Effect of Cyclic Loading on Motor Efficiency

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Abstract—**Electric motors are operated with cyclical loading on beam pumping units. This varying load impacts motor efficiency, energy consumption, and available torque. A method of sizing motors is proposed which will provide the least energy consumption and more starting torque. The technique compares steady-load motor efficiency with the efficiency when operated on various cyclic load configurations. The performance of unconventional and conventional pumping units is compared for calculated and measured torques. Field data is used to verify the model. For cyclic loads, motors are more efficient when operated near half-load. The reason for improved efficiency when using unconventional units is shown.**

Introduction

BEAM pumping comprises about 90 percent 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 the effective selecting of 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.

Mechanical System

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

$$\text{hhp} = \frac{Q\ \text{bbl} \mid H\ \text{ft} \mid \text{spgr} \mid 8.34\ \text{lb} \mid 42\ \text{gal} \mid\ \text{hp}\cdot\text{min} \mid \text{day}}{\text{day} \mid \qquad \mid \qquad \mid \text{gal} \mid \text{bbl} \mid 33\,000\ \text{ft}\cdot\text{lb} \mid 1440\ \text{m}}$$

$$\text{hhp} = QH\,\frac{\text{spgr}}{135\,663}.$$

where spgr denotes specific gravity. 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 Fig. 1.

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Power Company
          |
     _____|-------------- Electrical Hp
|     Motor     |
          |
     _____|-------------- Mechanical Hp
| Surface loss  |
          |
     _____|-------------- Polished Rod Hp
|   Rod loss    |
          |
     _____|-------------- Rod Hp Downhole
|   Pump loss   |
          |
     _____|-------------- Hydraulic Hp
|   Fluid Hp    |
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Fig. 1. System power.

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:

$$\text{prhp} = \frac{\text{hhp}}{(n_{\text{pump}})(n_{\text{rod}})}.$$

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:

$$\text{bhp} = \frac{\text{prhp}}{n_{\text{surface}}}.$$

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.

Traditional Procedures

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

Paper PID 88-16, approved by the Petroleum and Chemical Industry Committee of the IEEE Industry Applications Society for presentation at the 1987 Petroleum and Chemical Industry Committee Technical Conference, Calgary, AB, Canada, September 14–16. Manuscript released for presentation April 5, 1988.

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IEEE Log Number 8822933.

0093-9994/88/1100-1153$01.00

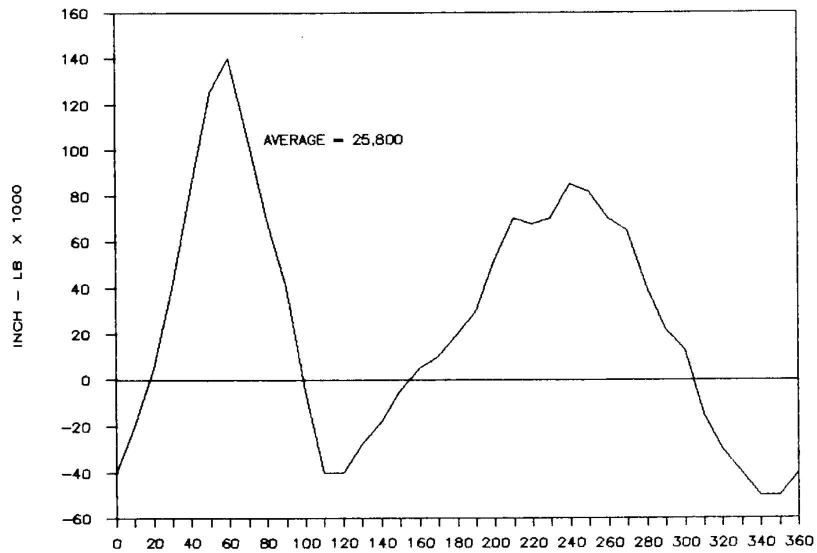

Fig. 2. Static torque curve, conventional unit.

easily applied [12]:

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

$$\mathrm{mhp} = \mathrm{bhp} \cdot \mathrm{clf}.$$

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 not longer appropriate to be close.

The most prevalent of the motor sizing approximations is given for conventional pumping units. The approximation is multiplied by 0.8 for unconventional geometry [3]:

$$\mathrm{mhp}_1 = \frac{Q \cdot H}{56\,000}$$

$$\mathrm{mhp}_2 = \frac{Q \cdot H}{45\,000}.$$

The first relationship is for high slip motors and slow speed engines. The second relationship is for normal slip motors and multicylinder engines. It is apparent that these relationships simply apply a factor to the mechanical horsepower equation. The assumptions that provide the factor are a 50-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 percent 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.

Pumping Unit Torque

Motor performance must be analyzed with the motor subjected to various pumping loads. Measured and API torque curves were used for both the conventional and unconventional geometry units [4]. 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 counterbalance 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 American Petroleum Institute (API) Specification 11E and is shown in Fig. 2. The API curve represents static torque calculated from the pumping unit geometry. The curve is appropriate for analyzing starting conditions and unbalanced conditions. A dynamic curve has been measured as shown in Fig. 3. The curve is appropriate for analyzing running conditions on a conventional unit, since it takes momentum into account.

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

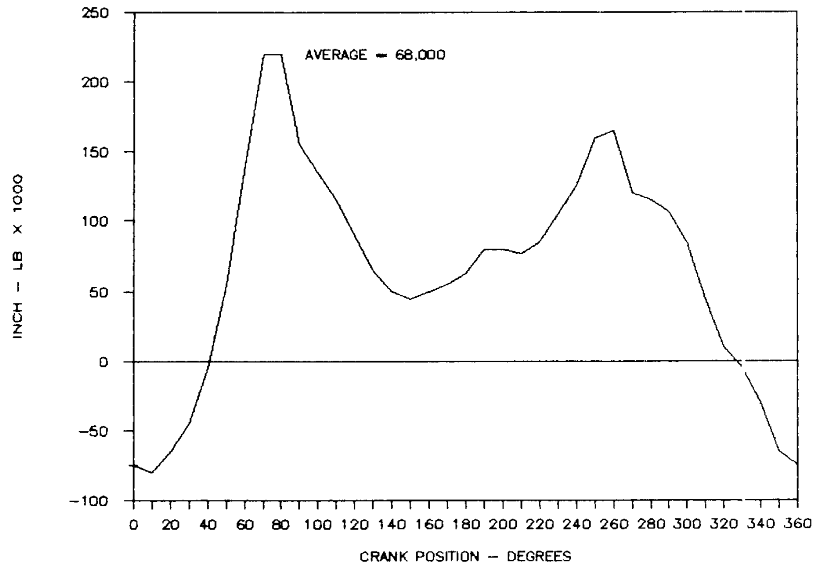

Fig. 3. Dynamic torque curve, conventional unit.

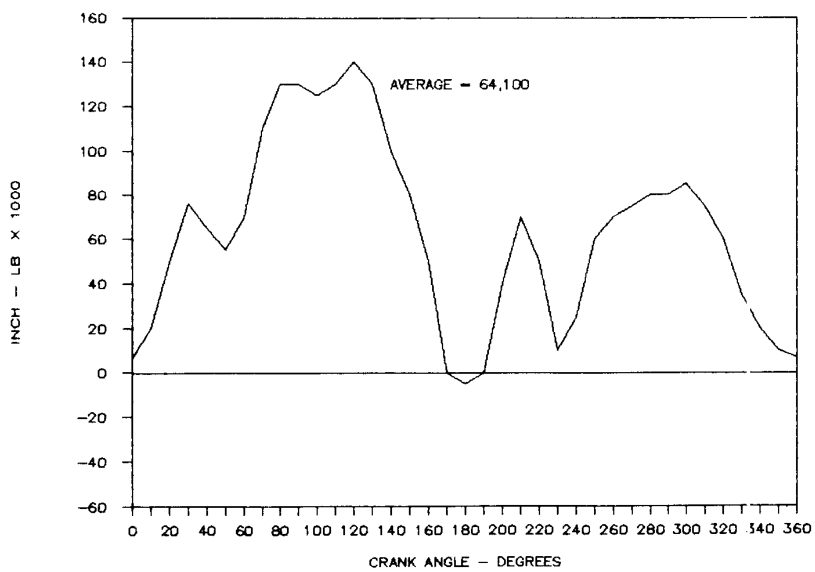

Fig. 4. Static torque curve, unconventional unit.

2) the quantity of negative torque. The ratio of 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 Fig. 4. A dynamic torque curve for a different type unconventional geometry is shown in Fig. 5. Both of these units have 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:

$$P = Tw.$$

If the torque is measured in in·lb, the angular speed is measured in r/min and the values are divided by a conversion factor of 63 025. The power is in units of horsepower. Averaging the speed and torque at discrete points on the unit

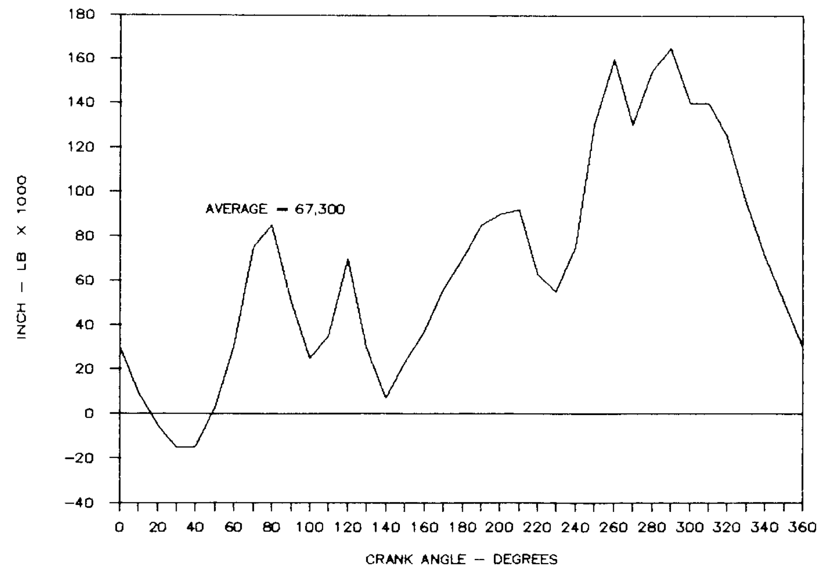

Fig. 5. Dynamic torque curve, unconventional unit.

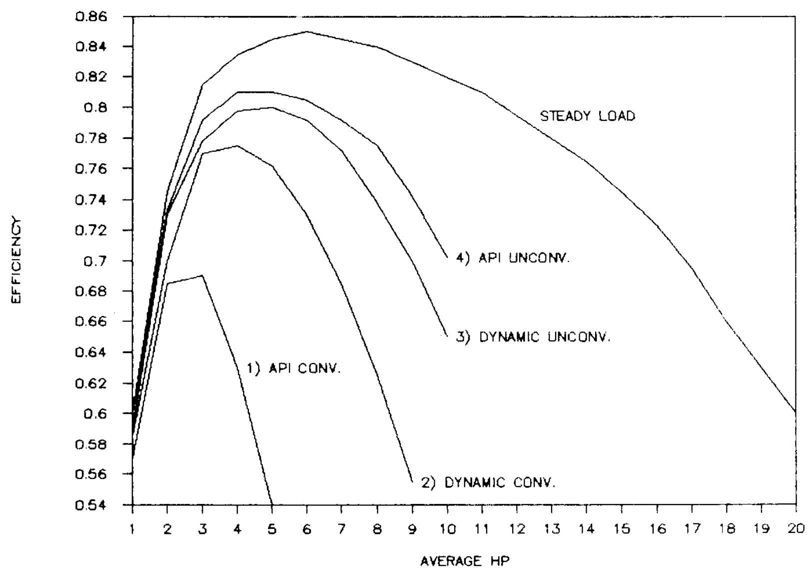

Fig. 6. Efficiency of 10-hp motor at various loads.

performance curve provides average horsepower over a complete cycle. Assuming the time increments between each point on the torque curve is the same, the torque can be integrated to determine an average torque. Dividing the sum of the discrete torque points by the number of points yields the average torque. If the time increments are the same, the average r/min can be used for the speed.

Motor Performance

As with all engineering solutions, the motor size rating is not an exact value but is a tradeoff between cost, size, and service to obtain a competitive device [5]. The manufacturer's steady-load performance curve for a very common 10-hp NEMA D torque characteristic motor is shown in Fig. 6 [6]. There is no particular point on the curve which dictates a rating of 10 hp. 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 is it 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 50-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 influenced primarily 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 are approximately proportional to load, this loss changes with the square of the load [7]. The constant losses dominate the efficiency at low load, while the copper losses dominate at loads greater than 50 percent:

$$\text{motor loss} = I^2R + \text{core loss} + \text{friction–windage loss}.$$

The motor efficiency curve given by the manufacturer is for a constant load over a normal operating range of 25–175 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.

It is necessary to extend the efficiency curves to cover the range of torques experienced by the motor. To determine the high end of the efficiency curve, a linear extrapolation was made up to locked rotor torque. This is the point at which the motor stalls and will no longer move the unit.

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. The losses are proportional to those experienced at an equivalent positive load [8]. To accommodate nonlinearities 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.

Overall motor efficiency was calculated for the various pumping-unit loading configurations. These are plotted on Fig. 6 with the standard steady-load motor efficiency curve. The average horsepower load on the motor is plotted on the ordinate. This is the actual horsepower required by the unit and delivered by the motor.

The average motor horsepower is the polished rod horsepower divided by pumping unit efficiency. Since the unit efficiency is very high, the curves closely represent polished rod horsepower.

The horsepower along the ordinate of the efficiency curves was obtained by scaling the pumping unit torque curves. For example, the average cyclical torque for the static conventional curve is 25 800 in·lb (4.9 hp at 12 r/min). If an average cyclical load of 258 000 in·lb (49 hp at 12 r/min) is required, the curve values are multiplied by ten.

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

$$\text{losses} = \frac{\text{bhp}}{n_{\text{motor}}} - \text{bhp}.$$

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:

$$n_{\text{cyc}} = \frac{\text{actual delivered horsepower}}{\text{actual delivered horsepower} + \text{actual loss}}.$$

The efficiency for the various pumping unit torque characteristics is plotted on the motor performance curve. 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 torque-characteristic conventional unit. The dynamic conventional, dynamic unconventional, and static unconventional operations are presented by curves 2, 3, and 4, respectively. It should be noted that curves 3 and 4 do not represent the same styles of unconventional pumping units.

From these curves several observations can be made. On a conventional unit, the maximum cyclical horsepower that can be started is 50 percent 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 percent 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 percent 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 percent 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:

$$\text{mhp} = \text{prhp}/0.4.$$

The curves indicate that 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 percent of its rated size.

Using the same motor, the efficiencies are consistently greater when 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-percent improved efficiency over the conventional unit.

ECONOMICS

If a motor is sized so that the cyclical load is at 40 percent of the motor rating, the amount of electricity consumed is significantly reduced. If the motor has a 75-percent load, a

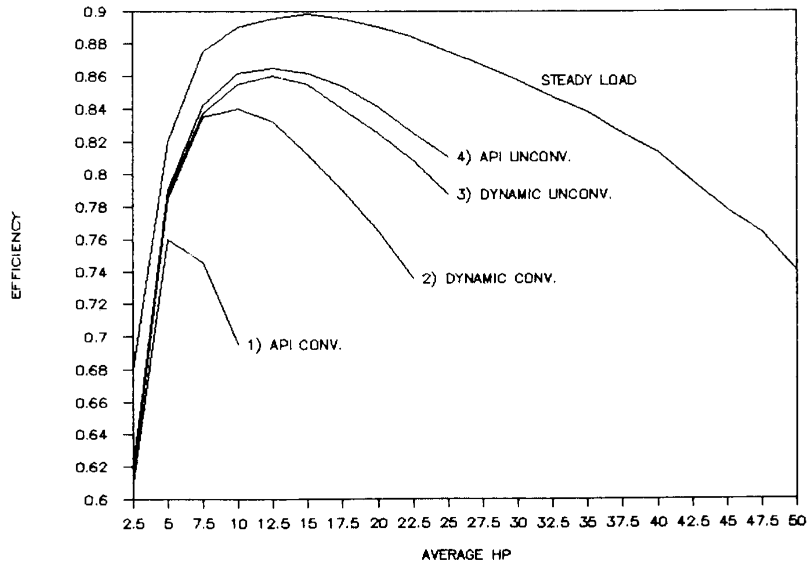

Fig. 7. Efficiency of 25-hp motor.

typical efficiency is 68-percent while a 40-percent load has a typical efficiency of 78-percent. The ten-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 65 350 kWh of energy at 100-percent efficiency, 83 782 kWh at 78 percent, and 96 103 kWh at 68-percent efficiency. The difference in efficiency is equivalent to 12 321 kWh per year. If the energy cost is \$0.06/kWh, the savings is \$739 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. Fig. 7 contains the motor performance data for a 25-hp motor. A motor rated at 25 hp has a maximum efficiency of 90 percent, while one rated at 10 hp has a maximum efficiency of 85 percent. A load of 8 hp represents 68-percent efficiency using a 10-hp motor and 84 percent using a 25-hp motor. This is an improvement of over 21 points, or a 33-percent improvement in energy usage.

The information indicates that motors should be sized to allow for peak gearbox 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 percent of the average mechanical horsepower calculated from producing rates, head, and downhole efficiency.

Other Consideration

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

One detrimental effect of oversizing motors is the lowering of the power factor from 0.87 to 0.74. However, this is easily corrected with capacitors.

One of the most positive effects is that the available starting torque becomes approximately twice as great. Hence 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 r/min for a slip of 6.6 percent, while at 10 hp the speed is 1160 r/min for a slip of only 3.33 percent. This increased slip is one reason for decreased motor efficiency as loading increases.

Field Experience

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 [9]. The study illustrated energy consumption compared to motor loading.

A graphical representation of the energy cost per barrel-foot is presented in Fig. 8. The empirical data supports results of the computer-generated curves in Figs. 6 and 7. The data show a 22-percent increase in cost at low load and a 25-percent 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.

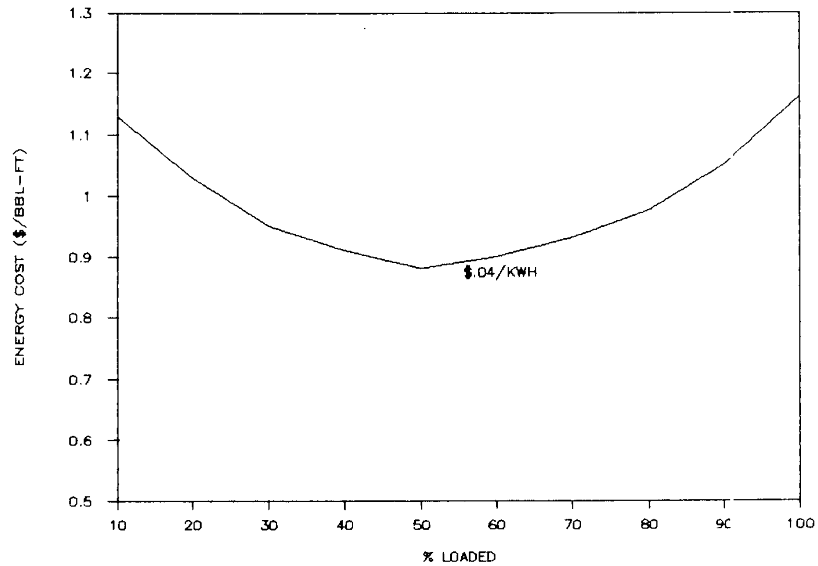

Fig. 8. Cost to pump wells versus percent loaded.

Conclusion

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 percent of its rating.
2) A ten-point improvement in efficiency is obtained when the motor load changes by a factor of two and the final motor load is near 50 percent of the motor rating.
3) A motor provides adequate starting torque for a conventional unit only when the motor rating is two times the average load.
4) Unconventional geometry units are at least five percent 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 percent more efficient than an equivalent conventional unit.

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