Optimum Design of High-Volume Sucker Rod Pumping Systems

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INTRODUCTION

Throughout the industry, operators have been faced with producing large volumes of fluid from wellbores not designed for a sucker rod pumping system. "If I had it to do over, I would do it this way," is the statement heard many times. This paper is the result of an operator having the opportunity of doing it "this way". From the problems experienced with shot holes, inadequate casing size and gas interference, proposed new producing wells in a waterflood expansion project were approached with the idea that all subsurface and surface equipment could be designed with high volume capabilities.

The resulting installations show a favorable reduction in polished rod horsepower with a considerable savings reflecting in power bills and failure-free rod life.

GENERAL

The success or failure of a sucker rod pumping system hinges on pump efficiency. A sucker rod pump cannot be efficient if it does not fill properly or is hampered by gas interference. Gas-oil ratio is the term most commonly associated with gas interference. A more practical relationship, when referring to pump efficiency, is gas-fluid ratio.

To produce a pumping well at maximum rate, a minimum formation back pressure must be maintained. The maximum producing rates cannot be achieved with high fluid levels. Experimental work in a cased wellbore utilizing a bottomhole pressure recording device with surface readout, an acoustical well sounder, and a dynamometer determined the correct pump intake pressure which would efficiently fill a two-inch bore stationary barrel pump that was equipped with a nonrestricting diptube and mud anchor. The minimum pump intake pressure for the above size pump was 80 psi. Although this is used as a rule-of-thumb number, it is recommended that minimum pump intake pressures be redetermined if conditions vary greatly.

It is desirable to produce wells with the entire producing interval submerged in fluid in areas where scaling across the formation is a possibility. This will minimize scale precipitation.

To achieve a failure-free rod life, short, fast strokes were ruled out. Rod stress, range of stress, and pump filling time were the basic factors considered. Permissible load diagrams for a particular type pumping unit were also considered.

API Bulletins 11L3 and 11L2, available from the American Petroleum Institute, proved invaluable to this author. Anyone involved in rod pumping design or analysis will benefit by using these books.

The following questions must be answered to achieve an efficient pumping system:

Pump

1. What bore size will give the desired volume?
2. What stroke length and frequency are desirable?
3. Will torsional requirements be prohibitive?
4. Will rod stress be excessive?

Tubing

1. Will required tubing size create casing size problems?
2. Will tubing size allow sufficient rod coupling clearance?

Rods

1. Can safe design limits be met?

Casing

1. Are costs of larger sizes prohibitive?
2. Can efficient gas separation be expected?
3. Can necessary pump intake pressure be achieved?

Pumping Unit

1. Can desired stroke length be obtained?
2. What unit geometry will fit the expected dynamometer card?

3. What are the expected prime mover HP requirements with different geometry units?

After these questions are answered, "What is the Optimum System?"

MUD ANCHOR DESIGN

Past experience in this area indicated some gas interference would be present. It also indicated that large inlet areas in the mud anchor and diptube were necessary for good pump efficiency in high water-cut wells. Therefore, the optimum mud anchor design in this case was simply utilization of the tubing-casing annulus as a quiet space. (Refer to Fig. 1). This type design allows extremely low pump setting depths. When used in conjunction with rat hole, it will allow optimum gas separation, pump intake pressure and formation back pressure. The main design criteria for this type mud anchor are:

1. Strainer nipple slots must have a minimum of four times standing valve area.
2. Slotted tubing nipple, slots minimum 4-6 times standing valve area, but not wide enough to pass ball sealers, etc.
3. Annular quiet space, at least two or three pump volumes.

If extremely high gas-fluid ratios are anticipated, it may not be wise to install a tubing anchor-catcher upon initial completion. The reduction in annular area may create higher pressures below the anchor. Although tubing stretch may be eliminated, the increase in pressure can retard gas separation and drastically reduce pump efficiency.

PUMP DESIGN

The most efficient sucker rod pump for high volume and gas handling ability is a stationary barrel, single-valved, bottom hold-down pump. Better valve size and flow area relationships are obtained with this design. A poppet valve can be attached to this pump which will eliminate downstroke fluid or gas pounds and produce a slight increase in minimum loads. In a high water-cut well, this pump can operate at 95 per cent efficiency if the pump intake pressure is above 80 psi. Normally, the general relationship of all subsurface components is best when an insert type pump is used in initial design. For example:

When necessary clearance to run an insert pump is available, a rod string may be designed within Goodman diagram limitations and with adequate rod coupling clearance. Necessary quiet space for gas separation can be achieved without an excessively long or large OD diptube and mud anchor.

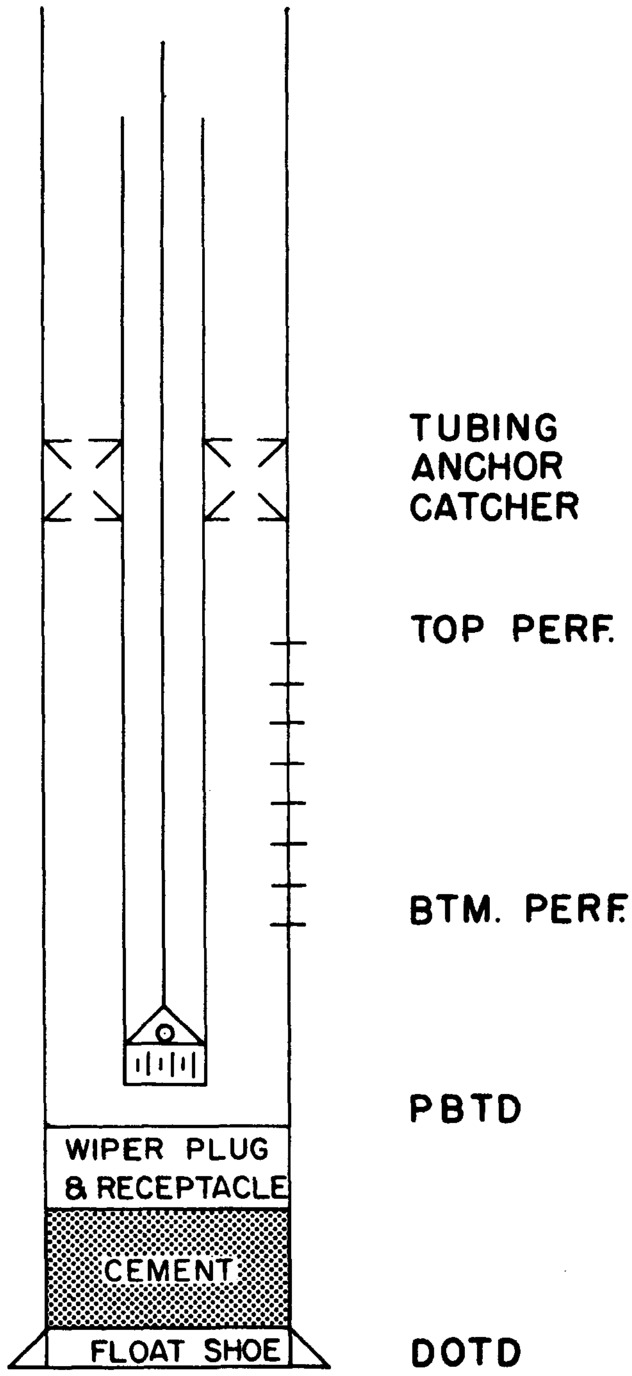
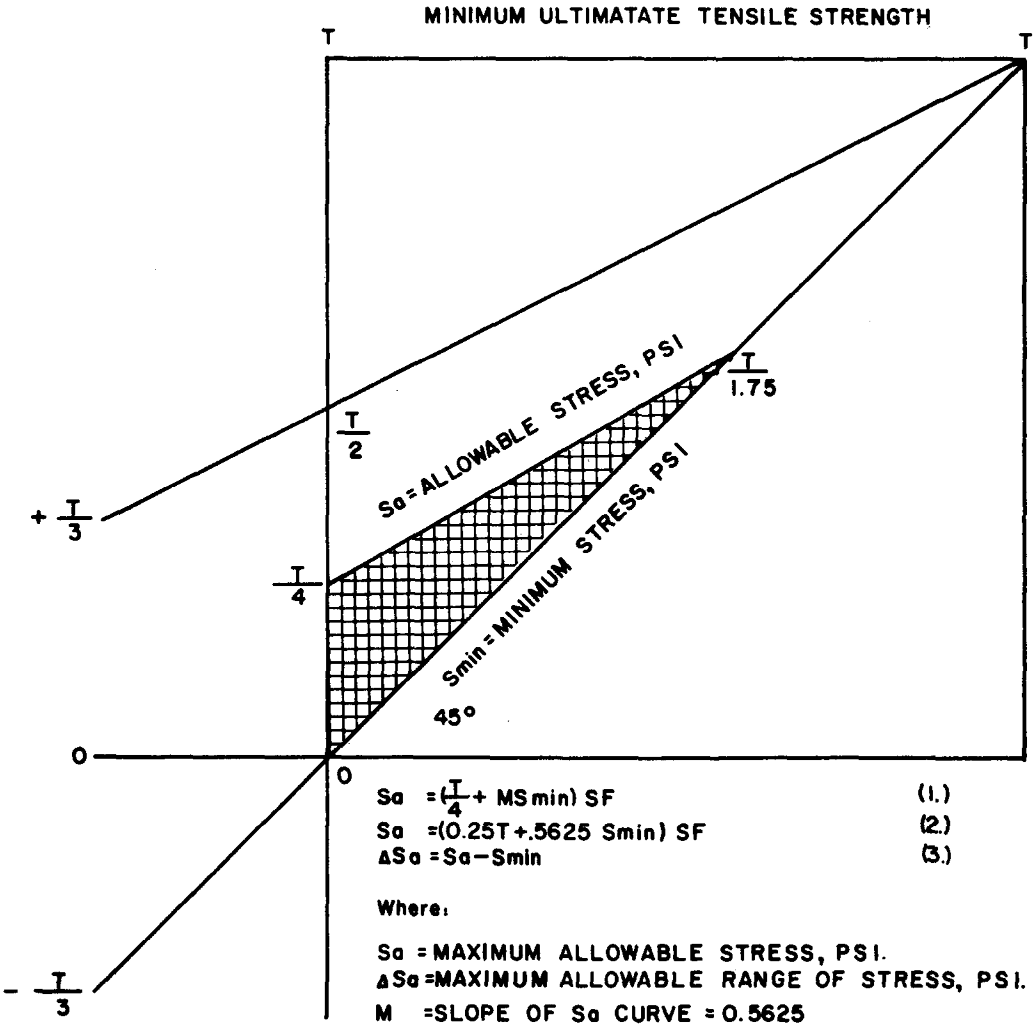


FIGURE 1

MODIFIED GOODMAN DIAGRAM FOR ALLOWABLE STRESS AND RANGE OF STRESS FOR SUCKER RODS IN NON-CORROSIVE SERVICE



$$Sa = \left(\frac{T}{4} + M S_{min}\right) SF \qquad (1.)$$

$$Sa = (0.25T + .5625\ S_{min})\ SF \qquad (2.)$$

$$\Delta Sa = Sa - S_{min} \qquad (3.)$$

Where:

Sa = MAXIMUM ALLOWABLE STRESS, PSI.
ΔSa = MAXIMUM ALLOWABLE RANGE OF STRESS, PSI.
M = SLOPE OF Sa CURVE = 0.5625
Smin = MINIMUM STRESS, PSI, (CALCULATED OR MEASURED)
SF = SERVICE FACTOR
T = MINIMUM ULTIMATE TENSILE STRENGTH, PSI.

FIGURE 2

RODS

Many factors must be considered to achieve optimum rod selection. Any functional forces on a sucker rod string increase peak polished rod loads, load range, and peak torque. Downstroke frictional forces decrease minimum loads and reduce maximum allowable downstroke velocity. The rod coupling-piston effect is a major frictional force in a high volume pumping well.

Table 1 gives optimum clearance with a given pump bore, tubing size and rod size. Obviously, these conditions cannot always be achieved, but use of this table in initial design will improve pumping conditions significantly.

A weighted section above the pump will improve maximum allowable downstroke velocity. The use of polished rods as a weighted section has created some problems. Coupling and pin design do not allow proper makeup; consequently, unscrewing and pin and coupling failures occur. Additional problems are encountered handling long sections of polished rod during normal pulling operations.

If tubing size is sufficient, sucker rods even larger than the top taper can be used. Slim hole couplings can be used without difficulty in the low stress area near the pump.

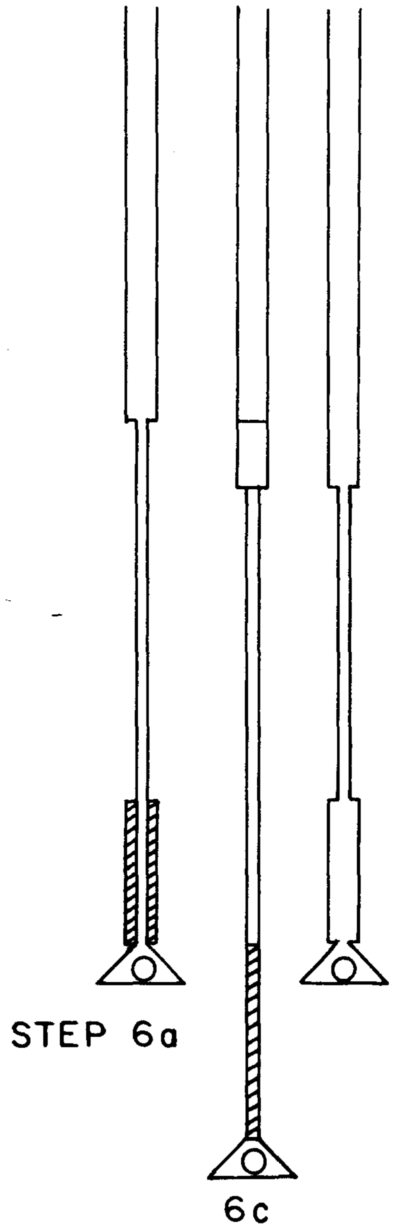


FIGURE 3

The attached Proper Rod Design Procedure, Table 2, adhering to the Goodman diagram, Fig. 2, and adjusting taper to compensate for a weighted section above the pump, has been developed to achieve optimum design. The taper adjustment involves lengthening the upper rod tapers to compensate for the weighted section. Figure 3 is a schematic illustrating taper adjustment. Steps 6a and 6c on Fig. 3 refer to Table 2, Proper Rod Design.

Optimum rod design is useless without optimum installation and operation practices. The API-recommended makeup procedure and handling practice should be used on each new string and every pulling job. The time and effort spent will be rewarded by eliminating all coupling pin or mishandling failures.

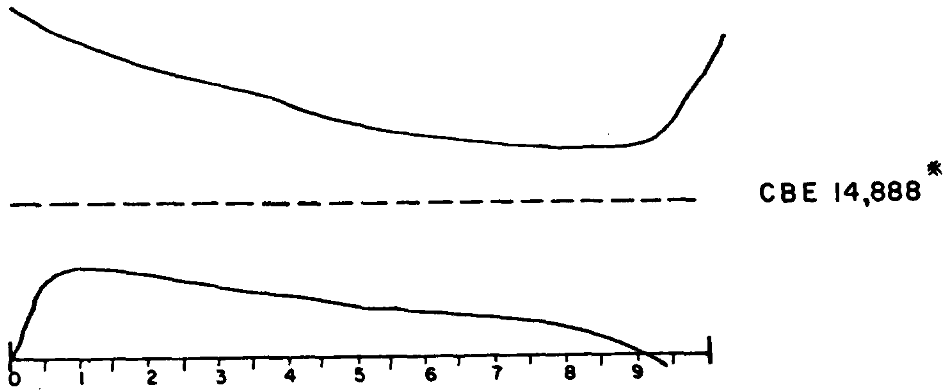
Fluid or gas pounds and corrosion can and must be controlled! Optimum design and operation practices will allow operation limited only by a rod string's fatigue endurance limits.

PUMPING UNITS

A pumping unit of conventional or improved geometry must be sized for the rod string and pump selected and also for the operating stroke frequency. One approach to unit selection is the use of permissible load diagrams, Fig. 4. The permissible load is the polished rod load necessary with a given amount of counterbalance to give a resulting net torque equal to the API rating of the gear reducer. When these loads are calculated and plotted versus polished rod displacement, the resulting diagram is a dynamometer card of the unit geometry. If a polished rod load occurs above the upstroke line or below the downstroke line, at the corresponding polished rod displacement, the reducer torque will exceed the API gearbox rating used in calculations. To use the permissible load diagrams in pumping unit selection, develop N/NO′ and FoSkr factors and select the predicted dynamometer card from the API Bulletin 11L2 card catalogue. The dynamometer cards can be scaled up and replotted with reasonable accuracy; however, the most important point is the polished rod position at calculated maximum and minimum loads. Different unit geometries can be examined versus the predicted card.

PRIME MOVERS

Unit geometry does have some effect on the required prime mover horsepower. Normally, a slight reduction can be expected with improved geometry units. For proper pumping unit control, an electric prime mover is the most desirable. Of the electric motors avail-



PERMISSIBLE LOAD DIAGRAM

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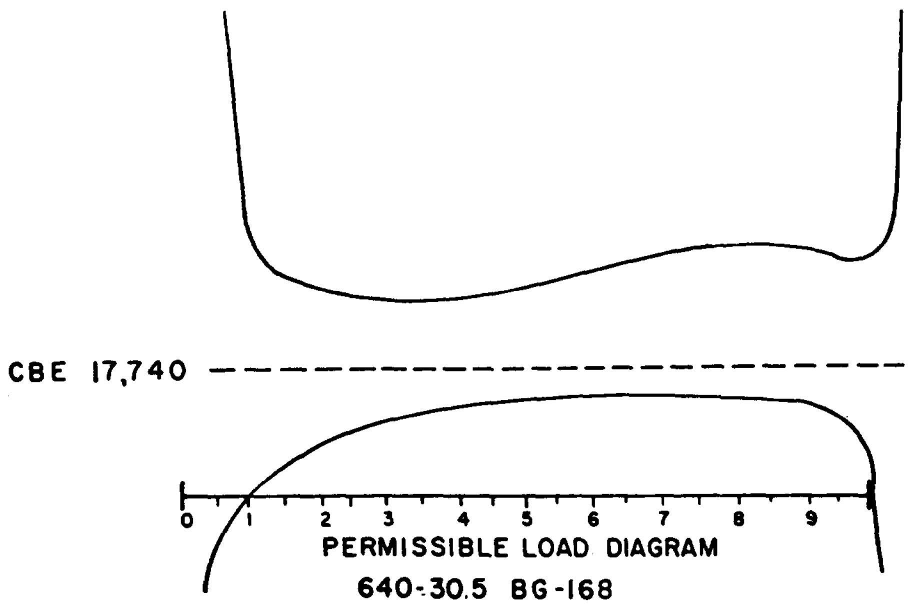
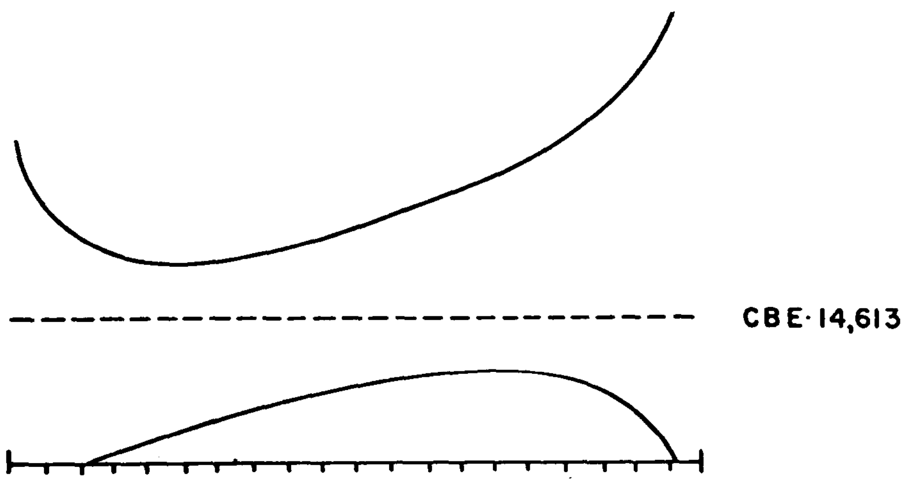


PERMISSIBLE LOAD DIAGRAM

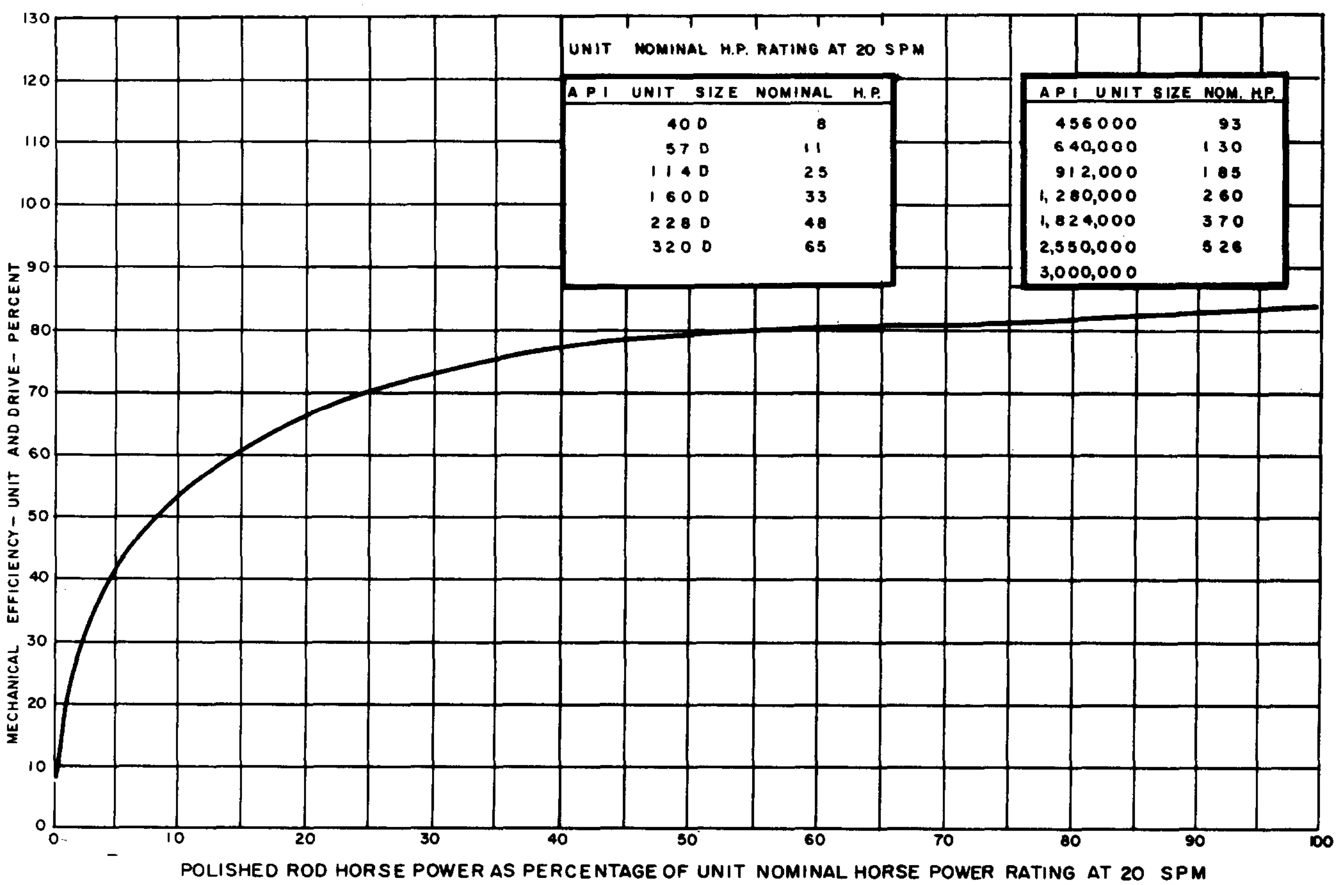
640-30.5 BG-168



PERMISSIBLE LOAD DIAGRAM

C 640D-305-168

FIGURE 4



$$H.P. = \frac{T \times N}{63,000 \times 1.57}$$

FIGURE 5

able, the Nema "D", KOF, or ultra high-slip motors will protect the gear reducer from high shock loading. The factors involving selection of an electric motor are:

1. If API gear reducer rating is or is not to be exceeded
2. What is necessary to make a motor slip and can this be achieved?
3. What are the economics of the system?

It has been stated, work input equals work output times a mechanical efficiency factor, or simply "you can't get something for nothing." When considering limited torque (high-slip) prime movers, the old conception of calculating gear reducer torque does not apply. The crank counterbalance inertia is utilized to the optimum by the high-slip motor. The net result is less gear box torque for the same amount of work output or the same gear box torque for more work output. In some cases, the use of high-slip motors will reduce the size pumping unit required.

To take advantage of limited torque motors, the loads applied must be of sufficient magnitude to enhance slipping characteristics. Manufacturers of these motors will be quite pleased to supply complete pumping system designs.

The method of prime mover selection outlined in Table 3 works quite well with ordinary engines and motors. Figure 6 is utilized with Table 3.

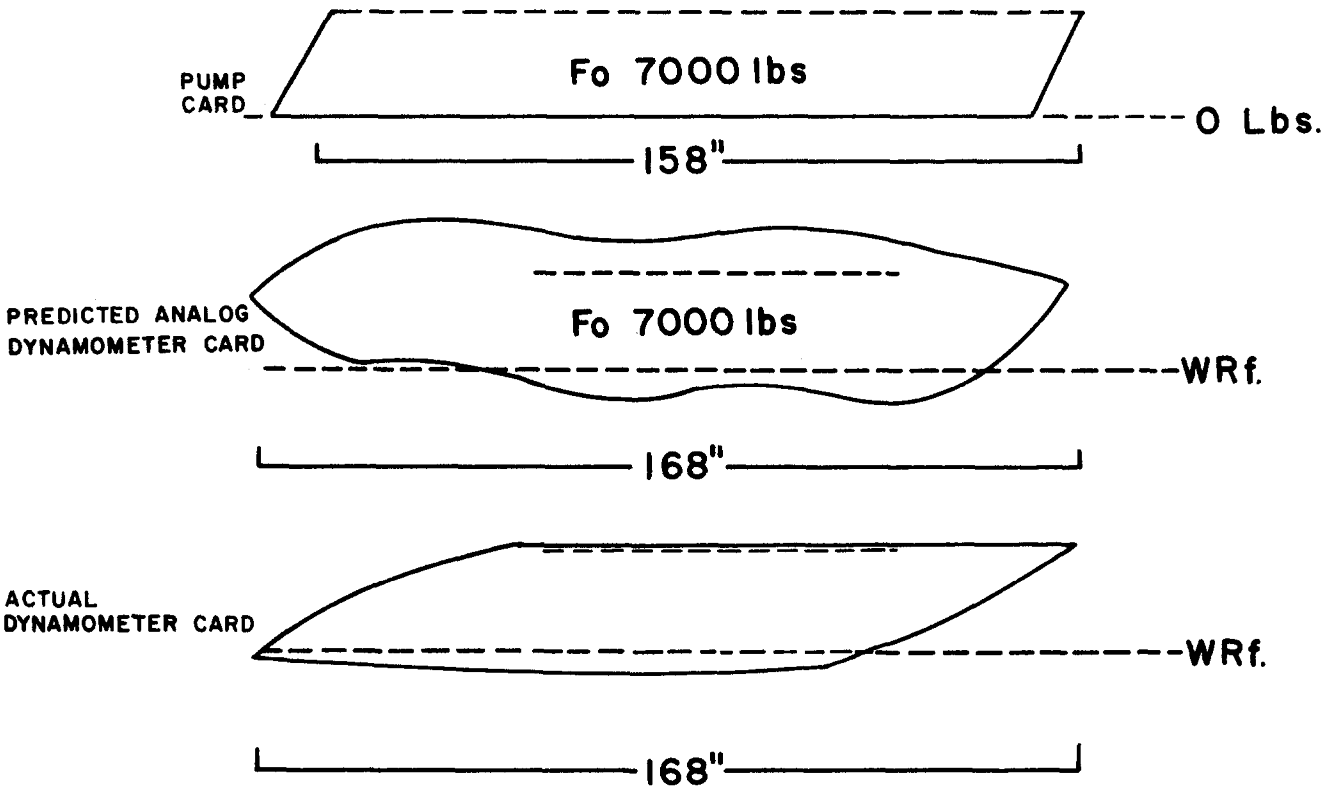
A controller equipped with a percentage timer will allow maximum production withdrawal rates. It will also allow the operator to control destructive fluid pounds; however, this will require periodic dynamometer surveillance.

A complete automation may be accomplished with pump-off devices or strain gauges. In either case, minimum down-time will provide maximum withdrawal rates.

WELL "A"

PUMP DEPTH 5275'

PRODUCING 520 BFPD 70% WATER



	HP	ELECT. COST FOR LIFE	ROD COST FOR LIFE
PUMP CARD	18.31	$31,460	
PREDICTED CARD	28.84	$103,680	$21,550
CONV. OR BG			
ACTUAL CARD	23.87	$85,900	$15,085
MK II			
SAVINGS		$17,180	$6,465
TOTAL		$23,645	
PNW		$ 5,766	$2,370
TOTAL		$ 8,136	

FIGURE 6

DRIVE SYSTEMS

To reduce shock loading into the gear reducer from the prime mover, belts and sheaves, and to allow maximum prime mover slip, the lightest belt sheave design should be used. A considerable increase in prime mover speed variation (slip) can be obtained by reducing the OD and weight of the gear box sheave. The new "V" section belts and sheaves offer a wide range of sizes, lightweight, high horsepower per belt ratings, and are very favorably priced. Proper alignment and belt tension must not be left to chance. Belt manufacturers will furnish tables, gauges for proper tension.

CONCLUSIONS

The optimum selection of components has produced a very favorable economic picture.

Figure 6 presents the results of optimum selection of rod pumping system components. It shows a theoretical pump card, predicted analog dynamometer card and the actual dynamometer card. The horsepower shown on Fig. 6 was calculated from the cards shown.

The electrical cost for life was calculated using \$0.012/kwh, and a 20-year operating life. The rod cost was calculated on a four-year change-out for the predicted analog card and a five-year change-out on the actual card.

The present net worth savings for this improved geometry unit is \$8136. The additional initial investment for the improved geometry unit over conventional units was approximately \$1000, making a net present worth savings of \$7136.

BIBLIOGRAPHY

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TABLE 1

SUCKER ROD COUPLINGS – MINIMUM TUBING SIZE
FULL SIZE COUPLINGS

ROD SIZE	OD x LENGTH	AREA	USE WITH OD TUBING	AREA	DIFF.	MAX. PLNGR. SIZE
5/8	1-1/2 x 4	1.767	2-1/16	2.405	.638	3/4"
3/4	1-5/8 x 4	2.074	2-3/8	3.140	1.066	1-1/16"
7/8	1-13/16 x 4	2.580	2-7/8	4.702	2.122	1-1/2"
1	2-3/14 x 4	3.758	3-1/2	7.031	3.273	2"
1-1/8	2-3/8 x 4	4.430	3-1/2	7.031	2.601	1-25/31"

SLIM HOLE COUPLINGS

ROD SIZE	OD x LENGTH	AREA	USE WITH OD TUBING	AREA	DIFF.	MAX. PLNGR. SIZE
1/2	1 x 2-3/4	.785	1.660	1.490	.705	7/8"
5/8	1-1/4 x 4	1.227	1.990	2.035	.808	1"
3/4	1-1/2 x 4	1.767	2-1/16	2.405	.638	7/8"
7/8	1-5/8 x 4	2.074	2-3/8	3.140	1.066	1-1/16"
1	2 x 4	3.142	2-7/8	4.702	1.560	1-1/4"

TABLE 2

PROPER ROD DESIGN

1. Find corresponding rod size, stroke length, SPM and pump bore in pumping system design tables. Note and record value of MPRL. (API BUL 11 L3) or use API-recommended Practice for Design Calculations for sucker rod pumping systems (API RP-11L).

2. Determine minimum stress: MPRL + Area of Top Rod = Smin

When rod size is:	1/2"	5/8"	3/4"	7/8"	1"	1-1/8"
Area of rod is:	0.196	0.307	0.442	0.601	0.785	0.994

3. Refer to Modified Goodman Diagram (Fig. 2) and use minimum tensile strength of type rod selected in formula.

4. Apply Smin determined in Step 2 to formula to determine allowable Smax. Use Corrosion Service Factor appropriate for well conditions. If calculated allowable Smax is greater than the stress predicted in design tables, the top rod size selected is large enough.

5. To determine the proper amount of weight necessary to make the travelling valve open on the downstroke:

   a. Determine fluid load, i.e.:

   $Fo = .34 \times G \times D^2 \times H$

   When G = Specific gravity of produced fluid
   D = Pump bore
   H = Net lift of S.N. depth

   b. Necessary weight is Fo x Cnst. listed below for appropriate pump bore:

Pump Bore:	1-1/16	1-1/4	1-1/2	1-3/4	2	2-1/4	2-1/2
Cnst:	.236	.219	.219	.156	.139	.120	.099

   Weight can be polished rods or large rods.

6. To correct API taper for weight above pump:

   a. Determine difference in selected sinker bars in wt/ft and normal bottom taper in wt/ft.
   b. Divide the difference in weight (from Step 6a) into the required weight from 5b.
   c. Add this distance to original S.N. depth for adjusted string length.
   d. Apply API percentage to obtain footage of top (and middle, if three-way) taper.
   e. Determine length of sinker bar section by dividing weight determined in 5b by the weight in lbs/ft of sinker bars selected.
   f. Determine necessary length of bottom taper.

7. Run weighted section immediately above pump.

   Run the rest of bottom taper.

   Run the determined adjusted API top or middle tapers.

TABLE 2 – CONTINUED

EXAMPLE

1. P. 265, API Bul 11L3

   Rod No. 76, Pump Depth 5000', Production 300 (2-7/8" tbg).

Pump Dia.	Stroke	SPM	PRRL	MPRL	Stress
1.50	120	10.1	15,541	4879	25.858

2. Smin 4879 lb + .601 in^2 = 8118 psi

3. For API Grade K and SF of 1.0: S max = (23,750 + .5625 Smin) SF

4. Smax = [23,750 + (.5625 x 8118)] 1

   = 28,316 psi

   Max. stress predicted in Step 1 = 25,858 psi

5. a. Fo = .34 x G x D^2 x H
   Fo = .34 x 1 x 2.25 x 5,00 = 3825 lbs

   b. Weight = Fo x Cnst
   = 3825 lbs x .216 = 826 lbs

6. Select 1" rods for weight

   a. 1" rods = 2.90 lbs/ft
   3/4 rods = 1.63 lbs/ft
   Difference = 1.27 lbs/ft

   b. Equivalent length = 826 lbs + 1.27 lbs/ft = 650 ft.

   c. Adjusted string length = 650 ft. + 5000 ft. = 5650 ft.

   d. Top taper = 5650 ft. x 30.9% = 1745.8 ft. or 1750 ft. of 7/8"

   e. Sinker Bars = 826 lbs = 2.90 lbs/ft or 300 ft. of 1"

   f. Middle Taper 5000 ft - (1750 ft. + 300 ft.) = 2950 ft of 3/4"

TABLE 3

APPLYING A PRIME MOVER TO SUCKER ROD PUMPING SYSTEMS

$$\text{PRIME MOVER HORSEPOWER} = \frac{\text{Polished Rod Horsepower}}{\text{Surface Efficiency x K}}$$

Where: (1) Polished Rod Horsepower is either measured or derived from API Standard RP-11L, "Design Calculations for Pumping Systems".

(2) Surface efficiency read from curve (Fig. 6)

(3) K for Non-Uniform torque units. (Air Balance and Conventional)

K = .80 for Nema D Motors and slow speed engines

K = .58 for Nema C Motors and multi-cylinder engines

(4) K for Uniform torque units (Mark II Units)

K = 1.00 for Nema D Motors and slow speed engines

K = .725 for Nema C Motors and multi-cylinder engines

EXAMPLE

What HP motors, Nema "C" and "D", would be required for a conventional unit when lifting:

a. 175 BPD
b. Depth—5000' (fluid level 4500')
c. Tubing 2" (unanchored)
d. SPM—16
e. Stroke 54"
f. Pump – 1-1/2"
g. S. G. – 0.9
h. Rods – 7/8" (30.9%) – 3/4" (69.1%)
i. Polished Rod Horsepower – 8.4
j. Peak (In-Balance) Torque – 132,000 in. lbs.

1. By observing 132,000 in. lbs. peak torque, the API speed reducer to handle this load would be an API 160,000 in. lbs.

2. From Fig. 6, 160,000 in. lbs. is nominal rated 33 HP.

3. Polished rod horsepower (8.4) divided by nominal HP 33 equals 25.5%.

4. From curve (Fig. 6), move vertically from 25.5% (on horizontal axis) upward to curve, then horizontally to the left to read 71%.

5. Select appropriate K value.

6. Horsepower requirement $= \frac{8.4}{.71 \times .8}$ = 14.8 HP for Nema "D" or slow speed engine. Select 15 HP motor.

7. Horsepower requirement $= \frac{8.4}{.71 \times .58}$ = 20.2 HP for Nema "C" or multi-cylinder engine. Select 20 HP motor.