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SOLUTIONS

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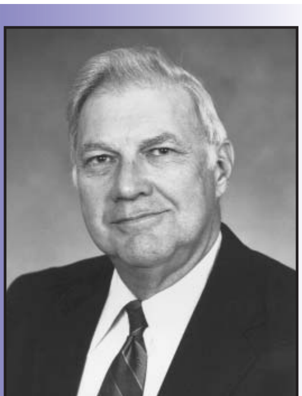
Controlling Chillers in Variable Flow Systems



Article by Gil Avery, P.E.

"This article presents a case study of an actual system that was not optimized to peak efficiency and discusses the changes made to improve its performance."

Since the evolution of primary-secondary pumping arrangements over 20 years ago, multiple chillers have been controlled by measuring flow in the secondary system and then operating the appropriate chiller(s) to handle the flow. This control method is flawed in that a variable flow chilled water system is rarely a constant ΔT system. Measuring only the flow does not provide an accurate indication of the load. As a result, most multiple chiller systems controlled in this manner do not operate at optimum efficiency and suffer from low ΔT.

This article presents a case study of an actual system that was not optimized to peak efficiency and discusses the changes made to improve its performance. These changes can be applied to other chilled water systems with multiple chillers.

The System

The cooling plant for Allied Signal, built in 1983 in Rocky Mount, N.C., includes two 700-ton (2462 kW) and two 350-ton (1231 kW) centrifugal chillers that serve the HVAC system and meet the cooling requirements for the industrial production heat exchangers. The facility produces and refurbishes controls and accessories for aircraft jet engines. The chilled water system utilizes primary chiller pumps and secondary load pumps similar to the system shown in *Figure 1*.

The author visited the plant in September 1992 to discuss methods of improving the system efficiency and optimizing chiller performance. The production heat exchangers required cooling water at a constant 45°F (7°C). However, the supply temperature fluctuated between 44°F and 52°F (6.6°C and 11°C) and could not be stabilized because of the ever-changing flow of return water from the bypass line that mixed with the supply water from the chillers. The low ΔT between the chilled water supply and returns caused some chillers to operate inefficiently at very light loads, because the chillers needed to be on-line to handle the flow instead of the load.

Installing the Check Valve

The recommendation was made that a valve should be installed in the bypass line (cross-over bridge) so that return water could not bypass the chillers. This change was made during the holiday shutdown at the end of 1992. The check valve is shown in the crossover in *Figure 1*.

The Results

Figure 2 charts the total annual kWh energy consumption of the individually metered chillers, for the years 1991 through 1996. The energy consumed by the chillers for the years 1993 through 1996, after the valve was installed, decreased an average of 20% per year. The average annual chiller

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utilization (a 350-ton [1231 kW] chiller operating one hour equals 350 utilization hours) decreased 28%, as shown in *Figure 3*. During these four years, the cooling load increased as production increased, and an additional secondary chilled water pump was added to serve new production equipment.

The reduced chiller utilization time verifies that the chiller efficiency increased because the chiller kWh consumption decreased as the cooling load was increasing. The estimated average annual savings was over $28,000 per year as detailed in *Figure 4*.

Because return water could not bypass the chillers after the bypass line valve was installed, the secondary supply water temperature stabilized at 45°F (7°C).

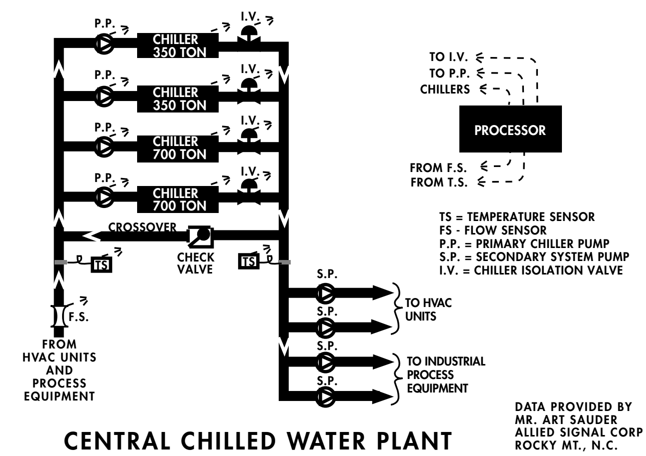


Fig. 1: Central chilled water plant

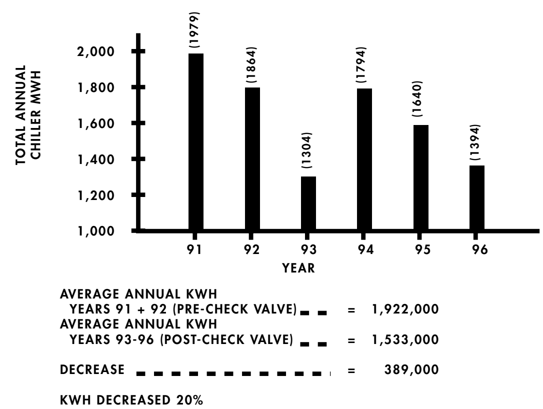


Fig. 2: Annual chiller MWH.

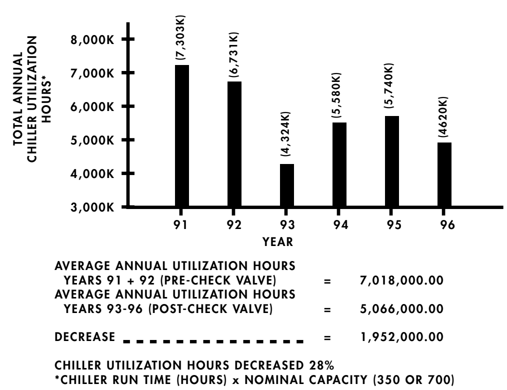


Fig. 3: Annual chiller utilization.

What May Reduce ΔT

The load on this system was sensed by flow (as it is on most primary/secondary systems). The logic behind this assumption is that the ΔT (generally 10°F to 16°F [5.55°C to 8.8°C]) is constant across the coils on a variable flow system. As long as the ΔT is constant, the chiller load is directly proportional to flow.

Factors that may reduce the ΔT and chiller efficiency include:

- The coils fouling on the air and water side enough to reduce the heat transfer. The chart in *Figure 5* is an approximation of entering water temperatures and flows required to maintain the same coil capacity (200,000 Btu/h [58 620 W]) as the coil fouling factor drops from 1.0 to 0.8. The figure depicts the typical degradation in a variable flow primary-secondary system as the system ages.

The physical size of the coil is the same for all five conditions. The second column is the "new" coil capacity if a new coil were operating with the reduced entering water temperature and increased gallons-per-minute of the "old" coil. (A new coil operating at an entering water temperature of 41.2°F (5°C) and with a flow of 26.0 gpm (1.64 L/s), [line 3] would have a capacity of 222,700 Btu/h (65 273 W). Note that over 6°F cooler water is needed when the fouling factor is 0.8. This is a large part of the increase in chiller energy which is experienced as the machine loses efficiency at cooler supply water temperatures.

- The airflow decreases as the filters and other air handling components become dirty.
- The chiller cooling capacity changes as the condenser water temperature changes from season to season.
- The thermostats controlling the chilled water valves may have throttling ranges that are too wide and thus permit the operating point to drift below design.
- A mix of two or three-way valves may exist in the system or a constant bypass may be used to maintain a minimum system flow.
- As cooling problems occur, maintenance personnel tend to lower supply air temperature set points.
- At reduced loads, chilled water flow may become laminar and coil heat transfer is impaired.
- The chilled water control valve seats may erode and leak or the valves may not control properly, because the valve actuators or valve bodies were not selected to endure the dynamic forces and close off pressures present in variable flow systems.

As the ΔT decreases, the secondary flow must increase and/or the supply water temperature must decrease. For example, if the ΔT is only at 50% of the design ΔT, then the coil flow must double or the inlet water temperature must decrease to handle the load. Because the supply water temperature from the chillers is generally fixed, the flow must increase. Eventually, the system flow requirements increase beyond the flow capacity of the online chillers. Typically, the flow capacity of the chillers is exceeded well before their cooling capacity is exceeded.

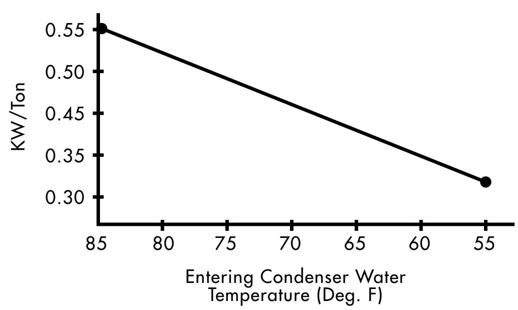
CHILLERS - 389,000 KWH x $.064	=	$24,896.00
AUXILIARIES (Pump, Towers, ETC) 15%	=	3,734.00
TOTAL ANNUAL SAVINGS	=	$28,630.00
COST TO INSTALL BYPASS VALVE	=	$1,000.00
RETURN ON INVESTMENT	=	2,860%
PAY BACK	=	.035 YEARS
	=	13 DAYS

ANNUAL SAVINGS DOES NOT INCLUDE ANY ALLOWANCE FOR REDUCTION IN MAINTENANCE, DEMAND CHARGES OR INCREASE IN EQUIPMENT LIFE.

Fig. 4: Annual savings.

COIL FOULING FACTOR	OPERATING CONDITION	NEW COIL CAPACITY B.T.U.	FOULED COIL CAPACITY B.T.U.	ΔT °F	EWT °F	LWT °F	GPM COIL FLOW
1.00	1	200,000	200,000	16.0	44.0	60.0	25.0
.95	2	210,600	200,000	15.7	42.7	58.4	25.5
.90	3	222,700	200,000	15.4	41.2	56.6	26.0
.85	4	234,900	200,000	15.1	39.9	54.8	26.5
.80	5	249,900	200,000	14.8	37.8	52.6	27.0

Fig. 5: The typical degradation in a variable flow primary-secondary system as the system ages.



Algorithm
KW/Ton = 0.34 + (0.007 x [EWT-55])

Fig. 6: Algorithm: kW/ton = 0.34 + (0.007 x [EWT-55])

When this happens, return water through the bypass starts to mix with the water leaving the chillers, thus raising the supply water temperature. The increased water temperature yields an increase in the flow requirement which further aggravates the problem.

Often, the operator will try to reduce the secondary supply water temperature by lowering the leaving water temperature of the active chiller(s) to compensate for the mixing effect. This makes it difficult to optimize chillers in a primary-secondary system without a bypass check valve and explains why so many field complaints about "inefficient chiller operation" and "too many chillers on-line" are made.

Many piping and control arrangements have been tried to improve the efficiency of chillers. Some of these are:
- Preferential loading with a backside bypass.
- Variable speed drives on the primary chiller pumps.
- Sequencing chillers based upon the direction of flow in the by-pass line, by the temperature of the supply chilled water or by the temperature of the return chilled water.

Most of these arrangements fail because they do not measure the correct load and because they do not prevent flow from the secondary return into the secondary supply line through the bypass. A check valve in the bypass line will prevent mixing return water with supply water and will insure the required flow through each on-line chiller whenever the primary flow exceeds the secondary flow. The check valve should be of the non-slamming type, and the bypass line and valve should be sized for the flow through the largest chiller.

When the secondary flow exceeds the primary chiller flow, the secondary and primary pumps will be in series. This may increase the flow through the on-line chillers by a small amount. Since the pump head is proportional to the flow rate squared, an increase in flow of 10% would require over a 20% increase in pump head. Even with the pumps in series, it is unlikely that the pumps would be able to generate this much additional head. Consequently, the additional flow created by the series pump arrangement is limited and is not an important factor. The chiller isolation valves in *Figure 1* will prevent return water through the inactive chiller(s) from mixing with cold supply water.

Summary

Load is proportional to flow times ΔT. Temperature sensors in the secondary supply and return chilled water lines and a flow sensor in one of these lines will provide an accurate indication of load. By determining the load, the correct number of chillers can be operated. As the measured load exceeds the calculated capacity of the on-line chiller(s), the next chiller can be started. The set point of the supply water from the chillers must be low enough to satisfy the load that requires the coldest water.

The capacity of a chiller changes with the temperature of the entering condenser water. The curve in *Figure 6* is a plot of entering condenser water temperature and kW/ton for a typical centrifugal chiller. This curve only applies to a particular make and model of chiller. The actual performance of the installed machine should be used to develop the kW/ton algorithm that can be entered into the software of the DDC system controlling the chillers. Additionally, the software should account for any penalty for increasing cooling tower fan kW when lowering the entering condenser water temperature. Under certain load and ambient conditions, this cost can outweigh the savings in chiller efficiency. Careful programming can achieve an optimum system efficiency.

The following example shows how chiller usage and efficiency are impacted by three different control strategies.
- 3,000-ton (10 551 kW) system with three centrifugals.
- Nominal capacity = 1,000 tons (3517 kW) each at 85°F (29°C) entering condenser water temperature.

• Maximum capacity = 1,250 tons (4396 kW) each at 55°F (12°C) entering condenser water temperature.
• Design supply water temperature = 45°F (7°C).
• Design return water temperature = 55°F (12°C).
• Typical operating conditions: (5°F ΔT [2.75°C ΔT]).
• Supply water temperature = 45°F (7°C).
• Return water temperature = 50°F (10°C).
• Load = 1,200 tons (4220 kW).
• Flow = 1,200 tons x 4.8 gpm/ton = 5,760 gpm (4220 kW x 0.085 L/s per kW = 360 L/s).

1. When control is from secondary flow without a bypass check valve, three chillers will be on line since the system has been programmed for a maximum of 2,400 gpm (151 L/s) per chiller.
2. When control is from measured load (flow x ΔT) with a bypass check valve, two chillers will be on line since the system has been programmed for a maximum capacity of 1,000 tons (3517 kW) per chiller and the load is 1,200 tons (4220 kW).
3. When control is from chiller capacity (1,250 tons (4396 kW) per machine at 55°F (12°C) entering condenser water temperature) with a bypass check valve, one chiller will be on line since the load is only 1,200 tons (4396 kW).

When strategies 2 and 3 are used, the flow and the leaving water temperature will decrease so that the flow times ΔT will equal the cooling load.

Generally, it is more efficient to operate the minimum number of chillers that will handle the load, even though the part load chiller efficiency may increase by using more chillers. The energy required for the additional pumps and auxiliary devices generally offsets any gain in efficiency. Therefore, matching chiller capacity to the load and eliminating the mixing of return and supply water at the supply side saves money.

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