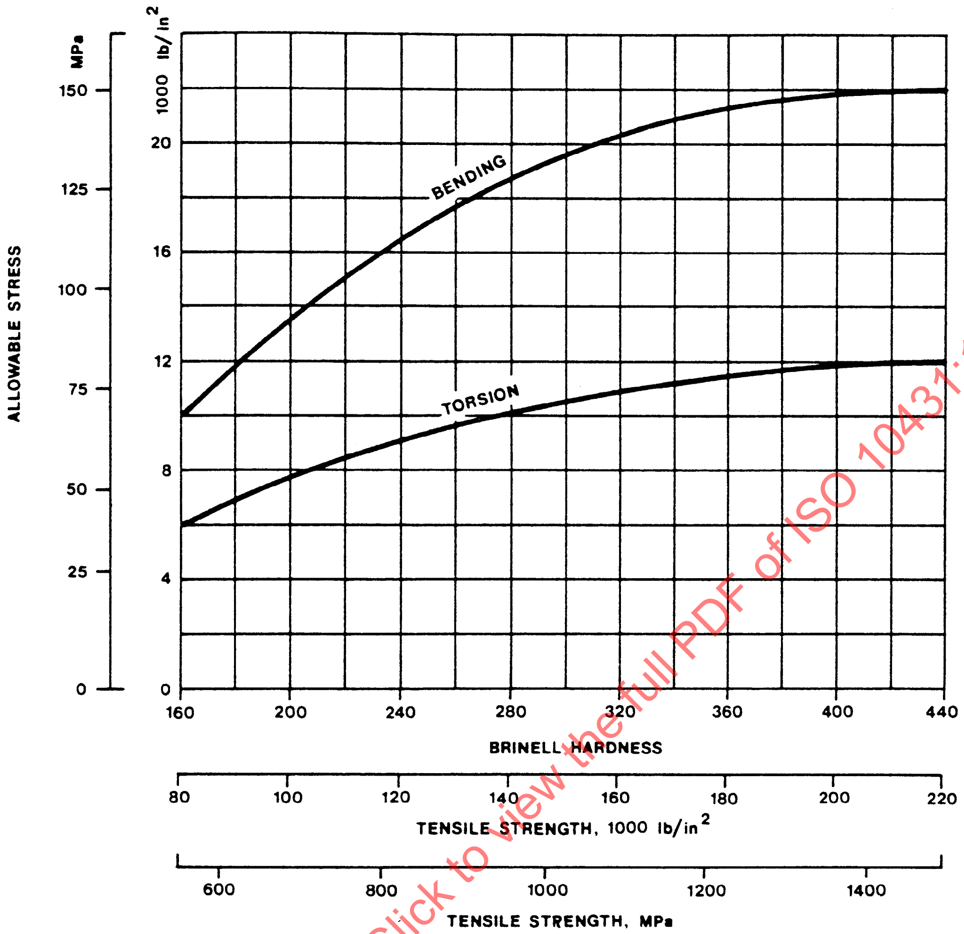


FIG. 3.10
ALLOWABLE STRESS — SHAFTING
FROM AGMA 422.03

TABLE 3.8
MAXIMUM ALLOWABLE TENSILE STRESS, FASTENERS

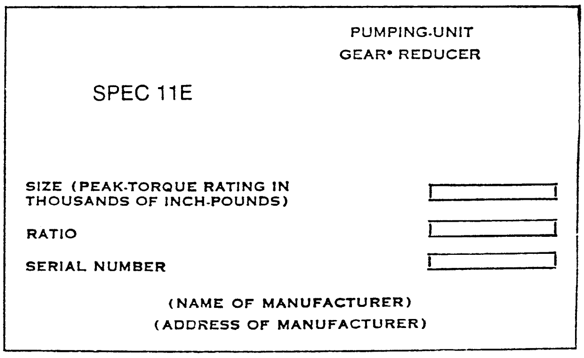
SAE and/or ASTM Designation	Threaded Fastener Diameter, inches	Hardness BHN	Yield Strength psi min.	Ultimate Tensile Strength psi min.	Allowable Applied Tensile Stress psi max.
SAE 2	Over ¼ to ¾ incl.	149-241	55,000	74,000	11,000
	Over ¾ to 1½ incl.	121-241	33,000	60,000	11,000
SAE 5	Over ¼ to 1 incl.	241-302	85,000	120,000	20,000
(ASTM A-449)	Over 1 to 1½ incl.	223-285	74,000	105,000	18,000
ASTM A-449	Over 1½ to 3 incl.	183-235	55,000	90,000	13,000
ASTM A-354	Over ¼ to 2½ incl.	217-285	80,000	105,000	17,000
Grade BB	Over 2½ to 4 incl.	217-285	75,000	100,000	17,000
ASTM A-354	Over ¼ to 2½ incl.	255-321	109,000	125,000	22,000
Grade BC	Over 2½ to 4 incl.	255-321	99,000	115,000	22,000
SAE 7	Over ¼ to 1½ incl.	277-321	105,000	133,000	24,600
SAE 8 (ASTM A-354 Grade BD)	Over ¼ to 1½ incl.	302-352	120,000	150,000	27,700

NOTE: *The basis for the values in Table 3.8 is to prevent joint opening at a peak rated load.*

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.

Special Seals and Breathers. It is recognized that oilfield pumping units operate outdoors under adverse atmospheric conditions and must be equipped with seals and breathers designed for these conditions.

3.9 LUBRICATION (See API RP 11G)



***Substitute "CHAIN" when appropriate.**

FIG. 3.11

PUMPING-UNIT REDUCER NAME PLATE

3.10 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 Table 3.9 for each gear reducer size manufactured.

3.11 MARKING*

Each pumping—unit reducer shall be provided with a nameplate substantially as shown in Fig. 3.11. The size (peak torque rating in 1,000 lb. in.) shown on the nameplate shall be one of those listed in Table 3.2. 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 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 reducer markings.*

*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, herein. No other use of the monogram is permitted.

CHAIN REDUCERS

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

3.13 Single, or multiple strand roller chain, conforming to American National Standards Institute (ANSI) 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.

3.14 Sprockets shall have ANSI tooth form.

3.15 The small sprocket shall have not less than eleven teeth.

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

3.17 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).

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

3.19 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.

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

3.21 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.**

3.22 The following formula shall be used for rating of chain:

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

Wherein:

T = peak-torque rating in inch-pounds.
S = ANSI ultimate tensile strength of chain in pounds.
R = pitch radius of large sprocket in inches.

TABLE 3.9
MANUFACTURER'S GEAR REDUCER DATA SHEET

MANUFACTURED BY: ________________________ , DATE SUBMITTED ______________________________

NOMINAL API REDUCER SIZE __________

CALCULATED VALUES

PITTING RESISTANCE TORQUE

First Reduction ______________________ lb. in.

Second Reduction ____________________ lb. in.

Third Reduction ______________________ lb. in.

BENDING STRENGTH TORQUE

First Reduction:

Gear __________ lb. in., Pinion __________ lb. in.

Second Reduction:
Gear __________ lb. in., Pinion __________ lb. in.

Third Reduction:

Gear __________ lb. in., Pinion __________ lb. in.

STATIC TORQUE

First Reduction:

Gear __________ lb. in., Pinion __________ lb. in.

Second Reduction:
Gear __________ lb. in., Pinion __________ lb. in.

Third Reduction:

Gear __________ lb. in., Pinion __________ lb. in.

NOTE: (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., _______ NPA, ________ TPA

TABLE 3.9 (Continued)

GEOMETRY FACTORS, I & 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 & PINION MATERIALS & HARDNESS

First Reduction:

Gear Material ________, Surface BHN/Rc ________, Core BHN§ ________

Pinion Mtl. ________, Surface BHN/Rc ________, Core BHN§ ________

Second Reduction:

Gear Material ________, Surface BHN/Rc ________, Core BHN§ ________

Pinion Mtl. ________, Surface BHN/Rc ________, Core BHN§ ________

Third Reduction:

Gear Material ________, Surface BHN/Rc ________, Core BHN§ ________

Pinion Mtl. ________, Surface BHN/Rc ________, Core BHN§ ________

§ Core hardness required for surface hardened gears and pinions only.

OTHER COMPONENTS

Crankshaft Material ________________, Hardness ________________

Housing Material ________________________________

Housing Type (Check ✓): Split ________, One Piece ________

BEARING SIZES*

High Speed Pinion ________________

**Intermediate Speed Pinion ________________

Low Speed Pinion ________________

Low Speed Gear ________________

*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.)

**Not applicable on double reduction reducers.

BEARING LOADING***

High Speed Pinion ________________

**Intermediate Speed Pinion ________________

Low Speed Pinion ________________

Low Speed Gear ________________

***For journal bearings list psi loading on each bearing. For roller bearings, list L-10 life as calculated in 3.8.

**Not applicable on double reduction reducers.

SECTION 4
INSPECTION AND REJECTION

4.1 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 which 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 so conducted as not to interfere unnecessarily with the operation of the works. The manufacturer shall furnish the inspector with gages, or other necessary measuring instruments, the accuracy of which shall be proved to the satisfaction of the inspector.

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

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

4.4 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.

APPENDIX A
APPROVED DATA FORMS

A1. Rating Form for Crank Counterbalances. An approved form for rating of pumping-unit crank counterbalances is shown in Fig. A-1. Manufacturers are urged to use this form when providing the information indicated.

A2. Stroke and Torque Factors. An approved form for submission of pumping-unit stroke and torque factors is shown in Fig. A-2. Manufacturers are urged to use this form when supplying such information.

Name of Manufacturer__________ Designation of Unit__________

1	2	3
Description[1]	Total Weight, lb.	Maximum[2] Moment about Crankshaft, in.-lb.

[1]Describe parts in use accurately enough to avoid any possible misunderstanding, showing on separate lines a series of practical combinations from minimum to maximum.
[2]Equals total weight (Col. 2) times distance to center of gravity in inches, with crank in horizontal position.

FIG. A-1
API RATING FORM FOR CRANK COUNTERBALANCE

Name of Manufacturer

Designation of Unit

Pumping Unit Structural Unbalance ____ pounds

1	2	3	4	5	6	7	8	9
Position of Crank,[1] deg.	Position of Rods[2]				Torque Factor[3]			
	Length of Stroke, in.				Length of Stroke, in.			
0								
15								
30								
45								
60								
75								
90								
105								
120								
135								
150								
165								
180								
195								
210								
225								
240								
255								
270								
285								
300								
315								
330								
345								

[1]For crank counterbalanced 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 counter-clockwise 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.

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

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

A	________	P	________
C	________	K	________
R_1	________	H	________
R_2	________	I	________
R_3	________	G	________

NOTE: *See Appendix B, C, D or E for symbol identification.*

FIG. A-2
PUMPING-UNIT STROKE AND TORQUE FACTOR

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)

Definition

B1. The torque factor for any given crank angle is that factor which, 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.

Method of Calculation

B2. 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 are given in Appendix A, Fig. A2.

B3. Torque factors and polished-rod positions are to be furnished by pumping-unit manufacturers for each 15-deg 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.

B4. Referring to Fig. B1, 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 crank-pin 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 crank-pin bearing to the center of the saddle bearing, inches.

ϕ = Angle between the 12 o'clock position and K, degrees; equals $\tan^{-1}\left(\frac{I}{H-G}\right)$

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

β = Angle between C and P, degrees.

α = Angle between P and R, degrees, measured clockwise from R to P.

Ψ = Angle between C and K, degrees, (equals angle χ — angle ρ).

Ψ_t = Angle between c and k, degrees, at top (highest) polished rod position.

Ψ_b = Angle between c and k, degrees, at bottom (lowest) polished rod position.

χ = Angle between C and J, degrees.

ρ = Angle between K and J, degrees.

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

$\overline{W}$ = Polished-rod load at any specific crank angle θ, 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 crank pins. 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 crank angle θ, (equals $\overline{TF} \times W_n$).

M = Maximum moment of the rotary counterweights, cranks, and crank pins about the crankshaft, inch-pounds.

T_r = Torque, inch-pounds, due to the rotary counterweights, cranks, and crank pins for a given crank angle θ (equals $M \sin \theta$)

T_n = Net torque, inch-pounds, at the crankshaft for a given crank angle θ (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 θ.

B5. By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crank angle θ.

$$\overline{TF} = \frac{AR}{C} \quad \frac{\sin \alpha}{\sin \beta} \qquad \text{B.1}$$

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

$$\phi = \mathrm{Tan}^{-1}\left(\frac{I}{H-G}\right) \quad \text{B.2}$$

This is a constant angle for any given pumping unit.

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

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

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

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

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

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

$$\Psi = \chi - \rho \quad \text{B.6}$$

$$\alpha = \beta + \Psi - (\theta-\phi) \quad \text{B.7}$$

$$\overline{PR} = \frac{\Psi_b - \Psi}{\Psi_b - \Psi_t} \quad \text{B.8}$$

$$\Psi_b = \cos^{-1}\frac{C^2+K^2-(P+R)^2}{2\,CK} \quad \text{B.9}$$

$$\Psi_t = \cos^{-1}\frac{C^2+K^2-(P-R)^2}{2\,CK} \quad \text{B.10}$$

Application of Torque Factors

B6. 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 formula:*

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

B7. The formula 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 non-dynamic 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 paragraphs B17., B18., and B19.

*This formula applies to pumping units where maximum counterbalance moment is obtained at θ equals 90 deg or 270 deg.

B8. 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 θ***, by the torque factor for the crank angle θ. The result is the rotary counterbalance effect, in pounds, at the polished rod.

B9. Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the cranks in the 90 deg or 270 deg 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 formula:

$$M = \overline{TF}\,(W-B) \quad \text{B.12}$$

To check measurements, the maximum moment, ***M***, should be determined with the cranks in both the 90-deg and 270-deg positions. Should there be a significant difference in the maximum moments calculated from measurements at 90 deg and 270 deg, 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.

B10. To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 4,000-ft well is shown in Fig. B2. 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 deg of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length which is proportional to the length of the stroke.

The length of the base line 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.*

B11. To further illustrate, a calculation will be made considering the point where the crank angle θ equals 75 deg. From polished-rod stroke and torque factor data for the particular 64-in. stroke 160-D pumping unit used for this example, it is found that the position of the polished rod at 75 deg 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 (Fig. B2). The dynamometer deflection at this point is read to be 1.16 in. which, with a scale constant of 7,450 lb. per in., makes the load, *(W)* at that point 8,650 lb.

B12. In a similar manner, the polished-rod load may be obtained for each 15-deg angle of crank rotation. The dynamometer card has been marked to show

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

the load and position involved for each 15 deg of crank angle. The structural unbalance, B, for the example unit equals + 650 lb. Therefore, the net polished-rod load, W_n at θ = 75 deg = $W—B$ = 8,650 — (+650) = 8,000 lb. The torque, T_{wn}, due to the net polished-rod load = $\overline{TF} \times W_n$ = 34.38 × 8,000 = 275,000 in.-lb.

B13. 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 Par. B9. 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.

B14. The horizontal dotted line drawn across the dynamometer card in Fig. B2 is the counterbalance effect measured with the dynamometer at the 90-deg crank angle and is 6,250 lb. The maximum moment can then be calculated as follows, using formula B.12:

$$
\begin{aligned}
M &= \overline{TF}\ (W—B)\\
&= 32.76 \times (6{,}250 - 650)\\
&= 183{,}000 \text{ in.-lb}
\end{aligned}
$$

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

Although not shown, the measured counterbalance effect for the 270-deg crank position was 6,410 lb. Using the torque factor of 32.04 at the 270-deg crank position for the example unit, the maximum moment is:

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

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

B15. The torque, T_r, due to the counterbalance at the 75-deg crank position would therefore be equal to 184,000 × sin 75-deg = 184,000 × 0.966 = 178,000 in.-lb. The net torque at the crankshaft for the 75-deg crank position would then be calculated from formula B.11 as follows:

$$
\begin{aligned}
T_n &= \overline{TF}\ (W—B) - M \sin \theta\\
&= 34.38 \times (8{,}650 - 650) - 184{,}000 \times 0.966\\
&= 275{,}000 - 178{,}000 = 97{,}000 \text{ in.-lb}
\end{aligned}
$$

These values may be calculated for other crank angle positions in the same manner as outlined above. Shown in Fig. B3 is a plot of torque vs. crank angle which includes the net polished-rod load torque curve, the counterbalance torque curve, and the net crankshaft torque curve.

B16. 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. (Fig. A2, Appendix A, Std 11E.) The position of crank, degrees, (Col. 1) is reversed, starting from the bottom with 15 deg and counting up in 15-deg increments to 360 deg.

B17. 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 sized 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.

B18. 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.

B19. 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° divisions is permissible without significant error.

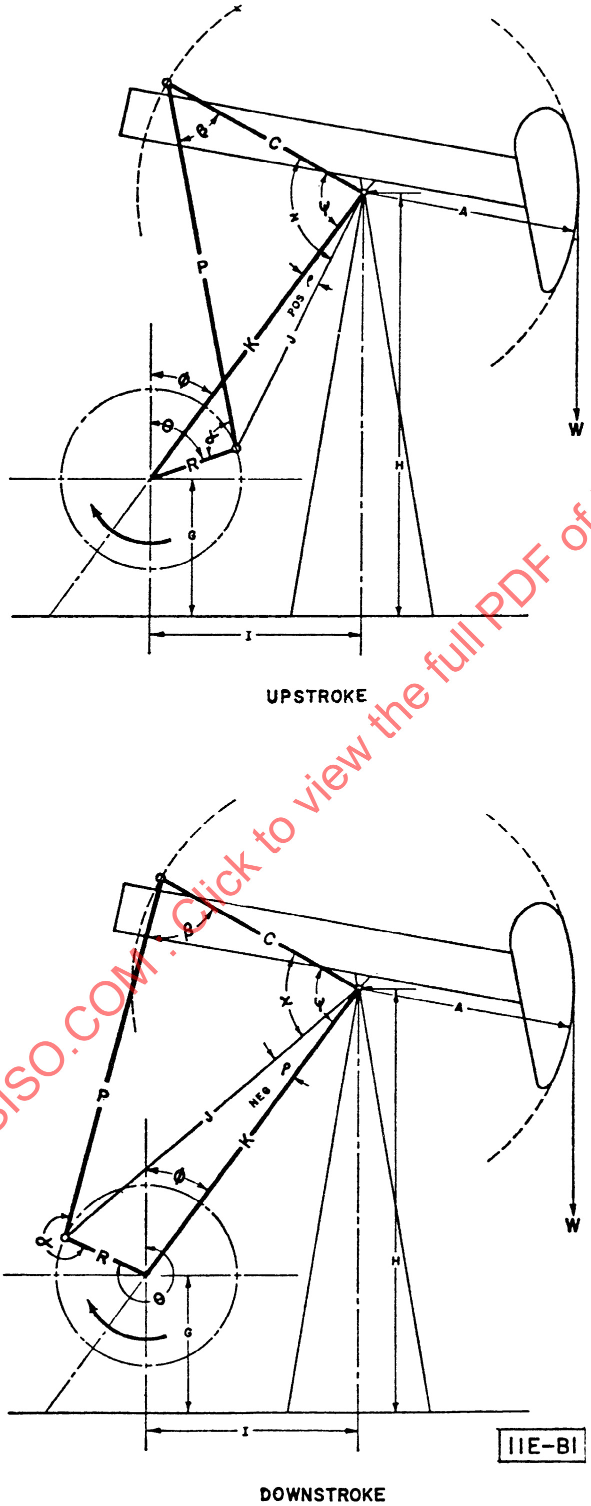


FIG. B1

PUMPING UNIT GEOMETRY

See Par. B4 for definition of symbols

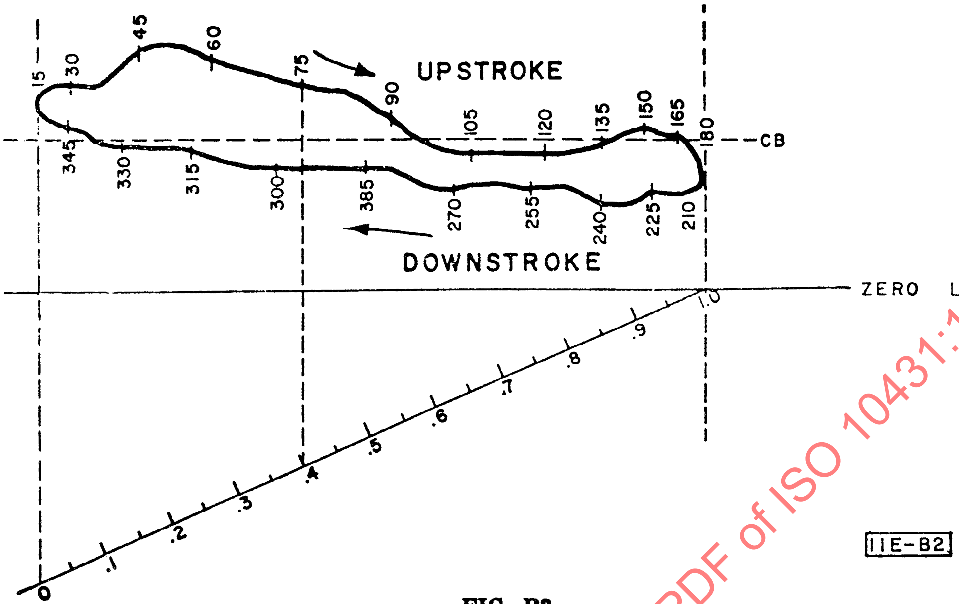


FIG. B2

DIVISION OF DYNAMOMETER CARD BY CRANK ANGLE USING API POLISHED-ROD POSITION DATA

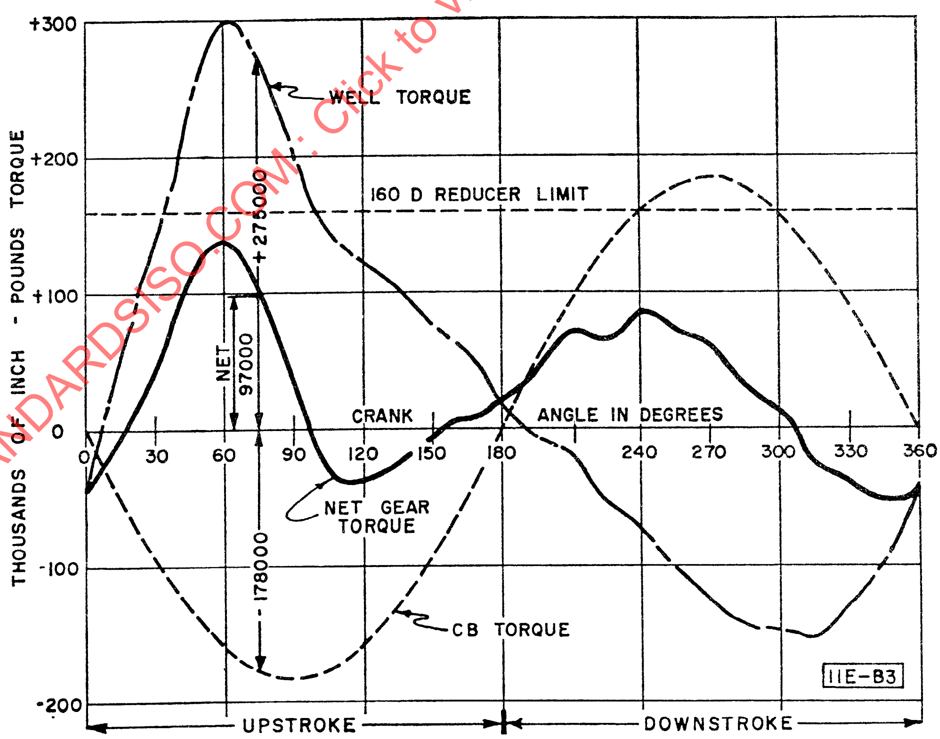


FIG. B3

TORQUE CURVES USING API TORQUE FACTORS

NET REDUCER TORQUE CALCULATION SHEET

(Conventional Crank Balanced Unit Only — CLOCKWISE ROTATION)

Company: ______________________

Location: ______________________

$Tn = \overline{TF}(W\text{-}B) - M \text{ SIN } \theta$

Well No.: ______________________

Unit Size: ______________________

θ	SINE θ	W	B	W—B	$\overline{TF}$	$\overline{TF}$ (W-B)	-M (SINE θ)	Tn
0	0						0	
15	.259						—	
30	.500						—	
45	.707						—	
60	.866						—	
75	.966						—	
90	1.000						—	
105	.966						—	
120	.866						—	
135	.707						—	
150	.500						—	
165	.259						—	
180	0						0	
195	- .259						+	
210	- .500						+	
225	- .707						+	
240	- .866						+	
255	- .966						+	
270	-1.000						+	
285	- .966						+	
300	- .866						+	
315	- .707						+	
330	- .500						+	
345	- .259						+	

Tn = Net Reducer Torque, in.-lbs

θ = Position of Crank

M = Maximum Moment of Counterbalance, in.-lbs

W = Measured Polished Rod Load at θ, lbs

B = Unit Structural Unbalance, lbs

$\overline{TF}$ = Torque Factor at θ, in.

CB at 90 ° = __________

M = (CB at 90° - B)($\overline{TF}$ at 90°) = __________

NET REDUCER TORQUE CALCULATION SHEET

(Conventional Crank Balanced Unit Only — COUNTER CLOCKWISE ROTATION)

Company: ______________________

Location: ______________________

$Tn = \overline{TF}\,(W\text{-}B) - M \text{ SIN } \theta$

Well No.: ______________________

Unit Size: ______________________

θ	SINE θ	W	B	W—B	$\overline{TF}$	$\overline{TF}$ (W-B)	-M (SINE θ)	Tn
0	0						0	
345	− .259						+	
330	− .500						+	
315	− .707						+	
300	− .866						+	
285	− .966						+	
270	−1.000						+	
255	− .966						+	
240	− .866						+	
225	− .707						+	
210	− .500						+	
195	− .259						+	
180	0						0	
165	.259						−	
150	.500						−	
135	.707						−	
120	.866						−	
105	.966						−	
90	1.000						−	
75	.966						−	
60	.866						−	
45	.707						−	
30	.500						−	
15	.259						−	

Tn = Net Reducer Torque, in.-lbs
θ = Position of Crank
M = Maximum Moment of Counterbalance, in.-lbs
W = Measured Polished Rod Load at θ, lbs
B = Unit Structural Unbalance, lbs

$\overline{TF}$ = Torque Factor at θ, in.
CB at 270° = ________
M = (CB at 270° − B)($\overline{TF}$ at 270°) = ________

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)

Definition

C1. The torque factor for any given crank angle is that factor which, 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.

Method of Calculation

C2. 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 are given in Fig. A2, Appendix A.

C3. Torque factors and polished-rod positions are to be furnished by pumping-unit manufacturers for each 15-deg crank position with the zero position at 6 o'clock. Other crank positions are determined by the angular displacement in a counter-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.

C4. Referring to Fig. C1, 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 crank-pin 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 crank-pin bearing to the center of the Samson Post bearing, inches.

ϕ = Angle between the 6 o'clock position and K, degrees; equals $\tan^{-1}\left[\frac{I}{H-G}\right] + 180°$

θ = Angle of crank pin rotation in a counter-clockwise direction viewed with the well-head to the right and with zero degrees occurring at 6 o'clock, degrees.

β = Angle between C and P, degrees.

α = Angle between P and R, degrees, measured clockwise from R to P.

Ψ = Angle between C and K, degrees, (equals angle χ — angle ρ).

Ψ_t = Angle between c and k, degrees, at top (highest) polished rod position.

Ψ_b = Angle between c and k, degrees, at bottom (lowest) polished rod position.

χ = Angle between C and J, degrees.

ρ = Angle between K and J, degrees.

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

W = Polished-rod load at any specific crank angle θ, pounds.

W_c = Counterbalance in pounds at the polished rod determined using dynamometer with crank pin at 90° 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 crank pins. 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, inch-pounds, due to the net polished-rod load for a given crank angle θ (equals $\overline{TF} \times W_n$).

M = Maximum moment of the rotary counterweights, cranks, and crank pins about the crankshaft, inch-pounds.

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

T_r = Torque, inch-pounds, due to the rotary counterweights, cranks, and crank pins for a given crank angle θ [equals M sin $(\theta + \tau)$]

T_n = Net torque, inch-pounds, at the crankshaft for a given crank angle θ (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 θ.

C5. By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crank angle θ.

$$\overline{TF} = \frac{AR}{C} \quad \frac{\sin\alpha}{\sin\beta} \qquad \text{.......... C.1}$$

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

$$\phi = \mathrm{Tan}^{-1}\left[\frac{I}{H\text{-}G}\right] + 180° \qquad \text{.......... C.2}$$

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° through 270°. It is positive for all other angles between 0° and 360°. When the angle $(\theta-\phi)$ is a negative number, it can be subtracted from 360° 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 ρ. The angle ρ is taken as positive when sin ρ is positive. This occurs for crank positions where $(\theta-\phi) = 0°$ to $(\theta-\phi) = 180°$.

The angle ρ is taken as negative when sin ρ is negative. This occurs for crank positions $(\theta-\phi) = 180°$ through $(\theta-\phi) = 360°$.

When the angle $(\theta-\phi)$ is negative, it can be subtracted from 360° and this new angle can be used to determine the proper sign.

$$\Psi = \chi - \rho \qquad \text{.......... C.6}$$

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

$$\overline{PR} = \frac{\Psi_b - \Psi}{\Psi_b - \Psi_t} \qquad \text{.......... C.8}$$

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

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

Application of Torque Factors

C6. 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 formula:

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

C7. The formula 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 non-dynamic 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 paragraphs C16, C17, and C18.

C8. Torque factors may be used to obtain the effect at the polished rod of the rotary counterbalance. This is done for a given crank angle by dividing the counterbalance moment, M sin $(\theta+\tau)$, by the torque factor for the crank angle θ. The result is the rotary counterbalance effect, in pounds, at the polished rod.

C9. Torque factors may also be used to determine the maximum rotary counterbalance moment. This is done by placing the crank pins in the 90 deg 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 formula:

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

C10. To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 2872 ft well is shown in Fig. C2. 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 deg of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length which is proportional to the length of the stroke.

The length of the base line 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.*

C11. To further illustrate, a calculation will be made considering the point where the crank angle θ equals 60 deg. From polished-rod stroke and torque

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

factor data for the particular 86-in. stroke 160-D pumping unit used for this example, it is found that the position of the polished rod at 60 deg is 0.405, and that the torque factor $\overline{TF}$ is 36.45. 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 (Fig. C2). The dynamometer deflection at this point is read to be 0.99 in. which, with a scale constant of 7,500 lb. per in., makes the load, (W) at that point 7425 lb.

C12. In a similar manner, the polished-rod load may be obtained for each 15-deg angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15 deg of crank angle. The structural unbalance, B, for the example unit equals —1535 lb. Therefore, the net polished-rod load, W_n at θ = 60 deg. = W—B = 7425 — (—1535) = 8960 lb. The torque, T_{wn}, due to the net polished-rod load = $\overline{TF} \times W_n = 36.45 \times 8960$ = 326,592 in.—lb.

C13. 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 Par. C9. Should the manufacturers' counterbalance data be used, it is suggested that a check be made using a polished-rod measurement technique.

C14. The horizontal dotted line drawn across the dynamometer card in Fig. C2 is the counterbalance effect measured with the dynamometer at the 90-deg crank angle and is 4594 lb. The maximum moment can then be calculated as follows, using formula C.12:

$$
\begin{aligned}
M &= \frac{\overline{TF} \quad (W_c - B)}{\text{Sin} \quad (90^\circ + \tau)} \\
&= 38.38 \times (4594 + 1535) \; / \; .891 \\
&= 264{,}008 \text{ in.—lb.}
\end{aligned}
$$

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

C15. The torque, T_r, due to the counterbalance at the 60 deg crank position would therefore be equal to 264,008 × sin (60° + 27°) = 264,008 × 0.999 = 263,744 in.—lb. The net torque at the crankshaft for the 60-deg crank position would then be calculated from formula C.11 as follows:

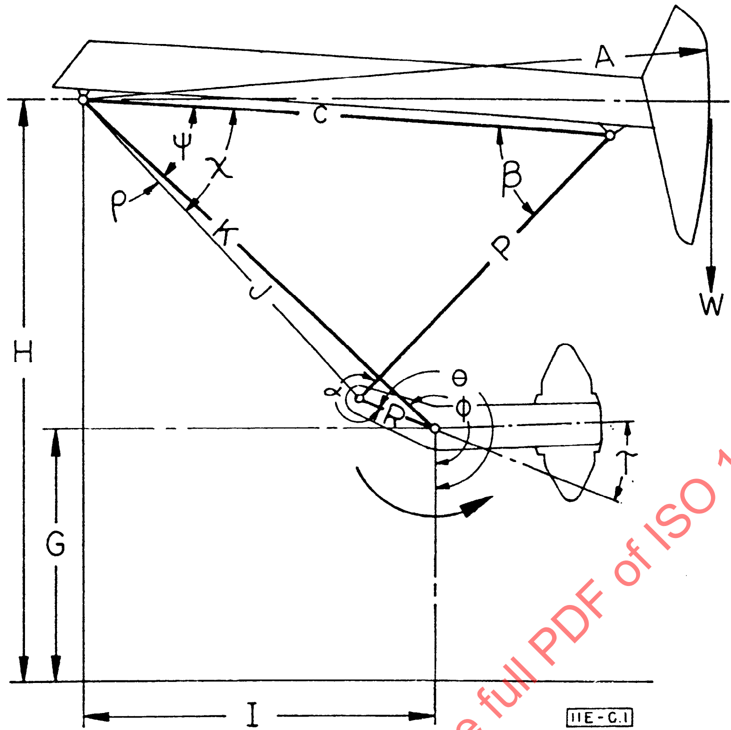
$$
\begin{aligned}
T_n &= \overline{TF}\,(W - B) - M \sin(\theta + \tau) \\
&= T_{wn} - T_r \\
&= 326{,}592 - 263{,}744 = 62{,}848 \text{ in.—lb.}
\end{aligned}
$$

These values may be calculated for other crank angle positions in the same manner as outlined above. Shown in Fig. C3 is a plot of torque vs. crank angle which includes the net polished-rod load torque curve, the counterbalance torque curve, and the net crankshaft torque curve.

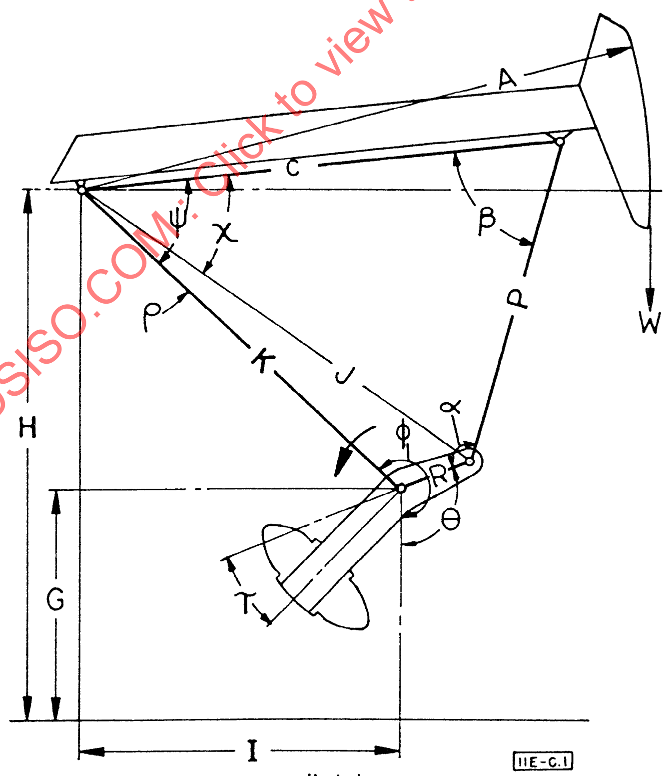
C16. The foregoing technique is generally accepted. Those wanting more precise results must realize the true stroke length can vary somewhat with a change in beam position in relation to the centerline of the saddle bearing due to an adjustable feature provided on most medium to large sized units or 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.

C17. 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.

C18. 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° divisions is permissible without significant error.



Downstroke



Upstroke

FIG. C1

FRONT MOUNTED GEOMETRY, CLASS III LEVER SYSTEM

See Para. C4 for definition of symbols.

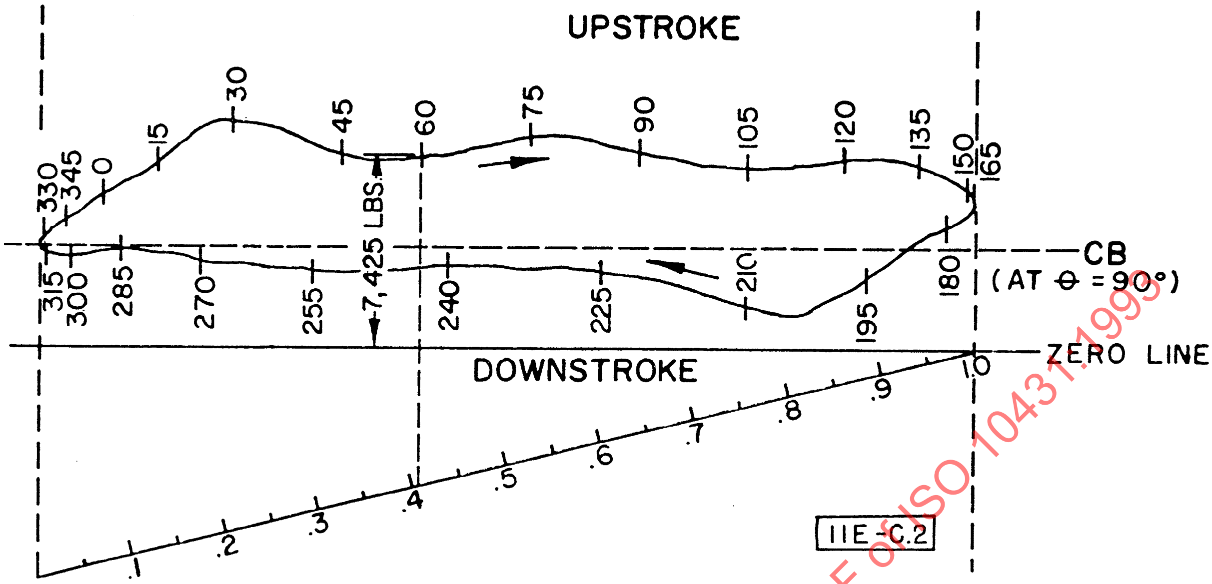


FIG. C2
DIVISION OF DYNAMOMETER CARD BY CRANK ANGLE USING API POLISHED-ROD POSITION DATA

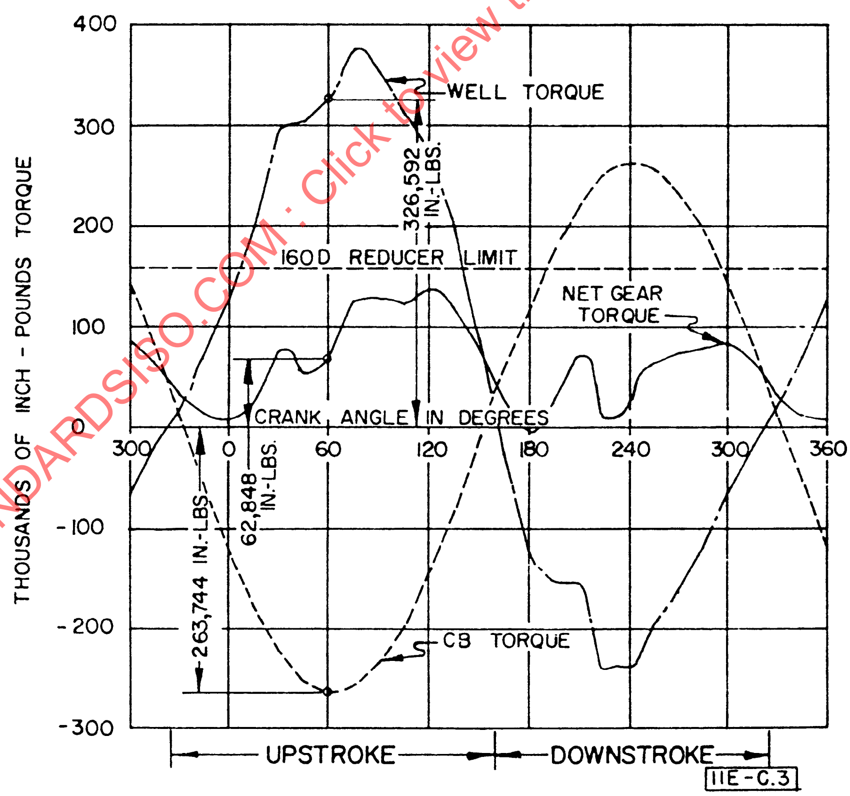


FIG. C3
TORQUE CURVES USING API TORQUE FACTORS

APPENDIX D

RECOMMENDED PRACTICE FOR THE CALCULATION AND APPLICATION OF TORQUE FACTOR ON PUMPING UNITS

(Front Mounted Geometry Class III Lever System Air Counterbalance)

Definition

D1. The torque factor for any given crank angle is that factor which, 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.

Method of Calculation

D2. 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 are given in Fig. A2, Appendix A.

D3. Torque factors and polished-rod positions are to be furnished by pumping-unit manufacturers for each 15-deg 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.

D4. Referring to Fig. D1, 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 crank-pin 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 crank-pin bearing to the center of the Samson Post bearing, inches.

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

ϕ = Angle between the 6 o'clock position and K, degrees; equals

$$180° - \tan^{-1}\left[\frac{I}{H-G}\right]$$

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

β = Angle between C and P, degrees.

α = Angle between P and R, degrees, measured counterclockwise from R to P.

Ψ = Angle between C and K, degrees, (equals angle χ — angle ρ).

Ψ_t = Angle between c and k, degrees, at top (highest) polished rod position.

Ψ_b = Angle between c and k, degrees, at bottom (lowest) polished rod position.

χ = Angle between C and J, degrees.

ρ = Angle between K and J, degrees.

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

$\overline{W}$ = Polished-rod load at any specific crank angle θ, pounds.

W_c = Counterbalance effect at the polished rod at any specific crank angle θ, pounds. Equals $M(P_a—S)$

T_n = Net torque, inch-pounds, at the crankshaft for a given crank angle θ.

M = Geometry constant for a given unit, sq. in. (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 θ.

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 θ.

D5. By application of the laws of trigonometric functions, the following expressions are derived. All angles are calculated in terms of a given crank angle θ.

$$\overline{TF} = \frac{AR}{C} \frac{\sin\alpha}{\sin\beta} \quad \text{D.1}$$

Sin α is positive when the angle α is between 0 deg and 180 deg, and is negative when angle α is between 180 deg and 360 deg.

Sin β is always positive because the angle β is always between 0 deg and 180 deg. A negative torque factor (TF) only indicates a change in direction of torque on the crankshaft.

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

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 deg and 90 deg moving clockwise, and is negative from 90 deg to 270 deg moving clockwise. When the angle $(\theta-\phi)$ is negative, it should be subtracted from 360 deg, 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 ρ is taken as a positive angle when sin ρ is positive. This occurs for crank positions between $(\theta - \phi) = 0$ deg to $(\theta - \phi) = 180$ deg.

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

$$\Psi = \chi + \rho \quad \text{D.6}$$

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

$$\overline{PR} = \frac{\Psi_b - \Psi}{\Psi_b - \Psi_t} \quad \text{D.8}$$

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

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

Application of Torque Factors

D6. 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 formula:

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

D7. The formula 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, crank; and neglects friction in the Samson 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 non-dynamic 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 paragraphs D14, D15, and D16.

D8. To illustrate the use of torque factors, a sample calculation will be made. A dynamometer card taken on a 5560 ft well is shown in Fig. D2. 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 deg of crank angle θ. Lines are projected down from the ends of the card, as shown, to determine its length which is proportional to the length of the stroke.

The length of the base line 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.*

D9. 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 psig and the minimum air pressure at the top of the stroke, 180 degree crank position, was 262 psig. Using the formula, $W_c = M(P_a - S)$ where $M = 52.5$ in.^{2} and S is 73 psig (as furnished by the pumping unit manufacturer) we calculate the following results:

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 lbs per inch gives us 1.185 inches.

Minimum counterbalance at the 180 degrees crank position is $W_c = 52.5\ (262-73) - 9923$ pounds. 9923 divided by 11,300 lbs per in. gives us .878 inch.

The counterbalance line can now be drawn on the dynamometer card as shown in Fig. D2.

D10. To further illustrate, a calculation will be made considering the point where the crank angle θ equals 75 deg. From polished-rod stroke and torque factor data for the particular 86 in. stroke 320-D Pumping Unit used for this example, it is found that the position of the polished rod at 75 deg 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 (Fig. D2). The dynamometer deflection at this point is read to be 1.45 in. which, with a scale constant of 11,300 lb. per in., makes the load, (W) at that point 16,385 lb.

D11. In a similar manner, the polished-rod load may be obtained for each 15-deg angle of crank rotation. The dynamometer card has been marked to show the load and position involved for each 15 deg 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 degrees crank position. The maximum dynamometer deflection at this point is 1.60 inches which when multiplied by the scale constant of 11,300 lb. per inch gives 18,080 pounds polished rod load.

D12. The net torque, T_n, can now be determined. In the formula $T_n = \overline{TF}\,(W - W_c)$, the value

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

(W—W_c) is represented by the difference in the dynamometer deflection between the card and the counterbalance line. Referring to the card in Fig. D2, we read the difference in the dynamometer deflection between the counterbalance line and the well card as .36 inch at 75° crank position. This value multiplied by the scale constant of 11,300 and the torque factor of 39.25 at 75° crank position gives 159,669 in-lb net torque. These values may be calculated for other crank positions in the same manner. Fig. D3 is a plot of the net torque curve.

D13. 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. (Fig. A2, Appendix A, Std 11E.) The position of crank, degrees, (Col. 1) is reversed, starting from the bottom with 15 deg and counting up in 15-deg increments to 360 deg.

D14. 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 sized 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.

D15. 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.

D16. 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° divisions is permissible without significant error.

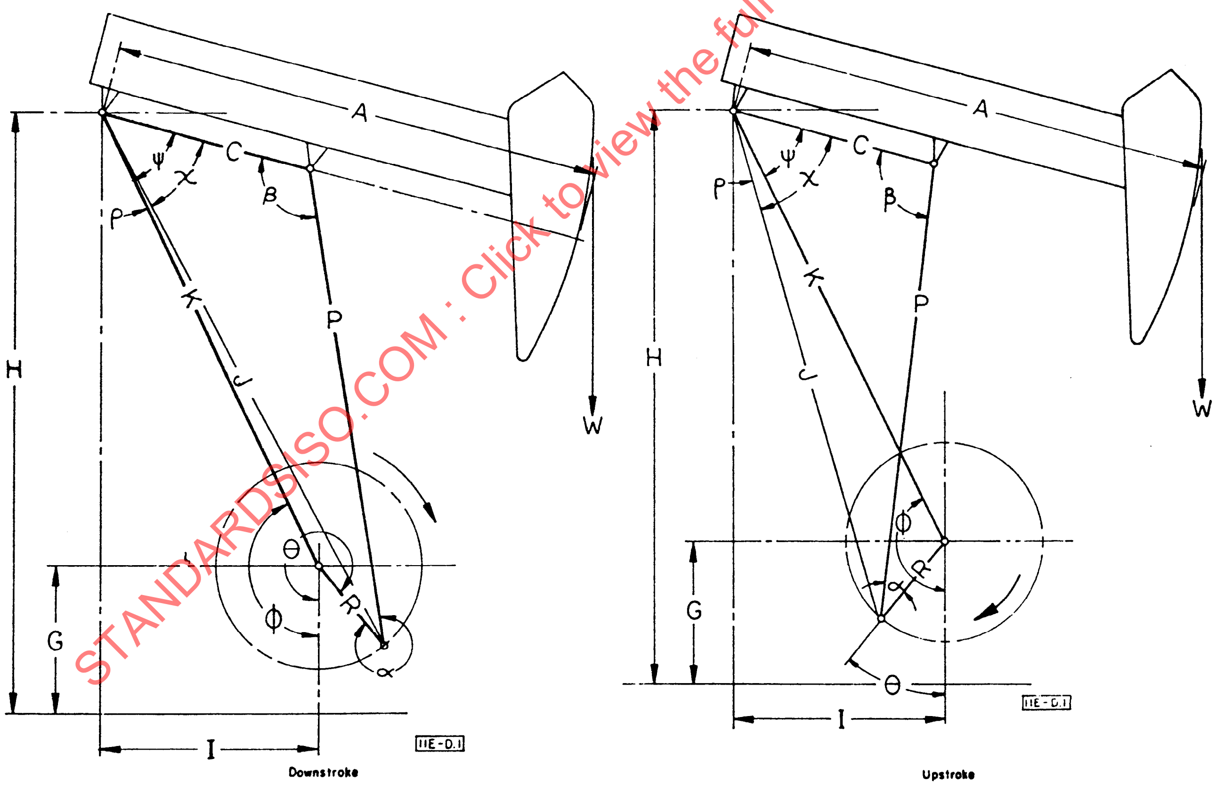


FIG. D1
PUMPING UNIT GEOMETRY
See Para. D4 for definition of symbols

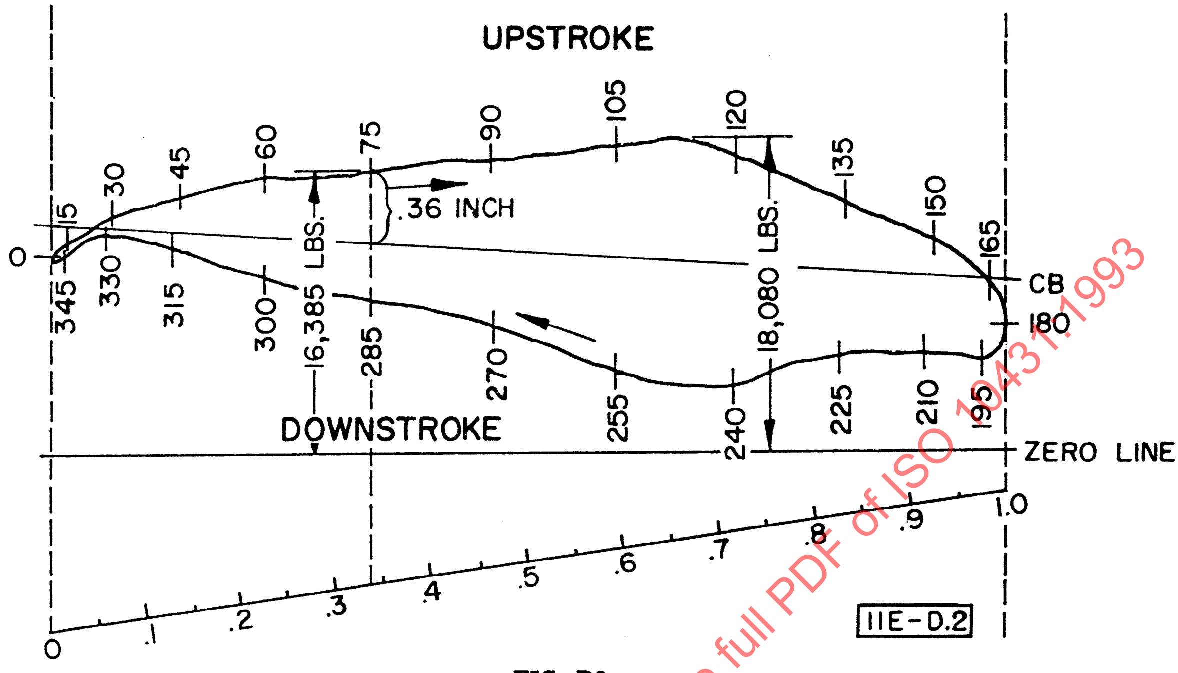


FIG. D2
DIVISION OF DYNAMOMETER CARD BY CRANK ANGLE USING API POLISHED-ROD POSITION DATA

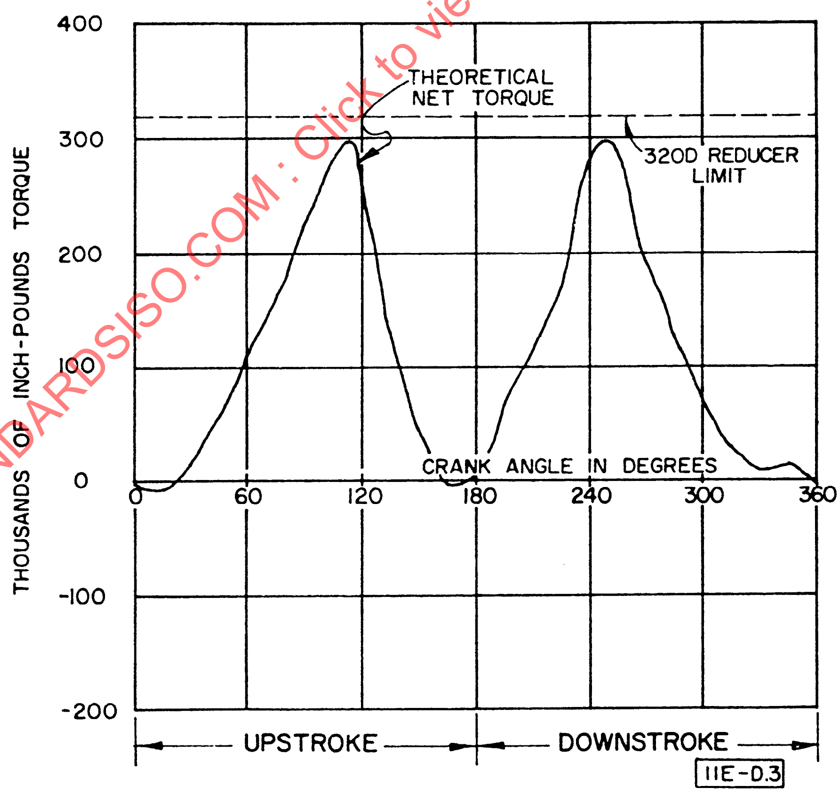


FIG. D3
TORQUE CURVES USING API TORQUE FACTORS