Converting From Primary/Secondary to

# All-Variable Flow

## Reducing chilled-water-plant energy consumption by 15 to 20 percent or more

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The control of all-variable-flow chilled-water systems is thought to be so complicated and, thus, difficult for operators to understand that many engineers and designers resist converting their primary/secondary systems, even though the return on investment of such a conversion is measured in weeks, rather than years, with chilled-water-plant energy consumption reduced by at least 15 percent and often by 20 percent or more.

The performance of a central chilled-water plant was documented for two years prior to and for four years following the plant's conversion to all-variable flow.[1] Following the conversion, energy consumption was reduced by 20 percent because the chillers were able to operate in the "max-cap" range. For chillers with water-cooled condensers and cooling towers, operation in the max-cap range is possible whenever the outdoor wet-bulb temperature is below design; for air-cooled chillers, it is possible whenever the outdoor dry-bulb temperature is below design. (Typically, max-cap operation is possible 98 percent of the time.) Chillers with halocarbon compressors often can produce 10- to 40-percent more cooling capacity than their nominal rating, with lower condensing temperatures (Figure 1).

Operation in the max-cap range is a win-win proposition: Cooling capacity goes up, and kilowatts per ton goes down. For the additional capacity to be utilized, the flow and/or the delta-T between the supply and return chilled water must increase. In primary/secondary systems, the flow through chillers generally is constant; therefore, the chillers cannot realize much of the benefit of operating in the max-cap range. In all-variable-flow systems, the control and staging of chillers is much simpler. Measuring or determining the direction of flow in the crossover is not necessary, and the flow through the online chiller(s) always is equal to or greater than design flow.

Following are the steps necessary to convert from primary/secondary to all-variable flow:

1) Replace all air-handling-unit (AHU) three-way valves and/or two-way modulating valves that do not close tight. Replacement valves must close tight at one-and-a-half times the total system head. The maximum differential the valves' wetted parts will withstand must be less than one-and-a-half times the pump head. All modulating valves must have a rangeability of at least 200-to-1 and meet the American National Standards Institute (ANSI) Class VI leakage rating. Converting to all-variable flow places chiller pumps

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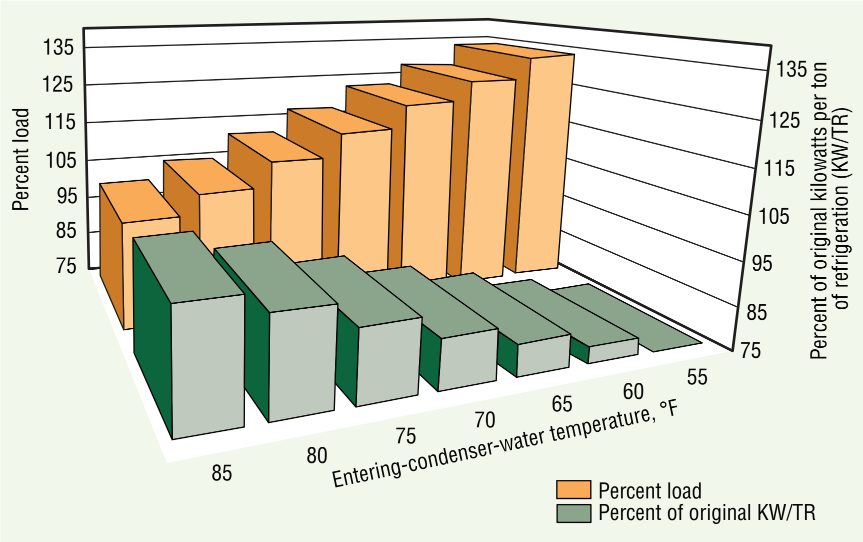


Courtesy of the York division of Johnson Controls

***FIGURE 1. Chiller maximum capacities and kilowatts per ton of refrigeration at reduced entering-condenser-water temperature.***

in series with variable-speed air-handler (secondary) pumps; therefore, the differential pressure across air-handler valves will be greater than in a primary/secondary system, where differential pressure across valves is determined by the secondary pump only.

2) Remove all balancing valves that do not serve as isolation or service valves. Those left in for service should be opened fully. Close or remove all bypasses between chilled-water supply and return lines. Fully open all triple-duty valves.

3) Close control valves when the AHU they serve is off, unless water is circulated through coils during cold weather to prevent freezing when the chillers are off.

4) Interlock primary pumps to run with the chiller they serve. These pumps can be manifolded together, as shown in Figure 2, or each pump can be piped to a chiller. When operating in the max-cap range, chiller pumps are in series with air-handler pumps, and flow through the chiller pumps may exceed the design flow of a primary/secondary system.

5) Control the variable-frequency drive on the air-handler pump from a differential-pressure sensor (DP-1 in Figure 2) across the branch serving the most remote AHU. If the air-handler-valve positions are known, reset DP-1 from the most demanding AHU valve. This will keep pump head as low as possible.

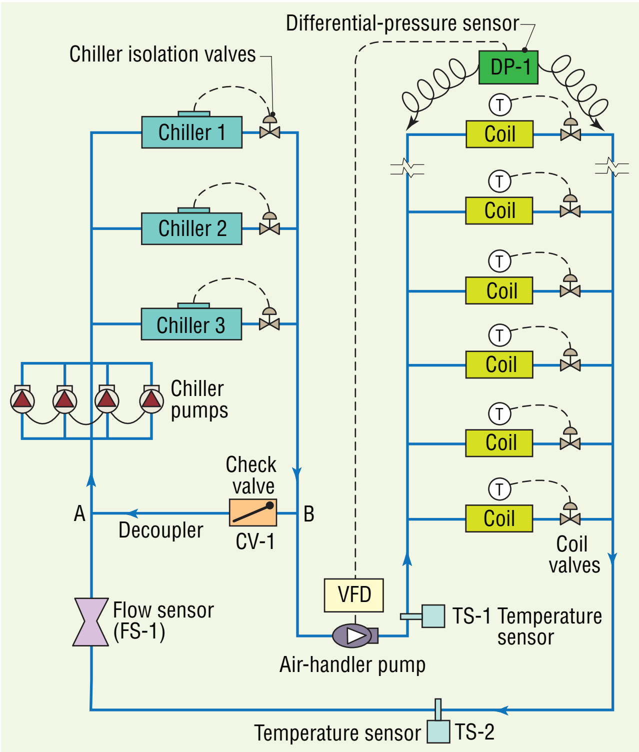


***FIGURE 2. Typical all-variable-flow system.***

6) Install a check valve (CV-1 in Figure 2) in the decoupler connecting the chilled-water supply and return mains (points A to B in Figure 2) to prevent return water from mixing with supply water. The check valve may be line size and have a manual or automatic opening mechanism so it can be opened and water circulated to the AHU during cold weather, when the chillers are off. A modulating two-way control valve may be used instead of a check valve. Control this valve from a differential-pressure sensor across the chiller evaporators. The valve must be sized for the minimum flow of the largest chiller, a close-off rating at least as high as the air-handler-pump cut-off head, and a pressure drop at or less than the setting of DP-1. The differential pressure across points A to B never will be less than the setting of DP-1. When a check valve is used in the decoupler, flow through the online chiller never will be less than design. When a control valve is used in the decoupler, flow always will be at or above the minimum flow required for the online chiller.

7) Install a two-position chilled-water isolation valve in the branch to each chiller. These valves may be line-size, metal-seated butterfly valves, unless they also will be used for servicing, in which case tight shut-off valves must be used.

8) Stage "on" the chillers from a temperature sensor (TS-1 in Figure 2), with the set point at the design air-handler water-supply temperature. This enables the chillers to operate in the max-cap variable-flow range to produce their maximum capacity, depending on the

entering-condenser-water temperature. The online chiller(s) will load fully (above rated capacity whenever the outdoor wet-bulb temperature is below design) before the next chiller is brought online. Table 1 shows the typical performance of a 1,000-ton chiller as evaporator flow increases from 2,000 to 2,464 gpm and condenser-water-supply temperature decreases from 85°F to 55°F. The increase in evaporator-pump water-transport energy is added to the chiller kilowatts to arrive at the total and net kilowatts per ton. The net kilowatts per ton decreases from 0.58 to 0.48, while the cooling capacity increases from 1,000 tons to 1,233 tons (23.3 percent).

9) For each chiller, set the control point of the supply-water temperature sensor 3°F below design. For example, if the design is 44°F, set the temperature sensor at 41°F. This will enable chillers to operate in the more efficient max-cap variable-temperature range. Table 2 shows the performance of a typical 1,000-ton chiller as delta-T increases from 12°F to 14°F. The kilowatts per ton decreases from 0.58 to 0.49, while the cooling capacity increases from 1,000 tons to 1,166 tons (11.6 percent).

10) Be aware that because chilled-water flow and delta-T vary, actual kilowatts per ton may be less than that shown in tables 1 and 2.

11) To stage off a chiller, close the isolation valve and stop the pump when the building load drops far enough below the capacity of the chiller to avoid cycling the chiller back on too quickly. Load is proportional to flow times delta-T. Flow can be measured with FS-1 in Figure 2, while delta-T can be measured with TS-1 and TS-2. Flow also can be measured with a differential-pressure sensor across the chiller evaporator, if dependable flow characteristics are known.

12) Vary the speed of and cycle the cooling-tower fans (and tower bypass valve, if used) from a temperature sensor in the condenser-water supply.

13) Reset the condenser-water-supply temperature with the outdoor wet-bulb temperature. Do not reset below what is recommended by the chiller manufacturer.

14) Specify equipment interlocks, flow- and temperature-stabilization time delays, chiller-isolation-valve timing, condenser-water controls, etc. as recommended by the chiller manufacturer(s) to complete the sequence of control.

| CDS, °F | Total kw | Tons | CWR, °F | CWS, °F | Delta-T | Kw per ton |
|---|---|---|---|---|---|---|
| 85 | 580.0 | 1,000 | 56.0 | 44.0 | 12.0 | 0.58 |
| 80 | 592.9 | 1,078 | 56.4 | 43.5 | 12.9 | 0.55 |
| 75 | 587.8 | 1,109 | 56.3 | 43.0 | 13.3 | 0.53 |
| 70 | 585.0 | 1,125 | 56.0 | 42.5 | 13.5 | 0.52 |
| 65 | 569.5 | 1,139 | 55.7 | 42.0 | 13.7 | 0.50 |
| 60 | 565.0 | 1,153 | 55.3 | 41.5 | 13.8 | 0.49 |
| 55 | 571.3 | 1,166 | 55.0 | 41.0 | 14.0 | 0.49 |
| CDS = condenser-water supply; CWR = chilled-water return; CWS = chilled-water supply | | | | | | |

***TABLE 2. Typical performance of a 1,000-ton chiller operating in the max-cap range with a variable evaporator delta-T and a constant evaporator flow of 2,000 gpm.***

| CDS, °F | Tons | CW gpm | Evaporator PD (FT) | Increase in EPTE | Total kw | Net kw/ton |
|---|---|---|---|---|---|---|
| 85 | 1,000 | 2,000 | 20.6 | — | 580.0 | 0.58 |
| 80 | 1,090 | 2,178 | 24.0 | 2.1 | 601.6 | 0.55 |
| 75 | 1,130 | 2,258 | 25.7 | 3.2 | 602.1 | 0.53 |
| 70 | 1,156 | 2,310 | 26.8 | 3.9 | 605.0 | 0.52 |
| 65 | 1,182 | 2,362 | 27.9 | 4.6 | 595.6 | 0.50 |
| 60 | 1,207 | 2,412 | 29.0 | 5.3 | 596.7 | 0.49 |
| 55 | 1,233 | 2,464 | 30.0 | 6.0 | 597.8 | 0.48 |
| CDS = condenser-water supply; CW = chilled water; EPTE = evaporator-pump transport energy (kw) | | | | | | |

***TABLE 1. Typical performance of a 1,000-ton chiller operating in the max-cap range with a constant evaporator delta-T of 12°F and an increasing evaporator flow.***

**REFERENCE**

1) Avery, G. (1998, February). Controlling chillers in variable flow systems. *ASHRAE Journal*, pp. 42-45.

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