PUMPING UNIT EFFECT ON MOTOR EFFICIENCY

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ABSTRACT

Electric motors are operated with cyclical loading on beam pumping units. However, motors are rated for steady loads. The performance of the motor changes when applied to a varying torque load. The motor efficiency, energy consumption and available torque are reduced. A method of calculating the effective ratings is presented. A comparison of operations on both conventional and unconventional pumping units is outlined. Economics of optimum motor sizing are discussed.

INTRODUCTION

Beam pumping comprises about 90% of the artificial lift systems used in the petroleum industry. In the past 20 years, electric motors have become the predominant drive mechanism for these units. There have been tens of thousands of motors applied to pumping units. Unfortunately, there have been almost as many methods of sizing the motors. Many of these procedures are radically different. Moreover, the results are equally diverse.

This paper will address effectively selecting motors for long term economic operation. The discussions will include the load of the fluid to be moved, losses in the mechanical equipment and fitting a motor performance curve to the beam performance. Theoretical and field data will be provided.

UNIT HORSEPOWER

The horsepower required for moving the fluid is a well defined problem. This is represented by the hydraulic horsepower (HHp) [1].

$$HHp = (Q\ BPD)(H\ ft)(sp\ gr)(350.28\ lb/bbl)(Hp\text{-}day/47{,}520{,}000\ ft\text{-}lb) \qquad (1)$$

$$HHp = (Q)(H)(sp\ gr)/135663 \qquad (2)$$

The flow rate "Q" is the total fluid that is moved. The head "H" is the total energy including friction loss in the pipe.

The hydraulic horsepower represents the energy required to move the fluid in a specified period of time. There are other losses in the system that must be overcome by the motor. The major groups of these are the pump, rod and surface losses as shown in Figure 1.

The power placed on the rods at the surface is polished rod horsepower (PRHp). It comprises all the downhole power including losses. The efficiency (n) of the pump and rods can be applied to the hydraulic horsepower to obtain polished rod horsepower.

$$PRHp = (HHp)/(n_{pump})(n_{rod}) \tag{3}$$

The mechanical or brake horsepower that the motor must deliver is proportional to polished rod horsepower and inversely proportional to surface efficiency. The surface efficiency is reduced by stuffing box friction, inefficiencies in the gears and belt slippage.

$$BHp = PRHp/n_{surface} \tag{4}$$

Electrical power purchased is not brake horsepower. There are electrical inefficiencies that must be considered. Through inappropriate choices of motors, these are often greater than the mechanical inefficiencies.

MOTOR SIZING

It has long been recognized that a motor rated only to meet the mechanical horsepower requirements would not perform adequately. Often the motor would not start because of inadequate torque or would overheat and burn out.

The mechanical horsepower is an average value based on moving a quantity of fluid per day. The motor horsepower rating (MHp) assumes a steady load and must be adjusted when the motor supplies power to a cyclical load. The cyclic load factor (clf) that has been used to compensate the motor rating for oil pumping service is a simple relationship but it is not easily applied [2].

$$clf = \frac{\text{root-mean-squared of pumping cycle current}}{\text{average of the pumping cycle current}} \tag{5}$$

$$MHp = (BHp)(clf) \tag{6}$$

Typical values of cyclic load factor range from 1.1 for low pumping speeds to 1.55 for high pumping speeds with normal slip motors. The cyclic load factor depends on the speed of the unit, the type of unit and the motor slip. Because of these variations it is not always applied consistently.

In an effort to reduce the task to a solvable problem, the pumping unit manufacturers have developed rule-of-thumb practices. As with most approximations, the rules are very usable and fit many applications. However, in the current environment of controlling both capital and operating expense costs, it is no longer appropriate to be close.

The most prevalent of the motor sizing approximations is given for conventional pumping units. The approximation is multiplied by .8 for unconventional geometry [3].

$$MHp_1 = (Q * H)/\ 56000 \tag{7}$$

$$MHp_2 = (Q * H)/\ 45000 \tag{8}$$

The first relationship is for high slip motors and slow speed engines. The second relationship is for normal slip motors and multi-cylinder engines. It is apparent these relationships simply apply a factor to the mechanical horsepower equation. The assumptions that provide the factor are a fifty percent system efficiency and a cyclic load factor of 1.5 for normal slip and 1.2 for high slip motors.

A major producer has reported the denominator factor is too conservative. Rather than 56,000, a value of 75,000 is used. In effect, the producer's motor is 33% smaller than the most prevalent approximate motor sizing.

Another major producer calculates the brake horsepower from the polished rod horsepower. A motor size is determined from the brake horsepower then the next larger size motor is selected. For example, a calculated BHp of 8 Hp would indicate a motor size of 10 Hp and a selected motor size of 15 Hp.

From these practices for sizing a motor, it is obvious a consensus does not exist. Moreover, reasonable determination of the appropriate size depends on broad generalizations rather than specific applications.

TORQUE CURVES

Motor performance must be analyzed with the motor subjected to various pumping loads. Actual pumping unit torque curves will be used as input to a computer program that determines the performance of the motor applied to the unit. The torque required by the unit varies somewhat with the particular motor applied. The variation is due to the changes in speed of the unit over the pumping cycle. Peak torques on the unit increases when the motor shaft speed variation decreases.

This paper addresses a technique to determine the performance on a particular motor when the unit torque load is known. It will not address the changes in torque on the unit because of the motor. That is the subject of a subsequent paper. This procedure will find the optimal operating point and size motor for a particular torque characteristic.

Therefore, the motor performance can be evaluated using fixed unit torque curves such as those in API RP 11E. API torque curves were used for both the conventional and unconventional geometry units [4]. A dynamic curve for a conventional unit was used based on Nabla data [5].

The instantaneous mechanical torque on the motor and the unit shaft varies as a distorted sinusoid. The well load produced by the polished rod is offset by a counter balance torque. The difference in these torques is the net torque on the shaft of the gearbox. The electrical power into the system changes with this cyclic relationship rather than the average mechanical horsepower.

A conventional pumping unit torque curve is given in API 11E and is shown in Figure 2. The API curve represents static torque calculated from the pumping unit geometry. The curve is appropriate for analyzing starting conditions and unbalanced conditions.

The dynamic curve is shown in Figure 3. The curve is appropriate for analyzing running conditions on a conventional unit, since it takes into account momentum.

The static performance provides a more conservative design which will yield larger equipment. The significant difference observed with the dynamic curve is: (1) the magnitude of the peak torque with respect to the average torque and (2) the quantity of negative torque. The ratio of the peak to average torque is a measure of the increased motor horsepower requirement for cyclic loads. Average torque is proportional to the polished rod horsepower.

The reduced negative torque directly reduces the amount of electricity regenerated into the system. Reduction of the generation with its associated positive losses, increases the electrical efficiency for the pumping cycle.

The static performance curve for an unconventional geometry unit is shown in API 11E and is given in Figure 4. This unit has significantly better peak to average ratios, less negative torque, and correspondingly better overall electrical efficiency than the conventional geometry units.

The horsepower required by the pumping unit can be calculated from the torque curves. The speed of the shaft as well as the torque must be used to determine the shaft horsepower. The power equation can be written in terms of units associated with beam pumps.

$$P = T\,w \qquad (9)$$

$$Hp = (\text{in-lbs} * \text{RPM}) \,/\, 63025 \qquad (10)$$

Averaging the speed and torque at discrete points on the unit performance curve provides average horsepower over a complete cycle.

MOTOR OPERATION

As with all engineering solutions, the motor size rating is not an exact value, but is a trade-off between cost, size and service to obtain a competitive device [6,7]. The manufacturer's steady load performance curve for a very common 10 Hp, NEMA D torque characteristic motor is shown in Figure 5 [8]. The curve is the motor manufacturer's data curve with extrapolations into the negative torque regions and from twice full load up to the locked rotor.

There is no particular point on the curve which dictates a rating of 10 horsepower. The horsepower rating for the motor is a value that will provide an average power when running at a constant load. It is not a peak rating nor a rating that will provide adequate performance on cyclical loads with large peaks.

One significant performance characteristic should be noted. The motor efficiency is much better when the unit is operating underloaded than when it is operating at greater than its rated horsepower. For this class motor the peak efficiency occurs near fifty percent load.

There are three main components that impact the efficiency of a motor: core losses, copper losses and friction-windage losses. The core losses are dependent upon the iron and electromagnetic fields. This loss is primarily influenced by the voltage. Since voltage is constant with load, this loss remains constant. The friction-windage represents the mechanical losses and is primarily influenced by speed. Since speed changes only a limited percentage with load, this loss is also approximately constant. Copper losses are dependent upon the wire size and I^2R heating. Since current changes approximately proportional with load, this loss changes with the square of the load [9]. The constant losses dominate the efficiency at low load, while the copper losses dominate at loads greater than fifty percent.

$$\text{Motor loss} = I^2R + \text{core loss} + \text{friction-windage loss} \tag{11}$$

The motor efficiency curve given by the manufacturer is for a constant load over a normal operating range of 25 percent to 175 percent of rating. Because of the cyclical nature of a pumping unit, the motor will operate over a much wider range. The minimum energy consumption of a unit will come when the motor is generating (pumping unit operating at negative torque). The maximum energy consumption is near locked rotor or peak torque.

On the low end of the curve the motor may become a generator. Although the motor has negative power consumption, it still has positive losses. These losses are proportional to those experienced at an equivalent positive load [10]. In order to accommodate non-linearities near zero load, it was assumed the losses are equal to no load losses if the motor efficiency is less than 0.5. This is an adequate approximation since the unit consumes relatively little energy while operating at low loads.

EFFICIENCY ANALYSIS

To determine the effect of pumping unit loads on the electric motor, the pumping unit torque curve characteristics were applied point by point to the motor performance curves. The superposition of the two is shown in Figure 6. It is important to note that this is one point of loading for one size of motor for the particular load characteristic.

The pumping unit torque was determined at a discrete crank angle. The torque was used on the motor curve to determine the speed and efficiency. The speed and torque were used to determine horsepower and the horsepower and efficiency were used to determine losses.

The efficiency at discrete points on the scaled horsepower curve is taken from the motor steady load curve. The instantaneous losses are then calculated.

$$\text{losses} = (\text{BHP} \;/\; n_{motor}) - \text{BHP} \qquad (12)$$

The horsepower from all the discrete points can be added to determine an average horsepower for the specified load conditions. Similarly, the losses at all the digitized points can be added to obtain the average losses. The overall motor efficiency under cyclical load can then be calculated.

$$n_{cyclic} = \frac{\text{actual delivered horsepower}}{\text{actual delivered horsepower + actual loss}} \qquad (13)$$

This representation has been the effective load on the motor for a conventional pumping unit. The same effective load on the motor for an unconventional pumping unit is depicted on Figure 7. At the same 5.8 horsepower loading, the losses are very different.

It is apparent that a different shape torque characteristic will affect the motor loading and require a specific analysis. Similarly a different magnitude torque characteristic or operating point will affect the motor loading and require a new analysis.

Each of the specific curves results in a single point of cyclic load (polished rod horsepower) and efficiency. By scaling the torque curve, a suite of these points can be used to develop a curve for the performance of the motor on a particular shape torque load. When other torque shapes are used a family of curves for a particular motor can be developed that represents its performance over a range of cyclic operating conditions.

A family of curves for a 10 horsepower, NEMA D motor are shown in Figure 8. This is a plot of motor shaft horsepower versus efficiency. The motor shaft horsepower is the polished rod horsepower divided by pumping unit efficiency. Since the unit efficiency is very high, the curves follows the polished rod horsepower.

The efficiency for the motor is plotted against the pumping unit horsepower load. The abscissa of the curves is the efficiency at each of the average horsepower points. Curve 1 represents efficiency of the motor operating on an API conventional unit. The dynamic conventional and API unconventional operation are represented by curves 2 and 3. Curve 4 represents the performance of the motor if it were applied to a steady load.

From these curves several observations can be made. On a conventional unit, the maximum cyclical horsepower which can be started is 50% of the motor rating, as shown at the maximum load point of curve 1. However, the motor can drive an average cyclical load at 90% of its rating as shown by the maximum point of curve 2.

The phenomenon is familiar to those who have had to "rock" a pumping unit to start it.

The best efficiency point occurs near 40% of the balanced cyclical load. This is at the peak of curve 2. The best efficiency point of an unbalanced unit is achieved by restricting the cyclical load to 25-30% of the motor rated load. To obtain the best efficiency a motor should have a rating about 2.5 times the polished rod horsepower requirement on a conventional unit.

$$MHp = PRHp \ / \ .4 \tag{14}$$

The curves indicate the unconventional geometry unit has significantly better performance than a conventional unit. The static and dynamic performance are very close. This is the result of the better average to peak torque relationship. Therefore, a motor can start a load with an average cyclical load equal to the motor rating. The maximum efficiency of the motor occurs when its cyclical load is 50% of its rated size.

Using the same motor, the efficiencies are consistently greater operating on an unconventional geometry. Five percent less energy will be consumed by the same load on an unconventional unit compared to a conventional beam pumping unit when properly sized. If the motor is heavily loaded, the unconventional geometry has as much as 30% improved efficiency over the conventional unit.

OPERATING COSTS

If a motor is sized so that the cyclical load is at 40% of the motor rating, the amount of electricity consumed is significantly reduced. If the motor has a 75% load, a typical efficiency is 65% while a 40% load has a typical efficiency of 75%. The 9 point improvement in efficiency will more than offset the investment in the larger equipment.

As an example, an average cyclical load of 10 Hp running continuously for one year consumes 65350 kwhr of energy at 100% efficiency, 87133 kwhr at 78%, and 99015 kwhr at 68% efficiency. The difference in efficiency is equivalent to 11882 kwhr per year. If energy cost is $.06/kwhr, the savings is $712 per year. This provides less than a three year payout simply in energy savings.

An even greater improvement in efficiency is realized when a larger frame motor is implemented. Typically the losses do not increase proportional to the motor rating. Figure 9 contains the motor performance data for a 25 Hp motor. A motor rated at 25 Hp has a maximum efficiency of 90%, while one rated at 10 Hp has a maximum efficiency of 85%. A load of 6.5 horsepower represents 66% efficiency using a 10 horsepower motor and 81% using a 25 horsepower motor. This is an improvement of over 15 points or a 23% improvement in energy usage.

The information indicates motors should be sized to allow for peak gear box torque and a cyclical load. The motor rating will be significantly larger than average polished rod horsepower. The pumping unit performance and the actual motor performance should be compared to arrive at the optimum selection.

Although the calculations are tedious by hand, with computer programs it is very viable to consider actual unit geometry and actual motor curves. Computer hardware is also developing that will permit direct monitoring of the true electrical horsepower and unit load. This will further aid maintaining minimum operating expenses.

In applying motors to projected loads, it is not always feasible to develop sophisticated models. Because of the shape of the motor performance curve and the pumping unit torque curve, in general, the motor should be sized to be 40% of the average mechanical horsepower calculated from producing rates, head, and downhole efficiency.

MITIGATING PARAMETERS

In the preceding analysis only the peak and average horsepower effect on motor efficiency have been compared. If the motor is sized to have only a 40% average load, other performance criteria must be considered.

One detrimental effect of oversizing motors is the lowering of the power factor from .87 to .74. However, this is easily corrected with capacitors.

One of the most positive effects is available starting torque becomes approximately twice as great. This assures the unit will start even if it is unbalanced.

Another significant improvement is reliability. Larger frame size motors have larger bearings capable of longer life with less loading. Furthermore the larger units have less I^2R heating. Since insulation life is reduced by approximately one-half for each 10°C rise in temperature, the mean time between failure should improve.

Because of the slope and shape of the speed curve, the speed change of the unit and resulting motor slip is greater at full load than one-half load. At 20 Hp the speed is 1120 RPM for a slip of 6.6%, while at 10 Hp the speed is 1160 RPM for a slip of only 3.33%. This increased slip is one reason for decreased motor efficiency as loading increases.

OPERATING VERIFICATION

To provide statistical verification, a field study was correlated with the analysis. The study conducted by Lovett and Richmond involved 181 wells in Kansas and Oklahoma [11]. The study illustrated energy consumption compared to motor loading.

A graphical representation of the energy cost per barrel-foot is presented in Figure 10. The empirical data supports results of the computer generated curves in Figures 8 and 9. The data shows a 22% increase in cost at low load and a 25% increase at high load. Since the majority of the data was in the center of the curve, the end points could have some error. The graph is the result of a fifth order curve fit to the measured data.

SUMMARY

Electric energy consumption can be dramatically improved by properly sizing the motor. The cyclical effect on average polished rod horsepower must be considered. By using motor efficiency curves and pumping unit torque characteristics, the optimum motor size can be calculated.

1. The best efficiency will be achieved with a motor operating at 40-50% of its rating.

2. A 10 point improvement in efficiency is obtained when the motor load changes by a factor of 2 and the final motor load is near 50%.

3. A motor provides adequate starting torque for a conventional unit only when the motor rating is 2 times the average load.

4. Unconventional geometry units are at least 5% more efficient than a conventional unit at the same load.

5. When a motor is heavily loaded, an unconventional unit is as much as 30% more efficient than an equivalent conventional unit.

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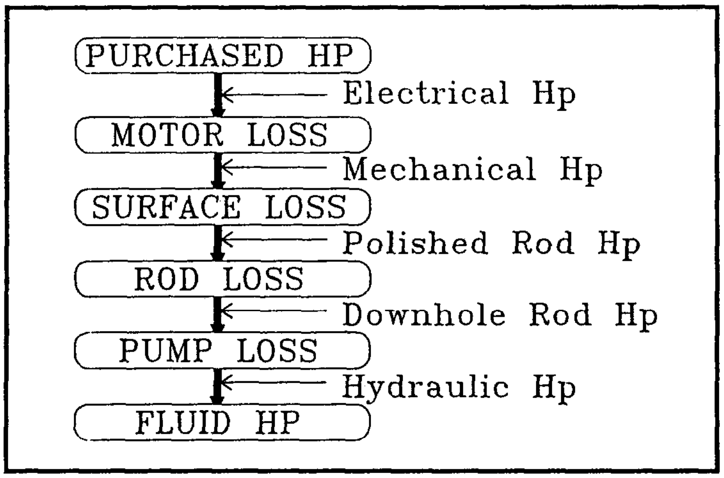

Figure 1 — System power diagram

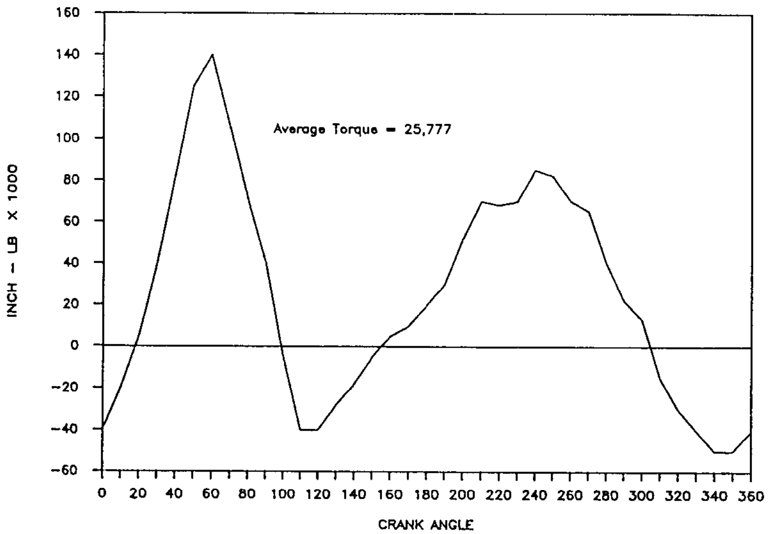

Figure 2 — Conventional pump torque —
API RP11E

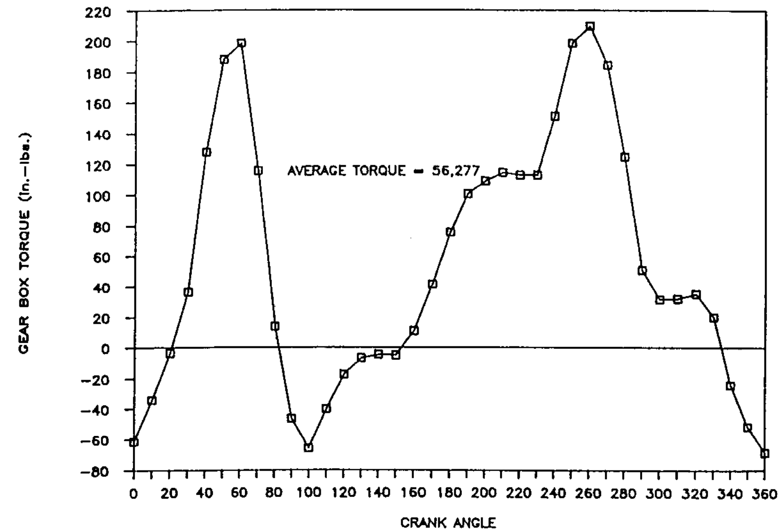

Figure 3 — Nabla — conventional pump torque

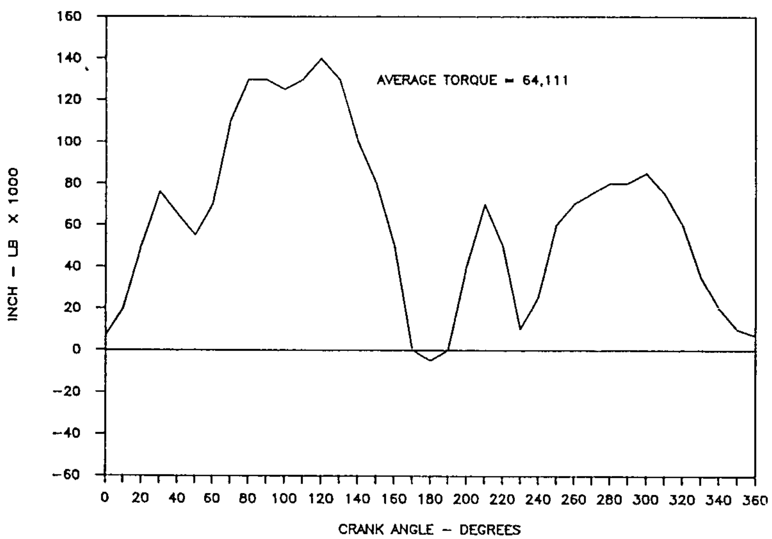

Figure 4 — Unconventional pump torque —
API RP11E

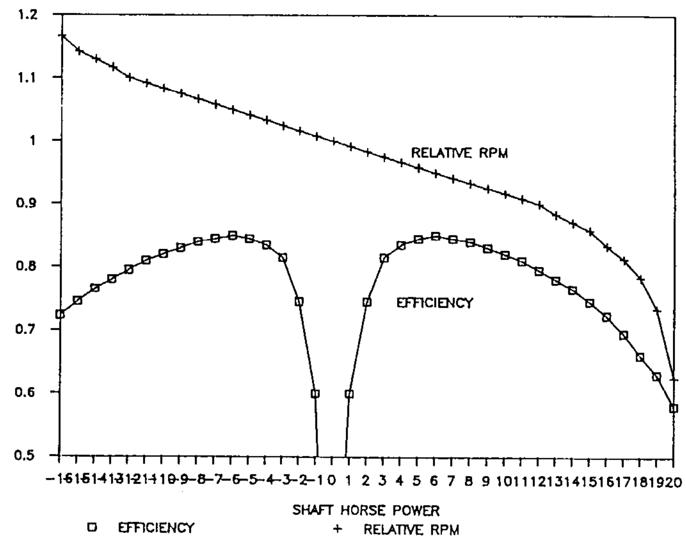

Figure 5 — 10 Hp Nema D motor —
efficiency and RPM

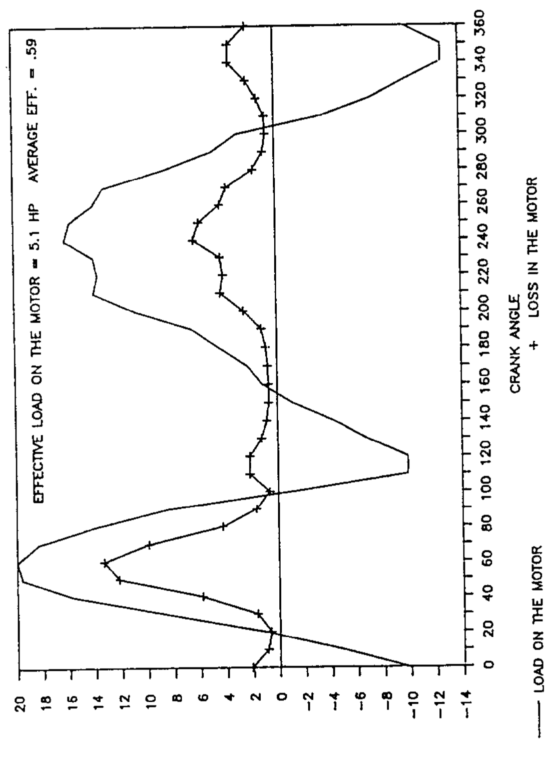

Figure 6 — API conventional with 10Hp — load and loss

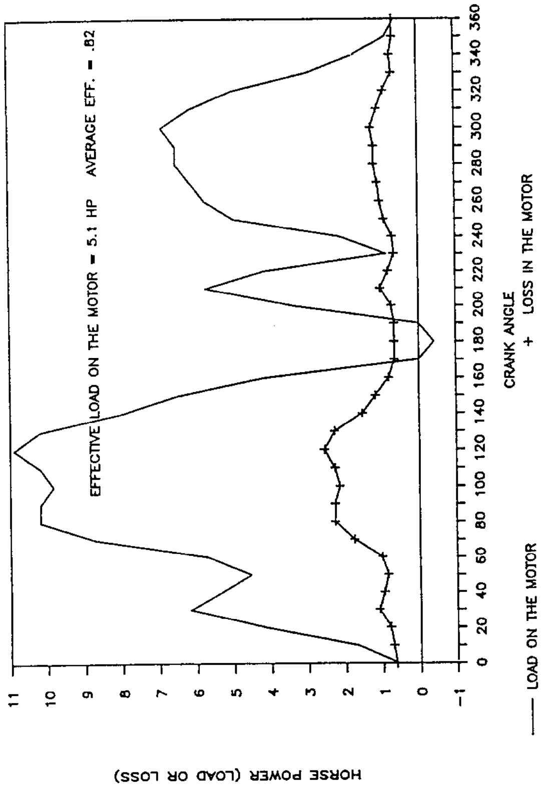

Figure 7 — Unconventional with 10 Hp load and loss

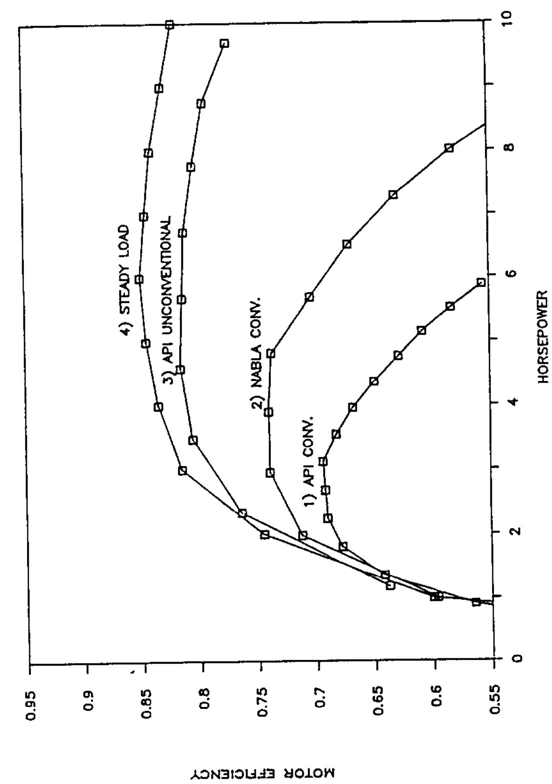

Figure 8 — 10 Hp Nema D motor efficiency

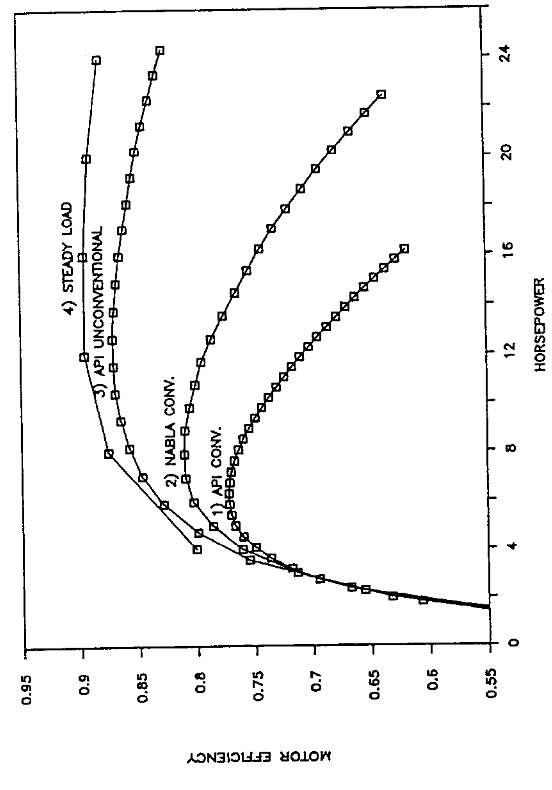

Figure 9 — 25 Hp Nema D motor efficiency

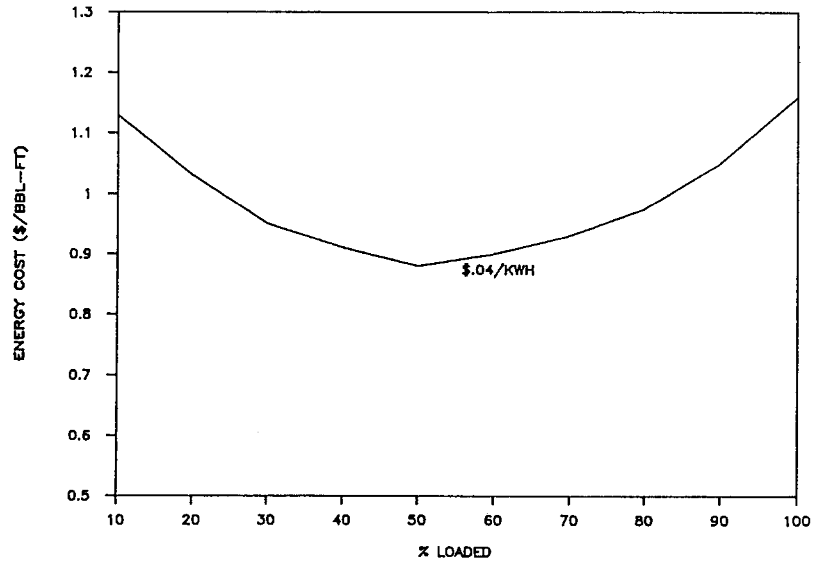

Figure 10 — Cost to pump wells vs. percent loaded