The Correct Method of Calculating Energy Savings to Justify Adjustable-Frequency Drives on Pumps

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Abstract—It is easy to make a bad business decision when using electrical energy savings as a justification to install adjustable-frequency drives (AFDs) on pumps. The simple hydraulic formulas and “rules of thumb” are easily misapplied and the errors will almost always economically favor the AFD installation. To use energy savings as a justification for an AFD installation it is necessary to accurately determine these savings over the life of the equipment. These savings are not dependent upon the AFD or motor characteristics but depend upon the characteristics of the process system. This paper is a tutorial in nature and will show why AFDs save electricity, give examples of the common errors that are made in performing the savings calculations, show how to do these calculations correctly, show how to mathematically model the process to assist in performing the analysis, and show how to perform the economic calculations to arrive at a rate of return and net present value on the AFD investment.

Index Terms—Adjustable-frequency drive, affinity laws, calculation errors, economics, electricity savings, energy-savings analysis, modeling, pump.

NOMENCLATURE

\[ Q \] Volumetric flow rate in gallons per minute (cubic meters per hour).

\[ SG \] Specific gravity. Ratio of density of liquid to that of water.

\[ N \] Speed of the pump impeller in revolutions per minute.

\[ hhp \] Hydraulic horsepower is the power required by the pump to deliver the required flow (i.e., brake horsepower plus pump losses). Per (8), \[ hhp = (H \times Q \times SG)/3900 \] assuming the units of \( H \) is feet and \( Q \) is gallons per minute. The constant 3960 is obtained by dividing the number of foot pounds for 1 hp (33 000) by the weight of 1 gal of water (8.33 lbs).

\[ bhp \] Brake horsepower is the total power required for the pump shaft to do a specified amount of work. \[ bhp = hhp/pump \] efficiency.

\[ hp \] Horsepower is the power required to do work at the rate of 550 ft-lbs/s (i.e., 33 000 ft-lbs/min). 1 hp = 0.7457 kW.

I. INTRODUCTION

ADJUSTABLE-FREQUENCY drives (AFDs) provide many benefits. One of these benefits can be a reduction in electrical energy consumption. Whether energy conservation is being used as the sole or partial justification for installing the AFD, accurate savings values are required to ensure the additional capital cost of the AFD installation will have an adequate rate of return (ROR) on the investment.

The pump formulas that are used to determine the savings are simple, but they are easily misapplied.

The logical method of using control valve power loss to determine the savings is inaccurate.

In essentially all cases, the errors overestimate the energy savings. The result can be a bad investment decision.

It is not difficult to accurately determine the energy savings. The process system must be mathematically modeled so that the pump hydraulic formulas can be properly applied. This modeling is simple, but necessary to achieve accurate results.

Properly applied, AFDs can save a significant amount of electrical energy. Whether or not the AFD is a good investment is dependent upon an accurate cost-savings evaluation. This paper will show how to perform this analysis.

The examples within the paper use American units. The Nomenclature contains the information necessary to convert to SI units.


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II. WHERE IS THE ENERGY CONSUMED?

To control a process, flow needs to be varied. With the pump operating at a fixed speed, at any given flow rate the pressure required by the process is lower than the pressure developed by the pump. The only exception is at the “natural operating point” or “test block flow” where the system and pump system curves intersect (see Fig. 1). To match the pressure requirement of the process, control valves are employed to either:

- directly drop the excess pump pressure (i.e., output pressure throttling); the valves are inserted between the pump discharge and the process as shown in Fig. 2;
- indirectly drop the pump discharge pressure (the control valve recycles additional flow through the pump as shown in Fig. 3); this technique takes advantage of the fact that there is an inverse relationship between pump discharge pressure and flow.

A. Output Pressure Throttling

With output throttling the control valve is directly dropping pressure. As with electrical systems, a device that drops pressure and has flow through it consumes power. This power must be supplied by the electric motor driving the pump.

B. Recycling

With recycling, the control valve increases the flow through the pump body to a point where the pump discharge pressure matches that of the system requirement. As seen from Fig. 1, the bhp requirement of the pump has a proportional relationship to flow, so the electric motor must supply this additional horsepower.

Both output pressure throttling and recycling are extremely inefficient, but, of the two, recycling is the worst.

III. HOW AFDs SAVE ENERGY

As can be seen in Fig. 1, the pump characteristic is not compatible with the system requirements. The process system requires less pressure for reduced flow while the pump discharge pressure increases at reduced flow. As the flow requirement is reduced, there is an ever increasing overpressure that must be dealt with. With a fixed-speed pump, the methods that deal with this overpressure waste electrical power.

In Fig. 1, there is only one flow point where there is no excess pressure to deal with. This is the “natural operating point.” If this intersection could be achieved at any flow rate, the power wastage associated with the over-pressures would be eliminated. This can be accomplished by changing the speed of the pump impeller via an AFD.

The important questions at any given flow rate are the following.

- What is the speed?
- What is the new bhp?

These questions are answered by the “fan laws” or “affinity laws” which govern the relationship between the speed of the impeller, head (or pressure) developed by the pump, and the input power to the pump (i.e., bhp). These laws are shown in

\[
\frac{Q_1}{Q_2} = \frac{N_1}{N_2}
\]
where

- $H$: pump discharge head (ft);
- $Q$: pump discharge flow rate (gal/min);
- $N$: pump impeller speed (r/min);
- $BHP$: pump shaft input horsepower;
- subscript “1” initial condition;
- subscript “2” final condition.

**IV. ERRORS IN CALCULATIONS**

To determine the electricity savings associated with using speed control, the bhp for the fixed-speed and the adjustable-speed case must be determined for the various flows under consideration. For the fixed-speed case, this is a simple matter of using the pump manufacturer’s flow versus horsepower curve (e.g., Fig. 1) or measuring the power into the motor and factoring in motor efficiency. For the adjustable-speed case, it is more complex and it is easy to make errors. As will be shown, the errors will almost always overestimate the power savings.

**A. Errors With Improper Use of Affinity Laws**

Combining (1) and (2) results in

$$
\frac{H_1}{H_2} = \left[ \frac{N_1}{N_2} \right]^2
$$

(4)

which states that head is proportional to the square of the flow.

It appears that as long as an initial flow and associated initial head and horsepower and final horsepower are known, the final conditions automatically fall out of (3) and (4). This is a common mistake. The variables within the affinity formulas are not independent of one another. When the pump speed changes, all the variables change. The result will be that the horsepower prediction will be too low. The following example will highlight this error.

Using the example system shown in Fig. 1, assume the following initial conditions: flow, 4000 gal/min; head, 3282 ft; horsepower, 4025 bhp.

This is shown in Fig. 4. These data could be field measured or could be the design condition of the pump.

What is the pump bhp requirement at, for example, 1000 gal/min?

Using (4) results in

$$
H_2 = 3282(1000/4000)^2 = 205 \text{ ft,}
$$

(5)

Combining (1) and (3) and solving

$$
BHP_2 = 4025(1000/4000)^3 = 63 \text{ bhp},
$$

(6)

The 63 bhp is a significant reduction from the 2506 bhp requirement if the pump was delivering this flow at a fixed speed.

Why the concern? Referring to Fig. 4, the process system curve indicates that 1054 ft of heat is required to deliver 1000 gal/min. This is far in excess of the results of (5).

Why the difference? In Fig. 4, (5) has been plotted over the range of flows and labeled the “Affinity Curve.” It is clear from this plotting of the result that the problem is that the affinity curve does not intersect the system curve at the desired operating point of 1000 gal/min.

The cause of the difference is that an assumption was made that the initial conditions were known, but they were not. None of the three initial conditions of $H_1$, $Q_1$, and $N_1$ required in (1)–(3) are known. Of the three final conditions of $HP_2$, $Q_2$, and $H_2$, only $Q_2$ and $H_2$ are known. There are three equations and four unknowns.

Referring to Fig. 5, the correct approach to use the affinity formulas is as follows.

- An affinity curve must be drawn which will intersect the system curve at the final flow rate.
- Where the affinity curve intersects the pump pressure/flow curve will be the initial flow and head conditions.
- Knowing the initial flow, the pump horsepower curve will give the initial horsepower.
- With the initial horsepower, initial flow, and final flow, (1) can be substituted into (3) to derive the final horsepower.

Redoing the example graphically, the affinity curve would intersect the pump curve at 1908 gal/min and 3837 ft. This corresponds to an initial horsepower of 2966 hp.

Combining (1) and (3) and solving

$$
BHP_2 = 2966(1000/1908)^3 = 427 \text{ bhp},
$$

(7)
This common error, in this simple example, overestimated the savings by 364 bhp. If the objective is to justify the installation of a drive, this error will help. However, with an objective of making a good business decision, this error must be avoided.

B. Errors in Use of Control Valve Power Loss

This is the second type of error that may overestimate the savings. Assume the system in Fig. 1. Just as in an electrical system, when a pipe or valve has flow through it and a differential pressure across it, it is dissipating power. Just as with electrical systems, the power is proportional to both flow and pressure. The formula for this power consumption is shown in

\[ HHP = \left[ \frac{H \times Q \times SG}{3000} \right] \]  

where
- \( H \) = head (ft);
- \( Q \) = flow rate (gal/min);
- \( SG \) = specific gravity;
- \( HHP \) = hydraulic horsepower.

It is common to calculate the horsepower loss across the control valve and assume this will be the amount of bhp that will be saved by using speed control. This seems logical, but it is incorrect, as will be highlighted in the following example.

A pump is connected into a process configuration as in Fig. 6. Assume the following conditions.
- The service is water (\( SG = 1.0 \)).
- The suction pressure of the pump is a constant 500 ft.
- The desired flow of the target line is 3573 gal/min.

The system characteristics are shown in Fig. 7.

What is the power loss across the control valve? Per Fig. 7, the target line requires 1687 ft to deliver 3573 gal/min. The pressure developed by the pump at 3573 gal/min is 3927 ft. The differential pressure across the valve is, therefore, 2240 ft (3927 ft minus 1687 ft). Per (8), the horsepower loss across the control valve is

\[ bhp = \frac{(2240 \times 3573 \text{ gal/min} \times 1.0)}{3000} = 2021 \text{ hp}. \]

Logically, this would lead to the belief that if we get rid of the control valve, the pump will have to supply 2021 hp less. The relationship between pump horsepower output to input at a fixed impeller speed is provided by the pump manufacturer as an efficiency curve shown in Fig. 7. At 3573 gal/min, the pump is 81% efficient. Therefore, for the pump to supply 2021 hp less output horsepower, it will need \( 2021/0.81 = 2495 \) less input bhp.

In reality, the amount of power saved by eliminating the control valve is much less than 2495 bhp. Application of the “fan laws” in (1)–(4) would show that the amount of power saved by using an AFD to control the flow is 2189 bhp, not 2495 bhp.

Why the difference? The assumption that was made in the previous example was that the pump efficiency at any given flow is the same at fixed speed and variable speed. This is not the case. For this particular process configuration, the pump efficiency is much lower at the lower impeller speed than the fixed speed case. This difference is shown in Fig. 8.

This efficiency curve of the pump under adjustable-speed service is dependent upon the process conditions. As shown in this example, using the control valve losses to estimate the power savings overestimated the savings by 306 bhp.

V. MODELING THE PROCESS SYSTEM

The previous examples were very simplistic and graphical tools were used. The examples assumed a pump suction pressure that never changed, a single pump, no parallel flows, and looking for a solution at only one flow point. This is electrically analogous to doing a “snap-shot” short-circuit study at a transformer secondary with an infinite bus, and a series connection of one transformer and no other impedances.

In reality, such a simple system does not exist. There are series and parallel configurations of pumps and lines and solutions at multiple flow points must be found. Just as with an electrical system, the pressures and flows are dynamic. As shown in the examples, the simple methods of determining the horsepower savings generally overestimate the benefits.

With even the simplest pumping/process system, it is impractical to use graphical analytical tools. Before the affinity laws
can be properly applied, the fluid system should be mathematically modeled.

This modeling basically consists of representing the process lines, pump pressure curves, and pump bhp curves as polynomial equations that a computer can manipulate. The individual formulas that determine the flow versus pressure relationship for the lines and pumps are applied to give an overall equation of the pressure and flow for the process system.

This is no different than writing an equation to solve for voltages and currents in a direct current system consisting of wires (process lines) and batteries with a known internal resistance (pumps). Basically, the only difference is that instead of a linear current versus voltage relationship in the wires and batteries, there is a squared relationship between the flow and pressure in the lines and pumps.

Process lines and pump flow versus pressure and pump flow versus horsepower curves can be represented by the polynomial equation

\[ H = a + bQ + cQ^2 \]  

(9)

where

- \( H \) is the head (ft);
- \( Q \) is the flow (gal/min);
- \( a, b, c \) are constants.

Practically, both characteristics can be reasonably represented by simplifying (9) to

\[ H = a + cQ^2. \]  

(10)

A. Process Lines

This can be represented by (10). For pump discharge lines, value “\( a \)” is the “static” head of the system. This is the amount of pressure that must be present on the process line before any flow is possible. For example, in Fig. 1, the static head (constant “\( a \)” ) is 1000 ft.

To find constant “\( c \)” in (10), the head at one flow rate is required. This can be a measured value or, in many cases, it has been superimposed on the pump curve diagram by the process designers.

B. Pumps

Flow versus Head Characteristic: This can also be represented by (10). In this case, constant “\( a \)” is the “shut-in” pressure. This is the pressure developed at the pump discharge at a given suction pressure and at zero flow. For example, in Fig. 1, “\( a \)” is 4000 ft. Both these can be taken directly from the pump manufacturer’s performance curve.

Constant “\( c \)” must be manipulated manually to achieve the “best fit” with the manufacturer’s curve. It may be necessary to manipulate constant “\( a \)” as well, to meet this objective.

Flow versus bhp: This is best represented by a variation of (10), but instead of \( Q \) taken to the power of two, the power is a variable. This is shown in (11). This allows the flexibility to manipulate the flow versus bhp characteristic to match the manufacturer’s pump efficiency curve

\[ \text{bhp} = a + cQ^d \]  

(11)

where \( Q \) is the flow (gal/min) and \( a, c, \) and \( d \) are constants.

This efficiency curve is derived from tests and is provided by the pump manufacturer. Peak efficiency is achieved at one flow condition. The typical shape of this curve is shown in Fig. 8.

All three pump performance curves (head versus flow, bhp versus flow, and efficiency versus flow) are related by a variation of (8) in the form of

\[ BHP = \left[ \frac{H \times Q \times SG}{3960 \times \text{Eff.}} \right] \]  

(12)

where

- \( H \) is the head (ft);
- \( Q \) is the flow rate (gal/min);
- \( SG \) is the specific gravity;
- \( BHP \) is the bhp;
- \( \text{Eff.} \) is the pump efficiency (%).

Since the purpose of the modeling is to determine a reasonably accurate value of energy savings, it is important that the bhp versus flow formula is reasonably accurate. Since the pump flow versus head curve cannot be represented exactly by (10), this inaccuracy can be compensated for by manually manipulating values “\( a \)” , “\( c \)” , and “\( d \)” in (11). This ensures that the bhp versus flow characteristic, when applied in (12) with the pump flow versus head characteristic, will closely match the actual tested values of efficiency.

Series and Parallel Connections: Pumps are often connected in series or in parallel. This is simple to model by representing the multiple pumps as one pump with new characteristics. Multiple pumps can be characterized like multiple batteries.

- **Pumps in Series**: As with batteries, with pumps in series the differential head is additive at any given flow. The bhp is the sum of the number of pumps in series.
- **Pumps in Parallel**: Again, as with batteries, with pumps in parallel the flow rate capability is additive at any given head. The bhp is the sum of the number of pumps in parallel.

C. Effects of Specific Gravity

The advantage of using feet of head as the pressure unit is that the differential head developed by a pump, and the head loss due to flow through a pipe is independent of specific gravity.

What specific gravity affects is the bhp requirement of the pump. The bhp requirement of a pump is directly proportional to the specific gravity of the fluid being pumped. For example, given a pump delivering the same flow of water (\( SG = 1 \)) versus crude oil (\( SG = 0.85 \)), pumping water would require 17.6% more horsepower than pumping crude oil.

D. Modeling Example

Assume a process configuration as shown in Fig. 9.

**Pump**: It has head versus flow characteristic as shown in Fig. 1. Applying (9), constant “\( a \)” is 4000 ft, and “\( c \)” is \(-4.49E-05\). The equation representing the pump is, therefore,

\[ H = 4000 - 4.49E-05(Q)^2 \]  

(13)

**Line A**: Assume we know that the pressure drop across this line is 200 ft at a flow of 4000 gal/min. This may be an actual...
measured value or a typical value based upon the pipe characteristics. Applying (10), constant “c” can be derived as follows:

\[
200 = c(4000)^2
\]

\[
c = \frac{15.6E - 0.5}{5.38E - 0.5}.
\]  

(14)

Since only the pressure drop across the line is required, there will be no value for the variable “a” in (10). The equation representing Line A is, therefore, (10) and (14)

\[
H = 1.56E - 0.5Q_1^2.
\]  

(15)

**Line B:** Assume that this line has the characteristic of the system line shown in Fig. 1. This line has a pressure drop of 861 ft at a flow of 4000 gal/min.

Applying (10), constant “c” can be derived as shown in

\[
861 = c(4000)^2
\]

\[
c = \frac{5.38E - 0.5}{5.38E - 0.5}.
\]  

(16)

Before any flow can be delivered through Line B, the pump must overcome the 1000 ft of static head. The characteristic of Line B is (10) with constant “a” of 1000 ft and “c” from (16) is shown in

\[
H = 1000 + 5.38E - 0.5Q_2^2.
\]  

(17)

The relationship between the flows is shown in

\[
Q_1 = Q_2 + Q_3.
\]  

(18)

The overall equation representing the head versus flow characteristic of the system in Fig. 9 is developed from adding the pressure drops in the system similarly to an electrical system. The system equation is shown in (19)

\[
500 - 1.56e - 0.5Q_1^2 + 4000 - 4.49E - 0.5Q_1^2 - (1000 + 5.38E - 0.5Q_2^2) = 0.
\]  

(19)

If Line C did not exist, \(Q_1\) and \(Q_2\) would be equal and (19) would reduce to

\[
1.14E - 0.4Q_2^2 = 3500
\]

\[
Q = 5541 \text{ gal/min}.
\]  

(20)

The 5541 gal/min in (20) represents the natural operating flow point of the system without the control valve installed. (i.e., the intersection of pump and system curves).

There is a difference from the natural operating point in Fig. 1. Fig. 1 assumed the pump suction pressure never changed (i.e., an infinite pressure bus). The example using Fig. 9 has included a process line between the infinite pressure bus and the pump suction. The suction pressure is, therefore, dependent upon the pump flow. The differential pressure across the pump has not changed from Fig. 1, however, the suction pressure has dropped, thus dropping the natural operating point.

If the same example in Fig. 1 is used but Line C is included, (19) becomes slightly more complex, but still very manageable with simple theorems.

With Line C controlled to a fixed flow of 500 gal/min, (18) and (19) result in

\[
500 - 1.56E - 0.5Q_2^2 + 4000 - 4.49E - 0.5Q_2^2 - (1000 + 5.38E - 0.5Q_2^2) = 0
\]

\[
3486 - 0.0605Q_2 - 1.14E - 0.4Q_2^2 = 0.
\]  

(21)

Using the theorem

\[
Q_2 = \frac{-b \pm \sqrt{b^2 - 4ac}}{2c}
\]  

(22)

when \(a + bQ_2 + cQ_2^2 = 0\).

Applying (22) to (21) results in \(Q_2\) being 5263 gal/min.

How, for example, is (19) used to determine the bhp requirement to achieve a certain flow in Line B by changing the impeller speed? When the impeller speed is lowered, the pump characteristic curve drops. The objective is to drop the curve to a new pump natural operating point that matches the flow in Line B. The unknown in the equation in this case is constant “a” in (9).

**VI. Tabulating Electricity Costs**

With the tools available to calculate the bhp at various flow conditions, the next step is to tabulate the electrical energy consumption for both the fixed-speed and adjustable-speed case. The difference is the savings.

Tabulation consists of five steps for each of the fixed-speed and adjustable-speed cases.

1) Determine the pump bhp requirement for each of the flow rates.

2) Determine the electrical power consumption at each of the flow rates. The inefficiencies to convert the electrical power to motor shaft power must be factored in. These are generally motor, transformer, and AFD efficiency factors. Considerations are the following.

   a) **Fixed-Speed Case:** Motor and captive transformer (if applicable) efficiency will be different at each particular flow rate. This information is readily available from the manufacturers or typical values could be used.

   b) **Adjustable-Speed Case:** Motor efficiencies will be different than the fixed-speed case since the motor speed is different at each flow rate and the drive output waveform may introduce heating in the motor which should be factored. The AFD efficiency must also be factored in. This is available from the AFD manufacturer or typical values can be used. Input transformer and output transformers may be present, whose efficiency should be included.
Fig. 10. Annual flow rate distribution.

### TABLE I
**AVERAGE ELECTRICAL HORSEPOWER REQUIREMENT: FIXED SPEED**

<table>
<thead>
<tr>
<th>Flowrate (GPM)</th>
<th>% of Time at Flowrate</th>
<th>Pump BHP</th>
<th>Efficiency Correction (a)</th>
<th>Electrical HP</th>
<th>Weighed HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1400</td>
<td>5</td>
<td>2.06</td>
<td>1.84</td>
<td>659</td>
<td>659</td>
</tr>
<tr>
<td>1750</td>
<td>5</td>
<td>3.13</td>
<td>0.90</td>
<td>347</td>
<td>347</td>
</tr>
<tr>
<td>2100</td>
<td>10</td>
<td>3.16</td>
<td>0.90</td>
<td>367</td>
<td>367</td>
</tr>
<tr>
<td>2450</td>
<td>15</td>
<td>3.44</td>
<td>0.90</td>
<td>363</td>
<td>363</td>
</tr>
<tr>
<td>2800</td>
<td>15</td>
<td>3.69</td>
<td>0.90</td>
<td>390</td>
<td>390</td>
</tr>
<tr>
<td>3150</td>
<td>20</td>
<td>3.84</td>
<td>0.94</td>
<td>409</td>
<td>409</td>
</tr>
<tr>
<td>3500</td>
<td>30</td>
<td>4.02</td>
<td>0.95</td>
<td>426</td>
<td>426</td>
</tr>
<tr>
<td>5347</td>
<td>0</td>
<td>4.96</td>
<td>0.94</td>
<td>523</td>
<td>523</td>
</tr>
</tbody>
</table>

**Average Electrical HP:** 3973

a) Composite efficiency correction for motors, transformers etc. Values are for example purposes only.

### TABLE II
**AVERAGE ELECTRICAL HORSEPOWER REQUIREMENT: ADJUSTABLE SPEED**

<table>
<thead>
<tr>
<th>Flowrate (GPM)</th>
<th>% of Time at Flowrate</th>
<th>Pump BHP</th>
<th>Efficiency Correction (a)</th>
<th>Electrical HP</th>
<th>Weighed HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1400</td>
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<td>659</td>
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<td>0.89</td>
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<td>1000</td>
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<tr>
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<td>0.91</td>
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<tr>
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<td>0.91</td>
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<tr>
<td>5347</td>
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<td>4.96</td>
<td>0.92</td>
<td>523</td>
<td>523</td>
</tr>
</tbody>
</table>

**Average Electrical HP:** 1559

a) Composite efficiency correction for motors, transformers etc. Values are for example purposes only.

3) Multiply by the annual percentage of time at the flow rate.

4) Sum the weighed electrical power requirements to arrive at an average annual power consumption.

5) Apply the kilowatthour electricity rate to the average horsepower values to arrive at an electricity cost, and subtract the values to determine the annual savings.

For example, consider the configuration in Fig. 9, where the annual flow rate distribution in Line B is shown in Fig. 10.

The average electrical horsepower consumption in this example is tabulated in Tables I and II and is 3973 hp for the fixed-speed application versus 1559 hp for the adjustable-speed case.

The electricity cost is calculated with

$$\$ / \text{yr} = \text{bhp} \times 0.0746 \text{ kW/HP} \times \$ / \text{kWh} \times 365 \text{ days/yr} \times 24 \text{ h/day}. \tag{23}$$

Assuming an electricity value of $0.03/kWh, (23) becomes

- for fixed speed $$\$779,000 \text{ per year}$$
- for adjustable speed $$\$306,000 \text{ per year} \tag{24}$$

### VII. ECONOMIC CALCULATIONS

The value in (24) is in units of present value dollars. It would not be meaningful to state this as the savings per year over the life of the equipment. What is required is to escalate (24) every year over the life of the equipment (i.e., determine the “future value”) and average the future value amounts to arrive at an average savings per year. This is shown in Table III, assuming a 20-year economic life and a 4% annual escalation rate.

When the differential capital cost of the adjustable-speed versus fixed-speed application is determined, this savings number can be used to determine the ROR and net present value (NPV) of the adjustable-speed drive investment.

#### A. NPV

What is NPV? Assume a project that provides an annual income costs $100,000 to implement. There are two choices.

1) Put the $100,000 into an investment with a known interest rate (i.e., discount rate).

2) Spend the $100,000 on the project.

The NPV will be the extra money that was made by putting the $100,000 into the project versus into the investment.

#### B. ROR

The ROR is the discount rate that will give an NPV of 0. In this example, it would be the interest rate of the investment, where, if that rate was available, there would be no advantage to putting the $100,000 in the project or into the investment.
For the AFD example, assuming the differential capital cost of the AFD installation is $650,000, the NPV can be determined using

\[
NPV = \sum_{i=1}^{n} \frac{Value_i}{(1 + Rate)^i}
\]  \hspace{1cm} (25)

where “Value” is the annual cash flow ($), “Rate” is the annual discount rate (%), and “i” is the year (e.g., 1, 2, etc.).

Assuming a discount rate of 15%, this would result in the following:

- NPV of $2,620,000;
- ROR of 77%.

In many cases, especially AFD installations on existing equipment, the differential capital cost may not be within the scope of responsibility or capability of the person doing the AFD savings analysis. In this case, a very useful number is the maximum amount of additional capital that can be spent on an AFD installation to realize the savings.

Most companies know what minimum ROR will be accepted. In the previous example with the cash flow as in Table III, if it was assumed that the minimum ROR that would be accepted was 15%, using (25) this translates into a maximum capital expenditure of $3,724,000.

VIII. EXAMPLE CASE STUDY

This was an analysis performed on a crude-oil shipping system (refer to Fig. 11). This is an existing system that consists of the following:

- ten storage tanks;
- six 600-hp booster pumps configured in three trains of two parallel pumps each;
- six 5500-hp booster pumps configured in two trains of three parallel pumps each;
- system of three parallel pipelines.

The request was to determine the savings by applying AFDs to the 5500-hp shipping pump motors.

Two AFD configurations were studied:

1) one AFD on one motor in one of the series trains;
2) two AFDs on one motor in each of the two series trains.

The flow demand and the pipeline configuration is constantly changing. Historical flow and pipeline availability was used to establish the process framework.
The flow demand and the pipeline availability determines the pressure requirement and, thus, the pump configuration at any point in time. The number of flow scenarios was determined from the following.

- Four different pump/AFD configurations were used in the analysis.
- With each of these four configurations, three pipeline configurations were considered.
- Within each configuration, five flow rates were used.
- The analysis was run for a 20-year period.

This resulted in approximately 600 flow scenarios being calculated to determine the savings.

Case Study Conclusion:

- With one AFD on one of the six shippers, an average of $444,000 per year would be saved.
- With two AFDs installed on one pump in each parallel train, an average of $800,000 per year could be saved.

The savings with two AFDs installed allowed a maximum of $4,200,000 in extra capital to be spent to install the AFD equipment.

This installation was not immediately implemented because the capital cost of retrofitting AFDs into this application was very close to the $4,200,000 maximum limit. Because of the heavy competition for limited capital, other projects, which had a higher ROR, took precedence. This was exactly what this analysis was intended to accomplish. It was not to justify installing AFDs, but to identify the opportunities and present factual data that could be used to allocate limited monetary resources.

IX. Conclusion

- Whether electrical energy savings is the sole or partial justification for installing AFDs, the savings calculations must be done correctly. Otherwise, the consequence may be a bad business decision.
- The simple formulas and “rules of thumb” for calculating the energy savings usually always overestimate the savings.
- The process system must be mathematically modeled to accurately determine the savings.
- The level of savings is totally dependent upon the process configuration (i.e., the efficiency of the motor and AFD are not significant in the overall economic calculations).

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