BULLETIN NO. TEH-910

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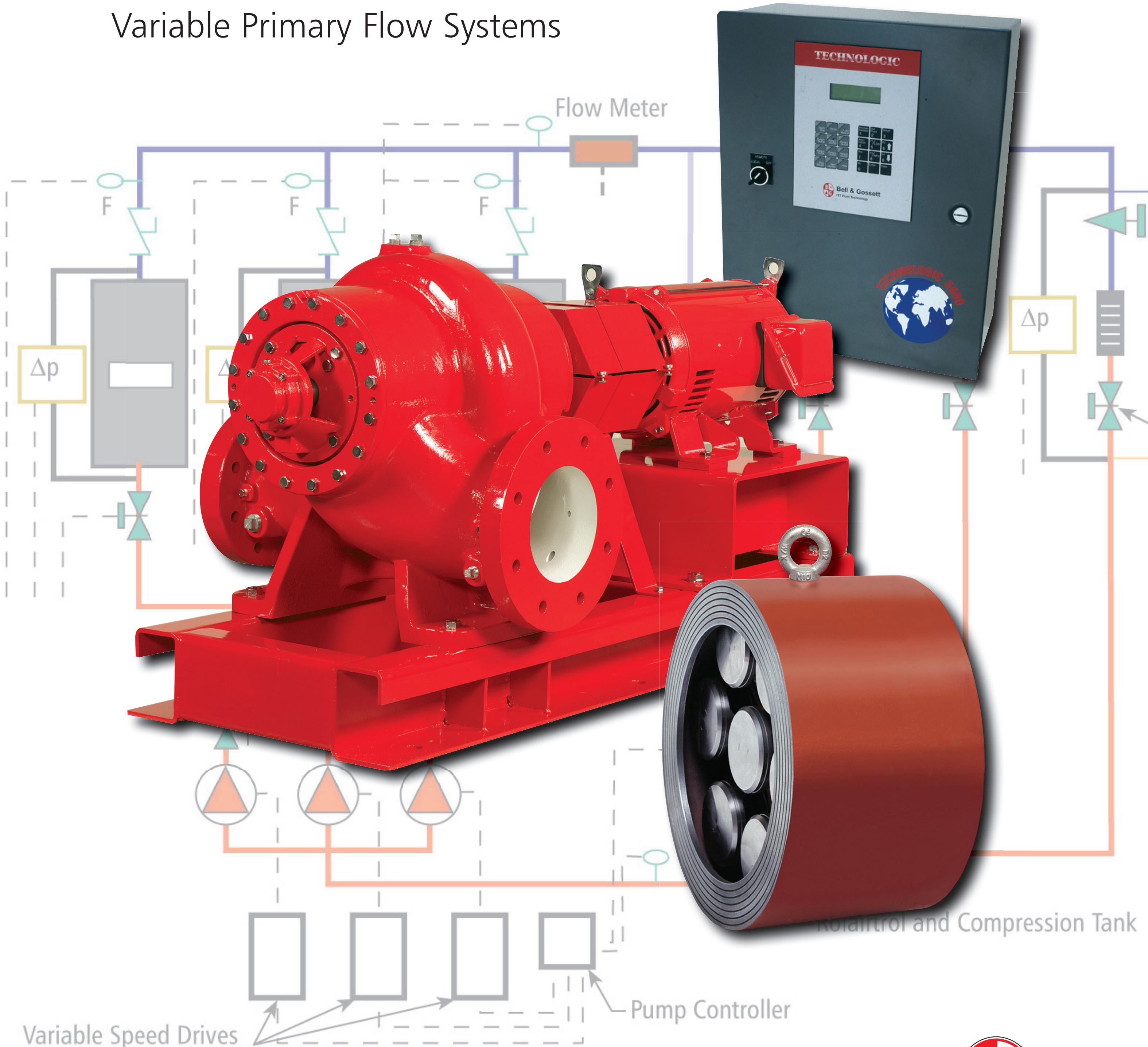
Bell & Gossett®

Variable Primary Flow Systems



Engineered for life

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NOTE:
Pump curves and other product data in this bulletin are for illustration only. See Bell & Gossett product literature for more detailed, up to date information. Other training publications as well as the Bell & Gossett design tools described in this booklet including the System Syzer, analog and digital versions, and ESP Plus are all available from your local Bell & Gossett representative. See www.fluidhandlingreps.com for your nearest Bell & Gossett representative.

Primary-Secondary Pumping Systems

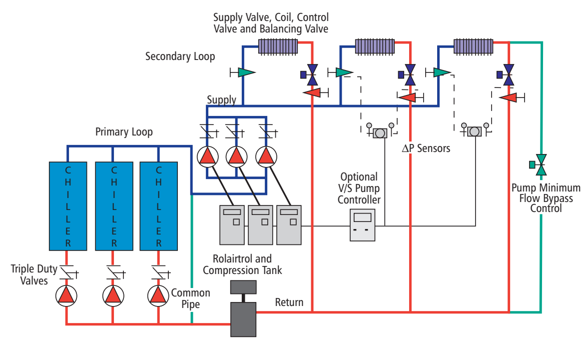
Primary-secondary (P-S) pumping, Figure 1, offers a lot of advantages in large chilled water systems. In Figure 1, each of the blue blocks represents a chiller evaporator; the chiller compressor and condenser are not shown. Each evaporator has a small, usually constant speed pump selected for the maximum chiller flow and the head loss of the "primary loop", e.g. the head loss of the evaporator, the Triple Duty Valve on the pump discharge, and the pipes and fittings in the primary loop. This loop is usually quite short, so the friction head loss is small, most of it caused by the evaporator. The larger pumps in the "secondary loop" are shown with common suction and discharge headers and with Triple Duty Valves at each pump discharge. This is necessary in order to permit staging pumps with changes in flow demand. The primary pumps could also have been installed with headers to allow more flexibility—any pump could then serve any evaporator. The secondary pumps must be selected for the maximum expected system flow, and the system friction head loss at that flow, assuming the secondary loop is a closed system. Since those pumps are providing chilled water to the entire system, they will usually have to provide much greater head to overcome losses in the long supply and return pipes, as well as the coil and control valve pressure drops. Most of the control valves are "two-way" which automatically reduce flow through the coil and the secondary pumps at part load. They are part of the automatic temperature control system as represented by the thermostat installed in a zone. Two-way valve systems are called "variable volume" systems.

This arrangement results in a "constant flow" primary loop and a "variable flow" secondary loop. The term "constant flow" may be a little misleading since the total flow in that loop varies with the number of pumps and chillers in operation, but the flow through any chiller is constant after the chiller and its pump have been staged on. The check valve feature of the Triple Duty Valve prevents backward flow through an idle chiller. The term "variable flow" in the secondary loop also requires a bit more explanation. If all the control valves in that loop were to close, for example, at very low part load, the secondary pumps would be operating at zero flow, in a "deadhead" condition which can damage the pumps. Large pumps always require some minimum flow to prevent such damage, so some provision must be made to insure that the secondary flow never falls below the minimum. A minimum number of three-way valves, or a piping bypass around the coils could be installed to provide minimum pump flow*. The maximum flow is determined by the end point of the secondary pump curve, or the capacity of the pump motor. It is not good practice to operate a centrifugal pump near the end point of its curve for prolonged periods, so a pump controller in a parallel pump installation can be used to stage on an additional pump, reducing the flow through the original pump causing both pumps to operate closer to the high efficiency portion of the impeller curve. The secondary flow can vary within these limits.

*The minimum recommended flow for Bell & Gossett pumps can be found in the Pump Details section of the ESP Plus Selection Software.

The piece of pipe shown in green is common to both loops: the "primary-secondary common pipe". It makes up for differences in flow between the two loops, allowing for "independent" operation of the pumps in each loop. The degree of independence is determined by the head loss in the common pipe. Low head loss in the common pipe results in more nearly independent operation, in the sense that flow in one loop has no effect on flow in the other. Higher head loss in that pipe causes the two sets of pumps to operate more like pumps in series; flow in one loop influences flow in the other. Proper design of the common pipe is crucial in large P-S systems. Bell & Gossett provides design guidance based on actual testing and research they completed many years ago.

Even a well designed "common pipe" cannot provide absolute independence between the piping loops. The loops still share the same fluid, and static pressure will still be communicated across the common pipe, but the loops are independent in the sense that flow in one loop will not have a material effect on flow in the other.



Primary-Secondary System
FIGURE 1

P-S Characteristics:

- It can provide nearly constant evaporator flow, one of the most important advantages. If the evaporator flow is constant, then the remaining variable is return water temperature which decreases with decreasing load. Temperatures in large volume systems with lower pumping rates generally change slowly. Given a reduction in return water temperature, there's enough time for the refrigerant control system to reduce refrigerant flow to avoid tripping the compressor or freezing the water.
- While no pumping system can absolutely prevent chiller damage, P-S pumping serves to minimize the consequences of operator error. For example, operating too many or too few chillers for a given load.
- Compared to constant volume, or three-way valve systems, a P-S variable volume system can use less

horsepower, reducing operating costs even with constant speed pumps.

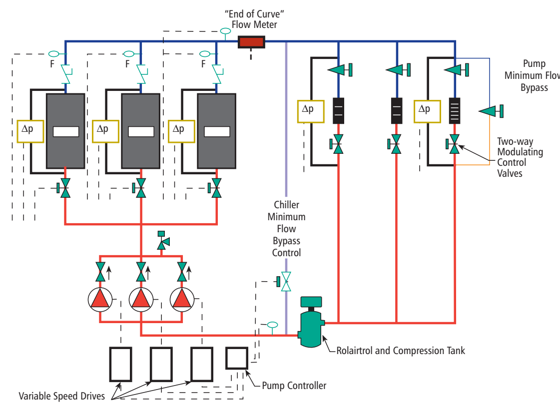
- The pumps in the variable volume secondary loop can be operated at variable speed if justified by a detailed analysis of the load profile, energy costs, and equipment costs.
- Manual or automatic chiller staging and protection is easily accomplished because a P-S system provides constant evaporator flow regardless of the number or size of the chillers in use.
- P-S allows the use of evaporators with low water side pressure drops allowing the primary pumps to be selected with relatively low motor horsepower requirements.

Given these advantages, P-S pumping became the preferred method for pumping the evaporator flow in large systems.

Variable Primary Flow (VPF) Systems

Modern chillers have improved refrigerant control systems that can provide much faster, more precise control over the refrigerant flow, so constant evaporator flow is no longer required. Figure 2 shows a typical VPF system.

- Compared to the P-S system, the most obvious difference is the reduced number of pumps. The pumps and chillers still require protection against extended operation at minimum flow, so a simple, fixed valve bypass is shown installed around the most distant coil. Instead of a simple fixed valve, a differential pressure control valve could be used. The disadvantage is higher cost and more complexity in setting the valve operating points, but the strong advantage is that the differential pressure control valve can remain closed when the system two-way valves are open, and bypass flow is not required.
- Since there is only one set of pumps, there's no need for the primary-secondary common pipe, the "decoupler" as it's sometimes called.
- Each chiller must have some kind of sensor to measure the actual flow and send that information to the pump controller. Even though the evaporator flow can be varied, there are still some limits. If the flow is too great, vibration and erosion will eventually damage the tubes used in modern evaporators. If the flow rate drops below the minimum for that chiller, there's a risk of unstable operation or freezing the water, so the refrigerant control system will shut down the compressor. Figure 2 shows either a flow meter or a differential pressure sensor at each chiller. There is no reason to have both, and each method for measuring flow has its merits.
- The VPF system must ensure minimum chiller flow at all staging points. One strategy uses coils with three-way control valves. Another uses a chiller minimum flow bypass with a two-way automatic control valve. This valve is normally closed, opening only in the event that the system flow is less than the minimum flow for the operating chiller. It could be that the minimum chiller flow bypass could also be used to provide minimum pump flow protection.
- The pumping system must use variable speed control. Typical variable speed control systems use differential pressure sensors across some of the coil and control valve branches to determine the pump speed. These are shown on two of the coils in Figure 2.
- The controller must be programmed to handle this more complex system. A more detailed discussion of the controller inputs and outputs follows later.
- System operators must understand how the system works, and operate it as instructed.

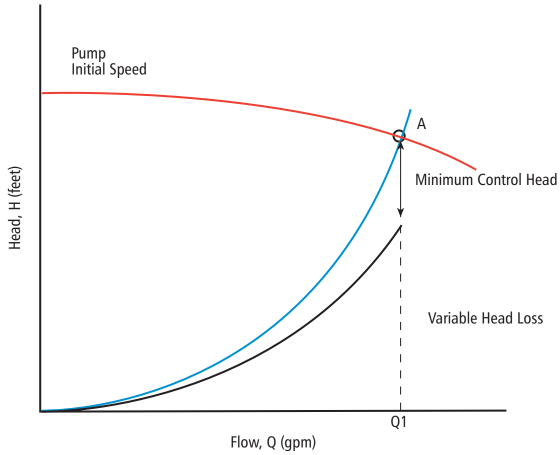


Variable Primary Flow System
FIGURE 2

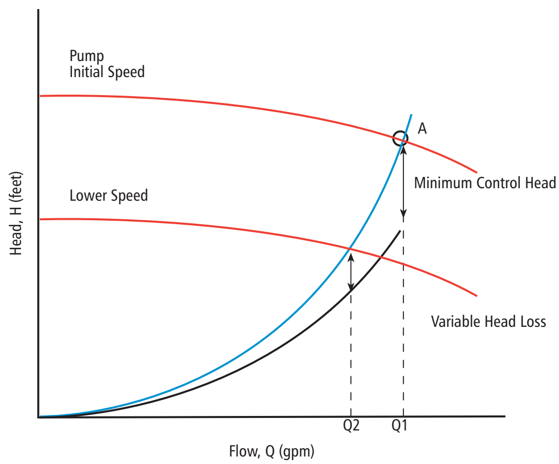
Variable Speed Pumping

Variable speed control of the system pumps will probably be more expensive than constant speed because of increased equipment and installation costs. On the other hand, properly designed and installed variable speed controls can reduce energy and other operating costs significantly over the lifetime of the pumping system, so the small increase in initial costs can be viewed as an investment with a very good rate of return. HVAC system design standards and local building codes encourage reduced energy use, so variable speed pumping becomes the normal practice, especially in larger systems that have a "load profile", e.g., a pattern of usage that includes many hours of part load operation.

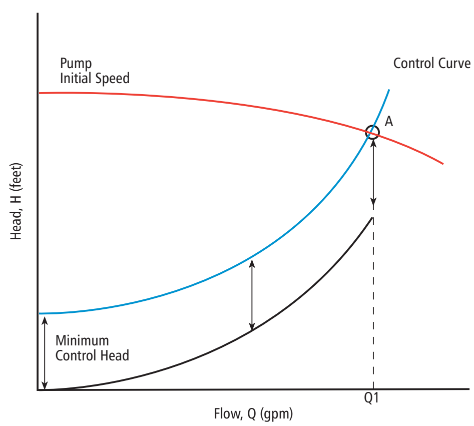
The most common way to control pump speed in HVAC systems is to install a differential pressure sensor/transmitter across the coil and control valve in the system's highest head-loss circuit. The total head loss at design flow across this branch becomes the setpoint, or "minimum control head", at the pump controller. The controller is programmed to maintain that differential pressure by lowering the pump speed as the control valve closes and raising it as the valve opens. Figure 3 shows this kind of control scheme in terms of the pump and system curves. The impeller head-capacity (pump) curve is shown for some initial rpm close to 100%. The system curve, which represents system head loss at full



Pump and System Curve, Initial Conditions
FIGURE 3

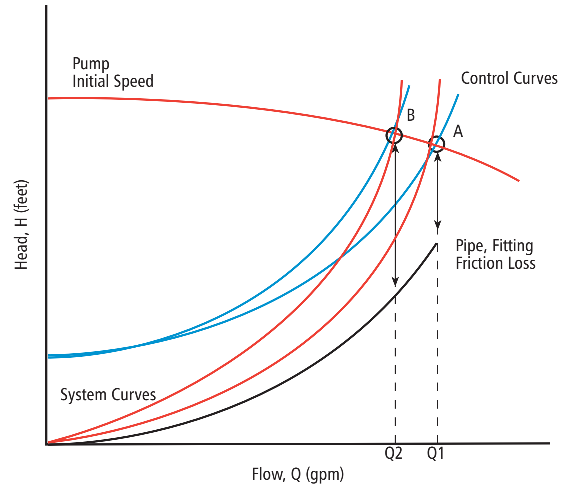


Reduced Speed at Part Load Changes the Branch Differential
FIGURE 4

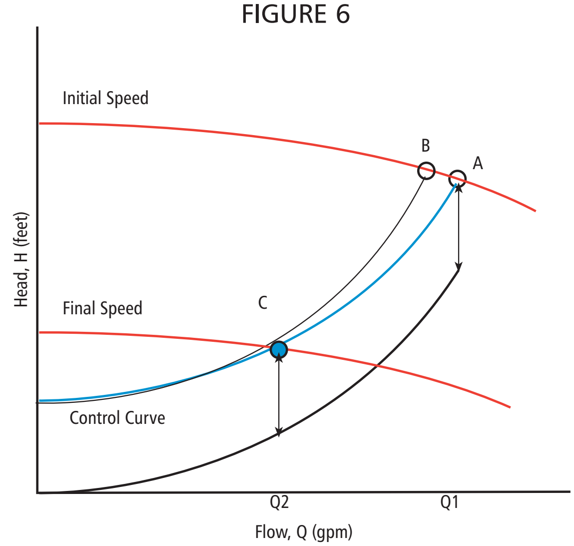


The Control Curve
FIGURE 5

flow, intersects the pump curve at point "A", defining the system design flow, Q1. Notice that this flow results in the total system head loss, represented by the sum of the Minimum Control Head, across the branch, and the Variable Head Loss in the rest of the system. If the pump were slowed to some lower speed, perhaps by using the manual speed control on the drive, the point of intersection with the system curve would shift; flow would be reduced to Q2 as shown in Figure 4. The total



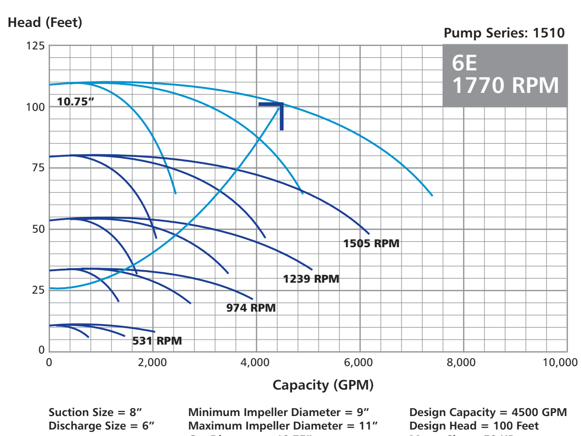
Temperature Control Valve Modulates at Part Load, Generates a Signal to Reduce Pump Speed
FIGURE 6



Reduced Pump Speed at Part Load, Branch Differential Restored to Set-Point
FIGURE 7

system head loss would be reduced because of the reduced flow, but the minimum control head across the branch would also be reduced, contrary to the programming built into the controller. The control system varies the pump speed to maintain the minimum control head automatically, so manual speed control isn't required in normal operation. This control action changes the system curve to create a modified "control curve", as shown in Figure 5. At full flow, the control curve and system curves both intersect the pump curve at Point "A", but the control curve is the sum of the constant minimum control head plus the variable head loss in the rest of the system, so at zero flow, the control curve intersects the head axis at the minimum control head, or the controller setpoint. Figure 6 illustrates how the control curve and pump curve change during automatic operation of the system. Near design conditions the pump and system operate at Point "A", but a decrease in heat load will eventually cause the control valve to modulate; reducing flow through the coil in order to reduce the rate of heat transfer from the room air to the chilled water system. The thermostat and valve controllers/actuators do this automatically. As the coil valve modulates to reduce flow, the point of intersection shifts to the left, with a rise in differential across the branch and the sensor/transmitter. The difference between the current measured value of differential pressure and the setpoint in the controller's memory will cause the pump speed to slow down in order to restore differential pressure to the setpoint. Figure 7 shows the result of this automatic control action. The system is stable at some new, lower speed, providing the reduced flow of chilled water required by the coil at the part load condition in order to maintain the thermostat's temperature setpoint. The minimum control head across the branch has been restored to the setpoint value and the pump is operating back on the original control curve. The brake horsepower being used by the pump has been greatly reduced because it's providing less head at the lower flow rate required by the part load condition. The pump can slow down under part load conditions because of the decreased head loss in "the rest of the system". The minimum control head has been restored to its original set value.

Parallel pumps are often used along with variable speed control in large systems. At design conditions, with all the system control valves open, all the pumps, operating at, or close to, full rpm, may be required in order to satisfy the minimum control head. At part load, speed is reduced and pumps de-staged to save even more energy. Figure 8 shows how three equally sized, variable speed pumps can satisfy any point on the control curve. Obviously, turning the pump motor off saves energy, but there are several other, more subtle factors that tend to increase savings even more. By careful programming, the controller can stage and de-stage pumps in order to keep the operating pumps at or near their best efficiencies. Additional savings over the life-time of the pump accrue because wear on mechanical components; e.g. coupler sleeves, shafts, bearings, is reduced by the inherent "soft-start" nature of variable speed pumps.



Bell & Gossett ESP Plus Display
Three V/S Pumps in Parallel
FIGURE 8

In a P-S arrangement, "the rest of the system" does not include the evaporator. It's on the other side of the de-coupler, operating at constant flow. In a VPF system, "the rest of the system" includes the head loss of the evaporator.

Minimum Chiller Flow

At very low loads, the pumps have slowed and de-staged. It's possible that only one of the pumps in parallel may be able to provide the required part-load flow. But if all the evaporators are open to this reduced flow, chances are good that each chiller's flow will be reduced below its minimum. Therefore, a decrease in demand for chilled water must also be accompanied by a reduction in the number of operating chillers. Turning off, or "de-staging" a chiller compressor isn't as simple as opening a set of contacts. Careful analysis is required in order to determine the part-load flow rate at which to de-stage a chiller, since isolating one chiller will increase the flow to the remaining chillers. A rapid increase in flow during this transition must be avoided by using slow closing valves on the de-staging chiller. Chiller plant operators should be involved in order to reduce the possibility and consequences of a failure in the automatic process. At the low end of the load profile, the system control valves may require less than the minimum single chiller flow. Only then will the controller send a signal to open the chiller minimum flow bypass valve. Ideally, this valve would never open, since it returns cold water from the supply to the warm water return, thus reducing the chiller entering temperature. If the temperature drops too low, the chiller's refrigerant control system will trip the compressor. Chillers used in VPF systems must have refrigerant control systems that are capable of allowing for changes in flow and temperature while avoiding unnecessary compressor trips.

Maximum Chiller Flow

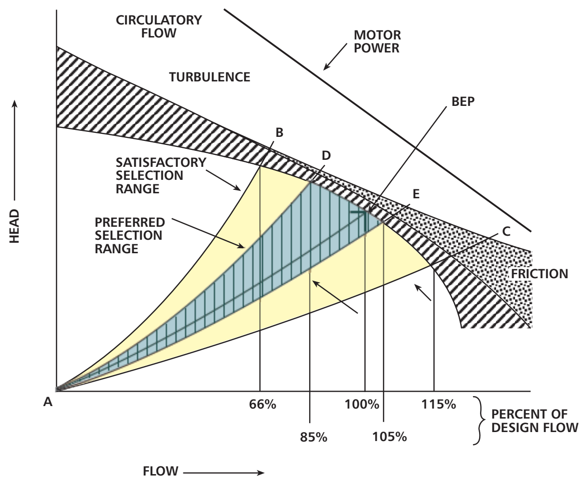
As demand for chilled water increases, the system control valves open under the control of the room thermostat and the automatic temperature control system. Additional pumps and higher speeds are required in order to meet the minimum control head setpoint. The controller must close the chiller minimum flow bypass valve as soon as flow increases above the chiller minimum. As single chiller flow increases toward its maximum allowable rate, another chiller must be staged on to handle the increased flow. The same kinds of concerns described previously now apply as the single (original) chiller flow drops and the oncoming chiller flow increases:

- The plant operators must be involved to monitor this staging action.
- The flow rate at which the next chiller is staged on must be carefully determined in order to prevent chiller upset
- Slow opening valves on the oncoming chiller will reduce the rate of change as the original chiller sees a reduction in flow.

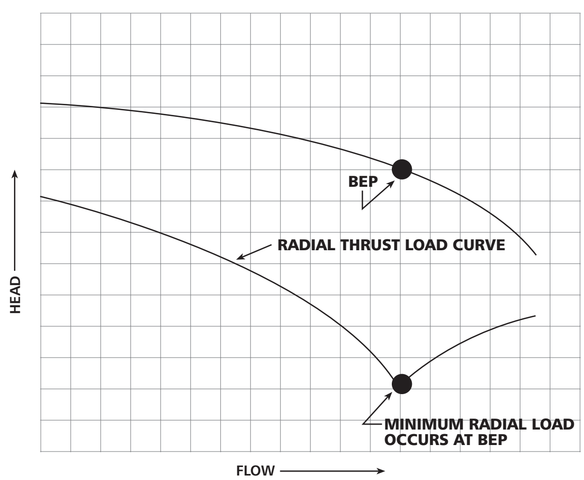
Pump "End of Curve" Protection

Just as pumps must not be allowed to operate for long periods at low flow rates, they must also avoid operation at very high flow rates. There are several good reasons for operating pumps close to their best efficiency flow.

Figure 9 comes from the Hydraulic Institute, it has also been published in the ASHRAE Fundamentals Handbook to help designers select pumps for high efficiency. For typical centrifugal pumps, there is a range of flow rates at which the best efficiency is achieved. Note that this range is not centered on the best efficiency flow, but is offset a little to the left. This means that if the pump can operate in the range of 85% to 105% of best efficiency flow, it will use less energy for a given head and flow.



Preferred Selection Range
FIGURE 9



Radial Forces Increase Away from Best Efficiency Flow
FIGURE 10

Figure 10 also comes from the ASHRAE Fundamentals Handbook. It shows that operation far to the right or the left of best efficiency flow will generate large radial forces which tend to deflect the pump shaft. If the shaft is made suitably stiff and strong, it won't actually deflect, but these forces will set up high radial loads in the pump bearings, reducing their service life. This adverse effect on bearing life is more pronounced at constant pump speed. Pumps operating at lower speed see smaller radial loads at a given point on the pump curve, which is another advantage of variable speed pumping.

Variable speed pumps in typical HVAC systems may sometimes be forced to operate at the extreme right end of their pump curve because the variable speed control system only requires that the minimum control head setpoint be maintained across the branch. It could happen that a single pump could satisfy the setpoint, but be operating well to the right of best efficiency flow, near or beyond its end of curve. In order to prevent this, one option is to install a flow meter, shown in Figure 2. This flow meter would signal that the single pump is operating too far to the right, causing the controller to stage on an additional pump—not because the set point wasn't satisfied, but because it was being satisfied at an unsatisfactory point of the pump curve.

Technologic® Pump Controller

The controller is a key component as the plant operates through its load profile. It must be programmed to handle the specific chillers, valves, and pumps used in the system.

- The chiller manufacturer must provide the actual minimum and maximum allowable flow rates for his evaporators, and the points where chiller staging and de-staging might best be accomplished
- The pump manufacturer's pump curves are required to determine pump staging, and permit end of curve protection.
- The control valve manufacturer must provide details about the valves and valve actuators at the system coils and at the chiller minimum flow bypass.
- It must be integrated into the Building Management System, if necessary.

A Bell & Gossett 5500 Technologic® Pump Controller, Figure 11, would have the following inputs and outputs for a typical three pump, three chiller system with two branch sensor/transmitters, the system shown in Figure 2.

Digital Inputs to the Technologic Controller

- Local/Remote/Off switch. Starts the pumping system manually or on receipt of an automatic signal from the BMS.
- AFD #1 on signal. Signals the controller of a successful pump drive start
- AFD #2 on signal
- AFD #3 on signal
- Pump #1 DP Switch. Signals the controller of a successful pump start.
- Pump #2 DP Switch
- Pump #3 DP Switch

Bell & Gossett Technologic 5500 Pump Controller
FIGURE 11

- Isolation Valve #1 open. Signals the controller that the on-coming chiller valve has opened.
- Isolation Valve #2 open
- Isolation Valve #3 open
- Chiller #1 start command present. Signals the controller that the chiller's internal controls have begun the process of starting the chiller.
- Chiller #2 start command present
- Chiller #3 start command present
- Chiller #1 running signal present. Signals the controller that the chiller has successfully started
- Chiller #2 running signal present
- Chiller #3 running signal present

If the controller fails to receive any of these expected inputs, an alarm condition would be displayed along with some diagnostic information for the benefit of the plant operators.

Digital Outputs from the Controller

- Open/Close Isolation Valve #1 (Chiller #1) Starts the process for opening or closing the slow acting automatic actuator for the chiller which is to be staged or de-staged.
- Open/Close Isolation Valve #2 (Chiller #2)
- Open/Close Isolation Valve #3 (Chiller #3)
- Request to stage/de-stage chiller light. Visual indicator that the load has changed enough to require chiller staging/de-staging.
- Request to stage/de-stage chiller audio alarm. Audible indicator to alert operators.
- Request to de-stage chiller. Initiates action to de-stage a chiller using its internal control system.
- Request to Stage chiller. Initiates action to stage a chiller using its internal control system.
- AFD #1 Start. Starts the pump drive.
- AFD #2 Start
- AFD #3 Start
- General alarm. Can be displayed locally or remotely at the BMS.

Analog Sensor Inputs from the System to the Controller

- DP sensor at Chiller #1. Usually an electrical current signal proportional to actual chiller flow rate.
- DP sensor at Chiller #2
- DP sensor at Chiller #3

-Or-

- Flow sensor at Chiller #1. Usually an electrical current signal proportional to actual chiller flow rate.
- Flow sensor at Chiller #2
- Flow sensor at Chiller #3
- Zone DP #1. Usually an electrical current signal proportional to the actual value of differential pressure at the branch where it's installed.
- Zone DP #2. Sensors are not required at every coil/control valve. Guidance on selecting the number and location of the sensors is available in other Bell & Gossett training resources.
- System Flow sensor. Usually an electrical current signal proportional to the total flow being provided by the pump. This is the value needed in order to protect against end-of-curve operation.
- Bypass valve feedback signal. Usually an electrical current signal proportional to actual bypass flow. Used by the controller to verify that the minimum chiller flow requested is actually being provided.
- Supply Temperature transmitter
- Return Temperature transmitter

Analog Outputs

- AFD 1 Speed. Usually an electrical voltage signal which sets the drive speed required to meet the minimum control head setting.
- AFD 2 Speed
- AFD 3 Speed
- Control Bypass Valve. Usually an electrical current signal proportional to the bypass flow required in order to meet the chiller minimum flow requirement.

VPF Systems
Part Load Operation

The system must be able to operate at reduced flow, so systems that serve nearly constant process loads may not be good candidates for VPF, especially if the initial cost of the system is greater than the cost of alternative systems. The greater the number of part load hours, the better the return on investment is likely to be. It is desirable to be able to reduce system flow at least 30% compared to design flow, so the greater the difference between full flow and part-load flow, the better.

Older systems which were equipped with constant-volume three-way valves are not good candidates for VPF or for P-S variable speed conversion unless the three-way valves can be replaced by two-way valves. Other variables which might make VPF less desirable include:

- Systems which may be sensitive to small, short term changes in supply water temperature like archives or small clean rooms.
- Systems in areas where there is high operator turnover, or where it may be difficult to find and train operators of acceptable quality.

Chiller Selection

The capacity of the installed chillers should be based on a comprehensive load calculation.

Both P-S and VPF chiller plants may benefit by considering system diversity. For example, if it's impossible to have full load on every chilled water coil at the same time, then the chiller plant can be designed to handle the smaller load imposed by the diversity factor. Both P-S and VPF systems can use economizers or "free cooling" from a cooling tower or cold lake water. It's always wise to select chillers in consultation with knowledgeable chiller manufacturer representatives.

Other sources of chilled water such as absorption cycle chillers should be considered, especially if suitable waste heat is available; although absorption chillers are best used in constant evaporator flow systems. Chillers operated in parallel must be able to provide the same water temperature change. Chillers that use the vapor compression cycle can usually provide much colder water than absorption cycle machines. Sometimes chillers can be installed in series. This is especially useful in systems designed to operate at a high design temperature difference.

It's very common to design the system with multiple, equally sized chillers. It simplifies control and staging since each chiller will have the same maximum and minimum flow rates. They will use common spare parts, and operator training requirements are minimized as well. Using chillers of different capacities may have an advantage in meeting unusual system load profiles more exactly, but they have the disadvantage of requiring a larger inventory of spare parts, and perhaps more extensive training if there are significant differences among the chillers.

Chillers in VPF systems must have refrigerant control systems that are able to handle variations in evaporator flow without upset.

Pump Selection

Multiple, parallel pumps are widely used for all of the reasons described earlier. Parallel pumps also provide a high level of redundancy. Depending on the load profile, there may be a significant number of operating hours where one or more pumps can be de-staged. Equal sized pumps allow great flexibility since any pump can supply any chiller. Other Bell & Gossett publications cover parallel pump applications in detail.

Equal capacity pumps are most commonly used, but different size pumps can be used as long as the designer is careful to avoid situations where the higher head of one pump closes the discharge check valve, (deadheads), another one. If different size pumps are installed in parallel, controls like automatic flow limiters may be required at a small chiller to avoid exceeding the maximum chiller flow if the largest pump is lined up to serve the smallest chiller.

Comparing the pump head required by a P-S system to the pump head required in a VPF system may not be as simple as it seems. Suppose a P-S system requires 25 feet in the primary pump and 100 feet of head for the larger secondary system. A simple analysis would predict that the pump head in the VPF system would be the sum of those two or 125 feet. But a closer look at the primary loop might be worthwhile. Experienced designers have long recognized that chiller manufacturers allow their chillers to operate in a range of flow rates from minimum to maximum. They have long stated this flow rate range in terms of the maximum and minimum allowable water side velocities. Most manufacturers would allow velocity to vary from 2 to 12 feet per second, consult your specific manufacturer to find the actual limits. An experienced designer may have developed the practice of selecting a chiller barrel to operate in the middle of that acceptable range at design flow, let's say 7 feet per second. That chiller head loss at the design flow at 7 feet per second was the major component of the primary loop head loss. But if that same chiller were used in a VPF system, it would have limited turndown; it would be limited to a flow reduction from 7 to 2 feet per second, reducing the energy saving potential. The chiller should be selected for design flow near the upper end of the allowable velocity range. Selecting a chiller for design conditions near the lower end of its range limits the chiller turndown, and may require the minimum flow bypass valve to open sooner. Selecting the chiller at higher head loss will then have the potential for increased turndown as flow modulates from 12 to 2 feet per second. A more accurate comparison between the P-S and the VPF systems might be on the order of 125 feet versus 135 feet.

System Rate of Change

As the flow varies between maximum and minimum at a given entering water temperature, the amount of heat being delivered to the evaporator also varies. Therefore the rate of heat transfer from the water to the refrigerant also must vary in order to maintain the leaving water temperature setpoint. Modern chillers, the kind required in VPF systems, have more precise and faster acting refrigerant controls to react to change without unexpected trips or freeze-up. Given a certain flow rate range, the next question then becomes "How rapidly can the flow go from one end of the range to the other without exceeding the ability of the refrigerant control system to maintain control?" Fortunately, system designers have been provided with several sources which answer this question. A study conducted by researchers from Pennsylvania State University, a complete citation is listed as reference 1, listed answers to this question from five different manufacturers of vapor compression cycle chillers. Answers are available from three of the five manufacturers, and they range from 25% per minute down to 2% per minute. In other words, some chillers can handle quite rapid change, others require much slower change. Some manufacturers have published more detailed information. For example:

- Trane chillers can handle flow rate changes up 30% per minute if leaving water temperature changes are not important, 10% per minute if it's important to maintain close to setpoint leaving water temperature.
- York chillers base their answer on a calculation of the "STR", where

$$\boldsymbol{STR} = \frac{\textit{System Volume (gallons)}}{\textit{System Flow Rate (gallons per minute)}}$$

Therefore, larger volume systems tend to have greater values of STR, and higher flow rate systems tend to have lower STR.

- For systems with STR>15, these chillers can handle changes from 100% to 50% in 15 minutes
- For systems with STR<15, these chillers require 15+(15-STR) minutes for changes from 100% to 50%.

In the years since that study or the detailed guidance was published, it's likely that the ability to cope with rapid change has improved; it's always best to get current information from the specific manufacturer on this issue.

Supply Water Temperature

Chillers often use the temperature of the water leaving the evaporator as the "controlled variable". Each pound of refrigerant carries a well defined amount of heat from the evaporator to the condenser. The control system varies the rate of refrigerant flow to accommodate changes in the water side flow rate and temperature in order to maintain a constant leaving water temperature. That's the reason that the rate of change discussed above had to include some concern for leaving water temperature-rapid changes in flow, particularly in small volume systems are accompanied by fluctuations in leaving water temperature in spite of the advanced refrigerant controls being used today. The temperature variations may not be large, and they will, over time, settle out as flow becomes more constant. In a large HVAC system, these changes are probably small enough to be ignored. In systems where thermal storage is available, they may not matter at all as the evaporator output is mixed with cold water from ice storage or a stratified chilled water storage tank. In small systems where close control over the room air temperature and humidity is important, these temperature variations could make a significant difference.

Minimum Chiller Flow Bypass

It is possible that a given minimum load always exists. This is especially likely if the chiller plant serves some process-related constant load that doesn't vary with time of day or season as typical HVAC loads do. But even if this constant load requires a flow rate greater than the minimum flow of the smallest chiller, the minimum chiller flow bypass is still required. It could be that the load is great enough to de-stage the smallest chiller, leaving a larger chiller operating below its minimum flow. Chiller manufacturer's recommend installing a controlled bypass so that the minimum flow rate of any active chiller can be provided at any staging condition. The cost of installing it is low, and the consequences of not having it may be severe. As system requirements change over time, it may well happen that the minimum flow chiller bypass will be needed in the future. The control valve in the chiller minimum flow bypass must be selected with a linear inherent characteristic, since the controller programming is based on the expectation that a given output signal will result in a given increase in flow through the bypass. An evaporator flow sensor of some kind is also required to provide the controller with feedback that the minimum chiller flow is, in fact, being provided. The valve must be selected with proper authority to avoid excessive distortion of the inherent characteristic. Other Bell & Gossett training materials describe important control valve issues like valve authority and rangeability in detail. Figure 2 shows the minimum chiller flow bypass installed in the equipment room, close to the pumps. There are some significant advantages in installing it out in the system, far from the pumps. During part-load hours when the minimum chiller bypass is open, the entire loop of chilled water supply and return pipes will be circulating cold water, in effect, a small thermal storage supply. If the bypass were to be installed closer to the pumps, then the water in the piping system could warm up as it absorbs heat through the pipe insulation. When one of the coil valves opens on an increase in demand, the chillers would see a short term flow of very warm water mixing with the bypass flow, which might be enough to cause an upset in their operation. The disadvantages in installing the bypass far away from the pumps are mostly in terms of finding space for the pipe, and sending the control signals over a longer distance.

Operator Capability

Central chilled water plant operators tend to come from mechanical or electrical trades, often with special training from chiller, tower, or drive manufacturers. A P-S, multiple chiller, variable speed, parallel pump installation is probably pretty familiar to all of them. Sometimes, in the press of day to day operations, as the system ages and equipment fails, they may modify the system to keep it on-line. P-S pumping systems are generally more robust than an equivalent VPF system. For example, an automatic drive bypass can seek an available drive on failure of the current drive, and bypass the drive electronics if there is none available. Certainly the operators should be aware of this situation and try to remedy it to restore variable speed operation as soon as possible. This kind of automatic response to drive failure is not recommended in VPF systems since a pump running at full rpm may increase evaporator flow well above the upper limit. Reference 2 is an example of a large system that was designed as P-S largely for that reason.

System Complexity

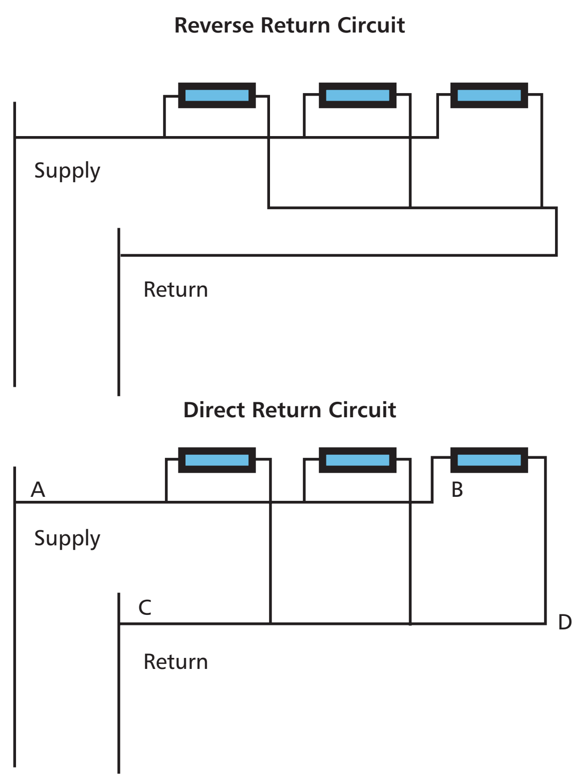
A system that is complex to one person may be very easy to understand for another. The P-S system is complex in that the piping system design includes that low pressure drop de-coupler. It usually includes some consideration for staging chillers in such a way as to limit the amount of cold water which flows across the common pipe to mix with the warm return water. On the other hand, the VPF system depends on accurate, timely, automatic sensor inputs at several points in normal operation. The selection, installation, periodic calibration and servicing of those sensors might be very complex to others.

Piping System Design

Figure 12 shows a two-pipe, direct return piping system in both the primary loop, and again in the secondary loop. Figure 12 also shows a two-pipe, direct return piping loop. While that piping system design is not the only alternative, it is among the most common for several good reasons.

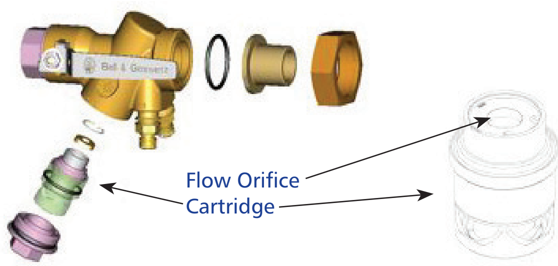
All coils see the same supply water temperature. In chilled water coils, that's usually an important factor because the cold water must drop the coil surface temperature below the dew point in order to condense ambient moisture. Sometimes it's advantageous to supply different temperatures to different coils. A variation on P-S pumping called "primary-secondary-tertiary pumping" is a convenient way to accomplish this. Details are available in other Bell & Gossett training materials.

Direct return piping uses fewer pipes, and takes up less space than a two-pipe, reverse return piping system. See Figure 12. A reverse return system uses an additional length of pipe to collect return water from the first than the second coil and so on until it carries all the return water, when it returns to the chiller plant. By doing this, the pipe length to and from each coil is equalized, taking one variable out of the balancing question. Achieving hydronic balance is a major issue in direct return systems. It's obvious in direct return systems that the first coil to receive water is also the first to return it, the last coil to receive water is the last to return it. This results in un-equal piping lengths through each coil circuit. That means a coil branch located near the pump will see a greater differential pressure, and therefore a greater flow than another branch located farther from the pump. Balancing valves of one sort or another must be installed in the lower head loss circuits in order to reduce the excess flow that would otherwise occur. Figure 13 is a Bell & Gossett Circuit Setter, a reduced port ball valve that can be manually set to add just enough resistance in the circuits that need it to make all circuits equal in head loss at design flow. The techniques for using Circuit Setters to achieve "proportional balance" are described in detail in other Bell & Gossett publications. An automatic flow limiting valve, the Bell & Gossett Circuit Sentry, is shown in Figure 14. This device can maintain design flow in spite of varying differential pressure across the branch. The size of the flow orifice determines the flow. The cartridge maintains a constant differential across the orifice in spite of variations in differential pressure across the branch which are caused by the action of other control valves, or changes in pump speed.



Reverse and Direct Return Piping In a Direct Return System, Branch to Riser Ratio is BD/AB+DC
FIGURE 12

Manual Balancing Valve
Bell & Gossett Circuit Setter ®
FIGURE 13

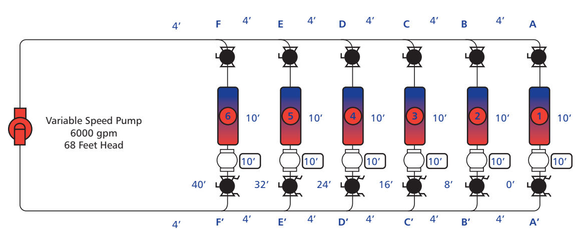


Automatic Flow Limiting Valve
Bell & Gossett Circuit Sentry ®
FIGURE 14

Although the balancing devices are important, they are best considered as the means to "fine tune" the piping system after the more important components like coils, control valves, and pipes have been selected.

Branch Head Loss

The "branch" contains the coil, control valve, isolation valves, balancing device and other accessories. A system with six identical branches is shown in Figure 15. The branch flow is determined by the heat transfer rate provided by the coil and the design delta tee. Assume that the flow required in each branch is 1000 gpm. The control valve must be selected with an appropriate inherent characteristic and great enough authority to minimize the distortion of that characteristic. The total head loss of each branch is therefore determined primarily by the coil and control valve. For this example, assume each branch head loss is 20 feet at design flow.



Direct Return Secondary Loop
Branch to Riser Ratio 0.4
FIGURE 15

Pipe Sizing Decisions

The flow rate in each section of piping is easily determined. The supply pipe from the pump discharge must carry 6000 gpm. At Point F, 1000 gpm goes to Branch 6, so section FE carries only 5000 gpm and so on. It is common practice to size pipe for the required design flow using friction loss rate in units of feet of friction head loss per 100 feet of length. Typical HVAC systems work best with a minimum of 0.85 ft/100' and a maximum of 4.5 ft/100'. Choosing pipe size within this range insures that velocities will be high enough to move unwanted air bubbles to the air separator, yet low enough to avoid noise or erosion. In large systems, the designer often has a number of choices of pipe size in this range for a given flow as shown in Figure 16. Note that three pipe sizes could carry a flow of 6000 gpm within the normal range. Of course, an experienced designer could also choose a pipe size slightly out of the normal range. For example, a 12" pipe carrying 5000 gpm is at the upper limit of the normal range, but if that section is short in length, the designer may choose to use it anyway since the next step in the process is to multiply the length of the section in feet by the friction loss rate in feet of head loss per 100 feet of length to obtain the total head loss in that section in units of feet of head loss. Friction loss in any pipe fittings in the section also has to be included, there are several ways of doing this, and they are described in detail in other Bell & Gossett publications. Returning to Figure 15, note that the total friction head loss for each section has been identified. There's four feet of head loss in each section of supply pipe between branches, and four more in each section of return pipe.

The pump head required is 68 feet, determined by adding all the section head loss values from the pump, through branch #1 and back. Assume a variable speed pump has been selected for a design condition of 6000 gpm at 68 feet of head. The differential pressure sensor/transmitter would then be installed across branch #1. The minimum control head of 20 feet set in the controller.

Flow (US GPM)	Pipe Size	Friction Loss (Feet)	Velocity (FPS)	Reynolds Number	Flow Type	Friction Factor
5000	12	4.48	14.34	1172764	Transition	0.0140
	14	2.77	11.86	1066660	Transition	0.0139
	16	1.41	9.08	933291	Transition	0.0138
	18	0.78	7.17	829403	Transition	0.0138
	20	0.45	5.77	743901	Transition	0.0138
	24	0.18	3.99	618839	Transition	0.0138
5500	14	3.33	13.04	1173326	Transition	0.0138
	16	1.70	9.99	1026626	Transition	0.0137
	18	0.94	7.89	912343	Transition	0.0136
	20	0.54	6.35	818292	Transition	0.0136
	24	0.22	4.39	680723	Transition	0.0137
6000	14	3.94	14.23	1279992	Transition	0.0137
	16	2.01	10.89	1119949	Transition	0.0136
	18	1.11	8.61	995283	Transition	0.0135
	20	0.64	6.92	892682	Transition	0.0135
	24	0.26	4.79	742607	Transition	0.0136

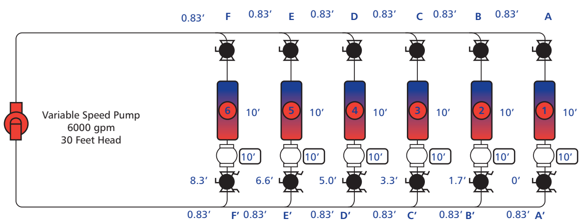
Pipe Friction Loss Rate
FIGURE 16

Branch to Riser Ratio

This represents a poorly designed system in the sense that water is not being properly distributed to each branch. The balancing valves are required to add significant resistance in the branches near the pump to prevent excess flow. Note that the balancing valve in branch #1 is set to provide no head loss at all, Theoretically, it's not needed in that branch because that's the branch that was used to determine the pump head. Additional resistance there would simply require greater pump head! The Circuit Setter in branch #2 provides eight feet of head loss to reduce the excess flow that would result from the extra eight feet of head loss required to force water out to branch #1, or the sum of supply and return pipe head loss in B-A plus A-B. Note that the Circuit Setter in branch #6 must add 40' of head loss in order to prevent excess flow in that branch close to the pump. That's hardly "fine tuning". The "branch to riser ratio" is a rule of thumb which can be used to evaluate the quality of a direct return piping system. Total head loss across each branch is 20'. The sum of supply plus return pipe head loss is 48'. The ratio is then 20/48 or 0.42.

Poor Design

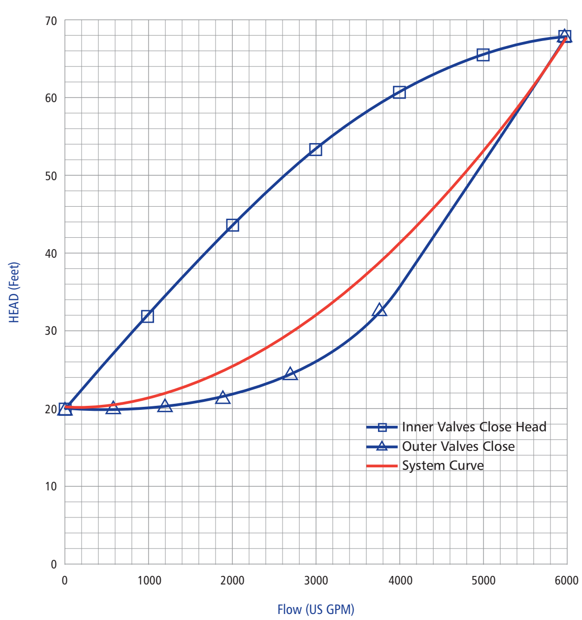
A system with a ratio this low reacts to part load in very unexpected ways. The discussion so far has assumed design, or full flow conditions, 6000 gpm. What happens if one of the branch valves closes? We would expect the flow to drop from 6000 to 5000 gpm, but in a poorly designed system, the answer depends upon which valve closed. Figure 17 shows the flow versus head loss control curve originating at zero gpm/20 feet and extending to design conditions, 6000 gpm/68 feet. There's only one point at each end of the control curve because there's only one way to get zero flow; all the valves have to be closed. There's only one way to get 6000 gpm; all the valves must be open. There are many ways to get 3000 gpm. The flow could be going to the three farthest branches, or the three nearest branches, or any combination between those extremes. The curve with the square data points represents the effect of valves closing from #6 out toward #1. The curve with the triangular data points represents the valves closing from #1 in toward #6. If valve #6 closes, the system flow drops from 6000 to about 5000 gpm as expected. But if the valve in branch #1 closes, the flow drops from 6000 to about 4000 gpm. The curves in this figure were derived using a mathematical analysis which is covered in more detail in other Bell & Gossett training materials.



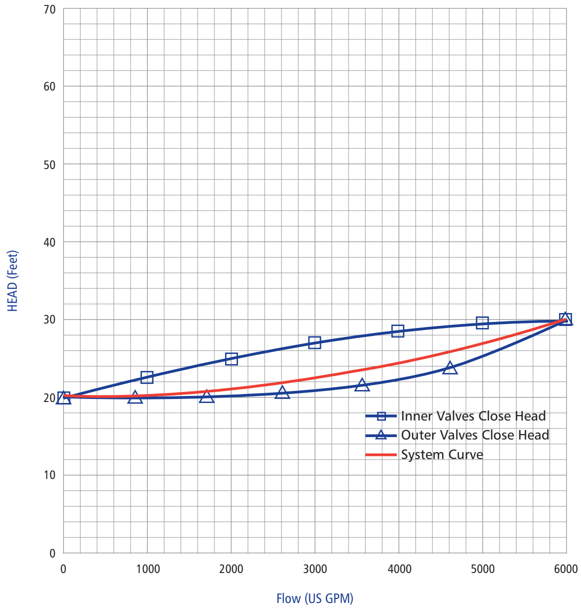
Direct Return Secondary Loop
Branch to Riser Ratio 2.0
FIGURE 18

Improved Ratio

Assume in Figure 18 that larger supply and return pipe sizes have been chosen at the lower friction loss rate. Since the length of a section didn't change, selection of a larger diameter pipe with a lower friction loss rate at the required flow would yield lower total head loss in each section as shown. The branch head loss is the same, but the branch to riser ratio is now 20/10 or 2.0. Note that the Circuit Setters have been re-set to much lower values, and the pump head has been reduced to 30 feet. The effect of part load is much more predictable. Closing only the outermost valve reduces flow to about 4700 gpm, Figure 19. Another consequence of this change is that the range of speed variation has been reduced, making variable speed pumping less attractive. Of course that's because this example was chosen to illustrate the effect of piping decisions. If the minimum control head could also be reduced, perhaps by choosing a different coil or control valve, that would change the situation too.



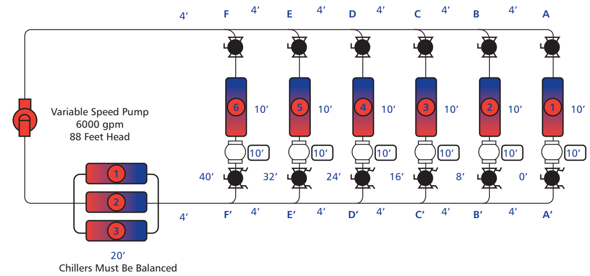
The Control Area – Ratio 0.4
FIGURE 17



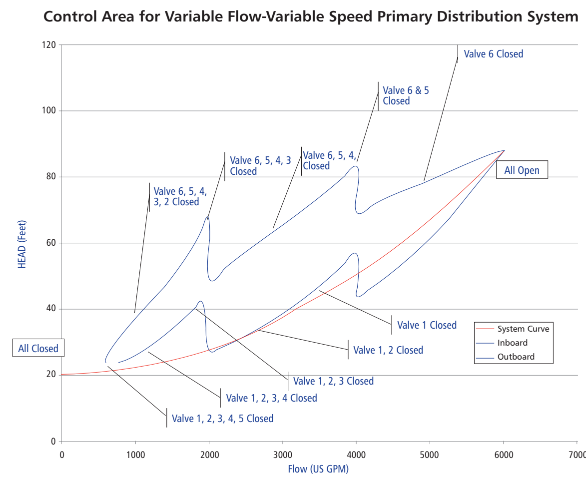
The Control Area – Ratio 2.0
FIGURE 19

The Effect of Poorly Designed Piping on VPF Systems

The example just completed assumed the chiller evaporators were on the other side of the de-coupler; a P-S pumping system. In a VPF system, there is no de-coupler, so the chiller head loss must be included in the branch to riser ratio calculation making it even more important to design the system piping to avoid unexpected, large, rapid changes in flow caused by closure of a single system valve. In Figure 20, three chillers are in parallel. Balancing devices have been used to insure that whatever flow exists will be evenly divided among the operating chillers. As discussed earlier, non-operating chillers will see zero flow. Analysis of the chiller's maximum and minimum flow rates has resulted in a decision to de-stage the third chiller if flow drops below 4000 gpm, and de-stage the second chiller at 2000 gpm. Figure 21 shows the possible consequences of poor piping system design. If the valve in branch #1 closed, the large, sudden drop in flow would tend to de-stage a chiller, leaving two operating. The abrupt rise in head loss at 4000 gpm represents the increased system head loss when only two chillers are in operation. A similar rise in head loss occurs when the second chiller de-stages at 2000 gpm. The effect of these rapid flow changes on the variable speed pump is not shown in the figure, but they have already been discussed. Good decisions in selecting coils, control valves and piping would tend to lessen these effects.



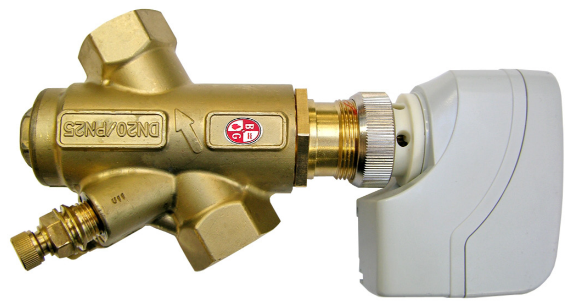
VPF System Branch to Riser Ratio 0.4
FIGURE 20



Control Area VPF with Ratio 0.4
FIGURE 21

Pressure Independent Control Valves

The valve pictured in Figure 22 has a cartridge similar to the one used in the automatic flow limiting valve. Unlike that device, this control valve has a variable orifice which reacts to signals from the thermostat and automatic temperature control system much like any other temperature control valve. Valves like this have proven to be very useful in any variable speed pumping system since they are not affected by changes in other control valves or pump speed, further reducing the rapid changes which result when all the other control valves are forced to reposition themselves due to a change in a single valve elsewhere.



Pressure Independent Control Valve
FIGURE 22

Summary

VPF systems have been successfully installed in many chilled water plants around the country, but VPF by itself is not a panacea, nor does it relieve the designer of the responsibility to apply good engineering judgment in the selection of all the components that make up the system.

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