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Predicted vs. Measured Loads in a Sucker Rod Pumping System

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ABSTRACT

In order to apply an oilfield pumping unit on the most economical basis, it is often necessary to predict its torsional and structural loading in advance of well completion. Frequently, a pumping unit is misapplied because the determination of these values by simple prediction formulae does not conform to eventually measured loads. An investigation of the simplified prediction formulae most commonly used in applying a sucker rod pumping unit illustrates why discrepancies between predicted and measured loads may exist.

INTRODUCTION

ONE of the numerous problems faced by the pumping unit manufacturer is to predict the torsional and structural loading that will be imposed on the surface equipment for any given set of well conditions.

As a precise determination of these loads involves complex formulation employing many variables (some of which are unknown prior to well completion), it becomes necessary to develop a set of prediction equations with solutions which are fast and simple and require knowledge of only a few known quantities, such as rod and fluid weight and polished rod acceleration.

In most cases, these simple and convenient formulae, employing only a small portion of the items which actually control torsional and structural loading, are entirely adequate to apply the surface pumping unit on a most economical basis — but the results obtained are admittedly only approximations and may differ considerably from measured loads.

In a recent study of two different types of sucker rod pumping units over a number of applications chosen at random, the predicted peak torque determined by these simplified methods was compared to the measured peak torque. The results showed that, in both types of units, approximately one-third of the applications measured greater than predicted, one-third measured less than predicted and one-third measured about the same as predicted. (See Table I).

In a sucker rod system, peak and minimum polished rod loads and consequently counterbalance, peak torque and horsepower are complex functions of many

TABLE I

Dr-No.	Unit Size	Calculated Torque	Measured Torque	Calc. Less Than Meas. by	Approx. Equal	Calc. More Than Meas. by
Type A Units						
017	M-160D-200-74	127,000	182,500	30%	—	—
019	M-160D-200-86	184,000	98,400	—	—	50%
026	M-114D-143-86	84,700	54,800	—	—	55%
036	M-228D-256-100	175,200	187,500	—	X	—
038	M-114D-143-74	112,600	59,800	—	—	88%
043	M-160D-173-74	53,800	80,700	33%	—	—
044	M-160D-200-74	142,000	112,000	—	—	27%
045	M-160D-200-74	92,000	102,340	—	X	—
049	M-228D-256-120	176,200	255,000	31%	—	—
050	M-640D-304-144	532,500	451,000	—	—	18%
053	M-320D-304-120	204,000	279,000	27%	—	—
054	M-320D-304-120	243,000	290,000	16%	—	—
057	M-228D-173-74	208,500	177,000	—	—	17%
059	M-160D-200-74	132,800	128,500	—	X	—
062	M-114D-143-74	162,000	73,600	—	—	120%
063	M-160D-200-86	112,800	125,100	—	X	—
074	M-456D-256-120	398,000	495,000	20%	—	—
079	M-228D-256-120	157,000	315,000	50%	—	—
081	M-228D-256-120	194,000	251,000	23%	—	—
082	M-160D-200-86	116,800	128,000	—	X	—
106	M-160D-246-74	90,900	102,000	—	X	—
142	M-456D-304-120	347,000	352,000	—	X	—
Type B Units						
003	C-320D-120-25	357,000	502,500	29%	—	—
004	C-228D-74-17.3	203,000	334,000	39%	—	—
007	C-320D-120-25	224,000	325,000	31%	—	—
009	C-114D-54RO-14	95,300	128,500	26%	—	—
014	C-320D-120-25	283,000	293,000	—	X	—
020	C-114D-64-16.1	39,100	47,100	17%	—	—
039	C-160D-173-74	149,000	101,800	—	—	46%
040	C-114D-64-14.3	92,000	110,400	17%	—	—
041	C-114D-64-14.3	88,900	116,100	24%	—	—
058	C-228D-64-20	143,000	168,500	—	—	15%
060	C-160D-174-74	198,500	133,300	—	—	49%
064	C-320D-84-30	355,000	372,000	—	X	—
065	TC-OALTR-41D	238,000	224,000	—	X	—
066	TC-OALTR-41D	310,000	270,000	—	X	—
067	TC-OALTR-41D	286,000	229,000	—	—	25%
080	C-160D-169-64	147,200	143,000	—	X	—
083	C-320D-120-25	210,500	140,000	—	—	50%
095	C-114D-64-14.3	94,200	105,500	—	X	—
099	C-160D-64-15.8	182,900	146,000	—	—	25%
104	C-160D-74-16.4	131,000	173,000	24%	—	—
108	C-160D-173-74	120,600	170,500	29%	—	—
115	C-320D-256-100	390,000	352,000	—	X	—

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downhole variables, some of which are: (1) rod weight, (2) fluid weight, (3) polished rod acceleration pattern, (4) viscosity and composition of well fluid, (5) rod-to-tubing friction, (6) subsurface pumping friction, (7) stuffing box friction, (8) fluid-to-tubing friction, (9) rod-to-fluid friction, (10) casing pressure, (11) pump submergence, (12) reservoir energy, (13) flowline back pressure, (14) fluid compressability, (15) abnormal well conditions such as fluid pound, striking up and down, (16) position of travelling valve on opening and closing, and (17) rod dynamics. The latter depends upon: (a) damping factors, (b) unit geometry, (c) spring constant of rods and tubing, (d) ratio of forced vibration to the natural frequency of the rod string, (e) pumping speed and stroke length, (f) variation of angular velocity of the cranks, (g) system inertia, (h) motor slippage, if electrically driven, (i) prime mover inertia, etc.

Obviously, any formula* for determining pumping unit loads that does not recognize and include *all* of the above variables will give only approximate results.

Fortunately, the three major known variables — weight of rods and fluid, and polished rod acceleration — are usually sufficient for predicting approximate torsional and structural loading so that, in most cases, pumping units can be effectively applied by simple and convenient formulation. There are, however, exceptions to this rule where unknown and/or unconsidered variables become relatively important.

For instance, it is not uncommon to have a rod harmonic contribute as much to peak polished rod loading as the accelerative force component itself. This is why many operators attempt to run a pumping unit at a non-synchronous pumping speed. Occasionally, combined friction forces influence rod loading nearly as much as fluid weight.

One of the reasons that these simplified prediction formulae have been so widely used by the industry during the past twenty-five years is because, frequently, many of the unconsidered variables are opposing and tend to cancel each other, which occasionally may make predicted loads conform closely to measured loads. For instance, during the upstroke, pump, rod and stuffing box friction are opposed by pump submergence forces and casing pressure. Consequently, when a measured load closely confirms predictions, it probably means that most of the unconsidered variables were canceling, rather than indicating that the prediction formula was particularly accurate.

The four basic prediction formulae most commonly used for sizing sucker rod pumping units are (1) the standard Mills equation for establishing peak polished rod loads, (2) the standard minimum load equation, which employs rod weight and acceleration, (3) a counterbalance formula which averages peak and minimum loads, and (4) the regular peak torque formula which uses peak and minimum loads along with their accompanying maximum up and down stroke torque factors.

*There are load equations which utilize not all, but more than the three basic items generally considered, and which give better results, but these formula are laborious to apply and, in many cases, their increased precision is not worth the extra effort required to solve them. These load formulae are often so complicated that it is necessary to use electronic computers for their application.

These four simplified formulae* are:

Peak Pol. Rod Load (P) = Wt. of rods (1 + Acceleration) + Wt. of fluid

or $$P = W_r (1 + K) + W_f \quad (1)$$

Minimum Pol. Rod Load (M) = (Wt. of Rods) (.873 − Acceleration)

or $$M = (W_r)(.873 - k) \quad (2)$$

$$\text{Counterbalance} = \frac{\text{Peak Pol. Rod Load} + \text{Min. Pol. Rod Load}}{2}$$

or $$CB = \frac{P + M}{2} \quad (3)$$

Peak Torque = ½[P x Torque Factor (Max. Up) − M x Torque Factor (Max. Down)

or $$PT = \tfrac{1}{2}(P \times TF_1 - M \times TF_2) \quad (4)$$

Individual analysis of these formulae should help illuminate and clarify the problem of reconciling measured and predicted loads in a sucker rod system.

Peak Polished Rod Load Formula

The most commonly used formula for predicting peak polished rod loading is the Mills formula:

$$P = W_r (1 + K) + W_f \quad (1)$$

where:

W_r = total rod weight
W_f = total fluid weight
K = maximum off-bottom acceleration – feet per second per second, divided by the gravitational constant, 32.2 – assuming that the cranks are turning with constant angular velocity and the polished rod is reciprocating with simple harmonic motion.

A significant discrepancy in this formula usually comes as a result of employing only three of the seventeen or so variables which control peak polished rod load.

This equation assumes that the fluid slug lifted with each stroke is not accelerated. This assumption may be valid in deeper holes, but, where large volumes are being elevated from shallow wells, it could be necessary to consider acceleration of both rods and fluid column. Failure to do this may create a substantial discrepancy between measured and predicted loads.

Differences in the time sequence in which the fluid mass and the rod mass are subjected to their maximum accelerative forces also contribute to incorrect prediction results. In a shallow well, rods and fluid are accelerated almost simultaneously; in deeper wells, acceleration of the rod mass precedes acceleration of the fluids.

Assumption of simple harmonic motion at the crankshaft is incorrect, because class I units (conventional) come off bottom stroke with an acceleration considerably greater than that of simple harmonic motion; class III systems (Mark II and air balance), on the other hand, come off bottom with an acceleration considerably smaller than that of simple harmonic motion.

A substantial error may result from predicting peak polished rod loading and assuming that there is no stuffing box friction, rod-to-tubing friction, fluid-to-tubing friction, rod-to-fluid friction or pump friction. Studies in shallow wells have shown that a

*These four equations may be used to predict torsional and structural loading for any beam-type pumping unit — conventional, air balanced, Mark II and special new geometries.

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tight stuffing box may account for nearly 40 per cent of the peak polished rod load. As far as the pumping unit is concerned, it cannot distinguish between the force necessary to accelerate the rods and fluid, or that required to overcome friction. Consequently, assuming frictionless well conditions will frequently lead to appreciable errors in predicting polished rod loads.

The formulation assumes that tubing pressure at the well head is zero. Commonly, however, there is a flowline back pressure which will tend to elevate peak polished rod loads.

Another important factor, overlooked in simple formulation, is the elastic and compressive nature of the rod and fluid loads and the harmonic forces produced in reciprocating the rod string. It is possible to pump in such a manner as to make the stress waves additive or subtractive to the force required to drive the rods. Simple formulation is not capable of properly accounting for the phenomenon of stress wave behaviour in the rod string.

Assuming incorrect acceleration of the rods, no acceleration of the fluid, a frictionless pumping system, a disregard of stress wave phenomena, and the omission of many other important factors all combine to make the prediction of peak polished rod load by simple formulation a hit-and-miss proposition. Without canceling errors and conservative pumping unit design, this formula would perhaps have been inadequate even as an approximation.

Minimum Polished Rod Load Formula

The formula for predicting minimum polished rod load is:

$$M = W_r (.873 - k) \qquad (2)$$

where:
.873 = total buoyant rod weight
k = maximum off-top acceleration — feet per second per second divided by the gravitational constant, 32.2 — assuming that the cranks are turning with constant angular velocity and the polished rod is reciprocating with simple harmonic motion.

Like formula (1) above, one of the greatest discrepancies in predicting minimum polished rod load results from accounting for only a small portion of the variables which actually control it.

This equation also assumes simple harmonic motion at the polished rod and constant angular velocity at the cranks, neither of which occurs during actual operation. Simple harmonic motion is not generated unless the pitman-to-crank ratio is infinitely large; in practice, however, it is approximately 3 to 1, and the cranks never turn uniformly.

A heavy fluid pound will also tend to create errors in predicting minimum polished rod loads. The formula assumes that the fluid load is transferred to the tubing at the top of the stroke, where polished rod velocity is zero. If it occurs at any other position a fluid pound will result, and the minimum load may be lower than formulated.

In this equation also, a substantial error arises in not accounting for friction. Frequently, a crooked hole or tight stuffing box will retard rod fall as much or more than the actual buoyancy of the fluid. Any force retarding the fall of the rod string, such as viscous fluid, parafin, etc., will create an artificially low minimum which is not accounted for by regular formulation. Thus, assuming frictionless well conditions leads to further errors in predicting minimum as well as maximum loads.

Counterbalance Load Formula

The common counterbalance relationship is:

$$CB = \frac{P + M}{2} \qquad (3)$$

where:
P = peak polished rod load
M = minimum polished rod load

Obviously any discrepancies (other than offsetting) in either peak or minimum predictions will create errors in the amount of counterbalance required. It should be recognized that one of the most common problems encountered by the surface unit is operating with too much or too little counterbalance. Either one can create serious gear reducer overload problems.

It is important to recognize that no surface pumping unit can be properly counterbalanced if the unit is pounding fluid. Although it is possible to equate torque peaks, this abnormal pumping condition still makes proper counterbalancing impractical if not impossible. When a fluid pound occurs and the well load approaches a sub-minimum value, the unbalanced counterweight torque is thrown on the gear reducer, producing a serious load problem and an unnecessarily large negative torque.

Peak Torque Formula

The general peak torque formula and its assumptions are shown below:

$$PT = \tfrac{1}{2}(P \times TF_1 - M \times TF_2) \qquad (4)$$

where:
PT = Peak Torque
P = Peak Polished Rod Load
TF_1 = Maximum Upstroke Torque Factor
TF_2 = Maximum Downstroke Torque Factor
M = Minimum Polished Rod Load

Assumptions:
1. Peak net upstroke torque = peak net dowsntroke torque.
2. Maximum upstroke counterbalance torque occurs at mid-upstroke position along with maximum upstroke well load torque.
3. Maximum downstroke counterbalance torque occurs at approximately mid-downstroke and at the same position as maximum downstroke well load torque.
4. No pounding or other abnormal well condition occurs.

This is the general net torque formula (4), which is valid for any type of *symmetrical or non-symmetrical* beam-type unit geometry. For symmetrical geometry, where the maximum upstroke torque factor is equal to the maximum downstroke torque factor, with both occurring at about midstroke position, and with the torque factors approximately equal to half the stroke length, the torque formula may be simplified to a special form:

$$PT = \frac{S}{2}(P - CB) \qquad (5)$$

where: PT = Peak Torque, in.-lbs.
S = Stroke length, ins.
CB = Counterbalance, lbs.
P = Peak Polished Rod Load, lbs.

It should be recognized that formula (4) and formula (5) give identical results, but the general formula can account for unequal up- and down-stroke torque factors, whereas the special formula assumes both maximum torque factors to be the same.

Determining peak torque from the general equation (4) or the special equation (5) involves two sep-

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arate and distinct operations. *First,* determination of peak and minimum polished rod load by the Mills equation; *second,* insertion of these peak and minimum values into one of the two peak torque formulae*. Consequently, errors in predicted peak torque may arise in several ways: (a) incorrect peak load prediction, and/or (b) incorrect minimum load prediction, and/or (c) a shift in phase between well load and counterbalance peak torque, and (d) differences in unit geometry.

(A.) Incorrect Peak Load Predictions

An operator wishes to apply beam-type pumping units to three similar 5,000-ft. adjacent wells — "A", "B" and "C" — by pumping 12-100-in. SPM with a 1½-in. pump and ⅞- to ¾-in. standard taper rods. In *predicting* the size of pumping unit required for each of these wells, the loads shown in Table II were determined.

Table II

PREDICTED LOADS	
1. Peak polished rod load	13,620 lbs.
2. Minimum polished rod load	6,040 lbs.
3. Counterbalance required	9,830 lbs.
4. Peak torque at 93% mechanical efficiency	212,000 in.-lbs.

Well "A"

On the basis of these predictions, a 228,000-in.-lb. unit was selected and purchased for this well.

On several occasions during the drilling of well "A", "doglegs" were noted, which made the operator suspect that a high friction component might be encountered when the well was eventually pumped.

Shortly after installing the 228,000-in.-lb. unit, the operator made a complete dynamometer analysis to see if the unit was properly applied. *Measured* dynamometer loads were then compared to *predicted* loads, as shown in Table III.

Table III

	PREDICTED LOADS	MEASURED LOADS
1. Peak polished rod load	13,620 lbs.	15,957 lbs.
2. Minimum polished rod load	6,040 lbs.	5,965 lbs.
3. Counterbalance	9,830 lbs.	11,238 lbs.
4. Peak torque at 93% mechanical efficiency	212,000 in.-lbs.	285,000 in.-lbs.*

*Determined by the torque factor method, employing measured loads.

Apparently, the operator's suspicion of a high friction component was realized, for the predicted peak polished rod load of 13,620 lbs. was substantially exceeded by the measured peak polished rod load of 15,957 lbs. Also, the predicted peak torque of 212,000 in.-lbs. was considerably less than the 285,000 in.-lbs. measured. Obviously, the gear reducer was appreciably overloaded.

*These equations assume zero phase shift between peak well load torque and peak counterbalence torque.

What caused these errors in predicted loads?

Investigation of the polished rod dynamometer card showed that a marked frictional component did exist during the upstroke, probably caused by a crooked hole. In addition to the predicted 13,620 lbs. of force required to lift rods and fluid, an additional 2,337 lbs. (15,957 — 13,620) was also required to overcome excessive friction. This particular discrepancy was unimportant structurally, because the chosen unit had adequate capacity to handle either load.

The differences between predicted and measured peak torque, however, were of concern to both operator and manufacturer, because the 228,000-in.-lb. unit recommended was overloaded torsionally.

Was this torsional error due to a discrepancy in predicting incorrect peak and minimum polished rod loads or was it because of an incorrect torque formula? By inserting the measured peak and minimum loads into the torque formula, it was found that the resultant net torque was 279,000 in.-lbs. — very close to the measured load of 285,000 in.-lbs.

Thus, the torsional inaccuracy was not due to limitations in the torque equation, but was due to the errors in predicting peak and minimum loads when using simple formulation.

(B) Incorrect Minimum Load Predictions

Well 'B"

The predicted loads for both wells "A" and "B" were the same, and identical 228,000-in.-lb. units were selected for both applications. After the 228,000-in.-lb. unit was installed on well "B", the operator made a complete dynamometer study to see how closely predicted loads conformed to measured loads.

The only unusual condition in the well was a substantial fluid pound which occurred about halfway through the downstroke, creating a sub-normally low minimum polished rod load. The *measured* dynamometer loads were then compared to the *predicted* loads, as shown in Table IV.

Table IV

	PREDICTED LOADS	MEASURED LOADS
1. Peak polished rod load	13,620 lbs.	13,826 lbs.
2. Minimum polished rod load	6,040 lbs.	5,030 lbs.
3. Counterbalance	9,830 lbs.	9,428 lbs.
4. Peak Torque at 93% mechanical efficiency	212,000 in.-lbs.	257,000 in.-lbs.*

*Determined from the torque factor method, employing measured loads.

The measured peak polished rod load shown in the table closely conforms to predictions. However, there is more than a 1,000-lb. discrepancy in the minimum load.

Also, a marked discrepancy exists between measured (257,000 in.-lbs.) and predicted (212,000 in.-lbs.) peak torque loading. This was a significant error, as it changed the size requirement of the pumping unit gear reducer.

Once more the question is raised — was the torsional error due to a discrepancy in the predicted minimum polished rod load or was it because of an incorrect torque formula?

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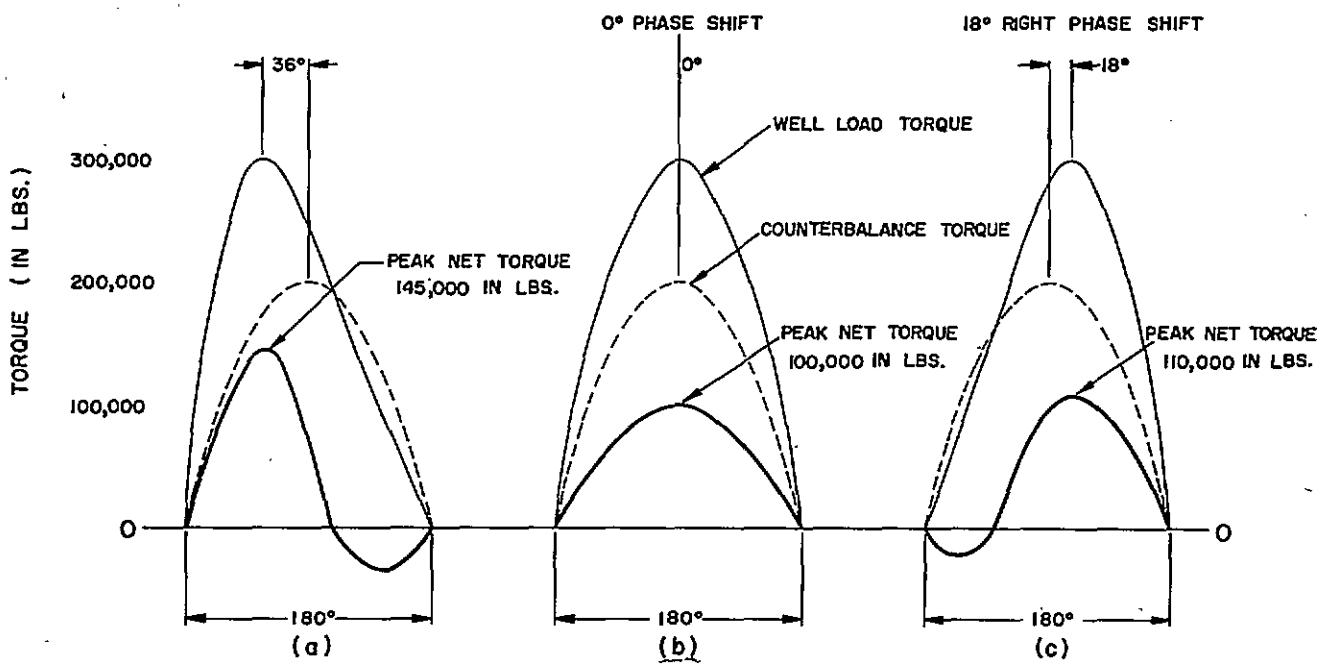

Figure 1.—Effect of Phase Shift on Peak Net Torque.

(Peak well load torque, 300,000 in.-lbs.; peak counterbalance torque, 200,000 in.-lbs.)

By inserting the measured peak and minimum loads into the general torque formula it was found that the resultant peak net torque was 246,000 in.-lbs., which was close to the measured load of 257,000 in.-lbs. Thus, the torsional inaccuracy occurred not because of limitations in the torque equation but because of an error in predicting minimum polished rod load.

In this case, the incorrectly predicted minimum load occurred because of a fluid pound, which is considered an abnormal pumping condition and is not accounted for in simple, unit application equations.

(C) Phase Shift Between Well Load & Counterbalance Torque

Well "C"

In Well "C", on the basis of predicted loads, a 228,000-in.-lb. unit was selected, and was operated under the conditions previously assumed. A dynamometer study was then made of the well. Both the *predicted* and *measured* loads are shown in Table V.

Table V

	PREDICTED LOADS	MEASURED LOADS
1. Peak polished rod load .	13,865 lbs.	13,507 lbs.
2. Minimum polished rod load.....	6,118 lbs.	6,005 lbs.
3. Counterbalance.....	9,992 lbs.	10,425 lbs.
4. Peak Torque at 93% mechanical efficiency .	210,557 in.-lbs.	268,712 in.-lbs.*

*Torque determined by the torque factor method, using measured loads.

Investigation shows that the predicted peak and minimum polished rod loads agree closely with the measured ones, but a sizable error exists in the predicted peak torque.

Indications are that the measured peak well load torque and the peak counterbalance torques are severely out of phase. Generally, this is because pumping speed, stroke length, rod string stiffness with relation to the fluid load and well depth combine to produce a peak polished rod load that, when multiplied by its accompanying torque factor, produces a peak well load torque occurring considerably before or after the position of the peak counterbalance.

It should be recognized that simplified formulation assumes that peak well load and counterbalance torque both occur simultaneously, and, if some condition interferes with this phasing, a substantial discrepancy between predicted and measured peak torque may arise.

Figure 1 illustrates the effect of dephasing on peak net torque. In each of the three examples — (A), (B) and (C) — the value of the well load torque is 300,000 in.-lbs., and the magnitude of the peak counterbalance torque is 200,000 in.-lbs. In (A), due to a 36-degree-left phase shift in the well load torque, the peak net torque is approximately 145,000 in.-lbs. In (B), with no phase shift, the peak net torque is approximately 100,000 in.-lbs., and in (C), with an 18-degree-right phase shift in well load torque, the peak net torque is 110,000 in.-lbs. Thus, in this typical illustration, an out-of-phase condition has increased the net peak torque by 50 per cent — although the value of both the peak well load and the counterbalance torque has remained unchanged.

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On a 12,000-ft. well in California, where the pumping unit was operating at approximately 4½-144-in. SPM, the elapsed time for the primary stress wave to reach the surface was great enough so that the peak polished rod load occurred at a high torque factor position. Because of this delayed stress wave, the position of the peak load multiplied by its maximum torque factor did not occur close to the position of the maximum counterbalance torque. This difference in the peak position of well and counterbalance torque produced a substantial net torsional load error that could not be accounted for by the simplified prediction calculations.

From the above, it can be seen that errors in predicting peak torque can arise from one or a combination of three things — incorrectly predicted peak and minimum polished rod load, or a substantial phase shift between well load and counterbalance peak torque.

Good operating practice dictates that pumping unit conditions should insure minimum phase shift between well load and counterbalance torque. This can be done by varying some combination of the stroke length, pumping speed, plunger diameter, unit geometry and the relationship of the fluid load to the spring constant of the rod string.

(D) Influence of Pumping Unit Geometry on Predicted Load Calculations

There are two general types of pumping unit geometries: (1) the conventional unit, which is designed as a Class I lever system, and (2) the air balance and Mark II units, which are designed as Class III lever systems. Because of a substantial difference in position of the torque factor peaks, different dynagraph shapes are favoured by these two different geometries.

In general, the Class I lever system favours dynagraph profiles sloping *upward* to the right, and the Class III geometry favours a dynagraph sloping *downward* to the right.

In *Figure 2*, dynagraphs X and Y from two different wells are shown. Both have the same measured peak and minimum loads, and simplified prediction formulae would show that their resultant peak torques are identical. However, a rigorous measured torque factor analysis of these cards shows that, for a particular type of pumping unit geometry, one card produces a peak net torque of 200,000 in.-lbs. while the other card produces a peak net torque of 100,000 in.-lbs. For another type of geometry, the reverse would be true. In both of the above cards, the position of the peak polished rod load produces a substantial phase shift between well load and counterbalance torque for one geometry, and little or no phase shift for another.

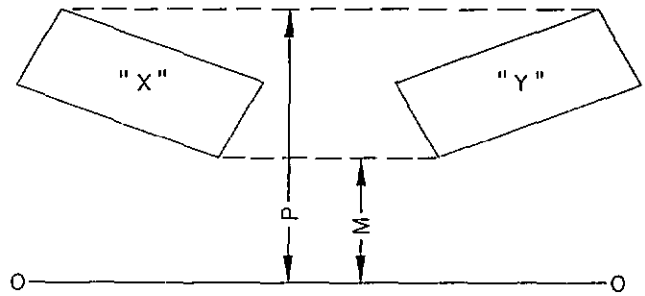

Figure 2.—Typical Over-travel and Under-travel Dynamometer Cards.

When one of these two geometries handles its favoured card shape, little or no torsional dephasing occurs, and the simplified torque formulae are approximately as accurate as the ability to predict peak and minimum polished rod loads. However, when a particular geometry is handling its non-favoured dynagraph shape, a severe phase shift may result and, if predicted peak and minimum polished rod loads conform closely to their measured values, a sizable discrepancy will exist between predicted and measured peak torque. Frequently, a dephasing error may offset a discrepancy in predicting peak and minimum polished rod loads.

In most cases, however, a phase shift between peak well load and counterbalance torque will result in a substantial difference between predicted and measured peak torques.

Conclusions

An attempt has been made to point out some of the difficulties encountered in trying to predict exact complex quantities using simplified and primitive approximations. Much study has recently been done toward improving pumping unit application formulae and, in the near future, it should be possible to apply pumping units using computer-generated curves which account for rod dynamics and many other effects as well as for the primary variables now considered. Until this time, however, these simple prediction equations that have served for so long, when used with judgment and discretion, can be counted on to assist the industry a little while longer in the important task of applying oilfield pumping units on a most economical basis.

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APPENDIX

Torque Equations, Beam-Type Pumping Units

(Refer to Figure 3)

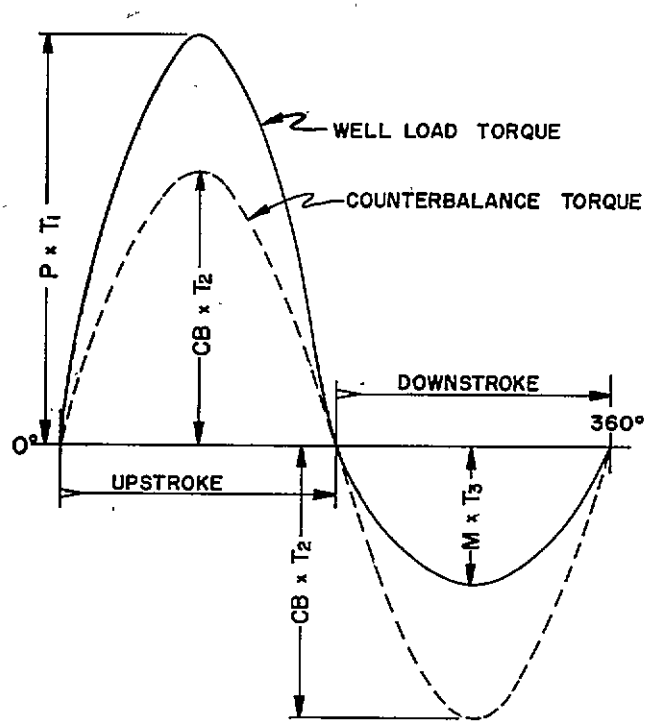

Figure 3.—Net Torque Diagram.

Where:

P	= Peak Load
CB	= Counterbalance
M	= Minimum Load
PTN_{up}	= Peak Net Torque, Upstroke
PTN_{down}	= Peak Net Torque, Downstroke
T_1	= Maximum Well Load Torque Factor, Upstroke
T_2	= Maximum CB Torque Factor
T_3	= Maximum Well Load Torque Factor, Down Stroke

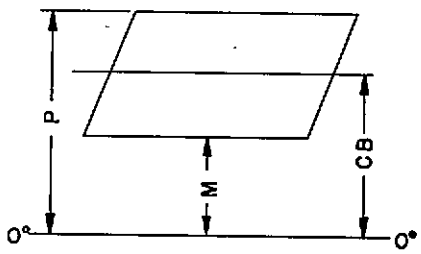

Figure 4.—Typical Dynamometer Card.

*General Net Torque Formula — All Beam-Type Geometries**

Assumptions:

1. Peak net upstroke torque (PNT_{up}) equals peak net downstroke torque (PNT_{down}).
2. Maximum upstroke counterbalance torque ($CB \times T_2$) occurs approximately at midstroke position.
3. Maximum upstroke well load torque ($P \times T_1$) occurs at approximately the same position as maximum counterbalance torque ($CB \times T_2$).
4. Maximum downstroke counterbalance torque ($CB \times T_2$) occurs at approximately mid-downstroke.
5. Maximum downstroke well load torque ($M \times T_3$) occurs at approximately the same position as the maximum downstroke counterbalance torque ($CB \times T_2$).
6. No pound or other abnormal well conditions occur.

Peak Net Torque Upstroke = Peak Net Torque Downstroke

$$PNT_{up} = P \times T_1 - CB \times T_2 \quad (1)$$
$$PNT_{down} = CB \times T_2 - M \times T_3 \quad (2)$$

Add (1) and (2)

$$2 \times PNT = P \times T_1 - M \times T_3 \quad (3)$$
$$PNT = \tfrac{1}{2}(P \times T_1 - M \times T_3) \quad (4)$$

*Special Net Torque Formula — Symmetrical Geometries**

Assumptions: Same as above, plus

7. Maximum upstroke torque factor (T_1) equals maximum downstroke torque factor (T_3). As $T_1 = T_3$, let $T_1 = T_3 = TF$
8. Maximum torque factor is approximately one-half the stroke length divided by

$$.93, \text{ or } \frac{S}{2 \times .93}$$

From *Figure 4:*

$$\frac{P + M}{2} = CB, \text{ or } P + M = 2\,CB \quad (5)$$
$$M = 2\,CB - P \quad (6)$$

Substitute (6) in General Torque Formula (4) above

Then:

$$PNT = \tfrac{1}{2}[P \times T_1 - (2\,CB - P) \times T_2] \quad (7)$$

but:

$$T_1 = T_2 = TF$$
$$PNT = \tfrac{1}{2}[P \times TF - (2\,CB - P)\,TF] \quad (8)$$
$$= P \times TF - CB\,TF \quad (9)$$
$$= P - CB\,(TF) \quad (10)$$

Substituting

$$TF = \frac{S}{2 \times .93} \text{ in (10)}$$

$$PNT = P - CB\left(\frac{S}{2 \times .93}\right) \quad (11)$$

*At 100% mechanical efficiency

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