Strategies for Design and Control of Low Energy Chilled Water Distribution Systems

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Achieving maximum efficiency in a chilled water based building conditioning system requires all aspects of the design to be carefully rethought. This paper supplements other materials on this subject and provides a specific example of designing and configuring a distribution system that minimizes the pumping power. While this example is for a hydronic system, a similar process is employed to maximize the fan power efficiency of an air distribution system.

The chilled water distribution portions of a building cooling system are a good place to look for energy reductions because outdated design and operating practices so frequently applied to these systems can result in a great deal of wasted energy. It is not uncommon to be able to cut distribution pumping energy use by more than half and often much more by incorporating advanced configuring and control practices in these parts of a chilled water cooling system. The outdated practice of developing system configurations around the principles of valve authority and the need for independence among valves was necessary before the advent of network based controls. But many modern control systems make it possible to configure chilled water distribution systems around new operating principles that allow much more efficient and effective operation. The new method of control described in this paper is called the "Valve Orifice Method" (VOM). The principle behind this new method of chilled water distribution control is, in simple terms, the replacement of independent, proportional control of the pump in response to a differential pressure setpoint with a process that continuously and automatically re-balances the system as flow requirements at the loads continuously change. The VOM is a "transitional" control strategy. As we work to develop more efficient systems, the industry must collaborate to raise the level of control applied as this is the key to next generation levels of building operating efficiency. In the long term, the engineering community must introduce new methods of applying and supporting more complex and robust controls for buildings. Because the VOM method is somewhat more complex, it is well suited to point a way for the industry to develop and support more advanced control. But it is also straightforward enough that it can be applied and supported with traditional approaches. So, it provides some insight as to the direction our industry must proceed to obtain full efficiency out of our building energy systems. What follows is a discussion of design criteria and strategies for VOM operation of chilled water distribution systems designers should consider incorporating into their systems along with a step by step example of how to apply this principle to substantially reduce chilled water distribution pumping energy consumption. This article does not discuss issues of expansion, air separation, venting, static pressure control, or make up. These and other fundamental system issues are unchanged and need to be addressed as with any closed loop water circulation system. To start in a description of the Valve Orifice Method, here are the key issues upon which designers should focus their attention in order to make the chilled water distribution system perform at maximum efficiency and effectiveness:

System Configuration: For successful Valve Orifice Method of operation, designers need to start by configuring the distribution system such that all chilled water flowing from the chillers passes through a load before returning to the plant, with a possible exception of very low flow conditions when some flow may need to be bypassed to maintain minimum flow through the chiller. This means the designer must eliminate any decoupled (primary-secondary) pumping or open bypasses, and refrain from employing any three way valves. Where multiple stages of pumping may be required, the pumping stages must be configured in series as booster pumps as shall be demonstrated. All pumping should be variable speed with pumps selected to operate efficiently at all expected flow, head and chiller sequence conditions. Piping layouts should minimize piping head loss so pipe should be sized generously (especially smaller sizes) and not employ extra piping for "reverse return" configurations. A good rule of thumb is to design the configuration so that maximum total dynamic piping pressure loss for each pump stage does not exceed 30 feet (9 meters) of pump head. If one or more lines must have a total piping head loss (including both supply and return lines) greater than 30 feet, add booster pumps as shown in Figure 1 and coordinate the operation of all pumps as described later to achieve maximum overall pumping efficiency. Designers should familiarize themselves with standard pipe fittings and use low loss fittings (such as sweeps in place of tight elbows) whenever possible.

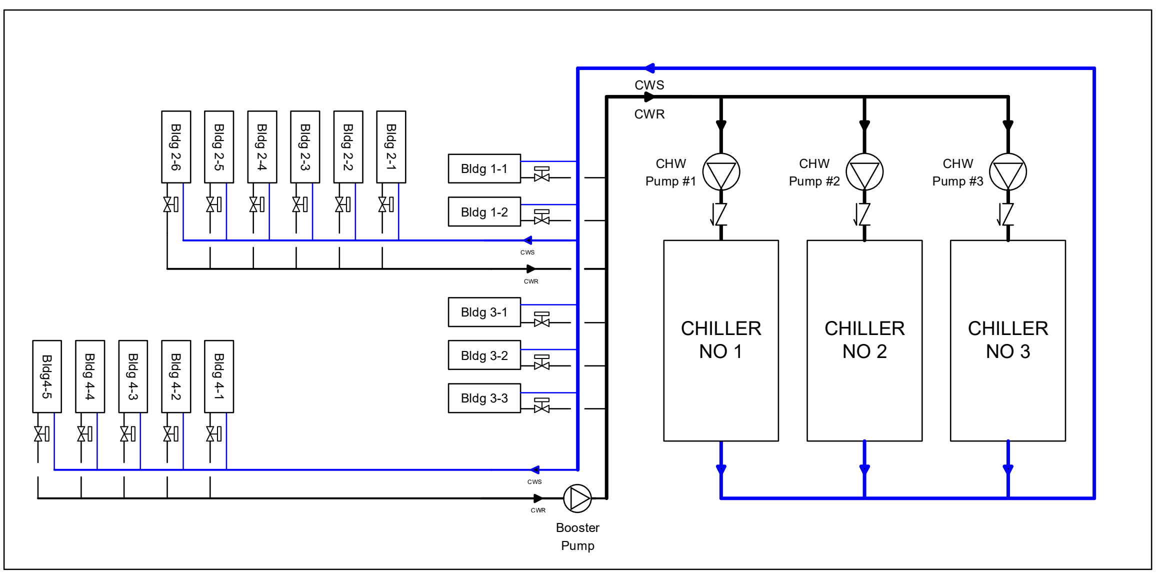

Figure 1: Schematic of Chilled Water Distribution system that serves 16 major loads in 4 buildings. Note that Building 4 has a long run and is served by a booster pump. This makes it unnecessary for the main CHW pumps to over-pressurize the other three buildings just to serve building 4. The booster pump allows better control and reduces overall pumping energy. All control valves are sized for maximum pressure drop of less than 1 psi.

Designers also need to consider how to keep the system as simple as possible and avoid the use of devices that can add to pumping pressure requirements. For example, balancing valves add to pressure losses which increase pumping energy and designers should avoid employing them. As discussed later there are more effective means to prevent overflow conditions at individual loads. If deemed necessary, it's recommended that strainers be placed near pump suction or chiller inlets where they are accessible and easily maintained rather than at each control valve.

Note that the larger control valves employed in the Valve Orifice Method (see valve selection discussion below) are far less subject to fouling or plugging than the much smaller orifices used in conventionally configured systems. Local strainers at the loads are generally not required.

Control Valves: In the Valve Orifice Method of operation, control valves are selected for a large flow coefficient (Cv) to minimize valve pressure loss. Simple but good quality electronically actuated ball type modulating valves (and butterfly valves for larger sizes) have equal percentage characteristics and make excellent control valves for the VOM method of operation. It's advised to use only full ported ball valves without any pressure compensation features and select the line size of each control valve so that the maximum pressure drop across the valve at design maximum flow is less than 1 psi (7 Kpa). Using the valve's flow coefficient makes this easy since the maximum Cv for any valve is the flow (in GPM) at which a 1 psi pressure drop across the valve will occur. So the designer simply needs to specify that only full ported valves shall be employed with a valve line size such that the valve has a Cv equal to or greater than the design maximum flow in GPM for the load it controls. Note that this larger valve Cv will make the actuation-to-coil capacity relationship non-proportional under many operational conditions. So, if a standard PID control is employed to modulate the valve, control should employ Integral control as the primary valve positioning means. But designers should also consider replacing or supplementing PID modulating valve control with new relational control strategies to make each non-proportional valve control regimen more robust and flexible. How to ensure effective valve control with the Valve Orifice Method will be discussed later.

Pump Selection and Operation: In compact distribution systems, just a single (primary only) pumping stage should be employed to pump water though the chillers and loads. Either dedicated (one pump per chiller) or header pump arrangements using an isolation valve on each chiller can be employed in the Valve Orifice Method of operation. If non-identical pumps are selected for parallel operation, the designer must be sure pump curves are consistent or analyzed to develop optimally efficient speed ratios among the dissimilar pumps. How to select non-identical pumps for parallel operation (which is usually necessary when dedicated pump configurations are applied with non-identical chillers) is an analysis regimen with which designers should become familiar.

In large, extended distribution systems the primary pumps should be supplemented with "booster" pumps so as not to exceed the maximum piping loss between the stages of pumping. For campus applications where a system serves multiple buildings it often works well to have a booster pumping stage at each building such that the primary pumps operate only to overcome the losses in the chillers and main distribution headers.

Operating with the Valve Orifice Method, pump speed is typically adjusted at 30 second intervals. The pump speed is based on the current orifice area open for all control valves served by the pump(s) rather than an end-of-line differential pressure setpoint. The percent of maximum orifice area for a valve with equal percentage characteristics can be determined by raising the fraction of valve opening to the 2.6 power. So, by multiplying the design flow for each load times the fraction of the current valve orifice area open (valve position fraction raised to the 2.6 power) for each load and valve and adding these together for all loads as the numerator, and then using the total pump design flow as the denominator, the percent pump speed is determined at each control interval. An adjustment factor is then added that ensures all loads are adequately served with absolute minimum pump head. Note that the control system incorporated to provide this control must have custom programming, robust auto networking, and advanced floating point math calculating capabilities. And accurate representation of the current position of each load control valve is required. Applying network based VOM control takes a little computation to set up, but once the control algorithms are completed, pump control will be smooth, stable and, most important, very efficient! If one or more booster pumps are employed as in Figure 1, they are considered the same as a load. For example, if a 500 GPM booster pump is operating at 50% speed and the primary pump has a 2,500 GPM design capacity, the load served by the booster pump will add 10% to the main pump speed (500 x 0.5 I 2500). System operation with the valve orifice method is quite straightforward and not at all difficult to apply once its setup has been worked through once or twice.

Energy Performance: Designers should strive to make total chilled water pumping power requirements as small as possible. Chilled water distribution systems should be designed to operate at no more than 0.03 kW/ton average annual chilled water pumping power, and much lower if the chilled water distribution system is compact. It is not uncommon to see conventional chilled water pumping consume 0.1 kW/ton or more in systems today so the potential reduction is substantial.

Putting It All Together - A Chilled Water Distribution System Example: Assume we are tasked with selecting pumps and control features for the chilled water distribution system application shown In Figure 1 that serves four separate buildings. Each of the loads shown are for comfort conditioning. And an analysis of the system dynamics indicates that the plant as shown with three equal sized chillers is adequate without operating at exceptionally low loading on any chiller. A design chilled water delta T of 15°F (S.3°C) has been selected. The cooling coils have been selected to provide the design maximum rated capacity at the 15°F (S.3°C) design delta T.

DISTRIBUTION PIPING LAYOUT

The starting point for such a distribution design is to layout the piping. The layout may be subject to some changes as the building design develops, but it is a good idea to get this in progress as soon as possible in order to be certain other disciplines understand space requirements for the distribution system piping as there is often competition within the design team for such space. To develop piping layouts two rules are recommended. First, a computerized piping system analysis program should be employed and kept up to date as the piping layout progresses so that pipe sizes and pressure losses are quantified as the design is developed, and updated to accommodate load adjustments, architectural changes and other design modifications. For simple distribution systems it's not difficult to develop spreadsheet analysis for the piping analysis. But with the availability of easy-to-use commercial piping analysis programs it's recommended that designers familiarize themselves with, and use, one of the many now available, especially if the piping system is more complex or incorporates multiple flow paths.

The second rule is to always consider the use of larger piping sizes and the least possible number of fittings in order to minimize pressure losses in the piping. The uncontested worldwide expert in low loss piping system design is Mr. Lee Eng Lock, and it's recommended that designers familiarize themselves with Mr. Lee's low loss piping design approach through his many well documented designs. It is also important that designers have a thorough understanding of the nature of pipe size selection. Pipe size is traditionally dictated by a combination of pressure loss and a maximum allowable velocity. Smaller piping is usually limited by pressure loss and larger piping by flow velocity limitations. So it is typically the smaller size piping that needs to upsized to minimize pressure and pumping energy losses. And upsizing this smaller piping is usually not costly. Furthermore, developing a layout that minimizes the number of fittings can reduce the installation cost. So the economics of larger piping and less fittings will reduce long term energy costs without adding Significantly to the system cost. Figure 2 is a schematic of the distribution system with the maximum design loads shown in RT. Figure 3 is the same schematic with the maximum design flows shown for each load in GPM.

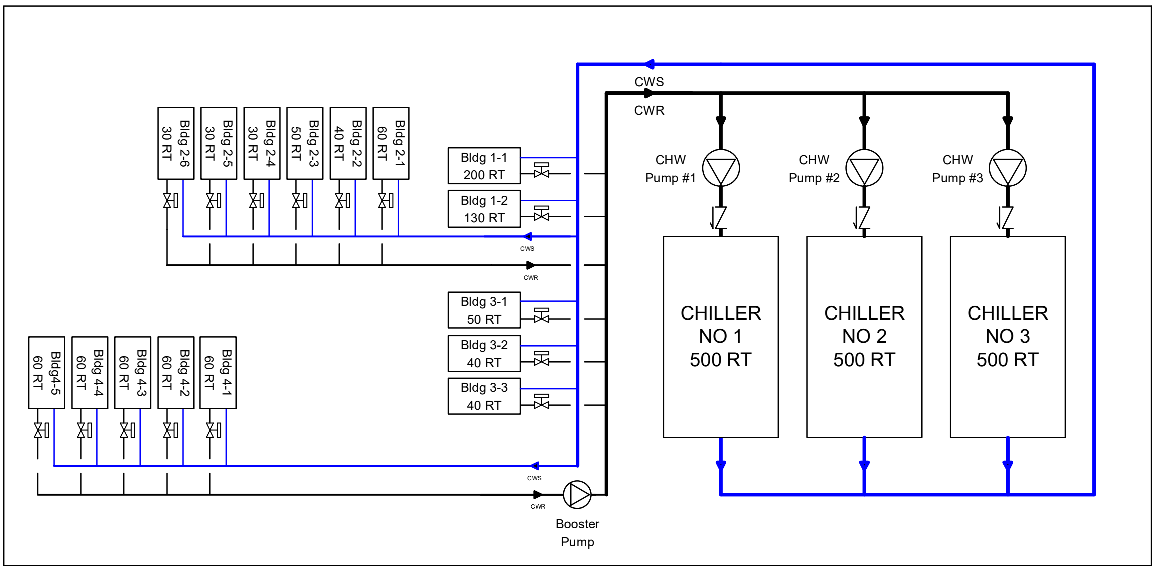

Figure 2: Figure 1 Schematic with loads and equipment capacities shown

In this example we assume the 30 feet maximum head rule can be observed with the primary pumps for all but the piping mains to Building 4. While piping losses and pump head calculations are not included in this paper, we should assume the system has been configured with booster pumping stage serving Building 4 to satisfy the rule that each pumping stage should pump through a maximum piping pressure loss of 30 feet (9 meters). Note: this piping loss rule does not include chiller barrel or load pressure losses. The booster pump in Figure 1 serves loads in the remotely located Building 4 to meet that rule.

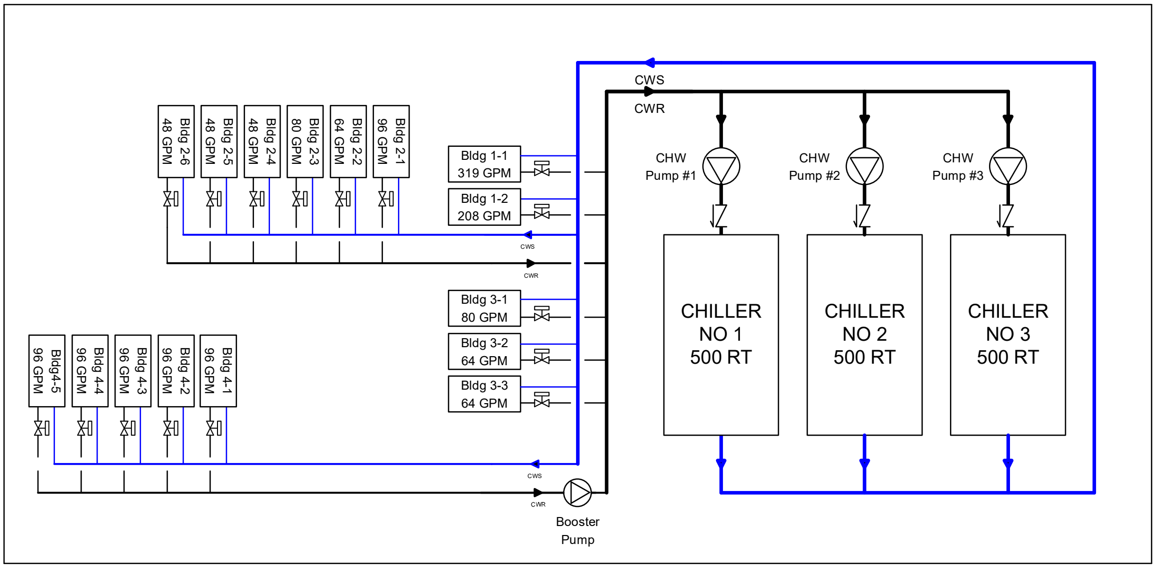

Figure 3: Figure 2 Schematic with design maximum flow (GPM) at each load shown

PUMP SELECTION

How the chilled water pumps are sized and selected is very important to the success not only of the distribution system, but the overall plant and cooling system operation as well. In terms of configuration, some designers, owners, and operators have a preference for dedicated pump arrangement as shown in the Figure 1 configuration and others for the header pump arrangement in Figure 4. A header pump arrangement has the advantage of easily adding an N+1 pump so that cooling capacity is not affected if a chilled water pump fails. It also has the advantage of the ability to pump greater flows through chillers to achieve full chiller capacity if delta T falls below design. And it does permit the use of more on-line pumps than chillers which may save pumping energy under certain conditions. However, header arrangements do require an automatic isolation valve for each chiller and more piping, often leading to higher system costs and higher system pressure losses. A designer may find it possible to achieve about the same overall efficiency and performance with a dedicated pumping arrangement but if a dedicated pump arrangement is selected, it is recommended to size pumps for a slightly lower "system" delta T so that sufficient additional flow is possible through each chiller if the system if the delta T falls below design. This can be especially important when replacing chilled water plants that serve existing loads with uncertain or varying delta T characteristics.

So let's run through the selection of the primary pumps for the system depicted in Figure 1 which incorporates the dedicated pump configuration. When a dedicated pump/chiller arrangement is employed, pumps should be sized to ensure full chiller capacity can be applied if the chilled water delta T falls below the design value. How much extra flow capacity is needed is a decision the designer should make based on experience and knowledge of the system. A good rule is to size pumps and pump head to be capable of circulating additional flow in order to meet the full load at with a delta T at least 2°F (1°C) below design delta T. In this example, the design delta T of 15°F (8.3°C) was reduced to 13°F (7.2°C) to size the pump flows and head requirements. The 920 GPM (212 CuM/Hr) pump capacity is based on that reduced delta T and ensures adequate pumping capacity is available to fully load each chiller even if the design delta T degrades to some extent over time. As noted, this up-sizing of the chilled water pumps may not be required when a header pumping arrangement is employed as an extra pump can be brought on line if delta T falls below design.

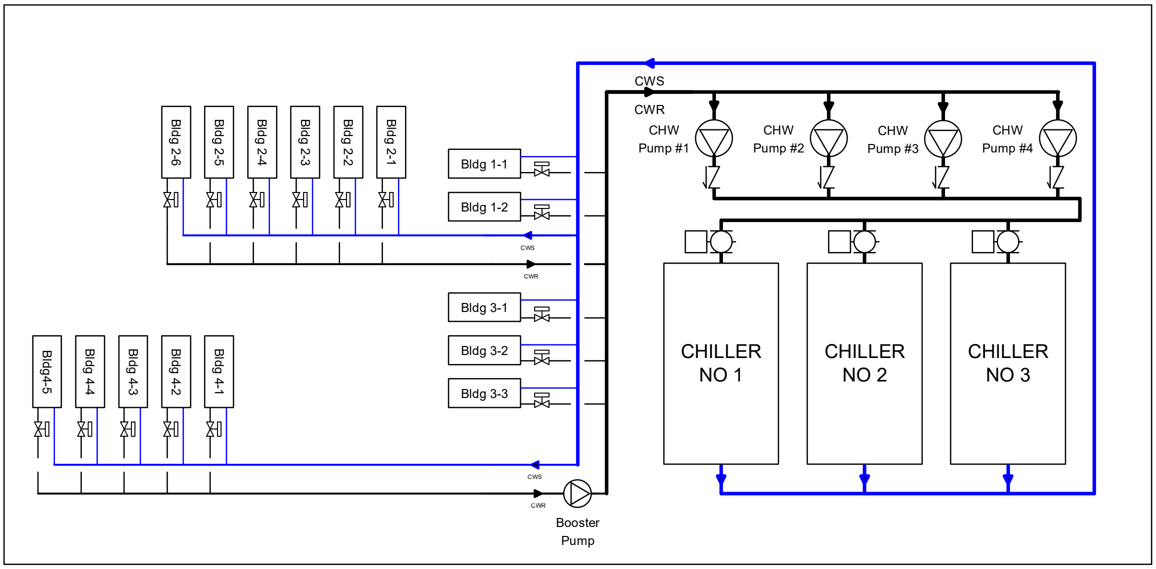

Figure 4: Schematic of same Figure 1 Chilled Water Distribution system except with a header type primary pump arrangement

The flows in Figure 3, based on the design delta T were used to size the piping, but now, the analysis is repeated to determine flows and pump head required to operate with the diminished delta T. The result of that analysis is shown in Figure 5. In this figure, the maximum chilled water flow to each load is based on the lower 13°F (7.2°C) value. The analysis of the system is reworked to determine the required pump head for these higher flows. The final primary pump selection is shown in Figure 5.

When selecting pumps, designers should also pay close attention to the shape of the pump curve, the motor type, and motor size. Generally, it is wise to select the highest efficiency pump that also has the highest shutoff head, so a "steep" pump curve is preferred. For the motor, designers should select high efficiency motors but be careful that the motor is sized only for the maximum power required by the application. Pump manufacturers typically prefer sizing the motor to ensure sufficient power at so called "end of curve" conditions. For this, the motor is sized such that under no operating conditions can it be overloaded. However, when variable frequency drives (VFDs) are applied to control motor speed, they are easily set up to limit speed and prevent motor overload in unanticipated high flow - low head operating conditions. Therefore it is unnecessary to size the motor any larger than required to meet the design maximum flow and head conditions. Oversizing motors adds cost and can also degrade operating efficiency. Designers should also consider employing ECM permanent magnet motors when possible, especially if it is expected that the pumps will be operating at and below 50% speed for substantial periods. The ECM motor is more efficient, especially at the lower speeds and loads.

Note that even with providing extra pumping capacity, the total chilled water pumping power 52.5 HP for a 1,500 RT plant. This ratio of nearly 30 RT of plant capacity per chilled water pumping horsepower is a good start, but to be really successful it will also have to be set up to be controlled for maximum efficiency. That is the task we will next concentrate our efforts upon.

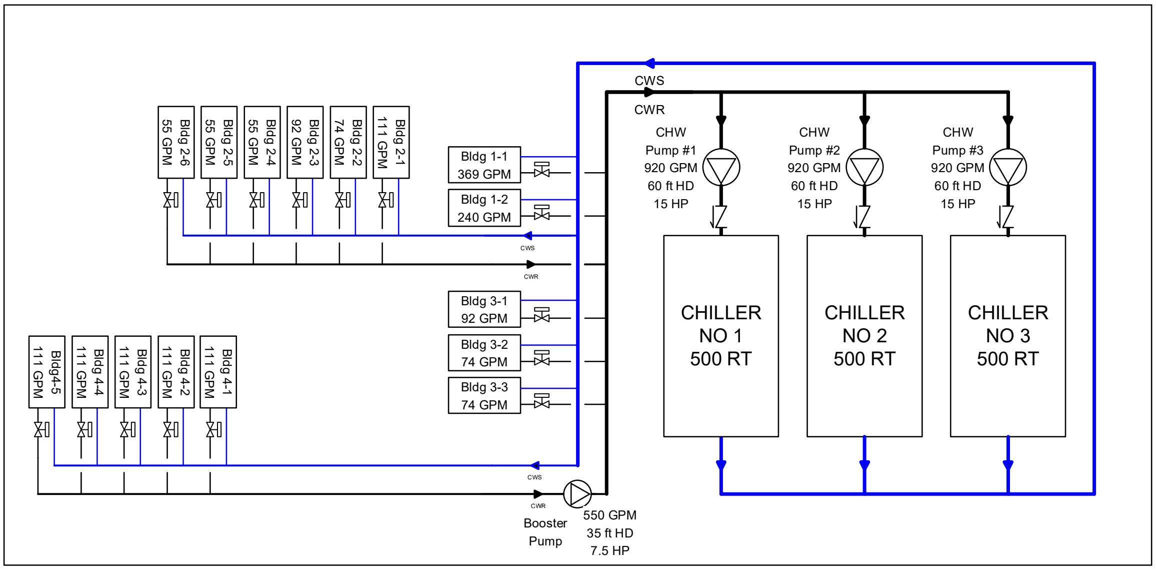

Figure 5: Figure 2 Schematic with design maximum flow (GPM) at each load at 13°F (8.3°C) Delta T. Pumps have been selected to meet operation at this lower Delta T to ensure full chiller capacity is available in the event Delta T falls below design.

CONTROL VALVE SELECTION

Notice that the system flow and head analysis was completed without considering the load control valves. That's because in the Valve Orifice Method of control, the control valves operate to continually "re-balance" the system as flow requirements at each load change. In this system operating strategy, the valves do not add to the overall head on the pumping system. In the past, control valve selection has been dictated by the valve authority method wherein the control valve is responsible for as much as 25% to 50% of total design system head requirements which results in extremely inefficient operation. To save this wasted pumping power, the Valve Orifice Method of valve and pump operation is based on employing valves with near-zero pressure losses at most operating conditions. Rather than mandating that each valve provide a proportional relationship between actuation and cooling capacity, and that all valves operate independently of one another, this new approach incorporates the control valves in a control strategy that continually re-balances the distribution system as the required flows at each load change. This holistic approach to pump and control valve operation means that control valves are selected for near zero nominal pressure loss which eliminates the wasted pumping energy of the valve authority method. This new method of control also ensures the operation of the valves and pumps is smooth, stable and more effective as well as efficient.

Simplicity, reliability and low cost make standard electrically modulated full ported ball valves ideal for the Valve Orifice Method of distribution system operation. Ball type modulating control valves are generally available in all nominal pipe sizes up to 2-1/2 inches or larger. There are several methods of controlling these valves, but the most effective are over a network in which the valve is commanded to a particular position, or with an analog signal (voltage or current loop) whose value determines the position. To implement the Valve Orifice Method of control, it is required that the position of each modulating valve is transmitted to the pump control software. For these types of control the properly scaled command signal can be used for the position signal since the internal electronics of the valve set the valve in a specific position based on the signal. For other types of digital control such as PWM, the valve should be provided with, and connected to, a separate position monitor input to the controls so that the system can read the position of the valve at all times.

Working in IP units, the control valve size selection for each cooling load is very easy. Since in IP units the Cv of a valve is the flow in GPM that produces a pressure drop across the valve of 1 psi, and since it is desired to have 1 psi or less pressure drop across the valve at design full flow, the designer simply specifies that the valve supplier select the full ported ball valve size for each load that has a Cv equal to or greater than the maximum design chilled water flow in GPM for each load.

As an example, consider the flow for each of the loads in Figure 3. The valve supplier needs simply to select and install the full ported valve size that has a CV equal to or greater than the design maximum GPM for each load.

Because the strategy of the Valve Orifice Method is to conduct a continuous re-balancing of the system, separate manually adjusted balancing valves at each load are unnecessary and can be counterproductive. If there is a concern that local valve control may allow overflowing at loads, it is recommended that a temperature sensor be installed on the chilled water return line from each coil and connected to the control system. Then, using the return chilled water temperature from the load and incorporating control that will be discussed later, the modulating valve itself becomes a multi-purpose, control, balancing, and flow limiting valve - but one that operates more efficiently and effectively than systems traditionally operated with separate valves, or with valves that employ mechanically operated flow control features.

OTHER SYSTEM COMPONENTS

This paper does not address the issues regarding system expansion, makeup and air venting or separation as those issues are not affected by this method of control. Selection of the cooling coils or heat exchangers is also beyond the scope of this paper. However, from the standpoint of the chilled water system, selection of each coil or heat exchanger with a low waterside pressure drop is always preferred. And it is preferred that all coils or cooling load devices have similar pressure drops at design full flow. It is also important to install line size full flow manual shutoff valves so that each chiller, coil, valve, pump or other system component can be easily isolated for maintenance or repair without requiring a more general system shutdown. And finally, it is valuable to incorporate pressure/temperature ports at the inlet and outlet of each coil and generously within the plant and throughout the system so that technicians have easy access to check temperature or pressure conditions throughout the system if flow or pressure problems develop.

Notice that this system configuration does not include a bypass line and valve to ensure the chillers maintain the manufacturer's minimum flow requirements. These low load minimum chiller flow requirements are generally falling as internal chiller controls improve. Some chiller manufacturers do not have a minimum flow requirement. In the Figure 1 configuration, the minimum flow limit, if present, can be maintained by simply opening valves of loads that are not currently operating if flow falls to that minimum limit. This is a simple solution that does not require any additional system components. However, if desired, the designer may choose to add a separate bypass line and control valve that is normally closed, but modulates open anytime a single chiller is operating. The bypass valve then modulates simply to maintain the flow in the system at the chiller manufacturer's minimum flow limit.

IMPLEMENTING VALVE ORIFICE METHOD CONTROL

There are two distinct steps to successfully implementing the Valve Orifice Method of chilled water distribution control. One is the development of an effective control regimen for the modulating control valve at each load. The second is the development of the pump speed control algorithm. In this section we will review the development of each of these control steps.

Valve Control: Selecting larger valves to control the flow of chilled water through the cooling loads requires several issues be addressed in developing a successful control regimen for these control valves. The most important is that the control provide adequate controllability so that the control does not cause the valve to continuously hunt. To help maintain controllability with the larger valve size employed by the Valve Orifice Method, designers should work to ensure the control provides the lowest possible pressure across the valves and that the chilled water temperature is maintained at the highest possible value that will achieve the temperature and humidity requirements of the loads the system serves. Providing these features not only assures controllable valve operation, but also result in a more efficiently operating system. This is indeed a case of managing closely related synergistic measures. The adjustment of chilled water supply temperature is beyond the scope of this paper, but the control of the pumping system to minimize the differential pressure at each load will be discussed in the pump speed control section.

However well the chilled water temperature is managed, the relationship between control valve actuation and coil capacity is not likely to be proportional. For the Valve Orifice Method of operation, it is recommended that if PID control is employed to position control valves, that the primary means of control be the integral part of the control, and that the control intervals be generously spaced so that the effect of the control change at each interval has a chance to influence the controlled variable before the next interval.

With this means of control, it is also suggested that at the startup of each air handler or other cooling load, the valve be immediately opened instead of waiting for the integral dominated control to react. This is very easy to do by simply providing a one-time command to pre-position the cooling coil valve anytime cooling is required when the air handler is started. In such a circumstance the valve can be set to a mid-point position at startup from which the valve control regimen then starts its positioning. An improved version of this approach involves setting the initial valve position as a function of initial fan speed and difference between the design and actual air temperature difference across the fan. For example, if at startup, the fan is operating at 75% speed and the actual air temperature difference across the fan (return air temperature minus the current supply air temperature setpoint) is 60% of the design difference, the valve would be commanded to an initial position of 45% (60% x 75%).

If there is a concern that the local load controls may not be well managed such that greater than design capacity may be programmed into one or more of the loads (for example, setting a supply air temperature setpoint below the design value), then it is wise to include a temperature sensor in the chilled water return line from the coil. This return chilled water temperature leaving the coil or load is then used to limit the flow through the coil or load. If the chilled water return temperature from the load falls below the design leaving temperature, the load is being overflowed and the valve opening is limited by the controls to ensure this temperature does not fall further. This will prevent a serious chilled water overflow condition through the coil. Generally it is recommended that the value used for this temperature limit is somewhat less than the design coil return water temperature value and that if it is reached and takes over control of the valve, a notification is made to the operator that the valve control is attempting to overflow the load so that the problem can be corrected. This flow limiting function is easily applied when a chilled water return temperature sensor is applied to each load.

Finally, if the chilled water plant is scheduled to start at the same time as the loads become active, and if chilled water is required to provide cooling, it is recommended that the start sequence for the chilled water plant be coordinated with the loads to ensure the plant is started and is supplying chilled water when the loads are started. Such coordination is accomplished by pre-positioning the active cooling coil control valves and starting the chiller plant before the air hander fans are started. This can improve both comfort and energy performance since many chilled water plants can take five to ten minutes to make chilled water after the command to start has been issued.

Pump Control: From a control standpoint the major difference between a conventional chilled water distribution system and one operating under the Valve Orifice Method is the speed control of the chilled water pumping. Pump speed is not controlled to maintain a differential pressure, but instead the speed is controlled based on the percent of valve orifice required at each load to maintain the needed capacity of the load. The valve position is controlled as described in the Valve Control section. Then the pump speed is controlled according to the number and capacity of pumps that are currently on line, the weighted average of the percentage of total orifice area active for the loads served, and a correction factor based on the maximum position of all control valves. What follows is a step by step development of the pump control algorithm for the primary pumps in Figure 1. This algorithm may be developed by the designer, or it may be required to be developed as part of the control sequences by the control contractor, or it may be implemented by a third party optimization specialist. However it is implemented, it is important that the process is well understood so that it can be properly reviewed, implemented and corrected if problems are found with its operation. Here then are the steps involved in developing the pump control algorithm. The information for these steps is shown in Table 1

CHILLED WATER PUMPING SPEED CONTROL SETUP TABLE

PRIMARY PUMPING

Primary Pump	Design GPM	Design CuM/Hr.
CHWP1	920	212
CHWP2	920	212
CHWP3	920	212
Total	2,760	635

PRIMARY PUMPING COOLING LOADS

	Design GPM	Design CuM/Hr	% of Pumping
Load 1-1	319	73	11.6%
Load 1-2	208	48	7.5%
Load 2-1	96	22	3.5%
Load 2-2	64	15	2.3%
Load 2-3	80	18	2.9%
Load 2-4	48	11	1.7%
Load 2-5	48	11	1.7%
Load 2-6	48	11	1.7%
Load 3-1	80	18	2.9%
Load 3-2	64	15	2.3%
Load 3-3	64	15	2.3%
Bldg 4 Pump	550	127	19.9%
Total	1,668	384	60.4%

BLDG 4 PUMP COOLING LOADS

	Design GPM	Design CuM/Hr	% of Bldg 4 Pump
Load 4-1	96	22	17.5%
Load 4-2	96	22	17.5%
Load 4-3	96	22	17.5%
Load 4-4	96	22	17.5%
Load 4-5	96	22	17.5%
Total	480	110	87.3%

Table 1:: Representation of Spreadsheet to establish Pumping Speed Control for Figure 1 Primary and Booster pumping.

Step 1: Summarize the total primary pumping capacity: In the left side of Table 1, the Design Maximum GPM (CuM/Hr) of the each selected primary chilled water pump is listed and totaled.

Step 2: Summarize the individual design maximum flow for each load served by the primary pumps: In the center of Table 1, the Design Maximum GPM (CuM/Hr) for each load served by the primary pump is listed. Note that the flows listed are the design flows at the design delta T. This data is taken directly from the coil or load data provided by its manufacturer. Note also that the Building 4 booster pump is included, but the Building 4 loads are not included in the listing for the primary pumps. The reason is the Bldg 4 booster pump is served directly by the primary pumping, but the Building 4 loads are not; they are served by the Bldg 4 booster pump. Those loads are shown in the right hand side of Table 1.

Step 3: Calculate the percent of the total pumping capacity that each load requires at design flow. In center portion of the table, the column labeled H% of Pumping" provides the percent of the total primary pumping capacity that each load represents at design flow (for example, Load 1-1 is 319 GPM I 2760 GPM (73 CuM/Hr 1635 CuM/Hr) or 11.6% of the total primary pump capacity. The Bldg 4 pump is treated just as another load served by the primary pumps. At design maximum flow it represents 19.9 percent (550 GPM / 2760 GPM).

For the Bldg 4 loads, their percent of pumping capacity is based on the booster pump. So, for example Load 4-1 is 96 I 550 or 17.5% of the design maximum booster pump capacity.

Step 4: Develop the Algorithm that will establish the pump speed at 30 second control intervals: This is accomplished by multiplying the percent of total orifice area for each valve that is currently open by the percent of pumping that each valve flow represents and totalizing it for all valves. Then, since multiple pumps are involved, this must be adjusted for the number of pumps that are currently operating. And finally, an adjustment is made to be certain maximum valve position.

Primary Pump Speed = (CHWV1-1POS ^ 2.6 x 0.116 + CHWV1-2POS ^ 2.6 x 0.075 + CHWV2-1POS ^ 2.6 x 0.035 + CHWV2-2POS ^ 2.6 x 0.023 + CHWV2-3POS ^ 2.6 x 0.029 + CHWV2-4POS ^ 2.6 x 0.017 + CHWV2-5POS ^ 2.6 x 0.017 + CHWV2-6POS ^ 2.6 x 0.017 + CHWV3-1POS ^ 2.6 x 0.029 + CHWV3-2POS ^ 2.6 x 0.023 + CHWV3-3POS ^ 2.6 x 0.023 + BLDG4BPSDP x 0.199) x (3 / NOPP) + ADJ

Where:

Primary Pump Speed: is the calculated speed command as a fraction of full speed that is sent to all operating primary pumps. This calculation is made and the new primary pump speed command is issued every 30 second control interval. Pump speed is subject to a minimum pump speed (usually 35% to 50%).

CHWV1-1POS through **CHWV3-3POS** is the current position (as a decimal fraction) of the control valve serving each load.

BLDG4BPSPD: is the speed as a decimal fraction of the full speed at which BLDG 4 Booster Pump is currently operating.

NOPP: is the number of primary pumps currently operating

ADJ: Is the valve position adjustment factor. It's value is determined as follows:

1. When chiller plant starts, its initial value is zero
2. If any valve is full open, its value is incremented by 0.5% each 30 second control interval
3. In no valve is more than 85% open, its value is decreased by 0.5% each 30 second control interval.
4. The value of ADJ is limited to the range of -15% to +15%

Table 2: Primary Pump Control Algorithm for Figure 1 Chilled Water System

The Final algorithm to control pump speed for the primary pumping system is shown in Table 2. The first factor consists of the summation of the current valve positions raised to the 2.6 power (to calculate the fraction of the orifice area that is open), and then multiplied by the percent of the primary pump flow that each load would require at full design flow. Then at the end of the last line, this term is corrected for the number of primary pumps on line. Finally, the ADJ factor is added to provide adjustment that ensures all loads will be satisfied without excessive pump speed.

Readers are encouraged to inspect and review this algorithm closely and when satisfied it is fully understood they are encouraged to work to construct the Booster Pump control algorithm as a next step to understanding the Valve Orifice Method process for operating chilled water distribution systems.

Following these rules and examples for chilled water distribution system design can not only reduce energy use and help make achieving an advanced Green Mark or other high efficiency performance rating for your cooling system a reality, it will also help point our industry toward new approaches that can succeed in applying the more robust control needed to achieve maximum efficiency from our building systems. I invite your thoughts, comments, and good ideas about this paper. Please contact me at thc@hartmanco.com