BEST METHOD TO BALANCE TORQUE OF A PUMPING UNIT GEARBOX

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ABSTRACT

There are three methods available to the operator to determine the net torque loading of a pumping unit's gearbox. Two dynamic methods determine the instantaneous torque throughout the pumping cycle: method 1) uses measured motor power, motor and drive efficiencies and the pumping unit speed to determine gearbox torque and method 2) Combines the measured surface dynamometer card and calculated torque factors together with measured or calculated counterbalance moments from the crank and weights. Performing a counter balance effect, CBE, test is a direct method of determining net gearbox torque at a specific crank position to estimate the counterbalance moment; this static test is where the cranks and counter weights are held level until no upward or downward movement is noticed when the break is released. Field case studies of applying all three methods to determining gearbox torque are presented in this paper. The pros and cons of using each method are discussed.

INTRODUCTION

The oldest and most common method of artificial lift used for producing oil wells is sucker rod pumping. In the United States sucker rod lift is used in over 85% of artificial lift wells. The sucker rod lift system is made up of four components 1) prime mover, 2) pumping unit, *3)* counterbalance to the rod loading, and 4) sucker rods and associated downhole equipment. The function of the pumping unit is to change the rotating motion of the prime mover into the vertical up and down linear pumping motion at the polished rod. The role of the prime mover is to furnish the necessary power to drive the system.

In a sucker rod pumping system, the polished rod work needed to lift the fluid column is required only during the upstroke. If the sucker rod load on the surface pumping equipment were not counterbalanced, then the total work required from the prime mover would be performed during the upstroke lifting the buoyed sucker rod load and fluid load. During the downstroke the prime mover would not be doing any work, while the force of gravity pulls the rods and pump back down to the bottom of the stroke. Operating in this inefficient manner would require an extremely powerful prime mover and gearbox. For this inefficient system, the uncounterbalanced sucker rod load would determine the torque on the gearbox. To improve the efficiency and to reduce the size of the prime mover and gearbox, plus to load the gearbox more uniformly, the sucker rod pumping system is furnished with some type of counterbalance system, where the counterbalance effect at the polished rod is approximately equal to the buoyant weight of the rods plus half the weight of the fluid. The "correct" counterbalance required to balance the loads on the gearbox is approximately equal to the weight of the rods floating in fluid plus ½ the fluid load.

BALANCED OR UNBALANCED GEARBOX LOADING

For each complete stroke the net torque load on the gearbox is cyclic, usually having two maximum peaks and two minimum valleys. The height of the peaks should be approximately equal for balanced operation. Pumping unit manufacturers use various types of counterbalancing and mechanical features to reduce the peak gearbox torques and to smooth out the cyclic effects of the load.

The pumping unit gearbox is underbalanced or rod heavy if the upstroke peak is greater. The pumping unit gearbox is overbalanced or weight heavy if the downstroke peak is greater. The net gearbox torque loading on the pumping unit is balanced if the peak upstroke torque is equal to the peak downstroke torque. **Fig. 1** shows typical torque (in-lbs) or power (kW) signatures of a pumping unit. If the net gearbox torque on a pumping unit is maintained in balance, then the peak torques imposed on the motor, the peak power delivered by the motor, and the peak current drawn by the motor are reduced, thus reducing the power cost.

As shown by **Fig. 2** the motor torque behaves in much the same way as the net mechanical torque on the gearbox, both motor output torque and gearbox torque have a peak (a maximum) and a valley (minimum) on the upstroke and downstroke portion of the pumping cycle. The relation of the "left" peak to the "right" peak determines whether the pumping unit is mechanically/electrically balanced or unbalanced, and whether the pumping unit is rod heavy (underbalanced) or weight heavy (overbalanced). In **Fig. 2** both the power and mechanical torque data show that this unit is weight heavy (overbalanced) and the counter weights should be moved in from the end of the crank to balance the peak torques. The net mechanical torque applied to the gearbox is derived from the measured polished rod loads and the output motor torque is derived from the power input to the motor. The relationship of the gearbox torque to the output motor torque shows the peaks and valleys of each parameter track closely. Also notice that although the pumping unit is electrically and mechanically unbalanced, when the gearbox torque exhibits negativity torque so does the motor. The negative motor torque is caused by the combination of moments from the various components (including cranks, counter-weights, beam, and rod loadings) driving the motor past its synchronous speed. When negative motor torque is displayed , then the motor is operating in a "regenerative horsepower" mode during this part of the stroke. The statement that "What the gearbox demands, the motor provides" is fairly obvious upon inspection of the **Fig. 2.**

PRIME MOVER

An important consideration in a pumping installation is the prime mover, electric motors and internal combustion engines are the two basic types in widespread use today. The selection of one type of prime mover over another depends upon local availability, fuel supply, local conditions, availability of maintenance, and personal experience or preference. The main advantages of electric motors over gas engines are their lower initial cost and lower maintenance costs. Electric motors also provide dependable all weather service and can be more easily fitted into an automatic system. The initial cost to electrify a well site can be large, if the well location is a long distance from electric power providers and the operator must pay the cost to set the power poles and run the wires. Gas engines have the advantage that they can be operated using produced lease gas for fuel. Additionally, gas engines have the advantage of more flexible speed control and can operate over a wider range of load conditions. Fuel costs for gas engines may be lower than comparable energy costs for electric motors, although as fuel power costs increase this condition may be reversed.

The selection of the type of prime mover has a direct impact on the method used to balance the torque loading on the gearbox. Only mechanical means can be used to balance gearbox torque when the prime mover is a gas engines. If the prime mover selected is an electric motor, then method 1) using measured motor power with motor and drive efficiencies and the pumping unit speed can be used to determine net gearbox torque. Inspection of the collected power data can be used to immediately determine, if the pumping unit system is balanced.

Power provided to the motor and the net torque resulting at the gearbox are directly proportional. Measurement of power using the power-current transducer during the pumping cycle is a quick and simple process. The power-current transducer consists of three voltage probes that are attached to the three wires to the motor. Two current transducers are installed around two of the three wires that power the motor. The power-current transducer assembly is compact and wires in the panel do not need to be removed or changed in any way to install the power-current transducer. Using power probes connected to the panel to directly measure electric power as a function of time, then software can use **Eq. 1** to calculate net torque at the gearbox during a pump stroke from the motor power data.'

$$T_N = 84.5 \times kW \times Eff / (SPM * SV) \qquad (1)$$

The power requirement on the upstroke should be balanced against the power requirement on the downstroke for more efficient operations. To balance the pumping unit the operator does not have to know the pumping unit API dimensions, weight of counterbalance, or center of gravities; all that is needed, is to know is the weight of the counterbalance that must be moved.

PUMPING UNIT GEOMETRY

The API dimensions are used to calculate torque factors at crank angles throughout one complete stroke and the torque factors are used to convert the measured polished rod load into torque at the gearbox. API dimensions, A, C, I, K, P and R, are shown in **Fig. 3** for a C-320D-256-100 conventional pumping unit. In the API description of the pumping unit, the prefix C indicates the type or API class of the pumping unit; 320 is the gearbox capacity in thousands of inch-lbs.; 256 is the rated beam load capacity in 100 lbs.; and 100 is the maximum stroke length for the pumping unit. In modern software programs used to calculate net gearbox torque, the pumping unit API dimensions are stored in a database and automatically get loaded when the user of the software selects the pumping unit manufacturer and API description. Generally conventional pumping units may rotate clockwise, CW, or counterclockwise, CCW, but other types of pumping units, gearbox lubrication requirements or specific manufacture requirements will require a certain direction of rotation. The direction of rotation, CW or CCW, of the crank is defined with wellhead to right of the gearbox, when the observer is looking towards the gearbox. Usually pumping units do not usually have symmetric rotation around the gearbox and the direction of rotation is important because the torque factors are different on the upstroke and downstroke for the same polished rod position.

POLISHED ROD LOADING

A Dynamometer is a device that measures the polished rod load applied to the pumping unit at increments of position over one complete stroke. Usually mounting a dynamometer between the polished rod clamp and the carrier bar makes this measurement, so that the entire weight of the rod string can be measured[2]. The accuracy of the measured load depends on the load cell used to acquire the data. Most portable load cells are calibrated and can accurately measure polished rod loads. Some error in the measured loads can be introduced while placing the portable loadcell between the carrier bar and polished rod clamp, if the carrier bar and clamp do not consistently contact the loadcell. Hydraulic[3] type load cells with a piston, usually have hysterises type of friction that causes an offset and drifting of the loads. Permanently mounted donut type loadcells are very accurate in measuring change in load, but a significant offset in the load can be developed due to overloading and the load cell becoming permanently deformed. The donut load cell should have a spacer and spherical washers to ensure the polished rod load is centralized on the load cell. **A** surface dynamometer card[4] is the plot of the measured rod loads at the various positions throughout a complete stroke; normally load is displayed in pounds of force and position is displayed in inches. Dynamometer cards are displayed by commercial diagnostic software for the purposes of determining rod loading and for torque calculations.

GEARBOX TORQUE FROM THE SURFACE DYNAMOMETER CARD

The standard method for determining the instantaneous torque throughout the pumping cycle uses torque factors and polished rod position data together with counterbalance moments as defined by standard API Spec 11E. This method is used in the Total Well Management[5,6,7], TWM software. **Table 1** is an example net torque calculations using the torque factors (TF) derived from the geometry of a Lufkin C-320D-256-100 pumping unit and are printed out for each 15-degree position of the crank. The instantaneous torque due to net well load at a given crank position is the torque factor at that position multiplied by the net well load at that position.

Net well load is:

$$\mathbf{W_, = net\ well\ load = (W - SU)} \qquad (2)$$

Torque due to net well load is:

$$\mathbf{T_{WN} = TF \times W_N} \qquad (3)$$

Torque due to crank and counterweights is:

$$\mathbf{T_{CN} = Me \times \sin(q + t)} \qquad (4)$$

The net torque gearbox torque, **T,,** about the crankshaft is the difference between the torque due to net well load and the torque due to the counterbalance moment of the crank and counterweights:

$$\mathbf{T_N = TF \times W_N - Me \times sine\ (q + t)} \qquad (5)$$

Fig. 4 plots torque due to net well load, torque due to crank and counterweights, and the net torque gearbox torque. Torque factors are positive (+) on the upstroke and negative (-) on the downstroke. The reference position for the beginning of the upstroke (12-o'clock or 6-o'clock) depends on the type of pumping unit. The manufacturer recommends that the Mark units rotate only counterclockwise (CCW), the Reverse Mark or Torquemaster only clockwise (CW), while a conventional unit may rotate either CW or CCW from their reference point. The TWM program computes the existing net gearbox torque using **Eq. 5** at each crank angle corresponding to a measured load using the API pumping unit dimensions, crank and master/auxiliary weight specifications using information stored in the Base Well File data base.

COUNTERBALANCE MOMENT

Fig. 5 shows the measurements for calculating counterbalance moments for a commonly used type of crank for a conventional type of pumping units with crank mounted counterweights. For convenience only one counterweight is shown on the top of the crank (this is the #1 counterweight). The numbering scheme to identify the counterweights and cranks is also shown. With the wellhead viewed to the right, the crank nearest to the observer is the #1 crank and the #2 crank is on the opposite side. With the crank pointing to the wellhead, the counterweights on the top of the crank are #1, those on the bottom are the #2 Counterweights.

$$Me = Mcr + \sum_{i=1}^{Nm} Wm_i \times (Dcg_i - Xi) + \sum_{i=1}^{Na} Wa_i \times (Dcg_i - Xi) \qquad (6)$$

Eq. 6 is used to calculate the existing counterbalance moment for conventional cranks by summing of the moments contributed by the cranks themselves (Weight x Center-of-Gravity) plus the moments of the master and auxiliary weights. **Fig. 6** displays how the cranks, master weights and auxiliary weights are selected from a database for a specific type of pumping unit. The calculation of the existing counterbalance moment for the current configuration displayed in **Fig. 6** is listed below:

	Crank #1	Crank #2
Name	8495B	8495B
Weight - Lbs	3510	3510
Center Gravity (CG) - inches	46.25	46.25
Mcr, Crank Moment (in-lbs):	162,338	162,338

	Master Weight #1	Master Weight #2	Master Weight #1	Master Weight #2
Name	3CR0	3CR0	3CR0	3CR0
Wmi, Weight (Lbs)	1327	1327	1327	1327
Dcgi (inches)	72.2	72.2	72.2	72.2
Xi. (inches)	40	40	40	40
CG -inches	32.2	32.2	32.2	32.2
M. W. Moment (in-lbs):	44,056	44,056	44,056	44,056

The existing counterbalance moment, **Me,** is calculated to equal 500,900 in-lbs. (2 X 162,338 + 4 x 44,056). If any auxiliary weights were present, then they would have been included in the calculation using the same procedure as the master weights.

COUNTERBALANCE EFFECT TEST

A counterbalance effect test is performed to measure the net load effect of the counterbalance moment at the polished rod in order to calculate the net torque on the gearbox. Various procedures are available to perform the test, but in general the counterbalance effect load is determined by stopping the unit on the upstroke with the cranks level. If the counterbalance effect load is between the buoyant rod weight plus fluid load and the buoyant rod weight, the pumping unit's crank will balance momentarily as the load is equalized due to fluid leakage from the tubing into the pump. **Fig. 7** shows the polished rod load trace versus time, where the operator stopped the unit on the upstroke with the cranks level. The initial polished rod load at 55 seconds was slightly greater than the buoyant rod load plus the liquid load, 12062 Ibs. As the liquid load leaks past the plunger, the polished rod load dropped toward the buoyant rod weight, 8041 Ibs. As the drop in load was occurring, the operator released the pumping unit brake periodically to determine whether the polished rod load is greater or less than the counterbalance effect load. The crank arm was horizontal when the counterbalance load was determined; so, minimal crank arm movement occurred on each brake release. Every 2 seconds or so, the operator released the brake and the brake drum was examined for any movement. The elapsed time from the beginning of the CBE test was 137 seconds until the brake could be completely released without any movement of the crank arm. A vertical line indicator was positioned with the arrow keys to the exact time when the crank arm did not move while the brake was released and the CBE value of 11024.6 pounds was measured.

Equation 7 is used to calculate the existing counterbalance moment from the measured counterbalance effect load:

$$Me = TF_{,,} \times (CBE - SU) / \sin(q + t) \qquad (7)$$

A properly functioning brake is a required component of this type of CBE test. The procedure for this CBE test normally works, unless the counterbalance effect load is greater than the traveling valve load or less than the standing valve load. This occurs when the pumping unit is very weight or rod heavy. This technique will not work when fluid slippage through the pump is rapid and the fluid load quickly transfers from the rods to the tubing.

FIELD DATA

Table 2 summarizes net gearbox torque calculations from field data collected on 10 different wells throughout the United States. The table displays the counterbalance moment and the distance to move the counterweights to balance the upstroke and downstroke peak of the net gearbox torques. Generally gearbox torque from power and CBM are in close agreement, where the CBE calculated gearbox torque shows the most difference.

Of the 10 wells, the distance required to move the counter weights is usually in good agreement when comparing balancing methods of CBM and power. When examining the field data there appears to be more discrepancies from the CBE method in determining the distance required to move the counterweights to balance the upstroke and downstroke peaks. **Fig. 8** shows that for most of the wells the CBE balancing method requires the weights to be moved from their current position different than the other two methods. For well 5 and ***6*** the CBE method said gearbox loading was rod heavy and to move the weights out from their current position, while the CBM and Power methods calculated to move the weights in. Wells 3 and 7 also show much different movement of the weights as calculated by the CBE, compared to CBM and power. Power method was not available for well 4, because the prime move was a gas engine. The other five wells (1, 2, **8,** 9, and 10) required approximately the same movement of the weights to bring the net torque loadings into balance.

POSSIBLE SOURCES OF ERROR

Peak torque balancing methods of CBM and CBE use many of the same parameters in the calculation of net gearbox torque, see **Eq. *5*** and **Eq. 7.** The API dimensions for a pumping unit are either hand entered or selected from a database. Some common sources of error concerning the API dimensions result in error in the torque factor calculations are: 1) the wrong pumping unit is select, 2) pumping unit not in the database, 3) field assembly of the pumping unit results in dimensions not matching database, 4) wrong radius/stroke length, and 5) direction of rotation. The wrong stroke length is usually detected by acquisition of the dynamometer card when an accelerometer is used to determine the stroke length, because the measured stroke length is determined from integrating the acceleration data and the user is notified that there is a discrepancy between the measured and data base stroke length.

For conventional units the direction of the crank rotation can be either CCW or CW, a common mistake is to leave the default CW direction of rotation selected and not select the correct direction of rotation. If power torque is acquired at the same time as the mechanical torque and the plots of the net torques do not overlay or do not have the same shape, then the most common cause usually is the direction of rotation causing the mechanical torque to be calculated improperly. A bad bearing or extra energy loss due to some type of friction between the input to the motor and the polished rod can result in the power and mechanical torque curves not aligning'. When the power and mechanical torque curves do not match, most often this mismatch is caused by an incorrect parameter the user has entered in the well database.

Both the CBM and CBE balancing techniques depend on accurately measured loads. The polished rod loads should be acquired using a calibrated load cell. Often the load measured with a permanently mounted, donut type loadcell can be in error due to 1) damage to the loadcell, 2) an error in the calibration of the load cell, 3) a load cell that is not centrally loaded at the top and bottom, or 4) an error in the calibration of the controller incorrectly converting the mV/V output from the loadcell into pounds of load. When acquiring data for the CBE load test, if the CBE load is measured too low, then the unit would appear to be more rod heavy. Both static friction and pumping unit inertia can cause the CBE load to be measured incorrectly. If a lower than actual CBE load is used, then the net gearbox peak torque would calculate too high when lifting the rods and too low when lifting the weights. When using a lower than normal CBE load, the location recommended to move the weights would be "out" closer to the end of the crank. For the CBM method the effects of using a measured polished rod load lower than actual load has just the opposite effect, the low polished rod load would make the pumping unit appear to be more weight heavy and the recommended location of the weights would be moved "in" from the end of the crank. Errors in load measurement effect the calculated location for the placement of the counterweights using the CBM and CBE mechanical methods, but errors in load measurement do not effect the motor power balancing method.

Calculation of the existing counterbalance moment was shown to be a fairly simple process in a previous section. The most common problem in determining the existing counterbalance moment is that the weights and center of gravities for a particular type of crank or counter weight are not known and the CBM method cannot be accurately used. The location of the weights in inches from the end of the crank is required for this method and the CBM method cannot be used if the distance from the end of the crank is unknown or was not recorded into the database from the last time the weights were moved. Different types of auxiliary weights are manufactured and the type of auxiliary weights are some times difficult to identify, because some types can be hidden in a pocket in the master weight and other types have slightly different thickness. Incorrectly identifying or ignoring a crank, master weight or, auxiliary weight will make a significant effect on the calculated counterbalance moment.

When using power to balance the torque loadings on the gearbox, the motor and belts efficiency is defaulted to 80% in the software program. Experience has shown this efficiency to be reasonably accurate for the purpose of calculating existing gearbox torque[9,10,11,12,13]. NEMA D motors operate at a nearly constant efficiency over a wide horsepower range, but a lightly loaded motor operates with a much lower efficiency". Rewound motors do not have the same efficiency performance of a new motor. The efficiency of a rewound motor depends on the quality of the repair and the efficiency may need to be de-rated or in some cases increased. When the actual efficiency of the belts and motor are much lower than the default, then calculated peak gearbox torques will be much higher and the gearbox loading could even be in excess of gearbox rating. Usually any error in efficiency affects the peak gearbox torques the same, therefore the distance to move the weights from their current location usually remains the same regardless of any error in efficiency.

BEST METHOD TO BALANCE A PUMPING UNIT

The best method for an operator to use to balance the net peak torques on a pumping unit gearbox is to use both power and mechanical methods at the same time to determine the existing net gearbox torque. When viewing the plot of net gearbox torque from power torque overlain by the net gearbox torque from mechanical torque, it is a simple matter to visually examine the plots and look for discrepancies.

Both dynamometer and power data can be further analyzed to determine instantaneous net gearbox torque. The upstroke and downstroke gearbox torque are both calculated and a recommended distance to move the counterweights to balance the unit is displayed. Pumping unit balance is easy using this combination of power measurement and dynamometer equipment.

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Table 1
Example Net Gearbox Torque Calculations

(1) Crank Angle Deg	(2) Phase Angle Deg	(3) Col 1 + Col 2 Deg	(4) Sin(Col3)	(5) Well Load (Lbs)	(6) Structural Unbalance (Lbs)	(7) Col. 5-6 Net PRL (Lbs)	(8) Torque Factor (In)	(9) Col. 7x8 Well Torqu (In-Lbs)	(10) M6nRent Moment (In-Lbs)	(11) CBoIT4rifu C.B. Torqu (In-Lbs)	(12) Col 9-11 let Torqu (In-Lbs)
0.0	0	0.00	0.000	8658	550	8108	1.58	12794	5009001	0	12794
15 0	0	1500	0 259	9005	550	8455	18 87	159545	500900	129442	29922
30 0	0	3000	0 500	10107	550	9557	32 11	306898	500900	250450	56448
45 0	0	4500	0707	11423	550	10873	41 87	455190	500900	354190	101001
40 0	0	6000	0 866	12767	550	12217	48 17	588525	500900	433792	154733
75 0	0	7500	0 966	13436	550	13086	51 14	669253	500900	483832	185421
90 0	0	9000	1000	12485	550	11935	50 76	605893	500900	500900	104993
105 0	0	10500	0.966	11408	550	10858	46 91	509394	500900	483832	25562
120 0	0	12000	0.866	11774	550	11224	39 59	444325	500900	433792	10533
135 0	0	13500	0707	12189	550	11439	29 35	341548	500900	354190	-12422
150 0	0	15000	0500	11761	550	11211	17 59	197159	500900	250450	-53291
1450	0	14500	0 259	11113	550	10563	4 01	43474	500900	129442	-46147
173 5	0	17350	0113	11131	550	10581	0 00	0	500900	56704	-56704
1800	0	18000	0 000	11240	550	10710	-4 28	-45869	500900	0	-45869
1950	0	195.00	-0.259	11504	550	10954	-13 12	-143701	500900	-129642	-14058
210 0	0	21000	-0.500	11433	550	10883	-20 87	-227123	500900	-250450	23327
225 0	0	225.00	-0 707	11203	550	10653	-28 04	-298702	500900	-354190	55488
240 0	0	24000	-0.866	11033	550	10483	-34 96	-366499	500900	-433392	47293
255 0	0	25500	-0 966	8449	550	7899	-41 64	-328890	500900	-483832	154942
270 0	0	27000	-1 000	7489	550	7139	-47 52	-339252	500900	-500900	141448
285 0	0	285.00	-0 966	5914	550	5344	-55 48	-297574	500900	-483832	184258
300 0	0	30000	-0 866	4706	550	6156	-50 99	-313910	500900	-433792	119882
315 0	0	315.00	-0.707	8364	550	7814	-44 72	-349428	500900	-354190	4761
330 0	0	330.00	-0 500	8344	550	7794	-32 18	-250796	500900	-250450	-346
345 0	0	34500	-0.259	8302	550	7752	-15 34	-118904	500900	-129643	10738
357 8	0	357.80	-0.038	8451	550	8101	0 00	0	500900	-19229	19229
360 0	0	36000	0000	8658	550	8108	I 58	12794	500900	0	12794

Table 2
Comparison of Net Gearbox Torque Balancing Methods

Well	Date	API Pumping Unit Description	CBM	CBE	Power	Move Counterweights Inches: IN(-) Out(+) CBM	CBE	Power
1	09/09/01	C-320D-256-100	229344	256630	209642	-3.6	-8.9	-4
2	12/19/01	M-456D-305-120	261935	251680	313281	-1.2	-0.1	3.6
3	12/19/01	C-640D-305-120	652468	518920	737500	12.3	1.1	14.7
4	10/18/01	RM-640D-405-156	731074	598643	N/A	-17.4	-10	N/A
5	01/12/01	M-640D-305-192	457988	399640	380900	-11.41	2.7	-3.3
6	09/25/00	C-640D-304-144	325000	384284	300900	-0.9	9.3	-3.1
7	09/26/00	M-640D-305-192	413224	484773	353800	-0.1	-9.4	-0.3
8	09/26/00	M-640D-305-192	409029	434129	356400	-7	-9.7	-3.7
9	08/29/00	M-640D-305-168	623343	414192	642800	2.8	2	4.5
10	11/10/00	M-456D-305-168	348207	381389	315300	0.4	3.2	-0.7

Nomenclature

W = well load at a specific crank angle
SU = structural unbalance of the pumping unit (either plus or minus value)
TF = torque factor, inches
M = existing counterbalance moment of the crank and counter weights
θ = the crank angle
τ = the crank phase angle.
Me = Existing counterbalance moment of the crank and counter weights (in-lbs)
Mcr = Crank counterbalance moment (in-lbs)
Wm = Weight of the master counterweight (Ibs)
Wa = Weight of the auxiliary counterweight (Ibs)
Dcg = Maximum distance from centerline of the Crankshaft to the counterweight center of gravity (in)
X = Distance from end of the crank to the outside edge of the counterweight
Xmax = Maximum distance along crank that counterweight can be moved
Nm = Number of master counterweights
Na = Number of auxilary counterweights
KW = Instantaneous motor power
Eff = Motor/Belts Efficiency
SPM = Strokes per Minute
SV = Speed Variation of the Motor

Figure 1 - Torque (in-lbs) or Power (kW) Signatures for a Unbalanced or Balanced Pumping Unit

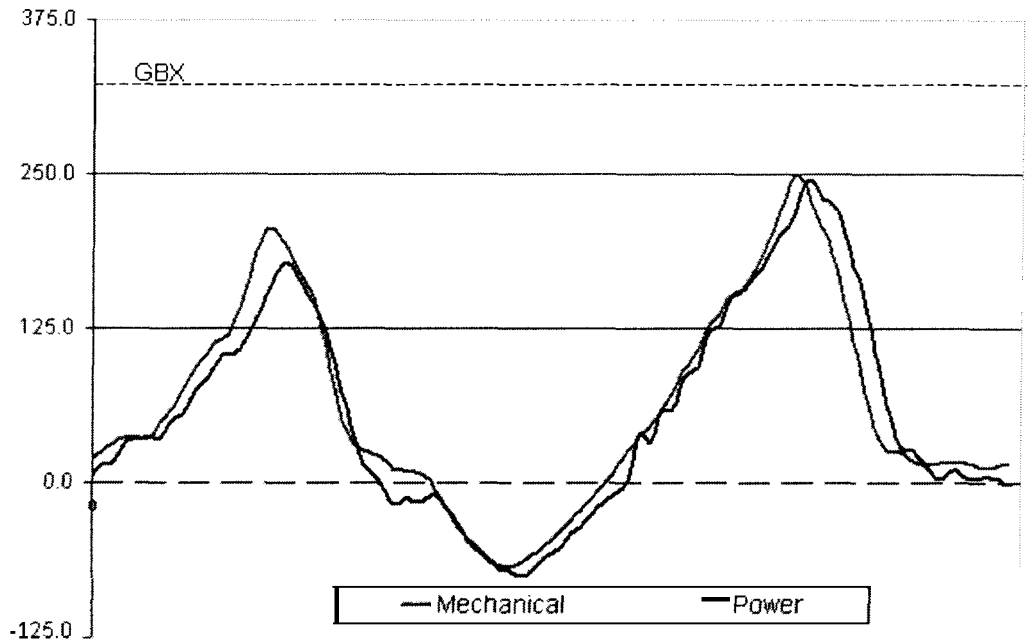

Figure 2 – Compare Motor Output Torque and Gearbox Torque

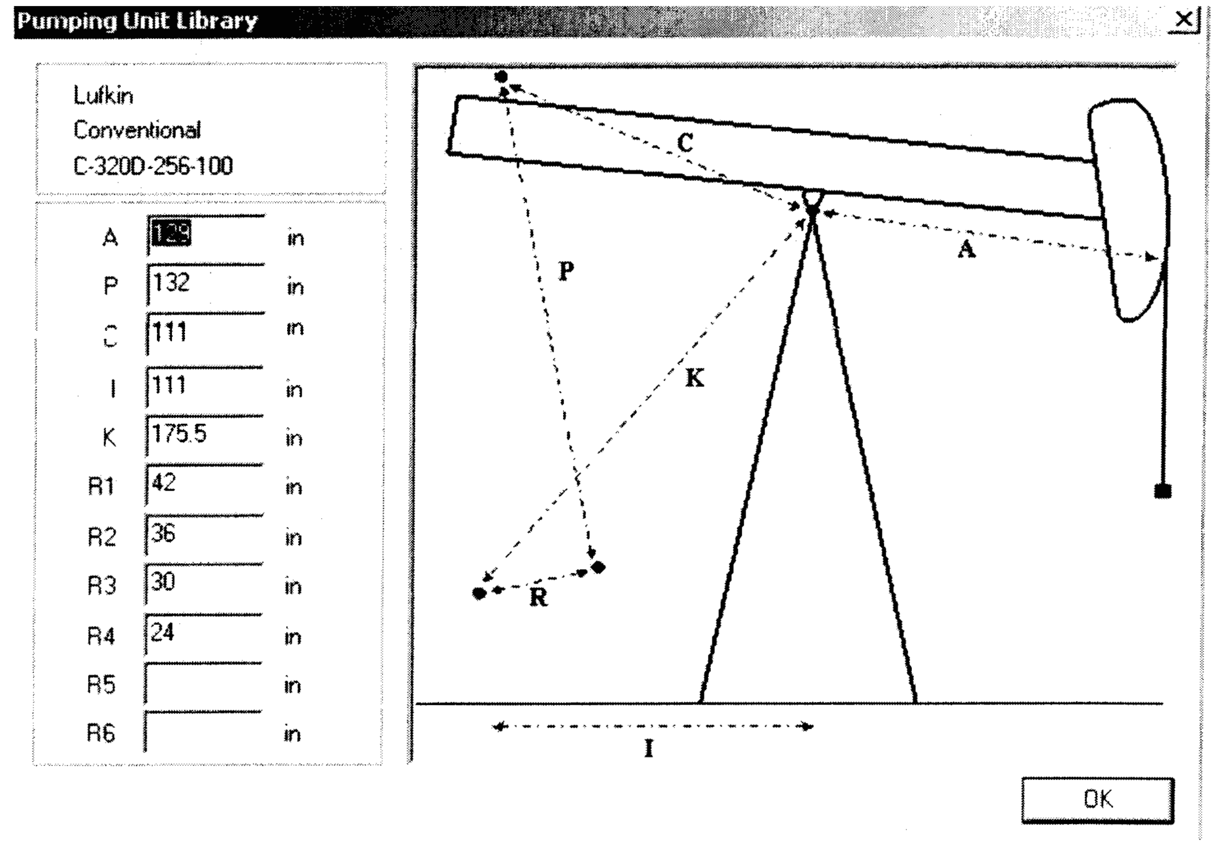

Figure 3 - API Dimensions for a Conventional Pumping Unit

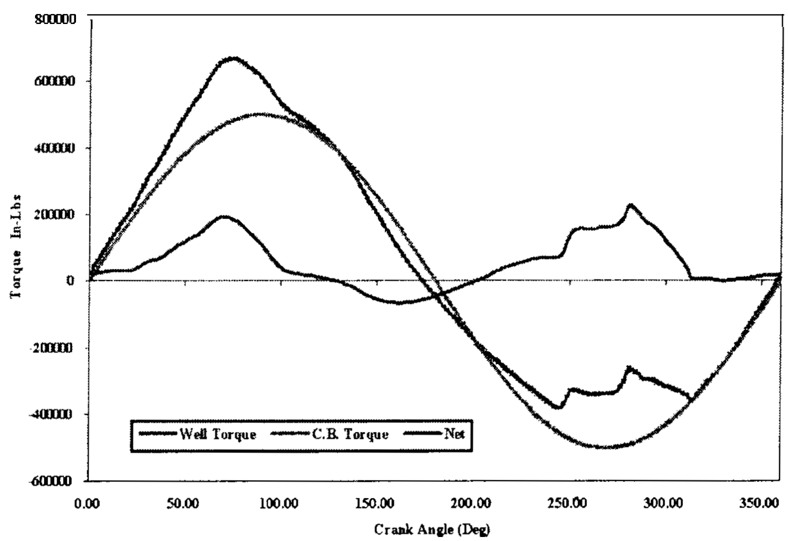

Figure 4 - Torques: Net Well Load, Counterbalance, and Net Gearbox

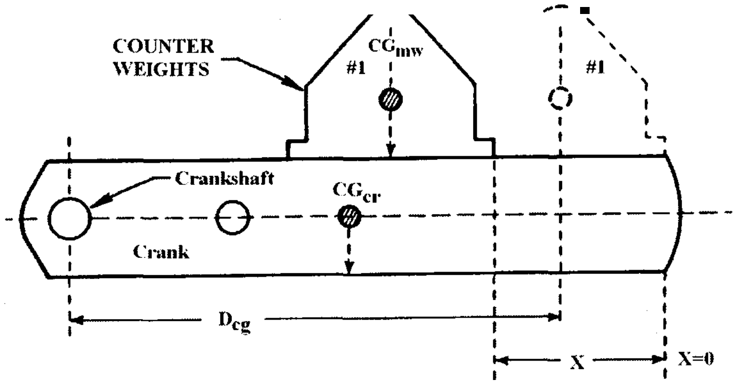

Figure 5 - Conventional Pumping Units with Crank Mounted Counterweights

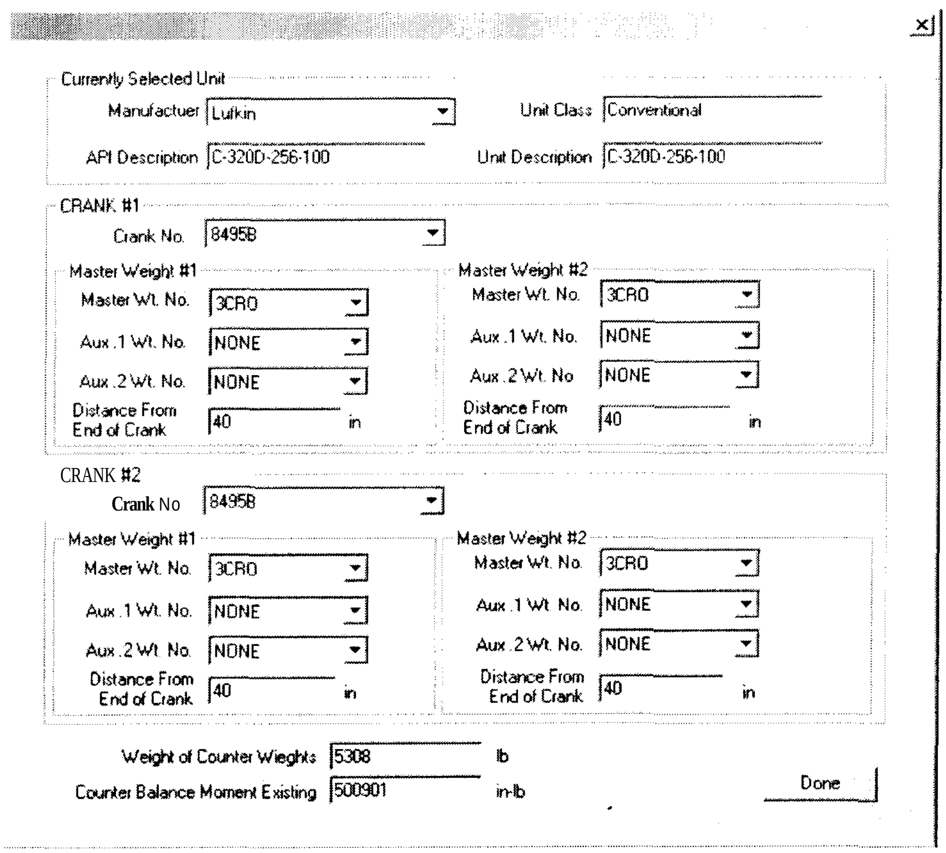

Figure 6 - Select Cranks, Master Weights and Auxiliary Weights

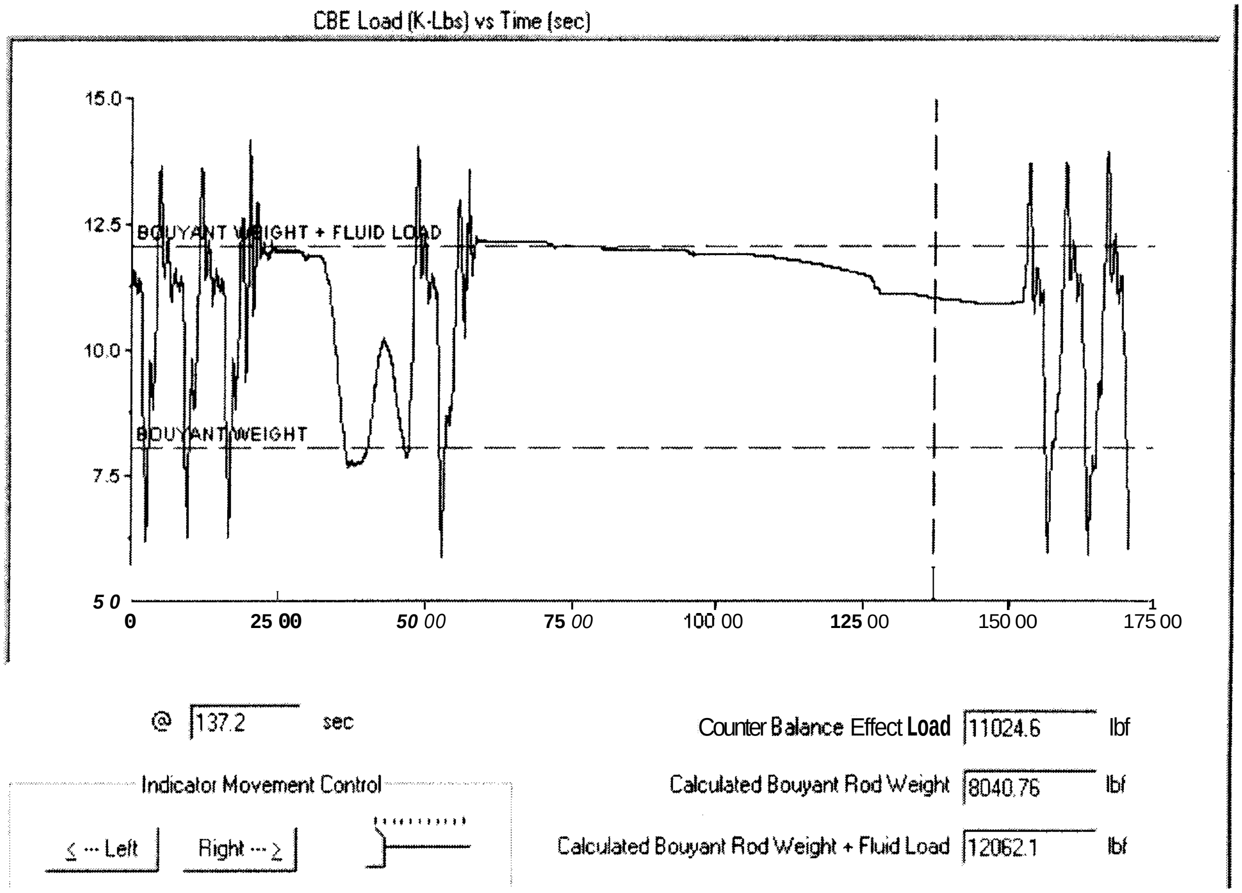

Figure 7 - Counterbalance Effect Test

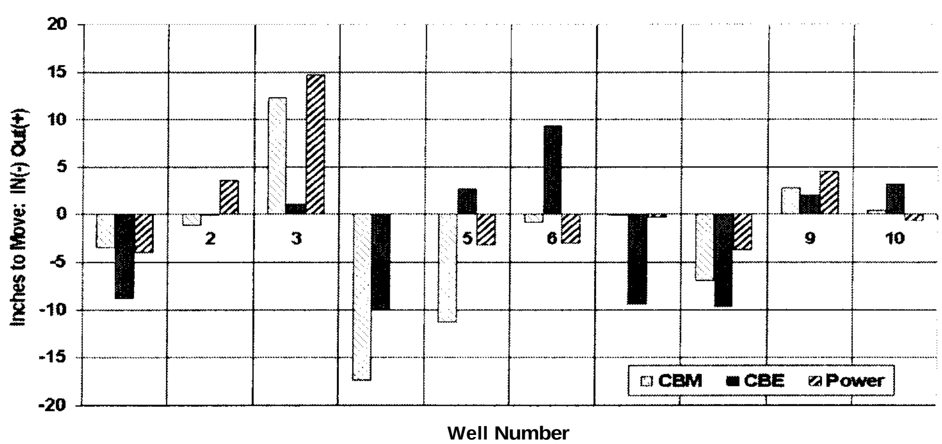

Figure 8 - Move Counterweights from Current Position for Wells 1-10