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SOLUTIONS

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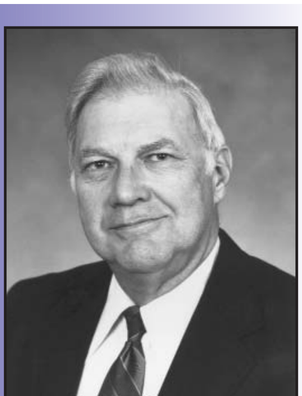

Article by Gil Avery, P.E.

Improving the Efficiency of Chilled Water Plants

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For the past 30+ years most central chilled water plants have utilized primary-secondary pumping arrangements as shown in Figure 1. The primary system includes all equipment to the plant side of points "A" and "B" and the secondary system includes all equipment to the load side of these points. The majority of these systems have the chillers staged from flow. As the flow in the secondary starts to exceed the flow in the primary, an additional primary pump and chiller is staged on. As the flow in the secondary decreases so that one less chiller can handle the reduced flow, a pump and chiller is staged off. This method of control assumes that the load is proportional to flow and that the ΔT between the chilled water supply and return is constant and at the design condition.

These assumptions are generally wrong. Systems rarely operate at design ΔT because of coils that are dirty, inadequate coil control valves, thermostats set wrong, excessive temperature drops in the supply and return mains, etc. Therefore, flow is not a reliable indication of load. As a result, the vast majority of central chilled water plants operate inefficiently with lower than design ΔT's and with more partially loaded chillers on line than would be necessary if the chillers were properly staged.

The Way It's Been Done

The typical primary/secondary system shown in Figure 1 is fully loaded at the outdoor design conditions of 95°F dry bulb and 78°F wet bulb. The system is operating as designed with 6,000 GPM of 45°F supply water and 57°F return water (2 GPM/ ton). Each of the three 1,000 ton chillers and companion pumps are handling 2,000 GPM. The fourth stand-by pump is off.

In Figure 2 the load is only 600 tons. The ΔT is 6°F (4 GPM/ton); typical of many systems designed to operate at a higher ΔT. Two pumps and chillers are on line, with each chiller inefficiently loaded to 300

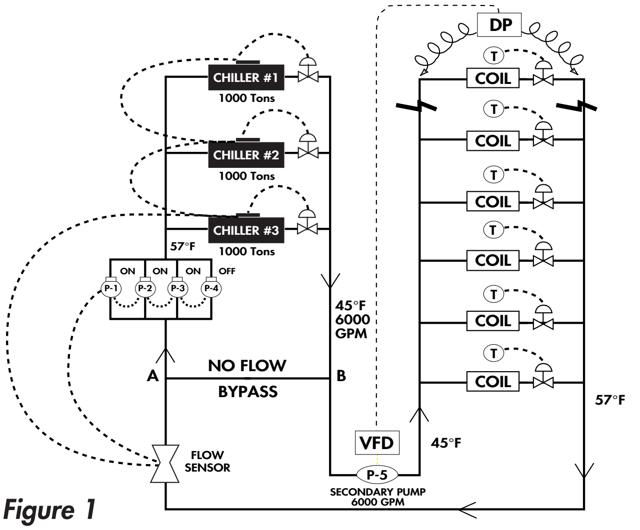

Figure 1

tons. 4,000 GPM is circulated in the primary, 1,600 GPM through the bypass and 2,400 GPM (600 tons x 4 GPM/ton) in the secondary. If the system were properly staged and piped one chiller could easily handle the 600 ton load.

In Figure 3 the load has increased to 1,500 tons. 6,000 GPM is circulating in the primary and secondary systems with no flow in the bypass. Three inefficiently loaded chillers and three pumps are on line. Two chillers could handle the load.

To satisfy the 1,600 ton load in Figure 4, 6,400 GPM (1,600 x 4 GPM/ton) is required. The flow in the secondary now exceeds the primary flow of 6,000 GPM, and return water through the bypass starts to mix with the supply water. This raises the supply water temperature and the air handling unit control valves start to open, thus increasing both the secondary and bypass flow. Temperature control in the conditioned spaces is lost, even though only about one-half of the central plant capacity is utilized. When chillers are staged from flow and are controlled at the design leaving chilled water temperature, the approximate maximum system capacity is:

(Operating ΔT)x(Central Plant Capacity)÷(Design ΔT)

Very often facilities with operating conditions such as these will add an additional chiller and pump even though the existing plant would have sufficient capacity if the chillers were piped and staged properly. Adding more equipment is not the solution. The secondary pump and piping is not designed for the additional flow.

The capacity of a chiller using hydrocarbon based refrigerants, increases from 1/2 to 1 and 1/2% for every degree decrease in entering condenser water temperature, but the increase in chiller capacity at a lower kW/ton cannot be utilized when chillers are staged from flow. Whenever the flow exceeds the design flow of the chiller in the typical primary/secondary system, the next machine is staged on even though the operating machine(s) are not fully loaded. The maximum capacity of any machine never goes above the 100% capacity abscissa line in Figure 5. Most chillers are rated at 85°F entering condenser water temperature, a condition that may exist less than 2% of the year. At other times whenever the outdoor wet bulb and entering condenser water temperature is less than design, the chiller capacity will increase and the kW/ton will decrease. This higher efficiency and lower kW/ton operating range is in the shaded area above the 100% capacity abscissa line in Figure 5. A high efficiency chiller may operate at 0.55 kW/ton with 85°F condenser water and at 0.34 kW/ton with 55°F condenser water. Most 1,000

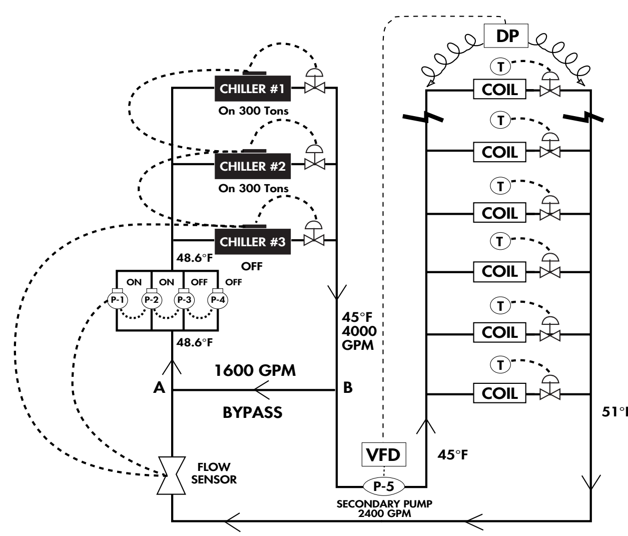

Figure 2

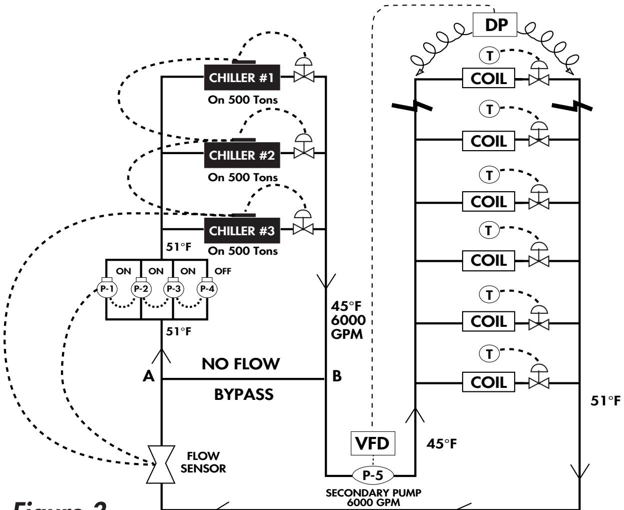

Figure 3

ton chillers may therefore produce over 1,100 tons when the entering condenser water temperature is 65-70°F.

The majority of primary/secondary chilled water plants operate similar to those depicted in Figures 2 and 3; with too many chillers on line for efficient operation. Minimizing the number of chillers on line reduces the plant kW/ton since the unneeded pumps and auxiliary devices are also off-line.

Improving The Efficiency

Although the remedy is relatively simple and inexpensive, operators are hesitant to change, thinking, "That was the way the system was designed to operate." Initially everything possible must be done to correct the issues contributing to the low ΔT. Then change the way the chillers are staged on and off and install a check valve in the bypass line at point "C" as shown in Figure 4 to prevent return water from mixing with supply water. The check valve and bypass line should be sized to handle the minimum flow of the largest chiller. A non-slamming check valve with a manual opening device is generally used. A temperature sensor in the secondary supply water, set low enough to satisfy the load requiring the coldest water, will stage the chillers on. They are staged off by another sensor in the secondary return water and by measuring the load (flow x ΔT). When the load decreases below the capacity of an on-line chiller, that chiller and pump is staged off. The sensors in the supply and return lines must be matched sensors with an accuracy of ±0.10°F. These changes have generally been shown to reduce the kW/ton by 10 to 20%.

Most chillers will operate satisfactorily with evaporator tube water velocities between 3 and 12 feet per second. Under certain conditions (e.g. start-up), the flow in the secondary will exceed the primary design flow and the primary and secondary pumps will then be in series. There will be a slight increase in the flow through the chillers under these conditions but since the flow varies as the square of the head, the actual increase will rarely exceed 10% above design. An increase in flow even this great would require an increase in pump head

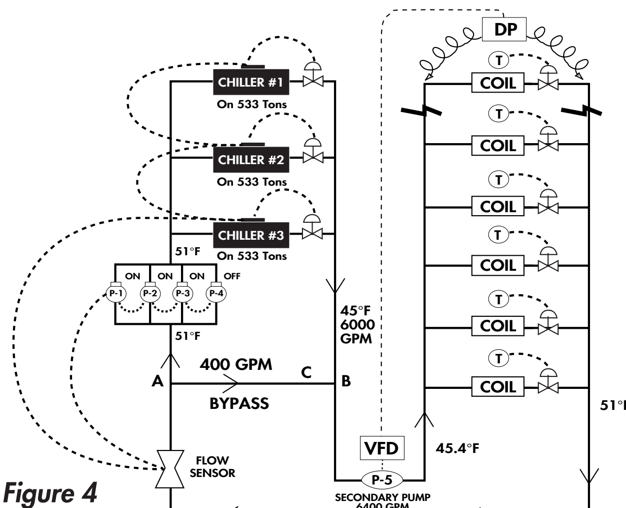

Figure 4

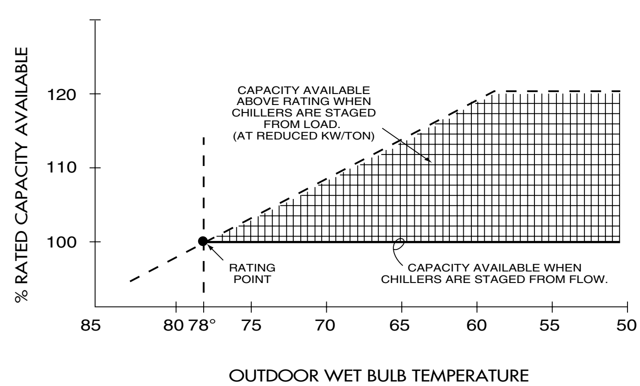

Figure 5

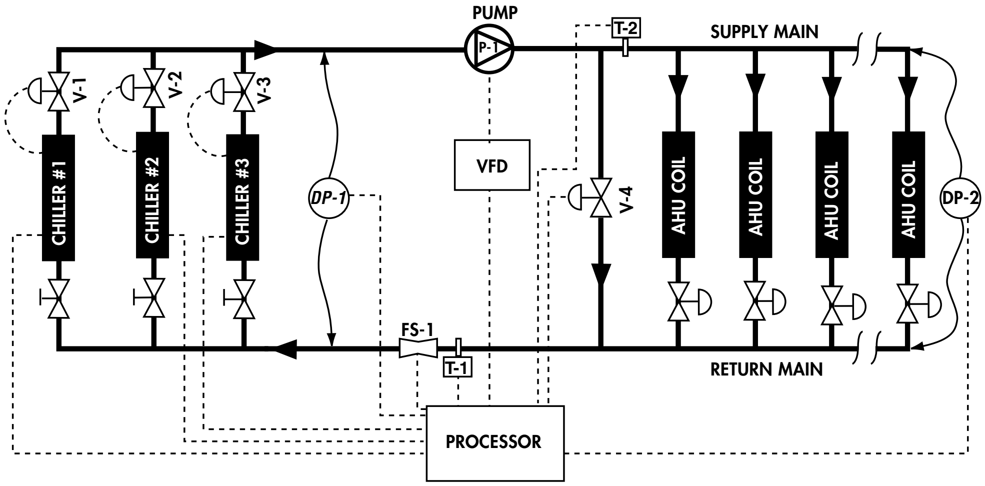

Figure 6

SINGLE PUMP PRIMARY - SECONDARY CHILLED WATER SYSTEM

of over 20%; a condition unlikely to occur with the usual centrifugal pumps that are chosen with flat pump curves.

Single Pump Primary/Secondary System

For new construction, the central plant can be designed using a single pump as shown in Figure 6. Conventional primary pumps are usually large, low speed, low head pumps that require an excessive amount of floor space. Their elimination can materially reduce the initial cost of the system.

The differential pressure sensor “DP-1” controls bypass valve “V-4” to maintain a minimum differential (minimum flow) across the chiller evaporators. When a chiller is staged on and the differential pressure decreases, the processor will modulate valve “V-4” open. When the coil valves or bypass valve “V-4” opens, the differential pressure at sensor “DP-2” will decrease and the processor will ramp up the speed of pump “P-1” through the drive "VFD" to maintain a constant differential across the ends of the supply and return mains. Flow at sensor “FS-1” and temperature at sensors “T-1” and “T-2” are used by the processor to stage the chillers “on” and “off” using the sequence described previously. Valves “V-1”, “V-2”, and “V-3” are closed when their respective chiller is off. Similar controls have been applied for many years to constant flow systems that were converted to variable flow and only one pump was available. Although flow sensor “FS-1” is shown, the differential pressure measured by “DP-1” across the chiller evaporators can be used to calculate the flow. The accuracy of this method of calculating flow increases in proportion to the number of chillers in the central plant. Each evaporator circuit acts as an orifice. As long as the system is closed to the atmosphere there will not be a significant change in pressure drop through the evaporator tubes as the system ages. On plants with three or more chillers, the accuracy approaches that of most flow sensors. The bypass valve “V-4” and bypass line should be sized to handle the minimum flow of the largest chiller. The drop across the bypass and valve must be less than the minimum operating differential (setting of “DP-2”). Valve “V-4” must be coupled to an actuator that can close off against the full head of the secondary system. Correctly sized industrial metal-seated butterfly or ball valves are a good choice if they are selected to withstand the system dynamic forces.

De-Centralized Systems

Outlying buildings on campuses and institutional complexes may have chilled water sources that can be connected to the chilled water system through the supply and return manifold as shown in Figure 7. Building “A” houses the chilled water or ice storage facility; building “B” has an existing chiller and load; buildings “C” and “E” have a load and source respectively; and building “D” houses the central chiller plant. All sources pump out of the return manifold through the cooling

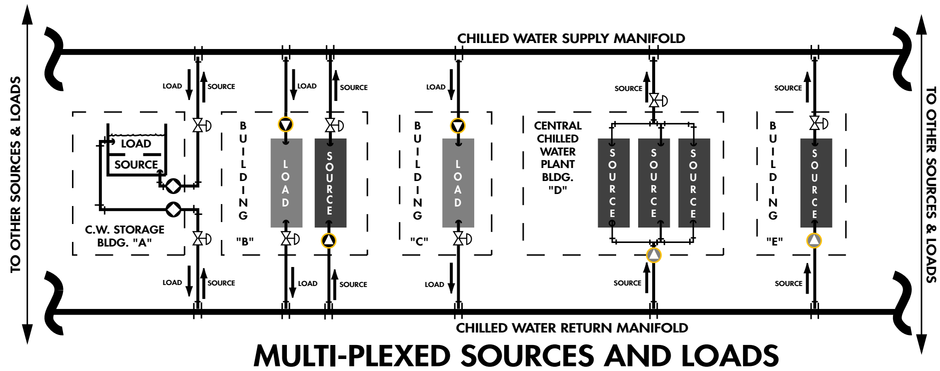

Figure 7

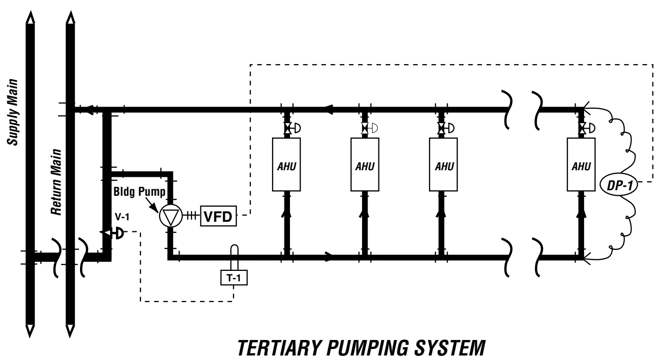

Figure 8

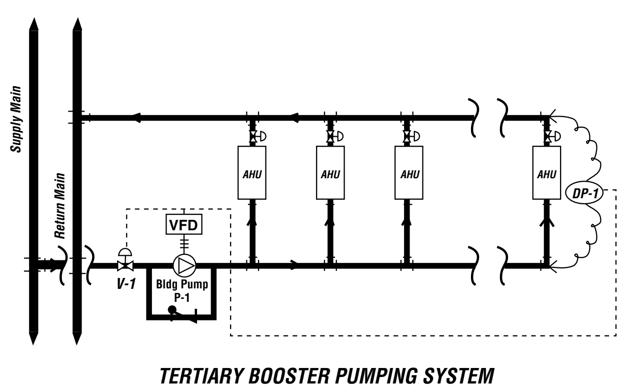

Figure 9

source and into the supply manifold. All loads are pumped out of the supply manifold through the load and into the return manifold. Two position valves "V-1" are used to isolate inactive loads and sources. Multiplexing all of the chilled water sources and loads in this manner has many advantages. Any source or combination of sources can serve any load or combination of loads any place on the manifold. Loads and sources can be matched to provide the most efficient operation. Manifold piping is smaller and less expensive than conventional loop piping. Chiller redundancy is maximized. A detailed method of controlling loads and sources is found in reference #1.

Tertiary Pumping Systems

Tertiary loads on central systems are generally piped in the customary fashion as shown in Figure 8. Temperature sensor "T-1" controls valve "V-1" to maintain the design supply chilled water temperature. Differential pressure sensor "DP-1" controls the speed of the building pump through the variable frequency drive "VFD." This method of piping and control does not take advantage of the differential pressure that is available between the supply and return mains. The pressure is wasted across valve "V-1." A preferred method of piping the tertiary system is shown in Figure 9.

The differential pressure sensor "DP-1" controls valve "V-1" in sequence with the speed of the pump through the variable frequency drive "VFD." As the load in the building increases, valve "V-1" modulates open. On a continued drop in differential pressure, pump "P-1" will start and ramp up to the speed required to maintain the design differential pressure at "DP-1." The cycle reverses on a rise in differential pressure. Utilizing the pressure between the supply and return mains reduces the building pumping energy.

Summary

The utilization of the typical primary/secondary paradigm shown in Figure 1 may be costing building owners millions of dollars in additional operating costs because of the inefficient nature of these systems. There are better ways of designing chilled water systems, and it's time to get our heads out of the 1950's and start using what we've learned since then.

Gil Avery PE

Ref. #1: "Microprocessor control for large chilled water distribution systems."
"October 1987 "Heating/Piping/Air conditioning."

Ref. #2: "Designing and commissioning variable flow systems."
"July 1993 ASHRAE Journal."

Ref. #3: "Controlling chillers in variable flow systems."
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Ref. #4: "Selecting valves and piping coils."
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