Comparing Energy Consumption – To VFD or Not to VFD

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You have probably noticed that there are several camps out there when it comes to centrifugal pump applications involving variable frequency drives (VFD's). One group believes that every pump should have a one while another thinks that they should be banned altogether. And, of course, there is the middle ground that says it depends upon the pump and application.

When we think of VFD control, the first thing that comes to mind is energy savings. There are, however, other benefits that can often justify the cost of this control technique whether there is energy savings or not. For example, soft start and stop reduces mechanical and electrical stress while virtually eliminating water hammer. In some instances, this alone can double the life of a pump and motor. Single to three phase conversion and open delta current balancing are two other popular, non-energy related applications and, you still get soft start and stop as a bonus. But if saving energy is the key purpose, it is very important that we understand how to compare energy consumption in the same application during constant and variable speed control. Lets take a look at a simple (water based) constant pressure / variable flow (cp/vf) application that uses the same pump and motor operating under both control schemes.

Motor, Pump, & Total Efficiency

We will start by defining efficiency since it is efficiency that will have a large influence on the amount of power consumed by either application. The efficiency of a typical machine is simply output power divided input power. If a motor consumes 1 kW of electrical power and produces 1 HP (0.746 kW), its efficiency is 74.6%. The same is true for the pump. Since 1 HP is equal to 33000 ft-lb/min, the following equation will give us the theoretical HP required (or generated) at any point on a pump performance curve:

HP = (head in feet X flow in gpm X 8.33 lb/gal) / 33000 ft-lb/min

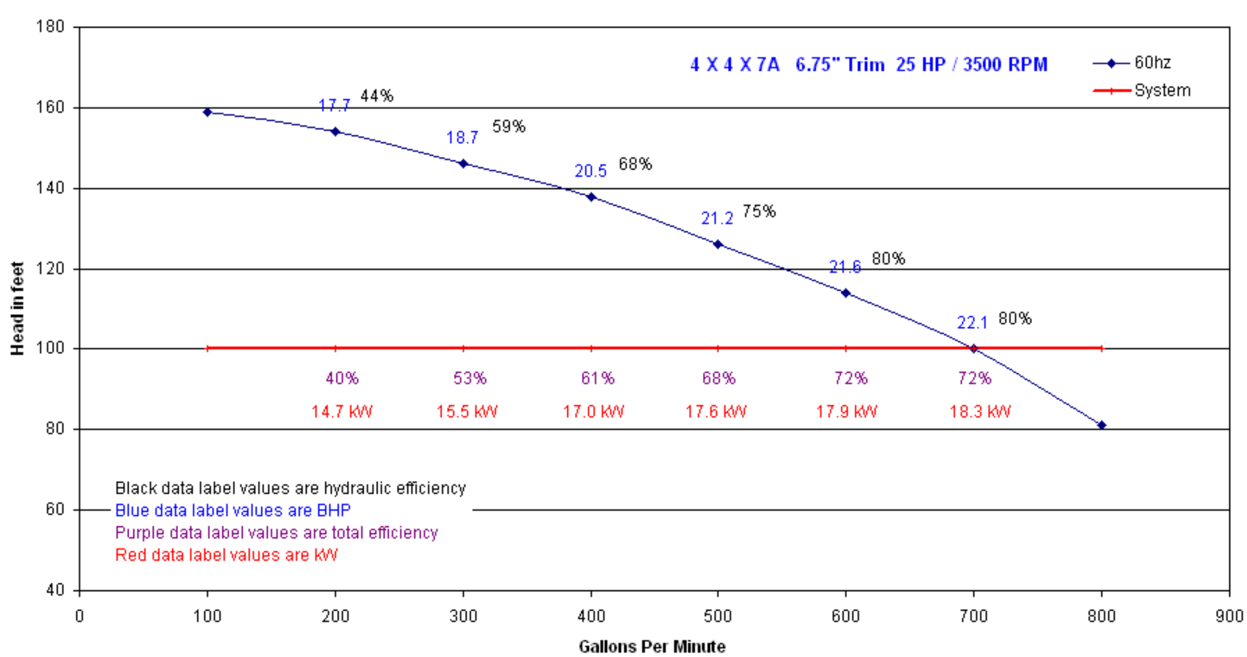
If you divide the theoretical HP by the actual HP measured during testing, the result is pump efficiency at that point.

Now, efficiency doesn't stop with the individual efficiencies. Any time we couple a motor to a pump we have to look at the efficiency of both machines working together (total efficiency). For example, suppose we have an electric motor that is 90% efficient at its rated speed and directly couple it to a pump that is 80% efficient at its BEP. How do we calculate the total efficiency? One approach would be to calculate the average of the two efficiencies. Another might be to assume that the system would operate at the lower of the two. Unfortunately, neither method is correct and either result would be quite a bit higher than the actual total efficiency. It turns out that the total or system efficiency is the product of the individual motor and pump efficiencies. In our example the total system efficiency at BEP is 72% (0.8 X 0.9). So at any point on a pump's performance curve, total efficiency will be the product of the motor and pump efficiencies at that specific operating point.

Another factor that can affect both motor and total efficiency is motor loading. The peak efficiency of a typical three phase motor occurs at about 75 – 80% of its rated load but it will maintain a fairly flat efficiency curve between 60% and 100% when operating at rated speed. When loading drops below 60%, efficiency begins to drop and around 40% it will drop rapidly. This is not an issue in a typical, constant speed pump application as long as the HP required at minimum flow is, at least, 60% of rated motor HP.

Constant Speed

Figure 1 puts our discussion, so far, into perspective. It shows a 4X4X7 end



suction centrifugal pump, with a manufacturer approved flow range of 200 - 800 gpm, operating in a constant pressure application. A pressure reducing valve (PRV) is employed to maintain pressure at 100' across a flow range of 200 to 700 gpm. The system curve (red) ignores any downstream changes due to friction and takes the form of a simple horizontal line. The blue data labels above the performance curve show the actual HP required at each major flow point and are based upon the hydraulic efficiency (black data labels) at that same point. As you can see the system requires approximately 22.1 HP at 700 gpm and drops to about 17.7 HP at 200 gpm. This reduction in power is just what we would expect to see when a centrifugal pump is throttled at the discharge by a valve. Since the minimum load HP (17.7) is well above 60%, we can assume that motor efficiency (90%) remains relatively constant across the full range of flow.

Now, what we have discussed so far is the HP required at each point - - not the power required to produce that HP. The purple data labels below the system curve show the total efficiency at each flow point and assumes a motor efficiency of 90%. In order to obtain the actual kW consumed we could recalculate the HP based on total efficiency and multiply by 746 or just use the following equation: [kW = (HP X 0.746) / motor eff]. This equation uses the original HP measurement and also takes into account motor efficiency. The red data labels show the power, in kW, consumed at each major flow point. As you can see, they range from 18.3kW at 700 gpm down to 14.7kW at 200 gpm.

Variable Speed Control

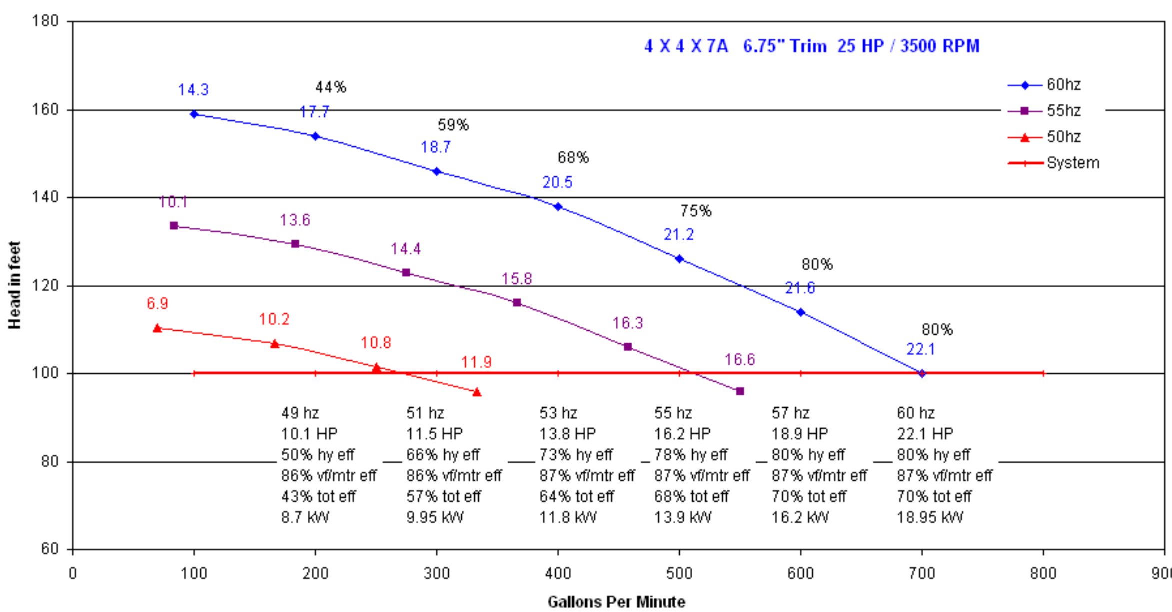
In this version of the application, the control valve is removed and a VFD is employed to control pump speed and system pressure across the same range of flow. When we introduce a VFD into the system we are essentially adding another machine to the equation so we have to take a look at its effect on the rest of the system.

The VFD impacts the system in several different ways. The first is its own efficiency - - how well it converts AC to DC and then pulses that DC to emulate an AC wave form. Energy losses are mostly in the form of heat and if you check the rated efficiencies of several high quality drives you will find that they range from about 97 to 99%. For our comparison we will take the middle ground and assume that the drive in our example is 98% efficient.

The second impact is due to the non-sinusoidal (pulsed) nature of the current supplied by the drive. This can give rise to harmonic losses in the motor and can result in an overall reduction in motor efficiency of about 1%. Since we will be using the same motor (90% eff) in this example we will assume that harmonic losses will reduce its efficiency at rated speed and load to 89%.

Finally reduced motor loading due to decreases in speed can also affect motor efficiency although the decreases can be much smaller than those seen at fixed speeds. This quantity is a bit harder to define because it varies by motor size. For example a typical 100 HP motor will experience a drop in efficiency of about 2.5% when operated at 30 hz (1/2 speed). Smaller motors, especially those under 10 HP, will experience a larger drop. From the information I have been able to obtain from several motor manufacturers, motors 10HP and above, under VFD control, will experience no drop in efficiency down to 50% loading and only a small drop (1%) around 40% loading. In our example we will assume an additional 1% drop in motor efficiency (88%) at the two lowest load points. (This is a topic that should be addressed in greater detail by the motor and drive industry.)

In figure 2 we see the original fixed speed, 60 hz curve plus the performance curves for the speeds produced at 55 and 50 hz. The colored data labels are HP required while hydraulic efficiency, at full speed, is again shown in black. The points where the frequency curves cross the system curve is the flow point for that particular frequency at 100 feet of head. Now, it might be more meaningful if we showed each individual frequency curve. But, since the typical VFD has a resolution of 1/100 of a hz, the number of curves (1200) would make the graph unreadable so you will have to visualize those additional curves on your own.



The tables below the system curve show the variable speed results for each major flow point. They include frequency to the nearest whole hz, HP required, hydraulic efficiency, vf/mtr efficiency (includes both drive and motor losses), total efficiency, and power in kW. For flow points where motor loading is greater than 50%, vf/mtr efficiency is equal to drive efficiency X motor efficiency (0.98 X 0.89 = 87%). At flow points where motor loading is less than 50% (200 & 300 gpm) motor efficiency is reduced to 88%.

If you compare the power required at 700 GPM you will notice that the variable speed application uses about 3.5% more than the fixed speed example even though the HP required is the same - - not exactly what we would expect from a power saving application. Fortunately, this is the only point on the system curve where this situation occurs and it is a function of the lower vf/mtr efficiency (87% compared to 90% for fixed speed) and the fact that head is the same for both applications. But, an interesting trend begins just below full flow. Even though vf/mtr efficiency remains lower at every flow point, the power required decreases quickly. At 600 gpm it is reduced by 9.5% compared to fixed speed operation. At 500 and 400 gpm it is reduced by 21% and 31% and at 300 and 200 gpm it is reduced by 36% and 41% respectively.

These rather large power reductions are explained by a slight variation of the equation we used earlier. The kW required at any point on the system curve is equal to 0.746((QXH)/(3960Xtot eff)). Flow (Q) changes equally in both applications but, unlike PRV control, a reduction in pump speed also affects the other two variables . First, head is reduced with a corresponding reduction in speed and kW is directly proportional to both head and flow at any point. For example, at 300 gpm operating head is reduced by 46' and even at 600 gpm it is reduced by 14'. Secondly, hydraulic efficiency moves to the left with flow as speed decreases and the kW required is inversely proportional to pump efficiency at those same points. At 300 gpm hydraulic efficiency is 59% under PRV control but under VFD control it increases to 66%. (See my November 2006 column "Preservation of Efficiency") Taken together, these two variables have a sizable impact on the power required at any flow point.

A very important result of this comparison is the fact that a relatively small change in speed can result in a much larger reduction in power. Over the entire range of flow, speed varies by a maximum of 641 RPM (18%) but the maximum power reduction over that range is 6 kW (41%) compared to PRV control. Even at 500 gpm a speed reduction of just 291 RPM (8%) results in a power savings of 3.7 kW (21%).

Other Considerations

The potential energy savings available through variable speed control depends, to a large degree, upon the pump selected for a particular application. BEP efficiency is important but the "range" of that efficiency is equally important. In our example BEP efficiency is a healthy 80% but, at mid point on the system curve (350 gpm) it is still 64% at rated speed and about 70% at its variable speed. Remember that efficiency moves to the left with a reduction in flow and that movement will result in higher efficiencies at lower flows. Another consideration during pump selection is the curve shape. Our example exhibits a rise in head of about 50 feet from full to minimum flow and allows a control range of 12 hz. Although flatter curves can be ideal for circulation applications (vp/vf), where head is primarily a function of friction, they may not offer the control range needed for applications that require higher static heads. Now, there are alternative VFD control techniques that can address these flatter curves but, with hundreds of pump models available from dozens of manufacturers, chances are you can find the proper curve shape for almost any application. Finally, pumps must be matched to the "actual" conditions of the application. Unfortunately, many systems tend to be over sized and have a margin of safety that is often equal to the expected flow! The pump used in our example is sized for a maximum flow of 700 gpm and a projected "typical" flow of 400 – 600 gpm. If the maximum flow remains the same but the projected typical flow is reduced to 200 – 400 gpm, a duplex system, consisting of two smaller pumps, would be a better choice. The control scheme could employ two VFD's for variable speed control of each pump and a PLC to supervise their operation. Another option would be to use a single drive that would engage an across the line starter and bring a constant speed pump on line when flow exceeds 400 gpm. We will discuss these control options in detail in a future column.

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