Increase Profits by Reducing Electricity Consumption: Cost Effective Procedures for Counterbalancing, Motor Sizing, and Friction Reduction Help Increase Sucker-Rod Pumping System Profits

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ABSTRACT

There are several ways to increase profit by reducing electricity consumption. (Counterweight balancing reduces electricity consumption. Correctly sized motors reduce electricity consumption. Reducing rod friction reduces electricity consumption.) The key is to control the maintenance costs.

Counterweight balancing costs may be minimized in two ways. One way eliminates engineering time. The other balances the pumping unit with one counterweight move.

Motor size is important. Oversize motors waste electricity. Undersize motors increase unscheduled maintenance. Undersize motors can cause lost production.

Sucker-rod friction is wasteful. A rule-of-thumb identifies low friction combinations of speed, stroke, and pump size.

PROBLEM STATEMENT

As the cost of electricity goes up, operating costs also rise. One way to increase profit is to reduce power consumption. There are several ways to reduce power consumption. Profitable electricity savings require low maintenance and capital costs.

SOLUTION STATEMENT

Electricity consumption goes down when the counterweight is moved to its ideal location, Fig. 1. (Appendix A describes the well.) This paper presents two ways to economically balance sucker-rod pumping units. One method uses inexpensive field data and simple calculations. The second method can balance a pumping unit with a single counterweight move. (The second method also estimates mechanical efficiency.)

Oversize motors waste electricity, Fig. 2. Field data identifies the smallest motor that can provide the necessary horsepower. Undersize motors increase maintenance and lost production costs.

Excessive sucker-rod friction wastes electricity. A rule-of-thumb reduces the friction horsepower, Figs. 3 and 4.

BACKGROUND

There were 463,854 U.S. stripper wells in 1990.[1] (Stripper wells produce a maximum of 10 bopd.) The average stripper produced 2.26 bopd, and the average U.S. well produced 12.41 bopd. These averages make it easy to believe previous claims that over 90% of the U.S. wells use sucker-rod pumping units for artificial lift.[2] The typical sucker-rod pumping unit is driven by an electric motor, Fig. 5.

The pumping unit converts the motor's rotating output to the up-and-down motion of the polished rod. The polished-rod forces determine how much torque is required from the motor. Fluctuating polished-rod loads require different motor torques during different portions of the polished-rod stroke, Fig. 6. Naturally, the motor must be large enough to supply the peak torque. Less obvious is the penalty for oversizing the motor, Fig. 2. An oversized motor uses extra electricity because the power factor and the efficiency are low when the motor is lightly loaded, Fig. 7.

Motor Size

The required amount of torque varies dramatically during each pump stroke. The torque for a typical rod-pumping system varies. The high approximates the nameplate rating of the motor. The low is negative, Fig. 8. (Negative torque means the well is actually running the motor as a generator.) Current also varies dramatically during one pump stroke. The current is proportional to the motor torque. Net crankshaft torque relates to motor horsepower.

$$P_{mot}(t) = T_c(t)\ N(t)\ /\ (63{,}025\ \eta_{dt}) \qquad (1)$$

The motor's ability to avoid overload damage is measured by the acceptable temperature rise. The operating temperature rise is related to the thermal horsepower. The motor's thermal horsepower is related to the root-mean-square (RMS) current. Mathematically, the RMS current is most affected by the peak current. The fluctuating current includes two peaks. The peak currents occur near the middle of the upstroke and the downstroke. Since the thermal capacity of the motor is related to the peaks, reducing the peaks reduces the load on the motor. The lowest current peaks occur when the upstroke peak equals the downstroke peak. Equating the current peaks is known as balancing the current peaks. Current peaks are balanced using crank-mounted weights.

Counterweights, Currents, and Torques

The counterweight on a pumping-unit is like the flywheel on a car engine. Without the flywheel, the car needs a larger engine. Counterweights, like flywheels, help reduce the pumping unit's motor size. This is because counterweights, like flywheels, even out the torque requirement. There are several counterweight-balancing criteria. Kemler mentions three: minimize the peak torque, make the upstroke torque equal to the downstroke torque, make the upstroke horsepower equal to the downstroke horsepower.[3] Later authors focus on equalizing either the peak currents or the peak torques.[4,5,6,7,8,9] API SPEC 11E recommends simplified equations that balance the counterweights by equating the peak crankshaft torques.[10] The counterweight balancing methods in this paper uses the difference between the upstroke and downstroke motor current peaks to predict the ideally balanced counterweight location.

Balanced Counterweight Location

When the counterweight is at its ideal location, the upstroke peak current equals the downstroke peak current. When the peak currents are equal, the root-mean-square (thermal) current is minimized. Minimizing the RMS current also minimizes the motor size. Minimizing the motor size allows the motor to operate with higher power factors and electric efficiencies, Fig. 7. Thus, equating current peaks minimizes power consumption by minimizing the required motor current.[11] This happens when the counterweights are ideally placed.

The ideal location is defined as the length from the crankshaft centerline to the counterweight's center of mass. The shorthand notation for the ideally balanced counterweight location is L_{bc}, which stands for balanced counterweight location. The L_{bc} minimizes power consumption.

Actually, each system has a range of potential L_{bc} values. The L_{bc} should be selected for the operating fluid level.[12] The preferred L_{bc} is for the maximum expected net lift. The maximum net lift depends on the minimum fluid level in the casing. The method described in this paper works for any fluid level. As a practical matter, this paper assumes the fluid level falls to the lowest possible level without actually pumping off. There are two reasons to design for a casing fluid level just high enough to avoid pumping off. First, the reservoir delivers the maximum amount of oil to the pump when casing fluid level is just above the pump. Secondly, the hydraulic horsepower requirement (at the pump) is higher for this fluid level than for any other fluid level. When the casing fluid is at any other level, the reduced hydraulic horsepower tends to compensate for the unbalanced current peaks, Fig. 9.

Torque Components

To find the proper L_{bc}, it helps to understand how the loads are transmitted through the pumping system. The following sections briefly discuss how each of the four loads affects the system.

Fig. 5 shows a typical rod-pumping installation. The major components are the pump, the sucker rods, the pumping unit, the speed reduction (gearbox, sheaves & belts), and the electric motor. The electric current into the motor provides the fluctuating torque that moves the traveling valve. In simple terms, the torque is a combination of four fluctuating loads. The four torque components are associated with: the polished rod load, the rocking inertia load of the beam, the rotating inertia load of the drive train, and the gravity load due to the masses of the crank and the counterweight. The combined fluctuations of these four load components determine the motor current fluctuations. (Appendix B gives a detailed discussion of the torque due to the rotating inertial load.)

Polished Rod Load

The most important torque requirement (of the four torque components) is due to the vertical forces in the polished rod. The equivalent crankshaft torque includes the product of the torque factor and the polished rod force.

$$T_{pr} = F_{tf} (F_{pr} - F_{su}) / \eta_{pu} \quad \text{(2)a}$$

Rocking Inertial Load

The angular accelerations of the beam contribute inertial torque to the total requirement. The equivalent crankshaft torque depends on the torque factor, the beam's angular acceleration, the beam's inertia, the distance from the Sampson bearing to the horsehead, and the pumping unit's efficiency.

$$T_{Ib} = 12 F_{tf} \alpha_b I_b / (L_A g_c \eta_{pu}) \quad \text{(2)b}$$

Rotating Inertial Loads

The angular accelerations of all the rotating parts in the drive train contribute to the rotating inertial loads. The only rotating parts between the horsehead and the crankshaft are the cranks and the counterweights.

$$T_{Ic} = 12 \alpha_c I_c / g_c \quad \text{(2)c}$$

The drive train includes other inertias plus mechanical efficiencies. The net motor torque equation includes all the drive train inertias and efficiencies, Appendix C. The other drive train components include the inertial torque due to the gearbox, the low speed sheave, the v-belts, the high speed sheave, and the motor rotor. The mechanical efficiencies of the pumping unit, the gearbox, and the v-belts add to the required output torque at the motor.

Crank and Counterweight Load

The gravity torque due to the cranks and counterweights changes as the crank rotates. The phase angle (τ) is defined by SPEC 11E.

$$T_{cwt,\theta} = T_{cwt} \sin(\theta + \tau) \quad \text{(2)d}$$

OVERVIEW

This paper presents several ways to increase sucker-rod profits by reducing electricity consumption. These suggestions include correct motor sizing, and rod friction reduction, and balancing.

Motors are designed to be most efficient for a steady, full load. A correctly sized motor uses less electricity than an oversize or an undersize motor. Neither oversize nor undersize motors run efficiently.

Sucker-rod friction consumes electricity. There are many combinations of pump size, stroke length, and stroke speed for any given production rate. Some combinations are preferred because they reduce the friction horsepower.

There are two balancing methods. One avoids calculations and balances pumping units in as few as two moves. The other balances the current peaks with a single counterweight move. The goal of both methods is to improve profit by reducing the balancing cost. The no-calculation method eliminates engineering time. The one-move method emphasizes engineering.

The no-calculation method uses simple, inexpensive field measurements. The field data from two counterweight locations estimates an ideal counterweight location that balances the peak currents (L_{bc}). A graphical prediction estimates the ideal L_{bc}. A simple calculation can also estimate the ideal L_{bc}. Either way, engineering time is zero.

The one-move method expedites the calculations with a computer. Program inputs include a measured dynamometer card and measured motor currents. The procedure assumes all vendor and well data are known, except for the efficiencies. The efficiencies are adjusted until the calculated motor currents match the measured currents. This identifies the mechanical efficiencies of the pumping unit, the gearbox, and the v-belt drive. Given all this information, it is theoretically possible to balance the peak currents with a single counterweight move.

EXISTING BALANCING METHODS, (Crankshaft-Torque and Motor-Torque)

There are several counterweight-balancing methods. API SPEC 11E uses a simplified torque equation for crankshaft torques. Gibbs and Svinos use a more accurate version of the SPEC 11E equation for calculating crankshaft torques.[13,14] The *motor* torque method extends the crankshaft torque equations to the motor output shaft.[15] (Appendix D discusses the relationship between the equations for crankshaft torque, motor torque, and motor current.)

The traditional balancing procedure involves measuring a surface dynamometer card. The measured forces and positions are converted to torques, using any of the net torque equations. There is an ideal counterweight torque that sets the peak upstroke torque equal to the peak downstroke torque. Any of the net torque equations can approximate the ideal counterweight torque.

The simplest net torque equations are the least accurate. Calculation accuracy suffers with all net torque equations. Accuracy suffers because the net torques are difficult to measure. Without measured torques, calculations are not validated. Given this limitation, the following assumption is made; assume that the calculated difference between the actual and the ideal counterweight torques is accurate.

An accurate torque difference can predict how much the counterweights must be moved to balance the pumping unit, *i.e.*, predict the difference between the present counterweight location and the ideal L_{bc}. (Vendor data are needed to convert the differential counterweight torque into a counterweight movement.) After moving the counterweights, the pumping unit is allowed to run long enough to stabilize the operating fluid level. Once the fluid level has stabilized, another dynamometer card is measured. At this point a second torque analysis can verify that the unit is balanced, *i.e.*, confirm that the counterweights are at the L_{bc}.

In actual practice, the second torque analysis probably shows that the unit is still out of balance, albeit closer to the ideal balance. Three limitations lead to this situation. First, there may be uncertainty about the actual counterweight mass and torque. Secondly, the pumping unit may run differently after the counterweights have been moved. Finally, the net torque equations vary in accuracy. In the past, multiple counterweight moves were required to truly balance a pumping unit. Unfortunately, balancing costs (crew time, engineering time, and lost production) tend to negate the power consumption savings. Keating presented a balancing method (based on the SPEC 11E equation) that needed as few as two counterweight moves to find the L_{bc}.[16]

MOTOR-CURRENT COUNTERWEIGHT BALANCING, NO-CALCULATIONS

Balancing peak currents without calculations and without trial-and-error is possible. The key is to balance the pumping unit over a period of time. Field data from two, previous partial balances predict an improved L_{bc}.

Figs. 1 and 2 show the electricity cost penalty for unbalanced operation. The unbalance penalty is on the order of $100 /yr. This is an upper limit on balancing cost for the example well. (Appendix A describes the example well.) Profitable balancing requires controlling balancing costs. One way to control costs is to eliminate engineering time. Another way is to minimize the balancing time. (Balancing with one counterweight move is discussed later.)

Background, Current Balancing, No Calculations

Historic field procedures for equating peak currents and for balancing peak torques are similar. Peak current measurements lead to counterweight moves, which modify the current peaks. There are three advantages to balancing current peaks. First, electricity consumption (hence cost) is minimized. Secondly, currents (unlike net torques) are easy to measure. Finally, there are no calculations. Unfortunately, "no calculations" is also a disadvantage.

The disadvantage is that there is no L_{bc} prediction. Without a L_{bc} prediction, equating the upstroke and downstroke current peaks is a trial-and-error method. Extra counterweight moves add to the cost of balancing.

Procedure, Current Balancing, No Calculations

A way to save time is to do partial balances. (Partial balancing recognizes that the next counterweight move can be on another day.) Scheduling partial balances in conjunction with other wellsite activity saves more time because the labor force is already present. Two partial balances give enough field data to approximate the ideal counterbalance location, L_{bc}. It takes at least two counterweight moves to find the ideal L_{bc}.

The procedure requires three field measurements for each partial balance. It is important to save the field data from past partial balances. Field data are only measured before moving the counterweights. The data are the upstroke peak current, the downstroke peak current, and the counterweight location.

Counterweight Location Measurements

The preferred counterweight reference point is the counterweight center of mass. $L_{cw,cm}$ is the distance from the crankshaft centerline to the counterweight's center of mass. An eyeball estimate of the counterweight's center of mass is probably close enough.

Current Measurements

Naturally, the well should be pumped long enough for the fluid level to stabilize before recording the peak currents. (I recommend an ammeter that captures the peak current. This reduces reading error and allows time to record one current before the other peak arrives.)

Activity Sequence

Measure $L_{cw,cm}$ and the peak currents. Estimate the ideal L_{bc}. Move the counterweights. This completes one partial balance. Come back another day for the next partial balance.

Continue making partial balances (over a period of time) until the pumping unit is balanced. Each improved L_{bc} is based on the two sets data that are closest to the ideal L_{bc}.

Minimum Cost Strategy

Waiting time wastes money.

Plan to arrive at the well shortly before the fluid level stabilizes at its normal operating level. If the well runs 24 hrs /day, anytime will do. Otherwise, arrive before pumpoff. (Hint: the upstroke current peak will approach its highest value as the well begins to pump off.)

Depart immediately after moving the counterweights. Waiting for the fluid level to stabilize wastes money. The next peak current data will be measured on another day. (Plan to arrive at the well shortly before the fluid ····)

L_{bc} Prediction Strategy

Move the counterweights closer to the ideal L_{bc}. Moving closer improves extrapolation accuracy. This assumes the correct movement direction is known.

Which Direction to Move

The direction of the move depends on whether the peak current is higher on the upstroke or the downstroke. When the upstroke current peak is larger than the downstroke current peak, then the counterweight torque is too low. The counterweight should be moved away from the crankshaft.

This implies one knows how far the counterweight needs to be moved to reach the ideal L_{bc}.

How Far to Move

The strategy for the first counterweight move is different.

First Counterweight Move

Guesstimate the distance to the ideal L_{bc}. A large move is more likely to bracket the ideal L_{bc}. Bracketing the ideal L_{bc} allows interpolation, which is preferred over extrapolation. If you have no idea how far to move the counterweights, be bold. Quickly (before the fluid level rises far) and temporarily move one weight to an extreme position and restart the pump. The peak currents will immediately tell you if this move is too little or too much. Now stop the pumping unit and move all the counterweights (including the one moved temporarily) closer to the ideal L_{bc}. (Caution: if there is little movement left in that direction, it may be necessary to add or remove counterweights. Calculations may wise.)

Other Counterweight Moves

After the first move, there are two known $L_{cr,cm}$ values and the two known upstroke and downstroke current peak differences. The known data define the slope and intercept of a straight line. The improved L_{bc} corresponds to zero difference between the peak, upstroke-and-downstroke currents, Fig. 10.

The difference between the peak currents associated with $L_{bc,prev}$ is dI_{prev}. The associated current difference for $L_{bc,new}$ is dI_{new}.

$$dI_{prev} = I_{prev,u} - I_{prev,d} \quad \text{(3)a}$$
$$dI_{new} = I_{new,u} - I_{new,d} \quad \text{(3)b}$$

Assume dI_{improv} is zero at the ideal L_{bc}. The interpolation for L_{bc} is

$$L_{bc,improv} = \frac{L_{bc,new}\, dI_{prev} - L_{bc,prev}\, dI_{new}}{dI_{prev} - dI_{new}} \quad \text{(3)c}$$

Summary, Current Balancing, No Calculations

Eliminate engineering time (by simplifying the calculations) to reduce the cost of counterweight balancing. Graph peak current and counterweight location (field) data to predict an improved counterweight location. Restart the pumping unit and leave after moving the counterweight. (Waiting for the fluid level to stabilize wastes time.) Balance the pumping unit over a period of time. Schedule partial balances in conjunction with other wellsite activity. Save field data from previous partial balances.

ELIMINATE OVERSIZE (AND UNDERSIZE) MOTORS

Oversize motors waste electricity.

Fig. 2 shows the electricity penalty for using an oversized motor. The oversize penalty is on the order of \$150 /(yr hp). (See Appendix A for a description of the example well.)

Bob Gault, a recognized sucker-rod authority, offers the following rules-of-thumb for estimating motor and switchgear costs. Installed motors cost about

$50 /hp. Installed switchgear costs about $30 /hp.

The cost of replacing an oversize motor is lowest when the oversize motor is used on another pumping unit. The payout approaches one year if the existing switchgear is reused.

Background, Eliminate Oversize Motors

The best motor is just big enough to withstand the actual thermal load.

Oversized motors waste electricity by running at a fraction of their rated capacity, Fig. 7. This is because electrical efficiency and power factor are low. The dark line on Fig. 7 is the product of efficiency and power factor. The product is high when the motor runs near its rated load. Of course, there are penalties for exceeding the motor's thermal rating. Knowing the thermal load helps pick a motor that is just big enough.

The thermal load capacity of a motor is its rated temperature rise. The temperature rise is related to the internal power losses. The power losses that heat a running motor are proportional to the square of the current.

$$P = V\,I = (I\,R)\,I = I^2\,R \qquad (4)$$

R is the motor resistance. The root-mean-square current is I. The square of the RMS current is proportional to the power losses that heat the motor.

Procedure, Eliminate Oversize Motors

Identify oversize motors and replace them with smaller motors. Use the replaced motors to replace other oversize motors, if possible.

The thermal load is measured with a thermal wattmeter. Field data guarantees the motor will be large enough, but not oversized. (Simulator prediction accuracy is suitable for most new system design.[17,18]

Thermal analyses may identify several pumping units with oversized motors. Ideally, the operator need only buy one, small motor. As a motor is removed from one pumping unit, it replaces a larger motor on another unit. In the end, there is one spare motor. Naturally, this will be the largest motor. The resale value of this extra motor may be higher than the purchase price of the small motor.

In this ideal example, the capital cost is negative (because the large, spared motor is worth more the cost of the new, small motor). The actual costs are the engineering time (to identify pumping units with oversized motors) and the labor to substitute motors.

Preventive Maintenance, Under Size Motor

Lost production and replacement motor cost reduce profit when a motor fails.

Thermal loads also identify undersized motors. There are optional remedies.

Try to schedule the changes in conjunction with other wellsite activities.

Monitor undersize motors for signs of deterioration. Replacement may not be necessary (or economical).

Balancing may help. A smaller pump, a shorter stroke, or a slower speed may help by reducing the instantaneous production rate. (This assumes the well runs less than 24 hrs /day.) Special operating conditions (*e.g.*, idle 12 hr /day) may allow the motor manufacturer to recommend a lower safety factor. Otherwise, consider substituting a correctly sized motor.

Payout depends on salvage value and on the date of the expected failure.

Summary, Motor Size

Oversize motors waste electricity. Replace oversize motors with correctly sized motors. Payout is on the order of one year.

Undersize motors have shorter useful lives. Monitor undersize motors for signs of deterioration. Consider alternate solutions, such as balancing, larger pump, shorter stroke, or slower speed.

REDUCED ROD-FRICTION HORSEPOWER

Use the largest pump that does not overstress the rod string.

An easy way to reduce energy consumption is to reduce the rod friction horsepower. An analysis of the equations shows that friction horsepower is proportional to velocity squared. Production (q) is proportional to the product of velocity and pump area. Assuming pump stroke is approximately equal to surface stroke, average rod velocity appears in the production equation. For a given production, the largest pump gives the lowest rod velocity. The lowest rod velocity gives the smallest friction horsepower.

$$F_{fr} = M_r\ C_r\ v_r\ /g_c \qquad \text{(5)a}$$

$$P_{fric} = F_{fr}\ v_r\ /550 = M_r\ C_r\ v_r^2\ /(g_c\ 550) \qquad \text{(5)b}$$

$$\bar{v}_r = 2\ N\ S\ /(12\ 60) \qquad \text{(5)c}$$

$$S_p \approx S \qquad \text{(5)d}$$

$$\begin{aligned} q &= A_p\ N\ S_p\ 60\ 24\ /(12\ 144\ 5.6145833) \\ &\approx A_p\ \bar{v}_r\ 60\ 60\ 24\ /(144\ 5.6145833\ 2) \qquad \text{(5)e} \end{aligned}$$

The obvious rule-of-thumb [**use the largest (tubing or rod) pump that does not overstress the rods**] is compatible with a widely quoted rule-of-thumb.[19,20] "Use long, slow strokes."

Wait for a workover before changing the pump. Since the S N product is the other variable, the choice of changing stroke (S) or speed (N) depends on which costs less. It may cost less to change the stroke by moving the wrist pin than it costs to replace sheaves and v-belts.

An easy way to reduce electricity consumption is to use the largest (rod or tubing) pump that fits in the casing without overstressing the rods.

Check on substituting a smaller motor after reducing the friction horsepower.

MOTOR-CURRENT BALANCING WITH ONE COUNTERWEIGHT MOVE

It is theoretically possible to balance the pumping unit with one counterweight move. Balancing on the first move assumes the availability of accurate information about the sucker-rod system.

Background, One-Move Current Balancing Calculations

Balancing peak currents involves some assumptions. First, the voltage is constant. Secondly, the voltage is the same for all three phases. Finally, the phases are balanced.

The current balancing equation is more accurate than the crankshaft torque equations. Currents are easy to measure. Comparing measured and calculated currents leads to efficiency estimates. The mechanical efficiencies are for the pumping unit, the gearbox, and the v-belt drive. (If the utility provides single phase power, then phase inverter efficiency is estimated.) In other words, efficiencies are inferred. (Appendix E estimates efficiency ranges.)

Preventive Maintenance, One-Move Current Balancing

Mechanical efficiencies add a new dimension to sucker-rod maintenance. Low efficiency wastes electricity. (*E.g.*, low pumping unit efficiency may indicate a lubrication failure.) Consider the economics of preventive maintenance when efficiency is low.

Procedure, One-Move Current Balancing

Balancing with a single counterweight move saves field time.

The objective of the one-move balancing method is to improve the economics of sucker-rod pumping units. This is accomplished by minimizing the field time needed to move the counterweight to the ideal L_{bc}.

Field Data and Calculated Efficiencies

The procedure starts with field measurements. Currents and a surface dynamometer card are measured. Record the measurements after the well has been pumped long enough to stabilize. (*I.e.*, pump until the fluid column in the casing falls to the normal operating level. Hint: calculated pump dynamometer cards identify the net lift.) Use measured, vendor, and well data in the motor current equation.

Motor Current Equation

Adjust the efficiencies until the calculated and measured currents match. This defines the mechanical efficiencies for pumping unit, gearbox, v-belt. The current equation leads to the incremental counterweight torque needed to balance the peak currents. The incremental torque is converted into a new L_{bc}, and the counterweights are moved. The current equation is

$$I = \frac{746\ N\ [\ T_{cwt}\ \sin(\theta + \tau)\ - T_{Inet}\ ]}{1.732\ V\ 63{,}025\ \eta_{vb}\ \eta_{gb}\ \eta_m\ F_{pf}} \quad \text{(6)a}$$

L_{bc} Prediction Equation

The following equation predicts the change in counterweight torque needed to balance the unit. (The derivation is given in Appendix C.) This equation considers all the loads and efficiencies between the polished rod and the electric wires.

$$dT_{cw} = \frac{T_{Inetu} - T_{Inetd}}{\sin(\theta_u + \tau) - \sin(\theta_d + \tau)} - T_{cwt} \quad \text{(6)b}$$

The equation for L_{bc} is

$$L_{bc} = L_{bc,prev} + dT_{cw} / W_{cw} \quad \text{(6)c}$$

The counterweights are then moved to L_{bc}.

Accuracy

The accuracy of L_{bc} depends on the accuracy of the vendor and well data. Given accurate mass and mass moment of inertia data, and given accurate power factor and efficiency data, it is reasonable to expect the first L_{bc} estimate to be accurate. The key to a single move balance is the availability of vendor and well details. Naturally, it takes time to collect the necessary details. An alternative to collecting vendor data exists. It is possible to estimate much of the input data, Appendix E. Some operators may find that it costs more to prepare accurate input than to make extra counterweight moves. (These operators are referred to the no-calculation section.)

Validation

The 1-move method requires field validation. Validation will verify that L_{bc} is the ideal counterweight location. Let the pumping unit has run long enough to stabilize the operating fluid level. Measure I_u and I_d. If teh current preaks are equal, the pumping unit is balanced.

Summary, One-Move Current Balancing

The current equation has two benefits. First, one counterweight move balances the pumping unit. Secondly, drive train and pumping unit efficiencies are available. Both benefits point to improved profit.

There is a downside. Computer and engineering labor substitute for physical labor. Vendor data must be collected, but only one time per well.

Current balancing with calculations works by substituting brain effort for brawn effort. Operators with inhouse engineering capability are more likely to have the necessary computers and well files.

RESULTS & CONCLUSIONS

1. **Counterweight balancing.**
   - A. This new method equates peak upstroke and downstroke currents to balance sucker-rod pumping units. (Balancing saves electricity.)
   - B1. The increased accuracy of the current equation reduces balancing cost by finding the ideal counterweight location in one move.
   - B2. The simplicity of the balancing without calculations reduces balancing cost by eliminating engineering time.
   - C. Reducing electricity consumption and reducing balancing cost combine to improve the economic reward for balancing rod-pumping units.
   - D. The 1-move method is based on a net torque equation that is more accurate than those used previously.
   - E. Given accurate vendor and well data, it is theoretically possible to balance a pumping unit with a single counterweight move.
   - F. Measured currents may be used to validate the current equation.
   - G. Validation provides mechanical efficiency estimates of the pumping unit, the gearbox, and the v-belt drive.
2. **Oversize motor replacement.**
   - A. It is sometimes possible to reduce electricity consumption by substituting a smaller motor.
   - B. Measured thermal loads confirm the correct motor size.
3. **Rod friction reduction.**
   - A. A new rule-of-thumb helps reduce electricity consumption by reducing the rod friction horsepower.
   - B. Rod friction is reduced when the largest possible (rod or tubing) pump is used.
   - C. After installing the largest possible pump, it may be possible to further reduce electricity consumption by substituting a smaller motor. (P_{prhp} goes down when rod friction goes down.)

APPENDIX A, EXAMPLE WELL DATA

Oil Production	9	bopd	Depth	3,000	ft
Water Production	81	bwpd	Stroke	64	in
WOR	9		Pump Diameter	1.75	in
Rod	66		Speed	4.9	spm
			Motor	NEMA D	

APPENDIX B, DISCUSSION OF FOUR CONTRIBUTIONS TO CRANKSHAFT TORQUE

Four torque components require motor torque:
the polished rod load,

the rocking inertia load of the beam,
the rotating inertia load of the drive train, and
the gravity torque of the crank and counterweight masses.

Polished Rod Load

The main motor load is the polished rod load. The polished rod load includes: a pump load and a rod load. The pump load is applied at the pump. The upstroke pump load is the force needed to lift and accelerate the fluid. The downstroke pump load is approximately zero. The combination is a fluctuating load. This fluctuating pump load is transmitted up the sucker rods to the polished rod. See Figs. 6 and 8.

At the polished rod, the combination of the rod load and the pump load can be measured. The rod load is the force needed to lift and accelerate the rods. The rod load is also a fluctuating load. Therefore, the polished rod load is a fluctuating load. The polished rod load can be measured with a dynamometer. Measured polished rod loads are an important part of balancing.

The polished rod load is transmitted through the unit to the gear box crankshaft, Fig. 5. When this polished rod load is transmitted through the unit, the efficiency of the unit must be considered. At the crankshaft, the polished rod load becomes a polished rod crankshaft torque.

Next, the polished rod crankshaft torque is transmitted through the gear box, sheaves and belts to the motor's shaft, Fig. 5. At the motor's shaft the polished rod crankshaft torque becomes the polished rod motor torque. To obtain the polished rod motor torque, one must consider the efficiency of the gear box, sheaves and belts.

Finally, this polished rod motor torque draws a current at the motor's wires. When this polished rod motor torque becomes a polished rod current, the power utilization of the motor must be considered. Therefore, the pump load draws fluctuating motor current. Another load that draws motor current is the articulating inertial load.

Articulating Inertial Load

The second load transmitted to the motor is the articulating inertia load. The only rotating parts between the horsehead and the crankshaft are the cranks and the counterweights

The articulating inertia load is the force needed to accelerate and decelerate the walking beam, the equalizer, and the horse head. The magnitude of this load is usually much less than the magnitude of the polished rod load. The articulating inertial load fluctuates as the walking beam accelerates. The beam acceleration is highest at the top and the bottom of the stroke and lowest near midstroke.

Like the polished rod load, the articulating inertia load is also transmitted through the unit, gear box (sheaves & belts), and the motor to the motor's wires. At the motor's wires, the articulating inertia load draws a fluctuating motor current. Again, the proper efficiencies and the power utilization must be considered. Another load that draws motor current is the rotating inertial load.

Rotating Inertial Loads

The third load transmitted to the motor is the rotating inertia load. The rotating inertia load is zero for a constant motor speed. Since the motor speed varies, the rotating inertia load must be considered. The magnitude of this load is usually much less than the magnitude of the polished rod load. The rotating inertia load consists of several loads: crankshaft, cranks, counterweights, gears, unit sheave, motor sheave, and motor rotor. It is convenient to divide these rotating inertia loads into three groups.

The first group includes the rotating inertia of the cranks and the counterweights. This rotating inertia load is applied at the crank (outside the gearbox). This is the force needed accelerate and decelerate the cranks and the counterweights. This load fluctuates with the angular acceleration of the crankshaft. Therefore, the more the rotating speed varies the greater the rotating crank inertia load will fluctuate. The magnitude of these fluctuations depends directly on the position of the counterweight. Thus, the magnitude of the rotating inertia load will change with every counterweight adjustment.

The second group includes the rotating inertia of the gears, and the unit sheave. The gears and sheave rotating inertias are applied at the crankshaft as a torque (inside the gearbox). This is the torque needed to accelerate and decelerate the gears and unit sheave. This rotating inertial torque also fluctuates with the angular acceleration of the crankshaft.

The third group includes the rotating inertia of the motor sheave and the motor rotor. The motor rotor and the motor sheave rotating inertia load are applied at the motor's shaft as a torque. This is the motor torque needed to accelerate and decelerate the motor and motor sheave. This rotating inertial torque fluctuates with the angular acceleration of the motor's shaft.

Like the articulating inertia load all of the rotating inertia loads are then transmitted to the motor's wires. At the motor's wires, the rotating inertia loads draw a fluctuating motor current. Again, all the proper efficiencies and the power utilization must be considered. Another load that draws motor current is the crank and counterweight load.

Crank and Counterweight Load

The final load transmitted to the motor is the crank and counterweight load. The crank and counterweight load is used to offset the other loads. This load is applied at the crankshaft, and it is the force needed to lift the cranks and the counterweights. This is a constant load but not a constant torque. This torque fluctuates because the torque arm changes as the crank turns.

Like all the other loads the crank and counterweight load is transmitted to the motor's wires. At the motor's wires, the crank and counterweight load draws a fluctuating motor current. Again all the proper efficiencies and power utilization must be considered. Therefore, all four of the pumping system loads draw fluctuating motor currents. The resulting current is also a fluctuating current. This resulting current exhibits two peaks: one during the upstroke and one during the downstroke.

These current fluctuations can be minimized by setting the peak upstroke current equal to the peak downstroke current. These peak currents can be balanced by moving the adjustment counterweight to the L_{bc}. This balancing of the peak currents will minimize the motor load and improve the power utilization efficiency. In other words, properly balancing your peak currents reduces your electrical costs.

APPENDIX C, DERIVATION OF THE MOTOR TORQUE EQUATION

Start with the torque at the motor shaft. It includes the inertia, mass and efficiency of each component between the motor and the polished rod.

Torque at the High Speed Sheave

The motor torque is T_m. Part of T_m is used to accelerate the motor rotor, the high speed sheave, and the v-belt. For simplicity,

$$T_{Ih} = 12\ (I_{mot} + I_{hs} + I_{vb})\ \alpha_c\ R_{vb}\ R_{gb}\ /g_c, \qquad (7)a$$

is the inertial torque needed to accelerate the high-speed, rotating components. This leaves $T_m - T_{Ih}$.

Torque at the Low Speed Sheave

The next step is to convert the remaining torque to the low speed sheave. There is a speed change. There is mechanical inefficiency. What remains is $(T_m - T_{Ih})\ \eta_{vb}\ R_{vb}$. Some of this torque accelerates the low speed sheave. This leaves $(T_m - T_{Ih})\ \eta_{vb}\ R_{vb} - T_{Il}$, where

$$T_{Il} = 12\ I_{ls}\ \alpha_c\ R_{gb}\ /g_c \qquad (7)b$$

Torque at the Crankshaft

Use η_{gb} and R_{gb} to convert the remaining torque to the crankshaft. This is $[(T_m - T_{Ih})\ \eta_{vb}\ R_{vb} - T_{Il}]\ \eta_{gb}\ R_{gb}$.

$$T_{Ig} = 12\ I_{gb}\ \alpha_c\ /g_c \qquad (7)c$$

is the torque that accelerates the gears, shafts, and bearings in the gearbox.

Net Crankshaft Torque

This gives the net torque at the crankshaft.

$$T_c = [(T_m - T_{Ih})\ \eta_{vb}\ R_{vb} - T_{Il}]\ \eta_{gb}\ R_{gb} - T_{Ig} \qquad (8)a$$

It is convenient to separate T_m from the inertial torques.

$T_c = T_m \eta_{vb} \eta_{gb} R_{vb} R_{gb} - T_{Idt}$, where..............................(8)b

$T_{Idt} = T_{Ih} \eta_{vb} \eta_{gb} R_{vb} R_{gb} + T_{Il} \eta_{gb} R_{gb} + T_{Ig}$......................(8)c

Gravity Torque, Crank and Counterweight

The crank and counterweight combination introduces two torque terms. The torque needed to accelerate the inertial mass is

$T_{Ic} = 12\ I_c\ \alpha_c\ /g_c$,..(9)a

and the gravity torque is $T_{cwt} \sin(\theta + \tau)$. This gives

$T_m \eta_{vb} \eta_{gb} R_{vb} R_{gb} - T_{Idt} - T_{Ic} + T_{cwt} \sin(\theta + \tau)$....................(9)b

Net Horsehead Forces

The remainder is the equivalent of the forces acting on the horsehead.

$T_{pr} = F_{tf} (F_{pr} - F_{su} + T_{Ib}/L_A)\ /\eta_{pu}$,................................(10)a

where the inertial torque of the beam is

$T_{Ib} = 12\ I_b\ \alpha_b\ /g_c$...(10)b

Net Motor Torque Equation

The motor torque equation is

$T_m \eta_{vb} \eta_{gb} R_{vb} R_{gb} - T_{Idt} - T_{Ic} + T_{cwt} \sin(\theta + \tau) - T_{pr} = 0$...........(11)

Efficiency Considerations, Motoring vs. Generating

Note: This form of the T_m equation assumes the motor is driving the polished rod. When the rod is using the motor as a generator, it is necessary to divide by η_{vb}, divide by η_{gb}, and multiply by η_{pu}. This is because the mechanical efficiencies continue to consume a portion of the torque.

Motor Current Equation

The goal is to balance peak currents. The T_m equation needs to be converted to motor current (I). The equations are

$P_{mot} = 1.732\ I\ \eta_m\ F_{pf}\ \eta_{pi}\ /746$, and.............................(12)a

$P_{mot} = T_m\ N\ R_{vb}\ R_{gb}\ /63{,}025$...................................(12)b

1.732 becomes 1.0 for single phase electricity. Solving for T_m gives

$$T_m = I \frac{1.732\ V\ 63{,}025\ \eta_{pi}\ \eta_m\ F_{pf}}{746\ R_{vb}\ R_{gb}\ N} \qquad (12)c$$

This assumes constant voltage, balanced phases, and a constant phase inverter efficiency. Substituting for T_m and solving for I gives

$$I \frac{1.732\ V\ 63{,}025\ \eta_{dt}}{746} \frac{\eta_m\ F_{pf}}{N} = T_{cwt} \sin(\theta + \tau) - T_{Inet} \qquad (13)a$$

Here, $T_{Inet} = T_{Idt} + T_{Ic} + T_{pr}$ .... (13)b

Torque Required to Balance Peak Currents

There is some dT_{cw} that forces the peak upstroke current (I_u) to be equal to the peak downstroke current (I_d). There is a unique value of N, η_m, and F_{pf} for each I. When the peaks are equal, the coefficient on the left hand side cancels. This leaves

$$dT_{cw} = \frac{T_{Inetu} - T_{Inetd}}{\sin(\theta_u + \tau) - \sin(\theta_d + \tau)} - T_{cwt} \qquad (14)$$

Iteration Considerations

Moving the counterweight sometimes changes the timing of one of the current peaks. A simple iteration resolves this. Calculate new values for each I using dT_{cw}. If A peak occurs at a different θ_u or θ_d, it will be necessary to update T_{cwt} and I_c. Then recalculate dT_{cw}, T_{Inetu}, and T_{Inetd}. This assumes the surface dynamometer card remains the same. Be sure to add the two dT_{cw} values before calculating the ideal L_{bc}.

APPENDIX D, CRANKSHAFT & MOTOR TORQUE vs. MOTOR CURRENT EQUATIONS

The T_c equation simplifies to the Gibbs and the Svinos (net torque) equation when η_{pu} is unity. If I_c and I_b are zero, the Gibbs and the Svinos (net torque) equation simplifies to the API SPEC 11E equation.

Eq. (12)c indicates that balancing the current peaks automatically balances the motor torque peaks. This is because each I has unique values for η_m, F_{pf}, and N. When I_u equals I_d, these unique values cancel. (Assume η_{vb}, η_{gb}, η_{pu}, and η_{pi} are constant.)

The $(T_c)_{max}$ probably balances at a slightly different crank angle. This is because $(T_c)_{max}$ ignores the drive train inertias and efficiencies. The difference between $(T_c)_{max}$ and $(T_m)_{max}$ may be insignificant.

APPENDIX E, ESTIMATES FOR MISSING VENDOR DATA

To solve for dT_{cw}, the unit geometry, the efficiencies, the articulating mass moment of inertia, and the rotating mass moment of inertia must be known.

Laine discusses inertia and efficiency estimates.[18] Svinos and SPEC 11E describe the equations needed to back calculate crank angles and torque factors from polished rod position.[10,13]

Efficiency, Drive Train

$\eta_{dt} = \eta_{gb} \ \eta_{vb}$ ....(15)a

$0.5 \leq \eta_{dt} \leq 0.83$

Efficiency, Gear Box[20]

$0.75 \leq \eta_{gb} \leq .88$

Efficiency, Phase Inverter

I have no data at this time. I can only guess the following

$0.5(?) \leq \eta_{pi} \leq 1.0(?)$

Efficiency, Pumping Unit

Pumping units and double reduction gear boxes both have three sets of bearings, which implies similar efficiencies. Since the gear box also has gear tooth contacts, pumping unit efficiency may be higher than the gear box.

$\eta_{gb} \leq \eta_{pu} \leq (\eta_{gb})^{0.5}$ ....(15)b

$0.75 \leq \eta_{pu} \leq 0.94$

Efficiency, V-belt[20]

$0.66 \leq \eta_{vb} \leq 0.94$

Inertia, Articulating (Beam) Mass Moment of

The mass moment of inertia of a uniform beam about its center of mass approximates the articulating inertia of the beam, equalizer, and horsehead about the Samson bearing.

$I_b = M_{be} \ (L_{be})^2 \ /12 = F_{WLbe} \ (g_c \ /g) \ (L_{be})^3 \ /12$ ....(16)a

Lufkin's Tables of conventional-pumping-unit articulating inertias ranges from 60 to 140% of $F_{WLbe} \ (g_c \ /g) \ (L_{be})^3 \ /3$. The smaller percentages apply to the larger pumping units; the larger percentages apply to the smaller pumping units. 40% error significantly affects the calculations. I recommend increasing the estimated I_b for the larger pumping units, and vice versa. (Vendor data are preferred.)

Inertia, Counterweight Rotating Mass Moment of

The inertia of an individual counterweight about the crankshaft is

$I_{cw} = (g_c \ /g) \ W_{cw} \ (L^2_{cm,cw} + r^2_{g,cw} \ /144 \ /2)$ ....(16)b

Inertia, Crank Rotating Mass Moment of

A uniform bar approximates the inertia for one crank.

$$I_{cr} = W_{cr} (g_c /g) [L^2_{cm,cr} +(L_{cr}/2)^2 /3]$$

Inertia, Motor Rotating Mass Moment of

The motor rotor inertia needs to be adjusted to the crank speed.

$$I_{mot,N} = I_{mot} R^2_{gb} R^2_{vb} \qquad (16)c$$

Inertia, Sheave Rotating Mass Moment of

Since the sheave rim contains most of the mass, the sheave mass moment of inertia (relative to crank speed) is approximately

$$I_{hs,N} = (g_c /g) W_{hs} (d_{hs} /12 /2)^2 R^2_{gb} R^2_{vb} = I_{hs} R^2_{vb} R^2_{gb} \qquad (16)d$$

for the high speed sheave and for the slow speed sheave it is

$$I_{ls,N} = (g_c /g) W_{ls} (d_{ls} /12 /2)^2 R^2_{gb} = I_{ls} R^2_{gb} \qquad (16)e$$

Sheave inertias are significant because $R^2_{gb} R^2_{vb}$ can exceed 10,000:1.

$$0.3 \leq I_{hs,N} /(R_{gb} R_{vb})^2 = I_{hs} \leq 3$$
$$5.0 \leq I_{ls,N} /(R_{gb} \quad)^2 = I_{ls} \leq 500.$$

Inertia, V-belt "Rotating" Mass Moment of

$$I_{vb,N} = F_{WLvb} L_{vb} (g_c /g) (R_{gb} R_{vb} d_{hs} /2 /12)^2$$
$$= F_{WLvb} L_{vb} (g_c /g) (R_{gb} \quad d_{ls} /2 /12)^2 \qquad (16)f$$

Inertia, Total Rotating Mass Moment of

$$I_r = I_{cw} +I_{cr} +I_{gb} +I_{mot,N} +I_{hs,N} +I_{ls,N} +I_{vb,N} \qquad (16)g$$

Motor KVA vs. rpm

$$\bar{P}_{kva} = (\text{phase})^{0.5} V A /1000 \qquad (17)a$$

Amperage incorporates the power factor and electrical efficiency.

Motor Output Torque vs. Motor rpm

$$T_m = 63,025 \bar{P}_{kva} \eta_m F_{pf} /(N R_{gb} R_{vb} 0.7456999) \qquad (17)b$$

Speed Ratio, Gear Box

$28 \leq R_{gb} \leq 32$ (double reduction)

Speed Ratio, V-belt

$2 \leq R_{vb} \leq 5$

Torque, Maximum due to Counterweight

$$T_{cw} = ncw\ W_{cw}\ L_{cm,cw}\ 12 \qquad (18)a$$
$0 \leq ncw \leq 8$

Torque, Maximum due to Crank

$$T_{cr} = 2\ W_{cr}\ L_{cm,cr}\ 12 \qquad (18)b$$

NOMENCLATURE

A	= amperage, amp
A_p	= pump area, in^2
C_r	= rod, viscous damping factor, 1/s
d_{hs}	= pitch diameter, high speed sheave, in
d_{ls}	= pitch diameter, low speed sheave, in
F_{fr}	= friction force on sucker rods, lbf
F_{pf}	= electric motor power factor, nondimensional
F_{pr}	= polished rod force, lbf
F_{su}	= structural unbalance (down = pos in SPEC 11E), lbf
F_{tf}	= torque factor, in
F_{WLbe}	= beam, lineal specific weight, lbf
F_{WLvb}	= v-belt, lineal specific weight, lbf/ft
g	= gravity acceleration = 32.174 048 556, ft/s^2
g_c	= gravity constant = 32.174 048 556, $lbm\text{-}ft/lbf/s^2$
I	= current, amps
I_b	= beam mass moment of inertia, $lbm\text{-}ft^2$
I_c	= crank & counterweight mass moment of inertia, $lbm\text{-}ft^2$
I_{cr}	= crank mass moment of inertia, $lbm\text{-}ft^2$
I_{cw}	= counterweight mass moment of inertia, $lbm\text{-}ft^2$
I_d	= downstroke current, amps
I_{gb}	= gear box mass moment of inertia, $lbm\text{-}ft^2$
I_{hs}	= high speed sheave mass moment of inertia, $lbm\text{-}ft^2$
$I_{hs,N}$	= hss mass moment of inertia relative to crankshaft, $lbm\text{-}ft^2$
I_{ls}	= low speed sheave mass moment of inertia, $lbm\text{-}ft^2$
$I_{ls,N}$	= lss mass moment of inertia relative to crankshaft, $lbm\text{-}ft^2$
I_{mot}	= motor mass moment of inertia, $lbm\text{-}ft^2$
$I_{mot,N}$	= motor mass moment of inertia relative to crankshaft, $lbm\text{-}ft^2$
I_r	= total rotating mass moment of inertia, $lbm\text{-}ft^2$
I_u	= upstroke current, amps
I_{vb}	= v-belt mass moment of inertia, $lbm\text{-}ft^2$
$I_{vb,N}$	= v-belt mass moment of inertia relative to crankshaft, $lbm\text{-}ft^2$
$L_{cm,cw}$	= length from crankshaft to counterweight center of mass, ft
L_A	= API length, Sampson bearing to polished rod, in
L_{bc}	= balanced counterweight location, in
L_{be}	= length of beam, ft
$L_{cm,cr}$	= length from crankshaft to crank center of mass, ft
$L_{cm,cw}$	= length from crankshaft to counterweight center of mass, ft

L_{cr} = crank length, ft
L_{vb} = v-belt length, ft
M_{be} = beam mass, lbm
M_r = rod mass, lbm
N = polished rod stroke rate, spm
ncw = number of counterweights
P_{fric} = viscous rod friction horsepower, hp
P_{mot} = motor nameplate horsepower, hp
P_{kva} = average billable power, kw
q = production rate, bpd
$r_{g,cw}$ = counterweight radius of gyration about center of mass, in
R_{gb} = gear box speed ratio, *e.g.*, 29/1, nondimensional
R_{vb} = v-belt speed ratio, *e.g.*, 4/1, nondimensional
S = surface stroke length, in
S_p = pump stroke, in
T_c = net crank torque, in-lbf
T_{cr} = maximum torque at crankshaft due to crank, in-lbf
T_{cw} = maximum torque at crankshaft due to counterweight, in-lbf
T_{cwt} = maximum crank and counterweight torque, in-lbf
T_m = (net) motor torque, in-lbf
V = voltage, V
v_r = polished rod velocity, ft /s
$\bar{v}_r$ = average polished rod velocity, ft /s
W_{cr} = crank weight, lbf
W_{cw} = counterweight weight, lbf
W_{hs} = high speed sheave weight, lbf
W_{ls} = low speed sheave weight, lbf

α_b = beam angular acceleration, rad /s^2
α_c = crank angular acceleration, rad /s^2
η_{dt} = drive train (combined gear and v-belt) efficiency
η_{gb} = gear box efficiency, nondimensional
η_m = motor efficiency, nondimensional
η_{pi} = phase inverter efficiency, nondimensional
η_{pu} = pumping unit efficiency, nondimensional
η_{vb} = v-belt efficiency, nondimensional
θ_d = crank angle at peak current on downstroke, rad
θ_u = crank angle at peak current on upstroke, rad
τ = crank counterweight phase angle

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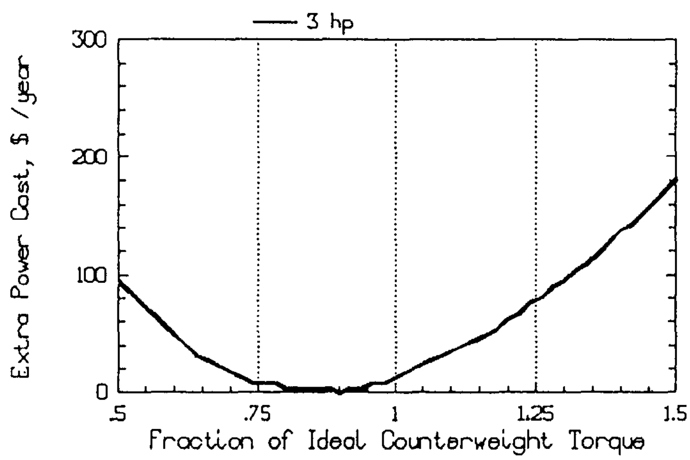

Figure 1 - Extra cost vs. motor hp and CWT torque

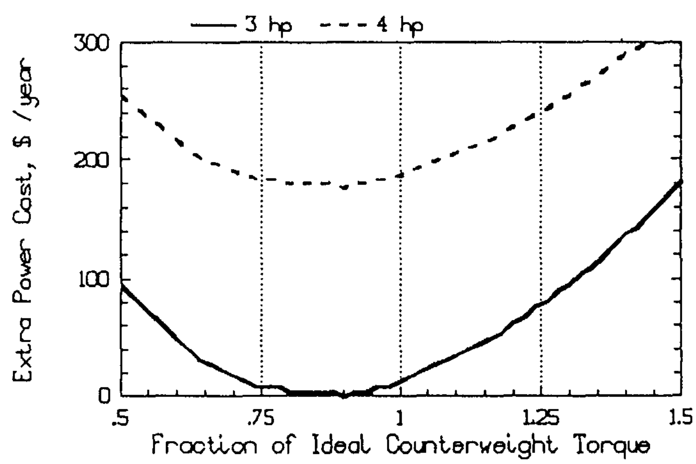

Figure 2 - Extra cost vs. motor hp and CWT torque

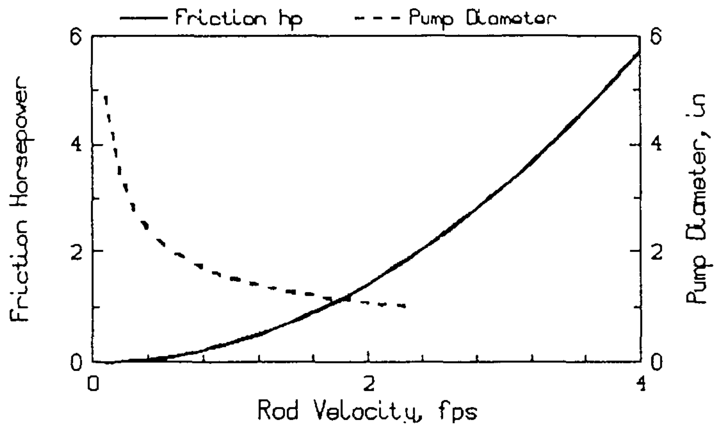

Figure 3 - Rod friction vs. pump area and rod velocity (Q = 100 bpd, L = 5000 ft, S = 60 in, N = 10 spm)

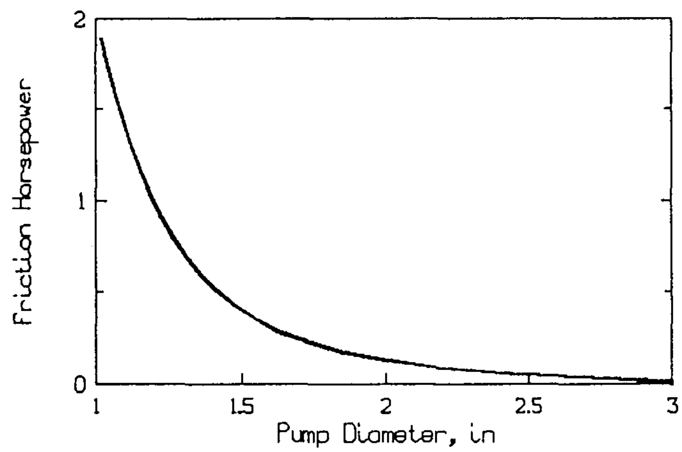

Figure 4 - Rod friction vs. pump area (Q = 100 bpd, L = 5000 ft, S = 60 in, N = 10 spm)

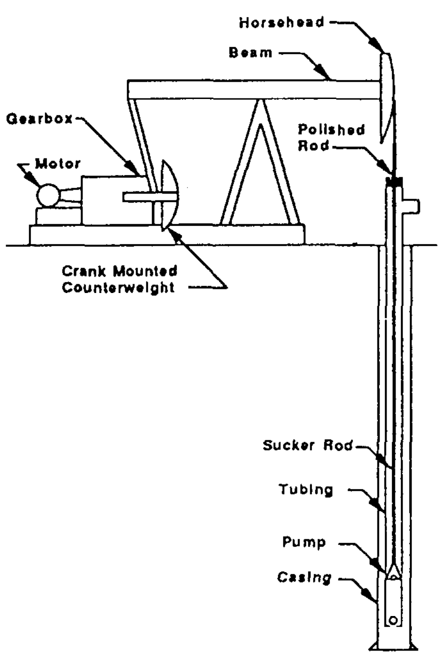

Figure 5 - Typical sucker rod system

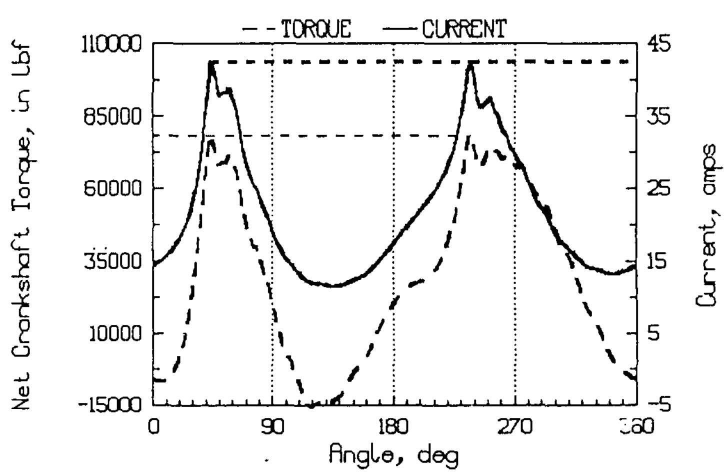

Figure 6 - Crank torque and phase current vs. angle

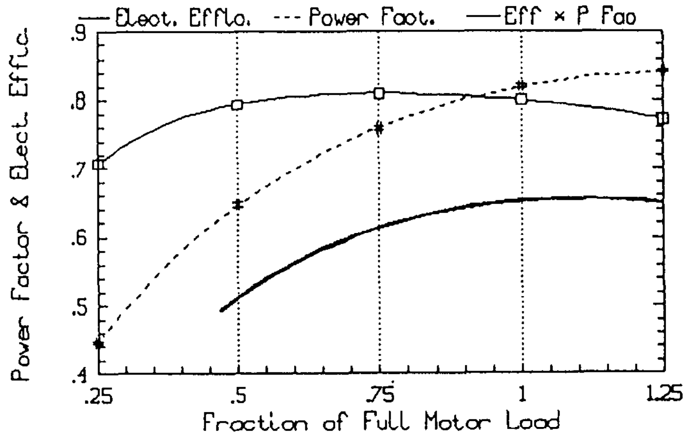

Figure 7 - Nominal motor properties
NEMA D, 3 to 7.5 hp

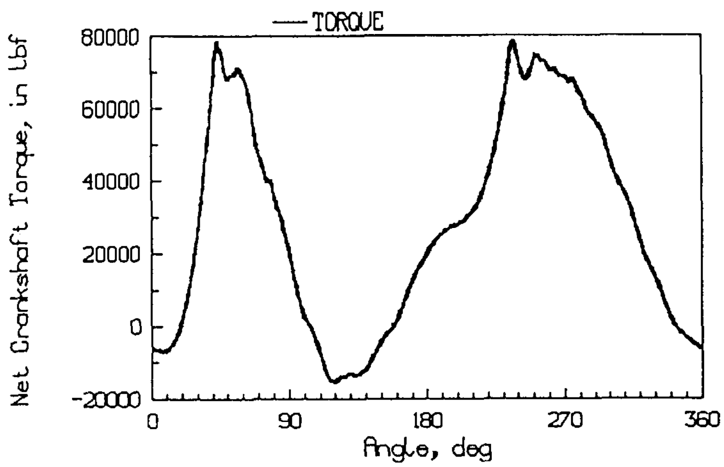

Figure 8 - Crank torque vs. crank angle

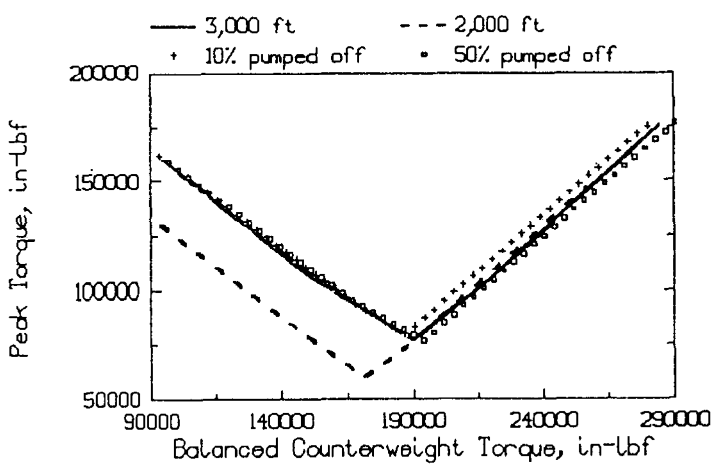

Figure 9 - Peak torque by net lift

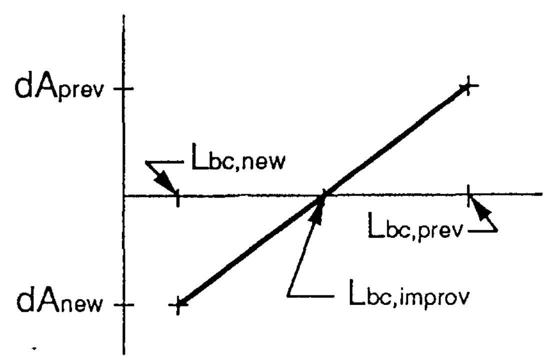

Figure 10 - Interpolation for ideal balanced counterweight location, $L_{bc,improv}$