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-six sessions of three types of well pumping short courses have been presented since November, 1961, to a total of 451 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 that our approach is on the right track: That fundamentals must be presented and these fundamentals must be presented in a stepwise 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 as listed in the Table of Contents, Subjects III through XVII. 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.

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I

INTRODUCTION

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

The well pumping knowledge of the participant has been increased by this formal training and by post-course application. This has made it possible, and technological advances have made it necessary, to continually upgrade the material presented. For example, the last three 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 but is only a portion of, the material presented at the most recent sessions. The material presented includes only Subjects III through XVII in the Table of Contents. Material covered at the most recent sessions included, but was not limited to, the subjects listed in "Contents of Course and Teaching Techniques". The importance of the teaching techniques used before and during the sessions cannot be overemphasized and will be thoroughly covered.

II

CONTENTS OF COURSE AND TEACHING TECHNIQUES

In general, the courses cover the following 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. Productivity indices as used to determine pump setting depth.
16. Gas Anchor design.
17. Systematic approach to solving well pumping problems.

The course teaching method is based on conferee participation. Pre-session homework is assigned. The API Divisions of Production PROFIT Series "Well Pumping" is furnished each participant approximately one month prior to the session. 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 that time. Approximately two weeks later, additional reference material and a 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 much 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 discussion are used as supplementary methods. The Provencial 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.

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 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 INTERPRETATION

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 used 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 to 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, 62.5 pounds divided by the weight of a cubic foot of steel, 490 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 = $0.434 \times 3.1416/4$.

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.

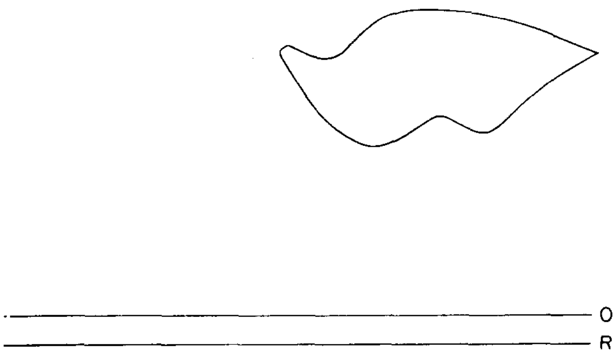
MPRL = The minimum load at the polished rod, in pounds, during the pumping cycle.

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.

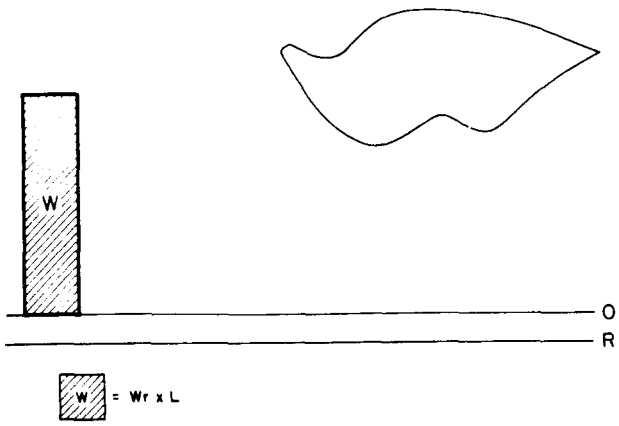


Zero Load

FIGURE 1

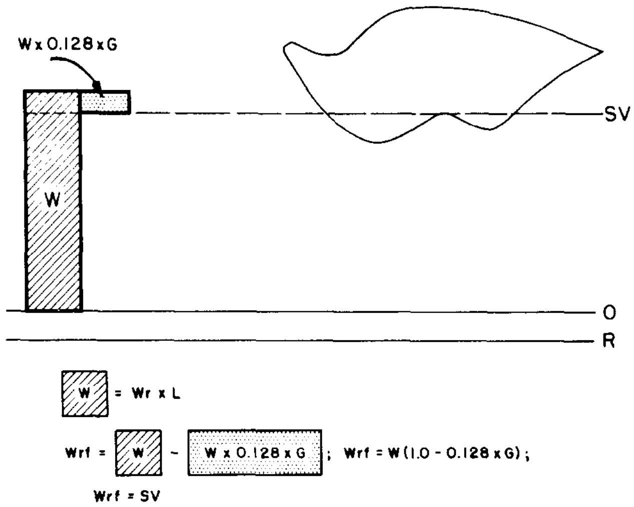
STANDING VALVE LOAD

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).



Weight of Sucker Rods in Air

FIGURE 2

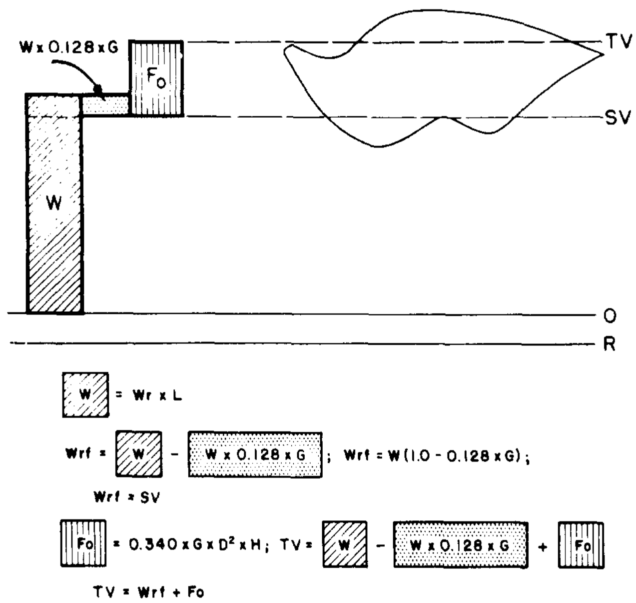


Standing Valve Load

FIGURE 3

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.

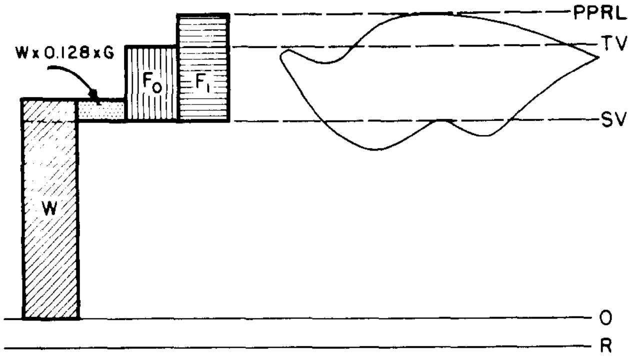


Traveling Valve Load

FIGURE 4

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.



$\boxed{W} = W_r \times L$

$W_{rf} = \boxed{W} - \boxed{W \times 0.128 \times G}$; $W_{rf} = W(1.0 - 0.128 \times G)$;

$W_{rf} = SV$

$\boxed{F_o} = 0.340 \times G \times D^2 \times H$; $TV = \boxed{W} - \boxed{W \times 0.128 \times G} + \boxed{F_o}$;

$TV = W_{rf} + F_o$

$\boxed{F_1} = \frac{F_1}{SK_r} \times SK_r$; $PPRL = \boxed{W} - \boxed{W \times 0.128 \times G} + \boxed{F_1}$;

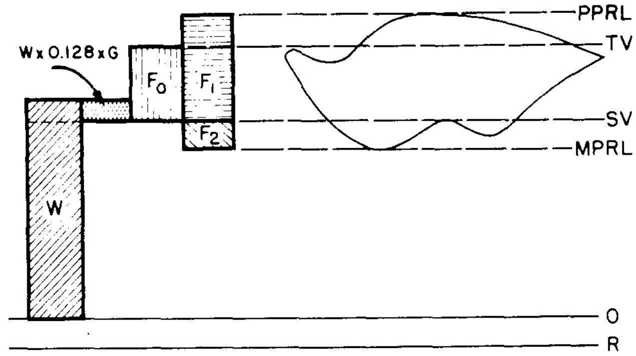
$PPRL = W_{rf} + F_1$

Peak Polished Rod Load

FIGURE 5

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 the dynamic effects (F_2) on the downstroke.



$\boxed{W} = W_r \times L$

$W_{rf} = \boxed{W} - \boxed{W \times 0.128 \times G}$; $W_{rf} = W(1.0 - 0.128 \times G)$;

$W_{rf} = SV$

$\boxed{F_o} = 0.340 \times G \times D^2 \times H$, $TV = \boxed{W} - \boxed{W \times 0.128 \times G} + \boxed{F_o}$;

$TV = W_{rf} + F_o$

$\boxed{F_1} = \frac{F_1}{SK_r} \times SK_r$; $PPRL = \boxed{W} - \boxed{W \times 0.128 \times G} + \boxed{F_1}$;

$PPRL = W_{rf} + F_1$

$\boxed{F_2} = \frac{F_2}{SK_r} \times SK_r$; $MPRL = \boxed{W} - \boxed{W \times 0.128 \times G} - \boxed{F_2}$;

$MPRL = W_{rf} - F_2$

Minimum Polished Rod Load

FIGURE 6

COUNTERBALANCE EFFECT

The counterbalance effect (CBE) is measured under static conditions for convenience but is applied under dynamic conditions. The counterbalance effect normally required is approximately equal to the weight of the sucker rods in air (W), minus one-half the buoyancy effect (W X 0.128 X G) plus one-half of the weight of the well fluid on the gross plunger area (F_O)

$$CBE = W - (W \times 0.128 \times G) + \frac{W \times 0.128 \times G}{2} + \frac{F_o}{2}$$

$$CBE = W_{rf} = 0.06\ W_{rf} + \frac{F_o}{2}$$ (Note: 0.064 is rounded to 0.06.)

$$CBE = 1.06\ W_{rf} + \frac{1}{2} F_o$$

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.

DESIGN CALCULATIONS SHEET
CONVENTIONAL SUCKER ROD PUMPING SYSTEM

Well____________________ Date____________________

Known or Assumed Data:

Fluid Level, H = ________________ ft. Pump Depth, L = ________________ ft.
Tubing Size______ in. Is it anchored? Yes______ No______ Pumping Speed, N = ______ SPM
Length of Stroke, S = ____________ in. Plunger Diameter, D = ____________ in.
Specific Gravity of Fluid, G = ____________ Sucker Rods____________
API Grade: C, 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)

Calculate Non-Dimensional Variables:

5. $F_o = .340 \times G \times D^2 \times H = .340 \times$ ______ x ______ x ______ = ______ lbs. (Gross Plunger Load)
6. $1/K_r = E_r \times L$ = ______ x ______ = ______ in/lb (line 2 x L)
7. $SK_r = S \div 1/kr$ = ______ ÷ ______ = ______ lbs. (S/line 6)
8. F_o/SK_r = ______ ÷ ______ = ______ (line 5/line 7)
9. $N/N_o = NL \div 245{,}000$ = ______ x ______ ÷ 245,000 = ______
10. $N/N_o' = N/N_o \div F_c$ = ______ ÷ ______ = ______ (line 9/line 3)
11. $1/K_t = E_t \times L$ = ______ 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. $S_p = [(S_p/S) \times S] - [F_o \times 1/K_t]$ = [______ (line 12) x ______ (S)] - [______ (line 5) x ______ (line 11)] = ______ in.
14. $PD - 0.1166 \times S_p \times N \times D^2 = 0.1166 \times$ ______ (line 13) x ______ (N) x ______ (D^2) = ______ bbls per day.

Determine Non-Dimensional Parameters:

15. $W = W_r \times L$ = ______ 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. $F3/SK_r$ = ____________ (Figure 6) (line 9 to line 8 to answer)
22. T_a = ____________ (Figure 7) (line 10 and line 8 intersection is %)
($T_a = 1.00 + (\%/100) \times [(W_{rf}/SK_r) - 0.3] /0.1 = 1.00 \pm$ ______

Solve for Operating Characteristics:

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

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FIGURE 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 (W_f)		xxxxxxxxxxx
S.V. + W_f (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:______________________________

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FIGURE 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 x 10^{-6}	1.000	....	....	....	....	....	100.0
54	1.06	0.892	1.697 x 10^{-6}	1.128	....	....	....	....	40.5	59.5
54	1.25	0.914	1.659 x 10^{-6}	1.139	....	....	....	....	45.9	54.1
54	1.50	0.948	1.597 x 10^{-6}	1.142	....	....	....	....	54.5	45.5
54	1.75	0.990	1.525 x 10^{-6}	1.130	....	....	....	....	64.6	35.4
54	2.00	1.037	1.442 x 10^{-6}	1.095	....	....	....	....	76.2	23.8
55	All	1.135	1.270 x 10^{-6}	1.000	....	....	....	....	100.0	....
64	1.06	1.116	1.441 x 10^{-6}	1.224	....	....	....	28.1	33.1	38.8
64	1.25	1.168	1.368 x 10^{-6}	1.222	....	....	....	31.8	37.5	30.7
64	1.50	1.250	1.252 x 10^{-6}	1.191	....	....	....	37.7	44.5	17.8
64	1.75	1.347	1.116 x 10^{-6}	1.137	....	....	....	44.7	52.7	2.6
65	1.06	1.291	1.150 x 10^{-6}	1.085	....	....	....	31.3	68.7	....
65	1.25	1.306	1.138 x 10^{-6}	1.093	....	....	....	34.4	65.6	....
65	1.50	1.330	1.119 x 10^{-6}	1.103	....	....	....	39.2	60.8	....
65	1.75	1.359	1.097 x 10^{-6}	1.111	....	....	....	45.0	55.0	....
65	2.00	1.392	1.071 x 10^{-6}	1.114	....	....	....	51.6	48.4	....
65	2.25	1.429	1.042 x 10^{-6}	1.110	....	....	....	59.0	41.0	....
65	2.50	1.471	1.010 x 10^{-6}	1.097	....	....	....	67.4	32.6	....
65	2.75	1.517	0.974 x 10^{-6}	1.074	....	....	....	76.6	23.4	....
66	All	1.634	0.883 x 10^{-6}	1.000	....	....	....	100.0	....	....
75	1.06	1.511	1.030 x 10^{-6}	1.168	....	....	22.6	26.1	51.3	....
75	1.25	1.548	1.006 x 10^{-6}	1.179	....	....	24.8	28.6	46.6	....
75	1.50	1.606	0.969 x 10^{-6}	1.185	....	....	28.3	32.6	39.1	....
75	1.75	1.674	0.924 x 10^{-6}	1.180	....	....	32.4	37.4	30.2	....
75	2.00	1.754	0.874 x 10^{-6}	1.160	....	....	37.2	42.8	20.0	....
75	2.25	1.843	0.816 x 10^{-6}	1.128	....	....	42.5	49.2	8.3	....
76	1.06	1.787	0.822 x 10^{-6}	1.061	....	....	25.9	74.1	....	....
76	1.25	1.798	0.818 x 10^{-6}	1.066	....	....	27.8	72.2	....	....
76	1.50	1.816	0.811 x 10^{-6}	1.073	....	....	30.9	69.1	....	....
76	1.75	1.836	0.803 x 10^{-6}	1.080	....	....	34.3	65.7	....	....
76	2.00	1.861	0.793 x 10^{-6}	1.087	....	....	38.5	61.5	....	....
76	2.25	1.888	0.782 x 10^{-6}	1.094	....	....	43.1	56.9	....	....
76	2.50	1.919	0.770 x 10^{-6}	1.096	....	....	48.3	51.7	....	....
76	2.75	1.953	0.756 x 10^{-6}	1.096	....	....	54.1	45.9	....	....
76	3.75	2.121	0.690 x 10^{-6}	1.043	....	....	82.5	17.5	....	....
77	All	2.224	0.649 x 10^{-6}	1.000	....	....	100.0	....	....	....
85	1.06	1.709	0.957 x 10^{-6}	1.237	....	15.9	17.7	20.1	46.3	....
85	1.25	1.780	0.919 x 10^{-6}	1.250	....	17.9	19.9	22.5	39.7	....
85	1.50	1.893	0.858 x 10^{-6}	1.242	....	21.0	23.4	26.5	29.1	....
85	1.75	2.027	0.786 x 10^{-6}	1.218	....	24.8	27.5	31.0	16.7	....
85	2.00	2.181	0.703 x 10^{-6}	1.180	....	29.0	32.3	36.3	2.4	....
86	1.06	2.008	0.757 x 10^{-6}	1.127	....	19.3	21.9	58.8	....	....
86	1.25	2.035	0.748 x 10^{-6}	1.136	....	20.7	23.5	55.8	....	....
86	1.50	2.079	0.733 x 10^{-6}	1.148	....	23.0	26.0	51.0	....	....
86	1.75	2.130	0.716 x 10^{-6}	1.157	....	25.6	29.0	45.4	....	....
86	2.00	2.190	0.696 x 10^{-6}	1.162	....	28.7	32.5	38.8	....	....
86	2.25	2.257	0.674 x 10^{-6}	1.158	....	32.1	36.5	31.4	....	....
86	2.50	2.334	0.650 x 10^{-6}	1.146	....	35.8	41.6	22.6	....	....
86	2.75	2.415	0.621 x 10^{-6}	1.125	....	40.3	45.6	14.1	....	....

TABLE I

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. Fig. 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.

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 con-

(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					
					1⅛	1	⅞	¾	⅝	½
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)

stant, 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).

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

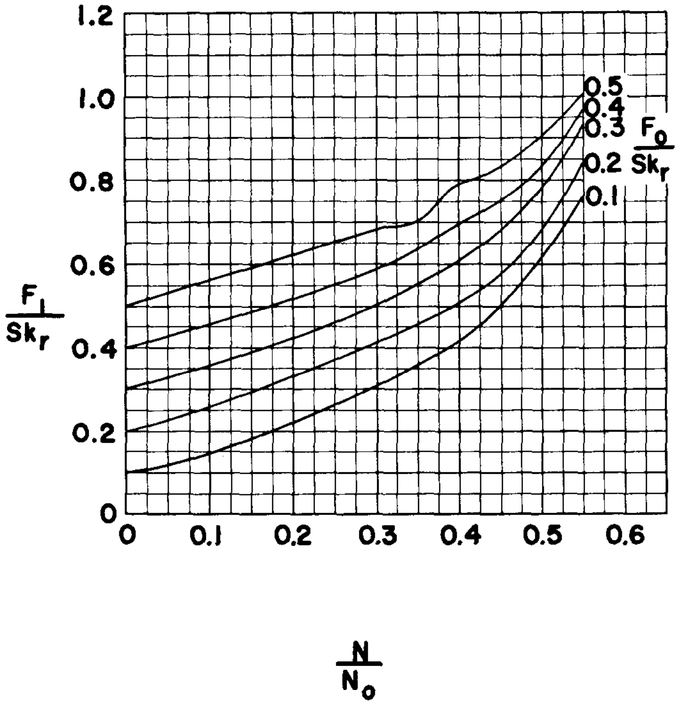
SK_r = $S/(1/K_r)$ = Pounds of load (static) necessary to stretch the total rod string an amount equal to the polished rod stroke, S.

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.

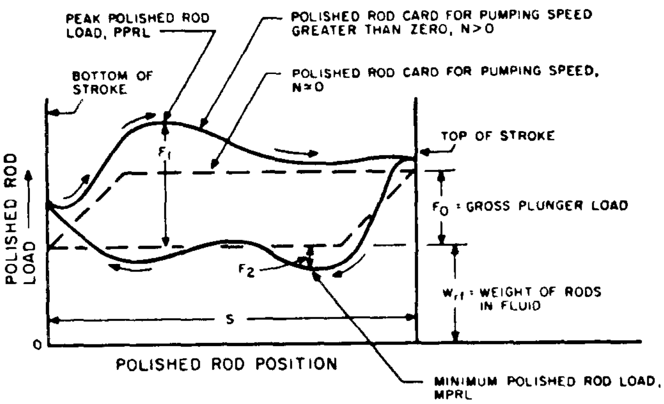
F_c = Frequency factor; F_c = 1.00 for a straight string but is greater than 1.00 for a tapered string of equal length, since the natural frequency of tapered strings is greater than the natural frequency of the same length straight string, Table 1, (API RP 11L, Table 1, Column 5).



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

(API RP 11L, Fig. 3)

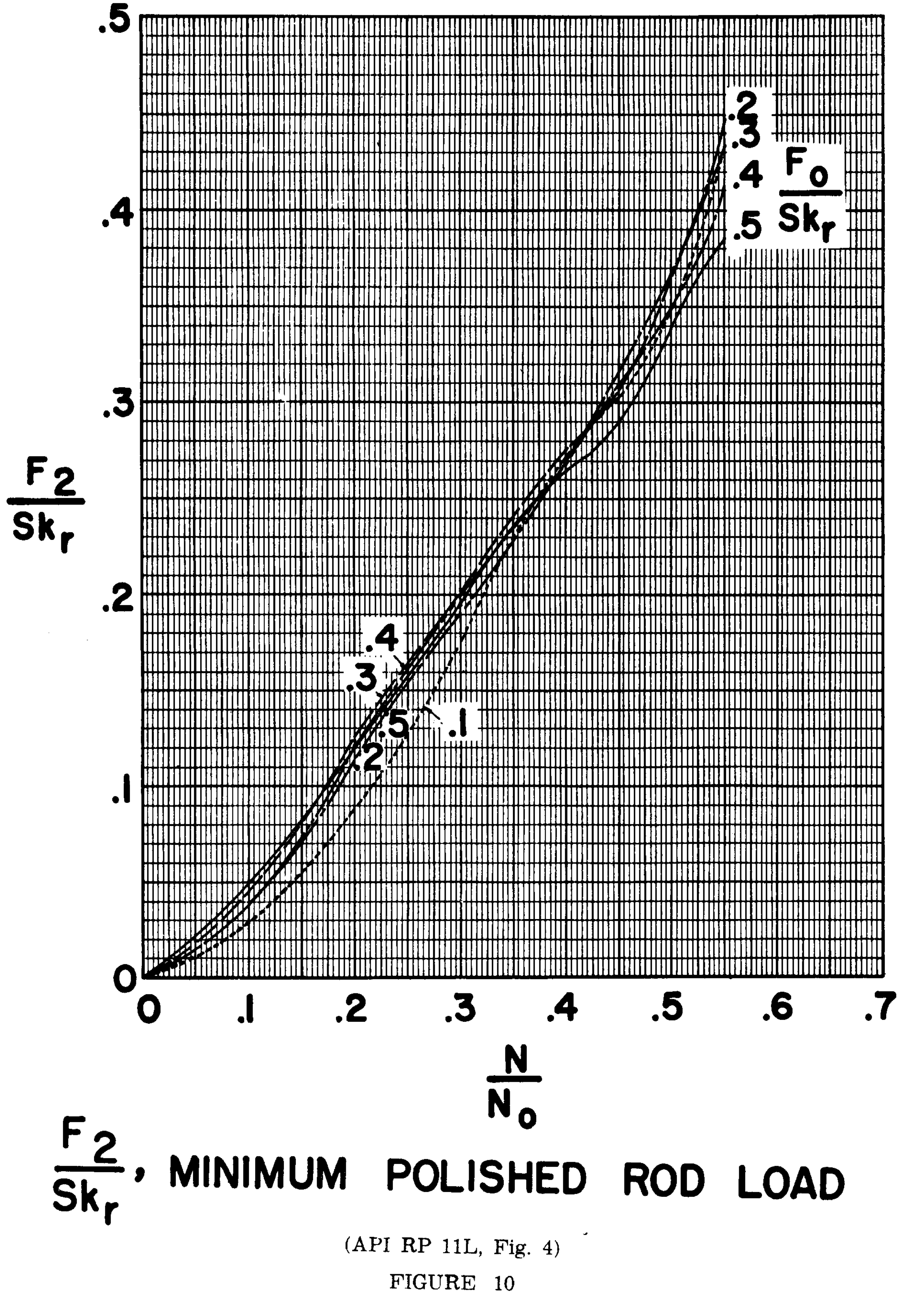
FIGURE 9



BASIC DYNAGRAPH CARD

(API RP 11L, Fig. 1)

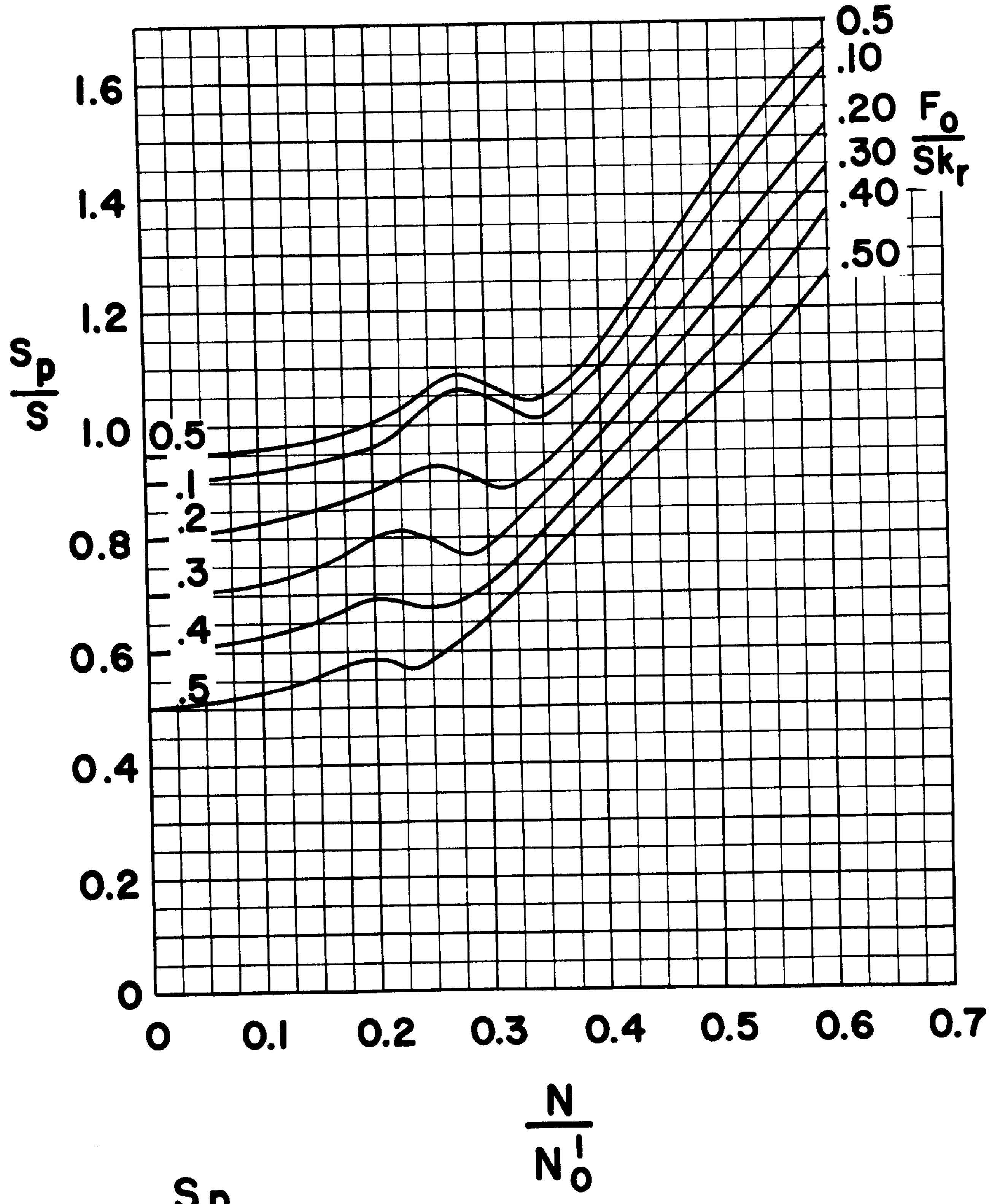
FIGURE 8



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

(API RP 11L, Fig. 4)

FIGURE 10



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

(API RP 11L, Fig. 2)

FIGURE 11

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

$\frac{N}{N_o'}$ = NL ÷ 245,000

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

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

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

$\frac{F_2}{SK_r}$ 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.

$\frac{F_o}{SK_r}$ = 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.

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}

(API TABLE 2)

TABLE II

$\frac{1}{K_t} = E_t \times L_{ua}$ = 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.

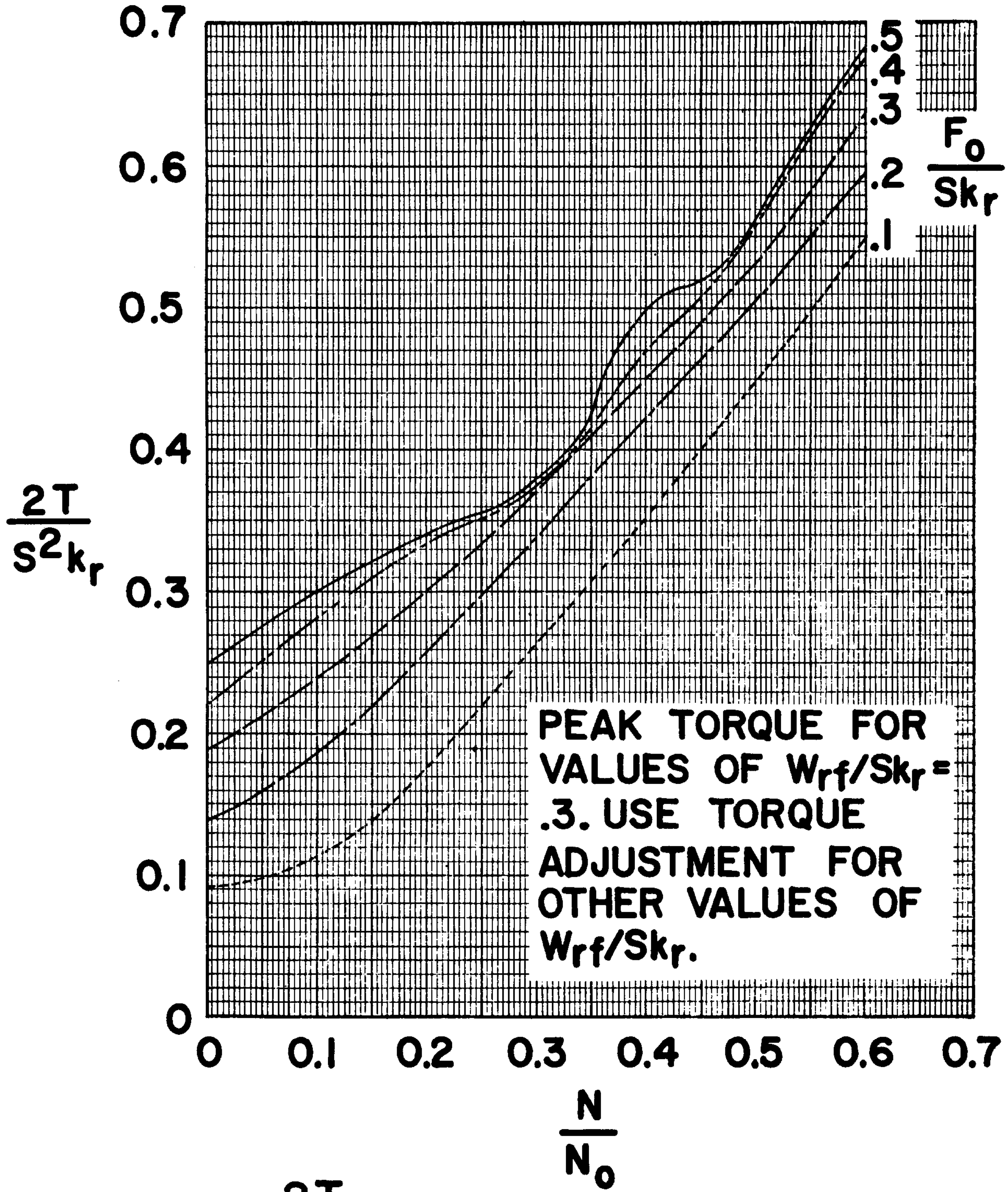
$\frac{S_p}{S}$ is a function of $\frac{N}{N_o'}$ and $\frac{F_o}{SK_r}$

$S_p = \frac{S_p}{S} \times S$ (if tubing is anchored at, or very near, the standing valve).

$S_p = \frac{S_p}{S} \times S - (F_o \times \frac{1}{K_t})$ (if tubing is not anchored or if anchor is far from standing valve)

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

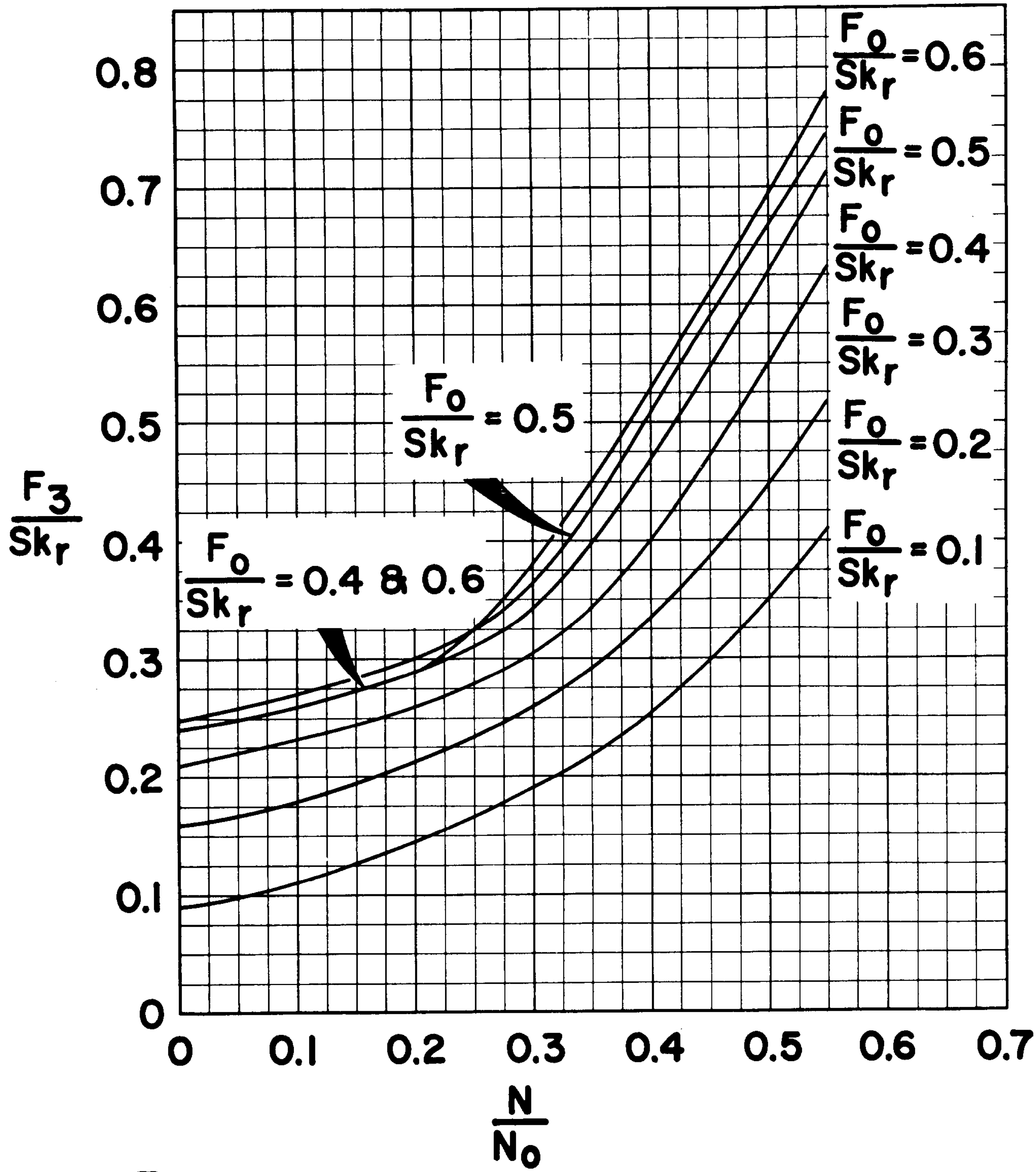
$PD \doteq 0.1166 \times S_p \times N \times D^2$



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

(API RP 11L, Fig. 5)

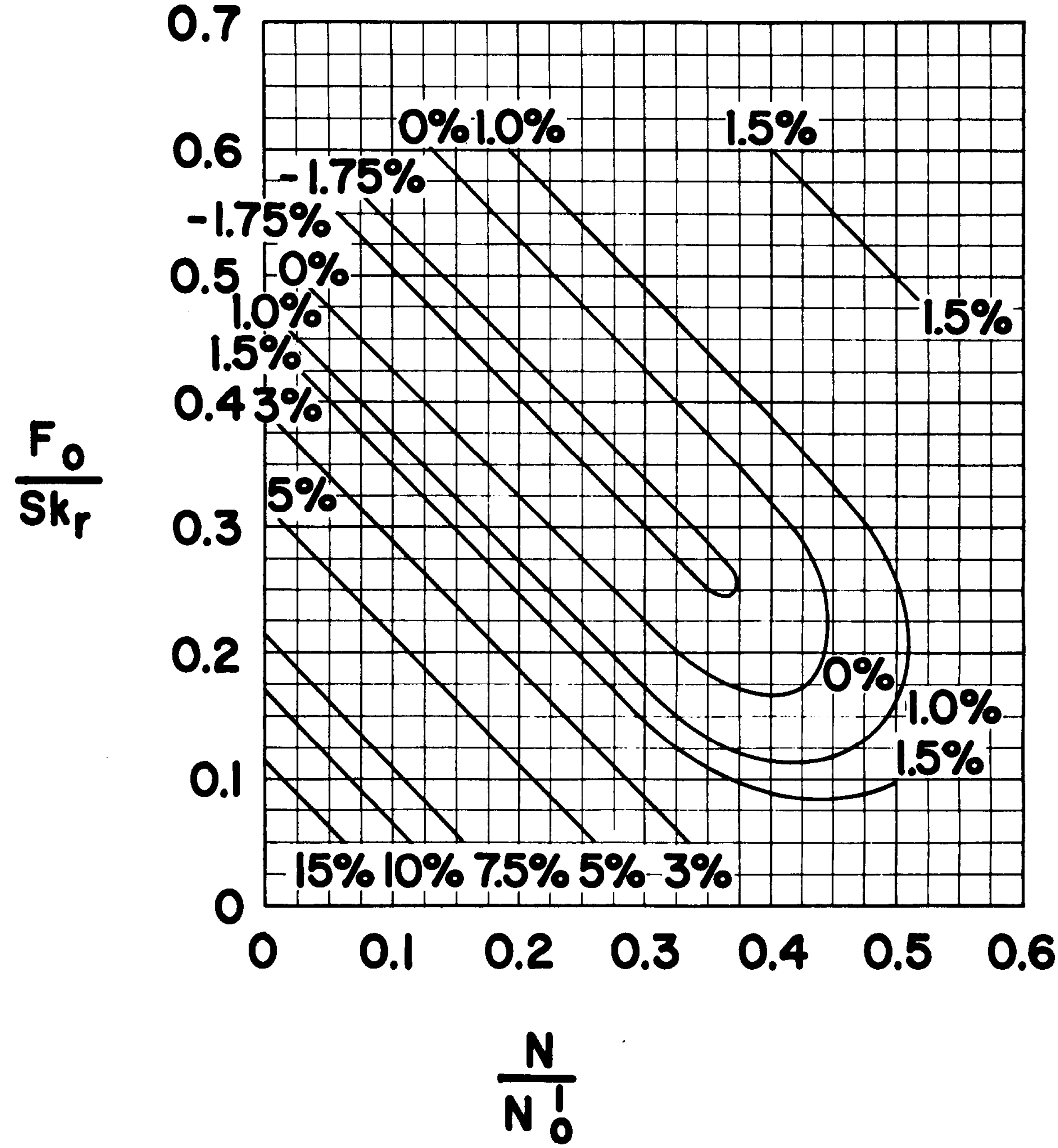
FIGURE 12



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

(API RP 11L, Fig. 6)

FIGURE 13



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

(API RP 11L, Fig. 7)

FIGURE 14

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 less than 0.3, torque must be adjusted downward (-), and if $\frac{W_{rf}}{SK_r}$ is greater than 0.3, torque must be adjusted upward (+).

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

$$T_a = 1.00 + (\%\text{ indicated on Fig. 14} \div 100) \times \left[\left(\frac{W_{rf}}{SK_r} - 0.3\right) \div 0.1 \right]$$

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

V

DETERMINING THE POLISHED ROD HORSEPOWER

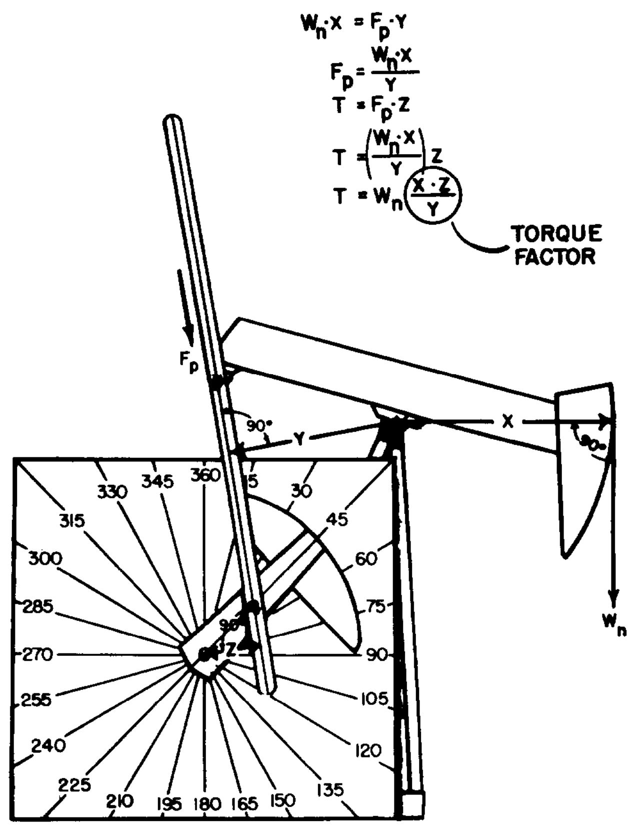
It is possible to determine the polished rod horsepower (PRHP) by using the method recommended by API RP 11L, which is as follows:

$\frac{F_3}{SK_r}$ 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}.

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



(After WORLD OIL, March 1965)

FIGURE 15

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.

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.

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

$\frac{X \times Z}{Y}$ 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 INSTANTANEOUS LOADS FROM DYNAMOMETER CARD

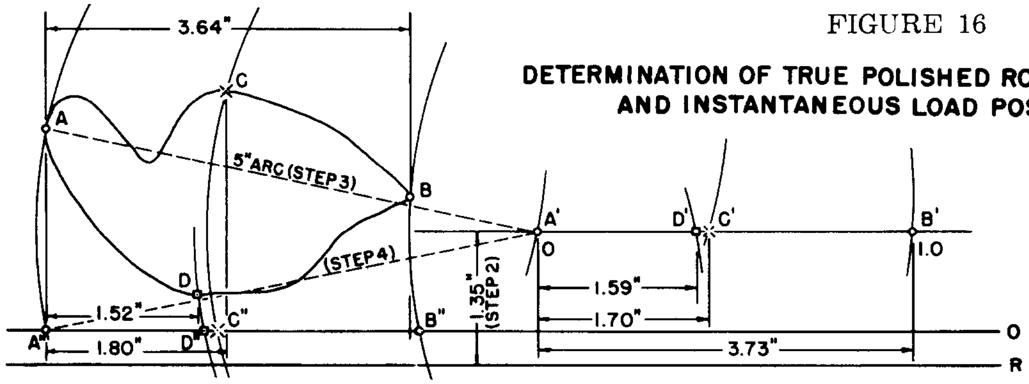
It is of definite advantage, and in most cases a necessity, to fix accurately the exact position of the polished rod with respect to instantaneous well loads. The impact and truth of this statement will be reinforced in a corresponding presentation under net torque determination presented later in the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

FIGURE 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 cards 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 differences 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 calculations illustrate 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

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

Per cent error = $\left(\frac{63.3 - 64.9}{64.9}\right)$ x 100 = $\frac{-160}{64.9}$ = -2.47 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

Per cent error = $\left(\frac{0.495 - 0.456}{0.456}\right)$ x 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 minimum load, using horizontal measurements = 1.52/3.64 - 0.418

Per cent error = $\left(\frac{0.426 - 0.418}{0.426}\right)$ x 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 dynamometer 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 the data furnished by the manufacturer does not indicate the crank position at the start and end of the stroke, or does not have torque factors marked plus or minus, plot a curve on linear graph paper of the "torque factor" on the abscissas (X-Axis) versus "crank angle" on the 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°. The upstroke will, in general, not start at 0°, nor will it end at 180°. If the crank rotates counterclockwise, the upstroke will start between 15° and 345° (either side of 0°) and end between 195° and 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 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 that 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 angle, 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 at 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 peak torque during the upstroke will equal the 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) TF at Θ_1 + (PRL at Θ_2 − SU) TF at Θ_2.

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

$$\frac{(\text{PRL at } \Theta_1 - \text{SU})\text{TF at } \Theta_1 + (\text{PRL at } \Theta_2 - \text{SU})\text{TF at } \Theta_2}{\text{TF at } 90°(\text{Sin } \Theta_1 + \text{Sin } \Theta_2)}$$

$+ \text{SU}$

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

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°

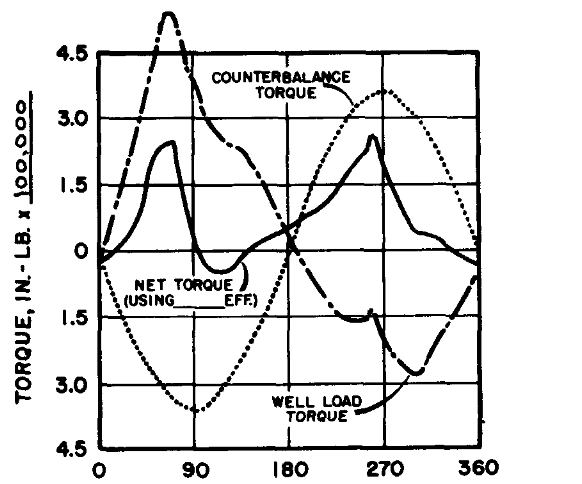
$$\text{P.R.H.P.} = \frac{\text{Area Card X D.C. X S.L. X SPM}}{\text{Card Length X 12 X 33,000}} = \frac{2.335 \times 8150 \times 86 \times 12.5}{4.73 \times 12 \times 33{,}000} = 10.93$$

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	.83		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.



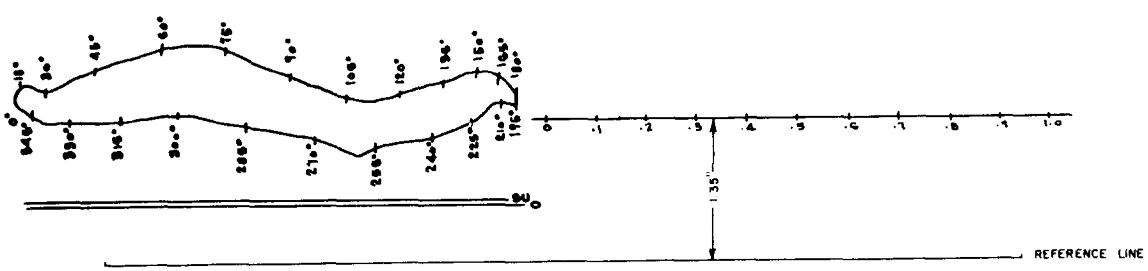


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FIGURE 17

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. There are reasons why it is necessary to have larger pumps than needed in some cases. 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 was 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 the basic subsurface pumps. The pump companies have publications which relate to the types of pumps for different well environmental 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 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.

Depth, Feet	FLUID PRODUCTION—Barrels Per Day—80 Percent Efficiency									
	100	200	300	400	500	600	700	800	900	1000
2000	1½ 1¼	1¾ 1½	2 1¾	2¼ 2	2½ 2¼	2¾ 2½	2¾	2¾	2¾	2¾
3000	1½ 1¼	1¾ 1½	2 1¾	2¼ 2	2½ 2¼	2½ 2¼	2¾ 2½	2¾	2¾	2¾
4000	1¼	1¾ 1½	2 1¾	2¼ 2	2¼ 2	2¼	2¼	2¼		
5000	1¼	1¾ 1½	2 1¾	2 1¾	2¼ 2	2¼				
6000	1¼	1½ 1¼	1¾ 1½	1¾						
7000	1¼ 1⅛	1½ 1¼								
8000	1¼ 1⅛									

In this tabulation surface pumping strokes up to 74 inch only are considered.

(Pump Plunger Sizes Recommended for Optimum Conditions)

TABLE 3

Courtesy Bethlehem Steel Company Sucker Rod Handbook, 1958 and WORLD OIL Magazine, Dec. 1957

The critical part of pump selection involves the determination of well capacity. By making certain assumptions, an approximation of the well fluid capacity can be made, both mathematically and graphically. The method presented does not conform to the engineering principles normally associated with the use of productivity indices. In the case of most pumping wells, the modifications made in the use of the productivity index data will not materially affect the calculated (or forecast) well capacities unless the greatest measured producing rate is small compared to the calculated capacity. The method used makes it necessary to alter the accepted definition of a PI to conform with the following assumptions:

(1) The specific gravity of the produced well fluid will remain the same from the measured producing rate during the PI test to the calculated capacity of the well.

(2) The oil formation volume factor is assumed to be approximately 1.0 and is assumed to be constant from the greatest measured rate to the calculated capacity. Therefore, bottomhole pressure versus well fluid production can be plotted on Cartesian coordinate graph paper.

(3) The fluid components considered in calculating the PI will be the stock tank barrels of oil and water produced, extrapolated to barrels of fluid per day.

(4) The amount of fluid in the reservoir is large compared to the well capacity.

(5) The pressure on the casing-tubing annulus, the gravity of the gas in the annulus, and the fluid level in the annulus can be measured.

(6) The gravity of the oil and water in the casing-tubing annulus can be determined, and the location of the pump intake and producing zone, relative to the casing-tubing annulus valve, is known.

ESTIMATING WELL CAPACITY FROM PRODUCTION TESTS

1. Assume that the producing capacity of the test pumping equipment is constant (only one producing rate can be economically secured).

(a) Determine the static (shut-in) reservoir pressure (P_{SI}) in the vicinity of the well. Note that the producing rate is zero BFPD.

(b) Pump the well until the producing rate (BFPD) and the bottomhole pressure (P_{STAB}) stabilize.

(c) Determine the stabilized producing rate (BFPD) and the stabilized pressure (P_{STAB}).

(d) Calculate the minimum bottomhole pressure (P_{MIN}) which will exist if the well is produced to capacity. This requires that the following be estimated and combined:

(1) the pressure, in psi held on the casing-tubing annulus

(2) the pressure due to the gas column between the casing pressure gauge and the fluid level in the annulus

(3) the pressure due to the fluid head needed to load the pump properly.

(e) Pumping PI = (BFPD − 0)/$P_{SI} - P_{STAB}$) = BFPD psi decrease in BHP.

(f) Well capacity = ($P_{SI} - P_{MIN}$) X PI.

Example 1

1. Given: Pump intake is 10 ft below 30-ft pay zone; P_{SI} = 130 psi (casing-tubing annulus pressure) + 8 psi (gas column pressure) + 180 psi (500 ft of 0.360 psi/ft gradient oil and water = 429 psi; Stabilized BFPD = 60 BOPD + 60 BWPD = 120 BFPD; Stabilized BHP (P_{STAB}) = 42 psi (casing-tubing annulus pressure) + 7 psi (gas column pressure) + 180 psi (500 ft of 0.360 psi/ft gradient oil) = 229 psi.

2. PI = (120 BFPD − 0 BFPD)/(429 psi − 229 psi) = 120/200 = 0.6 BFPD/psi decrease.

3. Assume: P_{MIN} = 45 psi (casing-tubing annulus pressure) + 9 psi (gas column pressure) + 18 psi (50 ft of the 75 ft of 0.360 psi/ft oil) = 72 psi, which is the pressure required to load the pump efficiently. (Note: The portion of the 75 ft of oil used is the 50 ft above the center of the pay zone, and the BHP is calculated from that point.)

4. Well capacity = $(P_{SI} - P_{MIN}) \times PI$ = (429 psi — 72 psi) X 0.6 $\frac{BFPD}{psi}$ = 357 X 0.6 = 214 BFPD. This well capacity is determined graphically in Fig. 18.

2. Assume that an additional producing rate and producing BHP are possible.

   (a) Determine the two stabilized producing rates ($BFPD_1$ and $BFPD_2$) and pressures ($P_{STAB-1} - P_{STAB-2}$).

   (b) $PI = (BFPD_2 - BFPD_1)/(P_{STAB-1} - P_{STAB-2})$.

   (c) Calculate the producing bottomhole pressure (P_{MIN}) needed to satisfy these conditions.

   (d) Well capacity = $BFPD_2 + (P_{STAB-2} - P_{MIN}) \times PI$.

Example 2

1. Given: $BFPD_1$ = 60 BOPD + 60 BWPD = 120 BFPD; P_{STAB-1} = 229 psi; $BFPD_2$ = 77 BOPD + 80 BWPD = 157 BFPD; P_{STAB-2} = 155 psi.
2. PI = (157 – 120 BFPD)/(229 – 155 psi) = 37/74 = 0.5 BFPD/psi.
3. Assume calculated P_{MIN} = 72 psi.
4. Well capacity = 157 BFPD + (155 – 72) 0.5 = 157 + 41.5 = 198.5, or 198 BFPD. This well capacity is also determined graphically in Fig. 18.

In these two examples, note that the calculated capacities differ. Neither will be the actual capacity, but the calculation made with the aid of the larger producing rate will be nearest the actual capacity. This again points out that the greatest measured producing rate should not be small compared to the calculated capacity, to maximize the chance of obtaining the most representative calculated capacity. A study of the simplified formulas used and of Fig. 18 will also indicate the errors that will result if the static BHP is measured before it ceases to increase or if the producing rates and producing BHP's are measured before they stabilize. In most cases when the calculated capacity differs from the actual capacity, the calcualted capacity will be too large. This is because the measured producing rate was obtained at too low a rate when compared to the actual capacity, or because production and pressure values used were obtained before the well had stabilized. It should also be noted that the PI of a specific well will usually decrease as the recoverable reserves decrease, as scale forms in the producing zone, etc. Therefore, the PI should be redetermnied periodically.

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. We do consider API Class D rods where the capabilities of API Class C rods are exceeded, if the system contains no hydrogen sulfide and is either non-corrosive or effectively inhibited. The discussion will be limited to situations involving the selection of those twc classes and will present a method of derating sucker rod strings which contain slim-hole couplings.

API Class C sucker rods much 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 hydrogen sulfide service if metal loss is controlled with an effective inhibitor.

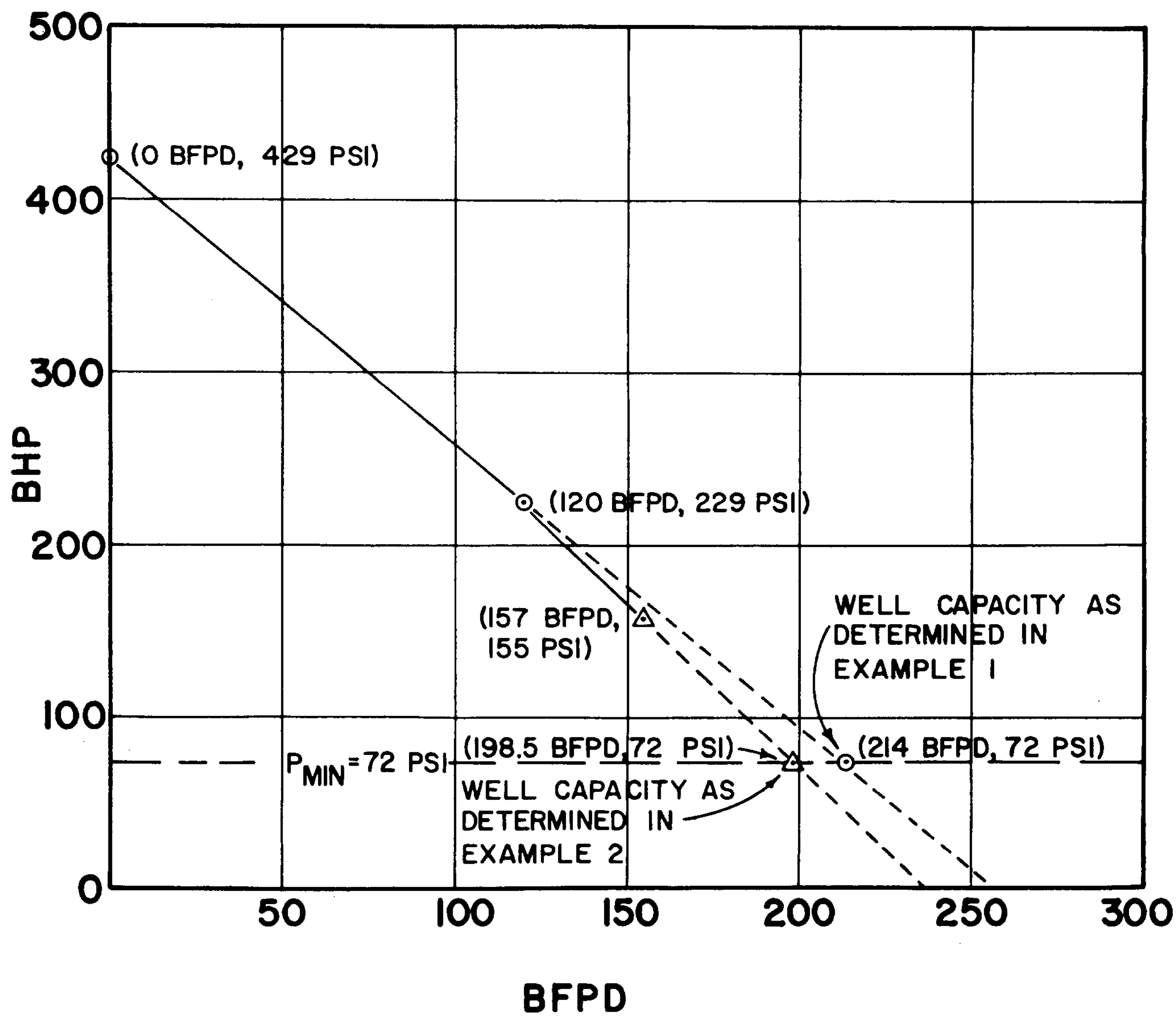
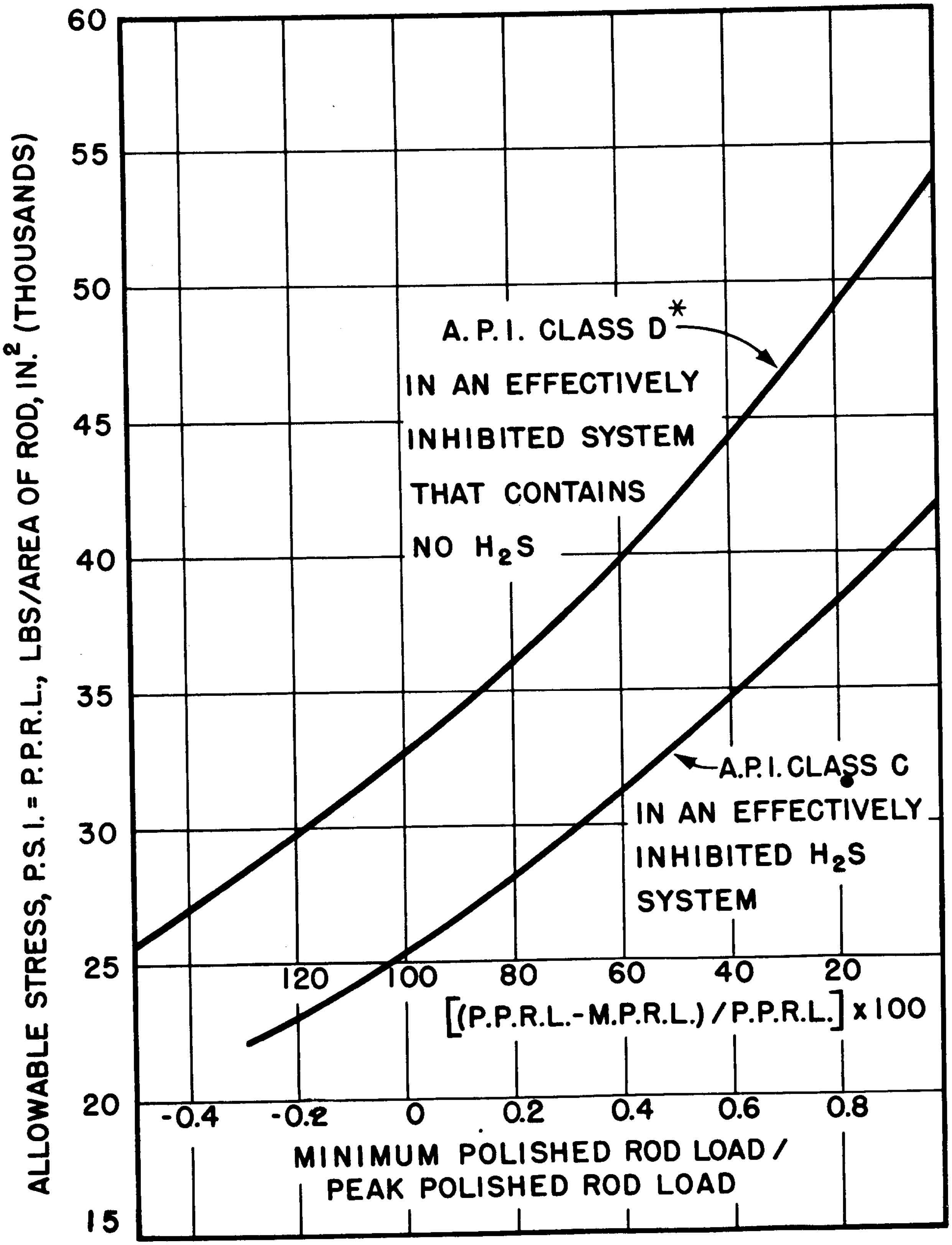


FIGURE 18

The API-suggested method of derating sucker rods for use in media other than air utilizes a modified Goodman diagram that present minimum stress versus maximum stress. It is contended that a plot of "stress" versus "stress ratio" is more meaningful. The API data indicates 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 the 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



SELECTION OF TYPE OF SUCKER RODS

FIGURE 19

* TO BE USED ONLY WHERE A.P.I. CLASS C ROD CAPABILITIES ARE EXCEEDED.

equals a minus one-third minimum tensile strength when the stress ratio is equal to minus one, etc.

From the above plot, it was assumed that the allowable maximum stress in an effectively inhibited corrosive medium 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

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 1/2"	1.77	.955	.72	1.05	.307	3.42
3/4"	1 5/8"	2.08	1.080	.92	1.16	.442	2.62
7/8"	1 13/16"	2.58	1.205	1.14	1.44	.601	2.39
1"	2 3/16"	3.76	1.393	1.53	2.23	.785	2.84
1 1/8"	2 3/8"	4.43	1.580	1.96	2.47	.994	2.48

TABLE 5

DATA ON SLIM HOLE SUCKER ROD COUPLINGS

NOMINAL COUPLING SIZE	O.D.	O.D. AREA	I.D. AREA	Δ AREA	$\left(\frac{\Delta \text{ AREA}}{\text{ROD AREA}}\right)$	$\frac{\left(\frac{\Delta \text{ AREA}}{\text{ROD AREA}}\right)}{(2.39)}$
5/8"	1 1/4"	1.22	.72	0.50	1.63	0.682
3/4"	1 1/2"	1.77	.92	.85	1.93	0.807
7/8"	1 5/8"	2.08	1.14	.94	1.565	0.655
1"	2"	3.14	1.53	1.61	2.05	0.857

Assume that a full size 7/8" coupling has an adequate Δarea/rod area, but just adequate.

Therefore, 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 x (1.565/2.39) = "$0.655S_A$".

The use of 1" slim hole couplings decreases the allowable stress from S_A to S_Ax (2.05/2.39) = 0.857 S_A, 5/8" to $0.682S_A$ and 3/4" to 0.807 S_A.

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 this 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 7/8-in. rods cause excessive failures. It can be assumed that a net coupling area divided by the net rod area ratio equal to the ratio (2.39) of the net area of a standard 7/8-in. coupling (1.44 sq. in.) divided by the area of a 7/8-in. rod body (0.601 sq in.) 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 with the above procedure. Further study by engineering personnel involved with this type of operation will be necessary to refine the derating factors.

The selection of the percentages of each size in a tapered rod string is presented as 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
- (a) New 0.98

(5) V-belt drive
- (a) Worn 0.96
- (b) New 0.98

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

(1) Worn unit $= 0.865 \times 0.98 \times 0.96 \times 0.96 = 0.781$

(2) New unit $= 0.92 \times 0.98 \times 0.98 \times 0.98 = 0.866$

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

(1) Worn unit $= 0.781 \times 0.96 = 0.750$

(2) New unit $= 0.866 \times 0.98 = 0.849$

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 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.*

*Assume that nominal pumping unit horsepower rating is equal to API gear box torque rating/4960.

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 = BFPD, barrels of fluid per day, 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 bbls 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 that 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, $\overline{SE} = L/1320000$, and the fundamental frequency (F) for any sucker rod system is: $F = 206/\sqrt{\overline{SE}}$ 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. Likewise, assuming F = 60 vib./min and SPM = 15, the order would be 4.0. 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 has been done and is quite useful as a quick reference when forecasting the shape of a dynamometer card corresponding with its order of pumping.

It is possible to determine the desirable (non-synchronous) pumping speeds in the case of tapered rod strings based on orders of pumping by the following approach, using API RP 11L as a basis:

N_o' = Undamped natural frequency of tampered rod string

$N_o' = 16{,}300 \text{ ft/sec} \times 60 \text{ sec/min} \times F_c \div 4L$ ft

$N_o' = (245{,}000 \times F_c) \div L$

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 = $N_o' \times 1.5$, $N_o' \times 2.5$, . . .

Undesirable pumping speeds = $N_o' \times 1$, $N_o' \times 2$, . . .

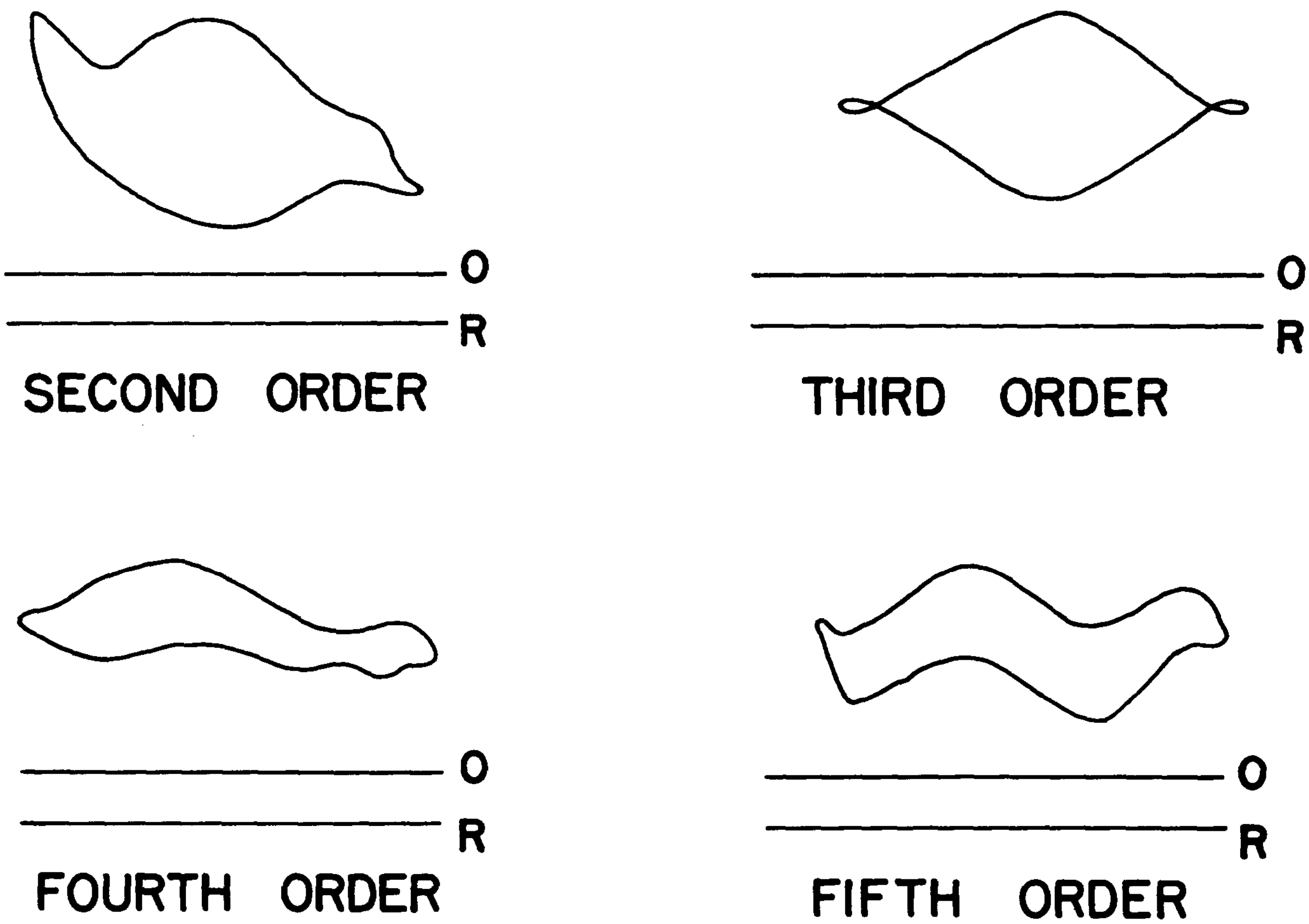
Maximum practical pumping speeds, N, strokes per minute, are non-synchronous speeds that are well below the free fall speed of the rods in the fluid being pumped. Pumping speeds and stroke lengths (S), in inches, which result in an acceleration factor ($S \times N^2/70{,}500$) greater than 0.3 are believed 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 rod, pin and coupling breaks and downtime.

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.

ORDERS OF CARDS

Sucker rod pumping systems conform, in general, to the principles of simple harmonic motion. Certain characteristic orders of cards have been developed and are normally a function of pump depth and strokes per minute. It is possible to estimate the orders of cards by using the methods previously mentioned using those two variables as the controlling coordinates.

The first order pumping situations are not encountered in oil well pumping, due to the high pumping speed required and the limitations of the free fall speed of rods at the required depth. Situations yielding second to fifth order cards



IDEALIZED ORDER
OF
DYNAMOMETER CARDS

FIGURE 20

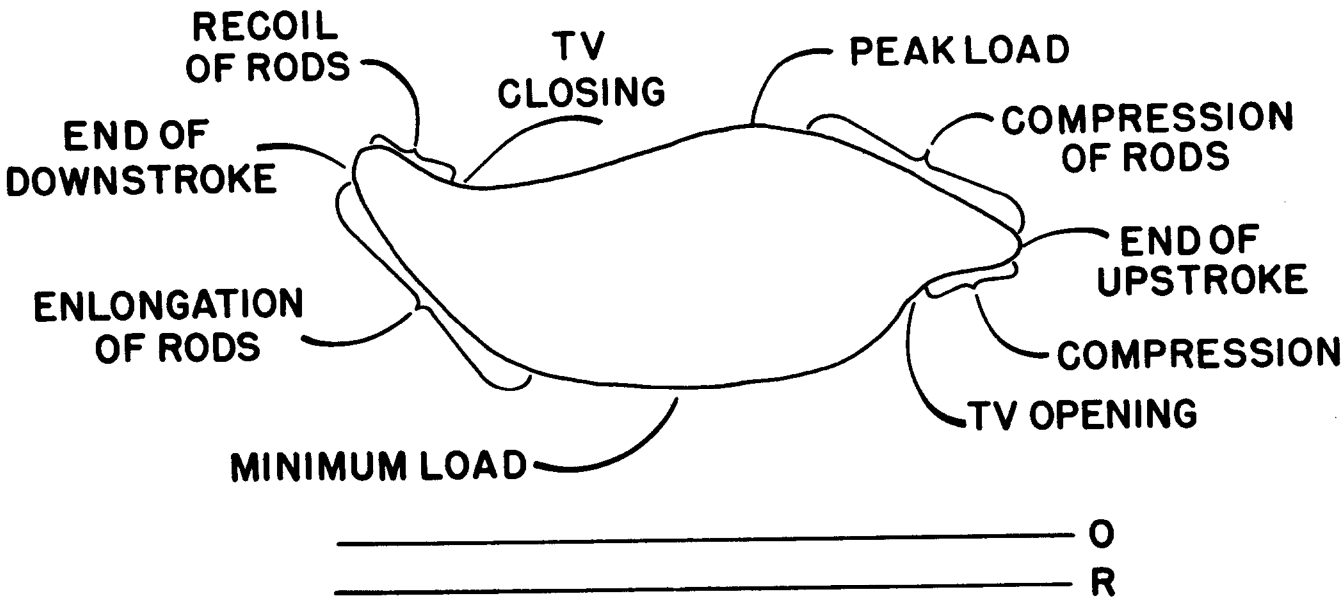


FIGURE 21

are the most common encountered today. Based on representative cards in the authors' files and other examples with which they are familiar, the following idealized cards have been constructed to portray the various orders.

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 indication of possible trouble. By varying the variables reflecting the natural frequency of vibration, a problem area can often be corrected, or induced, as the case may be.

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.

XIV
VALVE ACTION DURING THE PUMPING CYCLE

Figure 22 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

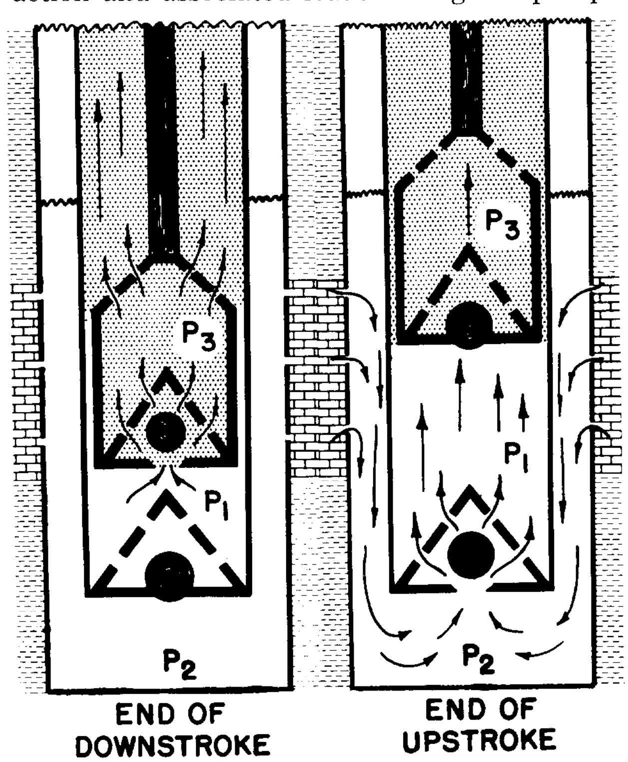


FIGURE 22

(After WORLD OIL, March 1965)

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.

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 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:

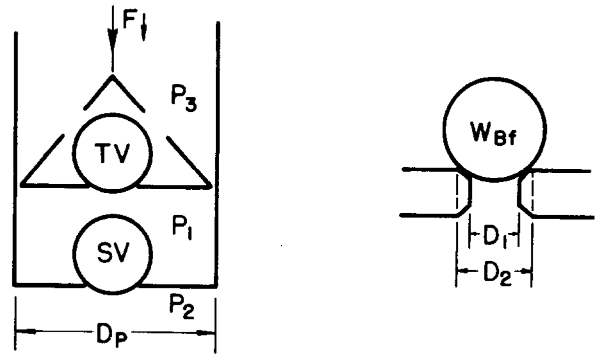


FIGURE 23

$$W_{Bf} = D_2{}^2 \times \frac{\pi}{4} \times P_3 < D_1{}^2 \times \frac{\pi}{4} \times P_1$$

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

$$P_1 > P_3 \frac{D_2{}^2 \times \pi/4}{D_2{}^2 \times \pi/4}$$

For 1½" pump, $D_1 = 0.656''$ and $D_2 = 0.723''$

$P_1 > P_3(1.216)$

$F\downarrow >\ 'P_1 - P_3)\ D_{p2} \times \frac{\pi}{4}$

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

$F\downarrow > 0.382\ P_3$

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.

2. Static pressure required to unseat SV on upstroke:

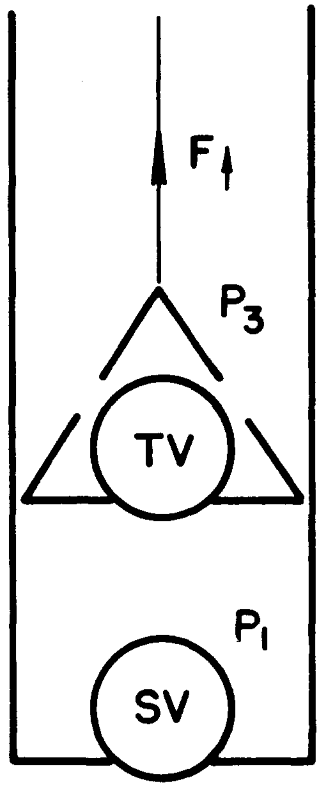


FIGURE 24

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 ft^3 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 $in.^3$.

$$P_1 \times D_2^2 \frac{\pi}{4} < P_2 \times D_1^2 \frac{\pi}{4}$$

$$P_1 < P_2 \frac{D_1^2}{D_2^2}$$

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

$$P_1 < 64.4 \times 0.89; \quad P_1 < 57.3 \text{ psia}$$

$$57.3 - 14.4 = 42.9 \text{ psig}$$

$$2 \text{ standard } \frac{Ft^3}{Bbl \times \frac{lb}{in.^2}} \times 1728 \frac{in.^3}{Ft.^3}$$

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

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

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

$$= in.^3 \text{ of gas}$$

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 $lb/in.^2$, 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 opened. 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 = $(126.85 - X) \times 0.356 \times 2.3 \times \frac{14.4}{50 - 2.3 + 14.4} \times 560/520 =$

$25.94 - 0.2045X$

$1.2045X = 25.94$

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

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

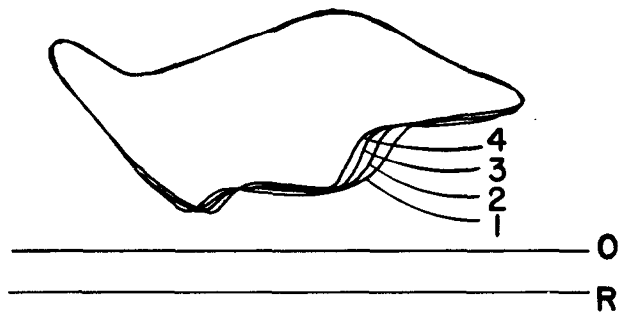
Check: $105.31 \times 0.356 \times 2.3 \times \frac{14.4}{62.1} \times \frac{560}{520} = 21.54$ in.^{3}

Assuming no slippage or pressure drop through the standing valve, volumetric efficiency = 100 (130 − 3.15 − 21.54)/130 = 100 X 105.31/130 = 81%

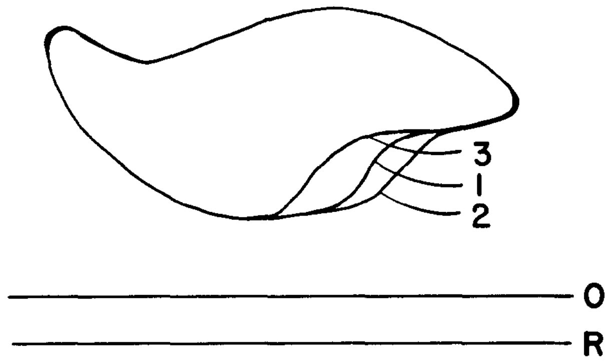
XV

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



GAS POUND

FLUID AND GAS POUNDS

FIGURE 25

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.

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 size, SPM, SL.	3. Some control by varying 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.

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 release gas that must be pumped.

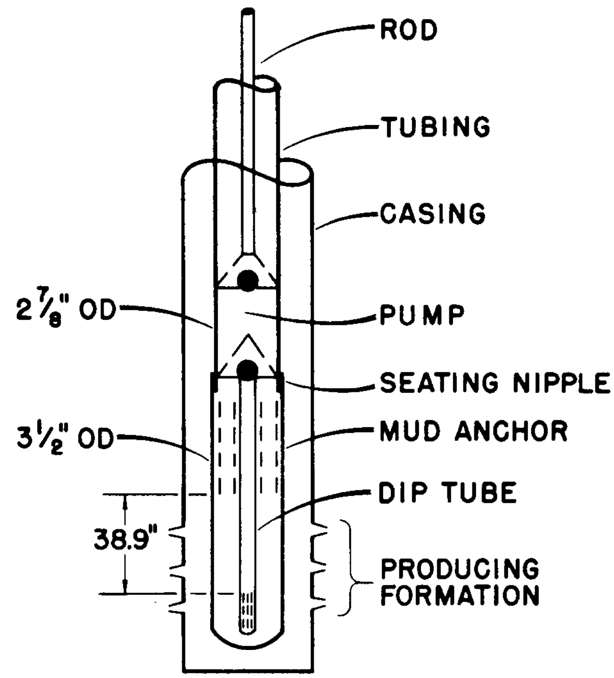
Assuming these rules are correct, it is concluded that:

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 valve 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 four 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 one and two 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.



"POOR BOY" GAS ANCHOR
(SEE GAS SEPARATION PROBLEM)
FIGURE 26

Solution:

1. Area of annulus between the mud anchor-dip tube an be determined from the following formula:

   Area of down pasage, in.^{2} = (0.00935 X BFPD)/(velocity $\frac{\text{ft.}}{\text{sec.}}$ X Pump volumetric efficiency) = 0.00935 X 300/0.5 X 0.70 = 8.01 in.^{2}
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 downhole 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?
 (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) Are the card shape and appearance the ones 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 non-synchronous 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 force caused by acceleration of fluid and/or rods which causes the plunger to travel more than it should or normally would.

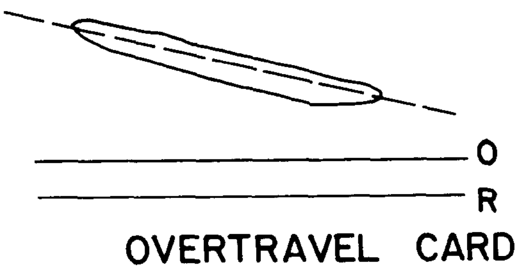


FIGURE 27

Undertravel — Some type of a restriction which causes the plunger to move less than it should or normally would.

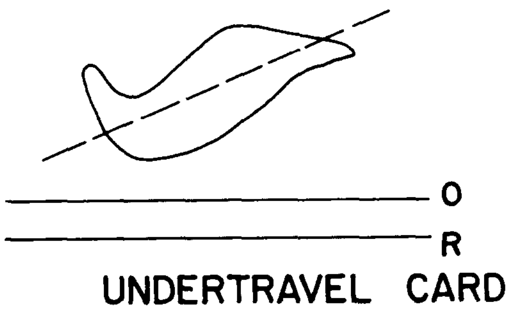


FIGURE 28

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.

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		

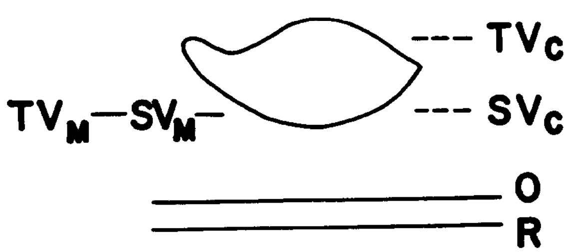
The following checklist can be used to identify possible causes when investigating undertravel situations or conditions:

Checklist B: For use in analyzing undertravel situaitons or conditions.

POSSIBLE SITUATIONS	Possible Cause Yes	Possible Cause 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 downhole friction		
11. Low API gravity fluid		
12. Stuck pump		
13. Improper lubrication of downhole pump		
14. Stuffing box too tight		
15. Tubing not anchored		
16. Rod guides, paraffin scrapers		

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

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



Facts:

(1) Normal card to possibly slight overtravel card
(2) Area of card normal
(3) Second order card
(4) SPM & Depth indicate should expect second order card.
(5) SV & TV weigh the same.
(6) SV_M & $TV_M \not\approx SV_C$
(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 ?

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

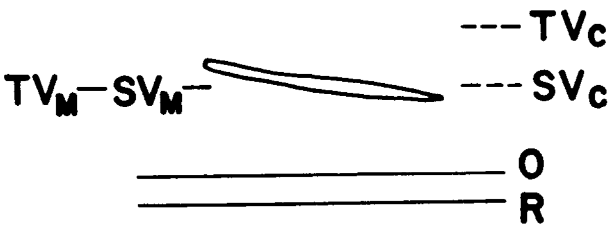
Probable Causes

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

Recommended Action

(1) If well is producing allowable, do nothing except make supervisor aware of potential problem.
(2) Test for tubing leak. If leak, pull well.
(3) If well not producing allowable, pull and repair pump.

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



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 $\not\approx$ SV_C
(6) No fluid production
(7) Fluid level indicates sufficient pump submergence.
(8) Well normally produces X bbls. oil and Y bbls. water per day.
(9) C P ?
(10) GOR ?
(11) Card representative for this well ?
(12) Past problems ?

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	

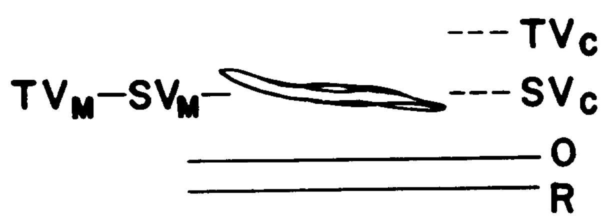
Probable Causes

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

Recommended Action

(1) Bump well.
(2) Test for tubing leak.
(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 $TV_M = SV_M$



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 $\not\approx$ SV_C
(6) Fluid production normal or slightly above normal
(7) CP ? TP ?
(8) GOR ?
(9) Card representative for this well ?
(10) Past problems ?

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	

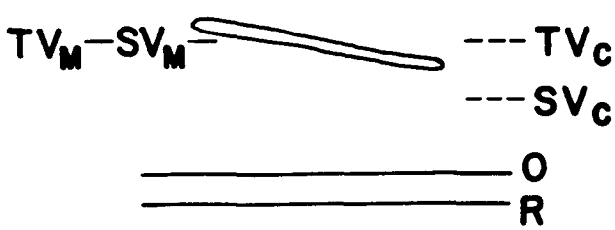
Probable Causes

(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 $TV_M = SV_M$



Facts:

(1) Overtravel card
(2) No significant card area
(3) SV and TV weigh the same
(4) SV_M & $TV_M \not\approx TV_C$
(5) Well producing some fluid but not as much as that normally expected.
(6) CP 350 psi
(7) GOR 6700 ft^3/bbls
(8) Card normal for this well ?
(9) Type of pump ?
(10) Waste space in pump ?
(11) Proper gas anchor ?
12. Past problems ?

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	

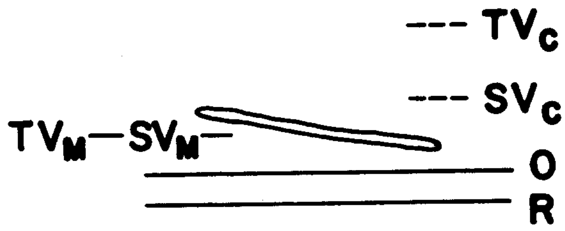
Probable Causes

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

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.

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



Facts:

(1) Normal card to slight OT card
(2) Card has area
(3) Order of card closely approximates that normally anticipated.
(4) TV_M and SV_M weigh the same.
(5) TV_M & $SV_M \not\approx TV_c$
(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 ?

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

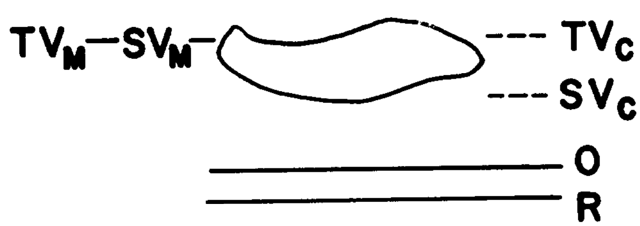
Probable Causes

(1) Bad standing valve

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 $TV_M = SV_M$



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 ?

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

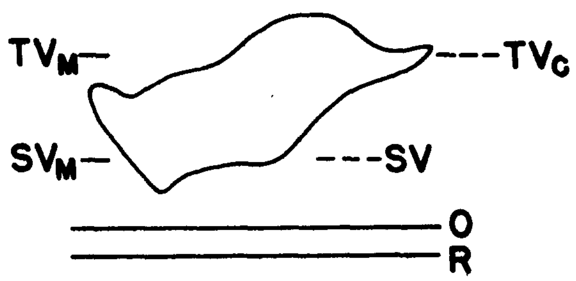
Probable Causes

(1) Parted rods above pump

Recommended Action

(1) Pull and repair rods.
(2) Run dynamometer card to determine rod problem.

Case 7: Undertravel card with card area.



Facts:

(1) Undertravel card

(2) Card has area.

(3) $TV_M \not\approx TV_C$, and $SV_M \not\approx SV_C$

(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.

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

Probable Causes

(1) Rods underdesigned

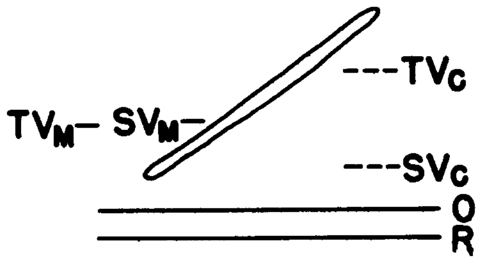
(2) Pump overdesigned

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 $TV \approx SV_M$



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.
(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. Undertravel cards.

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 downhole friction		X
11. Low API gravity fluid		X
12. Stuck pump	X	
13. Improper lubrication of downhole pump		X
14. Stuffing box too tight		X
15. Tubing not anchored		X
16. Rod guides, paraffin scrapers		X

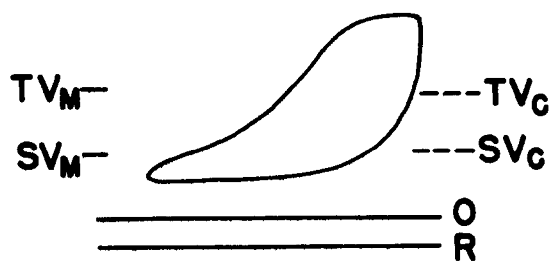
Probable Causes

(1) Sand problem
(2) Paraffin problem } Stuck pump
(3) Scale problem
(4) Stuck pump
(5) Too many rods
(6) Rods underdesigned } Design problem
(7) Pump overdesigned

Recommended Action

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

Case 9: Undertravel card with card area.



Facts:

(1) Undertravel card
(2) Card has area.
(3) $SV_M \approx SV_c$, and $TV_M \approx TV_c$
(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.

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 downhole friction	X	
11. Low API gravity fluid		X
12. Stuck pump		X
13. Improper lubrication of downhole pump		X
14. Stuffing box too tight		X
15. Tubing not anchored		X
16. Rod guides, paraffin scrapers	X	

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 operation 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.

XX
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XIX

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