Design Issues of Variable Chilled-Water Flow Through Chillers

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ABSTRACT

Variable-speed alternating current (AC) drive technologies are of particular interestfor heating, ventilating, and air-conditioning (HVAC) designs because controlling motor speed with variable-frequency AC drives to achieveflow modulation provides an opportunity to capture exceptionally high part-load operating efficiencies. Since HVAC systems spend long hours operating at part-load conditions, improvement in part-load efficiency results in substantial energy savings. Applying variable flow to chilled-water systems is particularly attractive because chilled-water pumping has two associated power costs, directly as pumping power and also as a load on the chiller plant. The small temperature differentials associated with chilled-water systems mean load-side stratification is generally not a concern. but the small temperature differentials do raise concerns about heat transfer at reduced flows.

Presently, typical variable-flow chilled-water systems are designed with two chilled-water circuits: primary and secondary. The primary circuit is usually a low-head circuit that maintains a constant chilled-water flow through the chiller, while the secondary chilled-water pump(s) provide variable flow to the loads based on their demand for cooling. Because the primary circuit is low head and requires relatively low power, it is often reasoned that two-circuit configurations involve only a small pumping energy penalty and only when operating at low loads. However, closer analysis uncovers the following true penalties..

- *a first-cost penalty for employing two separate pumps,*
- *a part-load chiller efficiency penalty from mixing bypassed supply chilled water with the return chilled water, and*
- *a chiller capacity penalty of underutilizing the full chiller capacity during high cooling demands at conditions not precisely congruent to the design peak conditions.*

For many building cooling applications, it is possible to design a chilled-water supply and distribution system with only a single variable-flow circuit. Such designs can avoid the problems listed here. However; there are potential pitfalls that must be considered before such a system can be successful. This ***paper*** *discusses the benefits and problems associated with a single-circuit variable-chilled-water-ji'ow system and offers a chiller plant control strategy that can provide safe, stable, and reliable chiller operation over the entire operating range employed in typical HVAC applications.*

INTRODUCTION

Traditional chilled-water plant design utilizing variable chilled-water flow involves primary/secondary loops with separate pumps, as shown in Figure 1. Typically, one low-head primary loop pump for each chiller in the primary circuit provides aconstant flow through the chiller, while one or more higher head variable-flow secondary loop pumps modulate to adjust secondary chilled-water flow to meet actual cooling demand. The imbalance in flow between the primary and secondary circuits results in flow through the bypass piping circuit. While this configuration satisfies the objective of maintaining constant chilled-water flow through thechiller, it may not achieve the highest chiller efficiency at part loads and can limit chiller capacity due to the fact that under almost all operating conditions, the flows in the two loops are not equal.

To achieve the full potential of variable chilled-water flow in an environment of integrated HVAC equipment operating under high-performance control strategies, it is necessary to rethink the physical configuration of variable-flow chilled-water systems. Recent work (Hartman 1993) has shown that integrated control strategies can be employed to operate variable-flow chilled-waterdistribution systems at much higher efficiencies by coordinating the pump speed directly to the load demands without employing pressure control. It is prudent also to analyze the

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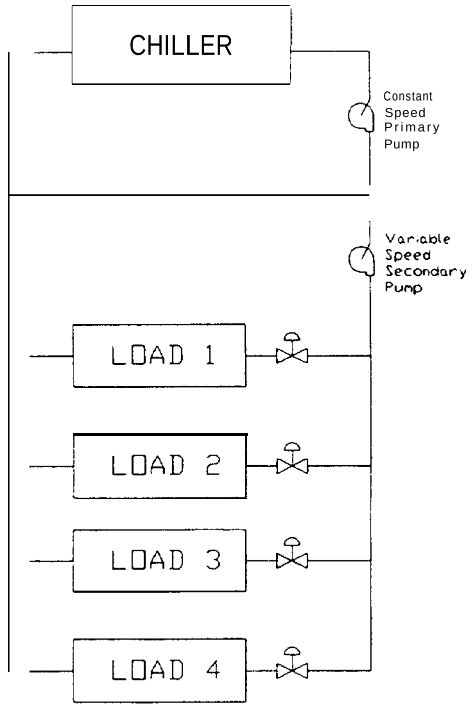

Figure I Primary/secondary chilled-water loop.

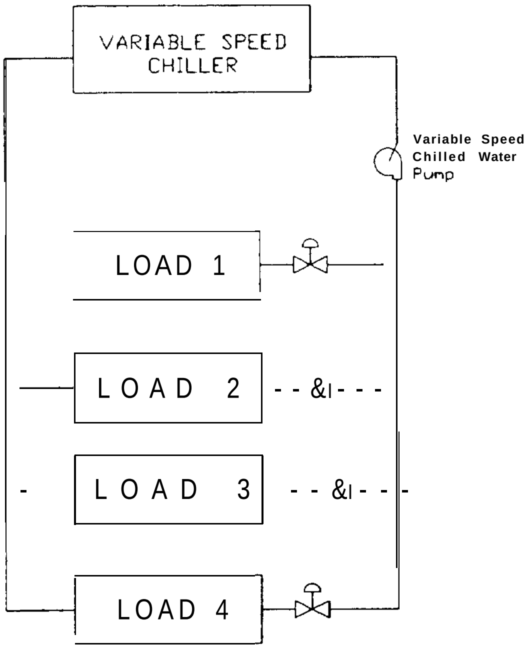

Figure 2 Single-circuit variable-flow flow chilled-water loop.

operation of the primary circuit before a particular configuration and control scheme is adopted.

The striking feature of Figure 1 is the necessity of having two separately powered chilled-water circuits. Designers should ask themselves whether this is really necessary. The constant-flow primary circuit has become accepted design practice because it is well known that below certain velocities of flow through heat exchangers, a switch to laminar flow may cause sudden substantial reductions in heat transfer capacity. The purpose of the primary pump is to ensure such a condition never troubles the system. However, at very low cooling capacity requirements, the heat transfer requirements are also greatly reduced, and by monitoring the chiller load, chilled-water temperature, and refrigerant temperatures, a properly integrated control system can easily adjust the overall system operation if water flow becomes too low to provide efficient heat transfer or may cause the chiller to approach operating limits If the control system is operating with suitable high-performance control algorithms, it can promptly make the necessary corrections to ensure efficient and stable operation of the entire system at all load conditions. With this in mind, consider the simpler piping configuration in Figure 2.

In Figure 2, the chiller itself may be a variable-speed unit, but in any case, it is one that offers a high turndown ratio and an increasing coefficient of performance (COP) as the cooling load is reduced. The required rate of flow through the chiller depends on the cooling load being delivered. This is a good design fit because the loads are connected with two-way valves such that load-side flow also varies with load. In such a scheme, both the chilled-water flow and chiller capacity are adjusted to effectively meet all load conditions. A threshold cooling capacity limit is defined below which the system does not operate, just as is the case with present chiller systems. The potential benefits of a single-circuit variable-flow chiller system as shown in Figure 2 are:

- lower first cost and lower maintenance costs,
- higher overall chiller plant operating efficiencies, and
- greater flexibility in utilizing full chiller capacity at peak conditions.

Before discussing these benefits in detail, let us consider the critical issues of such a design.

SINGLE-CIRCUIT VARIABLE-FLOW SYSTEM CONSIDERATIONS

Configuring a variable-flow chilled-water system as shown in Figure 2 does not mean it will work adequately under all load conditions without specific attention to the chilled-water flow over the wide range of potential operating conditions. To ensure effective and efficient operation of the Figure 2 configuration, several basic requirements must be met. First, the system must not be permitted to operate unless the cooling requirement is above a minimum threshold load. The threshold cooling load requirement is the lowest stable chiller operating load.

Next, the water flow through the chiller evaporator heat exchanger must always be sufficient to maintain evaporator temperature within suitable limits. Typically, the chiller manufacturer recommends a varying evaporator temperature range as a function of the chiller load, and the relationship between chiller efficiency and evaporator temperature is an important consideration as well.

Finally, for the Figure 2 system configuration to be effective, the nature of the loads must be such that chilled-water temperature can rise as the load decreases. Chilled-water systems that require low chilled-water temperature under low-load (low-flow) conditions must be carefully considered before such a configuration is adopted. Examples of systems with such special requirements are those that may be called upon to provide significant dehumidification at low loads or those supplying a variety of loads, a significant number of which may be shut off during peak-load conditions.

In many HVAC applications, these limitations do not pose an absolute barrier to employing the system configuration in Figure 2. However, a complete analysis under the entire variety of operating conditions that could be encountered must be accomplished to be certain the single-circuit scheme will perform satisfactorily under all conditions. In typical North American single-building applications, even though portions of the building may be unoccupied under certain conditions, the single-circuit configuration is usually a good candidate for effective and economical space cooling.

In addition to the load requirements listed above, a single-circuit variable-flow chiller system will be successful only as long as the direct digital control (DDC) system has the capacity to integrate the operation of the chiller(s), pump(s), and the loads **the** system serves with high-performance control algorithms. The start-up sequence for a single-circuit chiller system must provide calculations of present and upcoming cooling load requirements for all loads served. The cooling system is held off until the sum of the calculated load requirements reaches a threshold value that depends in part on anticipated upcoming conditions. Once enabled, high-performance control algorithms must be employed to coordinate the cooling loads, pump speed, and chiller capacity to meet the demand for cooling at the loads and meet the operational constraints of the chiller and other system components.

SINGLE-CIRCUIT VARIABLE-FLOW CHILLED-WATER SYSTEM CONTROL

Even if it were feasible, it is not optimal to operate the chiller in a single-circuit variable-flow system to maintain a constant chilled-water temperature. However, without a clear connection between chiller capacity and chilled-water pump flow, control can become indeterminate and result in erratic operation as the changes in chiller capacity and flow affect each other. One of the great resistances chiller manufacturers have to varying the flow of chilled water through chillers is the difficulty in establishing smooth chiller capacity control under varying flow conditions.

To control the chiller(s) and pump(s) most effectively, some simple mechanism of correlating the operation of the two together is required. Some approaches have been previously discussed (Hartman 1995). One approach that shows a great deal of promise is the use of coordinated chiller and pump (CCP) control.

In coordinated chiller and pump control, both the chilled-water pump and chiller capacity are controlled to react to changes in cooling demand at the loads, so that as loads change, pump speed and chiller capacity are adjusted in unison (percent chiller electric load is set proportionately to the pump motor load). This is a simple and effective way to coordinate the operation of the chiller and pump. The chiller capacity is adjusted in proportion to chilled-water pump power (or to the cube of pump speed). In this scheme, at approximately 93% of the design water flow through the chiller (and the loads), the chiller is operated at 80% of maximum electrical demand (0.93^3). At approximately 60% of the design maximum chilled-water flow, the chiller is operated at approximately 22% of maximum electrical demand (the same percentage of maximum electrical demand as required by the chilled-water pump).

It is important to note that because the COP of the chiller rises as the load decreases, the cooling capacity of the chiller (in most typical circumstances) does not fall by the same amount as its electrical load reduction (which is controlled to adjust chiller capacity). The exact change of chiller capacity with respect to electrical draw depends on the type of chiller (variable-speed or constant-speed) and evaporator and condenser conditions. Furthermore, there may be limits to the capacity adjustment range that depend on current operating conditions. For this reason, a minimum (and in some cases maximum) evaporator and chilled-water temperature algorithm and minimum capacity algorithm operate in parallel with the direct pump control algorithm as a limit to the primary chiller capacity control algorithm.

CCP control does not directly control chilled-water temperature. However, in regions where it may be required, the CCP control strategy may be extended to provide some flexibility in adjusting chilled-water temperatures for more or less dehumidification under part-load conditions. In this way, the chilled-water flow to chiller capacity algorithm can be adjusted slightly, depending on humidity conditions, to provide more or less latent cooling. A sample algorithm in the operators' control language (THC 1988) for a simple CCP control is shown in Figure 3.

BENEFITS OF SINGLE-CIRCUIT VARIABLE-FLOW CHILLED-WATER SYSTEMS

Because traditional design has steadfastly adhered to the concept of constant chilled-water flow through chillers, the industry has never had adequate discussion on the benefits of employing variable-flow schemes. Generally, it has been assumed that the benefits are limited to savings in the cost of the primary pump and a small energy reduction. At part-loads, however, if one focuses on a comparison between the operation of the systems under various conditions, the potential benefits are seen to be far more substantial.

Consider the flow difference between the primary and secondary circuits in a traditional two-circuit system as repre-

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"DETERMINE IF CHILLER PLANT SHOULD RUN"
DOEVERY 1 M
        CLG-DMD-L1 = MAX((AHU1MAT-AHU1SAT-SP)*1.086*AHU1FLOW), 0)
        CLG-DMD-L2 = MAX((AHU2MAT-AHU1SAT-SP)*1.086*AHU2FLOW), 0)
        CLG-DMD-Ln = MAX((AHUnMAT-AHUnSAT-SP)*1.086*AHUnFLOW), 0)
        CLG-DMD = CLG-DMD-L1 + CLG-DMD-L2 + CLG-DMD-L3 + ...
        CLG-DMD = CLG-DMD * (PROJ-HIGH-TEMP / 70)
"START AND STOP CHILLER BASED ON DEMAND
        IF CLG-PLANT OFF-FOR 30 M  AND CLG-DMD > 0.4*CAPACITY THEN
                START CLG-PLANT
        END
        IF CLG-PLANT ON AND CLG-DMD < 0.2* CAPACITY OR PUMP-SPEED < 50
        OR EVAP-TEMP < 35 OR COND-TEMP > 88 THEN
                STOP CLG-PLANT
        END

"WHEN COOLING PLANT IS ON OPERATE PUMP ACCORDING TO VALVE POSITIONS"
        IFONCE CLG-PLANT ON THEN PUMP-SPEED = 75
        IF CLG-PLANT ON THEN
                IF MAX-VALVE-POS > 95 THEN
                        PUMP-SPEED = PUMP-SPEED + 1
                ELSE
                        PUMP-SPEED = PUMP-SPEED - (70 - AVE-VALVE-POS) /10
                END
        ELSE
                PUMP-SPEED = 0
        END
" OPERATE CHILLER CAPACITY IN ACCORDANCE WITH PUMP SPEED
        IF CLG-PLANT ON-FOR < 10 M THEN
                CHLR-DMD = 0.4
        ELSE
                CHLR-DMD = (PUMP-SPEED/100) ^3
                IF EVAP-TEMP < 40 - CHLR-DMD/25 THEN CHLR-DMD = CHLR-DMD_10
                IF COND-TEMP > 80 THEN CHLR-DMD = MAX(CHLR-DMD, 40)
        ENDIF

ENDDO
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CLG-DMD-L1..CLG-DMD-Ln ARE THE CURRENT COOLING DEMANDS FOR EACH OF THE COOLING LOADS SERVED BY THE CHILLER

PROJ-HIGH-TEMP IS THE DAY'S PROJECTED HIGH OUTSIDE TEMPERATURE AS CALCULATED BY THE DDC SYSTEM

Figure 3 *Sample variable-flow single-circuit chilled-water plant control program.*

sented in Figures 4a and 4b. Figure 4a represents flow at low demands for cooling. Under this condition, the flow in the primary loop is substantially higher than that in the secondary. The higher temperature chilled water returning from the loads mixes with bypassed supply chilled water, which reduces the chilled-water inlet temperature, which adversely impacts the overall chiller operating efficiency at part-load conditions. To see how important this part-load energy penalty can be, consider Figure 5. which shows occurrences of various chiller load conditions in typical office buildings in four different regions in North America. Notice that for an overwhelming majority of the time, the chiller plants in all regions operate at low loads. The right-hand portion of Figure 5 shows the annual chiller operating hours as a percentage of total building operating hours. Note from Figure 5 that although chillers in warm climates of North America operate longer hours, chiller plants in all climates operate at roughly the same overall annual load profile. Note also that the North American chiller load profile includes chiller plant operation at less than 60% of design load for more than 70% of the chiller operating hours. The loss of chiller efficiency because of primary/secondary flow differences at part-load conditions can be a substantial penalty in many building HVAC applications.

Now, consider Figure 4b for flow at high loads. For energy efficiency, the flow of the primary circuit is likely to be less than the maximum flow capacity of the chiller. Such a selection reduces primary pump horsepower from two perspectives-lower flow through the primary circuit and also lower pressure drop through the chiller. However, the penalty for less primary pump horsepower is that the secondary flow during peak cooling requirements may exceed the primary loop flow. During these periods, return water from the loads is mixed with chilled-water supply. Such conditions occur frequently in chilled-water-to-air coils because the load conditions (airflow or air psychrometric conditions) do not match the design assumptions precisely.

With the warmer chilled-water supply, the loads may not be satisfied and the chiller may not be capable of operating at full load because it cannot compensate below its minimum chilled-water temperature limit. Such a condition results in chiller underutilization. The cooling capacity of the chiller may be adequate, but the limitations of the primary circuit do not enable the full capacity to be utilized. The only.way to eliminate the possibility of this problem in a primary/secondary chilled-water distribution system is to increase the flow and pressure capacity of the primary pump(s). This then results in an increased energy penalty for the overwhelming majority of the hours the plant operates below peak capacity.

In the single-circuit scheme there is no bypass. All return chilled water enters the chiller without bypassed supply water. Furthermore, the variable-speed pump can be sized for a maximum flow that is somewhat above the design load assumptions. This way the designer is ensured that the full capacity of the chiller can be utilized when peak loads occur at conditions that do not match design conditions exactly.

SUMMARY AND CONCLUSIONS

While there are limitations to the employment of single-circuit variable-flow chilled-water systems for building cooling applications, the opportunities available to designers with the expertise necessary to apply integrated high-performance DDC to these systems make compelling reasons to consider such systems. Single-circuit variable-flow chilled-water systems that are carefully designed and operated with integrated high-performance controls offer the following:

1. Simpler equipment configurations with accompanying first-cost savings that can reduce the system costs or be invested in higher quality, longer lasting components.
2. Lower total system energy use than what is possible with nonintegrated configurations and control strategies.
3. Control precision that is superior to that of nonintegrated traditional control approaches.

To fully exploit the benefits of emerging integration of HVAC equipment with high-performance DDC systems, designers should consider single-circuit variable-flow chilled-water systems. For many typical applications, the benefits can be substantial.

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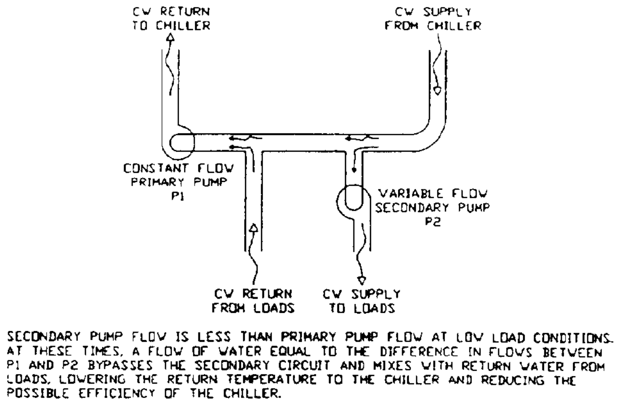

Figure 4a Chiller flow at load conditions.

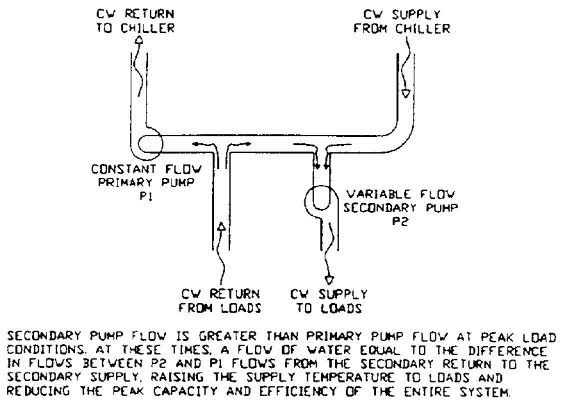

Figure 4b Chiller flow at peak load conditions.

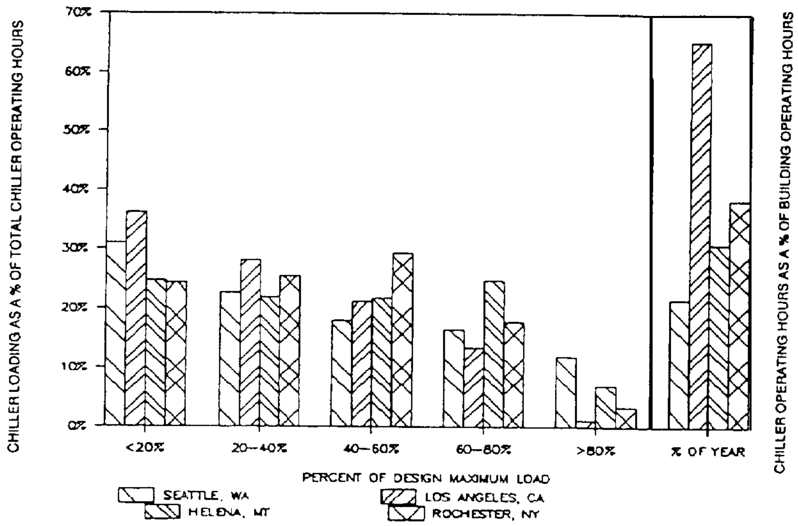

Figure 5 Office building chiller operation