APPLICATION AND ECONOMICS OF THE VARIABLE SPEED SUBMERSIBLE PUMP

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ABSTRACT

This paper discusses the techniques used to size available submersible pumping equipment to pump in a variable-speed, constant-head mode. Effects of variable-speed, constant-head operation on the pump and motor are described with their relation to optimum system performance. Energy-usage calculations for a conventional, constant-speed submersible pump and a variable-speed, constant-head submersible pump are compared. Finally, the economics of the variable speed submersible system are analyzed.

INTRODUCTION

The variable-speed submersible pump is an artificial lift system consisting of a standard oil-well submersible pump, motor, protector, and a variable-frequency control panel installed on the surface. Since a submersible pump motor is an induction motor, its speed will be proportional to the frequency supplied by the control panel. The frequency to the motor may be varied as long as the supplied voltage is varied in proportion.[1,2] An example of this may be seen in the submersible pump manufacturers' catalogs where the same motor is rated at 50 Hertz and 60 Hertz. More complete descriptions of how the control panel is capable of varying the frequency and voltage may be found in "An Introduction to Static Variable-Frequency Power Converters and Variable-Frequency Pump Drives" and "The Equipment For and Economics of Variable-Flow Well Pumping" cited above.

The purpose of this paper is to discuss the effects of this variable-speed capability on the submersible pump so as to apply the submersible in new and extended applications.

DEVELOPMENT OF CONSTANT HEAD CURVES

To apply the variable speed submersible pump, it is first necessary to understand the effects of varying the speed of the submersible pump. One way to describe this effect would be to look at the submersible pump manufacturers' curves. For a specific pump we can find a 60 Hertz curve and a 50 Hertz curve. These two curves can be plotted on the same graph to give pump curve for 2 frequencies as can be seen in Figure 1. However, we have a controller which can generate any frequency desired from 24 Hertz to 66 Hertz or higher.

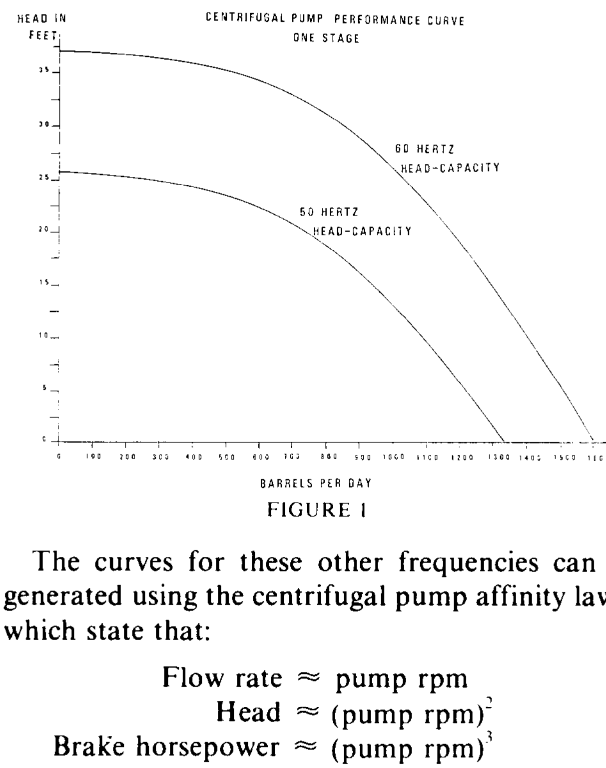

FIGURE 1

The curves for these other frequencies can be generated using the centrifugal pump affinity laws[3] which state that:

$$\text{Flow rate} \approx \text{pump rpm}$$
$$\text{Head} \approx (\text{pump rpm})^2$$
$$\text{Brake horsepower} \approx (\text{pump rpm})^3$$

Using these proportionalities, we can duplicate the 50-Hertz curve from the 60-Hertz curve.

First, by letting the head = 0, the no head flow rate at 50 Hertz will be

$$\frac{50}{60} \times 1600 = 1333$$

Next, by letting the rate = 0, the no flow head at 50 Hertz will be

$$\left(\frac{50}{60}\right)^2 \times 37.3 = 25.9$$

All other points on the 60-Hertz curve can be shifted to the 50-Hertz curve by multiplying the head at that point by $\left(\frac{50}{60}\right)^2$ and by multiplying the rate by $\frac{50}{60}$. The brake horsepower at 50 Hertz will be the brake horsepower at the 60-Hertz point multiplied by $\left(\frac{50}{60}\right)^3$. Finally, to find the efficiency for the new point on the 50-Hertz curve, we take the efficiency for the point we are moving from the 60-Hertz curve and move it over to the new rate point on the 50-Hertz curve.

This technique can now be used to develop a curve for any frequency within the useful limits of 40 Hertz to 60 Hertz. The equations for point conversion may be written as follows.

$$\text{New rate} = \frac{\text{New hertz}}{\text{60 hertz}} \times \text{60 Hertz rate}$$

$$\text{New head} = \left(\frac{\text{new hertz}}{\text{60 hertz}}\right)^2 \times \text{60 Hertz head}$$

$$\text{New Bhp} = \left(\frac{\text{new hertz}}{\text{60 hertz}}\right)^3 \times \text{60 Hertz Bhp}$$

New efficiency = efficiency at 60-Hertz point located at the new rate point.

With the use of these equations, a set of curves may be developed for 45 Hertz, 55 Hertz, and 66 Hertz, giving a family of curves as seen in Figure 2.

While this family of curves does a good job of explaining rpm effects on submersible pumps, it does not expressly describe the variable-speed submersible pump. Another way to approach the problem would be to consider the variable-speed submersible pump as a constant head device. That is, the frequency is varied according to a well's flow rate such that a constant head is maintained by the pump. To pick the constant head to be developed, we might first decide where submersible pumps have historically had long operational runs. Long runs are generally obtained when pumps are operating in a range from their best efficiency point to the high-rate side of their recommended range.

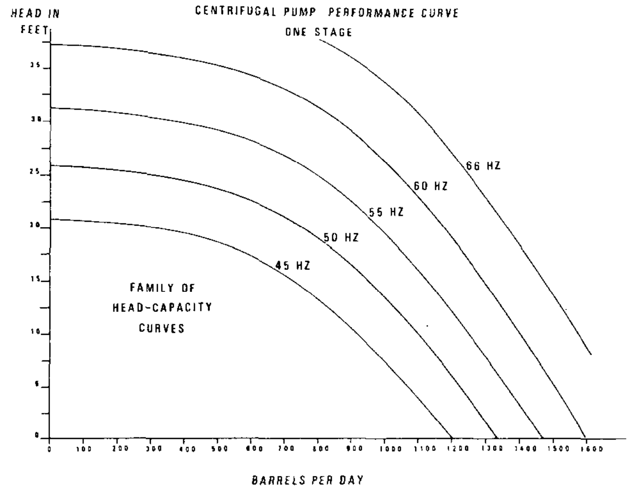

FIGURE 2

For an example, we'll select an operational head of 23 feet on the pump in the first example. That is, the 60-Hertz operation point picked at 23 feet of head and a flow rate of 1100 BPD. Next, a head load line is drawn across the pump curve. This is the head the pump will be required to develop for all productivity rates that a well may experience over its life. We are now required to find the frequencies which will cause the equipment to develop this head at all rates. To do this, a point may be picked along the original head curve. This point may be defined as H_2, R_2, E_2, and BHP_2. First a head conversion is made to determine the new frequency and new rate.

$$\text{Hertz}_2 = \left(\frac{H_2}{23}\right)^{1/2} \cdot \text{60 Hertz}$$

$$R1_2 = \frac{\text{Hertz}_2}{\text{60 hz}} \cdot R_2$$

$$E1_2 = E_2 \text{ placed at } R_2^1$$

$$BHP1_2 = \left(\frac{\text{Hertz}_2}{\text{60 hz}}\right)^3 \cdot BHP_2 \text{ placed at } R_2^1$$

$R1_2$, $E1_2$, and $BHP1_2$ are the new flow rate, efficiency, and BHP in the constant-head mode for Hertz_2. A complete curve is shown in Figure 3.

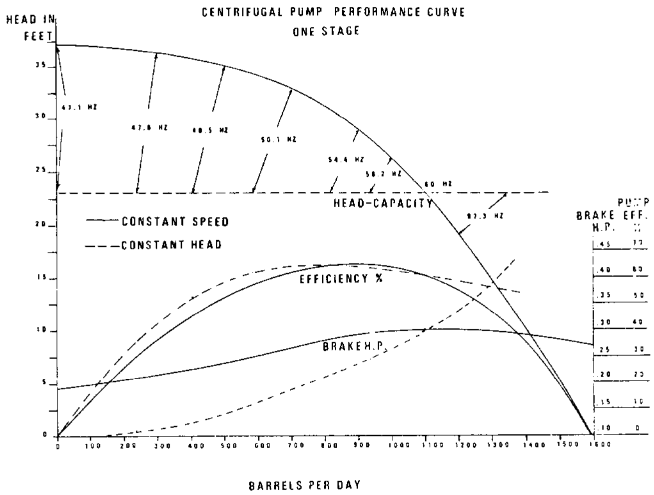

FIGURE 3

EFFICIENCY, THRUST, AND OPERATIONAL RANGE

The above affinity laws hold only over small percentage changes in speed, such as those shown in Figure 3. Even over these ranges, however, interesting effects occur. One of these effects is the ability to take a high-rate pump, slow it down, and run it more efficiently than if a smaller pump had been staged to run at the new rate. This effect occurs because the higher volume pumps have larger best-efficiency points than the smaller rate pumps.

An example is as follows. Pump "A," as shown in Figure 4, is designed to operate at 500 BPD, developing a head of 20.4 feet per stage at 60 Hertz and at 50 percent efficiency. Pump "B," as shown in Figure 5, is designed to operate at 950 BPD, developing a head of 21 feet per stage at 60 Hertz and at 64 percent efficiency. By maintaining 20.4 feet per stage across pump "B" and slowing it down to a speed proportional to 52 Hertz, the impeller would then be moving 500 BPD at 53 percent efficiency. The actual cost of the impellers in pumps "A" and "B" is the same. Of course, to turn pump "B" up to 60-Hertz at 950 BPD and at 21 feet per stage will require some additional investment in initial motor horsepower.

The next point to come to mind is the recommended capacity range of the pump. Pump "B" in the above example has obviously been forced to operate out of its recommended range. There are three reasons for its being necessary to operate a constant-speed pump in its recommended range. These are thrust on individual impellers, thrust on

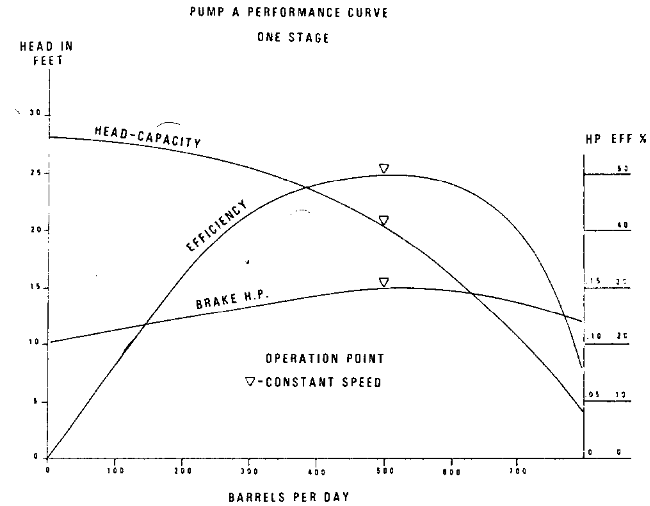

FIGURE 4

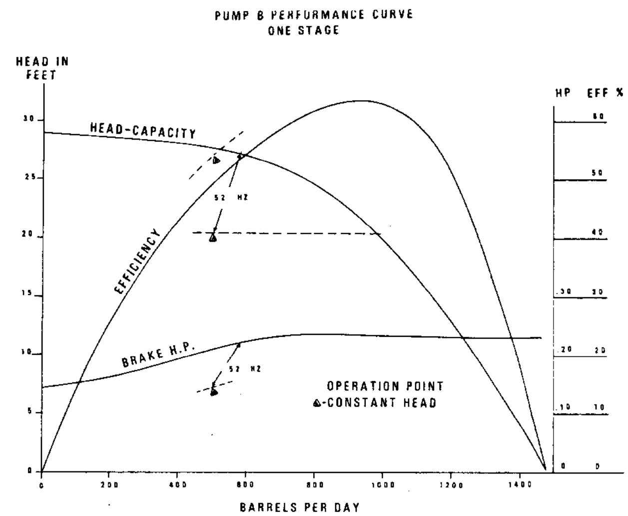

FIGURE 5

the pump's thrust bearing, and loss of motor cooling at low rates.

To look at these effects, it is first necessary to study the classical method of forcing pump "B" to operate at 500 BPD when it was staged for 21 feet per stage at 950 BPD. To make this pump produce 500 BPD, back pressure must be applied at the surface equivalent to 6.7 feet per stage. That is, at 500 BPD the impeller in pump "B" produces 27.7 feet per stage. The pump would be operating 49 percent efficient and requiring 0.21 Bhp per stage.

If the above pump is operated in the variable-speed mode to obtain 500 BPD, the additional 6.7 feet or 32 percent increase in head across the impeller is not necessary. It is this additional head which increases the down thrust on the impeller.[4] So while down thrust is increased somewhat in the variable-speed mode, it is not as intense as in the constant-speed mode.

In the thrust bearing, which carries the shaft load, the wear will be increased as the load increases, and the load increases proportional to the head the pump develops. In the constant-head mode, the load has not increased. Additionally, in the example, the speed or velocity of the thrust runner has been reduced by 13 percent which implies longer life. Also, the lubricating oil has not been heated to as high a temperature, which maintains film thickness, again leading to longer life.

Finally, regarding down hole losses, pump "B" in the constant speed mode requires 0.210 Bhp per stage. Pump "B" operating at 52 Hertz requires only 0.147 Bhp per stage, or 30 percent less power. This lower power requirement implies lower losses and less heating in the variable-speed mode. Pump "B" requires 0.235 Bhp per 950 BPD at peak efficiency or a ratio of $2.47 \cdot 10^{-4}$ Bhp/BPD. At 500 BPD in the constant-speed mode, the ratio is $4.2 \cdot 10^{-4}$ Bhp/BPD or a 70 percent increase. At 500 BPD in the variable-speed mode, the ratio is $2.94 \cdot 10^{-4}$ Bhp/BPD or only 19 percent increase.

It is the sum of the above which helps to verify the ability to extend the operational range of pump "B."

USING THE VARIABLE SPEED SUBMERSIBLE IN WATERFLOOD APPLICATIONS

In West Texas waterflood projects, it is often very difficult to determine the ultimate productivity of a well responding to flood. An operator may install increasingly larger production equipment on a well in attempting to lift fluid at a rate equivalent to the well's productivity.

For a 5000 ft reservoir, a producing well will typically be equipped with a small beam unit when the flood is initiated. As the well begins to respond to the flood, this small unit will become loaded to capacity and the fluid level in the well will begin to rise. The operator will now install larger lift equipment. Let us assume a 456,000 in-lb beam unit is installed. For 5000 feet of lift, the "456" will have a capacity of 400 to 500 BFPD. If the well's productivity continues to increase, the operator could be faced with installing either larger beam equipment or a submersible pump. On many wells in West Texas floods, lift equipment has been upgraded two or three times to keep up with productivity. Each equipment changeout represents significant investments, much of which is not recoverable.

An alternative is the installation of a variable-speed submersible instead of the "456" which was installed to replace the original unit. A variable-speed submersible pump system capable of lifting 900 BPD at 60 Hertz from 5000 feet can be installed at lower cost than the "456" unit on a new equipment basis. Therefore, over two times the capacity can be purchased for less initial investment.

Maintenance of the variable-speed submersible pump will be lower than for conventional submersible pumps. This is because the controller tends to protect all the subsurface equipment from the problems constant speed pumps encounter, such as the following.

1. *Heat:* The controller allows the pump to demand energy only as needed to lift fluid out of the well bore. Therefore, excessive heating is reduced.
2. *Thrust:* The controller allows the pump to develop only the head required to lift the fluid. Therefore thrust is minimized.
3. *Cable Failures:* The controller soft starts the motor. Therefore severe electrical transients are kept off the down hole power cable.

In summary, the variable-speed submersible pump can be installed with less capital investment than a new conventional 456,000 in-lb unit, with a production capacity of up to 900 BPD at 5000 feet of lift.

APPLYING THE VARIABLE-SPEED CONTROLLER TO EXISTING EQUIPMENT

The submersible pump at constant speed is a device which only has one rate for a given lift. This, of course, means that the chances of finding a pump operating in a secondary project which is correctly sized would be difficult. The pump will either have some artificial pressure applied at the surface in order to slow the production rate down, it will be ingesting some gas to make up the volume difference, or the well will have a high fluid level indicating a higher flow rate than designed. Within limits, all of these conditions can be handled by the equipment. These limits are the recommended ranges listed by the pump manufacturers.

When the pump is running at higher flow rates (and therefore lower heads) than it is designed for, we find the equipment will probably have an acceptable run, but the lost production due to a high fluid level is generally intolerable. As a matter of fact, the industry tends to over-size submersible equipment to guarantee that the high fluid level will not be present. The object of investing in a submersible in a secondary project is to pump the well down as close to the formation as possible. To spend 30 to 40 thousand dollars and still find the well not to be pumped down would typically cause a stir in the operations department. This implies that submersibles in secondary projects are probably running with some back pressure at the well head and therefore, at rates lower than design. But the wells are pumped down.

Operating a constant-speed submersible at rates lower than the design rate causes increased down thrust on the thrust rings and thrust bearings. Also, at lower rates, the pump is still requiring relatively high horsepower. This horsepower is causing heating down hole, but now there is less fluid flow past the equipment to cool it. Therefore, the equipment begins to run hotter. If the pump is forced to run at a sufficiently low rate, there will not be enough fluid passing by the equipment to keep its temperature at an acceptable level. This usually occurs after the rate has moved well out of the recommended range. At this point underload devices are used to shut the equipment down so that the fluid level in the well may build back up. The equipment is now cycling.

Cycling has several negative effects. First, there is some lost or deferred production. This will occur any time the fluid level in the wellbore is allowed to rise, creating back pressure on the producing formation. The magnitude of the lost production will be a function of several parameters, including the productivity of the well, the length of the down time period, and the pump capacity after restart. Unfortunately, the characteristics for longer equipment life, such as long down times and long run times, are the same characteristics which maximize production loss. Cycling is also very hard on the submersible pump equipment. The motor must now be started several times every day. Each start imposes severe electrical and mechanical stresses to the down-hole electrical cable, motor, and pump. In addition, the oil in the motor and protector is heated and then cooled with each on-off cycle. When the oil is heated, it expands. This expansion is compensated for in the motor protector. The protector may consist of a mechanical seal which moves or a tortuous fluid path. In either case, the expanding and contracting of the motor fluid will work the protector until an early failure occurs. This allows produced fluid into the motor which then soon causes a failure.

It can immediately be seen that the ability to simply turn the speed of the pump down to compensate for a low flow-rate condition would be an ideal solution. Excessive thrust is not generated. Horsepower requirements are lowered, substantially reducing down-hole heating. The equipment can run continually, eliminating the need to cycle, with its adverse effects including lost production.

To add a variable-speed controller to an existing installation requires a 480 volt, 3 phase source, a variable speed controller, and an appropriate step-up transformer to supply the correct down hole motor voltage. This equipment would replace the existing control panel and the special voltage primary transformers. Unfortunately, at current prices, this equipment may typically cost as much as the submersible equipment on the well. The next point then at hand is how such an expenditure might be justified.

To justify the expenditure, the following points might be considered.

1. An expected need for high lift capacity in the future.
2. Reduction of equipment maintenance costs.
3. Reduction of lost production due to equipment failure.
4. Reduction of lost production due to equipment cycling.
5. Reduction of energy costs.

There are many circumstances in secondary projects which cause a well's productivity to suffer a temporary decline. A well could be experiencing a low producing rate becuase of a temporary lack of injection capacity. In this case, the operator may know that the original lift capacity of the in-place submersible will be needed when injection capacity is returned. The producing well could have suffered some type of skin damage, such as scaling. Again, the operator may know that a scale cleanout will be performed, perhaps as soon as expense money is available and the equipment capacity will again be needed. Situations such as these should certainly make us look closely at the next points on the list to attempt to justify the variable speed equipment.

Submersible pumps had an overall domestic failure rate of 0.64 failures per year at an average cost of $10,906 per failure.[5] Of the $10,906, approximately $2300 is to cover the well servicing equipment. This would leave an average cost of $8600 per failure attributable to the equipment itself. While this average may be high for the West Texas area, we will continue to use it for the purpose of this example. The failure rate of 0.64 represents an average run time of 18.8 months. This, of course, includes all units which are properly sized and experiencing long runs and those units which are running out of range and experiencing short runs. Let us assume that improperly sized units (those more than 20 percent below the low end of their recommended ranges) are experiencing runs of 0.5 times the average or approximately 9 months. We might also assume that properly sized units (those operating from their best efficiency point to the high side of the recommended ranges) are experiencing runs of 1/.5 times the average (approximately 38 months).

If a unit is operating in the improper area as defined above, then we might expect to improve its life by installing the variable-speed controller and slowing the pump down until each impeller was generating only that head required to remove fluid from the hole. This would essentially be the same as operating the pump at its properly sized point. For maintenance benefits, we would first calculate failure costs for each operational case.

Case I
9 mo./failure = 1.333 failures/ year
1.333 failures/yr. · $10,906/failure = $14,541/yr.

Case II
38 mo./failure = 0.316 failures/yr.
0.316 failures/yr. · $10,906/failure = $3444/yr.

Net Savings	$11,097/yr.

Using this maintenance savings figure alone might give a payout of from 2 to 3 years. Of course, each field should be studied individually to determine the values assumed above, but the same methods apply.

Depending upon the circumstances of submersible-pump failure, the time to place a well back on production may be from 2 to 5 days. Assuming 3.5 days as an average and assuming lost production of 200 BOPD at an average cost of $10 per barrel, the following benefits could be generated.

Case I
1.333 failures/yr. · 3.5 days/failure
4.67 days/yr. · $2000 = $9340/yr.

Case II
.0316 failures/yr. · 3.5 days/ failure

1.11 days/yr. · $2000/day	= $2220/yr.
Net Savings	$7020/yr.

This production saving, coupled with the above maintenance saving, has now brought the payout of the variable speed controller down to 1 to 2 years.

Next, we can consider the loss of production due to cycling of the submersible equipment. As stated earlier, this fluid production loss is a function of many variables. After all the variables have been calculated, the basic result will be an average production rate in the cycling mode that is less than the average production rate if the well could be produced continuously at a minimized bottom hole pressure. For the purpose of this discussion, let us assume a production loss of 2 percent. We could then assign a production loss of 4 BOPD or $14,600 per year.

For this example case, with all its assumptions, there are now $32,717 of annual benefits. Assuming the variable-speed controller could be installed for $27,000, the payout period would be 10 months.

As any of the data regarding the well change, then the payout will change. As an example, we might assume that the well only produces 100 BOPD at an average price of $7 per barrel. The benefits now produced would only be $18,699 per year and the payout for the $27,000 investment would be 1.45 years. Of course, increasing production and oil value would shorten the original 10-month payout. Again, the point is that each well and field would have to be studied individually for economics.

Now, the effects of the drive with regard to energy requirements on existing equipment need to be studied. For this, a specific example should be considered. Assume that the pump depicted by the curves in Figure 6 is currently operating at a production rate of 400 BPD. Further, the pump is running at a rate of 600 BPD when on and is cycled 90 min off and 180 min on to achieve the desired production. Other data is as follows.

1. 226 stage pump
2. 60 hp, 840 volt 45 amp motor
3. 5000 feet of No. 4 cu cable
4. 5000 feet of total dynamic head is required to lift fluid into the production vessel.
5. The 60 Hertz design point is 22.12 feet/stage at 910 BPD.

First, the power required while running at the 600 BPD rate is calculated. This will be the power required by the pump divided by the submersible motor efficiency, all added to the I^2R loss in the power cable. From the curve, the BHP required by the pump at 600 BPD is 0.221 hp/stage.

The power required by pump is 0.221 hp/stage · 226 stages = 49.9 hp. Next, per cent motor load is calculated.

$$\% \text{ motor load} = \frac{49.9}{60} = 83\%$$

From motor curves, which can often be obtained from pump vendors, we find that at 83 percent motor load:

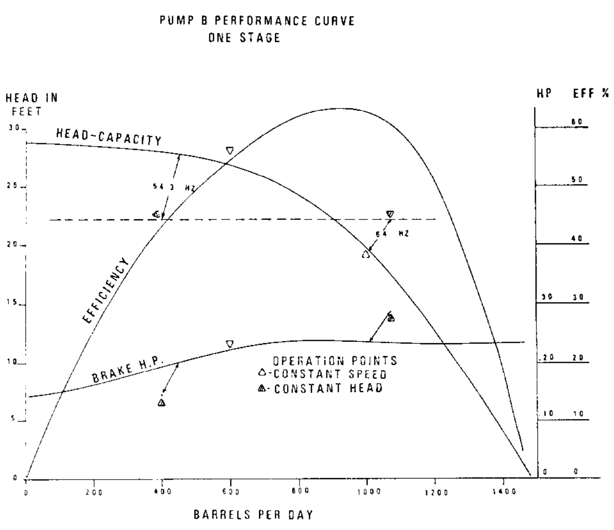

FIGURE 6

motor efficiency = 80.5%
motor amp = 84% of full load amps
= 0.84 · 45 = 37.8 amps

The resistence of No. 4 hard drawn copper is 1.52 ohms per conductor mile. The total resistence of the motor cable would then be:

$$\frac{3 \cdot 5000 \text{ feet} \cdot 1.52 \text{ ohms per mile}}{5280 \text{ ft. per mile}} = 4.32 \text{ ohms}$$

Total system power while running can then be expressed as:

$$\text{Total power} = \frac{49.9 \text{ hp} \cdot .746 \text{ kw/hp}}{.805 \text{ motor eff.}} + \frac{(37.8 \text{ amps})^2 \cdot 4.32 \text{ ohms}}{1000}$$

Total power = 52.41 kw

To find the energy required over a one month period, we calculate:

$$730 \frac{\text{hours}}{\text{month}} \times 0.67 \text{ duty cycle} \times 52.41 \text{ kw} = 25{,}629 \frac{\text{kwh}}{\text{mo.}}$$

Next, the energy required to run the system in the variable-speed mode must be calculated. First, the frequency or percent speed at which the minimum lift requirement of 22.12 feet per stage is satisfied at the required flow rate of 400 BPD is calculated. Through trial and error, it is found that 53.4 Hertz or 89 percent speed meets the lift requirements. Next we can calculate the power required by the pump.

$$\text{Pump BHP} = 0.201 \text{ hp/stage} \cdot \left(\frac{54.3 \text{ Hz}}{60 \text{ Hz}}\right)^3 \cdot$$

226 stages → 32 hp

Again % motor load is calculated, remembering that the motor name plate hp is now only 89 percent of the 60 Hz name plate value.

$$\% \text{ motor load} = \frac{32 \text{ hp}}{(60 \text{ hp})(.89)} = 60\%$$

Going back to the motor curves, we find that at 60 percent motor load:

Motor eff. = 78.5%
Motor amps = 66.5% of full load amps
= (.665) (45) = 29.9 amps

The resistance of the cable path is the same, therefore, the total system power may now be calculated.

$$\text{Total power} = \frac{32\text{ hp} \cdot .746\text{ kw/hp}}{0.785\text{ motor eff.}} + \frac{(29.9\text{ amps})^2 \cdot 4.32}{1000}$$

Total power = 34.3 kw

To find the energy required over a one month period, we calculate:

$$730\,\frac{\text{hours}}{\text{month}} \times 1.00\text{ duty cycle} \times 34.3\text{ kw} = 25039\,\frac{\text{kwh}}{\text{mo.}}$$

The energy savings downhole can now be seen as (25629-25039) kwh or 590 kwh. The really beneficial effects are that the losses in the 53.4 Hertz mode are continuous and spread over the full month. In the cycling mode, these losses are delivered at higher rates, causing larger temperature rises, and motor-oil expansion, and then energy delivery is stopped, allowing the motor oil to contract as discussed earlier.

We must now include the surface losses in our 53.4 Hertz system. From Alcock[2] we find that the variable speed controller is 95 percent efficient at 89 percent of base speed. Also, the step-up transformer will have an efficiency of 98.5 percent. The total system energy for the variable speed case may be calculated as follows.

$$25039\,\frac{\text{kw}}{\text{mo.}} \cdot 1.065 = 26{,}666\,\frac{\text{kwh}}{\text{mo.}}$$

This is within 4 percent of the 60 Hertz cycled pumping operation. The additional losses introduced by the non-sinusodial nature of the voltage and current waves will make the variable-speed case even less efficient. However, the deteriorating effects of cycled operation and excessive down thrust are avoided.

Comparing the variable-speed operation to a continuously operating pump at 60 Hertz and 400 BPD, we find the variable speed system saving over 6000 kwh per month. Of course, this would be a severe operating condition for the pump described.

This covers only an example case for a submersible installation which is running at reduced rates. Now the case may be considered where the submersible is operating at rates higher than the design rate. This implies a higher fluid level than required for pump suction and corresponding back pressure on the formation.

Assume that the pump specified in the above example is running at 60 Hertz, making 1000 BPD. From the curve, it can be seen that the head developed will be 19.7 feet per stage. For the 226 stage pump, this amounts to 4452 feet. This means that an additional 548 feet (5000-4452) of fluid is above the pump intake and, therefore, the formation. If the productivity of the well was such that removing this 548 feet of head would allow an additional 77 BFPD to be produced, then this would just match the capability of the existing 60 hp motor. That is, at 64 Hertz, the pump can lift the full 5000 feet at a production rate of 1077 BFPD. The horsepower required by the pump is 64 hp and the new horsepower rating of the motor is 64 hp. The motor could be overloaded for even higher rates at say 66 Hertz, but the motor manufacturer should be consulted. If the well has an 85 percent water cut, then an additional 11.5 BOPD would be produced. At $10 per barrel, the $27,000 equipment cost would payout in 8 months.

This application is particularly useful in wells in secondary projects whose peak productivity appears to be only slightly larger than the existing submersible's capability. The variable-speed equipment can be installed with very little down time, carry the existing equipment through the peak recovery of the additional oil, and would be available as the productivity of the well begins to decline in the later years of the flood.

In conclusion, it can be seen that justifying a variable-speed controller on an existing installation requires a very thorough study of each individual case. The variable-speed controller simply allows an alternative to up-sizing or down-sizing production equipment or waiting until an early failure occurs and then attempting to resize the equipment.

OTHER POSSIBLE APPLICATIONS OF THE VARIABLE SPEED SUBMERSIBLE

Wells producing from water-drive reservoirs would be a good application for the variable speed submersible system. As in secondary projects, it is often difficult to predict productivity.

Another problem could be an allowable limit. In the early stages of production, the well may be making 100 percent oil and limited to 200 or 300 BOPD. As the well begins to cut water, more total fluid needs to be lifted to make allowable. The variable-speed system could simply be turned up in speed to meet the allowable.

The variable-speed drive might also be used to control such problems as water coning. The variable speed system allows an operator to easily adjust the drawdown on a well after a test of water oil ratio has been made. This technique might be used to maximize production rates from such reservoirs. This might even be made automatic with a net-oil computer feeding water-oil ratios back to the variable speed controller for an automatic speed setting.

Gassy wells or wells with abrasives in the fluid might be successfully handled by submersibles by using very large, over-sized pumps to run slower. These larger pumps would have larger pathways for the abrasives to pass through, thereby minimizing wear contact. The slower turning parts might also have less abrasive wear. Gas in these pumps would have more room to migrate through instead of gas locking an impeller. If a gas lock did occur, a simple increase in speed would compress the gas and again get fluid moving.

The other alternative is sizing a smaller pump to run very fast, 5400 to 7200 rpm. This might have the effect of keeping abrasives in suspension, thereby minimizing wear.

CONCLUSION

The variable speed submersible system can have immediate application in secondary recovery projects. The system is a very viable alternative to larger sucker rod systems. While applications to existing submersibles are harder to justify economically, there should be many cases where the controller installation is the best alternative. As the price of these controls begins to come down due to the quantities being purchased, even more applications will arise.

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