Sizing a Beam Pumping Unit to a Well

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PROBLEM OF PREDETERMING REQUIREMENTS

Since the first oil well was put on the beam, the oil industry has been confronted with the problem of trying to predetermine requirements of the surface pumping equipment. Industry engineers have struggled with this problem for years. Although numerous methods of well load calculations have been devised, the sizing of the pumping unit installation is still regarded as an "educated guess". This problem is recognized by leading men in the oil industry as one that will require much study and should not be taken lightly.

At a recent API Subcommittee on Standardization Meeting held in Kansas City, a member of that committee made the following comment:

"Peak crankshaft torque on a pumping unit that is to be installed is estimated by using one of the half dozen or so formulas that have been imagined. These formulas will predict peak crankshaft torque within 10 or 15% under certain conditions and then may be 50% or 100% in error under other conditions. This is illustrated by the following table which shows peak torque calculated by two different formulas for two different well conditions.

	Well A	
	Condition 1	Condition 2
Calculated peak torque using Formula X	170,000#"	180,000#"
Calculated peak torque using Formula Y	382,000#"	328,000#"
Peak torque calculated from measurement of peak electric motor input and using motor characteristic curve	386,000#"	350,000#"

In this case, a check of the peak torque calculated from electric motor input measurement showed that the formula giving the higher torque peak was more nearly accurate."

It is readily seen that a more reliable method should be available for well load calculations to enable the operator to purchase equipment that is sufficient to meet well requirements; yet, the operator should not be penalized by purchasing equipment larger than needed. Also a standard method of well load calculations would enable equipment manufacturers to arrive at comparable size units when given the same well conditions. As shown in the above example, the unit furnished by using "Formula X" would be much too small for the well; while the unit furnished by using "Formula Y" would be more nearly correct. In both "Formulas" the results were obtained while using identical well conditions.

The standard method of well load calculations for sizing a beam pumping unit should be simple, yet reasonably accurate, and readily applicable to any set of well conditions.

It is the purpose of this paper to present such a method for well load calculations and to show how it is applied to a given set of well conditions in the selection of the correct size of pumping unit.

METHOD OF DETERMINING

The method proposed in this paper, one developed some years ago, owes its relative reliability to being based on a statistical study of a series of field tests. These tests used a torque meter in the V-belt sheave on the pumper reducer in addition to the regular dynamometer on the polished rod.

This approach permitted lumping of a large number of effects into two simple dynamic or motion factors, the "impulse factor" (IF) and the "counterbalance factor" (CF). Some of these variables, which can be studied in detail in a more theoretical approach to the problem, are listed below:

A. The friction of the oil in the tubing and the rods against the tubing are obvious factors.
B. Buoyancy of the rods in oil can justify a small correction on the downstroke.
C. Inertia or the resistance to speeding up or slowing down can be a factor. For example, the "flywheel" effect of the rotating crank counterbalance has a considerable ability to absorb peak loads.
D. Pumper geometry or proportions of the various elements and working centers can greatly affect the rate of acceleration and in turn the polished rod load at any particular point in the stroke. It will alter the effective length of the crank and at the peak load point.
E. The effect of the longitudinal vibration of the sucker rod string is another important consideration. Polished rod loads and pump performance vary considerably as pumping speeds are increased through resonant and non-resonant conditions, the vibration being excited at the resonant speeds.

In the interest of simplicity the above factors will not be further discussed. As stated above they are given reasonable weight in the impulse and counterbalance factors which have been developed.

Before the actual well load calculation can be made, it will be necessary to understand what occurs during a pumping cycle and define some of the terms used in the derivation of the equations for calculating peak polished rod load, required counterbalance and expected peak torque.

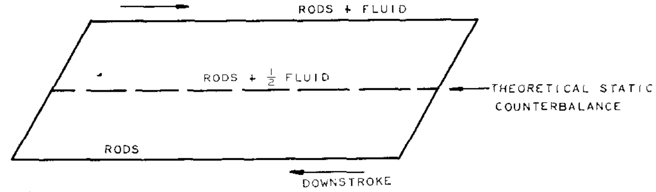

FIG. 1

TABLE 2

IMPULSE & COUNTERBALANCE FACTORS

Strokes Per Minute	STROKE LENGTH (IN.)																									
	16		20		24		28		30		34		36		42		48		54		64		74		86		120	
	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF	IF	CF
8	1.05	.996	1.06	.995	1.06	.994	1.06	.993	1.06	.992	1.06	.992	1.07	.991	1.07	.990	1.07	.988	1.08	.986	1.09	.984	1.10	.982	1.10	.978	1.14	.971
10	1.06	.994	1.06	.992	1.06	.991	1.07	.990	1.07	.989	1.07	.987	1.08	.986	1.08	.984	1.09	.981	1.09	.978	1.11	.975	1.12	.972	1.14	.968	1.18	.958
12	1.06	.991	1.06	.988	1.07	.987	1.08	.986	1.08	.985	1.09	.981	1.09	.980	1.10	.977	1.11	.974	1.12	.970	1.13	.966	1.15	.961	1.18	.955	1.26	.940
14	1.06	.988	1.07	.985	1.08	.983	1.08	.981	1.09	.979	1.10	.976	1.10	.974	1.12	.970	1.13	.967	1.14	.962	1.16	.957	1.18	.950	1.22	.942	1.35	.923
15	1.07	.987	1.07	.983	1.08	.980	1.09	.977	1.09	.976	1.10	.973	1.11	.971	1.12	.966	1.14	.963	1.15	.958	1.17	.952	1.20	.945	1.24	.935	1.39	.914
16	1.07	.985	1.07	.981	1.08	.977	1.09	.974	1.10	.972	1.11	.969	1.12	.967	1.13	.962	1.15	.958	1.16	.952	1.19	.945	1.23	.938	1.28	.927	1.48	.903
18	1.08	.980	1.08	.976	1.09	.972	1.10	.967	1.11	.965	1.12	.961	1.13	.959	1.15	.953	1.17	.947	1.20	.941	1.23	.932	1.29	.922	1.36	.911	1.67	.882
20	1.08	.976	1.09	.971	1.10	.967	1.11	.961	1.12	.958	1.14	.954	1.14	.951	1.17	.944	1.20	.936	1.23	.930	1.28	.918	1.34	.907	1.42	.895	1.85	.861
25	1.10	.965	1.11	.957	1.13	.949	1.15	.941	1.16	.937	1.18	.930	1.20	.927	1.24	.917	1.28	.907	1.34	.899	1.46	.881	1.60	.865	....	....	....	....
30	1.12	.950	1.14	.940	1.17	.929	1.20	.918	1.22	.913	1.26	.904	1.28	.899	1.35	.885	1.42	.872	1.53	.860	1.75	.841	....	....	....	....	....	....

IF = IMPULSE FACTOR. CF = COUNTERBALANCE FACTOR.

In analyzing the loads carried by the polished rod, we find that the basic load consists of the weight of the sucker rod string. As the polished rod moves upward from the bottom stroke position, the traveling valve closes and an additional load is added to the polished rod equal to the force caused by the head of the fluid acting on the net area of the pump plunger. (Where a tapered rod string is used, a small correction is made for the upward force acting on the difference in areas of the two rod diameters.) At extremely slow speeds this is the static upstroke load and is shown in Fig. 1 as the weight of rods plus the weight of fluid. As the polished rod starts on the downstroke, the standing valve closes and the traveling valve opens, transferring the load from the plunger to the tubing. Again the resultant load is the weight of the sucker rod string. In Fig. 1, the inclination of the ends of this static card is due to rod stretch and contraction as the load is picked up and released.

Length of Stroke Increased

As the length of stroke and strokes per minute are increased, there is superimposed on these static loads a dynamic effect that tends to increase the upstroke loads and to decrease the downstroke loads. A study of dynagraph readings taken from nearly 1500 pumping wells revealed that this dynamic effect on the upstroke was a function of polished rod acceleration, inertia, friction, rod string vibration, rod stretch, etc. For simplicity of application and with reasonable accuracy, all of these unknown effects on the first half of the upstroke load were combined with one empirical factor called the "impulse factor". These impulse factors were determined for various combinations of stroke lengths and strokes per minute and are shown as "IF" in Table 2.

From these studies it was found that the peak load on the upstroke could be determined by multiplying the static upstroke load by the appropriate impulse factor. This upstroke load is shown in Fig. 2 as the "peak polished rod load."

Dynamic effects, similar to those encountered in the first half of the upstroke load, also effect the downstroke load. However, these are subtractive, rather than additive, creating a lower minimum load in the first half of the downstroke.

In beam pumping applications it is necessary to counterbalance a portion of the well load in order to minimize the peak torque on the gear reducer and to keep the peak torque equal on both the upstroke and the downstroke of the pumping cycle. In the static condition of Fig. 1, the theoretical counterbalance is shown as that amount of counterbalance that gives equal net loads to the gear reducer on both the upstroke and downstroke. This obviously is the weight of the sucker rods plus one half the weight of the fluid.

However, extensive field experimental work has revealed that this amount of static counterbalance is approached only at very slow speeds and short strokes of the polished rod. For higher speeds and longer strokes, this theoretical counterbalance must be multiplied by a "counterbalance factor" to determine the actual required amount of counterbalance. From a series of torque studies and dynagraph readings, it was found that the dynamic factors which influence the upstroke and downstroke loads of the polished rod also influence the effectiveness of the counterbalance.

In the field tests underlying this method of pumping unit application, the counterweight was experimentally adjusted to equalize the upstroke and downstroke peak loads on the gear reducer. It was found that less counterweight was actually needed than the static calculations dictated, with the amount decreasing as the stroke was lengthened or the speed was increased. The resulting correction may be very significant where the weight of the rod string is high in proportion to the weight of the oil, such as will be found in deep wells with small bore pumps. The value for the counterbalance factor as determined experimentally is shown as "CF" in Table 2 for different combinations of stroke lengths and strokes per minute.

From the foregoing we can deduct that the net upstroke load to the cranks is the uncounterbalanced portion of the polished rod load, or the "peak polished rod load minus the required counterbalance." To find the approximate expected net gear torque, it is necessary to multiply the uncounterbalanced load by one half of the stroke length.

Equations

At this point we can establish equations to find peak polished rod load, required counterbalance and expected gear torque.

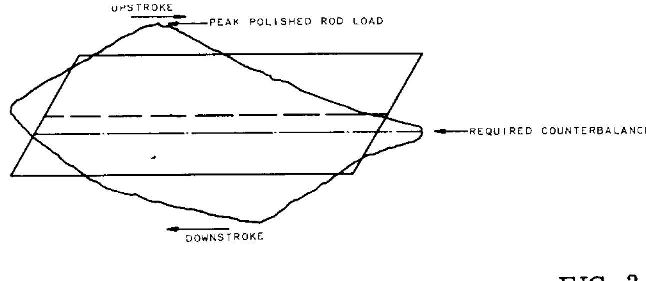

FIG. 2

PRL = (rods fluid) X (IF) Eq. I

CB = (rods 1/2 fluid) X (CF) Eq. II

T = (PRL - CB) X (1/2 stroke) Eq. III

Where,

PRL = Peak polished rod load, lb

CB = Maximum counterbalance, lb

T = Peak gear torque, in.-lb.

IF = Impulse factor

CF = Counterbalance factor

Rods = Weight of rods in air

Fluid = Net downward force (lb) created by fluid pressure of the column being lifted acting on the exposed surfaces of the sucker rod string and bottom hole pump

Stroke = Length of stroke (in.) required to get daily production of oil

In selecting pumping units, most operators try to be conservative; therefore, well load calculations are made so that the unit they choose will have ample capacity to meet the most severe pumping conditions of the well. This means that the fluid level is at the pump and the fluid being lifted has a specific gravity of 1.0. In this case the following conditions are required in the above equations to calculate pumping unit requirements.

(1) Maximum expected pumping depth
(2) Maximum expected plunger size
(3) Rod string size
(4) Maximum expected unit SPM
(5) Maximum stroke length necessary to get daily production with above plunger size and unit SPM

However, if the well loads are to be calculated for known conditions, the following additional factors should be considered:

(6) Specific gravity of oil
(7) Specific gravity of water
(8) Percent water
(9) Actual net lift of fluid

These factors affect only the fluid in the equations I and II and they have a tendency to decrease the fluid loads except in conditions where the specific gravity of the fluid is greater than one (1.0).

Table 1 has been developed to give the calculated loads for various combinations of plunger diameters and rod strings per 1000 feet of well depth. These loads are calculated for a specific gravity of 1.0 for the fluid and with a net fluid lift from the top of the pump plunger (no bottom hole pressure). The tapered rod strings are proportioned in such a manner that the stresses are equal at the top section of each size rod (this is the theoretical maximum stress in the rod string). Therefore, with a specific gravity of 1.0 for the fluid, the weight of rods per 1000 feet and the weight of fluid per 1000 feet in equation I can be combined to give a "polished rod dead load per 1000 feet" of well depth. Likewise, in Equation II the weight of rods per 1000 feet and one half the weight of fluid per 1000 feet can be combined to give the "counterbalance required per 1000 feet" of well depth. The following equations would apply for Table 1.

PRL = (polished rod dead load/1000 ft.) X depth (ft.) X IF Eq. IV

CB = (counterbalance required/1000 ft.) X depth (ft.) X CF Eq. V

Using the following given conditions and Equations IV, V, and III, a sample well load calculation can now be made.

(1) Pumping depth 4000 ft.

(2) Plunger size 1-1/2"

(3) Rod string size 3/4"

(4) Unit SPM 16

(5) Stroke length 42"

From Table I across from 1-1/2" plunger and under 3/4" sucker rods we find:

Polished rod load/1000 ft.=2184 lb.

Counterbalance required/1000 ft. =1897 lb.

From Table 2 across from 16 SPM and down from 42" stroke we find:

Impulse factor (IF) = 1.13

Counterbalance factor (CF) = .962

TABLE I **POLISHED ROD LOADS (LB)**

Pump Plunger Diameter	ROD STRING SIZES: 1"		1" & 7/8"			1", 7/8" & 3/4"				7/8"		7/8" & 3/4"			3/4"		3/4" & 5/8"			5/8"	
	Polished Rod Dead Load per 1000 Feet	Counter-balance Required per 1000 Feet	% 1" Rods	Polished Rod Dead Load per 1000 Feet	Counter-balance Required per 1000 Feet	% 1" Rods	% 7/8" Rods	Polished Rod Dead Load per 1000 Ft.	Counter-balance Req. per 1000 Ft.	Polished Rod Dead Load per 1000 Ft.	Counter-balance Req. per 1000 Ft.	% 7/8" Rods	Polished Rod Dead Load per 1000 Ft.	Counter-balance Req. per 1000 Ft.	Polished Rod Dead Load per 1000 Ft.	Counter-balance Req. per 1000 Ft.	% 3/4" Rods	Polished Rod Dead Load per 1000 Ft.	Counter-balance Req. per 1000 Ft.	Polished Rod Dead Load per 1000 Ft.	Counter-balance Req. per 1000 Ft.
1"	2888	2888	20.6	2374	2342	17.1	20.4	2051	1996	2241	2201	23.7	1873	1807	1759	1685	28.8	1464	1368	1344	1241
1 1/16"	2932	2910	21.1	2421	2368	17.5	20.9	2102	2026	2285	2223	24.3	1920	1832	1803	1707	29.8	1511	1395	1388	1263
1 1/4"	3079	2984	22.7	2579	2453	19.0	22.6	2273	2125	2432	2297	26.3	2077	1916	1950	1780	32.9	1672	1482	1535	1336
1 1/2"	3313	3101	25.3	2830	2627	21.2	25.4	2546	2284	2666	2414	29.4	2326	2049	2184	1897	37.9	1926	1621	1769	1453
1 3/4"	3590	3239	28.4	3127	2747	23.9	28.6	2869	2471	2943	2552	33.1	2621	2207	2461	2036	43.8	2229	1786	2047	1592
2"	3908	3398	31.9	3468	2931	27.0	32.3	3239	2686	3261	2711	37.4	2960	2388	2780	2195	50.7	2576	1977	2367	1752
2 1/4"	4268	3578	36.0	3854	3138	30.5	36.5	3660	2931	3621	2891	42.3	3343	2593	3140	2375	58.4	2968	2191	2727	1932
2 1/2"	4678	3783	40.5	4293	3374	34.5	41.1	4134	3206	4031	3096	47.7	3774	2824	3540	2575	67.1	3404	2429	3127	2132
2 3/4"	5118	4003	45.5	4765	3628	38.8	46.3	4651	3506	4471	3316	53.7	4249	3077	3990	2800	76.6	3894	2697	3577	2357
3 3/4"	7328	5108	70.1	7135	4903					6681	4421	83.5	6602	4336	6200	3905				5787	3462
4 3/4"	10218	6553								9571	5866				9090	5350				8677	4907

Eq. IV - Peak polished rod load

$$PRL = \frac{2184 \text{ lb}}{1000 \text{ ft}} \times 4000 \text{ ft.} \times 1.13 = 9872 \text{ lb.}$$

Eq. V - Required counterbalance

$$CB = \frac{1897 \text{ lb}}{1000 \text{ ft}} \times 4000 \text{ ft.} \times .962 = 7300 \text{ lb.}$$

Eq. III - Expected peak gear torque

$$T = (9872 \text{ lb} - 7300 \text{ lb}) \ \frac{42''}{2} = 54{,}012 \text{ in.-lb.}$$

These values would require a pumping unit with the following minimum specifications:

Structural capacity - 9872 lb

Required counterbalance - 7300 lb

Peak gear torque - 54,012 in.-lb

Maximum stroke - 42 in.

With the above minimum specifications as a guide, the operator is now in a position to select a pumping unit that meets his requirements. Referring to a unit manufacturer's "Unit Selection Chart," the pumping unit that more closely approximates the above specifications would have the following load capacities.

Unit structure rating	11,000 lb
Nearest available maximum counterbalance	7,465 lb
API gear reducer torque (API size 57)	57,000 in.-lb
Maximum stroke length	42 in.

It is sometimes necessary to check well loads after a well has been put into operation. The use of the dynamometer and the torque factors of the pumping unit in service is the most accurate method generally available. However, it is often necessary to make these calculations from known well conditions and without the use of a dynamometer. With the following pumping conditions, calculations can be made for the expected well loads of the present installation.

(1) Pumping depth	6000 ft.
(2) Plunger size	1-1/2"
(3) Rod string size 7/8"	1800 ft.
3/4"	4200 ft.
(4) Unit SPM	14
(5) Stroke length	64 in.
(6) Specific gravity of oil	40°API
(7) Specific gravity of water	1.2 (Brine)
(8) Percent water	20%
(9) Actual net lift of fluid	5000 ft.

TABLE 3 DEAD WEIGHT OF RODS & FLUID

ROD SIZE (IN.)	WT. OF RODS PER 1000 FT. (LB)	WEIGHT OF FLUID PER 1000 FT. (LB) PLUNGER DIAMETER (IN.)										
		1	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{3}{4}$
$\frac{5}{8}$	1137	207	251	398	632	910	1230	1590	1990	2440	4650	7540
$\frac{3}{4}$	1610	149	193	340	574	851	1170	1530	1930	2380	4590	7480
$\frac{7}{8}$	2161	80	124	271	505	782	1100	1460	1870	2310	4520	7410
1	2888	0	44	191	425	702	1020	1380	1790	2230	4440	7330

Tables 3 and 4 have been developed for use when it becomes necessary to calculate rod loads and fluid loads that are not covered by Table 1. In Table 3 the rod weights per 1000 feet are the dry weight of rods while the fluid weights per 1000 feet are the head of fluid (specific gravity 1.0) acting on the net plunger area. For a fluid with a specific gravity other than 1.0, the fluid weights per 1000 feet in Table 3 should be multiplied by the correct specific gravity to obtain the actual fluid weight per 1000 feet. Table 4 can be used to convert API gravity fluid to specific gravity. For tapered strings of rods the rod weights and fluid weights are calculated separately for each length of rod size. These rod and fluid weights are then totaled to give the total rod weight and the total fluid weight respectively.

From Table 2 across from 14 SPM and under 64 in. stroke

IF = 1.16
CF = 0.957

From Table 3 across from rods size

Wt. of 7/8" rods/1000 ft. = 2161 lb.
Wt. of 3/4" rods/1000 ft. = 1610 lb.

across from rod size and under 1-1/2" plunger size

Wt. of fluid/1000 ft. with 7/8" rods = 505 lb.
Wt. of fluid/1000 ft. with 3/4" rods = 574 lb.

From Table 4

Specific gravity of 40° API oil = 0.825

Then,

Specific gravity of mixture = 0.80 X 0.825 + 0.20 X 1.2 = 0.90

$$\text{Wt. of 7/8" rods} = \frac{2161 \text{ lb}}{1000 \text{ ft.}} \times 1800 \text{ ft.} = 3890 \text{ lb.}$$

$$\text{Wt. of 3/4" rods} = \frac{1610 \text{ lb}}{1000 \text{ ft.}} \times 4200 \text{ ft.} = 6762 \text{ lb.}$$

Total weight of rods = 3890 lb + 6762 lb = 10,652 lb.

$$\text{Wt. of fluid with 7/8" rods} = 0.90 \times \frac{505 \text{ lb}}{1000 \text{ ft.}} \times 1800 \text{ ft.} = 818 \text{ lb.}$$

$$\text{Wt. of fluid with 3/4" rods} = 0.90 \times \frac{574 \text{ lb}}{1000 \text{ ft.}} \times 3200 \text{ ft.} = 1653 \text{ lb.}$$

Total weight of fluid = 818 lb. + 1653 lb. = 2471 lb.

TABLE 4 GRAVITY CONVERSION

°A.P.I.	10	15	20	25	30	35	40	45	50	55	60	65	70
SP.GR.	1.000	.966	.934	.904	.876	.850	.825	.802	.780	.759	.739	.720	.706

SPECIFIC GRAVITY OF SATURED BRINE = 1.2

Using Equation 1, 2, and 3

$$PRL = (10{,}652 \text{ lb.} + 2471 \text{ lb.}) \times 1.16 = 15{,}223 \text{ lb.}$$

$$CB = \left(10{,}652 \text{ lb.} + \frac{2471 \text{ lb.}}{2}\right) \times 0.957 = 11{,}377 \text{ lb.}$$

$$T = (15{,}223 \text{ lb} - 11{,}377 \text{ lb.}) \times \frac{64 \text{ in.}}{2} = 123{,}072 \text{ in.-lb.}$$

It will be noted that the column of fluid with the 3/4'' rods was only 3200 feet. Since the net lift of the fluid was for 5000 feet, the fluid level was 1000 feet above the pump plunger.

CONCLUSION

Although there are many methods for sizing of beam pumping units and well load calculations, the method presented in this paper has been proved with many years of actual field experience. The values of impulse factor and counterbalance factor are empirical and were formulated from careful analysis of many actual pumping conditions. These factors adjust the calculated static well loads to the actual well loads as expected from a combination of stroke length and strokes per minute. Also these loads are reasonable for the average pumping well conditions.

The impulse factors and counterbalance factors in Table 2 apply to any well, whether single, dual or triple completion, as long as the static rod and fluid loads can be determined. For pumping wells with severe flow restrictions, such as pumping through "back pressure" valves, hollow sucker rods, etc., the fluid loads should be determined by using basic hydraulics.

The mystery of sucker rod pumping has been recognized by the industry through the formation in 1954 of the Sucker Rod Pumping Research, Inc., consisting of twenty-nine producing and manufacturing companies. A contract was let to Midwest Research Institute at Kansas City to construct a simulator to analyse the sucker rod pumping problem. From this study it is hoped that the many unknowns that enter into a given pumping cycle can be evaluated.