Understanding Variable Flow Systems

Part 2 of 3

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Figure 1 shows a typical Primary-constant/ Secondary-variable flow system. This system has been in vogue for many decades, for the following advantages:

1. Total dynamic head gets divided between two independent hydraulic circuits, reducing the head of each pump. Smaller pumps with small motors can be used. Maximum operating pressure of the system reduces, reducing the maximum pressure rating of equipment.
2. Pumping power forms substantial part of HVAC system energy consumption, and as HVAC systems operate at part load most of the time, varying the flow in secondary circuit to match the load results in substantial reduction in pumping power.
3. Constant flow in primary circuit through chillers eliminates the possibility of freeze-up related shut-downs.
4. Chillers can be staged to match load using simple control logic.
5. It is easy to add plant capacity or load to an existing system.

Chiller Staging

HVAC system loads keep changing all the time depending on the weather, occupancy or other similar factors. However, HVAC plants/ chillers have limited range of capacity control, and to overcome this multiple chillers and pumps are used. Chiller capacity control is followed by switching ON/OFF of the chillers, including primary chilled water pumps, condenser water pumps and cooling tower fans, to match the plant capacity to load.

The primary objective of chiller staging is to service the load with minimum number of operative chillers and associated plant, with minimum plant energy consumption. It must always be kept in mind that for proper load

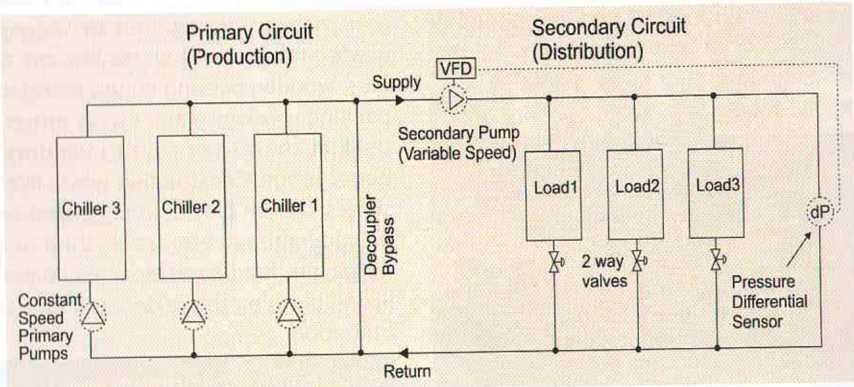

Figure 1: Primary-constant/Secondary-variable flow chilled water system

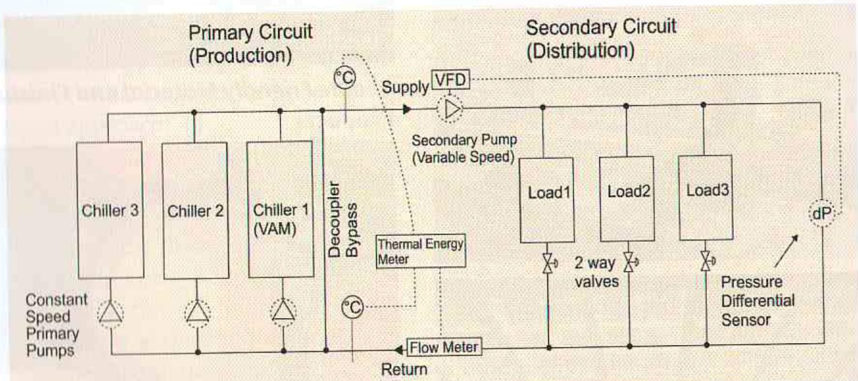

Figure 2: Primary-constant/Secondary-variable flow chilled water system

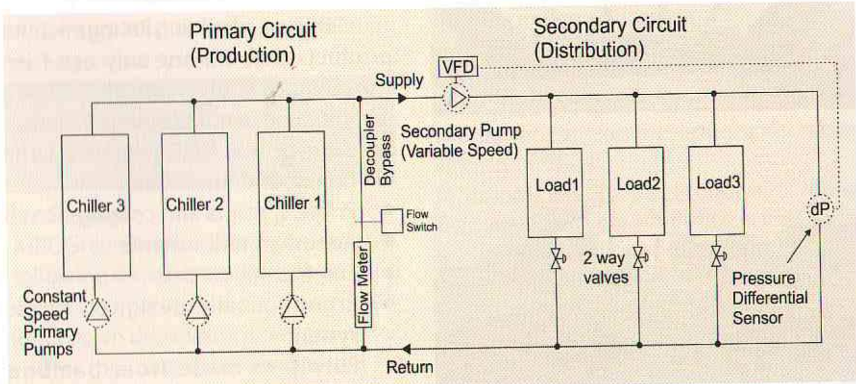

Figure 3: Primary-constant/Secondary-variable flow chilled water system

servicing the plant capacity must always be equal to or more than the load. Two schemes most often used for chiller staging are:

- Measuring total system load using a thermal energy meter
- Sensing flow direction and measuring flow rate in the decoupler bypass.

Figure 2 shows a system with a thermal energy meter connected to a flow meter and two temperature sensors. The flow meter is installed in the return header and measures flow rate in the secondary circuit. Temperature sensors are installed, one in the return header and the other in the supply header, to measure temperatures of supply and return water in the secondary circuit.

Load is calculated using the above data of flow rate and difference in water temperatures in the supply and return headers. A chiller is switched ON whenever the load increases above the operative plant capacity. However, to keep operative plant capacity higher than the load, and also to prevent hunting, a chiller is switched OFF only when the load drops below the operative plant capacity by at least 110% of capacity of a single chiller.

Figure 3 shows a system with a flow switch and a flow meter installed in the decoupler bypass. The flow switch is installed to detect and signal whenever there is reverse flow from return header to supply header, whereas the flow meter is installed to measure the rate of forward flow from supply header to return header.

Flow in the secondary circuit varies as per load, whereas it remains constant in the primary circuit unless a chiller is switched ON or switched OFF. Flow through the decoupler bypass indicates capacity mismatch between the load and the operative plant capacity. Forward flow in the decoupler bypass indicates surplus, whereas reverse flow indicates deficient operative plant capacity.

A chiller is switched ON whenever the flow switch installed in the decoupler bypass senses reverse flow. However, to keep plant capacity higher than the load and also to prevent hunting, a chiller is switched OFF only when the flow meter senses forward flow rate of at least 110% of flow rate through a single chiller.

Effect of Low ΔT on Chiller Staging

Both the above chiller staging schemes are based on the premise that the actual ΔT matches the design ΔT, which is seldom true, for reasons to be discussed later. Low ΔT is a major disadvantage of Primary-constant/ Secondary-variable flow systems; it disturbs automatic chiller staging and results in inefficient operation of plant.

From the heat transfer equation, heat transfer rate (H) is proportional to the product of water flow rate (Q) and the temperature difference of entering and leaving water (ΔT).

$$H \alpha Q \times \Delta T$$

Flow rate in the primary circuit is constant based on the number of operative chillers, and therefore, as per the above equation, any change in ΔT will change the chiller capacity proportionately. For a 100 TR chiller designed to operate at 5°C ΔT,

$$\text{Water flow rate} = \frac{100\text{ TR} \times 3000\text{ Kcal/hr}}{5^\circ\text{C}} = 60{,}000\text{ LPH.}$$

If ΔT drops to 4°C,

$$\text{Effective chiller capacity} = \frac{60{,}000\text{ LPH} \times 4^\circ\text{C}}{3000\text{ Kcal/hr}} = 80\text{ TR.}$$

On the other hand, flow rate in the secondary circuit is variable. In case of lower than design ΔT, AHU controls would force the control valve to open more and allow more flow than required to satisfy the load at the design ΔT. For a 100 TR AHU designed to operate at 5°C ΔT,

$$\text{Water flow rate} = \frac{100\text{ TR} \times 3000\text{ Kcal/hr}}{5^\circ\text{C}} = 60{,}000\text{ LPH.}$$

If ΔT drops to 4°C,

$$\text{Water flow rate} = \frac{100\text{ TR} \times 3000\text{ Kcal/hr}}{4^\circ\text{C}} = 75{,}000\text{ LPH.}$$

For better understanding, consider a system with 3 x 100 TR chillers with design ΔT of 5°C and instantaneous load of 180 TR operating at 4°C ΔT; then

$$\text{Flow rate in secondary circuit} = \frac{180\text{ TR} \times 3000\text{ Kcal/hr}}{4^\circ\text{C}} = 135{,}000\text{ LPH}$$

If chiller staging is carried out using thermal energy measurement,

Thermal energy meter would sense a load =

$$\frac{135{,}000\text{ LPH} \times 4^\circ\text{C}}{3000\text{ Kcal/Hr}} = 180\text{ TR,}$$

and would operate 2 chillers of 100 TR each with

Total flow rate of 2 x 60,000 LPH = 120,000 LPH, and

$$\text{Effective capacity of } \frac{2 \times 100\text{ TR} \times 4^\circ\text{C}}{5^\circ\text{C}} = 160\text{ TR}$$

As is evident from the above, under such conditions the plant with an effective capacity of 160 TR and flow rate of 120,000 LPH would be able to meet neither the load requirement of 180 TR nor the flow requirement of 135,000 LPH of the secondary circuit. Reverse flow will start in the bypass header; return water would mix with the supply water from the plant, increasing the supply water temperature in the secondary circuit. Increased supply water temperature could affect the coil efficiency resulting in user complaints of poor cooling and/or poor dehumidification, and manual intervention may be required to start another chiller to increase effective plant capacity as well as flow rate in the primary circuit.

In case of chiller staging based on flow direction and rate in the decoupler bypass, the increased flow rate of 135,000 LPH in the secondary circuit due to the reduced ΔT of 4°C will establish reverse flow in the bypass header. The flow switch will sense it and automatically start another chiller.

Although starting an additional chiller does solve the problem, it increases the energy consumption per TR of load. Increased flow rate in the secondary circuit increases the secondary pumping power; operation of additional chiller along with its primary chilled water pump, condenser water pump and cooling tower also adds to power wastage. ❖

The concluding part of this article will appear in the May-June 2014 issue of the Journal.