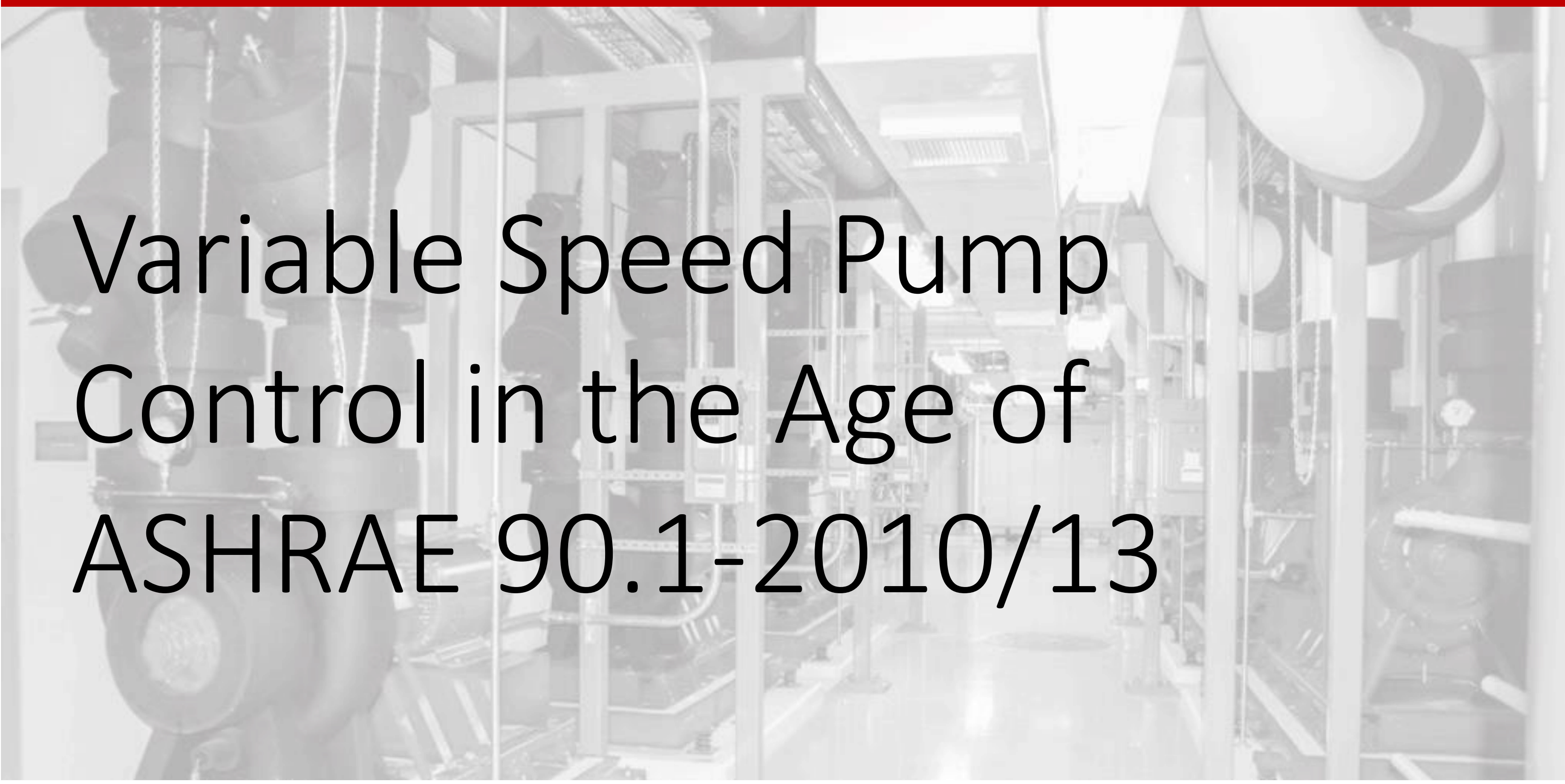
WHITE PAPER



Variable Speed Pump Control in the Age of ASHRAE 90.1-2010/13

By JMP Equipment Company

WHITE PAPER

ASHRAE 90.1-2010 and 2013 significantly ramp up efficiency requirements on chilled water pumping systems, both in terms of efficiency and in the application of variable speed pump control. This white paper will explain how these new efficiency standards impact chilled water design and drill down to solutions that will keep engineers within compliance.

ASHRAE 90.1-2010/2013 make variable speed pumping a requirement in most commercial chilled water systems. (See Addendum 1-3). The standard makes three main assertions about chilled water pumping systems:

(1) Any variable chilled water system (hot water does not apply) that utilizes motors of greater than 5 horsepower must have a control that limits motor demand to no more than 30% of design wattage at 50% flow.

(2) Pump controls can operate as a function of either the desired flow or the minimum required differential pressure at the heat exchanger requiring the greatest differential pressure, e.g. the most remote coil.

(3) If differential pressure control is used and the building has a DDC operated BMS (building management system), then the controller must continually change or "reset" the differential setpoint downward until one 2-way valve is nearly wide open.

How do we meet these requirements? It begins with a fundamental understanding of the two basic types of variable speed control, *curve control* and *area control*.

Area Control Versus Curve Control

A good analogy for curve control and area control is a first generation GPS that simply gets you from point A to point B versus one that can sense traffic conditions and provide alternate routes to keep you on the fastest route.

A strategy based on the control curve of a system is like a standard GPS; it doesn't provide alternate routes. As such, the pump is controlled to operate only at points on the system's control curve which represents the *theoretical* head loss in the system from zero to maximum pump flow. Wherever the selected pump curve intersects the control curve is where the pump will operate. These theoretical values are part of an algorithm that are programmed into the variable speed drive and will determine how fast the pump will operate under any given set of circumstances.

Area control is more dynamic. It can adjust pump speed based on *actual* system load in real time. Pump operation isn't tied to a predetermined control curve, but a multitude of curves that reflect actual load conditions, i.e. changing differential pressures as loads fluctuate. Thus, the pump may operate at many points above and below the control curve. This approach allows the

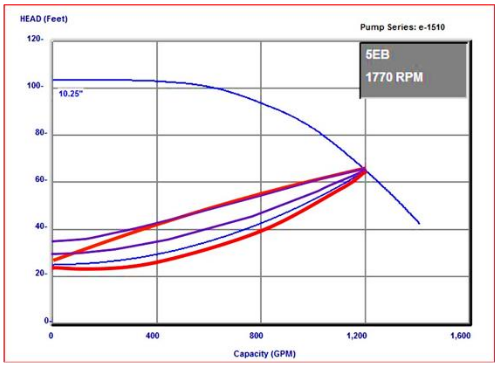


Figure 1

pump to adjust to fluctuating load conditions throughout the day.

The dark purple line in Figure 1 represents all the points where a pump controlled using curve control can operate. The area contained within the red lines represents all the points that a pump controlled using area control can operate.

In either case, it is the *Pump Affinity Laws* that are the basis for all variable speed pump control. These laws define the direct mathematical relationships between flow (GPM), pump speed (RPM/change in impeller diameter), head and brake horsepower (BHP). If you know any two of these values you can always determine the third. (See Addendum 4).

Where curve control and area control diverge is in the ability to react efficiently and effectively to *non*-theoretical circumstances. In the real world, no system operates on the projected control curve all the time. There are *misses*. Area control isn't confined to the points on the control curve, but rather an area contained within two curves so fewer misses occur.

Keep in mind that there are constant head losses and variable head losses.

Constant head loss or "control head" is the minimum amount of head required at all times to establish flow through the critical coil(s). That's why in Figure 1 the control curve starts 30 feet instead of 0 feet.

Variable head is the piping head loss throughout the system, which, because of the Pump Affinity Laws, varies with system flow. Variable head is calculated by subtracting control head from design pump head. The higher the variable head is compared to the control head, the more potential there is for saving pump energy.

Differential Pressure and Demand

To understand what happens in a closed-loop, variable speed system when demand drops, consider the system in Figure 2.

Admittedly, most systems have more than one coil and pump, but this simple example is perfect for illustrating the role of differential pressure sensing in variable speed pump control.

Notice we have a design flow of 1000 GPM and 80 feet (40' + 40') of variable piping head loss, plus 20 feet of control head. This is the minimum pump head we will have at all times, even at zero flow.

A pressure sensor located on the supply side to the coil and and immediately after the two-way control valve on the other side to sense the *differential pressure* across the coil. The differential pressure sensor is wired to the variable speed pump control. These components are necessary to make sure the control head setpoint -- which in this case is 20 feet -- is always maintained.

At full design conditions, we have 1000 GPM at 100 feet of head (40'+40'+20').

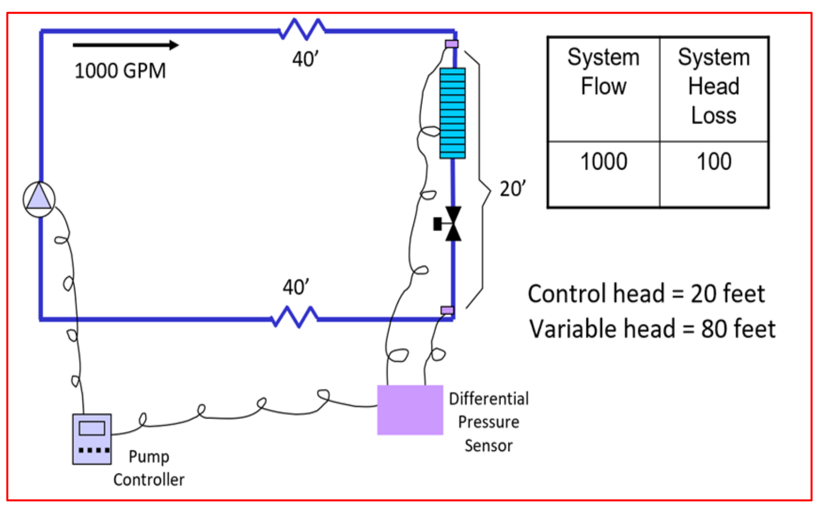


Figure 2

In this system the operational sequence would be as follows:

1. As demand drops, the control valve starts to close.
2. The closure of the control valve causes system flow to drop and an increase in differential pressure (DP) across the coil.
3. The DP sensor recognizes the increase in differential pressure and prompts the pump controller to slow the pump down.
4. The pump slows down until the control differential pressure (20 feet in this example) is restored.

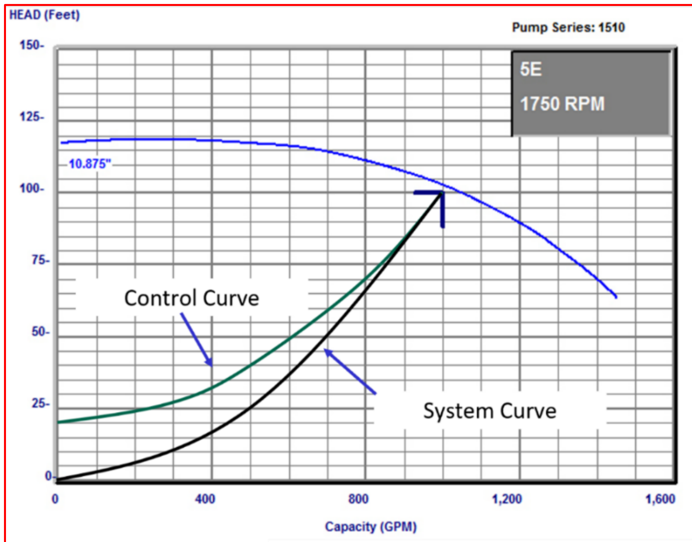
The rate at which the pump slows is determined by the Pump Affinity Laws. Specifically, if flow is cut in half, then the variable head loss is reduced by a factor of 4. Using the design coordinates of 1000 GPM and 100 feet of head as the starting point, every other operational point for the pump can be calculated, giving us the control curve. So, if flow drops from 1000 GPM to 500 GPM, then our system head loss drops from 100 feet to 40 feet.

If our piping loss on either side of the coil was 40 feet at 1000 GPM, then at 500 GPM, our head would drop to one quarter of 40 feet, or 10 feet of head on either side of the coil.

That is our variable head loss. Remember, we must maintain a pressure differential of 20 feet across the coil and control valve. 10 + 20 + 10 equals 40 feet of head. At half flow we have a system head loss of 40 feet.

Using the same formula, we can calculate the rest of the coordinates, which we can use to plot the control curve that is shown in Figure 3.

Variable speed control gets a bit more complicated in real-world, multizone systems that have valves opening and closing throughout the day. Figure 4



System Flow	System Head Loss
1000	100
800	71
500	40
300	27
0	20

Figure 3

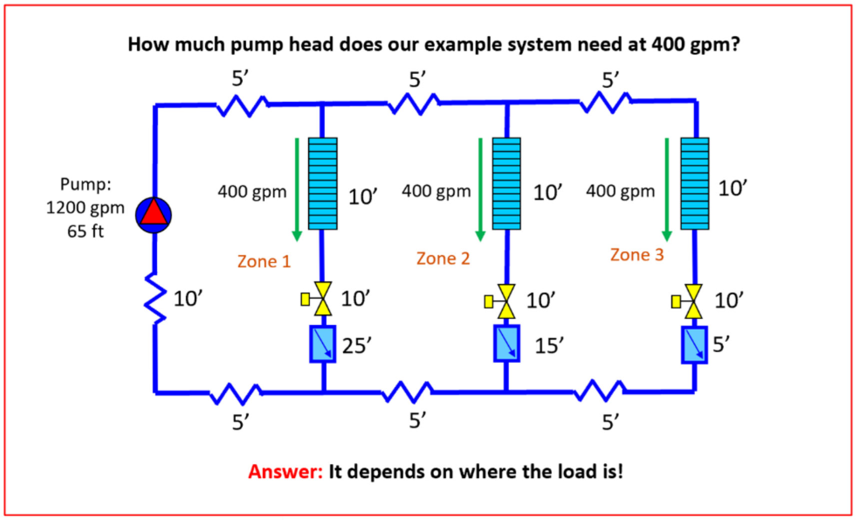


Figure 4

shows a fully loaded three-zone system with flow limiters at each zone.

Balancing is easy enough when all zones are at maximum 400 GPM design flow, but what happens when demand drops? Realistically, we know valves within a multi-zone system are not opening and closing in tandem. Rather, load varies from one zone to the next, depending on space occupation and the current solar load of individual zones.

Think of students in a high school migrating to the cafeteria between 11AM and 1PM. Clearly the zone serving the cafeteria is going to have a higher demand at that time of day than the classrooms, many of which may be unoccupied. We call this “system diversity.”

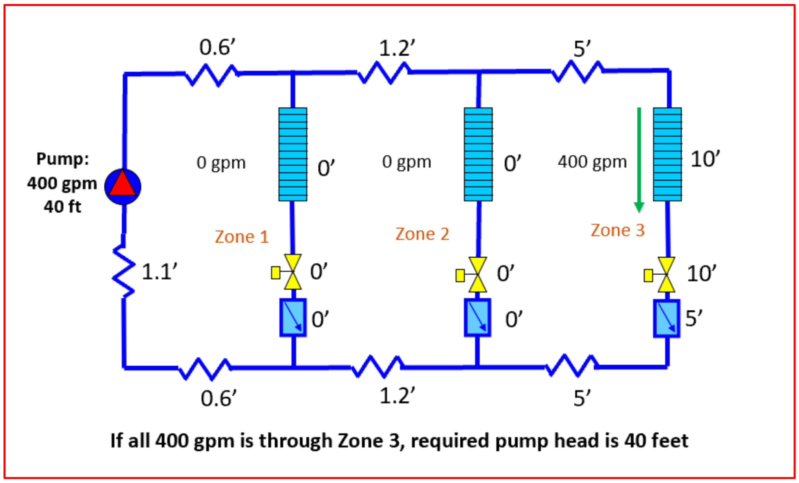


Figure 5

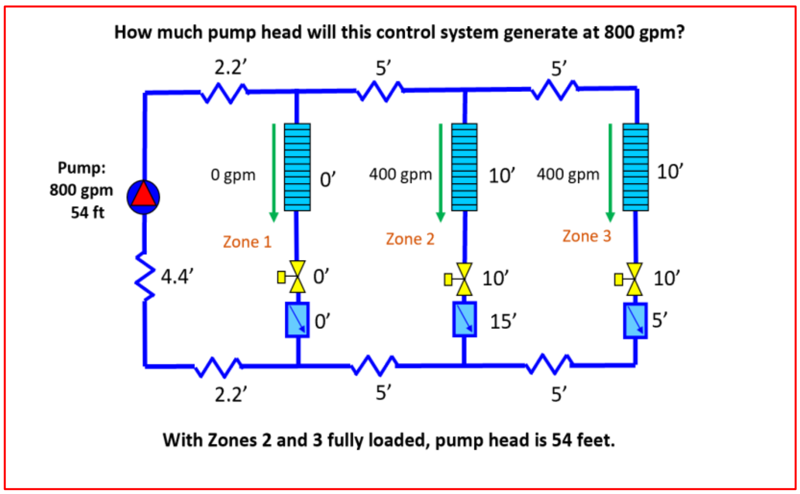


Figure 6

So what happens in our example system when load drops from 1200 GPM to 400 GPM? How much pump head do we need? It depends on *where* the load is. If, for example, all 400 GPM is through Zone 3, then the required pump head would be 40 feet -- technically, 39.7 if you do the math (Figure 5).

Now let's switch it up even more. What happens if the demand is 800 GPM?

Again, it depends on where the load is. Let's assume that Zones 2 and 3 are fully loaded. If that's the case, we will require a pump head of 54 feet (Figure 6).

Interestingly, if we plotted a control curve for this system, we would notice that these two reduced operational points (800 GPM @54 ft) and (400 GPM @40 ft.) do not fall on our control curve. They are, in fact, above it.

These are just two points for potential "misses". If we were to plot all of the possible operating points in this same system, it would encompass an area like what is shown in Figure 7.

The space enclosed by the red curve lines represents all the possible heads and flow rates a variable speed pump must deliver in order to satisfy the part load demands of this system. On a direct return system, the upper red curve represents the control valves that are located farthest from the pump; the lower red curve represents the control valves closest to the pump.

We prevent these misses by targeting our control strategy on the area contained *within* the red lines.

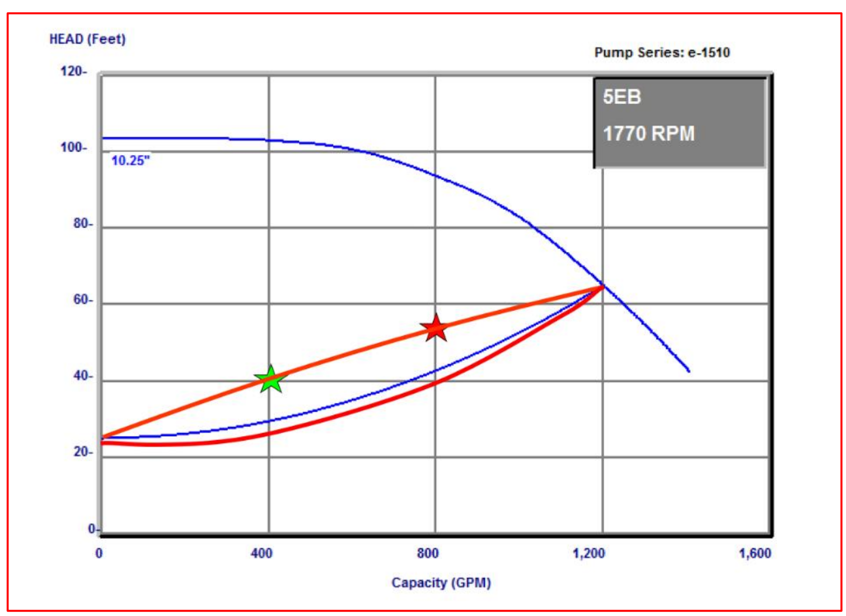


Figure 7

How System Diversity Impacts Control Strategy

The diversity characteristics of a given system should be considered when choosing a control strategy for variable speed pumps in a hydronic heating or cooling system. The amount of diversity and the time and location of loads will dictate whether curve control or area control is the best strategy.

System diversity refers to the fluctuating loads that occur in most any commercial or institutional building or facility throughout a 24-hour period. Loads naturally vary for all of the following reasons:

- The movement of people from one space to another throughout the day. Obviously, the people in a building or facility can't be everywhere at once. For example, students migrate from classrooms, to the cafeteria, to the dorms, etc. throughout the day.
- The east, south, west, and north exposures of a building heat up at different times throughout the day. This impacts the "solar" load on a building's cooling system.
- To a lesser degree, the opening and closing of doors also has some impact on the heat loss and heat gain of the building.

All of this creates diversity in the system, which means that peak loads do not (*cannot*) occur in every part of a facility at once. A simple way to see if a system has diversity is to add the total flow of all of the cooling coils and compare that value to your chiller plant design flow. In the above example the total connected load is 4000 tons, however the engineer has determined that the instantaneous maximum load at any given time is only 3000 (Figure 8). This gives the building a diversity factor of .75 or 3000 ÷ 4000.

If we know our peak block load, we can minimize chiller capacity and "move" cooling supply from one place to another

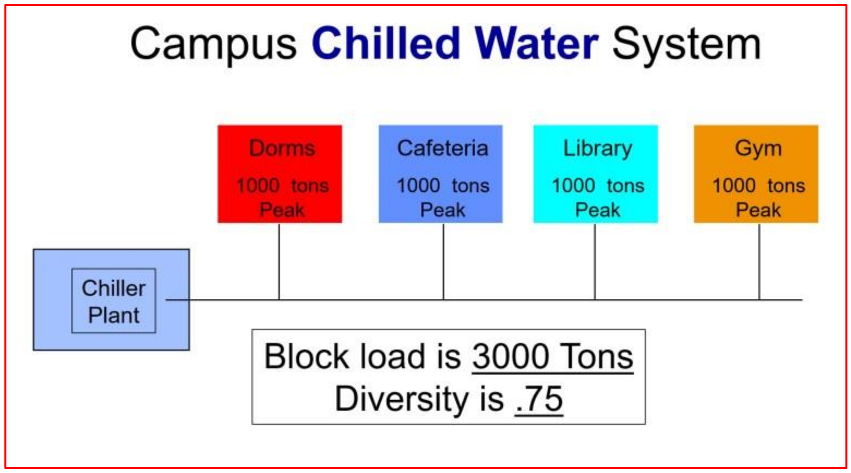


Figure 8

as needed. Variable speed pumping helps increase energy efficiency in these diverse systems; but there is a limit to how much we can slow the pump down while keeping the occupants comfortable.

Comfort and efficiency are the ultimate goals of variable speed pump control. Generally speaking, systems with a lot of diversity will experience more "misses" when pumps are controlled based on the control curve. When misses occur, comfort suffers and plant operators tend to increase the control head value on the pumps to keep occupants happy. Ultimately this solution costs the owner more money because it increases the horsepower of the pumps indefinitely.

How much a miss will impact comfort depends on where and when the loads occur. Some spaces may be able to tolerate the misses at certain times during the day. Other spaces may not.

Reset and Setback Considerations

Nighttime setback and chilled water reset can also cause problems and should be taken into consideration when deciding how to make the most of system diversity.

Think about a high school with nighttime setback. What happens when the system starts up in the morning and all zones call for heating or cooling at once? In our above example, this would mean that the system would suddenly require 4000 tons of cooling when the chiller plant can only provide 3000 tons. In this scenario, the coils closest to the chiller plant would get all the flow, while the more distant coils would be starved of flow.

A similar problem could occur in systems with chilled water reset. When the chilled water supply temperature is temporarily reset to a higher temperature during certain hours of the day, it can cause the system to flow more water than it was designed to flow.

Flow Tolerance

Neither of these are untenable problems; systems can be designed to allocate flow to various zones in a staged or sequential manner so that comfort is maintained. But one must keep in mind that nighttime setback and chilled water reset do impact variable speed control strategy, especially in systems that rely on non-dynamic curve control.

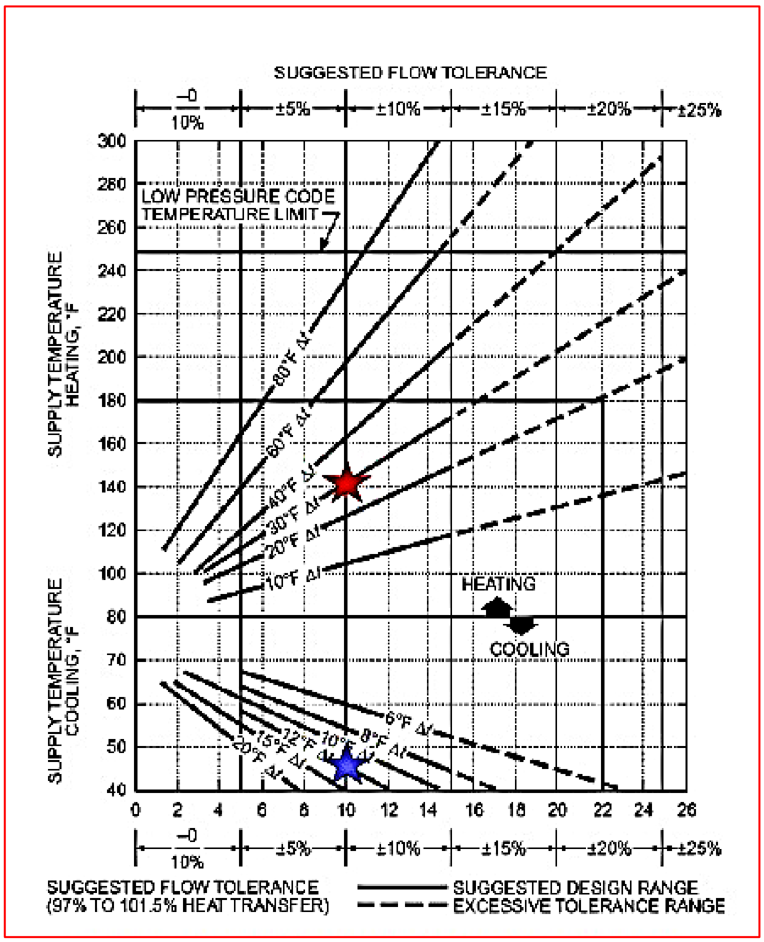


Figure 9 - FOR COIL FLOW TOLERANCE TO MAINTAIN 97% DESIGN HEAT TRANSFER (CARLSON 1981, 2016 ASHRAE HANDBOOK -- SYSTEMS AND EQUIPMENT)

Flow tolerance determines how much we can vary flow before comfort is negatively impacted. This is the amount flow reduction the system can tolerate and still maintain 97% of design heat transfer at design flow. ASHRAE suggests a minimum of 97% heat transfer to maintain a properly balanced system. Typical chilled and hot water design conditions require 10% flow tolerance to meet ASHRAE's definition for a "balanced system." In other words, you can only vary flow through a coil so much before you impair its ability to achieve sufficient heat transfer, leading to a miss and ultimately discomfort. A miss of greater than 10% is not acceptable.

Heating systems have a broader operating range than cooling systems when it comes to maintaining sufficient heat transfer. Let's look at the following graph adapted from the ASHRAE Handbook to get a better feel for the comparative flow tolerances of heating and cooling

This graph (Figure 9), originally developed by Gil Carlson of Bell & Gossett, has been adapted by ASHRAE and can be used to determine the coil flow tolerance for any given supply temperature at a given ΔT. The upper portion of the chart is for heating systems and the lower portion is for cooling.

Let's first look at heating. If we have a heating system with a supply temperature 140°F and a 30°F ΔT, we can use this chart to determine that under these conditions we have a flow tolerance of 10% -- meaning that as long as we maintain 90% of design flow, we will stay within ASHRAE's requirement for 97% heat transfer.

The same is true of a cooling system where the supply temperature is 45°F and the ΔT is 12°F. If we read across from 45°F on the lower portion of the supply side axis to where it intersects with the 12° F ΔT, we read down vertically to find that we have a recommended flow tolerance of 10%, meaning that we need to maintain 90% of design flow to maintain 97% of ASHRAE's required heat transfer.

Remember -- chilled water systems are less forgiving in terms of flow variance than heating systems. This is because with a cooling system you are have both sensible *and* latent (humidity) loads.

It's also important to note that ASHRAE says the system shall be designed to maintain the required flow tolerance. This includes part load. You must choose the right combination of variable speed pump control and balancing strategies to meet this requirement.

Balancing Variable Speed Systems

Good system balance is not only crucial to the efficiency of a variable speed system, it is also a requirement of ASHRAE 90.1, along with a written balance report which must be submitted to the owner or representative for the HVAC system. This requirement applies to any building with a conditioned space of greater than 5000 sq. ft., which includes most any non-residential building. (See Addendum 5).

ASHRAE also requires that we proportionally balance and thus be able to reduce the speed of the pump as much as we can and still meet design flow conditions. (See Addendum 6). In other words, we must trim the pump impeller or be able to reduce the speed of the pump so that we have just enough flow to deliver 100% of flow at the critical zone with the valve wide open.

Finally, we must provide the owner with the suggested setpoints for the system, including pressure in air duct, hot and chilled water temperature setpoints, Delta T's, and, most relevant to our current discussion, control head setpoint. (See Addendum 7). We may use any of the following components to balance, as long as the outcome is a verifiably balanced system:

- Calibrated balancing valves (i.e. circuit setters)
- Automatic system-powered flow control (flow limiting) valves
- Standard ball or butterfly valves with pressure gauges or test plugs for measuring pressure drop across the coil or heat exchanger so that appropriate flows can be determined based on the manufacturer's performance data
- Pressure-independent control valves
- Automatic control valves

Ball/butterfly valves and automatic control valves aren't really relevant options in today's sophisticated systems, so we will limit our upcoming exploration of control methods to the remaining three. Now let's look at each under specific scenarios to show how they can and can't be used to meet ASHRAE balancing standards and how each balance type changes the control area of the system.

Today's flow balancing devices for variable flow pumping systems fall into three main types:

1. Pressure Dependent Calibrated Valves
2. Pressure Independent Flow Limiting Valves
3. Pressure Independent Control Valves

(See Addendum 8 for advantages and disadvantages of various balancing devices.)

Pressure dependent calibrated valves, commonly referred to as circuit setters, are used for pre-set proportional system balancing. Circuit setters incorporate a ball valve and two pressure ports through which the entering and exiting pressures can be measured to determine the pressure drop across the valve. A calibrated plate makes it possible to balance and set flow. Circuit setters are field adjustable and may be preset prior to balancing. They are inexpensive and readily available, however getting them properly set up at system start-up requires expertise and can be time consuming, depending on the number of circuits.

Pressure independent flow limiting valves have cartridges on the inside that move back and forth in response to system pressure changes. This movement increases or decreases the size of the internal orifice. When the entering pressure is low, the spring-loaded cartridge inside the valve opens, and in doing so allows more flow to pass through the valve despite the low pressure. As entering pressure increases, the pressure acts on the spring, compressing it and reducing the orifice size so that flow is limited through the valve. In either case, flow is quickly stabilized despite system pressure fluctuations – as long as the pressures are within the operating range of the specific valve (Figure 10).

Flow limiting valves are shipped preset from the factory based on the maximum GPM, which saves balancing time at the jobsite. Proper sizing is extremely important for accurate control, because once the system gets out of a specific valve's operating differential pressure control range, the valve becomes a fixed orifice device and you're no longer limiting the flow.

Some flow limiters have dials that allow them to be adjusted for a specific flow once they are installed. This eliminates the time-consuming process of matching up valves to specific floors or equipment

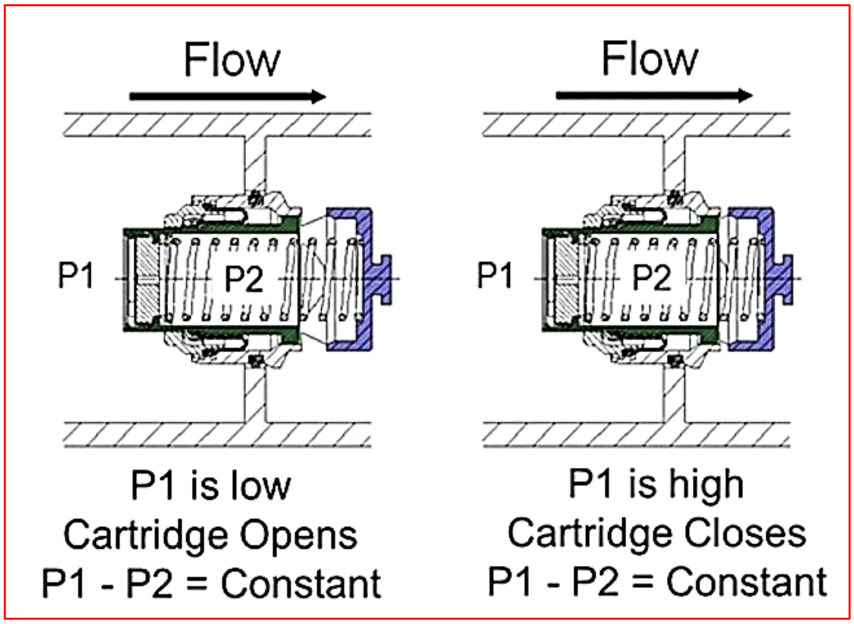


Figure 10

and provides an extra bit of flexibility when it comes to operating flow range.

Pressure independent control (PIC) valves combine the functionality of a balancing valve, control valve and a differential pressure regulator all into one valve body. We recommend using a valve that maintains its full stroke capability despite any preset maximum flow rate; therefore, the valve maintains full authority under all load conditions.

PIC valves incorporate a spring loaded differential pressure regulator, which constantly adjusts and compensates for fluctuations in system pressure. This internal element responds to pressure changes by moving up or down to maintain a constant flow despite these fluctuations. That is the key benefit of PIC valves - a change in differential pressure does not cause a change in flow!

PIC valves do not merely limit flow; they keep flow at a specific setting depending on the signal to the control valve. This eliminates underflows and overflows through the coil and ensures a much more consistent energy transfer. Figure 11 breaks out the functionality within the PIC, including the balancing, control, and differential pressure regulation.

To understand the operation of the PIC, let's assume we have installed the device on a coil sized for 900 GPM as shown. We have 34 PSIG entering the valve and 27 PSIG going out, meaning we have a 7 PSIG differential across the valve. The signal from the BMS is 8 volts to the valve actuator. We have 26 PSIG leaving the valve so the balance portion of the valve is absorbing 1 PSIG (Figure 11).

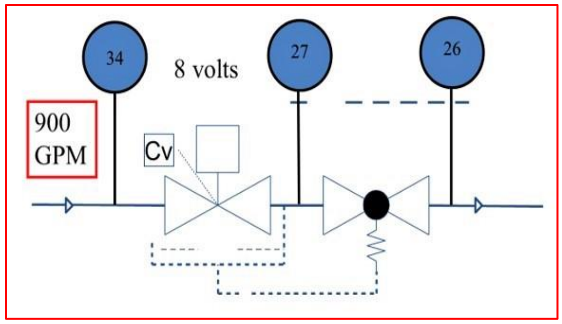


Figure 11

What happens if the demand drops in the zone next door, causing pressure to rise and subsequently increasing the entering pressure from 34 to 35 (Figure 12)? Keep in mind, the demand in the zone shown above has not changed; the control signal is still at 8 volts. However, the variation next door has created an increase in pressure differential across the control valve from 7 PSIG to 9 PSIG.

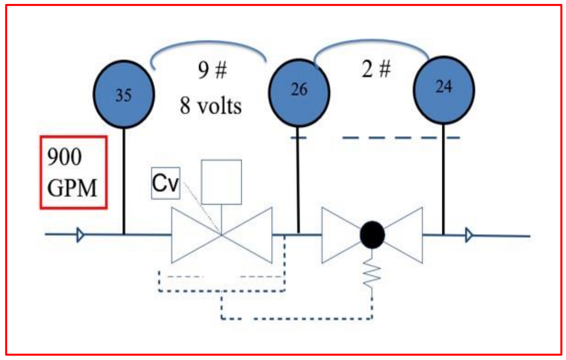


Figure 12

To keep things right within this zone and maintain proper flow, we need to reestablish a 7 PSIG differential across the control valve. The PIC responds to the current condition by throttling the balance portion of the valve to absorb the additional pressure – in this case increasing the differential from 1 PSIG to 4 PSIG, while also maintaining flow at 900 GPM (Figure 13).

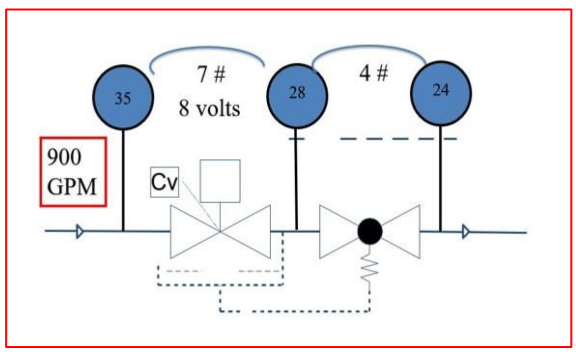


Figure 13

If this gets confusing, think of the PIC as coming complete with a tiny pipefitter who is constantly receiving data from the BMS, as well as pressure and flow information, and swinging into action by throttling the valve as needed.

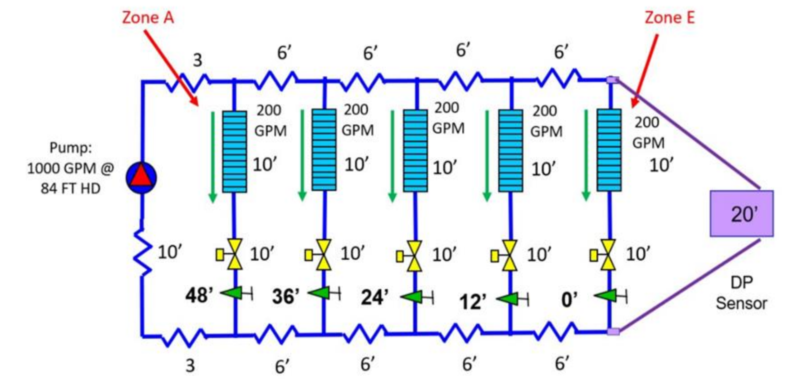
Only a change in load to a particular zone will result in a change in flow rate through the corresponding PIC valve. All this makes PIC valves the most accurate and efficient choice for variable speed pumping systems.

Manual Versus Automatic Balancing

How much pump head do you need in a manually balanced system at part load?

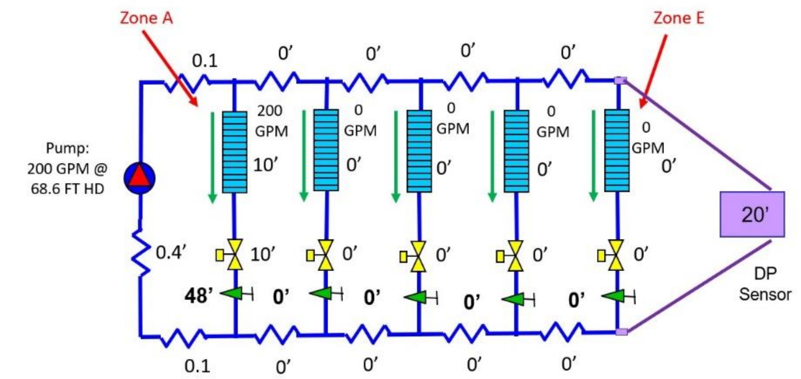
Again, it depends. Consider the proportionally and manually balanced system with a design flow of 1000 GPM and 84 feet of head shown in Figure 14. We are using differential pressure area pump control with a control head of 20 ft.

Balancing is easy enough at full load with 200 GPM going to all 5 zones. But what happens if system demand drops from 1000 GPM to 200 GPM? The question is unanswerable without knowing where the load is. It could all be in Zone A, B, C, D or E. Or it could be split between any two or more zones. In a typical HVAC system where the zone loads are never the same, it's anybody's guess where the load is. That's the challenge since head loss varies depending on the *location* of the load and the pressure losses through the branch pipe.



System is at 100% design flow, Manual Balance

Figure 14



System is at 200 GPM Thru Zone A, Manual Balance

Figure 15

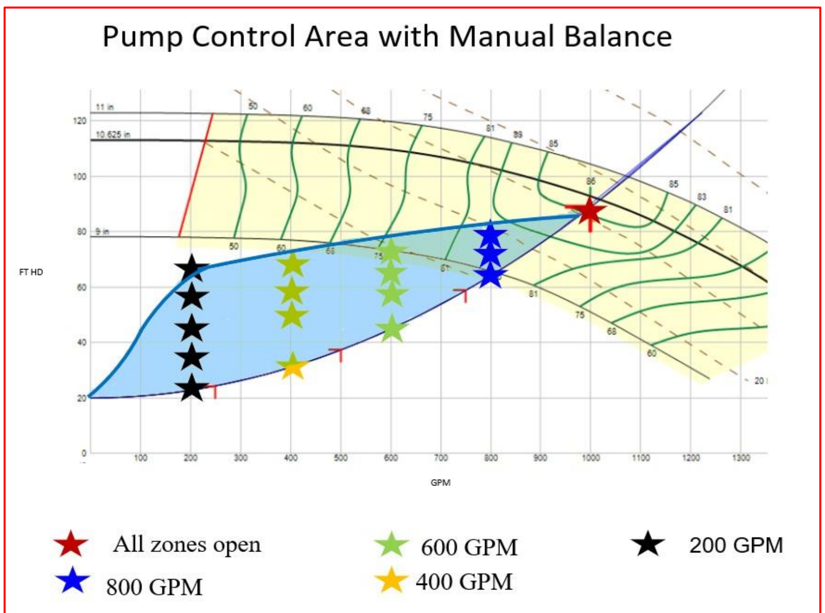


Figure 16

To better illustrate this dilemma and keep it simple, let's pretend that the 200 GPM load is all in Zone A. If that's the case our pressure drop adds up to approximately 69 feet. That's how much pump head we need to establish full flow through Zone A. Depending on the type of pump control we have, we will most likely experience a flow miss in the system if we are using manual balance valves (Figure 15).

The black stars on the curve in Figure 16 represent the head requirements for the system at 200 GPM for the multiple zones. If we repeat this for other flow points, we can plot the pump control area for the system.

So, pressure drop varies quite a bit depending on where the flow happens to be. For example, at 200 GPM, the actual pressure drop could exist anywhere along the vertical line of black stars. This represents a large pump control area which increases the potential for flow misses.

Now let's look at the same system utilizing automatic balance valves or pressure independent control valves. If this is the case, then our pressure drop adds up to only approximately 26 feet to give us full flow thru Zone A (Figure 17).

If we calculate the control area for the same system except with automatic balance valves, our control head setpoint is now 25 FT, taking into account the pressure drop needed for the automatic balancing valve (Figure 18).

Because automatic balancing valves can unload pressure drop, we are able to decrease the amount of pump head needed to fully flow Zone A at part load when compared to a manually balanced system. The amount of this decrease depends on the pressure drop in the branch headers. Therefore, a manually balanced system typically has a larger control area than an automatically balanced system.

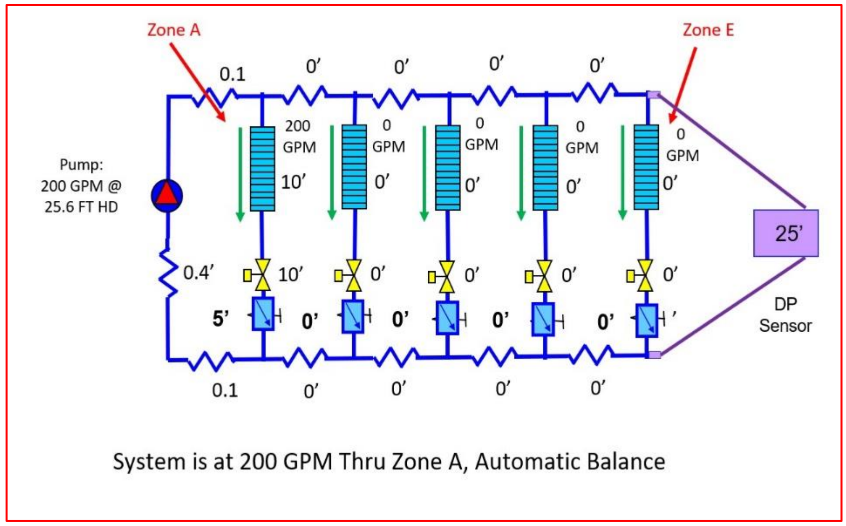


Figure 17

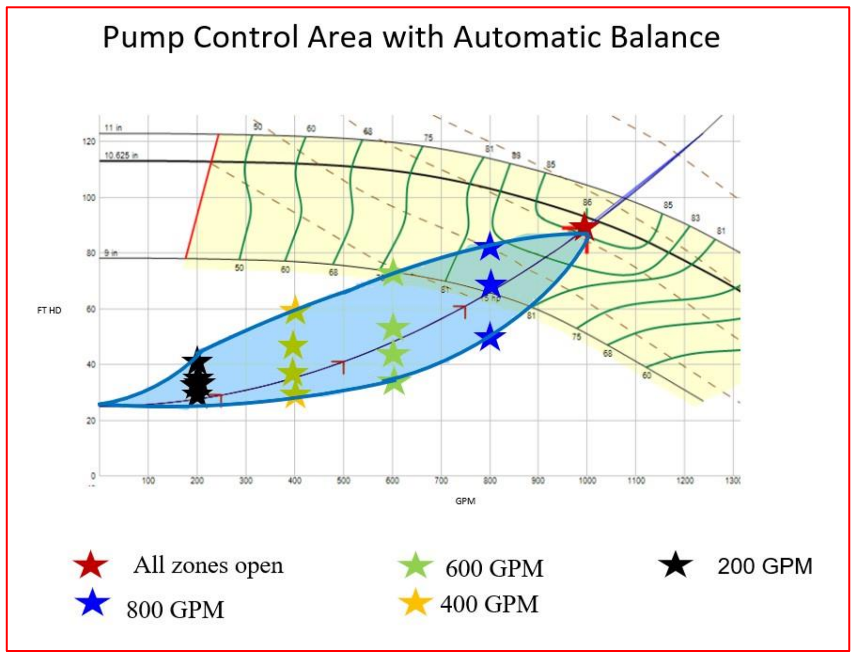


Figure 18

A Lesson in Curve Control

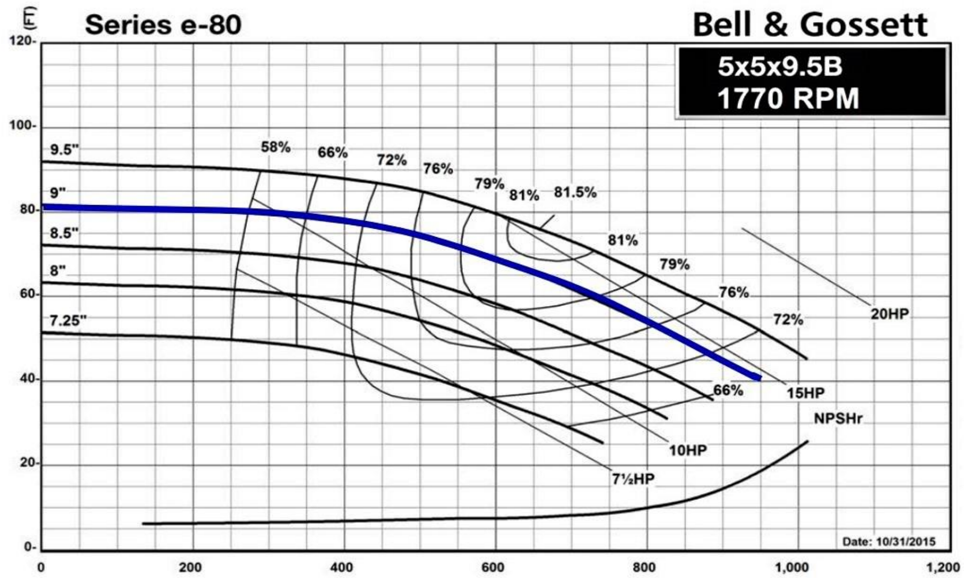
Many assume that curve control is always sensorless; this is not the case. In addition to sensorless control, there is also what can be thought of as "pump head curve control" and "full system flow curve control."

Any time curve control is used as a strategy to control variable speed pumps, the goal is to keep the pump operating on the building's control curve as established by the design engineer. For every point of flow on the control curve, there is a unique speed. So if you know one you can determine the other.

In sensorless curve control, the controller calculates the system flow based on the kW, RPM, and the impeller size of the pump. Using this estimation of flow, the pump controller adjusts the RPM of the pump so that it stays on the control curve. In essence, sensorless control uses the variable speed drive and the pump as a flow meter; but instead of physically measuring flow we are calculating a theoretical flow.

Not every pump is suited for sensorless control. Ideally, the pump curve will be shaped so that the horsepower line intersects with the pump curve at one easily defined point like the one in Figure 19.

Sensorless Flow Calculations



With 9” impeller, at 1750 RPM and 15 HP (both read from the VFD)

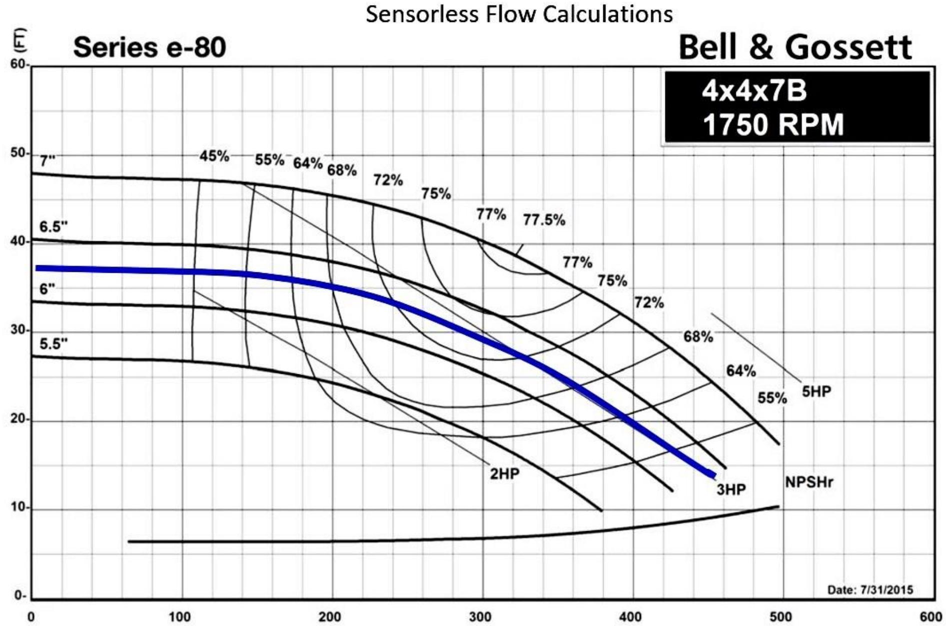
This pump IS suitable for sensorless

Figure 19

If, however, a pump has a curve that looks like the one below (Figure 20), it is not a good candidate for sensorless control. Notice that a pump with a 6.25 impeller at 3 horsepower follows approximately the same slope between 310 GPM and 450 GPM. Within this flow range, there is no way of knowing what the pump flow will be without a flow meter.

Sensorless control may also limit variable speed drive options. In sensorless

Sensorless Flow Calculations



With 6.25” impeller, at 1750 RPM and 3 HP (both read from the VFD)

This pump is NOT suitable for sensorless

Figure 20

applications the variable speed drive and its control are typically exclusive to the pump manufacturer, which can be a problem if there is a failure.

Curve Control *with* a Sensor

Pump head curve control compares the pump head pressure differential to the motor RPM to see where the pump is operating on the pump curve and then adjusts the speed to keep it on the control curve. This method requires a pressure sensor across the inlet and outlet of the pump to measure the pressure differential as shown in the diagram below (Figure 21).

Full system flow curve control works on the same principal as sensorless curve control except a flow meter is used to read real flow versus an estimated flow. The flow meter is typically located on the discharge of the pump. This, of course, provides more accurate pump control (Figure 22).

Once again if we know flow rate and speed we can slow down or speed up the pump as needed to keep it operating on the control curve.

The primary advantage of both full system flow and pump head curve control is that each is compatible with any drive manufacturer or any pump.

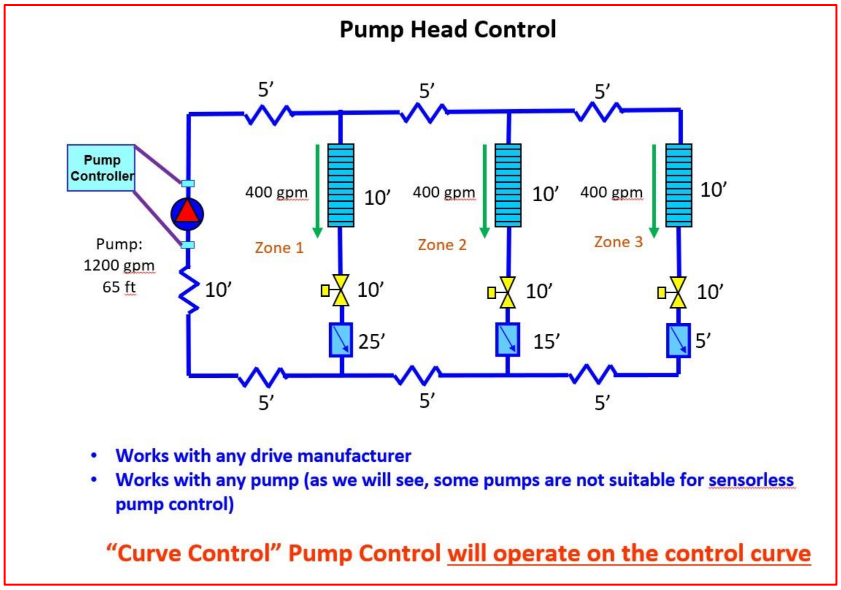


Figure 21

Finally, it is important to note that whenever a curve control strategy is used (sensorless or otherwise) most pump manufacturers will default the control head of the pump to a value that is 40% of the design head. For example, if the full design flow requires 100 feet of head, the control head (the required head a zero flow) would be 40 feet.

This is because most systems will not meet the following ASHRAE requirement with a control head setpoint greater than 40% of design head:

***6.5.4.1 Hydronic Variable Flow Systems**. HVAC pumping systems having a total pump system power exceeding 10 HP that include control valves designed to modulate or step open and close at a 50% or less of the design flow rate.*

Individual chilled water pumps serving variable flow systems having motors exceeding HP shall have controls and/or devices (such as variable speed control) that will result in pump

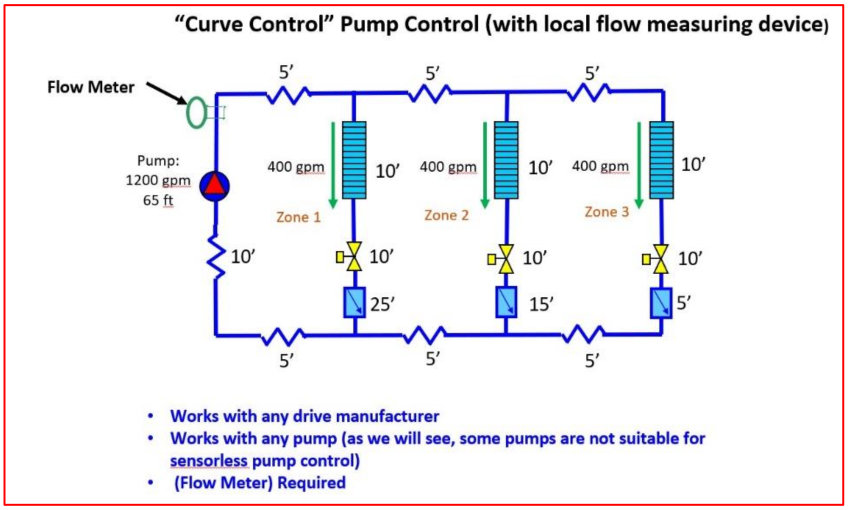


Figure 22

motor demand of no more than 30% of design wattage at 50% of design water flow.

To stay below 30% of design wattage at 50% of flow, the engineer will likely have to have the control head set at no greater than 40% of design head. Because manufacturers default to 40%, it is recommended that the control head needed for your system be programmed into the pump controller at the time of commissioning, otherwise the owner will be wasting energy indefinitely.

When Too Many Misses Occur

Frequent misses are fairly common in variable speed systems that utilize a curve control strategy. When misses occur, coils can become starved of flow. This results in unsatisfactory space temperatures and often uncomfortably high humidity levels, which leads to occupant complaints. The inevitable response is to increase the control head on the pumping system. Over time these adjustments cause the control head to creep up, sometimes exceeding efficiency parameters set by ASHRAE which require a control head of no greater than 40% of the total system head.

Consider the pumping system shown in Figure 23 which utilizes curve control and has only manual balancing. With only the zone closest to the pump open and calling for full flow (200 GPM), 68.7 feet of head is needed to overcome the resistance in that zone. However, if the pump control automatically defaults to 40% design head, only 36 feet of head will be available, giving us only 145 GPM of flow through the coil. Depending on the flow tolerance of the coil, this miss will likely result in complaints. The facility manager will likely respond by increasing the system control head, appeasing the occupants but significantly increasing operating cost. Furthermore, the system may no longer comply with ASHRAE 90.1.

Unfortunately, this has become an all too common problem due to the increasing number of curve controlled systems. The greater load diversity in the system, the more likely this is to be a problem.

A Lesson in Area Control

Systems with high load diversity are far more suited to area control than curve control.

Differential pressure (DP) sensors play a crucial role in an area control strategy by providing feedback to the pump controller to speed up or slow down the pump as needed to maintain the control head. With curve control, that head is planted squarely on the control curve. With area control, you've got more room to play. This comes in handy when you are working with systems that have a lot of diversity. In both cases, flow limiters or pressure independent control valves will help owners avoid misses under certain circumstances. These automatic balance devices help by unloading pressure drop at part load conditions.

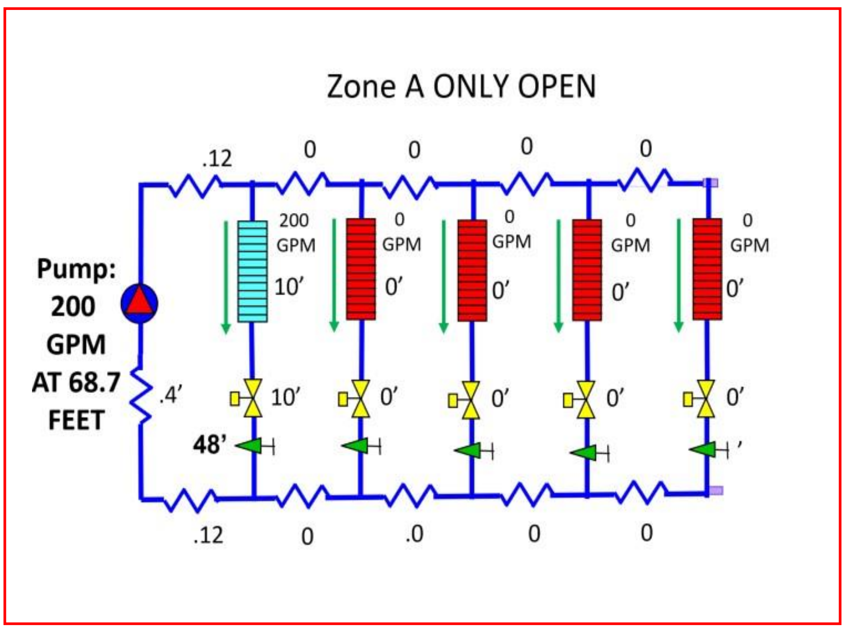


Figure 23

Notice that the system in Figure 24 utilizes automatic flow limiting balancing valves at each zone. If we are using a curve control pump control strategy without a differential pressure sensor at the most critical zone, they will do a fairly good job satisfying flow demand, but there are exceptions.

The worst-case scenario with this balance type occurs when all the load is in the most remote (critical) zone. So, what happens in our example system if we have full load in Zone E and no load in the other zones?

In that case, we will have a miss in Zone E (Figure 25).

Zone E is calling for 200 GPM, but we only get 162 GPM using a curve control strategy. To get the full 200 GPM through Zone E, we need a total of 42.8 feet of head. However, because we are using curve control, we only have 28 feet of head available to flow water through Zone E, which only gives us 162 GPM of flow. To stay within ASHRAE's requirement of 97% heat transfer, we have to maintain at least 90% of the design flow through that coil or 180 GPM.

We can solve this problem by adding a DP sensor to Zone E, which effectively changes our variable speed control strategy into area control strategy. With a DP providing feedback to the pump controller, we can make sure that our available head across the critical zone never drops below our control head, which in this example is 25 feet. This gives us 42.8 feet of pump head at this particular part load condition – enough to flow 200 GPM through Zone E (Figure 26).

The remote DP sensor operates within the control area instead of being bound to the control curve. Combined with automatic flow limiters or pressure independent control (PIC) valves, the approach provides better (and more efficient) load matching.

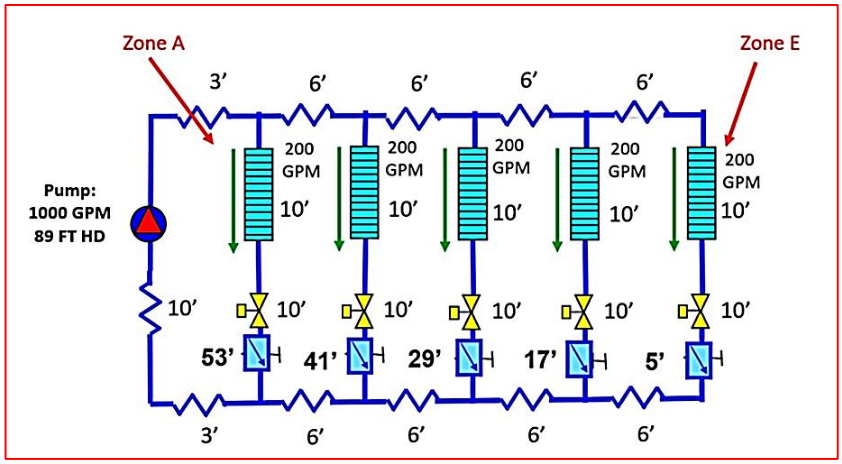
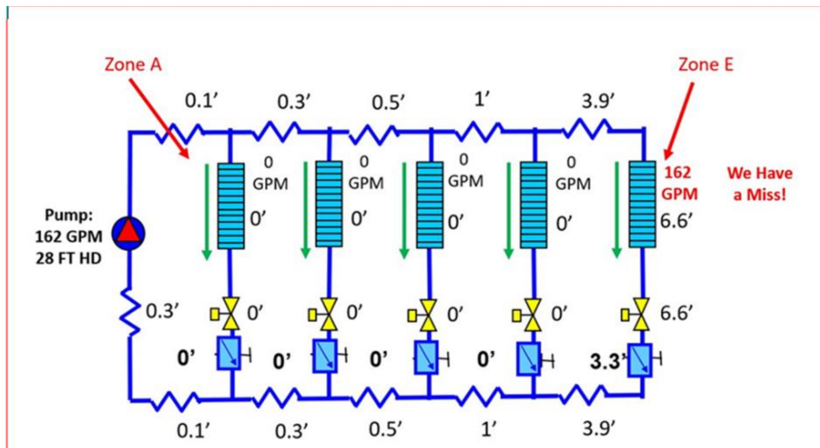
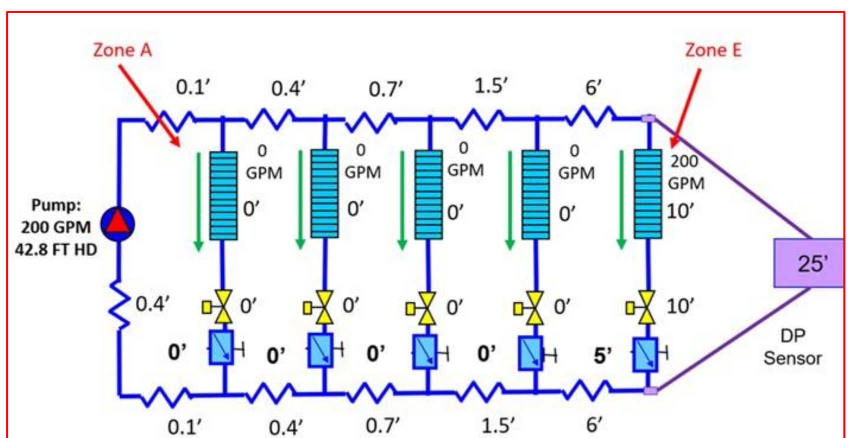


Figure 24



Control Curve Pump Control
Zone E Needs 200 GPM
Zone E Flows 162 GPM – We Have a Miss!

Figure 25



Control Area Pump Control
Zone E Needs 200 GPM

Figure 26

One word of caution when it comes to using PIC valves on variable speed systems that incorporate reset control. ASHRAE 90.1-2010/13 requires that facilities with building management systems (BMS) must continuously reset the control head based on the valve position so that under the least loaded conditions there is just enough head available to keep the most critical valve nearly wide open. This means that the position of each valve in the system must be monitored. However, differential pressure changes do not impact the flow through a PIC valve or its position. A built-in pressure regulator compensates for those increases and decreases in pressure, so the valve position doesn't change when the control head is reset. The only time the valve position will change is when differential pressure is lowered to the point that the valve is no longer in its control range. When this happens, we are no longer in control and are at risk of a big coil flow miss.

Summary

Variable speed control strategies must be applied based on the system design as well as what is practical for the given situation. There are no one-type-fits-all control solution.

In this white paper we've discussed the various options that are available for variable speed pump control and described in detail how each one works. Here are the bullet points:

- Area control with reset is the best option in terms of efficiency and avoidance of flow misses. Area control relies on a true measured setpoint and a differential pressure (DP) sensor as feedback for speed adjustment. This method can be used on a new or retrofit system without incurring too much expense. Just remember that reset only works if you can read the position of the zone valves. PIC valves won't work.

- Area control is a good option for systems that are ≤5HP and don't have a BMS. Area control uses a true measured remote setpoint so it is more accurate than sensorless curve control.

- Sensorless curve control can be a suitable variable speed control option but if air temperatures are critical, it may not be a good choice. Systems with a large amount of diversity are probably not candidates for sensorless control because there may be frequent and significant misses, resulting in occupant discomfort. Also, not every pump is suitable for sensorless curve control. If the horsepower line follows the same slope as the pump curve, then the pump flow would be indeterminable where the pump curve and horsepower lines overlap.

- Full system flow curve control uses a flow meter to read flow versus the estimate of sensorless curve control, which is based solely on the theoretical control curve. This is a good option in that it can be used on all pumps and with any drive manufacturer. It can also be applied to existing pumps already installed or in a retrofit application. But again, systems with high diversity and large control area may experience misses.

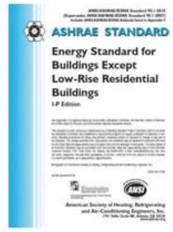
ADDENDUMS

Addendum 1

ANSI/ASHRAE/IES Standard 90.1-2010/13

HEATING, VENTILATING, AND AIR CONDITIONING

SECTION 6.5
Prescriptive Path



6.5.4 Hydronic System Design and Control.

6.5.4.1 Hydronic Variable Flow Systems. HVAC pumping systems having a total pump system power exceeding 10 hp that include control valves designed to modulate or step open and close as a 50% or less of the design flow rate.

Individual chilled water pumps serving variable flow systems having motors **exceeding 5 hp** shall have controls and/or devices (**such as variable speed control**) that will result **in pump motor demand of no more than 30% of design wattage at 50% of design water flow.**

Exceptions:

a. Systems where the minimum flow is less than the minimum flow required by the equipment *manufacturer* for the proper operation of equipment served by the system, such as chillers, and where total pump system power is 75 hp or less.

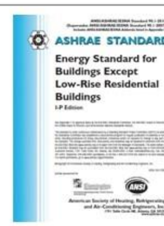
b. Systems that include no more than three control valves.

Addendum 2

ANSI/ASHRAE/IES Standard 90.1-2010/13

HEATING, VENTILATING, AND AIR CONDITIONING

SECTION 6.5
Prescriptive Path



6.5.4 Hydronic System Design and Control.

6.5.4.1 Hydronic Variable Flow Systems. HVAC pumping systems ….

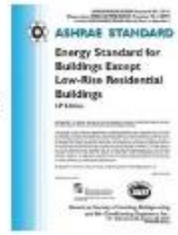
The controls or devices shall be controlled as a **function of desired flow or to maintain a minimum required differential pressure**. Differential pressure shall be measured at or near the most remote heat exchanger or the heat exchanger requiring the greatest differential pressure. **The differential pressure setpoint shall be no more than 110% of that required to achieve design flow through the heat exchanger.**

Exceptions:

a. Systems where the minimum flow is less than the minimum flow required by the equipment *manufacturer* for the proper operation of equipment served by the system, such as chillers, and where total pump system power is 75 hp or less.

b. Systems that include no more than three control valves.

Addendum 3



ANSI/ASHRAE/IES Standard 90.1-2010/13

HEATING, VENTILATING, AND AIR CONDITIONING

SECTION 6.5
Prescriptive Path

6.5.4 Hydronic System Design and Control.

6.5.4.1 Hydronic Variable Flow Systems. HVAC pumping systems ....

Where differential pressure control is used to comply with this section and DDC controls are used the setpoint shall be reset downward based on valve positions until one valve is nearly wide open.

Exceptions:

a. Systems where the minimum flow is less than the minimum flow required by the equipment *manufacturer* for the proper operation of equipment served by the system, such as chillers, and where total pump system power is 75 hp or less.

b. Systems that include no more than three control valves.

Addendum 4

Pump Affinity Laws.

These laws define the mathematical relationships between flow (GPM), pump speed (RPM/change in impeller diameter), head and brake horsepower (BHP). They are:

1. Pump GPM capacity varies DIRECTLY as the speed (RPM) or impeller diameter ratio change.
2. Total pump head varies directly as the SQUARE of the speed (RPM) or impeller ratio change.
3. BHP varies directly as the CUBE of the speed (RPM) or impeller diameter ratio

Addendum 5

SECTION 6.7
Submittals

6.7.2.3 System Balancing

6.7.2.3.1 General. *Construction documents* **shall require that all *HVAC systems* be balanced** in accordance with *generally accepted engineering standards* (see Informative Appendix E). *Construction documents* shall **require that a written balance report be provided to the building owner** or the designated representative of the building owner for HVAC *systems* serving *zones* with a total conditioned **area exceeding 5000 ft^2**.

Addendum 6

6.7.2.3 System Balancing

6.7.2.3.3 Hydronic System Balancing. Hydronic *systems* **shall be proportionately balanced** in a manner to first minimize throttling losses; then the pump impeller shall be trimmed or **pump speed shall be adjusted** to meet design flow conditions.

Exceptions: Impellers need not be trimmed nor pump speed adjusted

a. for pumps with pump motors of 10 hp or less, or

b. when throttling results in no greater than 5% of the *nameplate horsepower* draw, or 3 hp, whichever is greater, above that required if the impeller was trimmed.

Addendum 7

6.7.2.2 Submittal Manuals

Construction documents shall require that an operating manual and a maintenance manual be provided to the building owner ……..

e. A complete narrative of how each system is intended to operate**, including suggested setpoints**.

Addendum 8

PRESSURE DEPENDENT CALIBRATED VALVES	**ADVANTAGES**	**DISADVANTAGES**
 Circuit Setter	• Inexpensive • Field Adjustable • May be pre-set prior to balancing	Balancing process is tedious, especially for larger systems with many circuits
PRESSURE INDEPENDENT FLOW LIMITING VALVES	**ADVANTAGES**	**DISADVANTAGES**
 Flow Limiting Valve	• Maintains constant flow despite pressure fluctuations • Pre-balanced • Bell & Gossett Flo-Setter 2 has field adjustable operating range	• Some risk of losing control when system conditions exceed their operating range. At this point they become fixed orifices • Require some additional pump head (typically 5 to 10 feet) to make sure valve is in its operating range
PRESSURE INDEPENDENT CONTROL VALVES	**ADVANTAGES**	**DISADVANTAGES**
 Ultra Setter	• Maintains 100% authority at all times • Improves system efficiency • Lowers system energy cost • Provides stable flow and higher ΔT • An integrated solution - replaces balance and temperature control valves – so you have fewer valves to install	• Cost more • Must include some flow meters/ measure devices to measure flow—which is necessary to complete the required balancing report per ASHRAE 90.1.