The Utter Simplicity of
Contemporary HVAC Piping

By

James B. (Burt) Rishel, PE

November 14, 2004 Issue

The advent of digital control and variable speed pumping has greatly decimated the number of pieces of equipment needed to control the flow of water in chilled and hot water distribution systems. This applies to most hot and chilled water systems that have variable heating and cooling loads. Gone are balance valves, multiple duty valves, primary/secondary pumping, cross-over bridges and other energy-consuming piping accessories. Almost all that is left now are the pumps, piping, isolation valves, and coils with their control valves. We now have adaptive control that provides a continually adjusting program that has completely eliminated manual operation.

Without manual adjustment, we no longer need the tools that provided it such as balance valves and multiple duty valves. Also, precise control of pump speed has proved that piping arrangements that were used to control pump flow and pressure are now unnecessary. What is left now are the variable speed pumps and the coil control valves that give us the pressures and flows that are required in any part of a water system under all load conditions. In the following figures, the manual isolation valves will not be shown nor will the air separators or expansion tanks. All of these items will be substantially the same in all of the figures.

Let us start with a simple, single zone chilled water system that would have had constant speed pumps with total manual adjustment, Figure 1. This was the configuration regardless of any cooling load variation on it. Figure 1 was a fine system circa 1960 before the days of affordable variable frequency drives. If you have merely a constant load like a computer center without external exposure, it still would be worth consideration.

Today, this system with variable flow would most likely have only variable primary pumps with no balancing whatsoever, Figure 2. The coil control valves would be of a quality that would prevent overflow of any coil, regardless of its position in the water system. There are hundreds of systems like this in operation today. There is no need to control the flow of the pumps manually, since the pump speed control can be set for any desirable maximum speed. The advent of variable primary pumping began around 1994. Initially, chiller manufacturers had to be conservative with the allowable minimum flow. It was generally limited to a minimum evaporator tube velocity of 3 feet per second. Today, some chiller manufacturers have reduced this limit to as low as low as 1½ feet per second at part-load on the chiller. Ultimately, it is believed that this acceptable minimum flow question will be solved by the chiller manufacturers themselves by equipping each chiller with an integral circulator and by reducing the minimum flow requirement through improved refrigerant control. Due to this variation in minimum tube velocity requirement for different chiller manufacturers, it is imperative that each project have the minimum flow verified to insure compliance with the chiller manufacturer's requirements.

Many HVAC systems are much more complicated than the simple system of Figures 1 and 2. Figure 3 describes a traditional primary/secondary, constant flow, chilled water system with four zones; it has constant speed pumps with totally manual adjustment. All of the pumps would have been equipped with balance valves and check valves as well as isolation valves. The cross-over bridges would have had return control valves that decoupled the secondary and tertiary (zone) pumps.

It is apparent that there will be considerable energy waste in this manually balanced system if it were applied to a variable load application. The energy losses would be appreciable in the cross-over bridge valves in the building zones. If the zones are far apart, much of the energy of the secondary pumps is lost in the zone valves. Along with the energy loss through the bridge/return valves, the balance valves on all of the pumps and cooling coils would typically be throttled, resulting in energy wastes. This indicates that great energy waste occurs if we try to fit a constant speed system to a variable load installation. Conversions of constant speed systems to variable speed applications without the need for manual balancing have resulted in significant energy savings for the variable flow system.

This was demonstrated in the retrofit of a hotel in Southern California and was reported by Mr. Kent Peterson, PE in an article titled "Variable-Primary–Flow, Chilled–Water Plant Conversion", HPAC Engineering magazine, March 2004 issue. In this case, the cooling production and distribution equipment was downsized from the original chiller plant, and it defined the ability to reduce the capacity of the pumping equipment, chillers, and cooling towers when changing to variable flow. After conversion to variable primary pumping utilizing a smaller chiller plant, Mr. Peterson said: *"the data show that the VPF chiller plant saved approximately 68% of the pre-retrofit base energy consumption".*

There are several contemporary procedures for pumping a four zone system such as that shown in Figure 3. The distance between the zones is utilized to determine the best procedure for pumping this system. If the loads are close together in one building, only one set of pumps, Figure 4, is required; this represents a typical variable primary pumping system. The short distance to the farthest load does not impose a high distribution piping system friction on the load nearest to the chiller plant. If each zone is a building, and there is a considerable distance between the buildings, distributed pumping should be considered to avoid excessive pump head in the near buildings, Figure 5. The only chilled water pumps required in the chiller plant itself are the minimum flow circulators required to protect the chillers. This situation requires the use of an automatically controlled isolation valve on each chiller return piping to stop flow in that chiller when it is not supplying cooling to the chilled water system. Without these isolation valves, uncooled water would pass through an inoperable chiller and mix with cooled water from operating chillers. This would elevate the supply water temperature above the desired temperature.

A friction loss of 8 to 14 feet or more through pump fittings such as suction strainers, check valves, suction reducers, discharge increasers, isolation valves and piping is saved whenever a pumping system is eliminated. Quality design limits this loss to 8 feet while poor designs will reveal losses as high as 14 feet or more of pump head. In this case, variable primary pumping would, therefore, save as much as 16 to 28 ft of pump head by eliminating secondary and tertiary pumping installations.

Minimum flow circulators have a relatively low motor horsepower requirement. Assume that a chiller plant has 2,000 ton chillers, each with a design flow of 4,000 gpm. Additionally, assume that the chiller evaporators were selected at 7 fps with a minimum allowable flow at part load operation of 1½ fps. The minimum flow would 1.5/7 of 4,000 gpm or 860 gpm. If the friction loss through the evaporator were 21 ft. at 4,000 gpm, the design head for the minimum flow pumps would be $(1.5/7)^{2} \cdot 21$ or 1 ft. plus the piping and pump fitting loss. Assume the piping and fitting loss to be 16 ft, so the design head for the minimum flow pumps would be 17 ft. If the pumps have an efficiency of 80%, the brake horsepower would be 4.6 hp and the pumps would be equipped with 5 hp motors. The pumps would be variable speed so that the actual energy consumption would be quite low. Since the loss through the evaporator is so low that it may not be represented by a smooth curve, it should be verified by the chiller manufacturer. Control of these circulators should be accomplished either a chiller plant flow control as shown in Figure 5, or it can be by differential pressure transmitters connected across the chiller evaporators, Figure 6.

Recognizing the inefficiency of many existing chilled water systems, it is possible to retrofit a constant flow system of Figure 3 into a more efficient system through the use of the existing pumps and piping, Figure 7. All of the pumps would be converted to variable speed. The minimum flow through the chillers would be controlled by a new bypass valve. The speed of the chiller pumps would be controlled by a new differential pressure transmitter, DP_1, whose setting would be determined by the controller for all of the pumps. The secondary pumps would be controlled by the differential transmitter, DP_2, at the far end of the chilled water loop. This would be less costly than changing the pumps and piping to a variable primary pumping system as shown in Figure 4. However, a consulting engineer could make a careful evaluation to determine if it is economically feasible to downsize the pumps, chillers, and cooling towers as Mr. Peterson did and reported in the above referenced article.

The tertiary pumps in the buildings would be controlled by differential pressure transmitters, DP_2, at the correct location in each building, possibly at the top of each zone as shown. A bypass check valve would be located around each of the building pumps, and the return valves with all of their energy waste would be eliminated. Under light load, when there is adequate pressure provided by the secondary pumps, this will be indicated by a rise in the differential pressure at the building differential pressure transmitters, DP_2. After a time delay, the building pumps would be stopped and the building differential pressure transmitters would control the secondary pumps. Likewise, when the building load increases and the building pumps are needed, they will restart automatically. All of these piping and control procedures are now in operation on many chilled water systems.

A number of the diagrams in this article illustrate the use of differential pressure transmitters as the means of controlling pump speed. So far, this is a tried and proven method in use over the past twenty-five years. Contemporary digital electronics indicate the possibility of mass flow as a means of pump speed control. Unfortunately, this procedure also requires physical location of each individual cooling load because friction to a specific load is determined by the distance of that load from the pumping system. The total calculation then becomes quite complicated; so far, the author has not seen total pump control software that recognizes this fact.

Another explanation should be made concerning the piping diagrams in this article. All of the figures describe equal friction losses in the supply and return mains. Such is not always the case on large, actual systems. Often, the return piping has different circuiting and, therefore, unequal friction losses. The return piping should be evaluated as carefully as the supply piping.

One of the advantages of the use of cross-over bridges and tertiary pumps is the ability to blend warm return water with cold supply water to operate the zones at higher temperatures than those developed in the chiller plant. With the quality of today's piping insulation, however, there is little need for such blending. If there is a research or process facility that requires colder chilled water, it is usually better to provide a booster chiller for that facility rather than absorb the energy penalty produced by generating the entire cooling load at the reduced temperature. For example, if the research facility requires 40°F chilled water, the booster chiller could reduce the 45°F distribution water temperature. Obviously, this is an engineering design problem requiring an evaluation in each specific case.

All of the above figures have shown the chillers and their pumps assembled in tandem; it may be more economical to provide these pumps headered as shown in Figure 8. This could provide better programming of the pumps and offer the opportunity to overflow the evaporators in event of low chilled water return temperatures. Recognizing that chiller output is flow times the differential temperature, the chiller output can be sustained by increasing the flow during the existence of reduced differential temperatures. This is has become quite common now due to the many instances of low return temperatures in chilled water systems. All chillers have a maximum allowable velocity such as 10 fps in the evaporator. The chiller manufacturer should be consulted before overflow is attempted.

ASHRAE recognizes that the parasitic load of pumps and cooling towers may equal or exceed the energy consumption of the chillers themselves. Efforts are now being made to develop a Guideline to help in the determination of the total energy consumption of a chiller plant, not just the kW/ton of the chillers themselves.

Another significant improvement in HVAC water distribution systems is the realization that the quality of coil control valves must be such that they will withstand the broad differential pressures that have always existed in these systems. In the past, many of these coil valves were of a commercial grade that could not withstand the differential pressures imposed upon them. Today the use of industrial quality valves as well as pressure independent valves has improved the differential temperature and reduced dramatically the flow in these systems. This matter of valve authority and quality should be reviewed thoroughly in a separate technical paper.

Another development that has enhanced the efficiency of these systems has been that of a running limit on contemporary variable frequency drives. This prevents the pump from overloading the drive. This means that the drive and motor can be sized at the design condition rather than at the pump run-out condition. For example, assume that a chilled water pump requires 29 bhp at the design condition and 35 bhp at the run-out condition or the non-overloading point on the pump curve. With older drives, it would have been necessary to equip the pump with a 40 hp motor and drive. With this running limit, the pump can be furnished with a 30 hp motor and drive without concern for the hp requirement at the pump run-out condition. This reduces the first cost of equipment and total electrical installation; it also provides higher wire-to-shaft efficiencies at reduced loads on the system.

A recent development has been the use of geothermal energy for cooling and heating. Unfortunately, there has been some confusion about the capabilities of these thermal sources during variable flow. If the thermal source, such as a well field or lake, is capable of producing the energy required at design flow, there will be adequate thermal capacity at reduced flow. Misunderstanding this has resulted in the application of complicated pumping and piping systems, Figure 9. Consulting engineers, who understand the concept of the variation in heat flow at reduced flow, have developed simple pumping and piping systems as shown in Figure No. 10. A number of consulting firms have numerous installations in operation with this configuration. This demonstrates that there is no need for separate well pumps or for balancing valves on the circulating pumps or the building heat pumps.

Summary:

The most significant fact about all HVAC water systems is not the system's configuration, but what are the hours of operation at all percentages of flow, from minimum flow to maximum design flow? Only after receiving this information, can an engineer make use of the above suggestions on the reduction of first cost and increase of overall operating efficiency.

The increase in quality of coil control valves and the added software for the control of these valves and pumps has resulted in simplicity and greater efficiency for most of the HVAC industry's chilled and hot water distribution systems. Today's level of ability and competency of the designers and operators of these systems permits the use of this technology on all variable volume, HVAC systems and not just on special installations.

References:

Bahnfleth, W. and E. Peyer. 2004. *Varying Views on Variable Flow Chilled Water Systems,* HPAC Engineering 76 (3) S5-S9.

Bahnfleth, W. and E. Peyer. 2003. *Energy Use Characteristics of Variable Primary Flow Chilled Water Pumping Systems*. Proceedings of the 21st IIR International Congress on Refrigeration.

Avery, Gil. *Operating Chillers in the Max-Cap Range.* Engineered Systems, June 2003.

Taylor, Steven. *Primary Only vs. Primary Secondary Variable Flow Systems. ASHRAE Journal, February 2002*

Avery, Gil. *Improving the Efficiency of Chilled Water Plants,* ASHRAE Journal, May 2001.

Rishel, James B. (Burt). *Control of Variable Speed Pumps in HVAC Water Systems.* ASHRAE Transaction CH-03-4-3.

Rishel, James B. (Burt). *Use of Balance Valves in Chilled Water Systems.* ASHRAE Journal, June 1977.