BEAM PUMPING FUNDAMENTALS

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PREFACE

Continental Oil Company has found that the majority of the problems associated with sucker rod pumping are a direct result of ignoring rudimentary and basic principles. To aid production employees in acquiring and comprehending these fundamental principles, twenty-seven sessions of three types of well pumping short courses have been presented since November 1961, to a total of 470 operating personnel, test engineers, test engineer candidates, and engineers.

The results and benefits of the well pumping short courses include:

1. Improved pumping equipment design.
2. Increased well pumping efficiency.
3. Increased production.
4. Elimination of unnecessary pulling costs.
5. Longer rod and pump life.
6. Less frequent gear box failures.
7. Technical knowledge of test engineers, field operating personnel and engineers has been upgraded.
8. Field personnel have recognized the value of dynamometer and fluid level equipment and are taking the initiative in requesting the services of test engineers.
9. A better climate of cooperation has been established with respect to well pumping problems.
10. A trained engineer is now available in each producing office to furnish technical advice and assistance to test engineers and to design well pumping equipment and recommend operating practices.

It is believed that these results and benefits indicate our approach is sound. It is our conclusion that fundamentals must be presented, and these fundamentals must be presented in a stepwise understandable manner.

Our total well pumping course—as it would be presented in a one-week session—includes, but is not confined to, the seventeen subjects as listed in "Contents of Course and Teaching Techniques." The course presented herein includes the 15 subjects, Sections III through XVII, as listed in the Table of Contents. In addition, several of the "steps" in the stepwise analysis being taught have been shortened in the interest of time. However, there should be sufficient "meat" left in this course to satisfy all but the most hardy of sucker rod pumping analysts.

TABLE OF CONTENTS

I.

INTRODUCTION

Since 1961, Continental Oil Company has presented short courses to assist production employees in learning and applying the basic principles of sucker rod pumping.

The well pumping knowledge of the participant has been increased by this formal training and by post-course application. At a result of this training and subsequent application, technological advances have been realized, and it has become necessary and advisable to continually upgrade the material presented. For example, the last four sessions of the course incorporate the work of the Committee on Standardization of Producing Equipment of the Division of Production of the American Petroleum Institute, as presented in API RP 11L, "Recommended Practice for Design Calculations for Sucker Rod Pumping Systems (Conventional Units)," First Edition.

The material presented in this paper is selected from subject matter presented at the most recent sessions. Also discussed is the importance of the teaching techniques used before and during the sessions.

II.

CONTENTS OF COURSE AND TEACHING TECHNIQUES

In general, the courses cover the folllowing items and subjects, but each session is tailored to the audience level and needs of the participants in that particular session. All or part of the following subjects are covered in a stepwise fashion:

1. Description of dynamometer and fluid level instruments.
2. Typical loads during the pumping cycle which are critical to dynamometer card interpretation (building-block approach).
3. Pumping unit geometry, moments and torque factors.
4. Calculation of instantaneous net torque for both clockwise and counterclockwise rotation.
5. Counterbalancing.
6. Pumping unit efficiency.
7. Horsepower calculations.
8. Dynamometer card orders and prediction of actual orders.
9. Dynamometer card interpretation.
10. Fluid pound symposium.
11. Gas lock symposium.
12. Surface equipment selection and design.
13. Sucker rod symposium.
14. Subsurface pump symposium.
15. Inflow performance relationship curves as used to determine well capacity.
16. Gas Anchor design.
17. Systematic approach to solving well pumping problems.

The course teaching method is based on conferee participation. Approximately one month prior to the session, pre-session homework is assigned of which the API Division of Production PROFIT Series "Well Pumping" is a part. For those requiring it, "Applied Mathematics for the Petroleum Industry," published by the Petroleum Extension Service, University of Texas, in cooperation with other organizations, is also furnished at the time the first homework is forwarded. Approximately two weeks later, additional reference material and a second set of homework problems are mailed each student. With this thorough preparation, the audience level of the session can be pre-set, and the formal presentation of the course can commence at a more advanced level than would otherwise be possible.

The actual session is typified by continuous group participation. The Provincial Step or IPAT method is the teaching method primarily used, but lectures, demonstrations, illustrations and group discussions are used as supplementary methods. The Provincial Step or IPAT method is divided into four parts: (1) the introduction of the subject in which no new information is presented, (2) the presentation of pertinent subject information to be discussed, (3) the application of the information presented in some manner, and (4) some form of testing to determine the degree of comprehension of the subject presented. Problem solving, both individually and by groups, is used throughout the course.

Participant reaction and "feedback" are necessary and are secured daily during the session. One of the easiest traps into which an instructor can fall is that of believing the material being presented is understood by the group. Four primary methods have been used to assure that true feedback is received continuously. The first of these is a "morning report." The class is divided into work groups with preferably not more than four members in a group. Each group selects its own leader. After each day's session is completed, the group meets and discusses the subjects presented and evaluates the effectiveness of instruction. Those items or subjects not clear, or on which additional information or discussion is desired, are determined. Each group reports these items through its leader at the beginning of the session the following morning. These reports usually provide the instructors with an excellent opportunity of reviewing the salient subjects presented the previous day. In addition each participant submits his own comments on a daily critique sheet.

After the morning report period, a short, to-the-point, practical and pertinent quiz is given. Most of the time this will involve a problem in which the principles or subjects previously learned are applied. In addition, new subjects concerning previously assigned homework are introduced. The new subject may occupy the remainder of the morning. Quizzes are of the open-book type but are completed on an individual basis.

Direct questioning and group problem solving are the other two means of providing the instructor with the needed feedback. In the event a subject has not been presented in an understandable manner or is not understood, it is re-covered. The agenda is constructed with this flexibility.

III.

WELL LOADS CRITICAL TO DYNAMOMETER CARD INTERPRETATIONS

There are six basic loads which are critical to dynamometer card interpretation. It is possible to pre-calculate these loads in advance of actual well weighing operations, and the necessity for calculating these theoretical loads in advance cannot be emphasized too strongly. These loads, when considered with other indicators, can be used to diagnose operating and design problems. Four of these loads are measured under static conditions while the other two are measured under dynamic operating conditions.

The following definitions are applicable in determining static and dynamic polished rod loads:

DEFINITIONS

R = Reference line drawn on every card by the dynamometer reference stylus.

O = Zero line drawn on the card only when there is zero load on the dynamometer.

W_r = Weight of the sucker rod string in air, pounds per foot.

W = Total weight of the sucker rod string in air, pounds.

W_{rf} = Total weight of the sucker rods in well fluid, pounds.

L = Length of the sucker rod string, in feet.

0.128 = Weight of a cubic foot of fresh water, 62.4 pounds divided by the weight of a cubic foot of steel, 489 pounds.

G = The specific gravity of the fluid in the tubing above the pump.

0.434 = Weight of a column of fresh water 1 in. sq. and 1 ft. high, in pounds.

0.34 = 0.434 x 3.1416/4.

F_o = The static fluid load, in pounds per foot, on the gross plunger area multiplied by H, the net lift in feet.

F_1 = Fluid load on the gross plunger area plus maximum upstroke dynamic effects, pounds.

F_2 = Dynamic effects on the downstroke, pounds.

D = The diameter of the pump plunger, in inches.

H = Net lift, approximated by the distance from the surface of the ground to the operating fluid level in the tubing-casing annulus, in feet.

SV = The static load at the polished rod, in pounds, when the standing valve is closed and the traveling valve is open.

TV = The static load at the polished rod, in pounds, when the traveling valve is closed and the standing valve is open.

PPRL = The peak load at the polished rod, in pounds, during the pumping cycle.

MPRL = The minimum load at the polished rod, in pounds, during the pumping cycle.

SK_r = Static load necessary to stretch the total rod string an amount equal to the polished rod stroke.

CBE = Counterbalance effect, pounds.

ZERO LOAD

The zero (0) load is scribed prior to the time the well load is placed on the dynamometer. It is always good operating practice to obtain another zero line at the conclusion of the well weighing operation as a check against possible dynamometer malfunction.

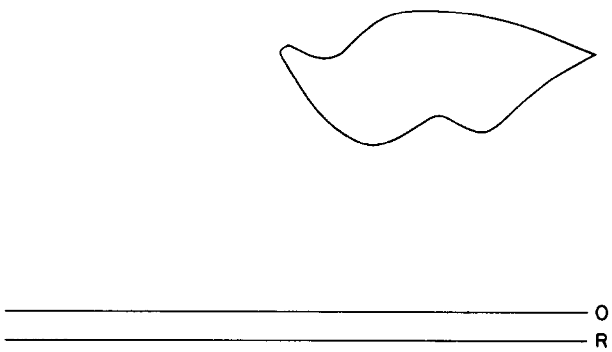


FIG. 1 — ZERO LOAD

STANDING VALVE LOAD

Although the standing valve is not actually measured, the effect of the weight of rods suspended in well fluid is measured and can also be calculated. That load is called the standing valve (SV) load. It is one of the two most important loads in dynamometer card interpretation. It is comprised of two basic components: (1) the weight of the sucker rod string in air (W), minus (2) the buoyancy effect (W x 0.128 x G).

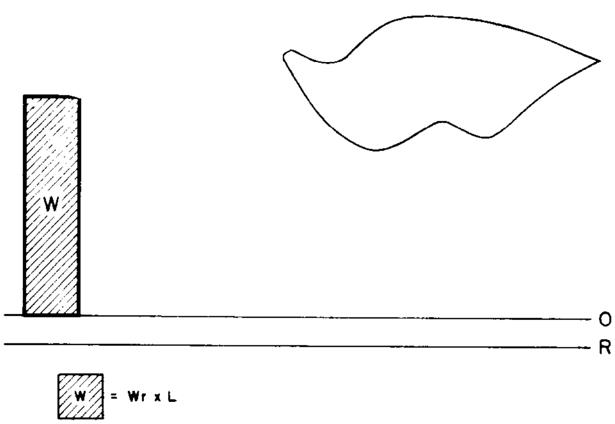


FIG. 2 — WEIGHT OF SUCKER RODS IN AIR

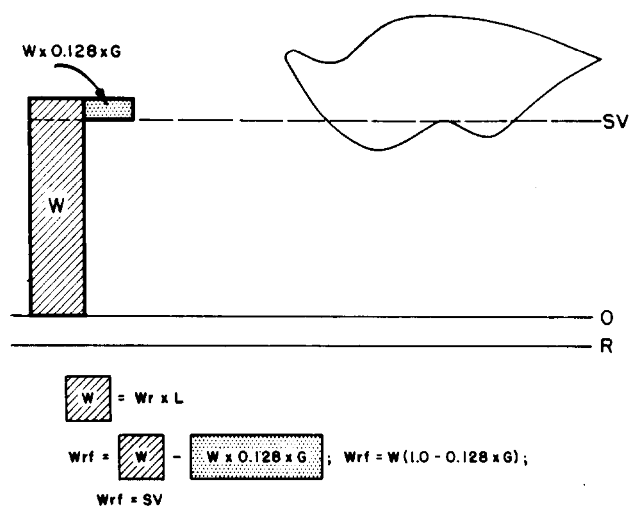


FIG. 3 — STANDING VALVE LOAD
(Weight of Sucker Rods in Well Fluid)

TRAVELING VALVE LOAD

The traveling valve (TV) load is comprised of the weight of the sucker rod string in air (W), minus the buoyancy effect (W x 0.128 x G), plus the net lift weight of the well fluid on the gross plunger area (W_f /ft x H = F_o). It can be precalculated and also measured. It is the other of the two most important loads critical to dynamometer card interpretation.

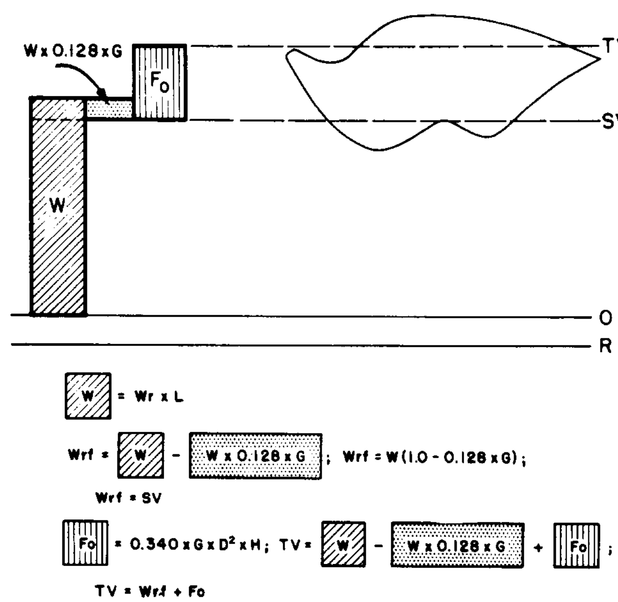


FIG. 4 — TRAVELING VALVE LOAD

PEAK POLISHED ROD LOAD

Four basic loads are involved in the peak polished rod load (PPRL). These are: (1) the weight of the sucker rods in air (W), minus (2) the buoyancy effect (W x 0.128 x G), plus (3) the weight of the well fluid on the gross plunger area (F_o), plus (4) certain dynamic effects on the upstroke. These latter two are combined and called F_1.

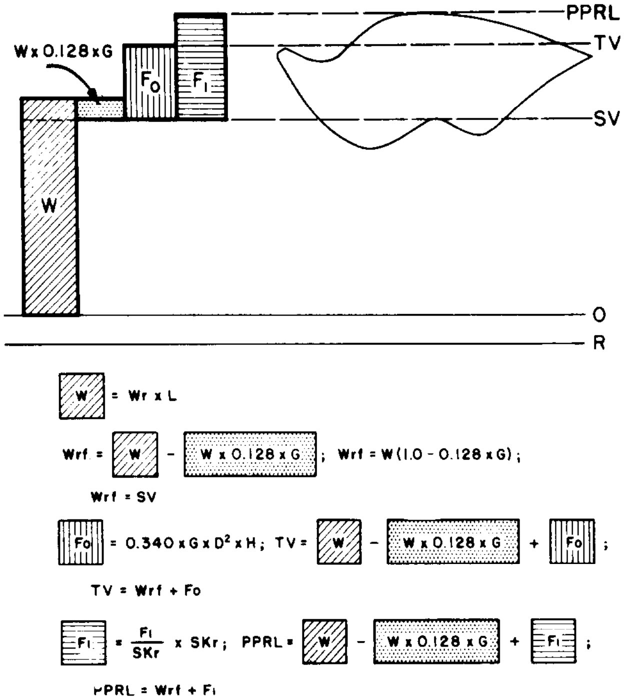


FIG. 5 — PEAK POLISHED ROD LOAD

MINIMUM POLISHED ROD LOAD

The new API method makes it possible to calculate the minimum polished rod load (MPRL) much more accurately. It is comprised of three basic loads: (1) the weight of the sucker rods in air (W), minus (2) the buoyancy effect (W x 0.128 x G), minus (3), the dynamic effects (F_2) on the downstroke.

COUNTERBALANCE EFFECT

The counterbalance effect (CBE) is measured under static conditions for convenience but is applied under dynamic conditions. The following counterbalance effect formula is given in API RP11L:

$$CBE = 1.06\ (W_{rf} + \tfrac{1}{2} F_o)$$

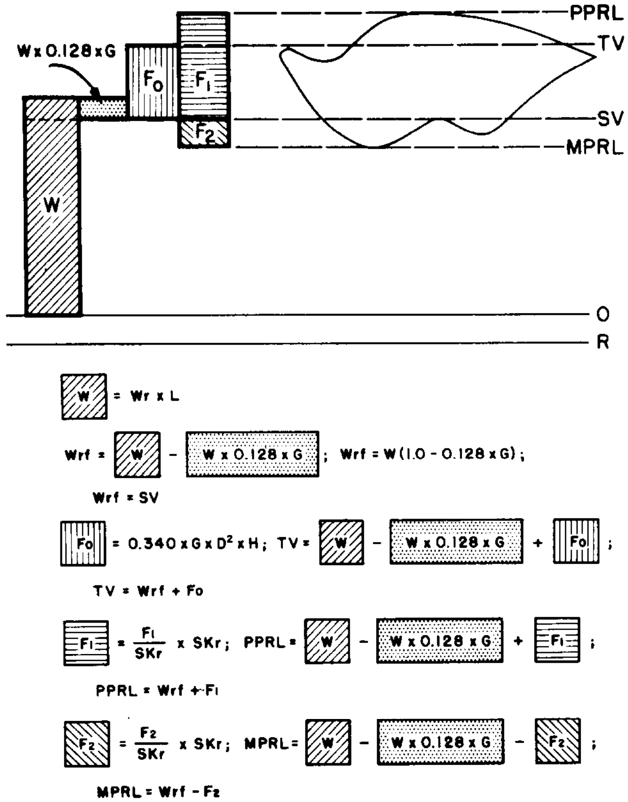


FIG. 6 — MINIMUM POLISHED ROD LOAD

There is probably as much over-all profit to be made by keeping the proper counterbalance effect on pumping wells as on any other item covered in this paper, with the possible exception of correct sizing of subsurface pumps.

The attached form, Fig. 7, "Design Calculations Sheet, Conventional Sucker Rod Pumping System", is used to pre-calculate loads and parameters for the new API method. Figure 7a, "Dynamometer and/or Fluid Level Sounder Test Report", is used to report complete individual well data, compare the pre-calculated and actual loads critical to dynamometer card interpretation, and make recommendations for corrective action.

DESIGN CALCULATIONS SHEET
CONVENTIONAL SUCKER ROD PUMPING SYSTEM

Well__________________ Date__________________ Calculated By__________________

Known or Assumed Data:

Fluid Level, H = __________ft. Pump Depth, L = __________ft. Tubing Size__________in.
Tubing Anchored? Yes_____ No. _____ Depth_____ft. Pumping Speed, N = __________ SPM
Length of Stroke, S = __________in. Plunger Diameter, D = __________ in.
Specific Gravity of Fluid, G = __________ Sucker Rods__________
API Class: C, D, S.S., K, H.T. (Circle one)

Record Factors from Tables 1 & 2:

1. W_r = __________ (Table 1, Column 3)
2. E_r = __________ (Table 1, Column 4)
3. F_c = __________ (Table 1, Column 5)
4. E_t = __________ (Table 2, Column 5)

> Note: With the exception of Figure 14a in Line 22, all references to Figures and tables on this calculation sheet correspond to API RP11L.

Calculate Non-Dimensional Variables:

5. F_o = .340 x G x D^2 x H = .340 x _____ x _____ x _____ = _____ lbs. (Gross Plunger Load)
6. $1/K_r$ = E_r x L = _____ x _____ = _____ in/lb (line 2 x L)
7. SK_r = S ÷ $1/K_r$ = _____ ÷ _____ = _____ lbs. (S/line 6)
8. F_o/SK_r = _____ ÷ _____ = _____ (line 5/line 7)
9. N/N_o = NL ÷ 245,000 = _____ x _____ ÷ 245,000 = _____
10. N/N_o' = N/N_o ÷ F_c = _____ ÷ _____ = _____ (line 9/line 3)
11. $1/K_t$ = E_t x L = _____ x _____ = _____ in/lb (line 4 x L)

Solve for S_p and PD:

12. S_p/S = __________ (Figure 2) (line 10 to line 8 to answer)
13. S_p = $[(S_p/S) \times S]$ - $[F_o \times 1/K_t]$ = $[\underset{\text{(line 12)}}{\_\_\_\_} \times \underset{\text{(S)}}{\_\_\_\_}]$ - $[\underset{\text{(line 5)}}{\_\_\_\_} \times \underset{\text{(line 11)}}{\_\_\_\_}]$ = _____ in.
14. PD = 0.1166 x S_p x N x D^2 = 0.1166 x $\underset{\text{(line 13)}}{\_\_\_\_}$ x $\underset{\text{(N)}}{\_\_\_\_}$ x $\underset{(D^2)}{\_\_\_\_}$ = _____ bbls per day.

Determine Non-Dimensional Parameters:

15. W = W_r x L = _____ x _____ = _____ lbs (line 1 x L)
16. W_{rf} = W [1-(.128G)] = _____ [1-(.128 x _____)] = _____ lbs.
17. W_{rf}/SK_r = _____ ÷ _____ = _____ (line 16/line 7)

Record Non-Dimensional Factors from Figures 3 through 7:

18. F_1/SK_r = __________ (Figure 3) (line 9 to line 8 to answer)
19. F_2/SK_r = __________ (Figure 4) (line 9 to line 8 to answer)
20. $2T/S^2K_r$ = __________ (Figure 5) (line 9 to line 8 to answer)
21. F_3/SK_r = __________ (Figure 6) (line 9 to line 8 to answer)
22. T_a = Torque Adjustment for Peak Torque for Values of W_{rf}/SK_r other than 0.3
    - a. T_a = 1.00 ± [(%/100) x ((W_{rf}/SK_r) - 0.3) x 10] (This is the general formula.)
    - b. (%/100) = (± _____/100) = (± _____) (Figure 7)(% is intersection of line 10 and line 8)
    - c. $[(W_{rf}/SK_r) - 0.3]$ = $[\underset{\text{(line 17)}}{\pm\_\_\_\_} - 0.3]$ = [± _____]
    - d. T_a = 1.00 ± $[(\underset{\text{(line 22b)}}{\_\_\_\_} \times \underset{\text{(line 22c)}}{\_\_\_\_}) \times 10]$ = 1.00 ± _____
    - e. T_a = _____

    Ta Can Also Be Determined Graphically As Follows:
    - f. % = _____ (Figure 7)(Intersection of lines 10 and 8 is %)
    - g. T_a = _____ (Figure 14a)(From % on Fig. 7 to W_{rf}/SK_r from line 17 to T_a)

Solve for Operating Characterictics:

23. PPRL = W_{rf} + $[(F_1/SK_r) \times SK_r]$ = $\underset{\text{(line 16)}}{\_\_\_\_}$ + $[\underset{\text{(line 18)}}{\_\_\_\_} \times \underset{\text{(line 7)}}{\_\_\_\_}]$ = _____ lbs.
24. MPRL = W_{rf} - $[(F_2/SK_r) \times SK_r]$ = $\underset{\text{(line 16)}}{\_\_\_\_}$ - $[\underset{\text{(line 19)}}{\_\_\_\_} \times \underset{\text{(line 7)}}{\_\_\_\_}]$ = _____ lbs.
25. PT = $(2T/S^2K_r)$ x SK_r x S/2 x T_a = $\underset{\text{(line 20)}}{\_\_\_\_}$ x $\underset{\text{(line 7)}}{\_\_\_\_}$ x $\underset{(S/2)}{\_\_\_\_}$ x $\underset{\text{(line 22)}}{\_\_\_\_}$ = _____ lb. in.
26. PRHP = $(F_3/SK_r) \times SK_r \times S \times N \times 2.53\times10^{-6}$ = $\underset{\text{(line 21)}}{\_\_\_\_}$ x $\underset{\text{(line 7)}}{\_\_\_\_}$ x $\underset{(S)}{\_\_\_\_}$ x $\underset{(N)}{\_\_\_\_}$ x 2.53×10^{-6} = _____
27. CBE = 1.06 (W_{rf} + 1/2 F_o) = 1.06 x ($\underset{\text{(line 16)}}{\_\_\_\_}$ + $\underset{\text{(line 5/2)}}{\_\_\_\_}$) = _____ lbs.
28. (PPRL - MPRL) x 100/PPRL = __________% [(line 23 - line 24) x 100/line 23]

(Revised: 1-6-69)

FIG. 7

DYNAMOMETER AND/OR FLUID LEVEL SOUNDER TEST REPORT

Well______________________________ Pool & Form.______________________________
TD/PD____________Ft., Interval Open to Production ____________Ft. to ____________ Ft.

EQUIPMENT DATA

Pump Size______________________________ Type______________________________
Pump Set @__________', SPM__________ SL__________ Possible Stroke Lengths__________
Pumping Unit: Make & Size____________________ Gear Ratio __________
Ratings: Gear Box__________"# Beam__________# CB__________#
Prime Mover: Type__________ Size__________ RPM__________
Sheave Sizes: Pumping Unit__________" Prime Mover__________" Tbg.Size__________"
Number of Belts ____ ____Rods: %________No.________ x __ = ________ x ____ = ________#
Sucker Rod Design: 1" Rods: %________No.________ x 25 = ________ x 2.90 = ________#
API Class: D, 7/8" Rods: %________No.________ x 25 = ________ x 2.22 = ________#
C,K, S.S., H.T. 3/4" Rods: %________No.________ x 25 = ________ x 1.63 = ________#
(Circle one) 5/8" Rods: %________No.________ x 25 = ________ x 1.13 = ________#
Calculated Total Weight of Rods in Air (W) = ________#

PRODUCTION DATA

Pumping is: Continuous__________ Intermittent__________(check one)
Well is pumped________ min. on and ________ min. off, ________ hrs/day or ________% of 24 hours
Type of pumping time control______________________________.
Daily Allowable:____________________BOPD Top Possible Allowable ____________________ BOPD.
Actual Production: Oil__________B/D, Water__________B/D, Total Fluid__________B/D__________%Water.
Normal Production: Oil__________B/D, Water__________B/D, Total Fluid__________B/D__________%Water.
Date of Production Test____________________.
Operating Fluid Level (Ft. to Fluid)____________________', Pump Submergence____________________'
T.P.__________, C.P.__________, Sp. Gr: Fluid__________ Gas__________, GOR__________
Pump Capacity (Net Plunger Travel @ 100% Vol. Eff.)____________________BFPD
Calc. Volumetric Efficiency__________%. Tubing anchored: Yes__________No__________Depth__________'

DYNAMOMETER ANALYSIS	CALC. LOADS	MEAS. LOADS
Rod Wt. in Fluid (S.V. Test)		
Fluid Wt. on Gross Plung. Area (F_o)		xxxxxxxxxxx
S.V. + F_o (T.V. Test)		
Peak Load		
Minimum Load		
Load Range		
Load Range - % of Peak Load		
Rod Stress		
Peak Torque		
Polished Rod Horsepower		
Counterbalance Effect		

FLUID LEVEL ANALYSIS		
TIME	FLUID LEVEL	PROD. RATE

Dynamometer Constants: 1" = ____________________Pounds: 1" = ____________________In. of Pol. Rod Stroke
Unit Rotation: Clockwise____________________ Counterclockwise____________________(check one)
Recommendations: (Use reverse side, if necessary, and attach dynamometer cards(s).

Distribution:______________________________ Prepared by______________________________
Date:______________________________

(Revised: 3-1-68)

FIG. 7a

(API TABLE 1)

ROD AND PUMP DATA

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1⅛	1	⅞	¾	⅝	½
44	All	0.726	1.990×10^{-6}	1.000	....	....	....	....	....	100.0
54	1.06	0.892	1.697×10^{-6}	1.128	....	....	....	....	40.5	59.5
54	1.25	0.914	1.659×10^{-6}	1.139	....	....	....	....	45.9	54.1
54	1.50	0.948	1.597×10^{-6}	1.142	....	....	....	....	54.5	45.5
54	1.75	0.990	1.525×10^{-6}	1.130	....	....	....	....	64.6	35.4
54	2.00	1.037	1.442×10^{-6}	1.095	....	....	....	....	76.2	23.8
55	All	1.135	1.270×10^{-6}	1.000	....	....	....	....	100.0	....
64	1.06	1.116	1.441×10^{-6}	1.224	....	....	....	28.1	33.1	38.8
64	1.25	1.168	1.368×10^{-6}	1.222	....	....	....	31.8	37.5	30.7
64	1.50	1.250	1.252×10^{-6}	1.191	....	....	....	37.7	44.5	17.8
64	1.75	1.347	1.116×10^{-6}	1.187	....	....	....	44.7	52.7	2.6
65	1.06	1.291	1.150×10^{-6}	1.085	....	....	....	31.3	68.7	....
65	1.25	1.306	1.138×10^{-6}	1.093	....	....	....	34.4	65.6	....
65	1.50	1.330	1.119×10^{-6}	1.103	....	....	....	39.2	60.8	....
65	1.75	1.359	1.097×10^{-6}	1.111	....	....	....	45.0	55.0	....
65	2.00	1.392	1.071×10^{-6}	1.114	....	....	....	51.6	48.4	....
65	2.25	1.429	1.042×10^{-6}	1.110	....	....	....	59.0	41.0	....
65	2.50	1.471	1.010×10^{-6}	1.097	....	....	....	67.4	32.6	....
65	2.75	1.517	0.974×10^{-6}	1.074	....	....	....	76.6	23.4	....
66	All	1.634	0.883×10^{-6}	1.000	....	....	....	100.0	....	....
75	1.06	1.511	1.030×10^{-6}	1.168	....	....	22.6	26.1	51.3	....
75	1.25	1.548	1.006×10^{-6}	1.179	....	....	24.8	28.6	46.6	....
75	1.50	1.606	0.969×10^{-6}	1.185	....	....	28.3	32.6	39.1	....
75	1.75	1.674	0.924×10^{-6}	1.180	....	....	32.4	37.4	30.2	....
75	2.00	1.754	0.874×10^{-6}	1.160	....	....	37.2	42.8	20.0	....
75	2.25	1.843	0.816×10^{-6}	1.128	....	....	42.5	49.2	8.3	....
76	1.06	1.787	0.822×10^{-6}	1.061	....	....	25.9	74.1	....	....
76	1.25	1.798	0.818×10^{-6}	1.066	....	....	27.8	72.2	....	....
76	1.50	1.816	0.811×10^{-6}	1.073	....	....	30.9	69.1	....	....
76	1.75	1.836	0.803×10^{-6}	1.080	....	....	34.3	65.7	....	....
76	2.00	1.861	0.793×10^{-6}	1.087	....	....	38.5	61.5	....	....
76	2.25	1.888	0.782×10^{-6}	1.094	....	....	43.1	56.9	....	....
76	2.50	1.919	0.770×10^{-6}	1.096	....	....	48.3	51.7	....	....
76	2.75	1.953	0.756×10^{-6}	1.096	....	....	54.1	45.9	....	....
76	3.75	2.121	0.690×10^{-6}	1.043	....	....	82.5	17.5	....	....
77	All	2.224	0.649×10^{-6}	1.000	....	....	100.0	....	....	....
85	1.06	1.709	0.957×10^{-6}	1.237	....	15.9	17.7	20.1	46.3	....
85	1.25	1.780	0.919×10^{-6}	1.250	....	17.9	19.9	22.5	39.7	....
85	1.50	1.893	0.858×10^{-6}	1.242	....	21.0	23.4	26.5	29.1	....
85	1.75	2.027	0.786×10^{-6}	1.218	....	24.8	27.5	31.0	16.7	....
85	2.00	2.181	0.703×10^{-6}	1.180	....	29.0	32.3	36.3	2.4	....
86	1.06	2.008	0.757×10^{-6}	1.127	....	19.3	21.9	58.8	....	....
86	1.25	2.035	0.748×10^{-6}	1.136	....	20.7	23.5	55.8	....	....
86	1.50	2.079	0.733×10^{-6}	1.148	....	23.0	26.0	51.0	....	....
86	1.75	2.130	0.716×10^{-6}	1.157	....	25.6	29.0	45.4	....	....
86	2.00	2.190	0.696×10^{-6}	1.162	....	28.7	32.5	38.8	....	....
86	2.25	2.257	0.674×10^{-6}	1.158	....	32.1	36.5	31.4	....	....
86	2.50	2.334	0.650×10^{-6}	1.146	....	35.8	41.6	22.6	....	....
86	2.75	2.415	0.621×10^{-6}	1.125	....	40.3	45.6	14.1	....	....

Table 1

In addition to those previously presented, the following definitions are applicable to API RP 11L issued as recommended practice for design calculations for conventional unit sucker rod pumping systems:

DEFINITIONS (continued)

S = Polished rod stroke, in inches. S is found by consulting the manufacturer's specifications, actually measuring the polished rod stroke length, or by correctly measuring the length of the dynamometer card, in inches, and multiplying that by the length constant, which is the inches of polished rod travel per inch of dynamometer card length. The latter method is the most accurate of the three, if it is performed correctly.

E_r = Elastic constant of rod string, in inches per pound foot, Table 1, (API RP 11L, Table 1, Column 4).

(API TABLE 1 (Continued))

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size					
					$1\frac{1}{8}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$
87	1.06	2.375	0.615×10^{-6}	1.048	....	22.3	77.7	....	....	....
87	1.25	2.384	0.613×10^{-6}	1.051	....	23.5	76.5	....	....	....
87	1.50	2.397	0.610×10^{-6}	1.055	....	25.5	74.5	....	....	....
87	1.75	2.414	0.606×10^{-6}	1.061	....	27.9	72.1	....	....	....
87	2.00	2.432	0.602×10^{-6}	1.066	....	30.6	69.4	....	....	....
87	2.25	2.453	0.598×10^{-6}	1.072	....	33.7	66.3	....	....	....
87	2.50	2.477	0.592×10^{-6}	1.077	....	37.2	62.8	....	....	....
87	2.75	2.503	0.586×10^{-6}	1.082	....	41.0	59.0	....	....	....
87	3.75	2.632	0.558×10^{-6}	1.082	....	60.0	40.0	....	....	....
87	4.75	2.800	0.520×10^{-6}	1.035	....	84.7	15.3	....	....	....
88	All	2.904	0.497×10^{-6}	1.000	....	100.0	....	....	....	....
96	1.06	2.264	0.698×10^{-6}	1.181	14.8	16.7	19.7	48.8	....	....
96	1.25	2.311	0.685×10^{-6}	1.203	16.0	17.8	21.0	45.2	....	....
96	1.50	2.385	0.664×10^{-6}	1.215	17.7	19.9	23.3	39.1	....	....
96	1.75	2.472	0.639×10^{-6}	1.218	19.9	22.0	25.9	32.2	....	....
96	2.00	2.572	0.610×10^{-6}	1.213	22.1	24.8	29.2	23.9	....	....
96	2.25	2.686	0.577×10^{-6}	1.197	24.9	27.7	32.6	14.8	....	....
96	2.50	2.813	0.540×10^{-6}	1.168	27.9	31.0	36.6	4.5	....	....
97	1.06	2.601	0.576×10^{-6}	1.103	17.0	19.1	63.9	....	....	....
97	1.25	2.622	0.572×10^{-6}	1.109	18.0	20.1	61.9	....	....	....
97	1.50	2.653	0.568×10^{-6}	1.117	19.3	21.9	58.8	....	....	....
97	1.75	2.696	0.558×10^{-6}	1.125	21.4	23.8	54.8	....	....	....
97	2.00	2.742	0.549×10^{-6}	1.132	23.4	26.2	50.4	....	....	....
97	2.25	2.795	0.539×10^{-6}	1.139	25.8	28.9	45.3	....	....	....
97	2.50	2.853	0.528×10^{-6}	1.144	28.5	31.7	39.8	....	....	....
97	2.75	2.918	0.515×10^{-6}	1.143	31.4	35.0	33.6	....	....	....
97	3.75	3.239	0.453×10^{-6}	1.108	45.9	51.2	2.9	....	....	....
98	1.75	3.086	0.472×10^{-6}	1.046	23.6	76.4	....	....	....	....
98	2.00	3.101	0.470×10^{-6}	1.050	25.5	74.5	....	....	....	....
98	2.25	3.118	0.468×10^{-6}	1.054	27.7	72.3	....	....	....	....
98	2.50	3.136	0.465×10^{-6}	1.058	30.1	69.9	....	....	....	....
98	2.75	3.157	0.463×10^{-6}	1.063	32.8	67.2	....	....	....	....
98	3.75	3.259	0.449×10^{-6}	1.076	46.0	54.0	....	....	....	....
98	4.75	3.393	0.431×10^{-6}	1.070	63.3	36.7	....	....	....	....
99	All	3.676	0.393×10^{-6}	1.000	100.0	....	....	....	....	....

***Rod No. shown in first column refers to the largest and smallest rod size in eighths of an inch. For example, Rod No. 76 is a two-way taper of 7/8 and 6/8 rods. Rod No. 85 is a four-way taper of 8/8, 7/8, 6/8, and 5/8 rods. Rod No. 77 is a straight string of 7/8 rods, etc.**

Table 1 (Continued)

$1/K_r$ = E_r x L = Elastic constant of total rod string in inches per pound.

SK_r = $S/(1/K_r)$ = Pounds of load (static) necessary to stretch the total rod string an amount equal to the polished rod stroke, S.

N = Pumping speed, strokes per minute (also equal to crank revolutions per minute).

N_o = Natural frequency of a non-tapered rod string, in strokes per minute.

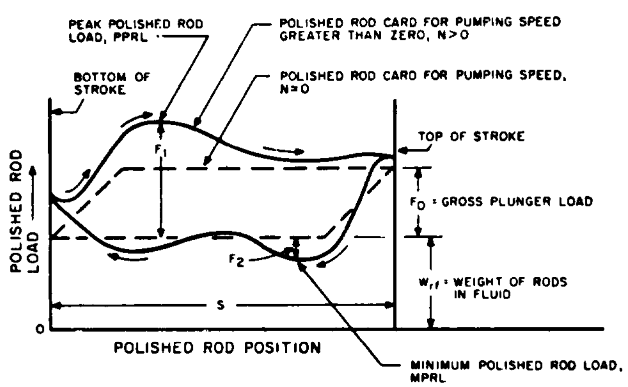
N_o' = Natural frequency of a tapered rod string, strokes per minute.

F_c = Frequency factor; F_c = 1.00 for a straight string but is greater than 1.00 for a tapered string of equal length, since the natural frequency of tapered strings is greater than the natural frequency of the same length straight string, Table 1, (API RP 11L, Table 1, Column 5).

$\frac{N}{N_o}$ = Dimensionless pumping speed.

$\frac{N}{N_o}$ = NL ÷ 245,000

$\frac{N}{N_o'} = \frac{N}{N_o} \div F_c$



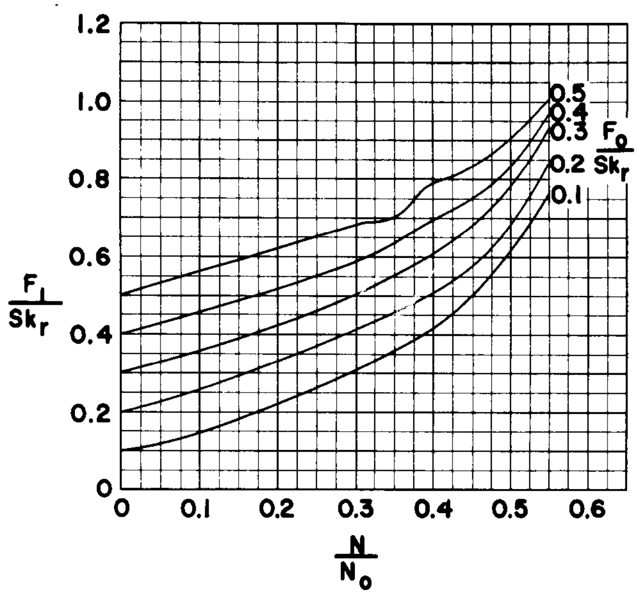
(API RP11L FIG. 1)

FIG. 8

BASIC DYNAGRAPH CARD

$\frac{F_1}{SK_r}$ is a function of $\frac{N}{N_o}$ and $\frac{F_o}{SK_r}$

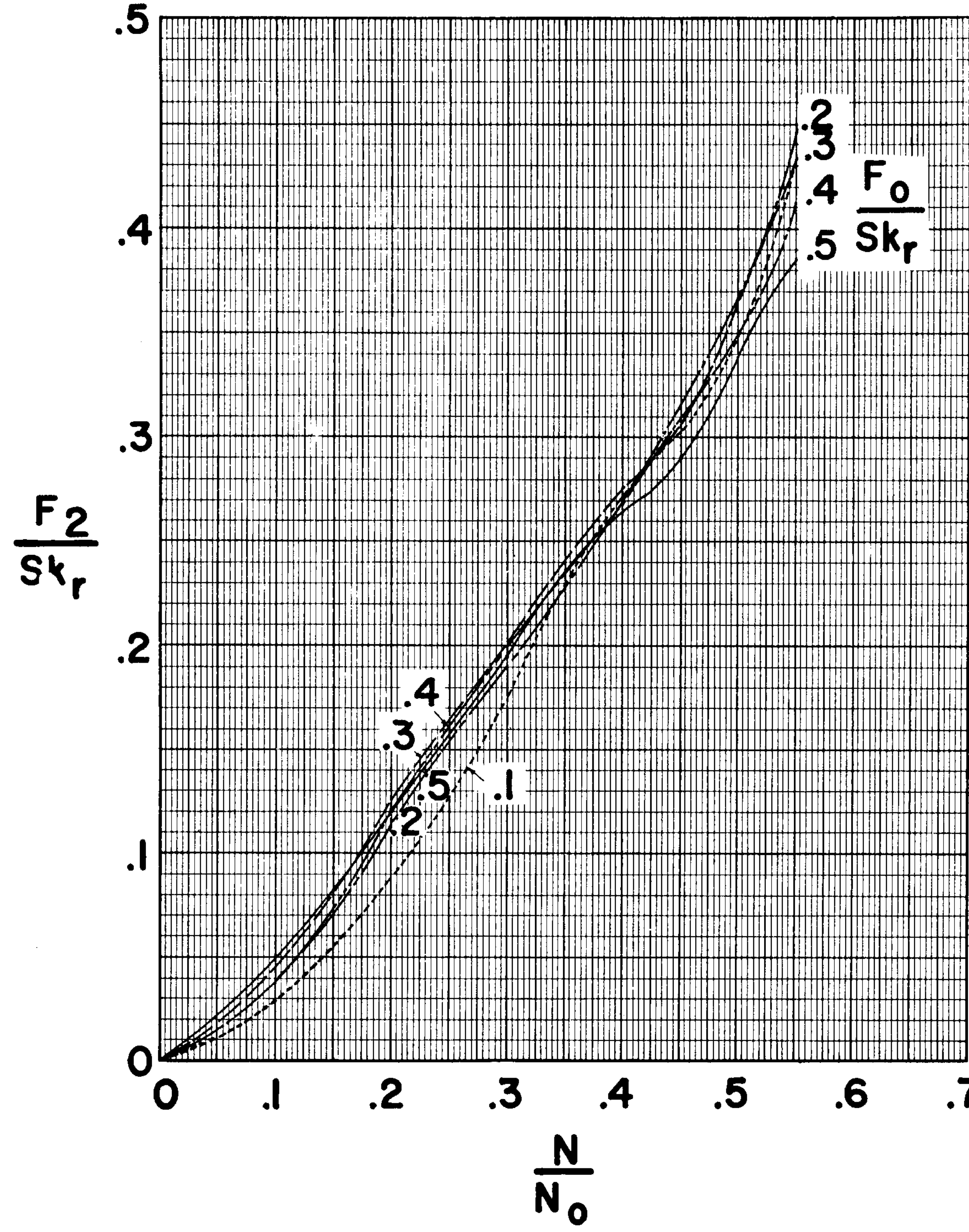
Where: F_1 = Fluid load plus maximum upstroke dynamic effects. This is added to the calculated standing valve load.



(API FIG. 3)

$\frac{F_1}{Sk_r}$, PEAK POLISHED ROD LOAD

FIG. 9



(API FIG. 4)

$\frac{F_2}{Sk_r}$, MINIMUM POLISHED ROD LOAD

FIG. 10

$\frac{F_2}{SK_r}$ is also a function of $\frac{N}{N_o}$ and $\frac{F_o}{SK_r}$

Where: F_2 = Dynamic effects on downstroke. This is subtracted from the calculated standing valve load.

$\frac{S_p}{S}$ is a function of $\frac{N}{N_o'}$ and $\frac{F_o}{SK_r}$

S_p = S_p x S (if tubing is anchored at, or very near, the standing valve).

(API TABLE 2)
TUBING DATA

1 Tubing Size	2 Outside Diameter, in.	3 Inside Diameter, in.	4 Metal Area, sq. in.	5 Elastic Constant, in. per lb ft E_t
1.900	1.900	1.610	0.800	0.500×10^{-6}
2⅜	2.375	1.995	1.304	0.307×10^{-6}
2⅞	2.875	2.441	1.812	0.221×10^{-6}
3½	3.500	2.992	2.590	0.154×10^{-6}
4	4.000	3.476	3.077	0.130×10^{-6}
4½	4.500	3.958	3.601	0.111×10^{-6}

Table 2

$\frac{F_o}{SK_r}$ = Dimensionless rod stretch.

E_t = Elastic constant for the tubing string, in inches per pound foot, Table 2 (API RP 11L, Table 2).

$\frac{1}{K_t}$ = Elastic constant for the unanchored portion of the tubing string, in inches per pound.

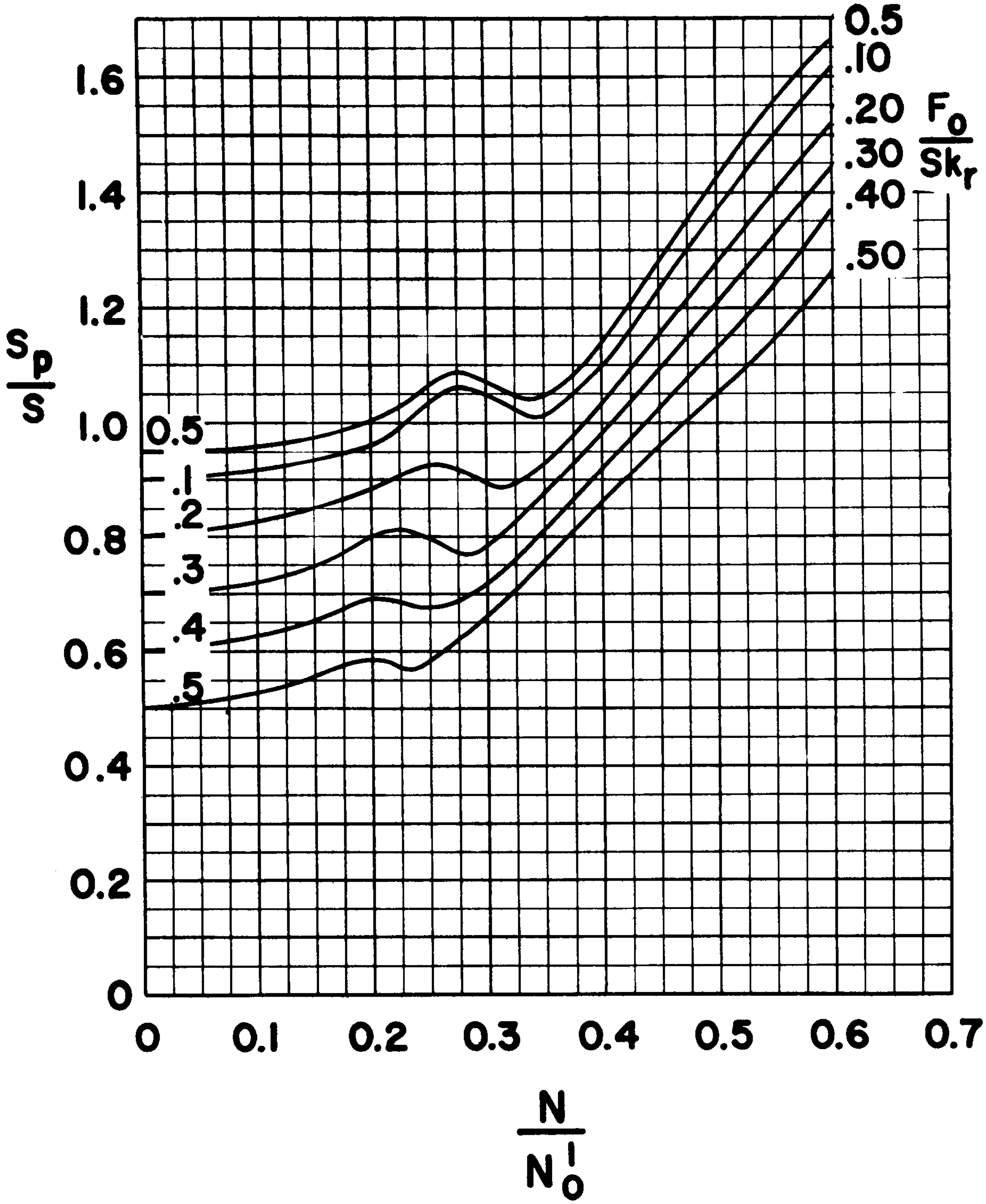
$\frac{1}{K_t}$ = E_t x L_{ua} = Elastic constant of the unanchored portion of the tubing string, in inches per pound, measured from the standing valve to the tubing anchor.

S_p = Bottom-hole pump stroke, in inches.

S_p = $\frac{S_p}{S}$ x S — (F_o x $\frac{1}{K_t}$) (This applies when the tubing is not anchored, or if the anchor is far above the standing valve. For example, if the tubing is anchored at one-half the distance from the standing valve to the surface, the value for F_o x $\frac{1}{K_t}$ would be equal to one-half of the unanchored tubing stretch and would be subtracted from $\frac{S_p}{S}$ x S.)

PD = Bottom-hole pump displacement in barrels per day, assuming 100% volumetric efficiency.

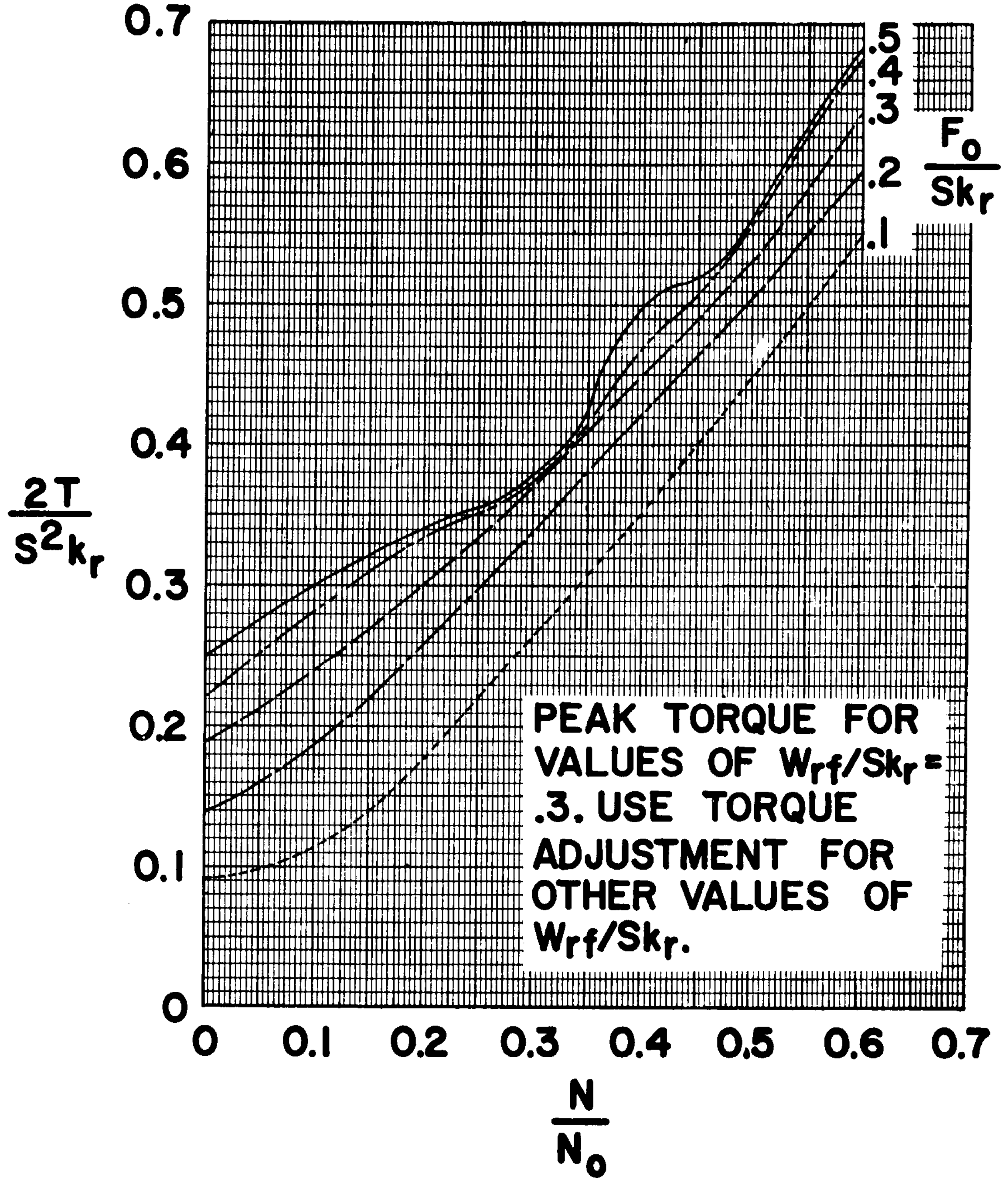
PD = $0.1166 \times S_p \times N \times D^2$



(API FIG. 2)

$\frac{S_p}{S}$, PLUNGER STROKE FACTOR

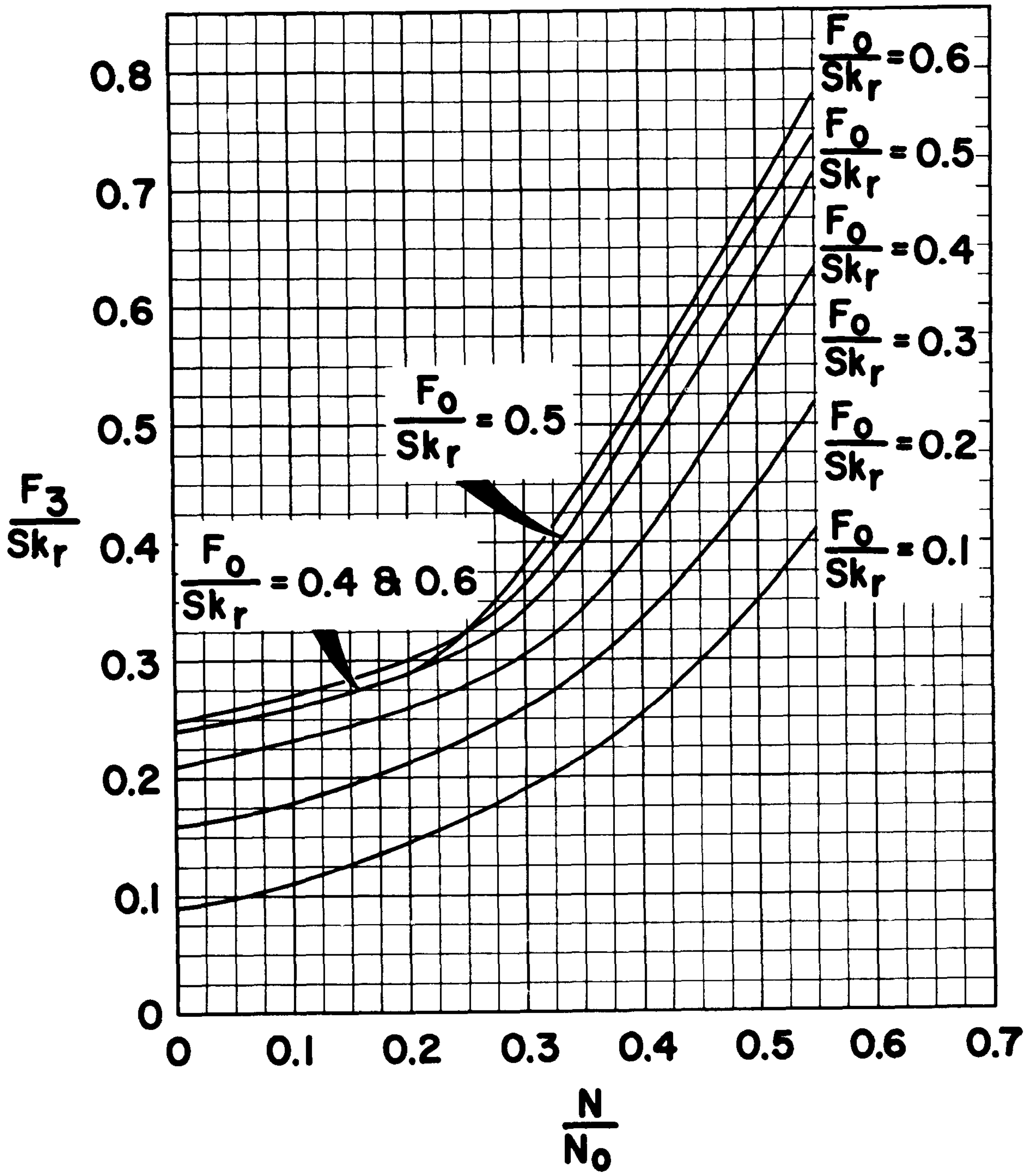
FIG. 11



(API FIG. 5)

$\frac{2T}{S^2k_r}$, PEAK TORQUE

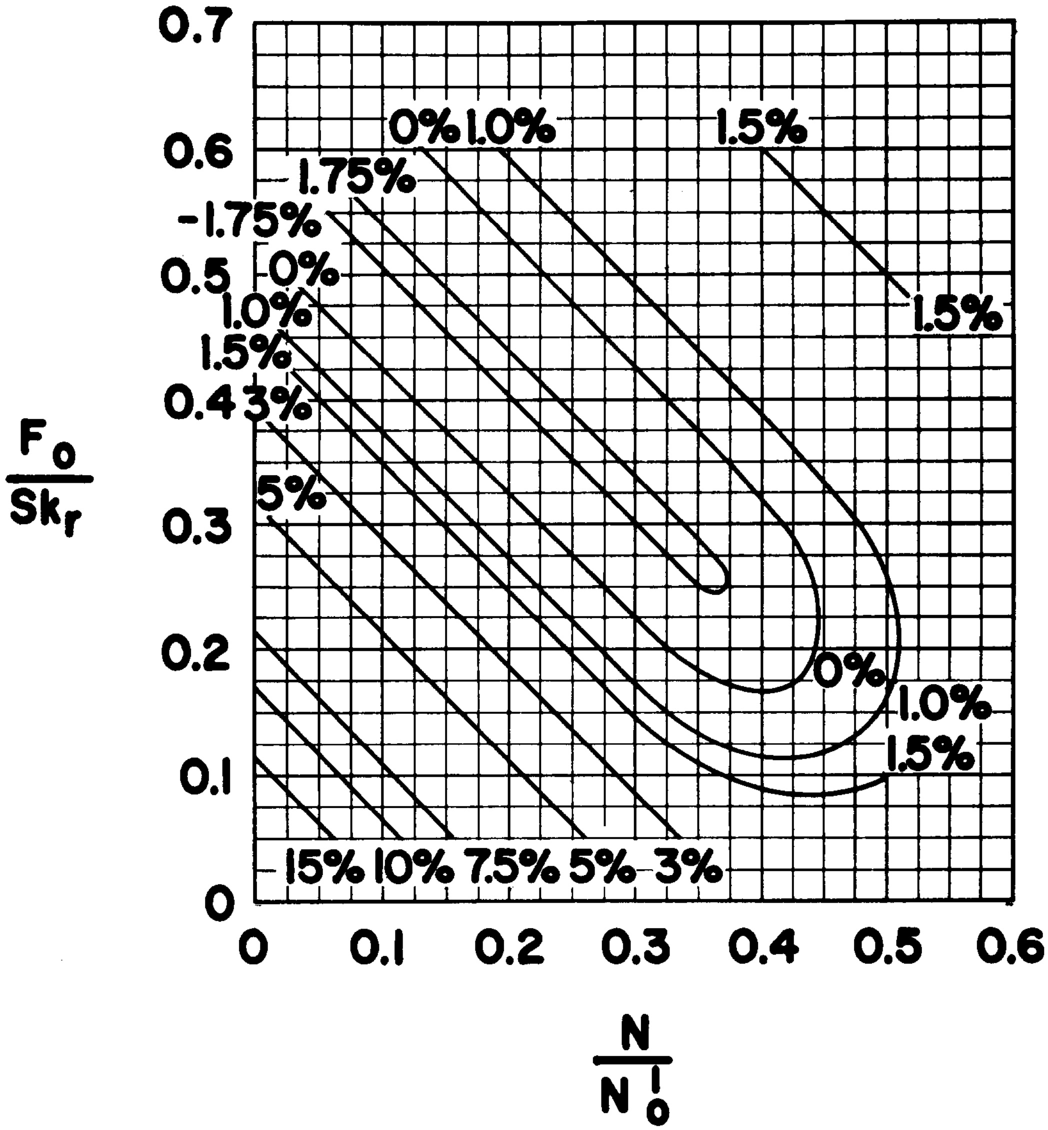
FIG. 12



(API FIG. 6)

$\frac{F_3}{Sk_r}$, POLISHED ROD HORSEPOWER

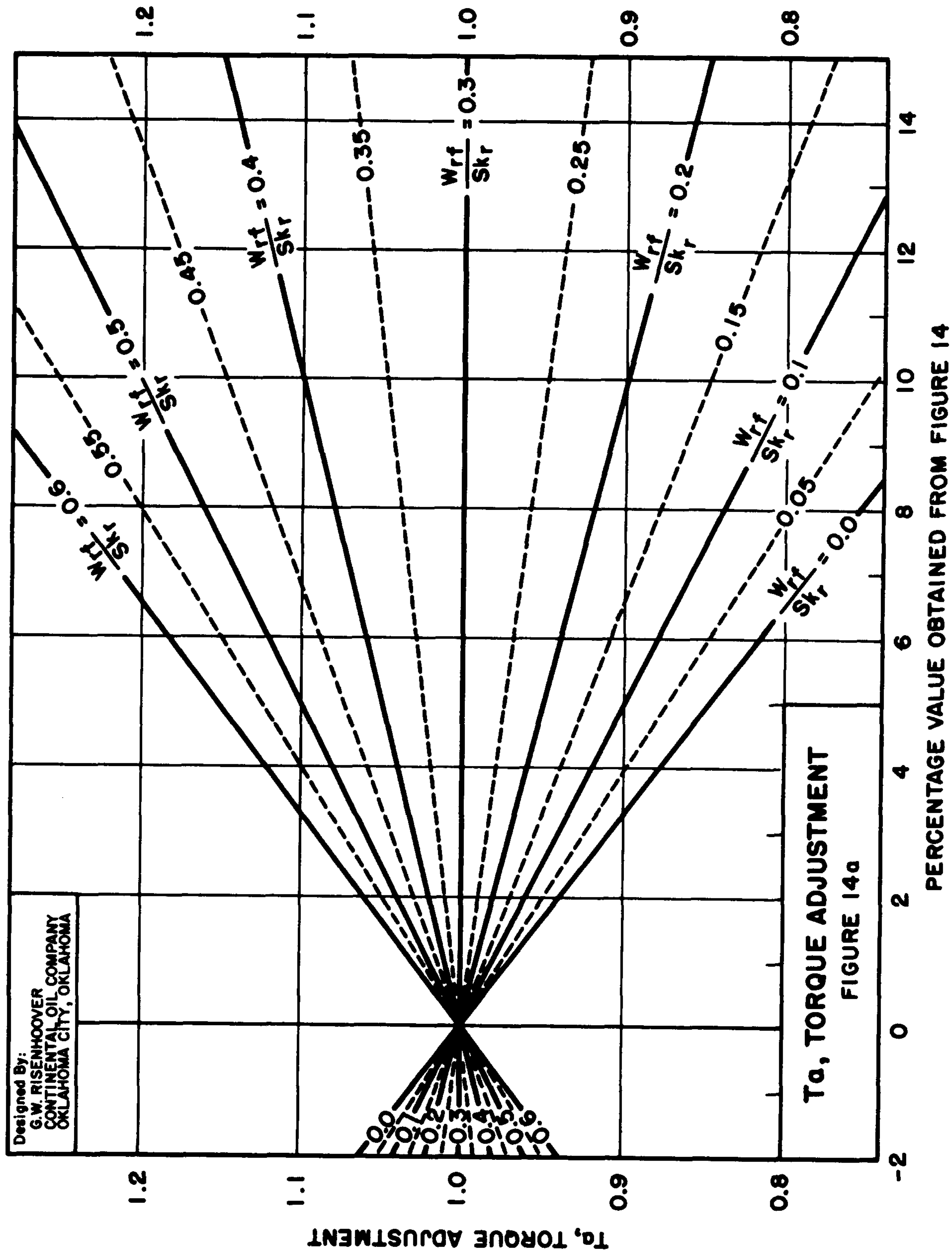
FIG. 13



(API FIG. 7)

Ta, ADJUSTMENT FOR PEAK TORQUE FOR VALUES OF $\frac{Wrf}{Sk_r}$ OTHER THAN 0.3

FIG. 14



Designed By:
G.W. RISENHOOVER
CONTINENTAL OIL COMPANY
OKLAHOMA CITY, OKLAHOMA
Ta, TORQUE ADJUSTMENT
FIGURE 14a
PERCENTAGE VALUE OBTAINED FROM FIGURE 14
Ta, TORQUE ADJUSTMENT

FIG. 14a

IV.

DETERMINING THE CALCULATED PEAK CRANK TORQUE

API RP 11L also provides a convenient method for determining the calculated peak crank torque (PT). The following procedure is used in this determination:

$\frac{2T}{S^2K_r}$ is a function of $\frac{N}{N_o}$ and $\frac{F_o}{SK_r}$

$\frac{F_3}{SK_r}$ is a function of $\frac{N}{N_o}$ and $\frac{F_o}{SK_r}$

If $\frac{W_{rf}}{SK_r}$ is greater or less than 0.3, an appropriate adjustment of torque must be made.

T_a = per cent adjustment, and is also a function of $\frac{N}{N_o}$, and $\frac{F_o}{SK_r}$

$T_a = 1.00 \pm \left[(\%\text{ indicated on Fig. 14} \div 100) \times \left(\frac{W_{rf}}{SK_r} - 0.3\right) \times 10\right]$ or obtained graphically from Figure 14 and Figure 14a.

$$PT = (2T/S^2K_r) \times SK_r \times S/2 \times T_a$$

V.

DETERMINING THE POLISHED ROD HORSEPOWER

It is possible to determine the polished rod horsepower (PRHP) by using the method recommended by API RP 11L, which is as follows:

$\frac{F_3}{SK_r}$ is a function of $\frac{N}{N_o}$ and $\frac{F_o}{SK_r}$

See Fig. 13, (API RP 11L, Fig. 6)

where F_3 = a force which will give horsepower when applied to the full stroke length at the speed of the pumping unit, and multiplied by the constant 2.53×10^{-6}.

$$PRHP = \frac{F_3}{SK_r} \times SK_r \times S \times N \times 2.53 \times 10^{-6}$$

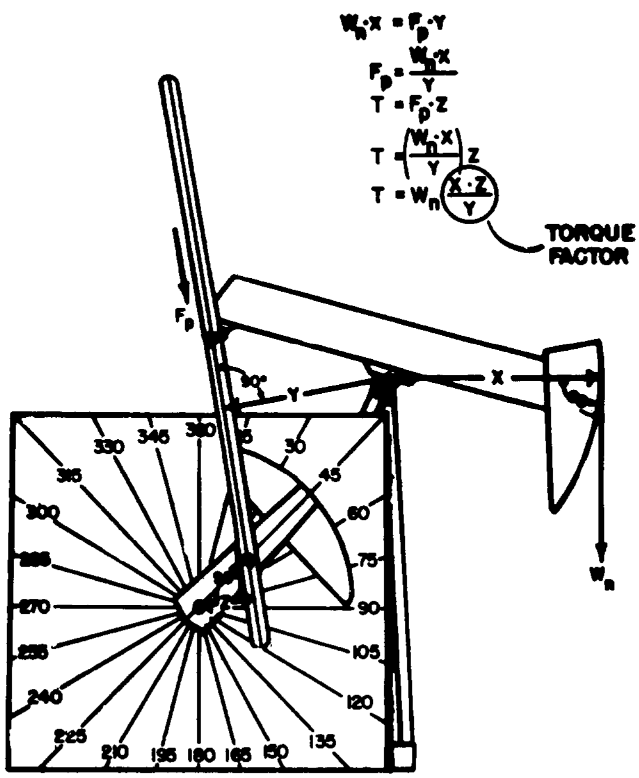


FIG. 15

(After WORLD OIL, March, 1965)

VI.

DETERMINING TORQUE FACTORS, TF, ON A MODEL CONVENTIONAL BEAM PUMPING UNIT

Torque factors are becoming a way of life in determining net torque values. It is essential that the correct torque factor be used with the corresponding load values, or an incorrect torque calculation may, and in all probability will, result. A torque factor is in essence a distance, measured in inches, which depends on the geometry of the pumping unit at certain crank angles. The use of Fig. 15, which is a 3-ft. high scaled model of a pumping unit, greatly simplifies the problem of explaining the various distance components of a torque factor and the associated crank angle.

Step 1. Assume that a load, W_n, is hanging from the front of the horsehead, perpendicular to the ground, and is at a horizontal distance, X in., from the center of the Sampson post bearing.

Step 2. Assume that the load, W_n, is being supported by a force, F_p, acting along the center line of the Pitman and that the length of the perpendicular from the center of the Sampson post bearing to the center line of the Pitman is equal to Y in.

Step 3. Assume that the force, F_p, is balanced by torque applied to the slow speed (crank) shaft and that the perpendicular distance from the center of the slow speed shaft to the center line of the Pitman is equal to Z in.

Therefore:

Step 4. $W_n \times X = F_p \times Y$

Step 5. $\frac{W_n \times X}{Y} = \frac{F_p \times Y}{Y}$ and $F_p = \frac{W_n \times X}{Y}$

Step 6. $T = F_p \times Z$

Step 7. Substitute value of F_p:

$$T = \frac{W_n \times X}{Y} \times Z; \text{ or } T = W_n \frac{X \times Z}{Y}$$

Step 8. $\frac{X \text{ in.} \times Z \text{ in.}}{Y \text{ in.}} = \frac{X \times Z}{Y}$ in.

$\frac{X \times Z}{Y}$ in. is defined as a "torque factor."

Step 9. The net well load, W_n, in pounds, at any crank angle, multiplied by the torque factor, in inches, corresponding to that crank angle will give the torque, in inch-pounds, which must be applied to the slow speed shaft to balance the net well load, W_n. In actual practice, the net well load, W_n, as used in Step 9, is equal to the well load at a specific crank angle minus the structural unbalance and minus any beam weight counterbalance effect measured at the polished rod.

VII.

DETERMINING POLISHED ROD STROKE LENGTH AND POSITION OF INSTANANEOUS LOADS FROM DYNAMOMETER CARD

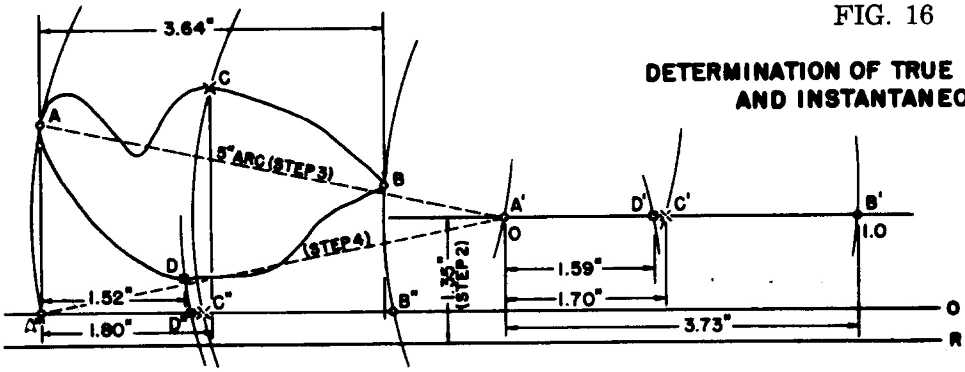
It is of definite advantage, and in most cases a necessity, to fix accurately the exact position of the polished rod with respect to instantaneous well loads. The impact and truth of this statement will be reinforced in a corresponding discussion under net torque determination presented later in the paper.

The following procedure presents the method to be used to determine the polished rod stroke length and to locate exact positions of the polished rod during the pumping cycle when using a Johnson-Fagg Dynamometer. Appropriate changes should be made when using other brands of dynamometers.

Step 1. The dynamometer card should be temporarily mounted on the left side of a sheet of paper that is at least as wide as the actual trace of the dynamometer card plus five inches.

FIG. 16

DETERMINATION OF TRUE POLISHED ROD STROKE LENGTH AND INSTANTANEOUS LOAD POSITIONS



Step 2. Construct a line 1.35 in. above and parallel to the reference line, R, and to the right of the dynamometer card. This line represents an imaginary line on the dynamometer card that corresponds to the distance from the reference line to the center of the main weight recording stylus shaft when the drum holding the dynamometer card rotates.

Step 3. The length of the main weight recording stylus is 5 in. Therefore, swing 5-in. arcs from the left and right ends of the dynamometer card to intersect the line constructed in Step 2. The distance between the intersections on this line is the correct dynamometer card length, "S."

Step 4. Swing 5-in. arcs from the two points found on the 1.35-in. line in Step 3 to the zero (0) line. The distance between these intersections is also the correct length of the dynamometer card.

Step 5. Place the dynamometer card back on the drum and measure the length of the string, in inches, that must be pulled from the drum to make the main stylus point move from the left zero line intersection point to the right zero line intersection point. This length is the exact polished rod stroke length, "S," in inches.

It is evident that an approximate "Dynamometer Length Constant" can now be determined by dividing the exact polished rod stroke length, in inches, by the exact dynamometer card length, in inches. This constant should be determined and recorded for each sheave since inexperienced personnel often forget to measure and report stroke lengths.

JUSTIFICATION

Drop perpendiculars from the ends of the card to the zero line. Compare this distance to the distance between the intersections found in Step 4. The magnitude of the error experienced is influenced by the location of the card end points relative to the line 1.35 in. above the reference line. It can now be seen that the location of any instantaneous load during the cycle relative to the position of the polished rod at that instant, must be determined by swinging a 5-in. arc from the instantaneous load to the line determined in Step 3. The left end of this line, the first point found in Step 3, is defined as the zero (0) position of the rods. The right end of the line, the second point found in Step 3, is defined as the 1.0 position of the rods. The exact position of the rods at any intermediate point, such as at the peak polished rod load, is found by measuring the distance between the zero point and the intermediate point and then dividing this distance by the "S" distance found in Step 3. Note the discrepancy in Fig. 16 between the apparent polished rod position and the true position. A serious instantaneous net torque determination error is possible unless this procedure is followed. It can be safely stated that there will be definite instantaneous difference between the apparent net torque and the actual net torque when attempting to analyze 15° crank angle loads throughout the pumping cycle unless this general method is followed. The following calculation illustrates this point:

Dynamometer length constant = 17.4 $\frac{in.}{in.}$

Actual length of card = 3.73 in.

Actual stroke length = 17.4 $\frac{in.}{in.}$ x 3.73 in. = 64.9 in.

Horizontal length of card = 3.64 in.

Stroke length, using horizontal measurements = 17.4 x 3.64 = 63.3 in.

$$\text{Per cent error} = \left(\frac{63.3 - 64.9}{64.9}\right) \times 100 =$$

$$-\frac{160}{64.9} = -2.47 \text{ per cent.}$$

Location of peak load = $\frac{1.70}{3.73}$ = 0.456 of polished rod travel.

Location of peak load, using horizontal measurements = $\frac{1.80}{3.64}$ = 0.495

$$\text{Per cent error} = \left(\frac{0.495 - 0.456}{0.456}\right) \times 100 =$$

$$\frac{3.9}{0.456} = 8.55\%$$

Location of minimum load = $\frac{1.59}{3.73}$ = 0.426 of polished rod travel.

Location of minumum load, using horizontal measurements = 1.52/3.64 = 0.418

$$\text{Per cent error} = \left(\frac{0.426 - 0.418}{0.426}\right) \times 100$$

$$\frac{0.8}{0.426} = 1.88\%$$

VIII.

DETERMINING THE THEORETICAL NET TORQUE AT THE CRANK, MEASURED AT THE POLISHED ROD

The theoretical net torque determination for conventional sucker rod pumping systems is one of the most important calculations which should be made. A pumping unit is normally designed to give approximately 20 years of service unless it is abused or overloaded. Only by making a net torque determination can it be determined that the gear box is overloaded. Careful attention to this feature often results in extending the life of a gear box, especially in waterflood operations. It has been stated by some manufacturers that the gear box represents approximately 40 per cent of the cost of a conventional pumping unit.

The procedure to follow in making net torque determinations from a Johnson-Fagg dynamometer card is presented below. Appropriate changes should be made when using other brands of dynamometers.

Step 1. Secure "API Pumping Unit Stroke and Torque Factors" and the "Structural Unbalance" from the manufacturer.

Step 2. The weight, in pounds, or counterbalance effect (CBE) at the polished rod at the 90° crank angle position measured during the dynamometer survey is determined, which is the distance from the zero line, in inches, and multiplied by the dynamometer constant, in pounds per inch. This measured counterbalance effect includes the structural unbalance.

Step 3. The torque (moment) exerted on the crank at the slow speed shaft by the crank counterbalance at the 90° crank angle position is then determined by subtracting the structural unbalance from the counterbalance effect measured at the polished rod, and multiplying the resultant, in pounds, by the torque factor, in inches, at 90°. Note: If the structural unbalance is negative, the crank counterbalance is greater than the counterbalance at the polished rod. Formula: Q, the maximum crank counterbalance moment = (CB at PR at 90° — SU) x TF @ 90°. The crank counterbalance moment at any other crank position is determined by multiplying "Q" by the sine of the crank angle position, Θ. On conventional units, the crank counterbalance moment is always at a maximum when the crank weights are horizontal and always zero when the cranks are vertical, either up or down. The moment, Q multiplied by sine θ, is positive if the prime mover is lifting the crank weights and negative if the crank weights are helping the prime mover lift the well load.

Step 4. Mount the dynamometer card to be studied on the lower left hand corner of a sheet of paper with the reference line near the bottom. The paper must extend a minimum of 5 in. beyond the right end of the card.

Step 5. As was done in Fig. 16, extend the reference line to the right and construct a line 1.35 in. above and parallel to the reference line. This line represents an imaginary line on the dynomometer card corresponding to the distance from the reference line to the center of the main weight recording stylus shaft when the drum holding the dynamometer card rotates. If the selected card does not have a 0 (zero) line, construct a 0 line the correct distance above the reference line.

Step 6. Swing 5-in. arcs from the extreme left and right ends of the card and intersect the 1.35 in. line constructed in Step 5. Label the left point 0 (zero). Label the right point 1.0 (one).

Step 7. Divide the line constructed in Step 6 into ten equal parts. Label the division mark to the right of the zero point 0.1, the next 0.2, etc.

Step 8. Determine which "rod positions" and torque factors correspond to the upstroke portion of the card and which ones correspond to the downstroke portion. This isn't always as easy as it might sound.

If data furnished by the manufacturer do not indicate the crank position at the start and end of the stroke, and the torque factors are not marked plus or minus, plot on linear graph paper a curve of the "crank angle" on the axis of abscissas (X-Axis) versus "torque factor" on the axis of ordinates (Y-Axis) and determine these points. The start and end of the stroke occur at the crank angles where the torque factor is zero, not at crank angles of 0° and 180°.

If the crank rotates clockwise when viewed with the polished rod to the right and the gear box to the left, the upstroke will be from approximately 0° to approximately 180°. <u>The upstroke will, in general, not start at 0°, nor will it end at 180°.</u> If the crank rotates counterclockwise, the upstroke will start between 15° and 345° (either side of 0°) and end between 195° and 165° (either side of 180°). By definition and logic, the torque factors are positive on the upstroke of the polished rod and negative on the downstroke of the polished rod.

Step 9. Swing arcs from each upstroke rod position of the polished rod to the upstroke portion of the dynamometer card Label the points on the card with the corresponding crank angle positions. For example, assuming a clockwise rotation, the first point determined after the start of the upstroke will be correctly labeled "15°", if the upstroke started between 0° and 15°. If the rotation was counterclockwise, the first point would be labeled "360" (or "0"), the second "345", etc.

Repeat the above procedure for the downstroke portion of the cycle.

Step 10. A work-saving short cut is to divide the "SU" (structural unbalance), in pounds, by the dynamometer weight constant, in pounds per inch, and construct a "SU" line at this calculated distance above the "0" line, if the "SU" is positive; or below the "0" line, if the "SU" is negative.

Step 11. Determine the net well load, in pounds, at each crank angle position to be studied. Net well load is equal to the load measured at the polished rod minus the structural unbalance (and minus any beam weight counterbalance effect, if present). Formula: NWL = PRL — SU; or NWL = (Distance from the polished rod load to "0" line minus the distance from "SU" line to "0" line) multiplied by the dynamometer weight constant; or NWL = distance from the polished rod load to "SU" line multiplied by the dynamometer weight constant.

Step 12. The theoretical net torque at the crank (slow speed shaft), measured at the polished rod, at a specific crank position is the algebraic sum of the net well load multiplied by the torque factor at the crank position, and the maximum crank counterbalance moment, "Q", multiplied by the sine of the crank angle, θ.

SUMMARY: Net well load torque is positive during the polished rod upstroke. The crank counterbalance moment is negative when the weights are falling. Both can be negative, or positive, at the start or end of the stroke, depending on unit geometry.

Crank counterbalance moment is positive when the weights are being lifted. Net well load torque is negative during the polished rod downstroke.

Step 13. Plot the theoretical net torque on the ordinates (Y-Axis) versus the crank angle position on the abscissas (X-Axis). Draw a curve through the points. The approximate peak torque during the upstroke and downstroke can be read from the curve. The crank angles, θ, at which the peak torques occurred can be read on the X-Axis.

If the plot suggested in Step 7 was made, the torque factors at the peak torque points can be read from this plot. Rod positions at the peak torque points can be approximated by extrapolating. If it is desired to determine the rod positions more exactly, plot rod position (Y-Axis) versus crank position (X-Axis) and read the rod positions desired. The theoretical net torque can then be calculated for the two peak points.

Step 14. If the peak torques are not almost equal, the unit probably should be rebalanced. Caution: If the correct counterbalance would cause the torque to reverse or increase negative torque during the high velocity portion of cycle, changing the counterbalance could cause more trouble than it would eliminate.

Ignoring negative torque, the counterbalance should be adjusted so that the theoretical peak torque during the upstroke will equal the theoretical peak torque during the downstroke. Let θ_1 = crank angle at peak torque on upstroke, and θ_2 = crank angle at peak torque on downstroke. Then if peak torques are equal, (PRL at θ_1 — SU) x TF at θ_1 — (CBE — SU) x TF at 90° x Sin θ_1 = (CBE — SU) x TF at 90° x Sin θ_2 — (PRL at θ_2 — SU) x TF at θ_2; (CBE — SU) x TF at 90° x (Sin θ_1 + Sin θ_2) = (PRL at θ_1 — SU) x TF at θ_1 + (PRL at θ_2 — SU) x TF at θ_2.

The optimum counterbalance, measured at the polished rod at 90°, CBE =

$$\frac{(PRL\ @\ \theta_1 - SU)TF\ @\ \theta_1 + (PRL\ @\ \theta_2 - SU)xTF\ @\ \theta_2}{TF\ @\ 90^\circ\ x\ (SIN\theta_1 + SIN\theta_2)}$$

$+ SU$

The following example reflects the net torque, the associated determinations and the report form used.

FIG. 17

98

TORQUE CALCULATION SHEET FOR CONVENTIONAL & AIR-BALANCED UNITS

Well: Johnson #1
Unit Make & Size Continental-Emsco DH-228-246-86
Structural Unbalance, S.U. = 360 lbs.
Possible Stroke Lengths 62-74-86 in.
Unit Rotation Clockwise Rotation
Assumed Unit Efficiency 83%
Counterbalance at Polished Rod @ 90° 9100 lbs.
Measured Stroke Length, S.L. = 4.73 in.

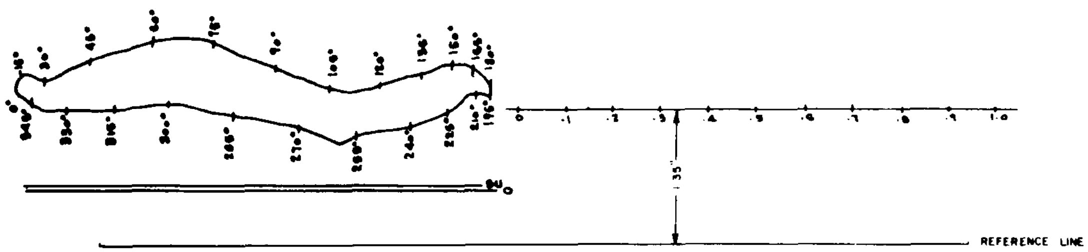
Calc. S.L. (from card) = 4.73 in. X 18.2 in./in. = 86 in.
Torque Factor Correction = Measured S.L./Mfg. S.L. = 1.000
Dynamometer Constant, D.C. = 8150 lbs/in.
Max CB Torque = (CB-SU) X TF @ 90° (9100-360)x41.208=360,160 in.-lbs.
Peak Load = 1.55 in. X 8150 lbs/in. = 12,620 lbs @ 70.57°
Minimum Load = .44 in. X 8150 lbs/in. = 3585 lbs @ 258.32°
P.R.H.P. = $\frac{\text{Area Card X D.C. X S.L. X SPM}}{\text{Card Length X 12 X 33,000}} = \frac{2.335\text{x}8150\text{x}86\text{x}12.5}{4.73\text{x}12\text{x}33{,}000} = 10.93$

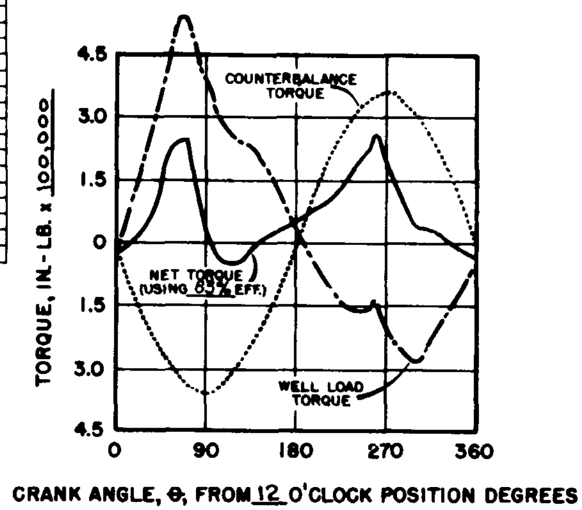
Conventional - Use all columns
Air balance - Use columns 1, 2, 3, 5, 10, 11

1	2	3	4	5	6	7	8	9	10	11
CRANK ANGLE	A FROM DYNAMOMETER CARD IN (a)	WELL LOAD (2 X D C) LBS	NET WELL LOAD (3- S U) LBS	TORQUE FACTOR IN (b)	POSITION OF RODS	WELL LOAD TORQUE (4 X 5) IN-LBS (c)	CB LEVER CORRECTION FACTOR	C B TORQUE (8 X MAX CB TORQUE) IN-LB (d)	THEORETICAL NET TORQUE (7+9) (e)	NET TORQUE IN-LBS (f)
0	.95		7,745	- 4.495	.001	- 34,815	0	-0-	- 34,815	- 28,900
15	1.04		8,475	10.739	.009	91,015	.259	-93,280	- 2,265	- 1,880
30	1.05		8,560	25.224	.064	215,915	.500	-180,080	35,835	43,200
45	1.25		10,190	36.630	.160	373,260	.707	-254,635	118,625	145,000
60	1.46		11,900	42.873	.282	510,190	.866	-311,900	198,290	239,000
75	1.43		11,655	43.872	.414	511,330	.966	-347,915	163,415	197,000
90	1.18		9,615	41.208	.548	396,215	1.000	-360,160	36,055	43,450
105	.99		8,070	36.380	.664	293,590	.966	-347,915	- 54,325	- 45,909
120	1.02		8,315	30.802	.768	256,120	.866	-311,900	- 55,780	- 46,300
135	1.14		9,290	24.975	.851	232,020	.707	-254,635	- 22,615	- 18,770
150	1.25		10,190	18.981	.917	193,415	.500	-180,080	13,335	16,080
165	1.17		9,535	12.487	.966	119,065	.259	- 93,280	25,785	31,100
180	1.07		8,720	4.995	.994	43,555	0	-0-	43,555	52,450
195	.92		7,500	- 3.663	.996	- 27,475	.259	93,280	65,805	79,350
210	.93		7,580	-13.403	.971	-101,595	.500	180,080	78,485	94,500
225	.76		6,195	-23.559	.913	-145,950	.707	254,635	108,685	131,000
240	.61		4,970	-32.717	.828	-162,605	.866	311,900	149,295	180,000
255	.49		3,995	-39.377	.717	-157,310	.966	347,915	190,605	229,750
270	.58		4,725	-43.123	.595	-203,755	1.000	360,160	156,405	188,500
285	.72		5,868	-43.872	.455	-257,440	.966	347,915	90,475	109,000
300	.33		6,765	-41.708	.326	-282,155	.866	311,900	29,745	35,850
315	.75		6,115	-36.786	.205	-225,005	.707	254,635	29,630	35,650
330	.76		6,195	-28.804	.105	-178,440	.500	180,080	1,640	1,980
345	.82		6,685	-18.065	.034	-120,765	.259	93,280	- 27,485	- 22,800
70.57°	1.50		12,225	44.45	.375	543,400	.943	-339,630	203,770	245,400
258.32°	.40		3,260	-40.10	.690	-130,730	.979	352,600	221,870	267,200

(a) A=Distance from zero line to dynamometer card on conventional units. A=Distance from air counterbalance line to dynamometer card on air balance units. A is positive if you measure up to card, negative if you measure down from C.B. line to card.
(b) Manufacturer's TF multiplied by TF correction.
(c) Positive on upstroke, negative on downstroke.
(d) Positive when CB rising, negative when CB falling.
(e) For air balance units, column 10 is 3 X 5.
(f) When 10 is positive, divide by efficiency; when 10 is negative, multiply by efficiency.

REMARKS: Peak Torque slightly exceeds the gear box rating. Negative Torque occurs in an undesirable part of upstroke in the high speed portion from between 90°-105° to between 135°-150°. The counterbalance is near optimum and to optimize counterbalance does not help eliminate the negative torque in middle of stroke in this case.





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IX.

SUBSURFACE PUMP SELECTION

The selection of the proper subsurface pump is a very important part of the pumping system design. There is a very close interrelation between the pump size and the design of the sucker rod string. Based on available information, over half the pumps in operation are larger than they should be for the most economical and profitable operation. In some cases there are reasons why it is necessary to use larger pumps than needed. However, a large majority have pumps installed which are too large when it is not necessary, and these will experience higher operating costs than would be the case if the proper pump size were used. Since this subject can be enlarged to constitute a textbook, the coverage in this paper will only be superficial.

API Standard 11AX pertains to the nomenclature and hardware of basic subsurface pumps. The pump companies have publications which relate to the types of pumps for different environmental well conditions. The selection of pump type must be tailored to specific conditions, and each design problem should be considered separately. No attempt will be made to solve pump type selection in this paper.

The sizing of the subsurface pumps to well capacity is another matter. There are "quick design" charts and other methods available to aid in selecting the correct or appropriate pump size.

Table 3, which is reproduced from a paper presented by Douglas O. Johnson at the Fourth West Texas Oil Lifting Short Course, is one such method. This table has been found to be accurate and is highly recommended.

Pump Plunger Sizes Recommended for Optimum Conditions

Net Lift of Fluid ft.	Fluid Production - Barrels per day - 80 pct efficiency									
	100	200	300	400	500	600	700	800	900	1000
2000	1 1/2 1 1/4	1 3/4 1 1/2	2 1 3/4	2 1/4 2	2 1/2 2 1/4	2 3/4 2 1/2	2 3/4	2 3/4	2 3/4	2 3/4
3000	1 1/2 1 1/4	1 3/4 1 1/2	2 1 3/4	2 1/4 2	2 1/2 2 1/4	2 1/2 2 1/4	2 3/4 2 1/2	2 3/4	2 3/4	2 3/4
4000	1 1/4	1 3/4 1 1/2	2 1 3/4	2 1/4 2	2 1/4 2	2 1/4	2 1/4	2 1/4		
5000	1 1/4	1 3/4 1 1/2	2 1 3/4	2 1 3/4	2 1/4 2	2 1/4				
6000	1 1/4	1 1/2 1 1/4	1 3/4 1 1/2	1 3/4						
7000	1 1/4 1 1/8	1 1/2 1 1/4			In this tabulation surface pumping strokes up to 74 in. only are considered.					
8000	1 1/4 1 1/8									

Table 3

(Courtesy Bethlehem Steel Company Sucker Rod Handbook, 1958, and WORLD OIL, December 1957)

WELL CAPACITY

The critical part of pump size selection involves the determination of well production. In the past, the procedure used involved an assumption that producing rates would be proportional to pressure drawdown, and by calculating a Productivity Index and determining the amount of available drawdown, the capacity of the well could be calculated. At the time this method was being used it was realized that it was not completely accurate when two or more well fluids (oil, gas or water) were being produced simultaneously. This was due to inherent changes in effective permeability with changes in fluid production and producing pressure which, in turn, limited straight-line projections. In 1966, J. V. Vogel of Shell Oil Company presented a method of determining producing rates at different producing intake pressures. He termed his method of analysis the "Inflow Performance Relationship" (IPR) after the terminology used in a paper written by W. E. Gilbert in 1954.

Prior to the advent of computers, the calculation of IPR curves was too tedious to be practical, but with modern day computer assistance, Vogel calculated IPR curves for wells producing from several fictitious solution gas-drive reservoirs. From those curves, he was able to develop a reference IPR curve which not only could be used for most solution gas-drive reservoirs in arriving at oil well productivity but would give more accurate projections. The limitation Vogel placed on his method was that it be used for two-phase flow conditions in a solution gas-drive reservoir. In general he felt it would not be correct to use it for other types of drive. One exception to this would be in a partial water-drive field where a large portion could be isolated from the encroaching water by barrier rows of producing wells near the encroachment front. He felt in that case the method could be used for at least a portion of the producing life of the shielded area. Another comparable exception was that the method could be used for a portion of a reservoir in which the expansion of a gas cap would be a significant factor.

In our opinion, well production projections are more reliable by using the Vogel method than by using the straight-line projection method.

In 1968, James R. Eickmeier of Shell Canada Limited published a paper in which he presented a refined method of using the "Inflow Performance Relationship" curves. He used production data from wells located in the House Mountain Field which is located about 130 miles from Edmonton. His experience was that the forecast IPR curves were substantiated by actual production figures.

Using the techniques presented by Vogel in his paper and the data and generalized IPR curves used by Eickmeier in his adaptation, a non-dimensional plot of rate versus pressure has been constructed for a hypothetical field having characteristics and production history similar to the House Mountain Field. It is felt that this plot, Fig. 18, can be used with confidence for fields similar to the House Mountain Field. If the reservoir under study has different reservoir parameters, then appropriate caution should be exercised in the use of the plot.

As an aid in using Fig. 18, Generalized IPR Curves, a hypothetical 4400-ft waterflood well capacity problem is presented. The steps involved in the solution are superposed on Fig. 18.

EXAMPLE PROBLEM

Determine the maximum well capacity in a 4400-ft. waterflood producer.

Given:

(1) Maximum pump setting depth = 4400 ft.

(2) Present static (shut-in) reservoir pressure = 810 psig measured at 4400 ft.

(3) Present well capacity with 49 psig back pressure at pump intake (at 4400 ft.) = 95 BFPD.

(4) Estimated shut-in reservoir pressure (ultimate reservoir pressure) at this well after waterflood fillup = 2450 psig.

(5) The field was originally a solution gas-drive reservoir.

Assume:

The Vogel method is valid and Eickmeier's Generalized IPR Curves can be used to determine well capacity when reservoir pressure increases or decreases, if the present static reservoir pressure, present stabilized producing rate, and present stabilized pump intake pressure are known.

Problem:

Determine the capacity of this well when the shut-in reservoir pressure reaches 2450 psig, using the Generalized IPR Curves.

Definition of Terms:

q_p = producing rate other than maximum possible

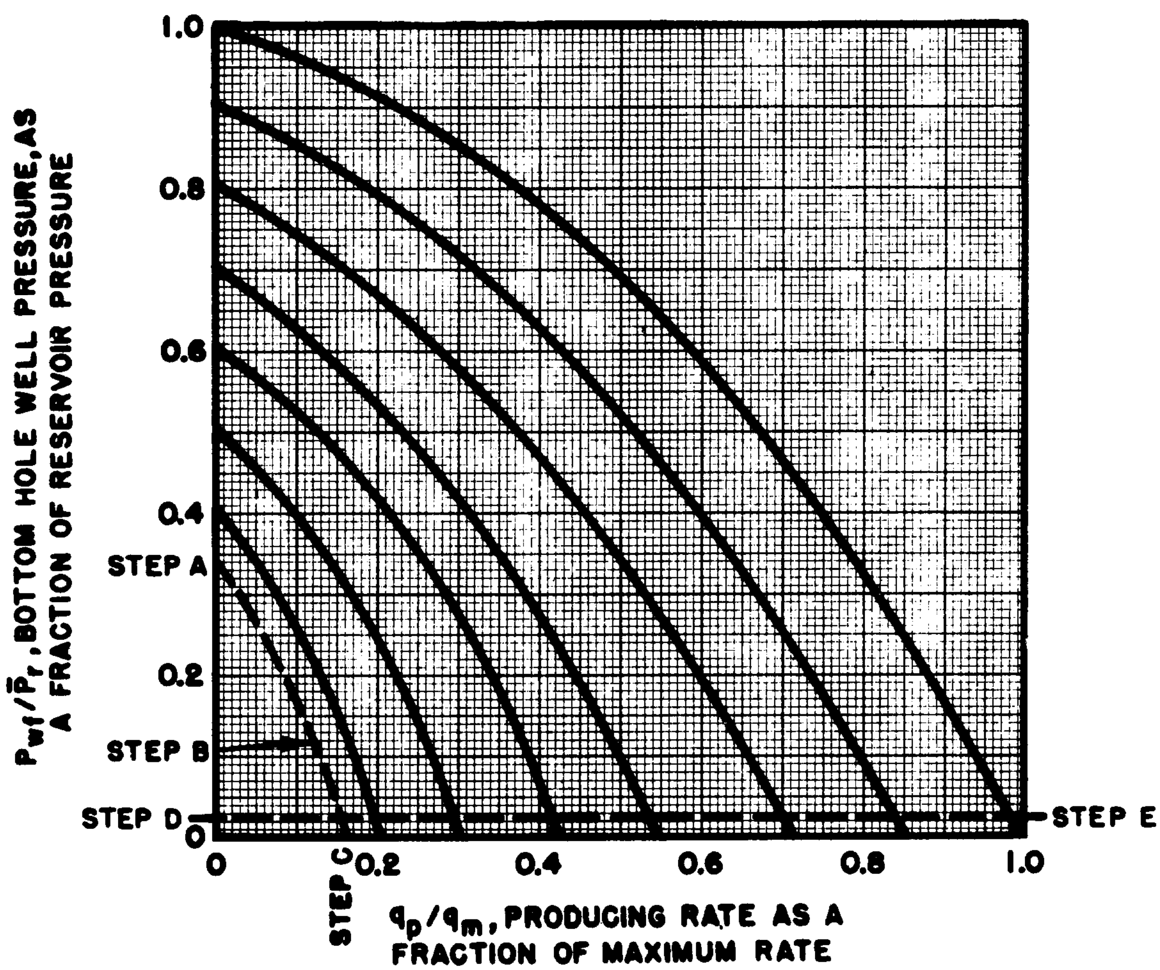
q_m = maximum possible producing rate

P_{wf} = bottom-hole pressure q_p

P_r = maximum reservoir pressure

P_{rp} = reservoir pressures less than maximum (usually present shut-in bottom-hole pressure)

GENERALIZED IPR CURVES



NOTE: DASHED LINES USED TO DETERMINE THE MAXIMUM CAPACITY OF A 4400' WATER-FLOOD PRODUCER IN EXAMPLE PROBLEM

FIG. 18

(After WORLD OIL, May, 1968)

Solution:

(a) Ratio of present shut-in bottom-hole pressure to maximum reservoir pressure, $\overline{P}_{rp} / \overline{P}_r = (810 + 14.7)/(2450 + 14.7) = 0.335$.

(b) Draw a curve through 0.335 on the Y-Axis and parallel to the specific Generalized IPR Curve that passes through approximately 0.4 on Y-Axis and 0.2 on X-Axis.

(c) Read on the X-Axis that this curve passes through 0.16. This is the ratio of the present capacity with zero producing bottom-hole pressure to the maximum possible producing rate.

(d) Ratio of present producing bottom-hole pressure to maximum reservoir pressure, $P_{wf} / \overline{P}_r = (49 + 14.7)/(2464.7 = 0.0258$. Draw a line parallel and 0.0258 units above the X-Axis. From where this line crosses the curve drawn in Step "b" read down to 0.158 on the X-Axis. This (0.158) is the ratio of the present producing rate to the maximum possible producing rate.

(e) From where the line drawn in Step "d" crosses the curve that passes through 1.0 on the X and Y Axes, read down to 0.98 on the X-Axis. This is the ratio of the producing rate at a reservoir pressure of 2450 psig with 49 psig producing bottom-hole pressure to the producing rate at a reservoir pressure of 2450 psig with zero producing bottom-hole pressure.

(f) Calculations:

(1) Present capacity with reservoir pressure of 824.7 psia and 63.7 psia (49 psig) back pressure = 95 BFPD. (Given)

(2) Present capacity with C psia back pressure = (0.16/0.158) x 95 = 1.01 x 95 = 96 BFPD.

(3) Capacity when reservoir pressure is increased to 2464.7 psia with 63.7 psia back pressure =(0.98/ 0.158) x 95 = 6.2 x 95 = $\underline{\underline{589}}$ BFPD.

(4) Capacity when reservoir pressure is 2464.7 psia with 0 psia back pressure = 95/0.158 = 601 BFPD.

Although in this instance the adaptation may not be entirely accurate due to the introduction of water as a recovery mechanism, and the fact that the produced fluid will be a mixture of oil and water, it is felt it is much more reliable for determining well capacity than the previously used PI straight-line extrapolation method. With that word of caution, the procedure is recommended.

X.

SELECTION OF TYPE OF SUCKER RODS

There are several grades of sucker rods which can be used in a pumping system. Some of these are better than others because of the nature of the fluid to be lifted and the investment and operating costs involved.

ALLOWABLE STRESS

The first consideration which must be made pertains to the allowable stress limits. Although there are situations which call for the use of other rods, the majority involve the use of API Class C rods. API Class D rods are considered where the capabilities of API Class C rods are exceeded and if the system contains no hydrogen sulfide and is either noncorrosive or effectively inhibited. The discussion will be limited to situations involving the selection of those two classes and will present a method of derating sucker rod strings which contain slim-hole couplings.

API Class C sucker rods must have a minimum tensile strength of 90,000 psi, and Class D rods must have a minimum tensile strength of 115,000 psi. The tensile strength of 115,000 psi is secured by hardening the rods to approximately 265 Brinell typical. This is above a Rockwell C hardness of 22; therefore, API Class D rods are susceptible to sulfide cracking and must not be used in hydrogen sulfide systems. API Class C rods give satisfactory service in an hydrogen sulfide environment if metal loss is controlled with an effeçtive inhibitor.

The API-suggested method of derating sucker rods for use in media other than air utilizes

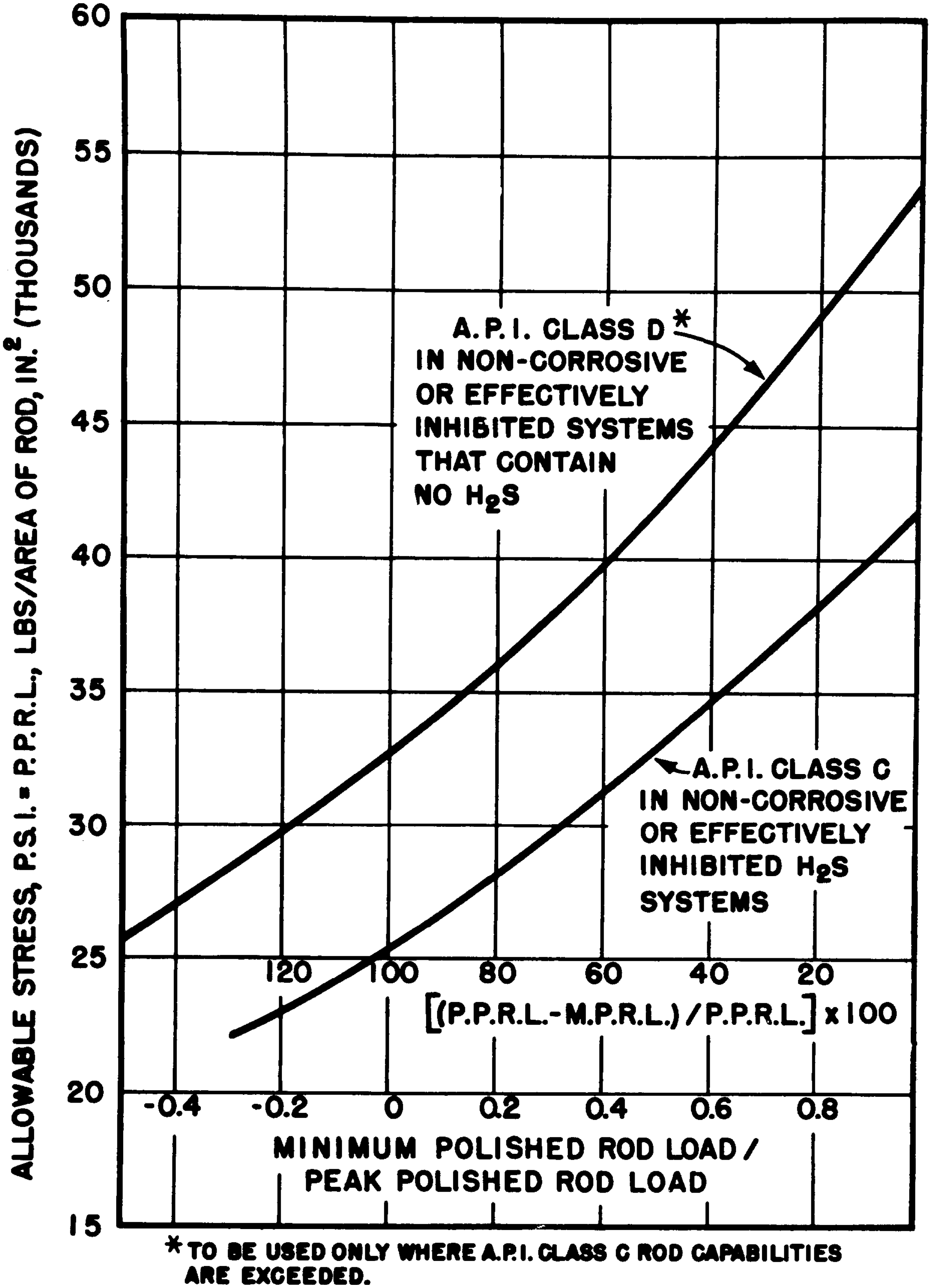


FIG. 19

SELECTION OF TYPE OF SUCKER RODS

a modified Goodman diagram that presents minimum stress versus maximum stress. It is contended that a plot of "stress" versus "stress ratio" is more meaningful. The API data indicate that the allowable stress in air with complete reversal is one-third of the minimum tensile strength. In other words, a fatigue life in air of 10 million cycles can be expected when a Class C rod is loaded with a maximum (tensile) load of 30,000 psi (90,000 x 1/3) and a minimum (compressive) load of minus 30,000 psi. Our plot of "stress" versus "stress ratio" indicates that the air endurance limit stress is two-thirds of minimum tensile strength when the minimum load is zero. The API plot indicates that the air endurance limit is only one-half of minimum tensile strength when the minimum load is zero.

Minimum stress curves were constructed which passed through three points: where minimum stress equals minimum tensile strength when the stress ratio is equal to one; where minimum stress equals zero when the stress ratio is equal to zero; and where minimum stress equals a minus one-third minimum tensile strength when the stress ratio is equal to minus one.

In the above plot, it was assumed that the allowable maximum stress in a non-corrosive environment or one which is effectively inhibited is equal to 1/3 of the minimum tensile strength when the stress ratio is 1/3. Our experience indicates that this assumption is realistic. A line was then drawn through these points parallel to the "maximum allowable stress in air" lines. These lines intersect the minimum stress curves at a point that is the maximum allowable stress when the stress ratio is equal to 1. This point was found to be 41,750 psi for the API Class C rod and 53,500 psi for the API Class D rod. The other allowable maximums and minimums needed to calculate the maximum allowable stress when the stress ratio is equal to other values between 1 and -1 were picked. With these data the curves shown in Fig. 19 were constructed.

Loading rod strings to the stresses indicated by Fig. 19 does not allow for weak links. Slim-hole couplings are weak links, and the allowable stress at a specific stress ratio must be decreased if the rod section contains slim-hole couplings. At this time, it is not known exactly how much to derate because of slim-hole couplings. It is known that the slim-hole couplings for one-inch rods do not cause excessive failures. Therefore it is assumed that a ratio of net coupling area to net rod area of 2.0606 is the minimum ratio that will allow the string to handle the stresses shown in Fig. 19. Data on standard couplings are given in Table 4, and data on slim-hole couplings are given in Table 5. The last column of Table 5 contains the derating factors determined by using the above procedure. Further study by engineering personnel involved with this type of operation will be necessary to refine the derating factors.

Table 4

DATA ON STANDARD, FULL SIZE SUCKER ROD COUPLINGS

NOMINAL COUPLING SIZE	O.D., IN.	O.D. AREA, IN.^{2}	I.D., (Q) IN.	I.D. AREA, IN.^{2}	Δ AREA, IN.^{2}	ROD AREA IN.^{2}	$\left(\frac{\Delta \text{ AREA}}{\text{ROD AREA}}\right)$
5/8"	1.5000	1.7671	.955	.7163	1.0508	.307	3.4228
3/4"	1.6250	2.0739	1.080	.9160	1.1579	.442	2.6197
7/8"	1.8125	2.5802	1.205	1.1404	1.4398	.601	2.3957
1"	2.1875	3.7582	1.393	1.5240	2.2342	.785	2.8461
1 1/8"	2.3750	4.4301	1.580	1.9607	2.4694	.994	2.4843

Table 5

DATA ON SLIM HOLE SUCKER ROD COUPLINGS

NOMINAL COUPLING SIZE	O.D., IN.	O.D. AREA, IN.^{2}	I.D., AREA, IN.^{2}	Δ AREA, IN.^{2}	$\left(\frac{\Delta \text{ AREA}}{\text{ROD AREA}}\right)$	DERATING FACTOR, $\left[\frac{\left(\frac{\Delta \text{ AREA}}{\text{ROD AREA}}\right)}{(2.0606)}\right]$
5/8"	1.250	1.2272	.7163	.5109	1.6642	0.8076
3/4"	1.500	1.7671	.9161	.8510	1.9253	0.9343
7/8"	1.625	2.0741	1.1404	.9337	1.5536	0.7540
1"	2.000	3.1416	1.5240	1.6176	2.0606	1.0000

Assuming that a 1" slim hole coupling has an adequate Δarea/rod area, but just adequate, the derating factor is 1.0. The derating factors for the smaller size couplings are listed in Table 5.

As an example, the use of 7/8" slim hole couplings should cause the allowable stress on a rod string to be decreased from an allowable stress of "S_A" to an allowable stress of $S_A \times (1.5536/2.0606)$ = "0.7540 S_A".

The selection of the percentages of each size rod in a tapered string is presented in Table 1 (API RP 11L, Table 1).

XI.

SUCKER ROD SYSTEM PUMPING EFFICIENCY

The efficiency of a sucker rod pumping system is dependent on several variables. Some of these are related to the surface pumping equipment and operation, while others involve downhole equipment and operation. These will be divided into two basic efficiencies as far as this paper is concerned.

The following procedure can be used to determine pumping unit efficiency:

PUMPING UNIT EFFICIENCY

Assumed component efficiencies:

(1) Spur gear, including bearings

(a) Double reduction gears and bearings, worn:
0.93 per set, or $(0.93)^2 = 0.865$

(b) Double reduction gears and bearings, new:
0.96 per set, or $(0.96)^2 = 0.92$

(2) Crank pin roller bearing

(a)	Worn	0.98
(b)	New	0.98

(3) Equalizer bearing

(a)	Worn	0.96
(b)	New	0.98

(4) Saddle bearing

(a)	Worn	0.96
(b)	New	0.98

(5) V-belt drive

(a)	Worn	0.96
(b)	New	0.98

Efficiency from the driven sheave on gear box through the saddle bearing, unit fully loaded:

(1) Worn unit = 0.865 x 0.98 x 0.96 x 0.96 = 0.781

(2) New unit = 0.92 x 0.98 x 0.98 x 0.98 = 0.866

Efficiency from the prime mover sheave, through V-belt drive, through the saddle bearing, unit fully loaded:

(1) Worn unit = 0.781 x 0.96 = 0.750

(2) New unit = 0.866 x 0.98 = 0.849

The loss in efficiency, friction horsepower or friction torque, will not decrease appreciably as the loads at the polished rod are decreased. Assuming that friction horsepower and friction torque remain constant, the load on the gear box and the load on the prime mover sheave are approximated as follows:

Friction Torque:

(1) Worn unit = 1 — 0.781 = 0.219 of API gear box torque rating.

(2) New unit = 1 — 0.866 = 0.134 of API gear box torque rating.

Friction Horsepower:

(1) Worn unit = 1 — 0.750 = 0.250 of nominal pumping unit horsepower rating.*

(2) New unit = 1 — 0.849 = 0.151 of nominal pumping unit horsepower rating.*

*Assuming that the nominal pumping unit horsepower rating is equal to the API gear box torque rating divided by 4960.

Knowing the torque at the polished rod, the torque on the gear box can be determined as follows:

Example Problem:

To find the peak torque on the gear box when the polished rod peak torque is known:

Given: (1) Beam type pumping unit is new.

(2) Calculated peak torque at the polished rod = 91,200 in.-lbs.

(3) API gear box torque rating = 114,000 in.-lbs. at 20 strokes per minute.

(4) Pumping speed = 20 strokes per minute.

Find: Peak torque on the gear box.

Solution:

Step 1. Peak torque at the polished rod divided by the API gear box torque rating = 91,200 in.-lbs./114,000 in.-lbs. = 0.8.

Step 2. On Fig. 20, draw a line parallel to the Y-Axis through 0.8 on the X-Axis to intersect the new unit line.

Step 3. Draw a line parallel to the X-Axis through the point from where the line drawn in Step "2" intersects the new unit curve to the Y-Axis.

Step 4. Read 0.857 on the Y-Axis. This indicates that the peak torque at the polished rod will be equal to 0.857 of the peak torque on the gear box. This is defined as the beam pumping unit torque efficiency factor.

Step 5. The peak torque at the polished rod divided by the pumping unit torque efficiency factor = peak torque on the gear box = 91,200 in.-lbs./0.857 = 106,400 in.-lbs.

Knowing the polished rod horsepower, the prime mover brake horsepower can be determined as follows:

Example Problem:

To find the prime mover brake horsepower when the polished rod horsepower is known:

Given: (1) Beam type pumping unit is worn.

(2) Measured polished rod horsepower = 9.2

(3) API gear box torque rating = 114,000 in.-lbs. at 20 strokes per minute.

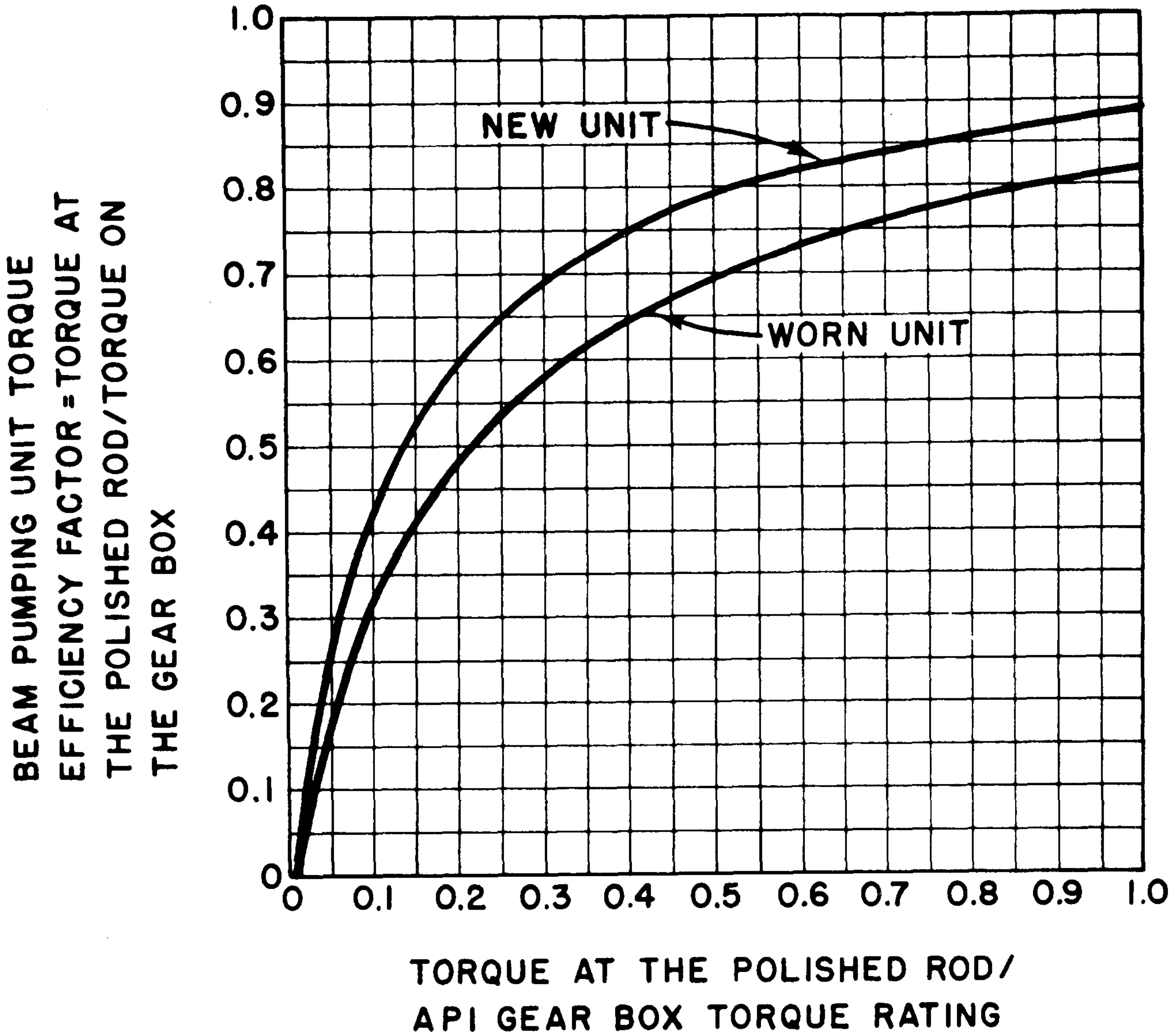
(4) Pumping speed = 20 strokes per minute.

Find: Prime mover brake horsepower.

Solution:

Step 1. 4960 times polished rod horsepower divided by API gear box torque rating = 4960 x 9.2/114,000 = 0.4.

Step 2. On Fig. 21, draw a line parallel to the Y-Axis through 0.4 on the X-Axis



BEAM PUMPING UNIT TORQUE
EFFICIENCY FACTOR

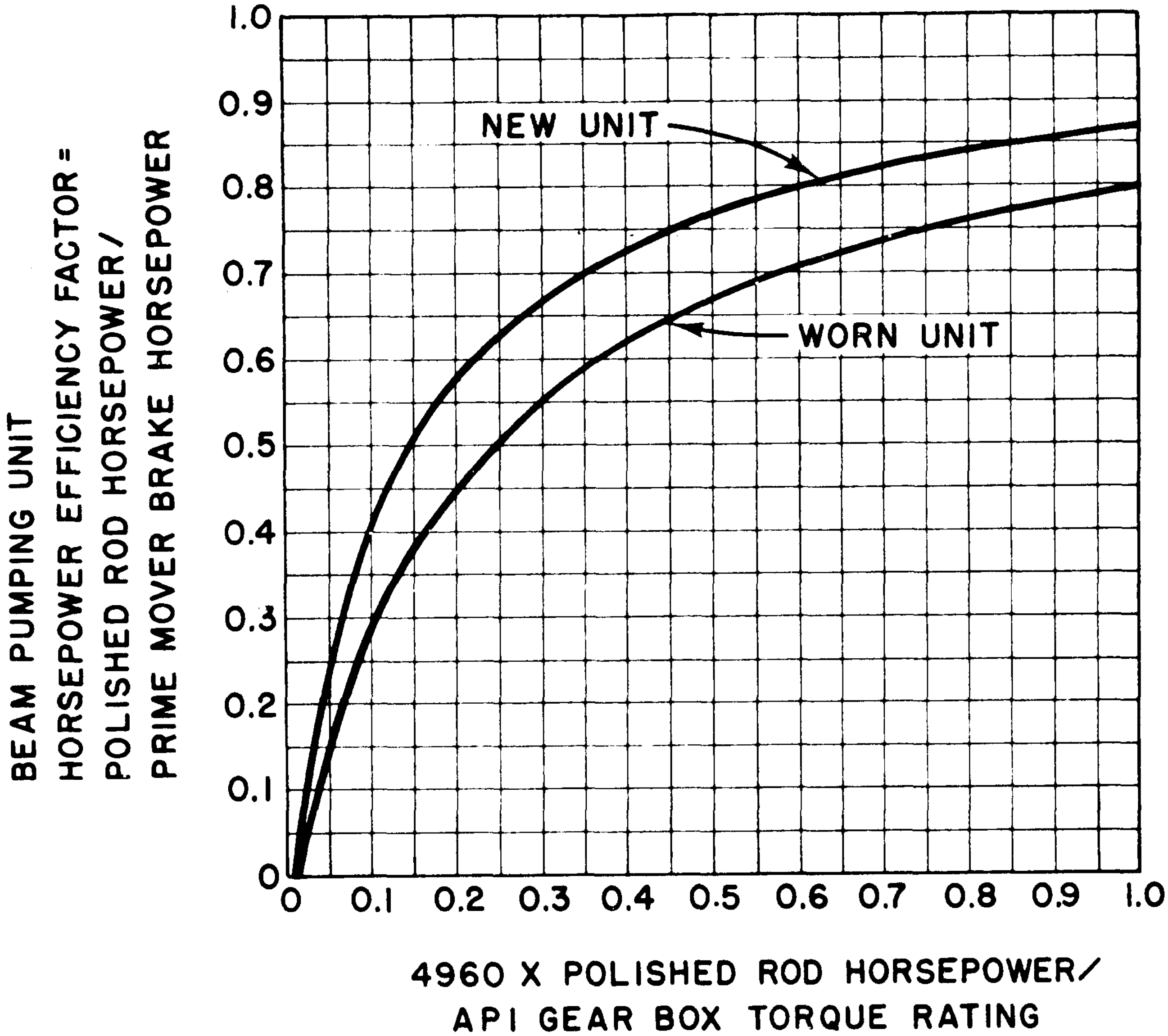
FIG. 20

to the "worn unit" curve.

Step 3. Draw a line parallel to the X-Axis through the point from where the line drawn in Step "2" intersects the worn unit curve to the Y-Axis.

Step 4. Read 0.615 on the Y-Axis. This indicates that the polished rod horsepower will be equal to 0.615 of the prime mover horsepower. This is defined as the beam pumping unit horsepower efficiency factor.

Step 5. The polished rod horsepower divided by the pumping unit horsepower efficiency factor = prime mover brake horsepower = 9.2/0.615 = 14.95.



BEAM PUMPING UNIT HORSEPOWER
EFFICIENCY FACTOR

FIG. 21

VOLUMETRIC EFFICIENCY

The second of the efficiencies to be discussed involves those conditions which affect the movement of fluid. The following definitions and calculations are applicable in determining volumetric efficiency:

DEFINITIONS (continued)

PD = Pump displacement, in barrels per day.

Vol. Eff. = Volumetric efficiency = Barrels of fluid per day, BFPD, measured in the stock tank at atmospheric pressure and 60° F, divided by pump displacement, PD, which is BFPD/PD.

S_L = Slippage = Leakage past the plunger, S_{LP} during the upstroke, plus leakage due to the delayed closing of the standing and traveling valves, S_{LV}, in barrels per day, divided by the pump displacement, in bbls per day.

With good design, leakage should not exceed 2 per cent of PD, and total slippage (S_L) should not exceed 3 to 5 per cent with a new, well-designed subsurface pump.

C = Clearance volume, which is the volume between the standing and traveling valves, in cubic inches, at the instant the traveling valve closes after completing the downstroke divided by the plunger displacement, in cubic inches. The plunger displacement, in cubic inches, is equal to the area of the plunger, in square inches, multiplied by the plunger stroke length, in inches.

K = Compressibility, which equals the change in volume of the fluid being pumped between the volume at pump discharge condition when compared to the volume at:
(a) discharge condition
(b) suction condition
(c) stock tank condition

B_S = Formation volume factor at suction conditions, which is barrels of fluid drawn into the pump per day, measured at the temperature and pressure existing between the standing and traveling valves at the end of the plunger upstroke, divided by the standard barrels of stock tank liquid per day corrected to atmospheric pressure and 60°F.

Vol. Eff. @ discharge conditions $= 1 - (K + CK + S_L)$

Vol. Eff. @ suction conditions $= 1 - S_L + CK/(1 - K)$

Vol. Eff. @ stock tank conditions $= (1 - S_L + CK/(1 - K))/B_S$

Sample Problem:

Given:

PD = 100 BPD

$S_L = 0.04$

$C = 0.10$

$K = 0.01$

$B_S = 1.30$

Find: Vol. Eff. @ discharge, suction and stock tank conditions.

(a) Vol. Eff. @ discharge conditions $= 1 - (0.01 + 0.10 \times 0.01 + 0.04) = 1 - 0.051 = 0.949$

(b) Vol. Eff. @ suction conditions $= 1 - 0.04 + (0.10 \times 0.01)/(1 - 0.01) = 1 - 0.04 + 0.001 = 0.961$

(c) Vol. Eff. @ stock tank conditions $= 0.961/1.30 = 0.74$

XII.

HARMONIC VIBRATION OF SUCKER ROD STRING

The motion of a reciprocating sucker rod string approximates simple harmonic motion. Obvious examples of simple harmonic motion include pendulum clocks, playground swings and the tone caused by the vibration of organ pipes. In the case of an organ pipe closed at one end, the fundamental frequency of the column of air inside the pipe is equal to the acoustic velocity of sound in air divided by four pipe lengths, assuming that the column of air in the pipe contains one-fourth of a wave length when sounding its fundamental tone. API RP 11L states that in actual practice it has been found the velocity of force propagation in a sucker rod system immersed in fluid is approximately 16,300 ft./sec. Adapting the principle of the organ pipe to a vibrating non-tapered sucker rod string, the undamped fundamental frequency of vibration can be calculated as follows, using the API recommended acoustic velocity value:

$$N_o = \frac{16,300 \text{ ft/sec} \times 60 \text{ sec/min}}{4L} = \frac{244,500}{L}$$

vib./min. where L = Length of sucker rod string, in feet.

N_o = Fundamental frequency of a non-tapered rod string.

In his book, "Dynagraph Analysis of Sucker Rod Pumping," J. C. Slonneger presents another method which involves the elongation of a non-tapered sucker rod string due to its weight alone. He referred to this as static elongation ($\overline{SE}$). Based on his work, $\overline{SE} = \frac{L^2}{1,320,000}$, and Slonneger states that the fundamental frequency (F) for any sucker rod system is: $F = \frac{206}{\sqrt{SE}}$, vib./min.

For all practical purposes F and N_o are basically equivalent. The API method also discusses the undamped frequency of tapered rod strings, N_o'.

Assuming the Slonneger equations are valid, then the fundamental rod frequency (F), divided by the strokes per minute of the pumping cycle (SPM), will indicate the order of pumping. As an example, assuming a fundamental rod frequency of 45 vib./min. and a speed of 15 SPM, the order of pumping is 45/15 or 3.0 vibrations/stroke. Likewise, assuming F = 60 vib./min. and SPM = 15, the order would be 4.0 vibrations/stroke. Following along this line, it is possible to construct a family of curves on a graph of SPM versus the length of the sucker rod string for the various orders of pumping. This was done by our "forefathers" and has been quite useful in the past as a quick general reference source when forecasting the shape of a dynamometer card corresponding to its order of pumping. API RP 11L has made it possible to further refine this approach.

API RP 11L has also made it possible to calculate the desirable (non-synchronous) pumping speeds in the case of tapered rod strings based on orders of pumping by using the following approach:

N_o' = Undamped natural frequency of a tapered rod string.

N_o' = 16,300 ft./sec. x 60 sec./min. x $F_c \div$ 4L ft.

N_o' = (245,000 x F_c) $\div$ L

Note: 244,500 has been rounded to 245,000 in the API approach

F_c = A constant of proportionality which depends on the rod design. Table 1, (API RP 11L, Appendix A and Table 1, Column 5).

L = Length of rod string, in feet.

Desirable pumping speeds are most likely to occur with a combination of N_o' and the following pumping orders: $N_o' \div 1.5$, $N_o' \div 2.5$, ... Likewise, undesirable pumping speeds occur with a combination of N_c' and the following orders: $N_o' \div 1$, $N_o' \div 2$, ...

When pump depth, pumping speed and F_c are known and the sucker rod string is vibrating without the influence of undue dampening effects, it is possible to calculate and construct families of curves which can be used to determine pumping orders. Figures 22, 23, 24 and 25 have been constructed on that basis. The following is an example of how to use these figures to determine which pumping order is likely to occur in a particular situation:

> Using Fig. 22, which is based on an F_c of 1.0, and assuming a pump depth of 6000 feet, and a pumping speed of 12 SPM, a 3.5 order non-synchronous pumping order should result.

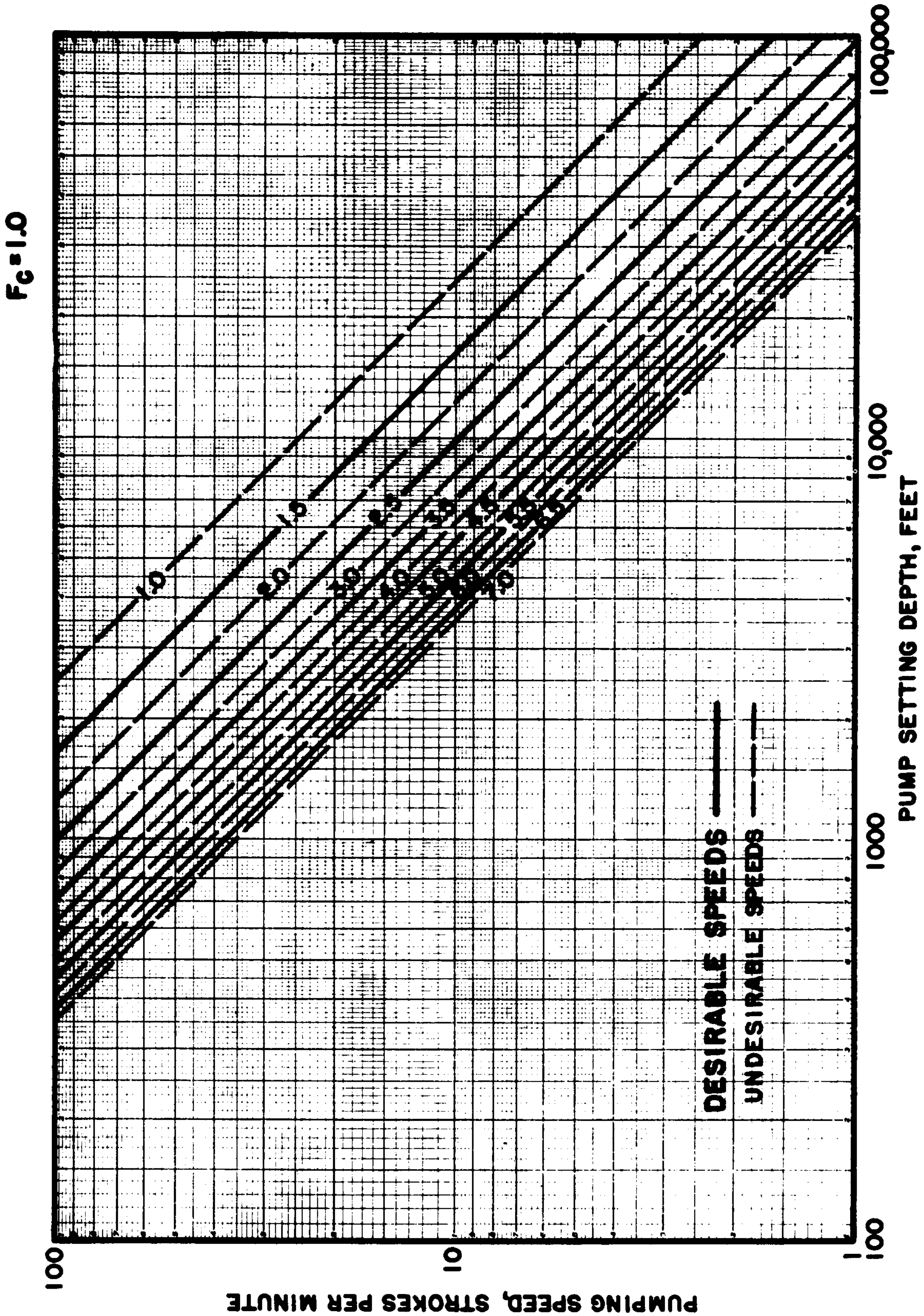
Maximum practical pumping speeds are non-synchronous speeds that are well below the free fall speed of the rods in the fluid being pumped. Pumping speeds (N), in stokes per minute, and stroke lengths (S), in inches, which result in an acceleration factor (S x N^2/70,500) greater than 0.3 are believed to be undesirable. Installations operating at acceleration factors approaching 0.5 are known to be in service, but the history of these installations indicates extensive downtime due to rod, pin and coupling breaks.

Minimum practical design pumping speeds are determined by several factors. Experience indicates that the industry is probably investing too much in the rod pumping installation if the acceleration factor is below 0.225 (three-fourths of the recommended maximum). Experience also indicates that a complete installation designed to operate at an apparent acceleration factor of 0.225 will result in a good balance between operating cost and investment.

CHARACTERISTIC DYNAMOMETER CARDS

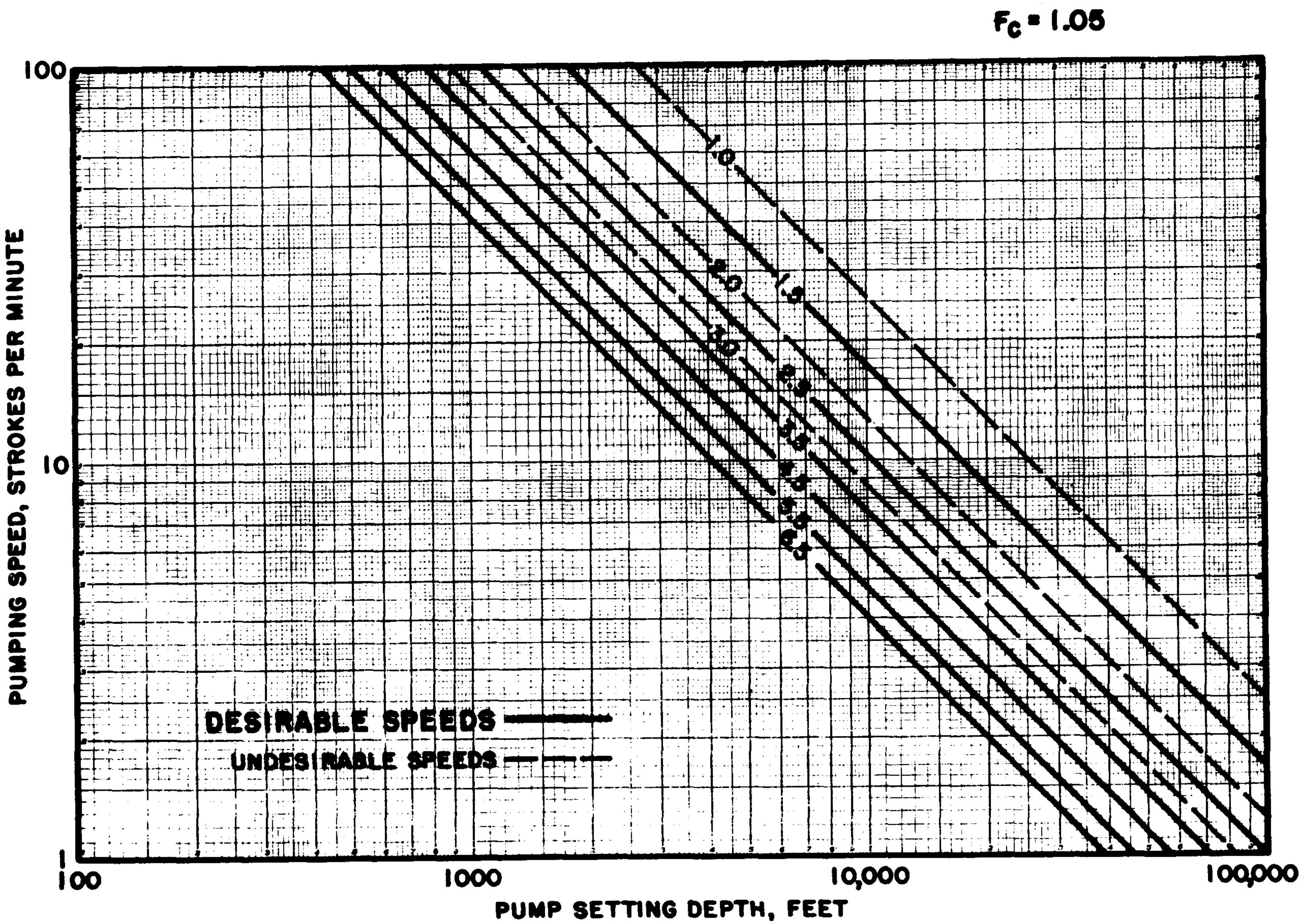
Since sucker rod pumping systems conform, in general, to the principles of simple harmonic motion, it is possible to predict the characteristic shapes of dynamometer cards if pump depth, pump speed and F_c are known. As mentioned earlier, Figs. 22, 23, 24 and 25 are very helpful in determining the orders of cards.

First order pumping situations are not encountered in oil well pumping due to the high pumping speed required and the limitation of



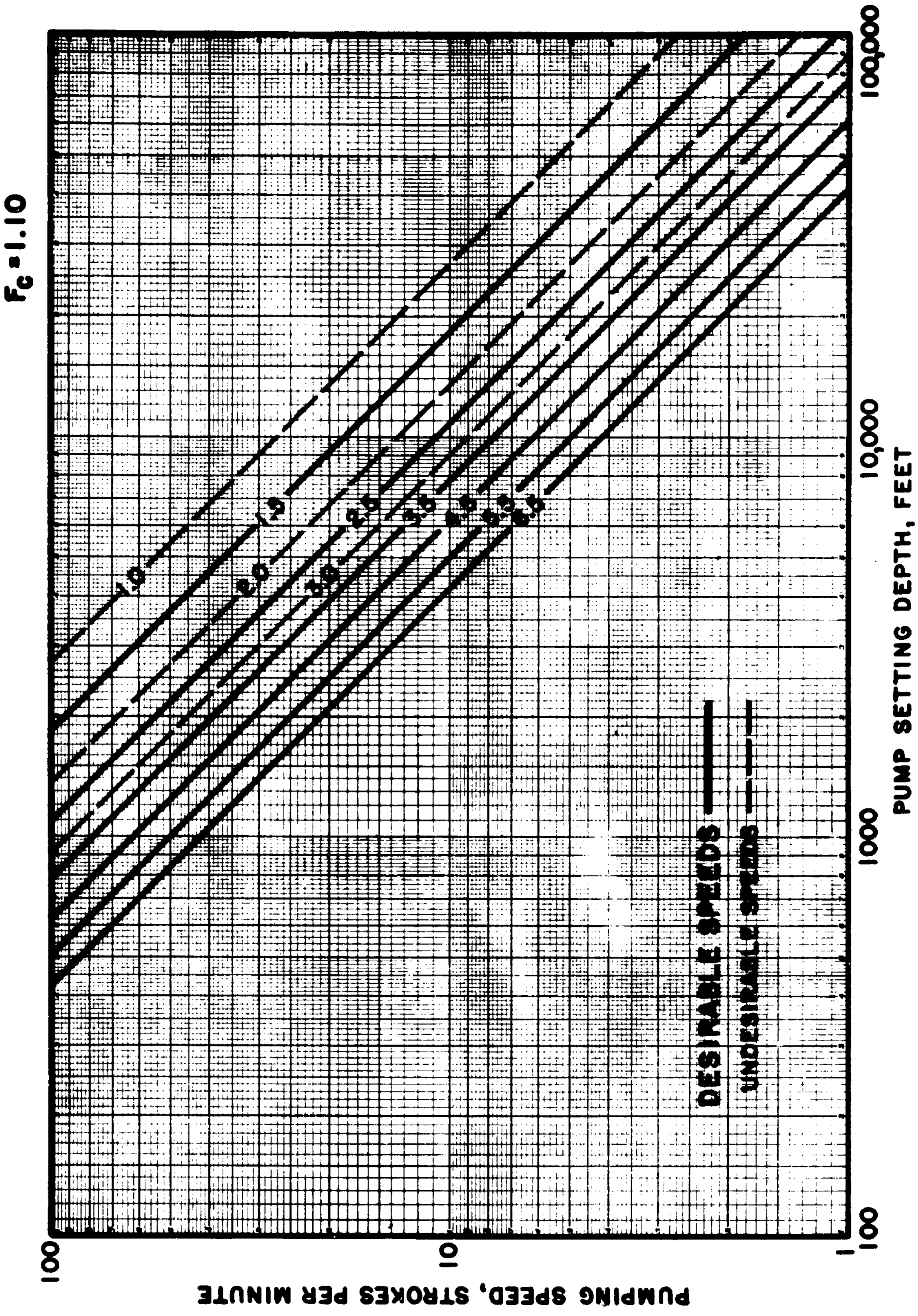
$F_G = 1.0$
PUMP SETTING DEPTH, FEET
100
1000
10,000
100,000
PUMPING SPEED, STROKES PER MINUTE
100
10
1
DESIRABLE SPEEDS
UNDESIRABLE SPEEDS

FIG. 22



$F_C = 1.05$
100
10
1
100
1000
10,000
100,000
PUMPING SPEED, STROKES PER MINUTE
PUMP SETTING DEPTH, FEET
1.0
1.5
2.0
3.0
DESIRABLE SPEEDS
UNDESIRABLE SPEEDS

FIG. 23



$F_C = 1.10$
PUMP SETTING DEPTH, FEET
100
1000
10,000
100,000
PUMPING SPEED, STROKES PER MINUTE
100
10
1
DESIRABLE SPEEDS
UNDESIRABLE SPEEDS
1.0
1.5
2.0
2.5
3.0

FIG. 24

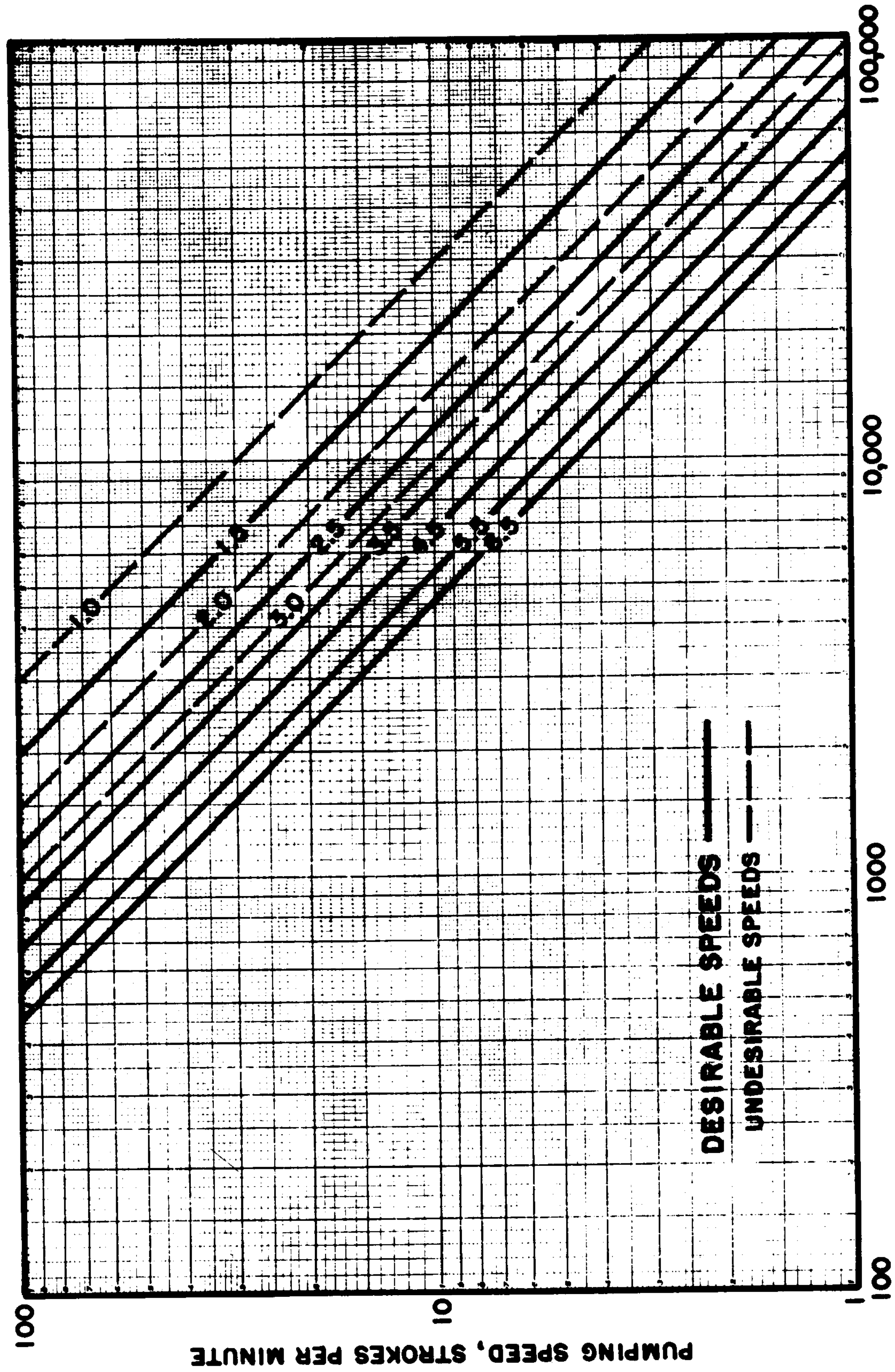
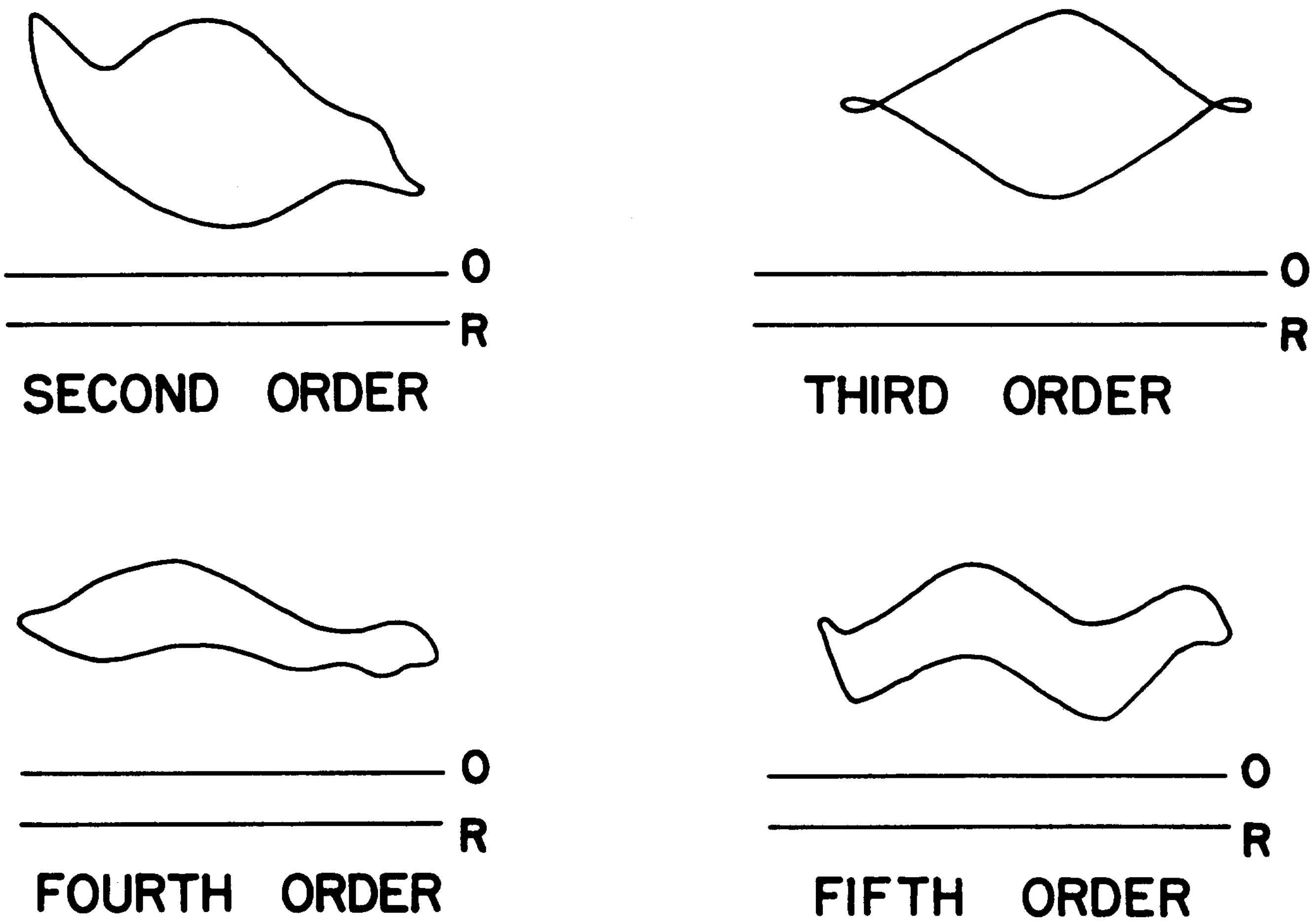

$F_C = 1.20$
PUMP SETTING DEPTH, FEET
100
1000
10,000
100,000
PUMPING SPEED, STROKES PER MINUTE
100
10
1.0
1.5
2.0
2.5
3.0
DESIRABLE SPEEDS
UNDESIRABLE SPEEDS

FIG. 25

the free fall speed of rods at the required depth. Situations yielding second to fifth order cards are probably the most common encountered today. Based on representative cards in the authors' files and other examples with which they are familiar, the idealized cards in Fig. 26 have been constructed to portray the various orders.



IDEALIZED ORDER OF DYNAMOMETER CARDS

FIG. 26

Following this approach, the characteristics of actual dynamometer cards can be compared to idealized orders or cards. If the actual card does not resemble the forecast card for that particular order, based on appropriate data, the card analyst has an indicator of possible trouble. By changing the variables reflecting the natural frequency of vibration, a problem area can often be corrected, or induced, as the case may be.

The use of the computer has made it possible to forecast the configuration of dynamometer cards using the dimensionless parameters N/N_o' and F_o/Sk_r as the controlling variables. By knowing these two variables, which are easy to calculate, a representative card can be obtained from a "library" of representative cards. The comparison of the actual card with the representative card also provides an indication as to whether or not problems exist. Figure 27 is a collection of representative cards using N/N_o' and F_o/Sk_r as the controlling variables.

It is interesting to note on Fig. 27 the change in the card from extreme overtravel when the N/N_o' value is large (.45) and F_o/Sk_r is small (.1) to extreme undertravel when the N/N_o' value is small (.1) and F_o/Sk_r is large (.6).

REPRESENTATIVE DYNAMOMETER CARDS

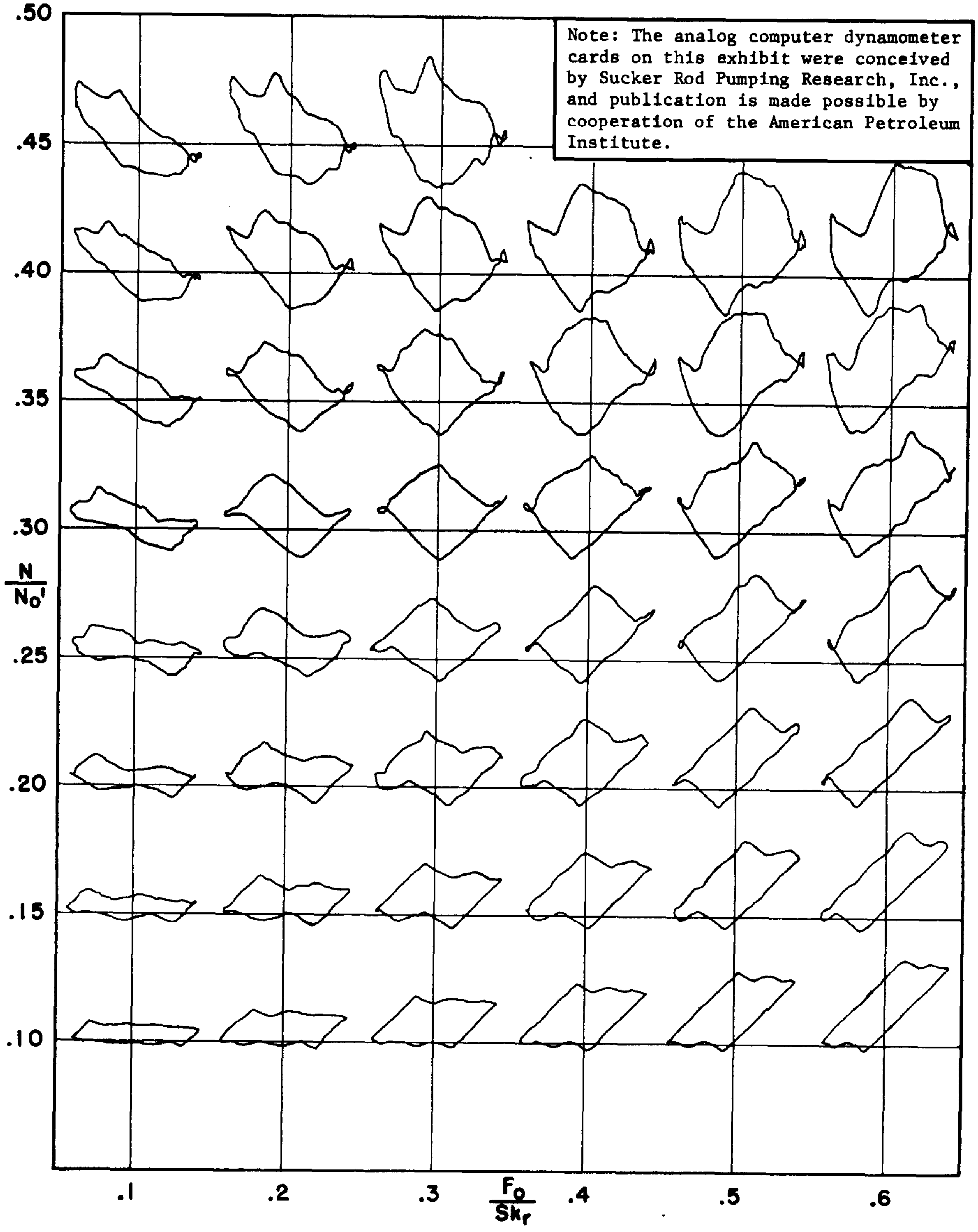


FIG. 27

XIII.

TYPICAL PUMPING CYCLE

There probably is no such card as a "typical dynamometer card" due to the inherent factors which influence its appearance. The following is presented as a typical card expressing a typical pumping cycle so that the major sequences occurring in a pumping cycle can be shown.

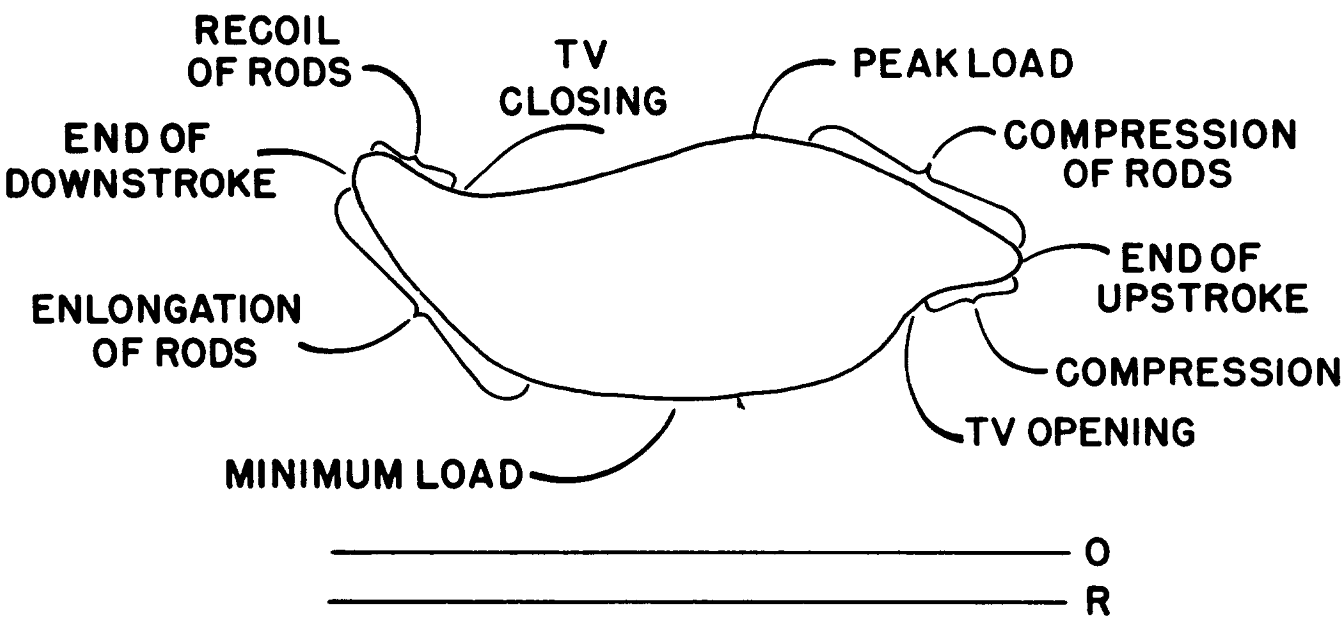


FIG. 28

XIV.

VALVE ACTION DURING THE PUMPING CYCLE

Figure 29 presents schematic diagrams of the standing and traveling valves during the pumping cycle. It is also helpful in visualizing valve operation during the "traveling valve" and "standing valve" tests.

The following is a discussion of the valve action and associated loads during the pumping cycle. The starting point is at the start of the plunger upstroke. At that time the standing valve (SV) is closed.

Step 1. The traveling valve (TV) closes when the pressure below the TV, P_1, approaches the pressure above the TV, P_3.

Step 2. The SV opens when the projected area of the top of the SV seat, multiplied by the pressure between the TV and SV, P_1, becomes less than the projected area of the bottom of the SV seat, multiplied by the pressure below the SV, P_2. There are other minor forces acting, but they will be ignored because of the small values involved.

Step 3. At the start of the downstroke of the plunger, the SV closes when the pressure above the SV, P_1, approaches the pressure below the SV, P_2.

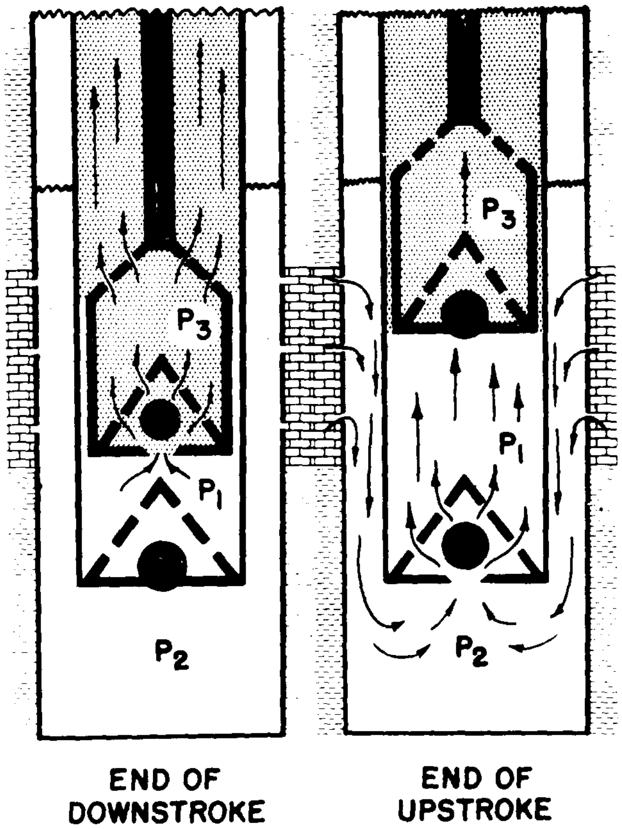


FIG. 29

Step 4. The TV opens if the projected area of the top of the TV seat, multiplied by the pressure above the TV, P_3, becomes less than the projected area of the bottom of the TV seat, multiplied by the pressure below the TV, P_1.

The pressure below a closed valve must be greater than the pressure above the valve before the valve can be opened. This causes difficulties which include:

1. Gas breakout: A barrel of "live" or saturated crude oil will normally release gas when the pressure is decreased.
2. Sucker rod buckling: On the downstroke, a portion of the required force must be obtained from the weight of the sucker rod string. Since the lower portion of the sucker rod string is in compression, rod buckling results unless the the necessary portion of the rod string for the required downward force is comprised of centralized sinker bars. Sucker rod buckling will cause excessive rod and tubing wear above the pump and many premature valve rod failures.

EXAMPLES OF VALVE ACTION PROBLEMS

1. Static force required to unseat TV on downstroke:

$$W_{Bf} + \left(D_2^2 \times \frac{\pi}{4}\right) x\ P_3 < \left(D_1^2 \times \frac{\pi}{4}\right) x\ P_1$$

Note: For the purpose of this illustration, the weight of the ball in fluid, W_{Bf}, is small and will be ignored.

$$P_1 > P_3 \left[\frac{D_2^2 \times \pi/4}{D_1^2 \times \pi/4}\right]$$

Using data from Figure 31, for a 1½" pump, D_1 = 0.656" and D_2 = 0.723"

$$P_1 > P_3\ (1.216)$$

$$F\downarrow > (P_1 - P_3)\left(D_p^2 \times \frac{\pi}{4}\right)$$

$$F\downarrow > 0.216\ P_3 \times \left(1.5^2\ \frac{\pi}{4}\right)$$

$$F\downarrow > 0.382\ P_3$$

If P_3 = 2000 psia, $F\downarrow > 0.382 \times 2000$

$F\downarrow$ > 764 lbs.

Wt. of 1½" polished rods = 6.008#/ft. in air.

$F\downarrow$ > 764/6.008 (1 - B), or 764/5.24, or 146' of 1½" PR.

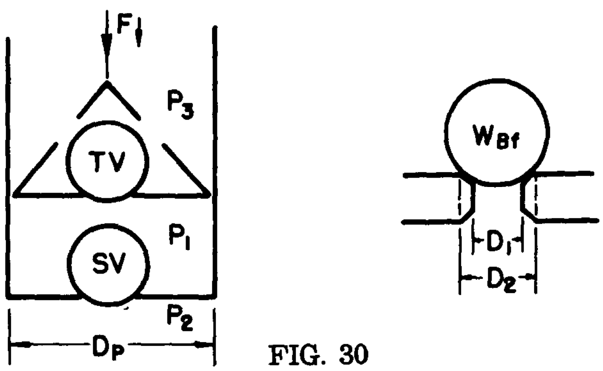
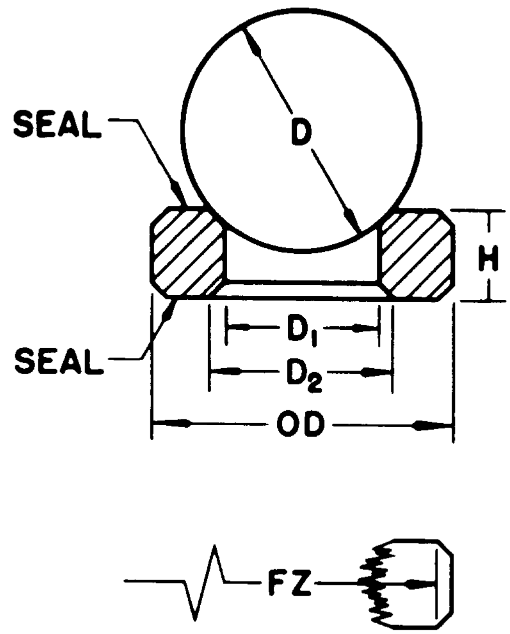


FIG. 30



DETAIL OF SEAL AND
45° CHAMFER OR RADIUS-
OPTIONAL WITH MANUFACTURER

V11-VALVE, BALL AND SEAT

1	2	3	4	5	6	7	8
Dimensional	Part Number						
Symbol	V11-106	V11-125	V11-150	V11-175	V11-200	V11-225	V11-250
D	0.625	0.750	0.938	1.125	1.250	1.375	1.688
H	0.500	0.500	0.500	0.500	0.500	0.500	0.500
FZ max.	0.767	0.892	1.111	1.331	1.421	1.631	1.921
OD $^{+.000}_{-.005}$	0.793	0.918	1.168	1.388	1.478	1.720	2.010
*D_1, in.	0.50	0.578	0.656	0.844	0.937	1.062	1.312
*D_2, in.	0.555	0.638	0.723	0.907	1.000	1.125	1.375
D_1^2, in.^{2}	0.250	0.334	0.430	0.712	0.878	1.130	1.721
D_2^2, in.^{2}	0.308	0.407	0.523	0.823	1.000	1.266	1.891
D_1^2/D_2^2	0.817	0.821	0.822	0.865	0.878	0.893	0.910
D_2^2/D_1^2	1.236	1.219	1.216	1.156	1.139	1.120	1.099

*Supplied by Harbison-Fischer Mfg. Co.

FIG. 31

2. Static pressure required to unseat SV on upstroke:

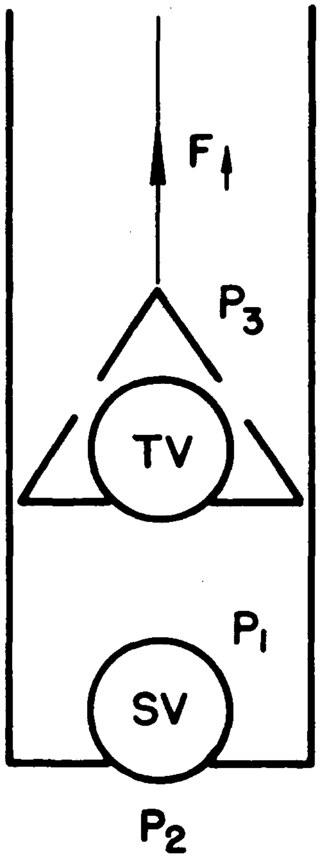


FIG. 32

Given: $P_2 = 50$ psig, $T = 100°F$, $D_1 = 1.062"$, $D_2 = 1.125"$, $D_p = 1\text{-}3/4"$, $S_p = 54"$, N = 20 SPM, gas anchor = 10' x 1-1/4" nominal line pipe with a pressure drop of 2.3 psi.

Assume: Neglecting any change in formation volume factor, 2 standard ft³ of gas are released per bbl of oil per psi decrease in pressure; std conditions = 14.4 psia at 60°F; 1 bbl = 9702 in.³; waste space (clearance volume) between TV and SV at bottom of downstroke = 5 in.³.

$$P_1 \times \left(D_2^2 \frac{\pi}{4}\right) < P_2 \times \left(D_1^2 \times \frac{\pi}{4}\right)$$

$$P_1 < P_2 \left(\frac{D_1^2}{D_2^2}\right)$$

$$P_1 < (50 + 14.4) \times \frac{1.13}{1.27}$$

$P_1 < 64.4 \times 0.89$; $P_1 < 57.3$ psia

$57.3 - 14.4 = 42.9$ psig

$$2 \text{ standard } \frac{\text{ft.}^3}{\text{Bbl.} \times \frac{\#}{\text{in.}^2}} \times 1728 \frac{\text{in.}^3}{\text{ft.}^3}$$

$$= 3456 \frac{\text{in.}^3}{\text{Bbl.} \times \frac{\#}{\text{in.}^2}}$$

$$\frac{3456 \dfrac{\text{in.}^3}{\text{Bbl.} \times \frac{\#}{\text{in.}^2}}}{9702 \dfrac{\text{in.}^3}{\text{Bbl.}}}$$

$$= 0.356 \frac{\text{in.}^3}{\text{in.}^3 \times \frac{\#}{\text{in.}^2}}$$

= 0.356 in.³ of gas that will be released from each cubic inch of oil per psi decrease in pressure, measured at standard conditions.

Therefore, if the pressure is reduced 7.1 #/in.², 5 in.³ of oil will release 7.1 x 5.0 x 0.356 = 12.6 in.³ measured at 14.4 psi and 60° F.

This 12.6 in.³ will occupy

$$12.6 \times \left(\frac{14.4}{42.9 + 14.4}\right) \times \left(\frac{460 + 100}{460 + 60}\right)$$

= 3.41 in.³ measured at 42.9 psi and 100°F, assuming the gas behaves as an ideal gas.

The total plunger displacement will be equal to the area of the plunger, $(1.75)^2 \times \frac{\pi}{4}$ in.² multiplied by the stroke length, 54 in., and will equal 130 in.³. 3.15 cu. in. of this displacement, measured at 50 psig, P_2, — 2.3 psig, the pressure drop through the gas anchor, will be filled by the gas released by the oil in the 5 cu. in. clearance volume before the standing valve opens. The remainder will be filled with oil and the gas released from the oil by the 2.3 psi pressure drop through the gas anchor.

Total volume to be filled with oil and gas during the upstroke = 130.00 — 3.15 = 126.85 in.^{3}.

Let the portion of this volume that will be filled with gas = X; let the portion that will be filled with oil = 126.85 — X. X will also =

$$(126.85 - X) \times 0.356 \times 2.3 \times \frac{14.4}{50 - 2.3 + 14.4}$$

$$\times\ 560/520 = 25.94 - 0.20X$$

$$1.2045X = 25.94$$

$$X = 25.94/1.2045 = 21.54 \text{ in.}^3$$

$$126.85 - 21.54 = 105.31 \text{ in.}^3$$

$$\text{Check: } 105.31 \times 0.356 \times 2.3 \times \frac{14.4}{62.1} \times \frac{560}{520} = 21.54 \text{ in.}^3$$

Assuming no slippage or pressure drop through the standing valve, volumetic efficiency = 100 (130 —3.15 — 21.54)/130 = 100 x 105.31/130 = 81%

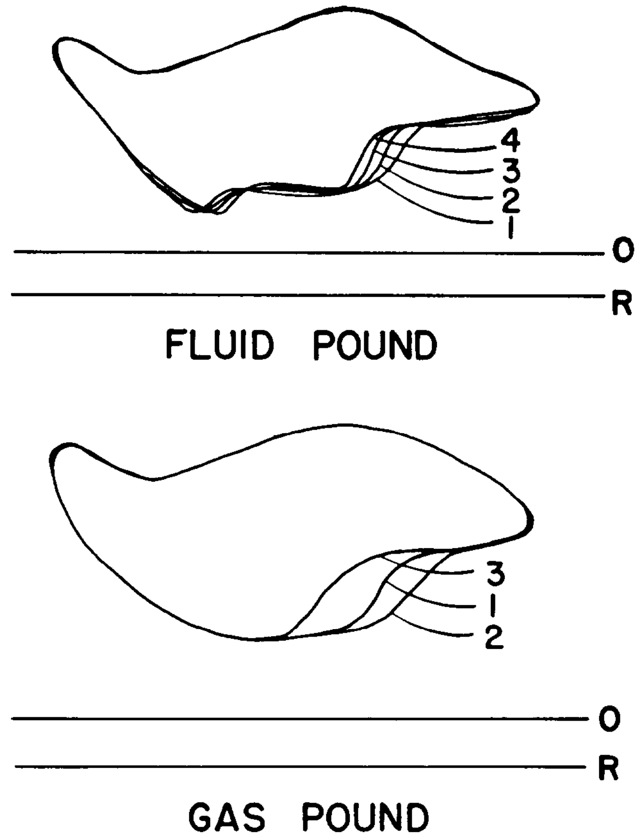


FIG. 33

FLUID AND GAS POUNDS

XV.

FLUID AND GAS POUNDS

There are two basic types of "pounds" experienced in well pumping: (1) fluid pounds and (2) gas pounds. In reality, both of these are fluid pounds but vary in nature. They are both caused by the pump not completely filling with liquid on the upstroke.

FLUID POUND

In the case of fluid pounds, the first portion of the downstroke will be gas compression until there is sufficient force generated to cause the traveling valve to open, causing a shock wave to travel through the pumping system. There will usually be only a slight change in load while compression is taking place. When the traveling valve opens, the weight of the fluid is transferred to the standing valve, and that transfer causes a sharp decrease in load. That change is referred to as the "fluid" pound. A fluid pound is undesirable but can be tolerated at either end of the stroke. When it occurs near the middle of the stroke, it becomes highly undesirable in that it will:

(1) Cause premature rod failure.
(2) Damage the pump.
(3) Damage the tubing.
(4) Damage the gear box.
(5) Result in deterioration of the entire pumping system at an accelerated rate.
(6) Increase lifting costs.
(7) Reduce fluid production in some cases.
(8) Often unseats the tubing anchor.
(9) May "drive" the tubing anchor deeper.

GAS POUND

A gas pound results when part of the fluid in the pump is in the form of gas, usually in a foamy or frothy condition. It closely resembles a fluid pound except that the liquid-gas ratio is inconsistent on each pump cycle, and more cushioning effect is present on the downstroke than would be experienced with a straight fluid pound. In most cases it is very difficult to distinguish between the two types of pounds merely by dynamometer card analysis. It is often possible to determine that a gas pound situation exists by two symptoms: (1) when the fluid level in the annulus fluctuates substantially due to a foamy condition, and (2) when the "gas pound trace" on a dynamometer card moves up and down but does not usually move progressively toward the end of the downstroke and stabilize as in the case of a fluid pound.

The major differences in fluid and gas pounds are as follows:

	Fluid Pound	Gas Pound
1. Place where pound occurs after pumping cycle has stabilized.	1. Fairly constant.	1. Moves up and down.
2. Progress of pound.	2. Continuous and toward the downstroke end of the cycle until condition stabilizes or pump gas-locks.	2. Initially moves toward the downstroke end of the cycle but will fluctuate back and forth.
3. Size of pump, SPM, SL.	3. Can control by varying pump size, SPM, SL.	3. Some control by varying pump size, SPM and SL, but cannot completely control.
4. Slope of pound on card.	4. Steep when pound occurs in the middle of stroke.	4. Generally less steep when the pound is in the middle of stroke.
5. Fluid level in annulus.	5. Fairly constant when pumping conditions stabilize.	5. Fluctuates up and down.

CONTROL OF POUNDS

A great deal of money can be saved by eliminating or controlling fluid pounds. Commencing with the most economical solution, the following can eliminate or dampen the effects of a fluid pound:

(1) Reduce the SPM.
(2) Increase pump submergence by removing casing pressure.
(3) Shorten the stroke.
(4) Time clock the well so that pump capacity will not exceed well capacity.
(5) Install a back pressure valve on the flow line in some cases.
(6) Reduce the pump size.
(7) Decrease the pump capacity to the well capacity.
(8) Increase the pump compression ratio.
(9) Change the pump setting depth to increase submergence or control type of fluid entering pump.
(10) Be sure tubing (mud anchor) perforations are of sufficient area.
(11) Install a correctly designed gas anchor.

In some cases it is possible to control a gas pound, but in a large number of instances only partial control is possible. In numerous instances when effective gas separation is possible before the fluid enters the pump, it is possible to exercise control over a gas pound. When this is not possible due to the nature of the fluid being produced, only partial control can be exercised.

Possible ways to control gas pounds are:

(1) Materially change the pressure at the pump intake by changing casing pressure, remedial action, etc.
(2) Install a back pressure valve on flow line.
(3) Lower the pump if possible, but in any case be sure the pump intake perforations are not opposite casing perforations, or opposite the producing formation in open-hole completions.
(4) Reduce the differential pressure experienced at the pump intake by corrective design of the subsurface hardware.
(5) Install a properly designed gas anchor when necessary, or use other means

of obtaining a more effective separation of gas prior to pump intake.

XVI.

GAS SEPARATION

It is extremely important to maximize the separation of gas from the produced fluid before it enters the subsurface pump. The following gas separation rules-of-thumb are taught at the well pumping short courses:

1. Large bubbles of gas will rise at a velocity of 0.5 feet per second in a typical well fluid being produced.
2. Pressure drops cause scale precipitation.
3. The pressure drop caused by fluid flowing through the perforations, or slots, in the dip tube, the pressure drop that results from friction in the dip tube, and the pressure drop across the standing valve releases gas that must be pumped.

Assuming these rules are correct, it is concluded that:

1. The area of the perforations, or slots, in the mud anchor should have an area equal to between two and four times the area of the annulus between the mud anchor and the dip tube. Note that this would be the area between the mud anchor and the pump, if the pump were equipped with a top holddown. The ratio should approach 4 if it is known that the well fluids are, or will be, capable of precipitating scale or/and paraffin under adverse gas separation situations.
2. The average downward velocity in the mud-anchor dip-tube annulus must be less than 0.5 feet per second as velocities below this value will normally permit the gas bubbles to separate from the fluid, rise through the downcoming fluid, and pass through the mud anchor slots. If the downward velocity is greater than 0.5 feet per second, only a portion of the gas will be separated, and the volumetric efficiency of the pump will be decreased.
3. The area of the perforations, or slots, in the dip tube should have an area equal to a minimum of 4 times the area of the standing valve. The dip tube should not be installed open-ended, unless there is a valid reason. It is usually run in the hole with the pump, and if it were open-ended, it could and probably would be packed full of paraffin scraped from the tubing.
4. The internal area of the dip tube should be as large as is practical. Therefore, thin-wall pipe should be considered. In addition, friction can be reduced approximately 20 per cent by plastic-coating the interior of the tube, or by using thin-wall plastic pipe.
5. The length of the dip tube should be held to a minimum, but it must be long enough to provide for an adequate quieting volume between the bottom of the mud anchor slots and the top of the dip tube slots. It is suggested that the volume of the quiet space be between 1 and 2 pump volumes. A controllable variable is the length and location of the slotted mud anchor section. For example, if the slots are spaced over a four-foot length, and the slots commence 1.5 feet below the seating nipple, the dip tube would be three feet longer than if the slotted section were only two feet long and started 0.5 feet below the seating nipple.

GAS SEPARATION PROBLEM

Given: Pump capacity must be 300 BFPD; I.D. of casing is 4.892 inches; O.D. of upset tubing is 2-7/8 inches; adequate gas separation will be secured if the average velocity of the fluid in the mud-anchor dip-tube annulus is less than 0.5 feet per second; pump plunger diameter is 1.75 inches, operating at 15 - 80 inch strokes per minute; the pump intake will be above the casing perforations; pump volumetric efficiency = 70 per cent.

Problem: Design a "poor boy" gas anchor.

Solution:

1. Area of annulus between the mud-anchor dip-tube can be determined from the following formula:
   Area of down passage, in.^{2} = (0.00935 x BFPD)/(velocity, ft./sec. x pump volumetric efficiency) = 0.00935 x 300/0.5 x 0.70 = 8.01 in.^{2}

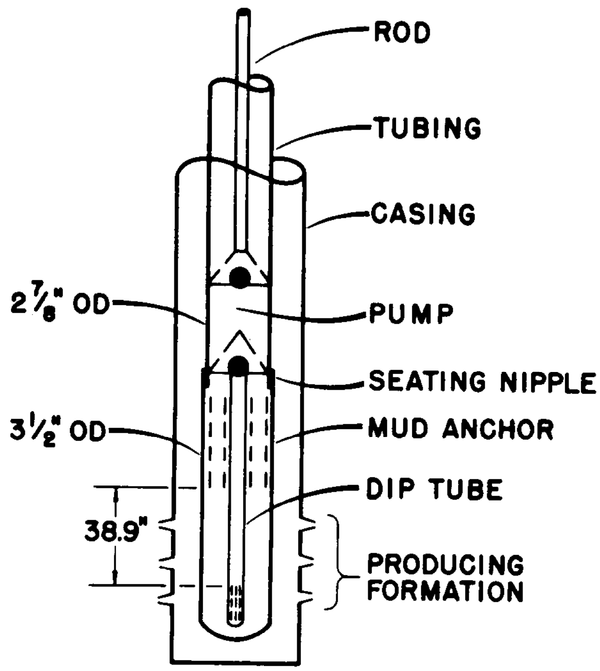


FIG. 34

"POOR BOY" GAS ANCHOR
(See Gas Separation Problem)

2. Area of the mud anchor slots = 8.01 in.^{2} x 4 = 32.04 in.^{2} = 32 — ¼ in. x 4 in. slots.
3. Area of standing valve = 1.062 in.^{2}; area of dip tube slots = 4.25 in.^{2} = 9 — 1/8 in. x 4 in. slots.
4. Size of dip tube = 1¼-in. nominal line pipe; O.D. area = 1.66 in.^{2}, I.D. area = 1.38 in.^{2}.
5. I.D. area of mud anchor = 8.01 in^2 + 1.66 in.^{2}, or 9.67 in.^{2}. Therefore, select 3½-in. O.D. pipe with an I.D. area of 9.90 in.^{2}.
6. Let length of quieting space between bottom of mud anchor slots and top of dip tube slots result in a volume = 2 pump volumes = 2 x 80 x $(1.75)^2$ x 0.7854 = 2 x 192 in.^{3} = 385 in.^{3}. Length of quieting space = 385 in.^{3}/9.90 in.^{2} = 38.9 inches.
7. O.D. of a 2-7/8 in. upset collar = 3.5 in. Therefore, it is suggested that the 3.5-in. O.D. mud anchor be butt-welded to a 2-7/8 in. upset collar.
8. Bottom of mud anchor should be closed to keep out well trash while running the anchor and to keep gas from entering the dip tube.
9. The exact lengths of the mud anchor and dip tube should be determined, considering make-up lengths required.

XVII.

INDICATORS OF MALFUNCTION OR TROUBLE

Several valuable indicators can be used in diagnosing well pumping trouble. These are:

(1) Accurate, complete and representative well tests.

(2) Past history of well and equipment performance.

(3) A "healthy" dynamometer card taken when the well producing equipment and down-hole pumping conditions are representative of the normal producing characteristics of the well.

(4) "Before" and "after" dynamometer cards and fluid level charts to pinpoint causes of trouble.

(5) A dynamometer card taken at the time trouble is being experienced which may show:

   (a) Overtravel or undertravel.
   (b) SV and/or TV measured values which do not correspond to the appropriate calculated values, especially when both valve tests measure the same.
   (c) Card area.
   (d) Fluid or gas pounds.
   (e) Abnormal peak or minimum loads.
   (f) Measured counterbalance effect with respect to actual dynamometer card trace.
   (g) Actual order of card which can be used to compare with expected order.
   (h) Sharp changes in loads, such as sticking plunger, well bumping bottom.

INSTANCES WHEN TV AND SV MEASUREMENTS ARE THE SAME

When the traveling and standing valves measure the same, a situation thought process can be used to narrow the possible causes to a minimum number. Even with this minimum number the exact cause may not be apparent, but it will usually fall into one of the two or three major group possibilities. In many cases the course of action will be practically the same, so in essence the cause has been pinpointed.

The procedure for solving the problem of the TV and SV measuring the same can follow several approaches, but the most effective one will usually become apparent if a thought procedure, such as the one presented below, is established. It will be noted that some items are mentioned which are not directly related to the problem of the TV and SV measuring the same but which may have a bearing on the problem in connection with equipment design and operation. A TV-SV problem can sometimes be prevented when other primary problem areas are corrected.

(1) Well Conditions

- (a) Has amount of fluid production materially changed?
- (b) What is the current production compared to that normally experienced?
- (c) Is the well producing top allowable (in prorated areas)?
- (d) Has the GOR increased, or is it high?

(2) Annulus Fluid Level Conditions

- (a) Is there sufficient submergence?
- (b) Is there indication of "foamy" conditions?
- (c) What is the casing pressure? Is it too high?
- (d) Is there an indication of too much submergence?

(3) Surface and Subsurface Equipment Situation

- (a) Is the pump size optimum for the volume of fluid production?
- (b) Has the sucker rod string been optimized?
- (c) Is there a section of heavy rods above the pump?
- (d) Is the sucker rod string compatible with the pump size?
- (e) Have the stroke length and number of strokes per minute been optimized?
- (f) Is the pump setting depth satisfactory?
- (g) Does the well have a gas anchor? If so, has the design been optimized?
- (h) Is the over-all pump efficiency satisfactory?

(4) Dynamometer Card Characteristics

- (a) Is the card shape and appearance the one normally obtained on this well.
- (b) Are there indications of overtravel or undertravel?
- (c) Does the general card configuration correspond to the harmonic card order expected under the prescribed operating conditions?
- (d) Does the card have area?
- (e) Do the TV and SV measurements correspond to the calculated values?
- (f) Are the PPRL and MPRL normal?
- (g) Is the over-all card at the proper location on the building-block load diagram?
- (h) Is there a fluid or gas pound present?
- (i) Is there an indication of sufficient submergence?
- (j) Are there any load anomalies or sudden load changes on the card?

OVERTRAVEL AND UNDERTRAVEL

Under normal operating conditions and recommended pumping speeds, forces are acting that will result in both overtravel and undertravel. The degree of each of these depends upon such factors as synchronous or nonsynchronous pumping speeds, too fast or slow pumping speeds, the order of harmonic vibration at which the system is operating, friction such as is experienced in cases of crooked hole, paraffin or scale accumulation, either at the pump or higher in the system, friction caused by sand production, too many sucker rods or an improperly designed sucker rod-subsurface pump relationship. Overtravel and undertravel may be defined as follows:

Overtravel — A condition caused by the acceleration of fluid and/or rods which results in the plunger traveling more than it should or normally would.

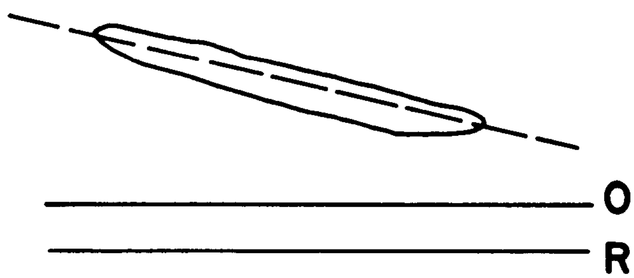


FIG. 35

OVERTRAVEL CARD

Undertravel — A condition caused by some type of a restriction which results in the plunger moving less than it should or normally would.

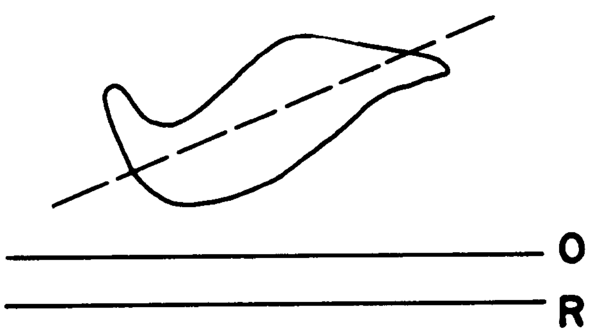


FIG 36

UNDERTRAVEL CARD

By proper design criteria and application, the net plunger travel can be either increased or decreased by the conditions of overtravel or undertravel. Undesirable side effects can result unless careful consideration is given these factors.

IDENTIFYING POSSIBLE CAUSES IN OVERTRAVEL AND UNDERTRAVEL SITUATIONS

When both valve measurements are the same and the card appears normal, or when the card can be classified as an overtravel card, the following check list will be helpful in narrowing the possible cause of the problem. Commencing with examining the dynamometer card, certain possibilities are evaluated. These are checked either "yes" or "no", based only on the appearance of the dynamometer card. As factual data are analyzed, some of the "yes" answers may not be possible; then the previous "yes" becomes "no". After all "yeses" are evaluated, those which still appear as possibilities then become "probable" where previously they had been classified "possible".

In cases where more than one "recommended action" is possible, the approach should be to consider the most economical and practical one first, and if it does not solve the problem, proceed sequentially through the remaining ones on the same basis until a solution or a course of action is reached.

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

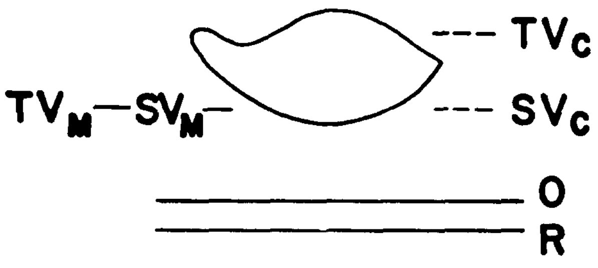
POSSIBLE SITUATIONS	Possible Cause	
	Yes	No
1. Rods parted at pump		
2. Rods parted above pump		
3. Rods unscrewed at pump		
4. Unseated pump		
5. Valve rod failure		
6. Flowing well		
7. TV stuck open		
8. SV stuck open		
9. TV and SV both stuck open		
10. TV bad		
11. SV bad		
12. TV and SV both bad		
13. Pump worn out		
14. Split pump barrel		
15. Gas lock		
16. Tubing leak high		
17. Tubing leak at pump		
18. Pump underdesigned		
19. Rods overdesigned		
20. SPM too high		

Checklist B. For use in analyzing undertravel situations or conditions.

POSSIBLE SITUATIONS	Possible Cause Yes	No
1. Sand problem		
2. Paraffin problem		
3. Scale problem		
4. Too many rods		
5. Rods underdesigned		
6. Pump overdesigned		
7. Too much tubing		
8. Crooked tubing		
9. Crooked hole		
10. Other types of down-hole friction		
11. Low API gravity fluid		
12. Stuck pump		
13. Improper lubrication of down-hole pump		
14. Stuffing box too tight		
15. Tubing not anchored		
16. Rod guides, paraffin scrapers		

The following idealized situations illustrate the thought process involved in connection with the use of Checklists A and B:

Case 1. Normal-appearing card with area, but $TV_M = SV_M$



CASE 1

Facts:

(1) Normal card to possible slight overtravel card.

(2) Area of card normal.

(3) Second order card.

(4) SPM & Depth indicate should expect 2nd order card.

(5) SV & TV weigh the same.

(6) SV_M & $TV_M \approx SV_C$

(7) Well produces less fluid than normal.

(8) Well normally produces X bbls. oil and Y bbls. water per day.

(9) C P ?

(10) GOR ?

(11) Card normal for this well ?

(12) Past problems ?

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

CASE 1

POSSIBLE SITUATIONS	Possible Cause Yes	No
1. Rods parted at pump		X
2. Rods parted above pump		X
3. Rods unscrewed at pump		X
4. Unseated pump		X
5. Valve rod failure		X
6. Flowing well		X
7. TV stuck open		X
8. SV stuck open		X
9. TV and SV both stuck open		X
10. TV bad	X	
11. SV bad		X
12. TV and SV both bad		X
13. Pump worn out	X	
14. Split pump barrel	X	
15. Gas lock		X
16. Tubing leak high	X	
17. Tubing leak at pump	X	
18. Pump underdesigned		X
19. Rods overdesigned		X
20. SPM too high		X

CASE 1

Probable Causes

(1) TV bad
(2) Worn out pump } Valve problem
(3) Tubing leak
(4) Split pump barrel } Tubing leak problem

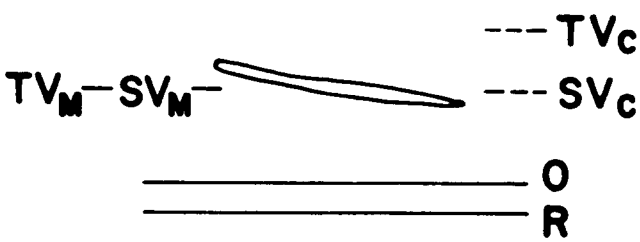
Recommended Action

(1) Test for tubing leak. If there is a leak and the production loss is significant, pull well.

(2) If well is producing allowable, do nothing except make supervisor aware of potential problem.

(3) If well not producing allowable and the production loss is significant, pull and repair pump.

Case 2. Overtravel card with little or no area, and $TV_M = SV_M$



CASE 2

Facts:

(1) Overtravel card.

(2) No significant card area.

(3) Order of card not what is normally anticipated.

(4) SV_M and TV_M weigh the same.

(5) SV_M and $TV_M \approx SV_C$

(6) No fluid production.

(7) Fluid level high indicating excessive pump submergence.

(8) Well normally produced X bbls. oil and Y bbls. water per day.

(9) C P ?

(10) GOR ?

(11) Card representative for this well ?

(12) Past problems ?

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

CASE 2

POSSIBLE SITUATIONS	Possible Cause Yes	No
1. Rods parted at pump	X	
2. Rods parted above pump		X
3. Rods unscrewed at pump	X	
4. Unseated pump	X	
5. Valve rod failure	X	
6. Flowing well	X	
7. TV stuck open	X	
8. SV stuck open		X
9. TV and SV both stuck open	X	
10. TV bad	X	
11. SV bad		X
12. TV and SV both bad	X	
13. Pump worn out	X	
14. Split pump barrel	X	
15. Gas lock		X
16. Tubing leak high		X
17. Tubing leak at pump	X	
18. Pump underdesigned		X
19. Rods overdesigned		X
20. SPM too high		X

Case 2

Probable Causes

(1) Rods parted at pump. } Parted
(2) Rods unscrewed at pump. } rods
(3) Unseated pump.
(4) TV & SV bad. } Valve
(5) TV & SV stuck open. } problem
(6) TV bad.
(7) Worn pump.
(8) Split pump barrel. } Tubing leak
(9) Tubing leak at pump. } problem

Recommended Action

(1) Bump well.

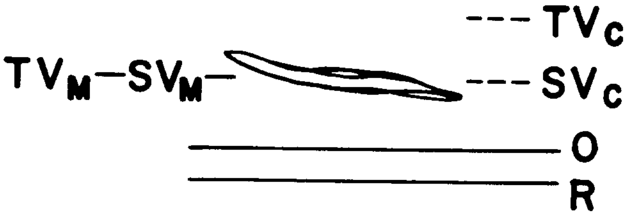
(2) Test for tubing leak. If leak indicated, pull well.

(3) Try to screw rods on pump. If that fails, repair rod string.

(4) Pull and repair pump.

(5) Run dynamometer card after well is restored to production to determine problems if trouble is parted rods.

Case 3. Overtravel card with little or no area, and $TV_M = SV_M$



CASE 3

Facts:

(1) Overtravel card.
(2) No significant card area.
(3) Order of card not what is normally anticipated.
(4) SV_M and TV_M weigh the same.
(5) SV_M and TV_M $\approx$ SV_C
(6) Fluid production normal or slightly above normal.
(7) CP ? TP ?
(8) GOR ?
(9) Card representative for this well ?
(10) Past problems ?

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

CASE 3

POSSIBLE SITUATIONS	Possible Cause Yes	No
1. Rods parted at pump	X	
2. Rods parted above pump		X
3. Rods unscrewed at pump	X	
4. Unseated pump	X	
5. Valve rod failure	X	
6. Flowing well	X	
7. TV stuck open	X	
8. SV stuck open		X
9. TV and SV both stuck open	X	
10. TV bad	X	
11. SV bad		X
12. TV and SV both bad	X	
13. Pump worn out	X	
14. Split pump barrel	X	
15. Gas lock		X
16. Tubing leak high		X
17. Tubing leak at pump	X	
18. Pump underdesigned		X
19. Rods overdesigned		X
20. SPM too high		X

Case 3

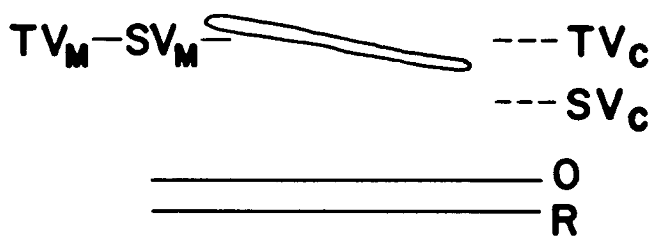
Probable Causes

(1) Flowing well or "flumping" well.

Recommended Action

(1) Install back pressure valve, if well is to continue being pumped.

Case 4. Overtravel card with little or no area, and $TV_M = SV_M$



CASE 4

Facts:

(1) Overtravel card.

(2) No significant card area.

(3) SV and TV weigh the same.

(4) SV_M & TV_M $\approx$ TV_C

(5) Well producing some fluid but not as much as that normally expected.

(6) CP 350 psi.

(7) GOR 6700 ft^3/bbl.

(8) Card normal for this well ?

(9) Type of pump ?

(10) Waste space in pump ?

(11) Proper gas anchor ?

(12) Past problems ?

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

CASE 4

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause No
1. Rods parted at pump		X
2. Rods parted above pump		X
3. Rods unscrewed at pump		X
4. Unseated pump		X
5. Valve rod failure		X
6. Flowing well		X
7. TV stuck open		X
8. SV stuck open	X	
9. TV and SV both stuck open		X
10. TV bad		X
11. SV bad	X	
12. TV and SV both bad		X
13. Pump worn out		X
14. Split pump barrel	X	
15. Gas lock	X	
16. Tubing leak high		X
17. Tubing leak at pump		X
18. Pump underdesigned		X
19. Rods overdesigned		X
20. SPM too high		X

Case 4

Probable Causes

(1) SV stuck open } Valve problem
(2) SV bad }
(3) Well pumped off
(4) Gas lock
(5) Split pump barrel

Recommended Action

(1) Obtain more dynamometer cards and fluid level charts.

(2) Check fluid level for pump submergence.

(3) Check to see if legal obligations and/or field operating requirements will allow CP to be reduced to a minimum.

(4) Check to see if polished rod can be lowered.

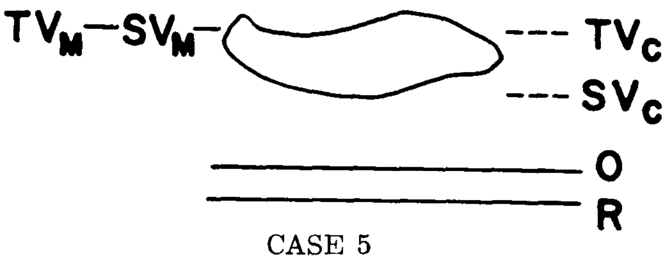
(5) Bump well. If normal card does not appear and well is not bumping bottom, check spacing.

(6) Check possibility of wrong type of pump.

(7) Check for gas anchor and design of gas anchor.

(8) If necessary to pull pump, check for split barrel.

Case 5. Very slight overtravel card with area, but $TV_M = SV_M$



CASE 5

Facts:

(1) Normal card to slight OT card.

(2) Card has area.

(3) Order of card closely approximates that normally anticipated.

(4) TV_M & SV_M weigh the same.

(5) TV_M & $SV_M \approx TV_C$

(6) Well not producing as much fluid as it should.

(7) Well normally produces X bbls. oil and Y bbls. water per day.

(8) Fluid level indicates excess submergence.

(9) CP ?

(10) GOR ?

(11) Card normal for this well ?

(12) Past problems ?

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

CASE 5

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause No
1. Rods parted at pump		X
2. Rods parted above pump		X
3. Rods unscrewed at pump		X
4. Unseated pump		X
5. Valve rod failure		X
6. Flowing well		X
7. TV stuck open		X
8. SV stuck open		X
9. TV and SV both stuck open		X
10. TV bad		X
11. SV bad	X	
12. TV and SV both bad		X
13. Pump worn out		X
14. Split pump barrel	X	
15. Gas lock		X
16. Tubing leak high		X
17. Tubing leak at pump		X
18. Pump underdesigned		X
19. Rods overdesigned		X
20. SPM too high		X

Case 5

Probable Causes

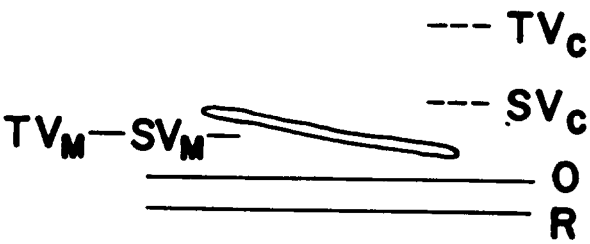
(1) Bad standing valve.

(2) Split pump barrel.

Recommended Action

(1) If well is not capable of producing desired volume with present pump, pull and repair pump.

Case 6. Overtravel card with little or no card area, and $TV_M = SV_M$



CASE 6

Facts:

(1) Overtravel card.

(2) No significant card area.

(3) Order of card not that normally anticipated.

(4) SV_M and TV_M weigh the same.

(5) SV_M and $TV_M < SV_C$

(6) No fluid production.

(7) FL indicates sufficient pump submergence.

(8) Normal production X bbls. oil and Y bbls. water per day.

(9) CP ?

(10) GOR ?

(11) Card representative for this well ?

(12) Past problems ?

Checklist A. Normal cards or overtravel cards with SV and TV measuring the same.

CASE 6

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause No
1. Rods parted at pump		X
2. Rods parted above pump	X	
3. Rods unscrewed at pump		X
4. Unseated pump		X
5. Valve rod failure		X
6. Flowing well		X
7. TV stuck open		X
8. SV stuck open		X
9. TV and SV both stuck open		X
10. TV bad		X
11. SV bad		X
12. TV and SV both bad		X
13. Pump worn out		X
14. Split pump barrel		X
15. Gas lock		X
16. Tubing leak high		X
17. Tubing leak at pump		X
18. Pump underdesigned		X
19. Rods overdesigned		X
20. SPM too high		X

CASE 6

Probable Causes

(1) Parted rods above pump.

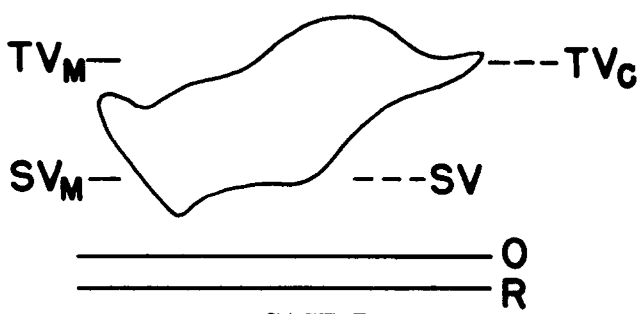
Recommended Action

(1) Pull and repair rods.

(2) Run dynamometer card in an attempt to determine rod problem.

(3) If parted rods failure has occurred frequently, study to determine possible cause.

Case 7. Undertravel card with card area.



CASE 7

Facts:

(1) Undertravel card.

(2) Card has area.

(3) $TV_M \approx TV_C$, and $SV_M \approx SV_C$

(4) Production not as much as it should be.

(5) Well normally produces X bbls. oil and Y bbls. water per day.

(6) Order of card normal ?

(7) Have experienced frequent rod breaks.

Checklist B. For use in analyzing undertravel situations or conditions.

CASE 7

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause No
1. Sand problem		X
2. Paraffin problem		X
3. Scale problem		X
4. Too many rods		X
5. Rods underdesigned	X	
6. Pump overdesigned	X	
7. Too much tubing		X
8. Crooked tubing		X
9. Crooked hole		X
10. Other types of down-hole friction		X
11. Low API gravity fluid	X	
12. Stuck pump		X
13. Improper lubrication of down-hole pump		X
14. Stuffing box too tight		X
15. Tubing not anchored		X
16. Rod guides, paraffin scrapers		X

CASE 7

Probable Causes

(1) Rods underdesigned.

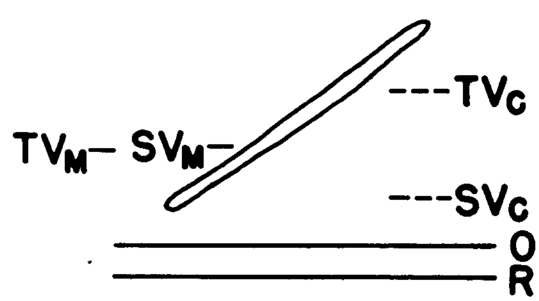
(2) Pump overdesigned.

Recommended Action

(1) Pull well and redesign sucker rod-pump relationship.

(2) Continue surveillance over well to determine if rod parting problem has been corrected if the same rods are used. The chances are that all of the present rod string is damaged.

Case 8. Undertravel card with small or slight area, and $TV_M \approx SV_M$



CASE 8

Facts:

(1) UT card.
(2) No significant card area.
(3) TV_M and SV_M are approximately equal but fall between the calculated TV and SV values. (This is coincidental.)
(4) No fluid production.
(5) Well has sufficient submergence to produce.
(6) Well normally produces X bbls. oil and Y bbls. water per day.
(7) Card normal for this well ?
(8) Past problems ?

Checklist B. For use in analyzing undertravel situations or conditions.

CASE 8

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause No
1. Sand problem	X	
2. Paraffin problem	X	
3. Scale problem	X	
4. Too many rods		X
5. Rods underdesigned		X
6. Pump overdesigned		X
7. Too much tubing		X
8. Crooked tubing		X
9. Crooked hole		X
10. Other types of down-hole friction		X
11. Low API gravity fluid		X
12. Stuck pump	X	
13. Improper lubrication of down-hole pump		X
14. Stuffing box too tight		X
15. Tubing not anchored		X
16. Rod guides, paraffin scrapers		X

CASE 8

Probable Causes

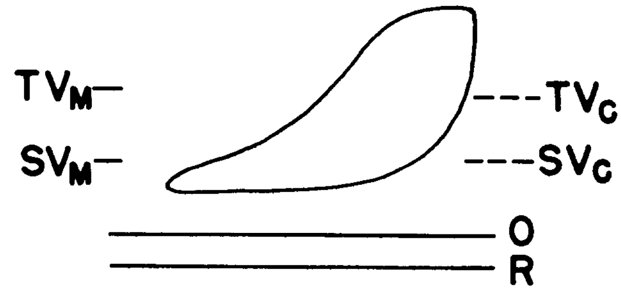
(1) Sand problem
(2) Paraffin problem
(3) Scale problem
(4) Stuck pump

stuck pump

Recommended Action

(1) Re-check TV and SV tests.
(2) Check for stuck pump.

Case 9. Undertravel card with card area.



CASE 9

Facts:

(1) Undertravel card.
(2) Card has area.
(3) $SV_M \not\approx SV_C$, and $TV_M \not\approx TV_C$
(4) Card not normal for this well.
(5) Well not producing as much as normal.
(6) Well normally produces X bbls. oil and Y bbls. water per day.

Checklist B. For use in analyzing undertravel situations or conditions.

CASE 9

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause No
1. Sand problem		X
2. Paraffin problem	X	
3. Scale problem		X
4. Too many rods		X
5. Rods underdesigned		X
6. Pump overdesigned		X
7. Too much tubing	X	
8. Crooked tubing	X	
9. Crooked hole	X	
10. Other types of down-hole friction	X	
11. Low API gravity fluid		X
12. Stuck pump		X
13. Improper lubrication of down-hole pump		X
14. Stuffing box too tight		X
15. Tubing not anchored		X
16. Rod guides, paraffin scrapers	X	

CASE 9

Probable Causes

(1) Too much tubing.

Recommended Action

(1) Remove excess tubing.

XVIII.

CONCLUSION

There is fertile ground for increasing production, reducing costs and increasing efficiency when personnel directly related to the selection and operation of sucker rod pumping equipment understand the principles involved. API RP 11L now makes it possible to pre-calculate accurately the loads critical to equipment selection and dynamometer card interpretation. If this publication and other associated principles of well pumping are presented in an understandable manner, and are understood by field operating personnel and design engineers, the end result will be much more efficient well pumping operations and increased profits. Personnel in whose field of responsibility this type of artificial lift lies can be up-graded and will take a more intelligent interest in their work.

XIX.

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