COUNTERWEIGHT TORQUE (and COUNTERBALANCE EQUIVALENT) versus GEARBOX AND MOTOR LOADS

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ABSTRACT

Capital and power consumption costs for sucker-rod pumps depend, in part, on the counterweight torque. Counterweight torque is the primary variable because peak torque depends on how well the well is balanced. The secondary variables affect the ideal counterweight torque (motor slip, pumpjack efficiency, net lift, and torque equation). High slip motors allow more crank speed variation. Pumpjack efficiency (preventive maintenance) affects power consumption. Net lift changes the net crankshaft torque as the well pumps off. The general form of the net torque equation more accurately converts polished rod forces into equivalent moments at the crankshaft than the simplified counterbalance equivalent equation.

Each combination of the primary and secondary variables may have a unique counterweight torque that minimizes the peak net crankshaft torque by equalizing the largest upstroke and downstroke torques. The peak torque affects capital costs for the assumed operating and preventive maintenance conditions by determining the smallest acceptable gearbox and motor sizes. Fully loaded motors use less electricity because power factor and electrical efficiency go up with the ratio of brake horsepower to motor horsepower.

In short, it is possible to cut capital and power consumption expenses by managing the difference between the ideal and the actual counterweight torques throughout the life of the well.

Pumpjacks are counterbalanced for an ideal operating condition that considers motor slip, mechanical efficiency, and a fluid level that falls to the pump. The latter element indicates that normally operating pumpjacks have multiple ideal counterweight torques. Careful torque predictions for the anticipated range of operating conditions can save capital by reducing gearbox and motor requirements. Specifying the smallest acceptable motor reduces power consumption because the power factor increases when the nameplate rating is close to the brake horsepower.

Brake horsepower is affected by mechanical efficiency and counterweight torque. High mechanical efficiency comes with maintenance. A preventive maintenance goal is to reduce total operating cost by lowering power consumption. Thus power consumption and maintenance costs are an economic tradeoff for the life of the well. Counterweight torque also affects power consumption because it takes a bigger motor to handle the higher peak torques caused by improper counterweight torque. A well-balanced pumpjack uses less power because the ratio of brake to motor horsepower is higher when the crank is ideally counterweighted. Calculated counterweight torque accuracy depends on the selected equation. The net crankshaft torque equations are more accurate than the simplified counterbalance equivalent equations.

BACKGROUND

Balancing the counterweight equalizes the largest upstroke and downstroke torques at the crankshaft. Equalizing the largest torques allows us to specify the smallest suitable gearbox and motor by minimizing the peak torque. The minimum torque goal dates from the 1920's and may still apply.

Griffin's peak torque study finds an average 19.7% difference between the ideal and the actual counterweight torques for 77 wells.[1] Furthermore, over 50% of the 77 pumpjacks are out of balance. Griffin uses the API RP 11L from of the counterbalance equivalent equation.[2]

Counterbalance equivalents can be misleading because it includes several assumptions. These assumptions, which have little impact on slide rule calculation accuracy, become significant with personal computers. PC's provide speed and accuracy that warrants using the general net torque equation.

The generalized equation works equally well with measured and simulated dynamometer cards. Simulated cards introduce important flexibility because the simulator can model a range of mechanical efficiencies, motor slips, and fluid levels.

$$T_n = F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)/\eta_{pj} - T_{cwt}\ \sin(\theta + \tau) + I_c\ \alpha_c \quad \text{......(1)}$$

when $F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)$ is positive. Multiply by pumpjack efficiency if $F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)$ is negative because friction consumes some of the polished rod work before it reaches the crankshaft. See Appendix A for the nomenclature. Development of the net torque equation is in Appendix B. The pumpjack vendor normally provides F_{tf}, F_{su}, I_b, L_A, τ, and I_c values. It is possible to develop simple equations for estimating I_c and I_b.

The ideal counterweight torque equalizes the net, peak, upstroke and downstroke crankshaft torques. The procedure calculates the net torque for each simulated or measured timestep. The solution is iterative because changing T_{cwt} can shift the peak torque to another timestep.

Counterweight torque affects motor size because the traditional equation for predicting the minimum motor rating accounts for the fluctuating torque requirement, pumpjack efficiency, and drive train efficiency.[3]

$$\bar{P}_{mot} \leq P_{prhp}\ F_{clf}\ /(\eta_{pj}\ \eta_{dt}) \quad \text{......(2)}$$

where polished rod horsepower is the area of the surface dynamometer card. The trapezoid rule is sufficiently accurate for most polished rod horsepower calculations.[4]

$$P_{prhp} = \sum_{m=1}^{nts} [(F_{pr,m+1} + F_{pr,m})\ (U_{pr,m+1} - U_{pr,m})\ /(2\ 550\ dT_{m1}) \quad \text{......(3)}$$

$$F_{pr,nts+1} = F_{pr,1} \quad \text{......(4)}$$

$$U_{pr,nts+1} = U_{pr,1} \quad \text{......(5)}$$

$$dT_{m1} = T_{m+1} - T_m \quad \text{......(6)}$$

$$T_{nts+1} = T_{1_{nts}} + T_{per} \quad \text{(7)}$$

$$T_{per} = \Sigma_{m=1} (dT_{m1}) \quad \text{(8)}$$

The cyclic load factor is the ratio of the root mean square and the average net torques.

$$F_{clf} = [\sum_{m=1}^{nts} (T^2_{n,m}/nts)]^{.5} / \sum_{m=1}^{nts} (T_{n,m}/nts) \quad \text{(9)}$$

Typical mechanical efficiencies for new and worn pumpjacks and gearboxes are recommended by Gipson and Swaim.[5]

Counterweight torque affects power consumption because oversized motors tend to lower power factor and electrical efficiency. See Fig E1, Appendix E.

$$\bar{P}_{kva} = P_{prhp} /(\bar{F}_{pf}\ \bar{\eta}_{mot}\ \eta_{pj}\ \eta_{dt}) \quad \text{(10)}$$

METHODOLOGY

This methodology demonstrates a technique for evaluating how the difference between the ideal and the actual counterweight torques affects the peak crankshaft torque. Motor slip, mechanical pumpjack efficiency, net lift, and torque equation also impact the ideal counterweight torque(s), peak torque(s), gearbox size, motor size, and power consumption for a common well. (Table 1) The outlined steps apply to existing wells and to new wells.

The study well closely matches the assumptions in the derivation of the counterbalance equivalent equation for predicting the ideal counterweight torque in order to check counterbalance equivalent accuracy. The well uses a conventional 114D-143-64 pumpjack running at 4.9 spm with a 64 in stroke to lift 90 bpd with 3000 ft of 0.75 in steel rod and a 1.75 in pump. The drive train efficiency is 95 percent. (Appendices C, D)

The complete methodology starts by history match field data. Simulator runs then predict the surface dynamometer card so a torque analysis program can calculate the ideal counterweight torque for each option. (Table 1) The torque analysis program then calculates the torques and horsepowers for a range of unbalanced counterweights. The method for an existing well has no hardware options; so, it is shorter. More time is needed for a new well even though the history match is skipped. The options (pumpjack, stroke, spm, pump, rod, etc) need more simulations.

The sucker-rod simulator used in this work solves the damped wave equation and the fluid inertia equations.[6] The simulator models several boundary conditions including net lift from zero to 50% pumped off, anchored and unanchored tubing, class I (conventional, special geometry) and class III (Mark II[tm]) pumpjacks, and motor slip. The torque analysis program uses the general form of the net crankshaft equation. (Appendix B) The general form finds the ideal counterweight torque by equating the largest upstroke and downstroke torques. This is an iterative solution because changing the counterweight torque affects the nonlinear portions of the net torque equation. Once the ideal counterweight torques is known the program finds the net torque, peak torque, cyclic load factor, brake horsepower, and power consumption for actual counterbalance torques from 50% to 150% of the ideal counterweight torque.

RESULTS

This section summarizes the graphical and tabular results (Figs 2-11 & Tables 2-5). The discussion section elaborates on the significance of the results.

Counterweight torque influences the gearbox and motor size for each sucker-rod installation. The gearbox must carry the peak crankshaft torque, and peak torque depends on a number of factors including motor slip, mechanical efficiency, net lift, and the torque equations. The smallest peak torque occurs when the ideal counterweight torque equalizes the largest upstroke and downstroke torques. Higher peak torques occur when the counterweight setting deviates from the ideal. Ideal counterweight predictions vary with the simplifying assumptions used to develop the torque equations. Torque affects the motor size because torque and horsepower are proportional. An oversize motor consumes extra electricity because the power factor and the electrical efficiency are low. An undersize motor fails by stalling or overheating.

Motor Slip

The smallest peak torque depends on motor slip. The smallest peak torque for the 7.5% slip motor (97,800 in-lbf) is 27% higher than the 77,500 in-lbf minimum with 18.5% motor slip. The peak torque increases as the counterweight torque deviates from the ideal. The net lift is 3,000 ft and the pumpjack efficiency is 90% for both examples. (Fig 2)

Pumpjack Efficiency

The smallest peak torque depends on pumpjack efficiency. The smallest peak torque for 80% mechanical efficiency (110,000 in-lbf) is 12% higher than the 97,800 in-lbf minimum with 90% efficient pumpjack. The motor slip is 7.5% and the net lift is 3,000 ft for both examples. (Fig 3)

Net Lift

The smallest peak torque depends on net lift. The smallest peak torque (60,700 in-lbf) is 21% lower than the 77,000 in-lbf minimum for the 50% pumpoff condition. The smallest peak torques are similar for 50%, 10%, and 0% pumpoff (77,000, 80,000, and 77,500 respectively) because the nominal net lift is 3,000 ft. The motor slip is 18.5% and the pumpjack is 90% efficient for the four examples. (Fig 4)

Counterbalance Equation

The predicted ideal counterweight torque depends on the selected equation (the simplified counterbalance equivalent equation vs. the general net torque equation). The counterbalance effect approach is four percent low (182,000 in-lbf) and the RP 11L version of the counterbalance effect approach is one percent high (192,000 in-lbf) compared to the general net torque equation. This highlights the hazard of relying on the counterbalance equivalent approach; the error is four (4) percent for a case that satisfies the assumptions behind the counterbalance equivalent approach. Clearly counterbalance equivalent will give *misleading* results for many wells. (Fig 5)

The general equation for net crankshaft torque predicts a different ideal counterweight torque for each net lift. Thus, the actual counterweight torque for a given sucker-rod installation is only truly balanced for a single fluid level. The general torque modifies API Specification 11E to include the flywheel effects of the rotating and articulating masses.[7] The motor slip is 18.5% and the pumpjack is 90% efficient for the four examples. (Fig 6)

Gearbox Rating

Selecting the gearbox is easy when one size covers a wide range of peak torques. Unfortunately, the peak torque for the example is only below 80,000 in-lbf when the actual counterweight torque is within two percent of the ideal. The two percent range exceeds simulator accuracy, unless field data confirms the predictions; so, an 80,000 in-lbf gearbox is too small. The motor slip is 18.5%, the net lift is 3,000 ft, and the pumpjack efficiency is 90%. (Fig 7)

Motor Size

Polished rod horsepower depends on motor slip and net lift, but not on pumpjack efficiency. The polished rod horsepower (1.98 hp) is 19% lower for the 18.5% slip motor than for then the 7.5% slip motor (2.44 hp) for a 3,000 ft net lift because the higher slip smoothes the polished rod loads. Increasing the fluid level in the annulus and restricting the amount of fluid in the pump barrel reduce the required work. The respective pumpjack and drive train efficiencies are 90% and 95%. (Fig 8)

The motor size depends on the net lift and on the difference between the ideal and actual counterweights. Changing the counterweight torque from 262,000 in-lbf to 265,000 in-lbf increases the motor size from 5 hp to 7.5 hp when the net lift is 3,000 ft. The difference is that the 1.98 polished rod horsepower (for the 3,000 ft lift) and the increased cyclic load factor (2.14 to 2.18) combine with the mechanical efficiencies (90% pumpjack and 95% drive train) to exceed 5 hp. The 10% pumpoff condition experiences a similar increase at a slightly higher counterweight torque because the pump barrel is partially empty. The motor slip is 18.5 percent. (Fig 9)

The motor runs at half load (5 hp), and the peak torque is below the NEMA D stall rating (275% full load torque). The peak torque is below 200% (120,000 in-lbf) while the counterweight torque is within 25% (140,000 to 240,000 in-lbf). The lift is 3,000 ft. (Fig 10)

Counterweight Torque

Select the ideal counterweight torque for the operating conditions with the higher peak torques; 185,000 in-lbf is a better choice than 170,000 in-lbf because the predicted peak torque for the expected range of operating conditions (85,000 in-lbf) is 9% lower for the 185,000 in-lbf setting. The motor slip is 18.5%, the pumpjack efficiency is 90%, and the drive train efficiency is 95%. (Fig 11)

The ideally counterbalanced crank uses a smaller gearbox (80,000 in-lbf) than the 125% balanced crank (143,000 in-lbf). (Tables 2, 4, and 5)

Operating Cost

Power consumption and capital cost are affected by motor size and by counterweight torque. An ideally balanced sucker-rod installation can use a smaller motor (3 or 4 hp) that uses less power (2.74 to 3.18 KVA) than the identical unit counterbalanced to 125% of the ideal. The latter balance needs a larger motor (5 or 7.5 hp) because the peak torque is higher. Further, the bigger motor uses more power (4.74 to 5.75 KVA) because the light load reduces the power factor and electrical efficiency. The lifting cost difference between 5 and 7.5 hp motors is $0.017 per barrel. The polished rod horsepower and production are identical for both motors. (Fig E1, Tables 2, 3, 4, and 5)

DISCUSSION

The results follow the steps for evaluating gearbox, and motor sizes as affected by counterweight torque. The secondary variables are motor slip, pumpjack efficiency, net lift, and counterbalance equations.

Motor Slip

Motor slip smooths the torque fluctuations when the net torque is high by slowing down. Deceleration further reduces the motor load by drawing torque from the flywheel action of the rotating and articulating (rocking) masses. This positive feedback cycle lowers loads and allows smaller gearboxes and motors. Naturally, smaller rotating equipment allows more deceleration by further reduces the rotating inertia. Low inertia sheaves are important because one lbm ft^2 of mass moment of inertia at the motor is equivalent to 40,000 lbm ft^2 at the crankshaft when the rpm to spm ratio is 200. (Appendix B) Since low inertia improves high slip applications it is important to use the smallest suitable high slip motors. This motor may operate close to the manufacturer's ambient temperature and temperature rise limitations.

Pumpjack Efficiency

Since low slip motors tend to increase motor size by holding the spm speed within tighter limits it might be tempting to oversize the motor. This may well reduce repair and maintenance costs, but it also increases power consumption. Thus, equipment selection impacts a tradeoff between maintenance and operating expenses. S Gault[8] reports that shifting lubrication responsibility from pumpers to a designated contractor can significantly increase reliability. Such reliability is essential if gearbox, motor, and counterweight torque decisions assume high pumpjack and drive train efficiencies.

Net Lift

Net lift is moderately important to minimum peak torque. The lowest peak torque for 2,000 ft net lift is 19,000 in-lbf less than for the pumpedoff conditions. (Fig 4) This shows that smaller gearbox and motor sizes are appropriate when inflow performance exceeds the installation's capacity. However, it may be possible to increase capacity by changing the spm, stroke, pump size, or time clock settings. The total number of choices is prohibitive. Fortunately, Gipson[9] suggests a number of shortcuts, and slow speeds is near the top of the list. Slow speed saves power, as demonstrated by R. Gault[10], S. Gault and Takacs.[11] The reason is that friction horsepower due to viscous drag on the rods increases with speed. Running more hours per day, with a longer stroke, or with a larger pump all increase production without aggravating the friction horsepower. Conversely, if a well pumps off too fast, it may be possible to reduce lifting costs by decreasing spm, stroke, or pump size. Such operating changes may lead to new ideal conterweight torques.

In summary, if the inflow performance is such that 2,000 ft really is the maximum net lift, then an 80,000 in-lbf gearbox (Fig 4) and a 3 hp motor (Fig 9) are sufficient when the (ideal) counterweight torque is 170,000 in-lbf (Fig 11).

Counterbalance Equations

The counterbalance equivalent equation uses statics to estimate the peak torque and the ideal counterweight torque. Counterbalance equivalent gives the best results for low speed, deep wells with fully stressed rods and conventional pumpjacks. Low speed reduces the dynamic loads relative to the static forces because mass times low acceleration equals low force. Deep wells reduce the dynamic fluid load at the traveling valve relative to the hydrostatic fluid load because fluid friction has more distance to damp the pressure waves. Fully stressed rods reduce the dynamic rod force relative to the static weight of the buoyed rods by stretching. This reduces the dynamic rod force for all pumpjacks. Conventional pumpjacks are most likely to be symmetrical. This means the torque factor is half the stroke length, an integral assumption in the counterbalance equivalent derivation. (Appendix B)

The example well intentionally satisfies the counterbalance equivalent assumptions in order to *highlight the hazards*. Erroneous results show that the counterbalance equivalent approach to crankshaft torque is clearly an approximation that is questionable even when the well meets the assumptions behind the approach. (Fig 5) The RP 11L version improves counterbalance equivalent accuracy by including 94% pumpjack efficiency; however, RP 11L continues to neglect dynamic fluid loads and the flywheel effect of the rotating and articulating (rocking) masses. RP 11L is also limited to nominal motor slip, nominal viscous drag, and conventional pumpjack assumptions. These limiting assumptions coupled with the simplifying assumptions needed to develop the counterbalance equivalent equation from the general torque equation make it clear that the general torque equation gives much better results. (Appendix B)

The choice of torque equations depends on the available computing capacity. The algebraic counterbalance equivalent equation is suitable for slide rule calculations, and the API RP 11L equation includes pumpjack efficiency with one more slide rule multiplication. Personal computer availability mean the repetitious calculations of the general net torque equation are now sufficiently fast and accurate to supercede the simplified, counterbalance method.

Non-Ideal Counterweight Torque

Non-ideal counterweight torques tend to require larger gearboxes and motors to handle the peak torque increases. The larger hardware sizes tend to increase power consumption by lowering power factors, electrical efficiencies, and speed variation. Increasing the counterweight torque to 125% of the ideal for the example well increases the gearbox by one size, increases the motor by two sizes, and adds about 50% to the power consumption. (Tables 3, 4, and 5)

Normal operating variations such as transient inflow performance, waterflood breakthrough, degrading mechanical efficiency, and voltage fluctuations modify the ideal counterweight torque. Judicious selection of the actual counterweight torque can limit the range of peak torques. Careful balancing limits the peak torque and helps keep the loads within the capacities of the gearbox and motor.

Limitations

Each combination of rod, pump, pumpjack, speed, stroke, and viscosity contributes to counterweight torque. API BULL 11L3 is a good starting point[12] for selecting reasonable alternatives. Several authors offer guidance for minimizing the simulation options (Gipson and Swaim, Gipson, R. Gault, S. Gault, and Takacs).

Simulators can model a wide range of conditions, and accuracy is good when field data is available for history matching. Unfortunately there is no measured data for a new well. New wells present three options each of which has advantages: oversize the equipment, be willing to change the equipment once field data is available, or test the well with a rental system. Oversize equipment not only costs more, but it probably wastes power with low electrical efficiency and power factor. A better choice is to temporarily use one set of oversize hardware for all new wells. Each option provides field data for history matching. (The economics of renting vs. owning a temporary system exceed the scope of this paper.)

Maintenance, repair, and power consumption records are important factors that are beyond the scope of this study. Similarly, the cost of hardware options (such as ultra high slip vs. 5 to 8% slip NEMA D motors) are important. This paper assumes capital cost, operating expense, quality, and reliability are proportional to size.

CONCLUSIONS

The example well introduces a method for dealing with non-ideal counterweight torques for motor slip, pumpjack efficiency, and net lift.

Motor Slip

The high slip motor reduces the gearbox and motor sizes by varying the crank speed as the net torque load fluctuates. Since load smoothing is more pronounced when the mass moments of inertia are smaller, it is important to limit the rotating inertia by using smaller sheaves with high capacity vbelts.

Pumpjack Efficiency

Mechanical efficiency directly affects net torque and brake horsepower because higher pumpjack and drive train efficiency allow smaller gearboxes and motors. Since efficiency affects preventive maintenance, it is important for engineering to consider production's maintenance philosophy when sizing gearboxes and motors. It is possible to reduce capital and power consumption expenses when efficiency is high and when the maximum difference between the actual counterweight torque and the ideal counterweight torque(s) is small.

Net Lift

Each net lift has its own ideal counterweight torque. Thus, the ideal counterweight changes as the well pumps off. Fifty and ten percent pumpoff have different peak torques as do annular fluid levels of zero and 500 feet; so, the variable net lift guarantees differences between the ideal counterweight torque(s) and the actual counterweight torque. The actual counterweight torque should be a compromise that minimizes the peak torques for the entire range of expected operating conditions. Using measured dynamometer cards to history match simulator output predictions makes it possible to confidently predict the effect of the anticipated operating conditions.

Counterbalance Equation

Computer solutions to the general form of the net torque equation give the *best* ideal counterweight torque values. The oversimplified counterbalance equivalent gives *misleading* estimates for the ideal counterweight torque(s).

Gearbox Size

The gearbox rating must be suitable for the largest peak torque caused by the range of differences between the actual counterweight torque and the compromise value based on the expected range of ideal counterweight torques. Capital cost can be lower if counterweights are carefully adjusted on a regular basis and if preventive maintenance keeps mechanical efficiency high.

Motor Size

The smallest suitable motor delivers the peak torque without stalling and without exceeding the manufacturer's ambient temperature and temperature rise limitations. The manufacturer's temperature recommendations are more important because the brake horsepower is closer to the motor rating.

Non-Ideal Counterweight Torque

Time affects the ideal counterweight torque, and the actual counterweight torque should reflect the anticipated range of ideals until the next scheduled balancing. The next check may be in a short or long time frame. Short term variations include fluid level changes, unsteady inflow performance, voltage fluctuations, and efficiency variations between lubrications. Long term variations include mechanical wear, flood front breakthrough, and reservoir decline. Seasonal changes in lubricating properties may be either short or long term considerations.

Limitations

A complete analysis necessarily includes other pumpjacks, rod, strokes, spms and pumps. While we expect the generalizations based on this limited study to be valid for it is possible that an exhaustive study will lead to some modifications.

Operating cost is inversely proportional to the ratio of brake horsepower to motor horsepower because a higher brake to motor horsepower ratio gives a higher product of power factor and electrical efficiency. The product of power factor and electrical efficiency is assumed identical for both ultra high slip and NEMA D motors. This favors ultra high slip motors with a power consumption advantage because the motor horsepower is lower.

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APPENDIX A NOMENCLATURE

dT = timestep, s
F_{cbe} = counterbalance equivalent, lbf
F_{clf} = cyclic load factor, nondimensional
F_{hr} = net hydrostatic force on rod, lbf
F_{pf} = motor power factor, nondimensional
F_{pr} = polished rod force, lbf
$F_{pr,m}$ = mth timestep polished rod force, lbf
$F_{pr,m+1}$ = polished rod force, next timestep, lbf
$(F_{pr})_{max}$ = peak polished rod force, lbf
$(F_{pr})_{min}$ = minimum polished rod force, lbf
F_{rf} = rod & coupling wt hanging in fluid, lbf
F_{su} = struct. unbal. (down=pos, SPEC 11E), lbf
F_{tf} = torque factor, in
$(F_{tf})_{max}$ = maximum torque factor, in
g = gravity acceleration = 32.1740, ft/s^2
g_c = gravity const. = 32.1740, lbm ft/(lbf s^2)
i = step counter
I_b = beam mass moment of inertia, lbm ft^2
I_c = crank & cwt mass moment of inertia, lbm ft^2
I_{dt} = drive train mass moment of inertia, lbm ft^2
j = step counter
L_A = API length, sampson bearing to pol. rod, in
m = timestep index, nondimensional
nts = number of timesteps, nondimensional
N = stroke speed, spm
N_{rpm} = motor rpm
P_{bhp} = brake (output) horsepower, hp
P_{kva} = average billable power, kW
P_{mot} = motor nameplate horsepower, hp
P_{prhp} = polished rod horsepower, hp
S = stroke, in
T = time, s and torque, in-lbf
T_c = crank net torque, in-lbf
$T_{c,m}$ = mth timestep net crank torque, in-lbf
T_{cwt} = maximum counterweight torque, in-lbf
T_{max} = peak torque, in-lbf
$T_{max,d}$ = downstroke peak torque, in-lbf
$T_{max,u}$ = upstroke peak torque, in-lbf
T_m = net motor torq, in-lbf & mth timestep, s
T_{per} = period, s
$U_{pr,m}$ = mth timestep polished rod position, ft
α_b = beam angular acceleration, rad /s^2
α_c = crank angular acceleration, rad /s^2
η_{dt} = drive train (gear & v-belt) efficiency
η_m = motor efficiency, nondimensional
η_{pj} = pumpjack efficiency, nondimensional
θ = API crank angle (meas. from vert.), rad
τ = crank counterweight phase angle, rad
ω_c = crank angular velocity, rad/s
ω_m = motor angular velocity, rad/s

APPENDIX B TORQUE EQUATIONS

The counterbalance equivalent helps approximate the peak crankshaft torque.[13,14] The general form of the API SPEC 11E equation for the net crank at the crankshaft determines helps calculate the ideal counterweight torque or the net crankshaft torque for any counterweight torque. A similar equation applies to the net torque at the motor.

The static counterweight torque equation assumes a symmetric, conventional pumpjack, neglects structural unbalance, neglects the crank phase angle, neglects mechanical efficiency, and neglects acceleration loads in order to calculate the peak and minimum polished rod forces. The peak polished rod load is the upstroke force, and the minimum rod load is the downstroke force. The average rod load is also the weight needed for the ideal counterweight torque. Thus, the average rod load (now called the counterbalance equivalent) times half the stroke length (now called the torque factor) is the ideal counterweight torque. The weight needed for the ideal counterweight torque is numerically equal to the force at the horsehead due to the counterweight torque, thanks to pumpjack symmetry. (Fig B1)

$(F_{Pr})_{max} = F_{rf} + F_{hr}$ or field measured ....(B1)

$(F_{Pr})_{min} = F_{rf}$ or field measured ....(B2)

$F_{cbe} = F_{rf} + F_{hr}/2$ ....(B3)

$(F_{tf})_{max} = S/2$ ....(B4)

$T_{cbe} = F_{tf}\ F_{cbe} = [(F_{Pr})_{max} + (F_{Pr})_{min}]\ S/4$ ....(B5)

$T_{max,u} = F_{tf,i}\ (F_{pf,i} - F_{cbe}) = F_{tf,j}\ (F_{cbe} - F_{pf,j}) = T_{max,d}$ ....(B6)

The API RP 11L equation adjusts for pumpjack efficiency.

$F_{cbe} = 1.06\ (F_{rf} + F_{hr}/2)$ ....(B7)

The API SPEC 11E equation neglects mechanical efficiency and the acceleration loads.

$T_n = F_{tf}\ (F_{pr} - F_{su}) - T_{cwt}\ \sin(\theta + \tau)$ ....(B8)

The general equation for the net torque at the crankshaft is

$T_n = F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)\ /\eta_{pj} - T_{cwt}\ \sin(\theta + \tau) + I_c\ \alpha_c$ ....(B9)

when $F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)$ is positive and

$T_n = F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)\ \ \eta_{pj} - T_{cwt}\ \sin(\theta + \tau) + I_c\ \alpha_c$ ....(B10)

if $F_{tf}\ (F_{pr} - F_{su} + I_b\ \alpha_b/L_A)$ is negative. The general net torque equation derivation equates the sum of the torques about the crankshaft to zero, then solves for the net torque at the crankshaft.

To derive the counterbalance equivalent torque equation, substitute

$F_{tf}\ F_{cbe} = T_{cwt}\ \sin(\theta + \tau)$ ....(B11)

and let F_{su}, τ, $I_c\ \alpha_c$, and $I_b\ \alpha_b\ F_{tf}\ /L_A$ approach zero to get

$T_n = F_{tf}\ (F_{pr} - F_{cbe})$. ....(B12)

The net crankshaft torque equation becomes net motor torque equation by adding drive train inertia and efficiency.

$T_m = I_{dt}\ \alpha_c + T_n/\eta_{dt}$ ....(B13)

when is T_n positive and

$T_m = I_{dt}\ \alpha_c + T_n/\eta_{dt}$ ....(B14)

when is T_n negative.

The drive train inertias convert to equivalent inertias at the angular velocity of the crankshaft by conserving angular energy,

$I_1\ \omega_1^2 = I_{dt}\ \omega_c^2$ ....(B15)

where ω_1 is the angular velocity at the motor or at the large v-belt sheave. If the motor has a 0.031081 slug ft^2 mass moment of inertia at 980 rpm, and if the crank turns at 4.9 spm, then the motor has an equivalent inertia of 40,000 lbm ft^2. (One slug equals 32.1740 lbm.)

The final equation is for motor current is

$I_{3ph} = 0.746\ T_m\ N\ /(63025\ 3^{.5}\ \text{voltage}\ \eta_m\ F_{pf})$ ....(B16)

for three phase and for single phase the equation is

$I_{1ph} = 0.746\ T_m\ N\ /(63025\ \text{voltage}\ \eta_m\ F_{pf})$. ....(B17)

APPENDIX C DATA (3,000 ft deep, 90% Pumpjack Effic. 18.5% Motor Slip)

0.08	DD	1	DTHETA
0.05	DU	0.9	EM
11	NELEM	360	BU
4.9	SPM	0	TAU
3	NCYCLES	90000	M
1	NSECTS	0	IM
30.5E6 .75 1.634 490. 3000.	E,DIA,WTA,RHOM,XLEN	00	STUFF
0.83	GT	0	DAMPP
0.83	GA	0	PUMP LOAD ADJ
1.75	DPUMP	1.0	CLF ADJ
0.8	PUMP EFF	0.06	COST
3000	DEPLIFT	10	HPRAT
30.	PTUB	0	IOUT
20.	PCSG	1	IROD
1	IFLAG	1	IFLD
0.5	FILL	14.7	PO
2.375	ODTUB	1.5E-6	COMP
1.995	IDTUB	0.5	DAMPF NOTE: KEEP 0.5 < DAMPF < 2.5
1	IUNIT	1	IVS
114	PTRAT	240	SPERATIO
143	PRLRAT	0.75	EMOT
64.	STROKE	8	NOPMC (Low Power Ultra High Slip)
84.	XA	0. 47.0 19.0	rpm ft-lbf amps
84.	XP	200. 44.0 18.0	
72.0625	XC	400. 40.0 15.	
72.	XI	700. 36.0 12.	
112.929	XK	1000. 21.0 5.7	
27.0	XR	1200. 0. 1.5	
1	JR	1250. -6.0 2.1	
71000	XIG	1300 -13.5 2.7	
56000	XIB		

APPENDIX D OUTPUT FILE (3,000 ft deep, 90% Pumpjack Effic. 18.5% Motor Slip)

PRIME MOVER		TORQUE ANALYSIS	
MAX SPEED (spm):	5.2	COUNTERBAL TORQ (in-lb):	189244.
MIN SPEED (spm):	3.6	EFFECTIVE CNTRBAL (lbf):	6547.
SPEED VARIATION (%):	30.568	PUMP ANALYSIS	
CYCLIC LOAD FACTOR:	1.4097	NET PUMP STROKE (in):	56.0
CLF AVE POWER (hp):	2.9	PROD 100% EFF (bbl/day):	90.4

TORQUE AVE POWER (hp)	2.1	PRODUCTION (bbl/day):	72.4
PUMPING UNIT		PUMP EFFICIENCY	.8
AVERAGE SPEED (spm):	4.528	TUBING STRETCH (in):	2.4
POLISHED ROD POWER (hp):	1.822	LOSS ALONG ROD STRING (hp)	.1
		PUMP POWER (hp)	1.7

KINEMATIC TABLE

ANGLE (deg)	POSITION (normalized)	TORQ FACTOR (in)	ANGLE (deg)	POSITION (normalized)	TORQ FACTOR (in)
.0000	.00043922	-1.5710400	186.7089	1.00000000	.0000000
2.0785	.00000000	.0000000	189.9996	.99935610	-1.4654500
10.0000	.00647364	6.1087090	199.9996	.98915150	-6.2029290
20.0000	.03323554	13.7732500	209.9995	.96575020	-11.2646200
30.0000	.07987607	20.8070000	219.9995	.92858310	-16.4055500
40.0000	.14385040	26.5891300	229.9994	.87787940	-21.2873900
50.0000	.22113060	30.6634600	239.9994	.81480350	-25.5692200
60.0000	.30683940	32.8672300	249.9993	.74133470	-28.9957700
70.0000	.39608630	33.3441200	259.9993	.65997700	-31.4321200
80.0001	.48467470	32.4448700	269.9992	.57345500	-32.8421800
90.0001	.56946920	30.5867500	279.9991	.48450840	-33.2432500
100.0001	.64841460	28.1432000	289.9991	.39580480	-32.6657600
110.0001	.72033350	25.3901800	299.9990	.30994180	-31.1290000
120.0001	.78464620	22.4981700	309.9990	.22949170	-28.6321100
130.0000	.84110470	19.5473700	319.9989	.15705430	-25.1570200
140.9999	.89830370	15.9404500	329.9989	.09528719	-20.6816000
150.9999	.92989890	13.4625200	339.9988	.04688671	-15.2041700
159.9998	.96173220	10.2134600	349.9987	.01448631	-8.7805570
169.9998	.98451570	6.7045710	360.0000	.00043983	-1.5720090
179.9997	.99741510	2.8359570			

SURF POS (in)	SURF FORCE (lbf)	PUMP POS (in)	PUMP FORCE (lbf)	CRANK ANG (deg)	NET TORQUE (in-lbf)
-.01	4523.86	-.02	-476.32	.5	-5626.2
.08	4562.55	.00	-426.52	6.1	-5975.4
.61	4697.33	.15	-257.31	11.8	-4955.6
1.57	5016.25	.35	-32.75	17.5	-762.1
2.95	5355.92	.73	385.54	23.1	7365.0
4.75	5880.53	1.16	870.50	28.8	21968.5
6.91	6465.00	1.71	1485.37	34.5	41060.8
9.41	7145.42	2.36	2153.93	40.2	64700.9
12.19	7391.15	4.58	2243.95	45.8	77183.8
15.17	7060.77	8.17	2248.65	51.5	67176.2
18.32	7235.91	11.04	2253.05	57.2	70559.8
21.57	7264.56	14.35	2154.32	62.8	67378.0
24.85	6994.28	18.11	2113.64	68.5	50628.6
28.14	7041.87	21.33	2130.59	74.2	43287.1
31.39	7178.46	24.30	2157.78	79.8	38942.4
34.55	7118.52	27.47	2208.64	85.5	27160.5
37.61	7134.59	30.51	2203.03	91.2	18128.9
40.54	7147.34	33.42	2215.79	96.9	9795.9
43.33	7135.40	36.25	2187.54	102.5	1681.3
45.97	7046.12	39.05	2139.90	108.2	-5776.5
48.45	6935.81	41.73	2130.29	113.9	-11855.7
50.77	6975.92	44.06	2107.52	119.5	-13299.1
52.93	7010.96	46.13	2111.50	125.2	-13552.7
54.92	6980.47	48.10	2144.24	130.9	-13761.1
56.75	6985.05	49.92	2151.51	136.5	-12160.6
58.40	7016.13	51.54	2167.19	142.2	-9201.7
59.89	7030.84	52.97	2177.32	147.9	-5716.0
61.20	7012.02	54.32	2148.17	153.5	-2028.4
62.33	6919.62	55.62	2139.16	159.2	1717.1
63.27	6906.62	56.63	2113.15	164.9	7205.4
64.02	6936.25	57.33	2098.55	170.6	12875.7
64.55	6900.10	57.89	2118.97	176.2	17589.8
64.87	6892.70	58.22	2117.29	181.9	21732.4
64.96	6917.01	58.27	2111.81	187.6	24813.0
64.79	6871.59	58.22	2057.96	193.2	26688.8
64.11	6657.07	58.07	1885.46	201.3	28473.5
63.29	6420.99	57.86	1653.14	206.9	30036.1

SURF POS (in)	SURF FORCE (lbf)	PUMP POS (in)	PUMP FORCE (lbf)	CRANK ANG (deg)	NET TORQUE (in-lbf)
62.19	6093.32	57.59	1361.20	212.6	33062.1
60.80	5712.71	57.23	962.22	218.3	37965.7
59.12	5223.35	56.82	507.70	223.9	46558.0
57.16	4696.80	56.31	-50.26	229.6	58262.3
54.94	4085.82	55.62	-476.32	235.3	74150.3
52.48	4182.43	53.01	-476.32	240.9	73581.8
50.05	4453.82	50.09	-476.32	246.1	65876.1
47.45	4207.08	47.91	-476.32	251.3	73085.2
44.72	4177.36	45.21	-476.32	256.5	74064.8
41.89	4314.32	42.15	-476.32	261.7	68626.8
38.97	4231.39	39.35	-476.32	266.9	69639.3
36.00	4217.89	36.42	-476.32	272.1	68467.6
33.00	4245.15	33.36	-476.32	277.3	64909.7
29.99	4246.88	30.36	-476.32	282.5	61803.3
26.99	4252.04	27.37	-476.32	287.7	58033.1
24.03	4255.89	24.39	-476.32	292.9	53657.5
21.14	4263.18	21.47	-476.32	298.1	48745.1
18.33	4275.36	18.66	-476.32	303.3	43320.0
15.64	4290.22	15.96	-476.32	308.5	37490.8
13.07	4306.42	13.37	-476.32	313.7	31505.9
10.46	4327.48	10.72	-476.32	319.4	24858.6
8.25	4341.21	8.48	-476.32	324.6	19021.6
6.25	4368.67	6.46	-476.32	329.8	13180.1
4.49	4394.94	4.67	-476.32	335.0	7874.5
2.98	4421.52	3.12	-476.32	340.2	3212.2
1.76	4451.74	1.84	-476.32	345.4	-592.5
.84	4489.29	.89	-476.32	350.6	-3112.9
.28	4502.64	.31	-476.32	355.3	-4604.7
.01	4522.19	.00	-476.32	360.0	-5567.6

APPENDIX E TYPICAL POWER FACTOR and ELECTRICAL EFFICIENCY

These equations curve fit vendor data for 3 to 7.5 hp motors when the load ratio spans 0.2 and 1.25.

Load ratio = (brake horsepower) /(motor horsepower) ....(E1)

F_{pf} = (1.411936) $(.5800495)^{(\text{load ratio})}$ $(\text{load ratio})^{(.7327973)}$ ....(E2)

η_{mot} = .9263102 -.05882107 /load ratio -.06875724 $(\text{load ratio})^2$ +.0003167309 $/(\text{load ratio})^3$ ....(E3)

Table 1
Study Variables

Case i.d.	Motor Slip	Net Lift	Pump Off	Pumpjack Efficiency
a	18.5%	3,000 ft	na	90%
b	7.5%	3,000 ft	na	90%
c	7.5%	3,000 ft	na	80%
d	18.5%	2,000 ft	na	90%
e	18.5%	na	10%	90%
f	18.5%	na	50%	90%

Table 2
Minimum Motor Size

Polished Rod Horsepower	1.98 hp
Pumpjack Mech. Efficiency	.90
Drive Train Mech. Efficiency	.95
Avg. Brake Horsepower	2.32 hp
Cyclic Load Factor	1.42
Minimum Motor Size	3.29 hp
Full Load Speed	980 rpm
Peak Torque @ 100% ideal cwt	77,500 in-lbf
Peak Torque @ 125% ideal cwt	124,500 in-lbf

Table 3
Power Consumption vs. Motor Size

Average Motor hp	Minimum Motor hp*	Motor Size hp	Average Load Ratio	Average Power Factor	Average Electrical Efficiency	Average KVA	Save KVA	Save $/bbl **
2.32	3.29*	3	.77	.77	.82	2.74	-	-
2.32	3.29	4	.58	.68	.80	3.18	.44	.0070
2.32	3.29	5	.46	.61	.77	3.68	.94	.0150
2.32	3.29	7.5	.31	.50	.73	4.74	2.0	.0320
2.32	3.29	10	.23	.43	.70	5.75	3.0	.0482

* (polished rod hp) (cyclic load factor) /(mechanical efficiency)

** ($0.06 /kW hr) (24 hr/day) /(90 bpd)

Table 4
Ideal Peak vs. Stall (Breakdown) Torque with 100% Ideal Counterweight Torque

Peak Torque in-lbf	Motor Size hp	Full Load 980 rpm	Torque 4.1 spm	Peak/ Full Load	NEMA D Stall -- Torque --- %FL	in-lbf	Peak/ Stall Ratio
77,500	3	193	46,300	1.67+	275%	127,000	.61+
77,500	4	257	61,800	1.25+	275%	170,000	.46+
77,500	5	322	77,200	1.00	275%	212,000	.37
77,500	7.5	482	116,000	0.67	275%	319,000	.24

\+ potential motor selection

Table 5
Peak vs. Stall (Breakdown) Torque with 125% Ideal Counterweight Torque

Peak Torque in-lbf	Motor Size hp	Full Load 980 rpm	Torque 4.1 spm	Peak/ Full Ratio	NEMA D Stall -- Torque -- %FL	in-lbf	Peak/ Stall Ratio
124,500	3	193	46,300	2.69	275%	127,000	.98
124,500	4	257	61,800	2.01	275%	170,000	.73
124,500	5	322	77,200	1.61+	275%	212,000	.59+
124,500	7.5	482	116,000	1.07+	275%	319,000	.39+

\+ potential motor selection

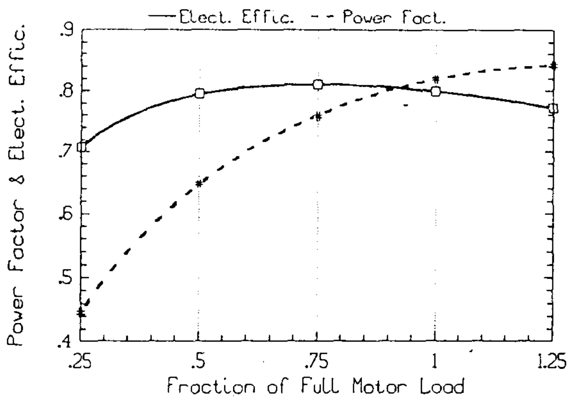

Figure E1 - Nominal motor properties
NEMA D, 3-7.5 horsepower

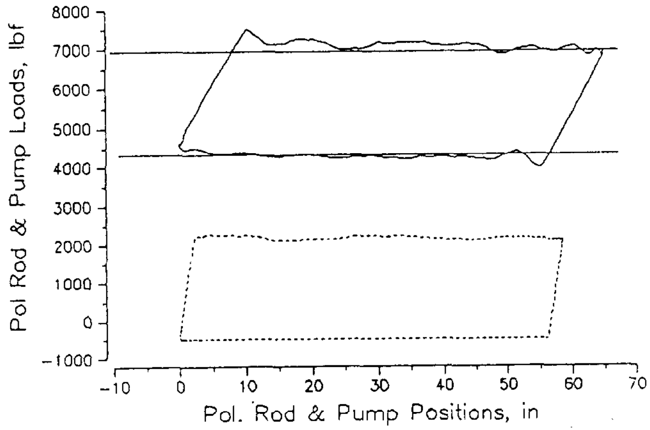

Figure 1 - Surface and pump dynamometer cards

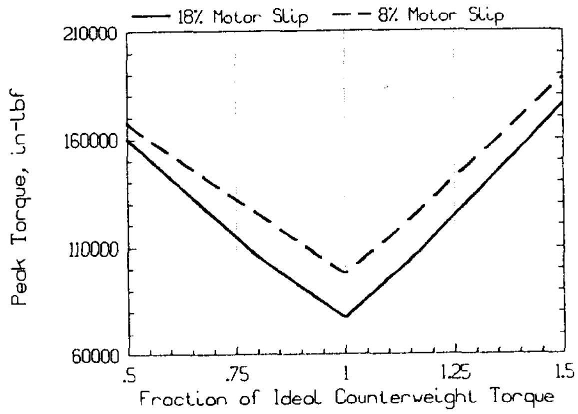

Figure 2 - Peak torque vs. motor slip

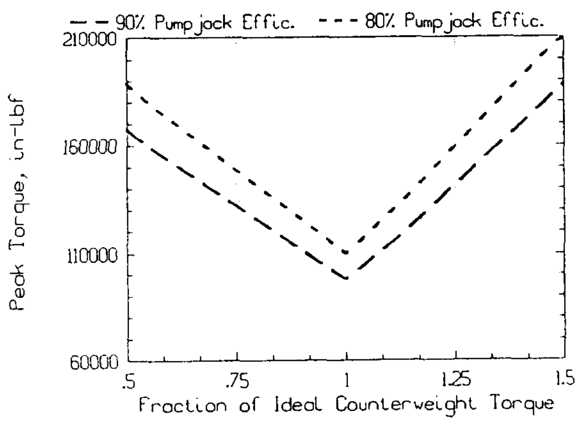

Figure 3 - Peak torque vs. mechanical pumpjack efficiency

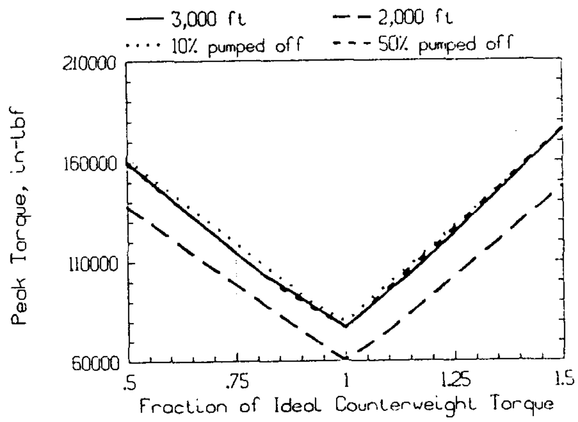

Figure 4 - Peak torque vs. net lift

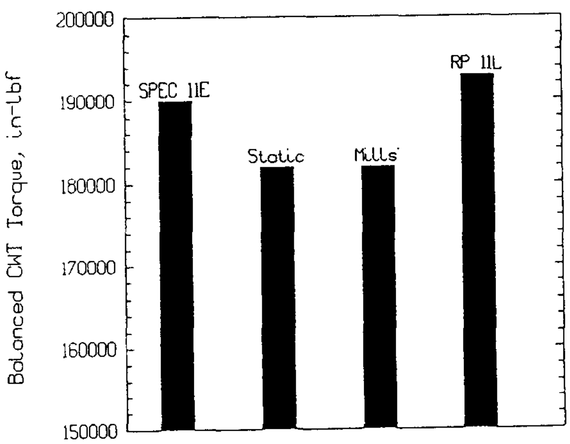

Figure 5 - Ideal counterweight vs. calculation method

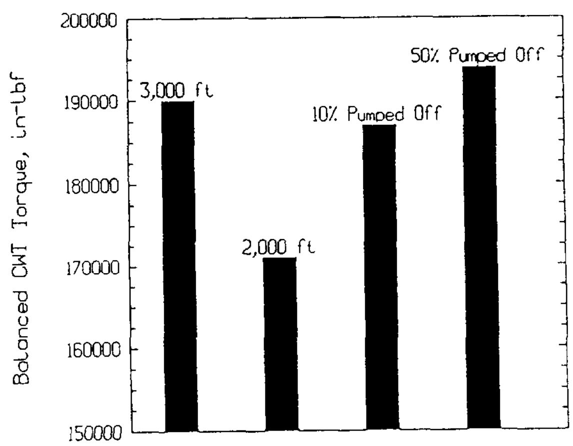

Figure 6 - Ideal counterweight vs. net lift

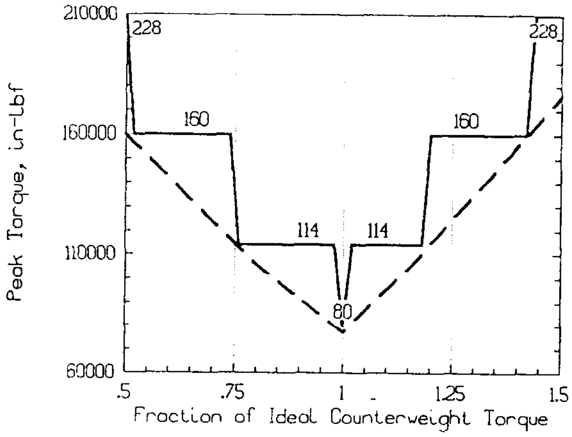

Figure 7 - Gearbox size vs. counterweight torque

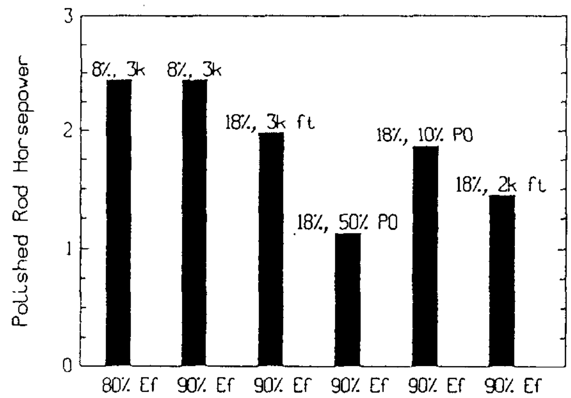

Figure 8 - Polished rod horsepower vs. pumpjack efficiency, motor slip, net lift

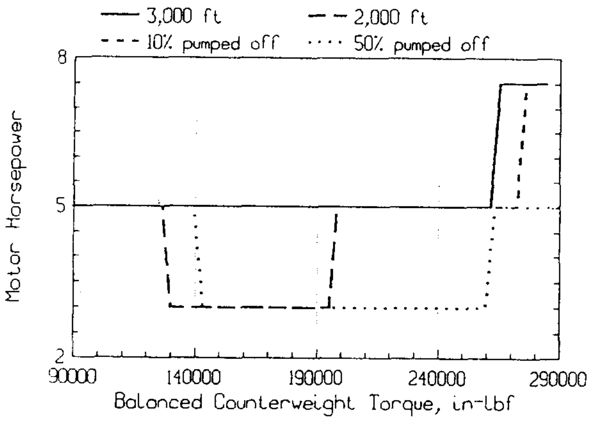

Figure 9 - Motor horsepower by net lift

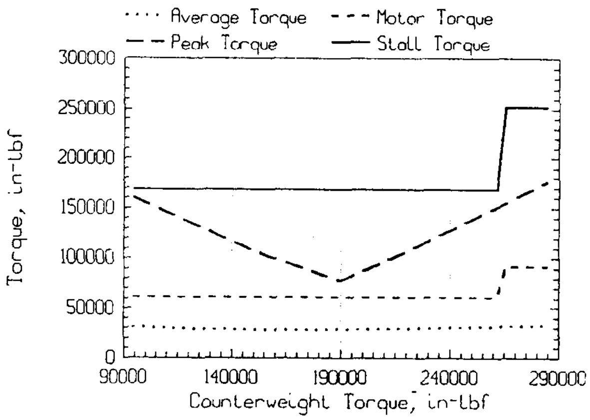

Figure 10 - Torque vs. ideal counterweight

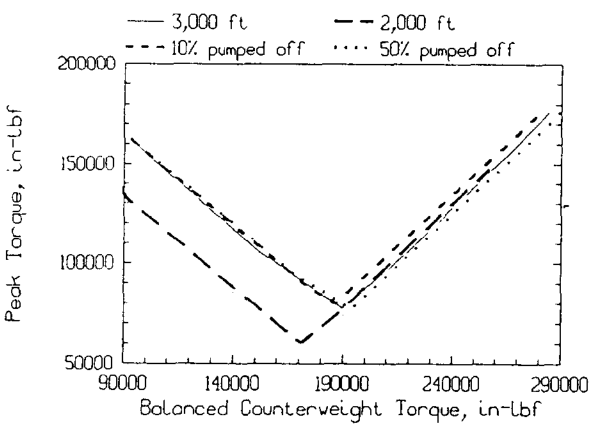

Figure 11 - Peak torque by net lift

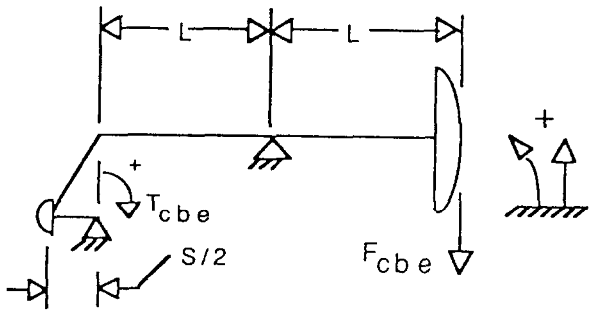

Figure B1 - Counterweight equivalent