ARE YOUR VARIABLE SPEED PUMPING APPLICATIONS DELIVERING THE PREDICTED SAVINGS? IMPROVING CONTROL TO MAXIMIZE RESULTS

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ABSTRACT

About one-fifth of all U.S. utilities incentivize variable frequency drives (VFDs) (NCSU 2014), and many of these drives control pumping systems. Field studies and research show that few variable-flow systems are optimally controlled and the fraction of actual-to-ideal savings is frequently as low as 40% (Kissock 2014a; Ma 2015; L. Song, Assistant Professor, Department of Mechanical Engineering, University of Oklahoma, pers. comm., July, 2013.). Utility incentive programs that rely on ideal energy saving calculations could overestimate savings by 30% (Maxwell 2005).

Previous work has shown that excluding the effects of changing motor efficiency, VFD efficiency, pump efficiency, and static head requirements results in overestimating savings (Bernier and Bourret 1999; Maxwell 2005). This work considers the difference between actual and ideal savings caused by bypass, position and setpoint of control sensors, and control algorithms. This paper examines the influence of these factors on energy savings using simulations, experimental data, and field measurements. In general, energy savings are increased when bypass is minimized or eliminated, pressure sensors for control are located near the most remote end use, and the pressure control setpoint is minimized.

INTRODUCTION

According to some estimates, pumps account for between 10% and 20% of world electricity consumption (EERE 2001; Grundfos 2011). In industrial applications, pumps frequently account for 25% of plant energy use (EERE 2001). Unfortunately, about two thirds of all pumps use up to 60% too much energy (Grundfos 2011). A primary reason is that, although pumps are designed for peak flow, most pumping systems seldom require peak flow and the energy efficiency of flow control methods varies significantly. Before variable-frequency drives (VFD), bypass and throttling were common, but inefficient, methods of varying flow to a specific end use. Today, the most energy-efficient method of varying flow is by varying pump speed with a VFD. Previous work has shown that excluding the effects of changing motor efficiency, VFD efficiency, pump efficiency and static head requirements results in overestimating savings (Bernier and Bourret 1999; Maxwell 2005).

The quantity of energy saved in variable-flow systems is highly dependent on other factors in addition to motor, pump, and VFD efficiencies. Field studies and research show that few variable-flow systems are optimally controlled and the fraction of actual-to-maximum savings can be as low as 40% in poorly-controlled flow systems (Song 2013; Kissock 2014a; Ma 2015). In fact, it is not uncommon to find instances where VFDs actually increase energy use when VFD efficiency losses are greater than pump power savings (Seryak, John. 2014. "Fan and Pump System Low-Cost/No-Cost Energy Efficiency Opportunity Identification”. Triple Point Energy Saving Seminar, Chillicothe, OH, April 10, 2014.).

This work considers the difference between actual and ideal savings caused by bypass, position and setpoint of control sensors, and control algorithms. The paper begins by defining “ideal” flow control as the most energy-efficient type of flow control, and compares pump power from reducing flow by throttling to the ideal case. Since some minimum flow is required in most pumping systems, the effect of bypass flow on pumping energy use is considered. The control variable for most variable flow pumping systems is pressure; hence, the effect of the location and setpoint value of pressure sensors on pump power is considered. Finally, a case study is presented which demonstrates how pumping energy can be reduced through application of these ideas.

IDEAL FLOW CONTROL

In order to consider the effect of bypass and control on pumping system energy use, it is useful to define the maximum savings that can be expected from reducing flow. Figure 1 shows two operating points of a pumping system. Point 1 represents the pump operating at full flow. Point 2v is the operating point if pump speed is slowed by an optimally-controlled VFD. Point 2v lies on a system curve in which dh (the pressure head) approaches zero as volume flow rate approaches zero and pump head varies with the square of volume flow rate. This ideal case represents the minimum pumping power that can be expected when flow is reduced from $\dot{V}_1$ to $\dot{V}_2$. The reduction in pump power is also defined by the pump affinity law shown in Equation 1, where W is fluid work and $\dot{V}$ is volume flow rate at respective operating points.

$$W_{2V} = W_1 \times \left(\frac{\dot{V}_2}{\dot{V}_1}\right)^3 \qquad 1$$

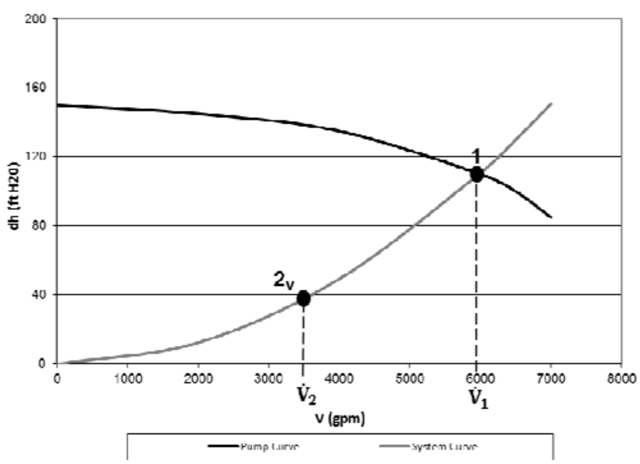

Figure 1. The ideal system curve approaches zero head at zero flow.

Few actual pumping systems achieve the power reduction defined by the pump affinity law because of throttling, minimum flow constraints, static head requirements due to changes in elevation, velocity and pressure between the inlet and outlet of the piping system, and control losses. In the following sections, these deviations from the ideal case are investigated.

THROTTLED FLOW CONTOL

One of the most common methods of varying flow is to throttle flow by partially closing a valve in the piping system. In this section, we compare pump power from reducing flow by throttling to reducing flow by slowing the pump with a VFD. The University of Dayton Hydraulics Lab (UDHL) is equipped with two pumps, a VFD, a parallel piping network with four branches, pressure sensors, and flow meters. In Figure 2, Point 1 is the operating point with the pump at full flow. Point 2_T is the operating point when flow was throttled to volume flow rate, $\dot{V}_2$. Point 2_V is the operating point when pump speed was slowed by a VFD to volume flow rate, $\dot{V}_2$. Pump power is proportional to the product of head and flow, which is represented by the rectangular area defined by each operating point. The data show that when flow was controlled by throttling at Point 2_T, the pump ran at 1,800 rpm and consumed 8.5 kW. When flow was controlled by the VFD at Point 2_V, the pump ran at 1,180 rpm and consumed 3.25 kW; pump power decreased by 62%. Clearly, reducing flow with a VFD is more energy-efficient than throttling flow.

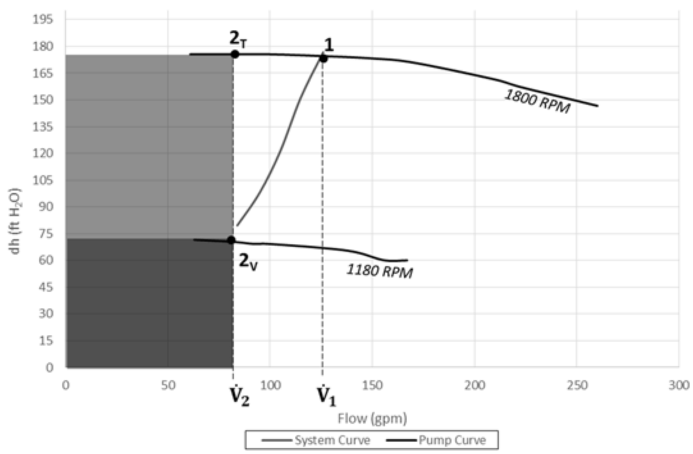

Figure 2. Measured energy penalty due to throttled flow.

In pumping systems the power transmitted to the fluid, W_{fluid}, is given by Equation 2 where dh is the pressure head across the pump and $\dot{V}$ is the volume flow rate.

$$W_{fluid}[kW] = \frac{dh[ft\ H_2O\ ] \times V[gpm]}{3{,}960\ [gpm \cdot ft \cdot H_2O/hp]} \times 0.746 \left[\frac{kW}{hp}\right] \qquad 2$$

At Point 2_V, the fluid work was:

$$W_{fluid}[kW] = \frac{71.6\ ft\ H_2O \times 81\ gpm}{3{,}960\ gpm \cdot ft \cdot H_2O/hp} \times \frac{0.746\ kW}{hp} = 1.09\ kW$$

However, the electrical power to the pump motor is always greater than fluid work because of efficiency losses in motor, pump, and VFD. Using the measured power draw and calculated fluid work, the combined efficiency, $\eta_{combined}$, is given by Equation 3.

$$\eta_{combined} = \frac{W_{fluid}[kW]}{P[kW]} \quad 3$$

At Point 2_V, the combined efficiency was:

$$\eta_{combined} = \frac{1.09\ kW}{3.25\ kW} \times 100\% = 34\%$$

The combined efficiencies at Points 2_T and 2_V were 32% and 34%, respectively. These results indicate that about 70% of pump power was lost due to inefficiencies in the motor, pump, and VFD.

MINIMUM FLOW CONSTRAINTS

Most pumping systems require some minimum flow due to constraints such as minimum VFD speed, minimum flow through the pump, or minimum flow through equipment such as a chiller evaporator. Hence, a bypass is needed to provide a path for minimum flow when end uses reduce flow below this minimum. In the ideal case, no bypass flow is permitted when the end uses require more than minimum flow. Flow in excess of the minimum required flow increases pumping power and wastes energy.

Ideal Bypass Flow Control

"Primary-only" pumping systems use a single pump (or set of pumps in parallel) to move fluid through a chiller and to end uses. In primary-only pumping systems, best practice to meet a minimum flow constraint is to add bypass piping with an actuated 2-way valve to the piping system (Avery 2009). The bypass valve is closed when the end uses require more than minimum flow. The valve opens to allow minimum flow based on a flow meter as shown in Figure 3 (Taylor 2012) or based on the pressure difference through the chiller as shown in Figure 4 (J. Fauber, Senior Mechanical Engineer, Heapy Engineering, E, pers. comm., May 2015).

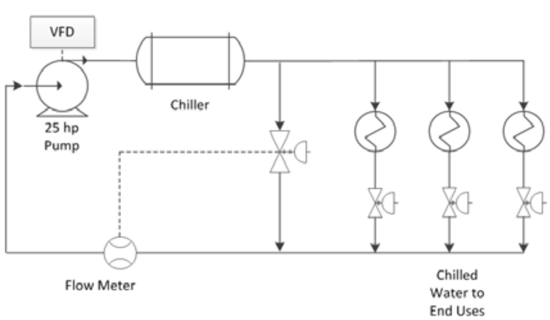

Figure 3. Primary-only system with 2-way bypass controlled by flow meter.

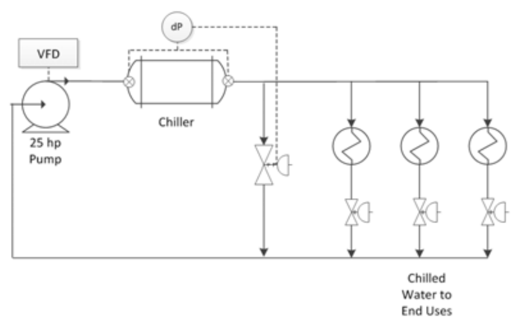

Figure 4. Primary-only system with 2-way bypass controlled by differential pressure.

Another way to guarantee minimum flow through chillers is by dedicating "primary" pumps to move water through the chiller(s) and "secondary" pumps to move water to end uses. In primary-secondary systems, constant-speed primary pumps provide minimum flow to the chiller(s) which eliminates the need for a bypass valve as shown in Figure 5.

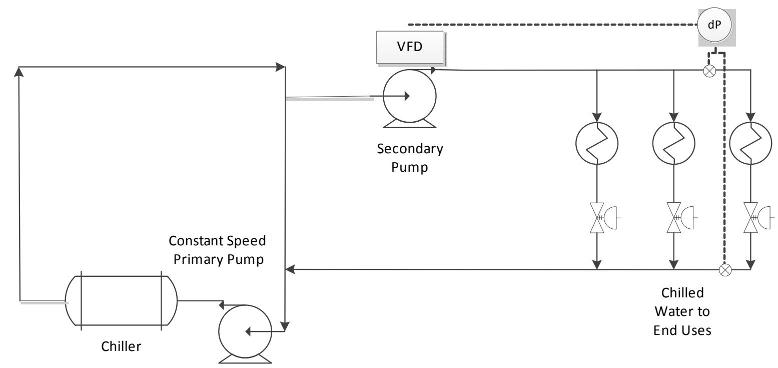

Figure 5. Primary-secondary system eliminates the need for a bypass valve

Excess Bypass Flow

The effect of excess bypass flow on energy savings in variable-flow pumping systems was experimentally measured in the UDHL. In this experiment, flow through "the process" end use was 81 gpm with the manually-controlled bypass valve closed. Next, flow through "the process" end use was maintained at 81 gpm with the bypass valve about 50% open. Figure 6 shows that with the bypass valve closed, the VFD ran at 1,180 rpm, total flow was 81 gpm, and the operating point is 2_{VC}. With the bypass valve 50% open, the VFD ran at 1,212 rpm, total flow was 134 gpm, and the operating point shifted to 2_{VO}. Since flow through the process end use was maintained at 81 gpm, 53 gpm traveled through the bypass loop. In this case, pump power increased from 3.25 kW to 4.12 kW or about 27%. When the manually-controlled bypass valve was fully open, pump power increased by 54%. This indicates that if the effect of excess bypass flow is neglected in savings calculations, those calculations will significantly overestimate savings. These results also suggest that savings can be significantly increased by minimizing excess bypass.

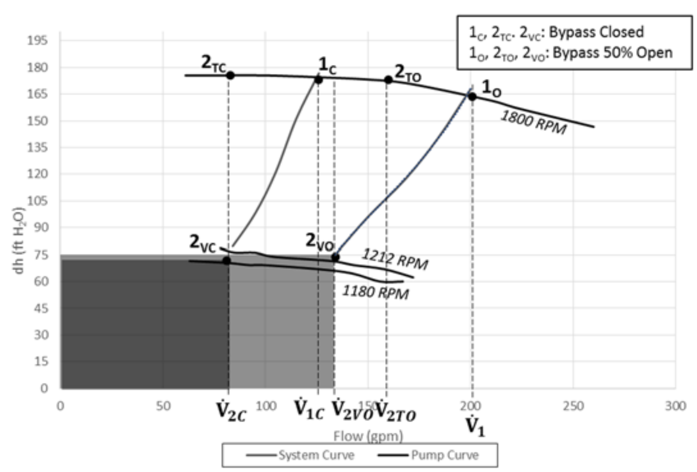

Figure 6. Measured energy penalty associated with bypass 50% open

Three common piping systems that result in excess bypass flow are shown below. The first is a 3-way valve at one or more end uses as shown Figure 7. The 3-way valve(s) always allows some bypass flow, thus wasting pumping energy.

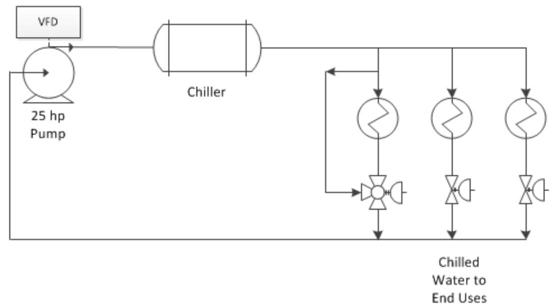

Figure 7. 3-way bypass valve at one end use.

Another piping system that allows excess bypass flow uses a manually-controlled bypass valve as shown in Figure 8. Manually-controlled valves are always at least partially open and hence allow bypass flow even when no bypass is required. The recommended commissioning practice in a system with a manually-controlled bypass valve is to close all end use loads and throttle to allow minimum required flow (J. Kelley, Energy Engineer, Plug Smart, LLC, E, pers. comm., February 2015.). However, when end use valves close, flow through the bypass valve increases due to rebalancing of flow, allowing more than minimum flow through bypass. Rebalancing of flow is shown experimentally by the difference in flows $\dot{V}_{2TO}$ and $\dot{V}_{2VO}$ in Figure 6. In industrial systems, manually-controlled bypass valves are often neglected after installation and allow unmonitored excess bypass flow.

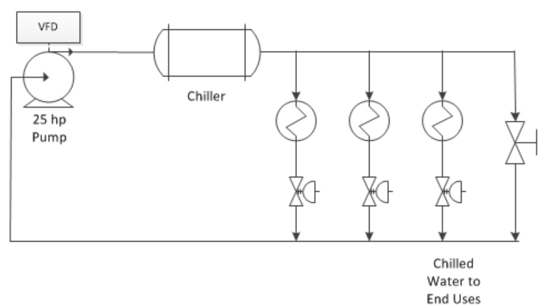

Figure 8. Manually-controlled bypass valve.

A third common piping system that allows excess flow uses an automatic flow-limiter on the bypass pipe as shown in Figure 9. Automatic flow-limiters are better than manually-controlled valves since they prevent rebalancing of flow. However, they still allow a fixed amount of bypass flow when no bypass is required, resulting in wasted pumping energy.

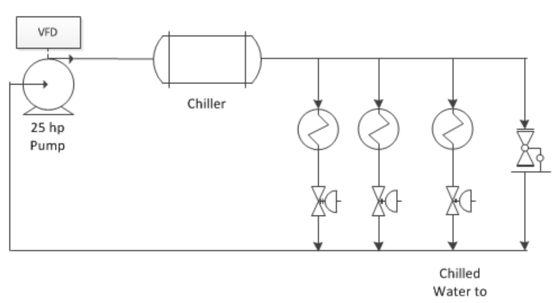

Figure 9. Automatic flow-limiter bypass valve.

PUMP SPEED CONTROL

Fixed Pressure Setpoint Control

Automatically-controlled VFDs modulate pump speed based on data from a control variable; the most common control variable is pressure. The setpoint pressure determines the y-intercept of the system curve; hence, a high pressure setpoint increases pump power at all flows. This concept is demonstrated in Figure 10. The rectangle defined by Point 2_{V80} represents pump power when the pressure setpoint value is 80 ft. H_2O. The rectangle defined by Point 2_{V40} represents pump power when the pressure setpoint value is 40 ft. H_2O. The difference in the size of the rectangles represents the additional energy associated with setting the dP at 80 ft. H_2O compared to setting it at 40 ft. H_2O.

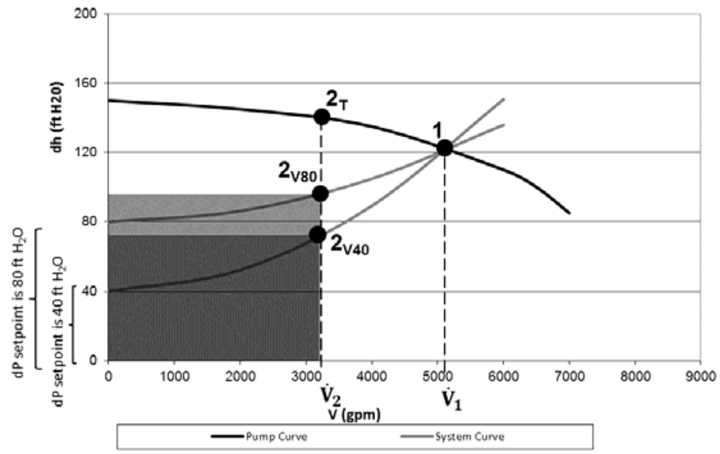

Figure 10. Additional energy associated with a higher dP setpoint is represented by the difference in the rectangles.

The setpoint pressure must be large enough to push fluid through the end uses and depends on the location of the sensor in relation to the pump and end uses. The influence of sensor location on pressure setpoint is demonstrated by considering four common pumping systems.

Figure 11 shows a closed-loop chilled water system with a sensor located at the pump outlet measuring differential pressure between the supply and return headers. At this location, the pressure setpoint has to be high enough to push fluid from point 2 through the supply header, through the most remote end use, and back to point 3 through the return header. If the differential pressure sensor were located near the most remote end use, the pressure setpoint would only have to be high enough to push fluid through the most remote end use. Thus, controlling the VFD based on the differential pressure near the most remote end use results in a lower setpoint pressure and reduced energy use. Locating a differential pressure sensor at the most remote location is recommended by ASHRAE Standard 90.1-2010 (ANSI/ASHRAE Standard 90.1-2010).

Another way to consider pumping energy use is to characterize energy savings in terms of flow reductions and head reductions. In Figure 11, from reference points 1-to-2 and 3-to-1, energy savings would be realized from flow and head reductions. From reference points 2-to-3, energy savings would only be realized from flow reduction in end uses. Thus, the total pressure drop across the pump would decrease only slightly due to reduced flow through the chiller.

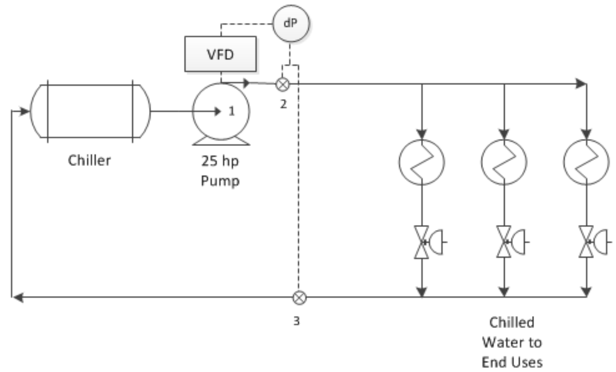

Figure 11. Closed-loop chilled water system with differential pressure sensor at the discharge of the pump.

Figure 12 shows a closed-loop chilled water system with a sensor measuring pressure at the discharge of the pump. At this location, the pressure setpoint has to be high enough to push fluid from point 2, through the end uses, through the chiller, and back to the pump at point 2. Thus, controlling the VFD based on pump discharge pressure results in an even higher setpoint than in Figure 11, and increased energy use.

As before, pumping energy use through Figure 12 can also be described in terms of flow reductions and head reductions. From reference points 1-to-2, energy savings would be realized from flow and head reductions. From reference points 2-to-1, energy savings would only be realized from flow reduction in end uses. Thus, the total pressure drop across the pump would remain nearly constant even as flow is reduced, and energy savings from pressure reduction are minimal.

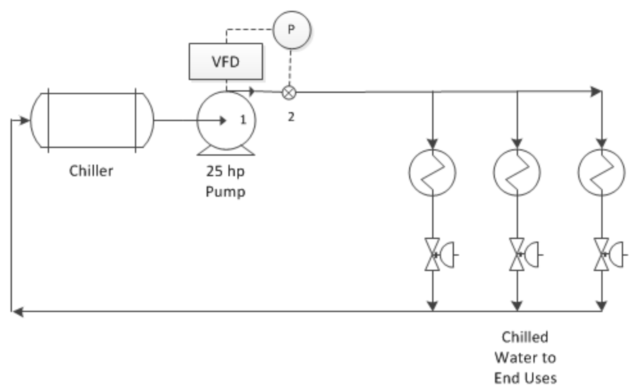

Figure 12. Closed-loop chilled water system with single pressure sensor at the discharge of the pump.

Figure 13 shows an open-loop cooled-water system with a sensor measuring pressure at the discharge of the pump. At this location, the pressure setpoint has to be high enough to push fluid from point 2 through the end uses and into the open tank. Moreover, additional pressure head is required to lift the fluid from 3-to-2. Thus, controlling the VFD based on pump discharge pressure results in a high pressure setpoint.

As before, from reference points 1-to-2 and 3-to-1, energy savings would be realized from flow and head reductions. From reference points 2-to-3 and to lift the fluid through the elevation gain, energy savings would only be realized from flow reduction.

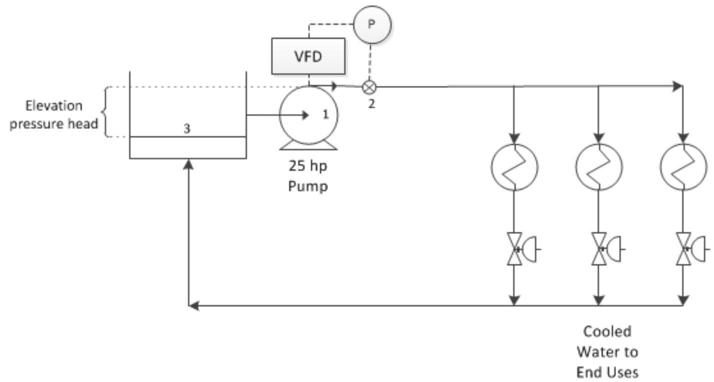

Figure 13. Open-loop cooled-water system with single pressure at the discharge of the pump.

Figure 14 shows an open-loop cooled-water system with a sensor measuring pressure at the discharge of the pump. At this location, the pressure setpoint has to be high enough to push fluid from point 2 through the entire supply header and through the most remote end use. If the pressure sensor were located near the most remote end use, the pressure setpoint would only have to be high enough to push fluid through the most remote end use. Thus, controlling the VFD based on the pressure near the most remote end use results in a lower setpoint pressure and reduced energy use.

As before, from reference points 1-to-2 and 4-to-1, energy savings would be realized from flow and head reductions. From reference points 2-to-3 and to lift the fluid through the elevation gain, energy savings would only be realized from flow reduction.

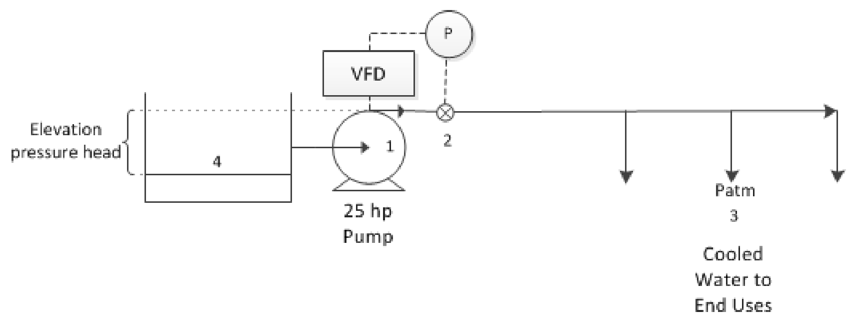

Figure 14. Open-loop system with single pressure sensor at the discharge of the pump.

Resetting Pressure Setpoint Based on Valve Position

As demonstrated above, a high pressure setpoint reduces energy savings when reducing flow. In fixed pressure setpoint control, the pressure setpoint must be high enough to maintain flow at peak conditions. However, many pumping systems seldom require peak flow. In these cases, the pressure setpoint can be dynamically reduced to save pumping energy. In valve position control, the pressure setpoint, and hence VFD speed, decreases until at least one valve is almost fully open. Thus, valve position control supplies the required flow, at minimum pressure, without starving any end uses. The effect is to approach zero head at zero flow and approach the ideal pumping system shown in Figure 1.

Valve position control is especially effective in parallel-flow piping systems where pressure based control may starve some end uses. For example, if the parallel-flow piping system shown in Figure 15 used fixed setpoint pressure control, locating the differential pressure sensor in circuit A may starve end uses in the circuit B if circuit B requires more flow and pressure than circuit A. Valve position control is less effective on systems with multiple end uses of different sizes. Moreover, valve position control does not set the lower limit on VFD speed; this is done elsewhere in the control algorithm.

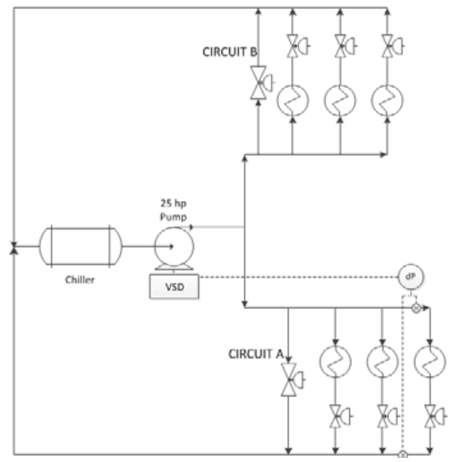

Figure 15. Parallel-flow pumping system.

CASE STUDY

The following case study demonstrates energy saving potential from minimizing bypass flow and pressure control losses. The variable-flow HVAC chilled water system shown in Figure 16 employs best practice by locating a differential pressure sensor near the most remote end use. In addition, the pressure sensors across the chiller evaporators can be used to measure flow through the evaporators to ensure minimum flow. The use of reverse return balances unregulated flow through the air handlers (Taylor 2002). The use of a three-way bypass valve at AHU 1 is not a recommended practice because some flow is always being pumped through the bypass.

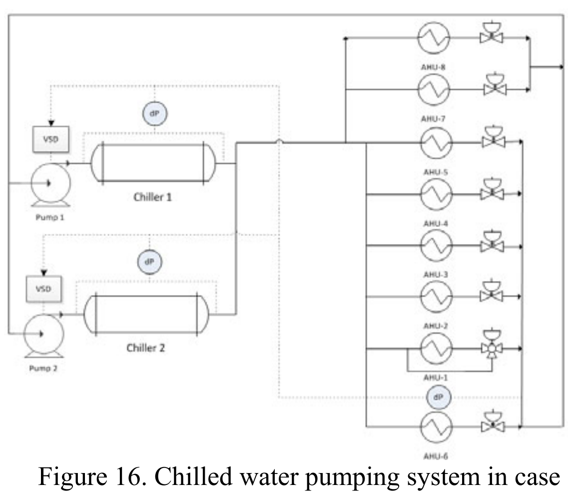

Figure 16. Chilled water pumping system in case study.

Despite the existence of VFDs and well-located pressure sensors, data from the control system indicate non-optimal control. Figure 17 shows measured five-minute VFD speed versus building differential pressure (dP) data from April 2014 to November 2014 when only Pump 1 was operating. Figure 18 shows measured five-minute VFD speed versus chiller evaporator dP data from April 2014 to November 2014 when only Pump 1 was operating. In a properly controlled system, the VFD should vary speed to maintain a constant building dP. However, the data show that the building dP varies between 1 psi and 8 psi. In addition, in a properly controlled system, the VFD speed and building dP should be highly correlated. However, statistical analysis shows an R^2 value of 0.02 between VFD speed and building dP. In fact, Pump 1 operates at 70% for 55% of the time. According to the chiller specification sheets, the chiller evaporator dP must be between 1.4 psi and 6.4 psi. However, it maintains an average chiller evaporator dP of 6 psi and often exceeds the limit of 6.4 psi. All this indicates very poor control.

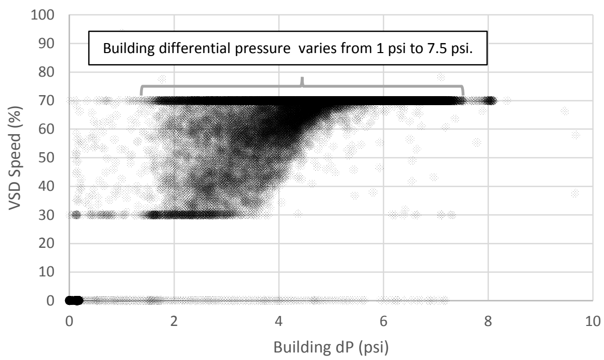

Figure 17. Building dP vs. VFD Speed.

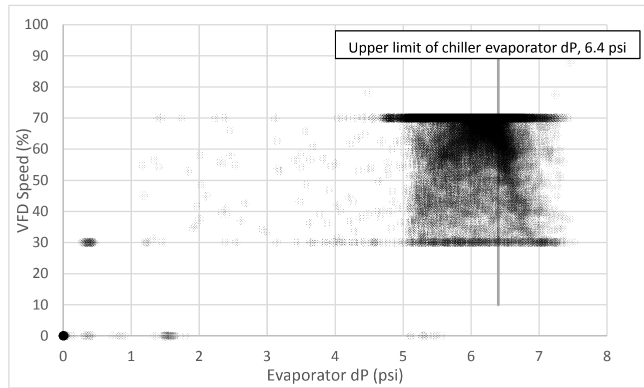

Figure 18. Chiller Evaporator dP vs. VFD Speed.

Time series data offers more insight into the control problems. As seen in Figure 19, the VFD operated at an average speed of 70% on Saturday, May 10, when the average outdoor air temperature was 65.5 ˚F, the average relative humidity was 70% and occupancy and internal loads were small. In Figure 20, the VFD operated at an average speed of 64% on Thursday, July 10, when the average outdoor air temperature was 70.7 ˚F[1] and the average relative humidity was 66%[2] and occupancy and internal loads were high. If properly controlled, the VFD speed would be higher on a hotter, wetter day with higher occupancy and internal loads. However, the opposite is true. The VFD speed is stuck at 70% on May 10 and drops to 30% on July 10. This indicates inefficient control.

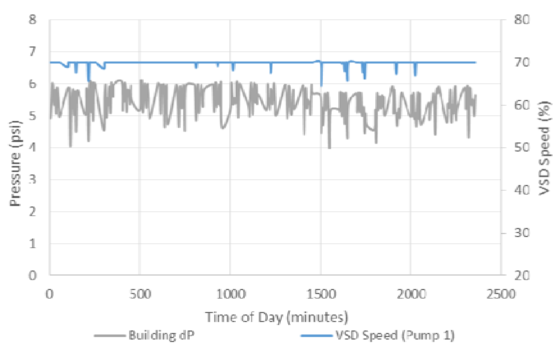

Figure 19. Trend data on Saturday, May 10, 2014.

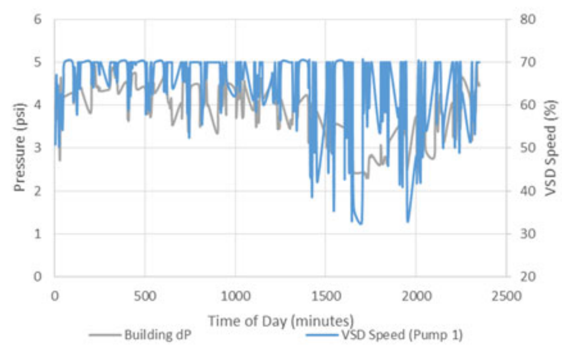

Figure 20. Trend data on Thursday, July 10, 2014.

[1] Average daily temperatures were taken from Kissock's Average Daily Temperature Archive: http://academic.udayton.edu/kissock/http/Weather/

[2] Relative humidity taken from Weather Underground: http://www.wunderground.com

The original control algorithm executes the following steps:

1. Chiller 1 and Pump 1 turn on when the outdoor air temperature is above 65 ˚F. For outdoor air temperatures between 55 ˚F and 65 ˚F, economizers on the air handling units (AHU) are able to meet the cooling load.
2. Differential pressure is read across the building and Chiller 1 evaporator:
   - The chiller evaporator dP must be maintained between 1.5 psi and 6.4 psi. If it is higher or lower, the VFD ramps up or down to maintain the upper and lower bounds.
   - The building dP setpoint is 10 psi. The VFD simultaneously ramps up or down to maintain 10 psi across the building.
3. If the building dP drops below 2.3 psi for 30 minutes, pump 2 turns on to increase building pressure.
4. If the chilled water supply temperature is greater than 48 ˚F, Chiller 2 comes on to provide more cooling capacity to the building.

Because of the 6.4 psi upper bound on the chiller evaporator dP, the VFD is always hunting for, but never reaches, the building dP setpoint of 10 psi. To improve energy-efficiency, the building dP setpoint should be reset to 2.3 psi. According to maintenance personnel this is sufficient to supply all air handlers with adequate flow. In addition, it guarantees a minimum of 1.4 psi pressure drop, and minimum flow, through the evaporator. This change will stop the VFD from hunting for a set-point it can never attain. Instead, the VFD speed will vary with the thermal building load; resulting in better controlled and more energy-efficient pumping. Using the pump affinity law with an exponent of 2.5, the estimated annual savings from implementing the reduced building dP setpoint are 47% of chilled water pumping energy use, or about 9,600 kWh per year and about $960 per year. Implementing the reduced building dP setpoint will require minimal cost and maintenance personnel time.

After this change is successfully implemented, we will recommend replacing the 3-way bypass valve with an actuated 2-way bypass valve controlled to provide a minimum pressure drop, and hence flow, across the chiller evaporator. Next, we will recommend resetting the building dP setpoint based on valve position. We will begin implementing these changes after sufficient baseline data is collected so energy savings can be measured.

CONCLUSIONS

This paper describes several best practices for controlling variable-flow pumping systems. As variable-flow pumping systems become more common, adhering to these best practices are important in order to maximize energy savings. In our experience, many variable-flow pumping systems are not optimally controlled, so the potential for savings is great.

To maximize energy savings in variable-flow pumping systems, bypass flow should be minimized or eliminated. Best practice to eliminate excess bypass flow is to employ an actuated two-way bypass valve or primary-secondary pumping. If the effect of excess bypass is neglected, savings calculations will significantly overestimate savings.

Best practice in controlling VFD speed is locating a differential pressure sensor near the most remote end use in the system. To increase energy savings, the dP setpoint should be as low as possible. Further, valve position can be used to reset the pressure setpoint downward to allow static head to approach zero and maximize energy savings.

ACKNOWLEDGMENTS

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