2002

A New Approach To Set Point Control In Chilled Water Loops

L. Lu
Nanyang Technological University

W. Cai
Nanyang Technological University

A NEW APPROACH TO SET POINT CONTROL IN CHILLED WATER LOOPS

*Lu Lu, Research Associate, Nanyang Technological University, Nanyang Avenue,
Singapore 639798; Tel.: (0065) 6790-5472; Fax: (0065) 6790-5471
E-Mail: ellu@ntu.edu.sg *Author for Correspondence

Wenjian Cai, PhD, Associate Professor, Nanyang Technological University, Nanyang Avenue,
Singapore 639798; Tel.: (0065) 6790-6862; Fax: (0065) 6790-5471
E-Mail: ewjcai@ntu.edu.sg

ABSTRACT

This paper presents a new approach to control variables in chilled water loops for energy conservation. Based on the analysis of the relationships between differential pressure and chilled water flow rate in chilled water loop, a series of optimal set points can be found to reset the controlled variables. With these reset set points, chilled water pumps only provide the necessary energy for water distribution loops. An optimization problem accompanied with its constraints is proposed for direct-return chilled water loops. A simple algorithm is provided to find the optimal set points, which can be used in real time to control the variable speed drive pumps to save energy under part-load conditions. With some modification, the proposed approach can be extended to different type of chilled water loops. An application example shows that the chilled water loop operating under the optimal set points has the potential energy savings compared with the conventional method.

NOMENCLATURE

a_0,a_1,a_2: curve fitting constant of pump energy;
C_{pw}: specific heat of water under constant pressure;
H_{BL}: head friction loss of piping branch;
H_P: distribution pump head;
H_{VL}: head friction loss of control valve;
K1: friction factor of cooling coil;
m: the total number of subsystems;
n: the total number of coils;
Q_{coil}: cooling load of coil;
T_{CHWS}: chilled water supply temperature;
η_{pump}: efficiency factor of pump;
$c(H_{VL})$: constraint function of H_{VL};
g_c: a constant;
H_{CL}: head friction loss of cooling coil;
H_{PL}: head friction loss of pipings;
$H_{VL,0}$: pump head ;
$K2$: friction factor of pipes;
m_w: water flow rate;
P_{pump}: power consumption of pump;
T_{CHWR}: chilled water return temperature;
η_{motor}: efficiency factor of motor;
η_{VSD}: efficiency factor of VSD;

Subscript

B: branch;
j: the jth coil/terminal;
m: the measured data;
i: the ith coil/terminal;
$i.j$: the ith coil in the jth subsystem;
des: the designed data;

INTRODUCTION

The chilled water loop is one of important subsystems in a typical central cooling system. It consists of pumps, pipes, valves, and controls (ASHRAE 1999). Chilled water loops transfer the heat from the conditioned areas to central plants. The chilled water from the chillers in the central plant is supplied to coils and terminals of various zones at chilled water supply temperature and water returns at chilled water return temperature. The chilled water pumps provide the energy for water to circulate through the distribution loops. The water flow rate across each cooling coil is controlled by corresponding valve. Our objective is to provide every coils or terminals appropriate chilled water flow matched with their cooling loads by the least power consumption.

Before Variable Speed Drive (VSD) and Direct Digital Control (DDC) techniques existed, the chilled water pump delivered the chilled water to distribution loops at the constant flow rate regardless of the variation of cooling loads. This method not only wasted the pump energy but also resulted in the low ΔT problems. When VSD technique tended to maturity, many experts considered it to improve the controls of chilled water loop. It was believed by Hegberg (1991) that VSD pumping gave systems a high degree of control and potential operating power savings. In Rishel's paper (1991), he gave the objectives of the control of VSD pumps in chilled water systems. However, controlling the chilled water loop only by VSD technique is considered not enough by some other experts. Nowadays, VSD and DDC techniques are widely accepted to be the powerful tools for energy conservation to control chilled water loops in Heating, Ventilating and Air-Conditioning (HVAC) systems. However, some problems still exist in implementation of VSD and DDC techniques. Ahmod (1991) proposed a DDC-based system for a direct-return hydronic system by measuring and calculating water flow rate for each coil. However, it is too complicated for engineering and almost impossible to install flow sensors and differential pressure sensors in each coil in practical. Using valve position to control pump speed through DDC system was proposed by Tillack and Rishel (1998), but the authors only give the general idea and no practical methods for common systems. In 1998, Hartman pointed out that VSD and DDC technologies were overwhelmingly underutilized and fixed differential pressure control could not reduce the pumping energy significantly at part-load conditions. He claimed that DDC network could be employed to operate the pump speed, however he didn't give any practical methods to implemented his ideas. By analysis the traditional system curves, fan laws and pump affinity laws, Bynum and Merwin (1999) concluded that the design system curves could not be used to calculate part-load energy savings with variable flow systems. He provided a control concept of differential pressure reset for a specific application example according to engineering experiences. Because the characteristics of chilled water loops are different from system to system, Harris's control concept was hard to extend to the other systems. ASHRAE (1999) states that the best strategy for a given chilled water set point would be to reset the differential pressure set point in order to maintain all discharge air temperatures with at least one control valve in a fully-open condition. But how to find the fully open valves and implement it in real time with variation of cooling load is main difficulty.

A new approach to set point control of VSD pump speed and control valve positions is proposed in this paper to deal with the difficulty. Based on analysis of the relationships between differential pressure and chilled water flow rate in chilled water loop, an optimization problem accompanied its constraints are given. Through a simple optimization algorithm, the optimal set points can be found out promptly and be implemented into control in real time for energy conservation. Compared with the traditional control method, potential energy savings can be achieved by the new approach.

SYSTEM DESCRIPTIONS

The typical variable flow chilled water distribution system is shown in Figure 1. Each distribution piping branch consists of a cooling coil and a two-way control valve. The two-way control valve is modulated by off coil air temperature (a room thermostat) in response to the varying room load. The VSD pump speed is modulated by a differential pressure controller usually located at the far end of the direct-return piping load as the system flow varies in order to keep a constant pressure differential set point. The differential set point is often chosen on the basis of the design condition and after being applied a safety factor. This kind of chilled water system was criticized by some experienced experts (Hartman 1998, Bynum and Merwin 1999) because it could not save the operating energy as expected in design phase.

Fixed differential pressure control is the main reason of the inefficiency. Because the fixed differential pressure must be maintained, the head of VSD pump cannot be reduced as the pump law and the pump energy is wasted. In order to resolve this problem, the following system (Figure 2) by different control strategy is proposed to save the pumping energy under part-load conditions. In the new scheme, the VSD pump is no longer merely controlled by differential pressure sensors. It is controlled by optimal set points provided by a DDC controller, which collects the cooling load data of each coil. The control valves are no longer controlled by room temperature sensors. Instead, they are controlled by a DDC controller too.

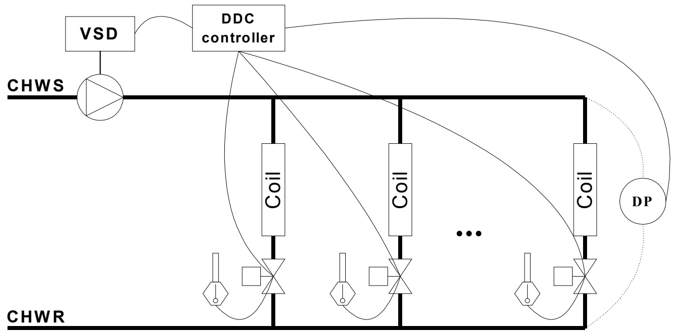

Figure 1. Typical chilled water distribution system

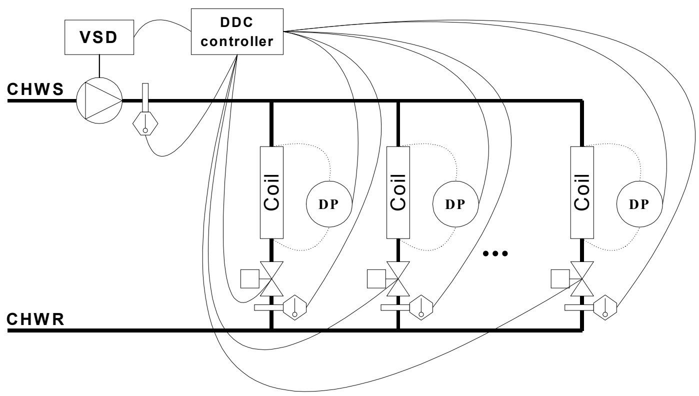

Figure 2. Proposed chilled water distribution system

The control strategy is given in following.

- The DDC controller collects the chilled water supply temperature (T_{CHWS}) and the chilled water return temperature of each distribution branch ($T_{CHWR,i}$).
- The DDC controller collects the differential pressure values of all the cooling coils ($H_{CL,i}$) and determines the measured chilled water flow rate across the coils ($m_{w,i,m}$) according to the coil characteristics, $m_{w,i,m} = f(H_{CL,i})$.
- Based on the measured data, the DDC controller calculates the cooling load of each coil ($Q_{coil,i} = m_{w,i,m} C_{pw} (T_{CHWR,i} - T_{CHWS})$) and the new required water flow rate ($m_{w,i} = \dfrac{Q_{coil,i}}{C_{pw}(T_{CHWR,des} - T_{CHWS})}$) to achieve the designed chilled water return temperature ($T_{CHWR,des}$) by the following two equations.
- By optimization method introduced in the next section, the DDC controller determines the new set points of VSD pump speed and the control valve positions in every distribution branches. The VSD pump speed is determined by water flow rate ($m_w = \sum_i m_{w,i}$) and head of the pump ($H_{VL,0}$). The control valve position is given in term of head loss across the valve ($H_{VL,i}$).
- The optimal set points are sent to local controllers or actuators to control the whole system.

PROBLEM FORMULATION AND OPTIMIZATION

The objective of optimizing the chilled water system is to deliver the enough chilled water to all the cooling coils at the least expenditure. The main energy consuming components are the distribution chilled water pumps equipped with VSDs. Therefore, the objective function is described in the following function according to mechanism model.

$$\text{Min: } P_{pump} = \frac{m_w \cdot H_{VL,0}}{\eta_{pump}\eta_{motor}\eta_{VSD}g_c} \tag{1}$$

The total required water flow rate m_w has been determined in the previous step. The efficiency factors: η_{pump}, η_{motor} and η_{VSD}, are the functions of m_w and $H_{VL,0}$. The polynomial can be employed to approximate the effect of all the efficiency factors when evaluating P_{pump}. There are also two limits about $H_{VL,0}$. One is that the pump head should be within the normal operating region. The other is that the head loss of all the coils should be maintained with such pump head.

For the commonly used direct-return hydronic system, the following equations are formed according to the principle of conservation of mass flow at a node and uniqueness of pressure at a given point in the loop. The first equation describes the head losses of the coils. The second equation describes the head loss of the distribution pipes. The third and fourth equations give the initial value of head loss and flow rate. The last equation expresses the relationship of branch head between coils. The variables in the equations are labeled out in Figure 3.

$$H_{CL,i} = K1_i \cdot m_{w,i}{}^2 \quad (i = 1,2,\cdots n) \tag{2}$$

$$H_{PL,i} = K2_i \cdot \left(\sum_{j=i}^{n} m_{w,j}\right)^2 \quad (i = 1,2,\cdots n) \tag{3}$$

$$m_{w,0} = 0 \tag{4}$$

$$H_{CL,i-1} + H_{VL,i-1} = H_{CL,i} + H_{VL,i} + 2H_{PL,i} \quad (i = 1,2,\cdots n) \tag{5}$$

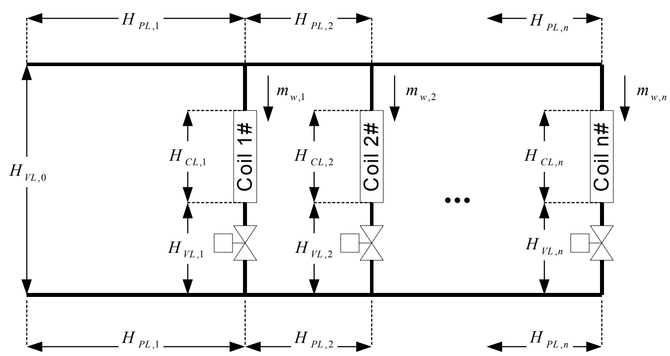

Figure 3. Variables appeared in the optimization problem

Rewrite the problem in a concise way; the optimization problem is as the following.
Objective function:

$$\text{Min } P_{pump} = f(H_{VL,0}) = a_0 + a_1 H_{VL,0} + a_2 H_{VL,0}{}^2 \tag{6}$$

Constraints:

$$H_{VL,0} - \min(H_{VL,0}) \geq 0 \tag{6a}$$

$$\max(H_{VL,0}) - H_{VL,0} \geq 0 \tag{6b}$$

$$H_{VL,i} \geq 0 \quad (i = 1,2,\cdots n) \tag{6c}$$

$$\left(K1_i m_{w,i}^2\right) + H_{VL,i} + 2 \cdot K2_i \left(\sum_{j=i}^{n} m_{w,j}\right)^2 - \left(K1_{i-1} m_{w,i-1}^2\right) - H_{VL,i-1} = 0 \quad (i = 1,2,\cdots n) \tag{6d}$$

In the objective function, for the specific water flow rate, the energy consumption of pump (P_{pump}) can be treat as the function of pump head ($H_{VL,0}$). The polynomial is employed to approximate P_{pump} according to catalog data and the parameters (a_0, a_1, a_2) are derived by curve-fitting method. There are totally n+1 unknown variables: $H_{VL,i}$(i = 0, 1, … n). The other variables have already known by the previous steps. The constraints consist of n+2 inequality constraints and n equality constraints.

The general way to solve the problem is to construct a Lagrangian function in the following equation.

$$L(H_{VL}, \lambda) = f(H_{VL}) - \sum_{i=1}^{2n+2} \lambda_i c_i(H_{VL}) \tag{7}$$

$c_i(H_{VL})$ are n+2 inequality constraint functions where i = 1, 2, … n+2 and $c_i(H_{VL})$ are n equality constraint function where $i = n$+3, … 2n+2.

To find the local minimum of this optimization problem, Karush-Kuhn-Tucker conditions are needed. After that, this local minimum also needed to be proved as a global minimum. However, it is too complicated to use Karush-Kuhn-Tucker conditions to get the results. If we observe the constraints carefully, we can find a much more easy way to solve the problem. Equation (6d) can be rewrite in another form.

$$H_{VL,i} = H_{VL,0} - K1_i m_{w,i}^{\ 2} - \sum_{l=1}^{i}\left(2 \cdot K2_l \left(\sum_{j=i}^{n} m_{w,j}\right)^2\right) \ (i = 1,2,\cdots,n) \tag{8}$$

If the unknown variables in Equation (6d) are substituted by Equation (8), this problem will be chanced into a one-variable optimization problem. It has a simple objective function, binomial, and n+2 linear constraints, which limit the feasible region of the only variable, $H_{VL,0}$. This simple problem can be solved by finding the feasible region of the only variable limited by n+2 constraint functions and searching the minimal point by any feasible methods.

IMPLEMENTATION ISSUES

The new approach proposed in this paper is capable of tackling most of chilled water pumping systems with two-way control valves (Luther 1998) (Wang 2001), such as:

- Primary-secondary pumping system (Plant-building loop);
- Primary variable speed pumping system (Plant-through-building loop: variable flow);
- Plant-distributed pumping loop;
- Primary-secondary-tertiary pumping system (Plant-distributed building loop).

For the typical primary-secondary pumping system, the new approach can be implemented directly. The set points of secondary pump head ($H_{VL,0}$) is determined by optimization algorithm stored in central control system and controlled by VSD. All of the set points of control valve pressure loss is also determined by the algorithm and controlled by setting the appropriate valve position according to the valve characteristic curves.

In a primary variable pumping system, there is a common pipe (bypass) between chilled water supply and return pipe to ensure that minimum flow is always maintained through the chillers. Therefore, the new approach needs some modifications before implementation. In theoretic, the common pipe can be treated as the first distribution branch, where no coil exists ($H_{CL,1} = 0$) and the head drop across the valve must larger than the minimum value ($H_{VL,1} - \min(H_{VL,1}) \geq 0$). The minimum valve of the control valve in the common pipe is determined by the minimum flow requirements of chillers.

The configuration of plant-distributed pumping loop is similar with that of primary variable speed pumping system. The differences of two systems lie in two points. One is that the primary pumps in plant-distributed pumping loop have constant speeds, whereas the primary pumps in primary variable speed pumping system have variable speeds. The other difference is that all the control valves in distribution branch in primary variable pumping system are changed into VSD pumps in plant-distributed pumping loop except the common pipe. Therefore, the common pipe can be treated same as the former implementation and the head losses of valves ($H_{VL,i}$) are substituted by the negative pump head loss of pumps ($-H_{P,i}$). The optimization problem becomes into the following form.

Objective function:

$$\text{Min } P_{pump} = \sum_{i=1}^{n} f(H_{P,n}) = \sum_{i=1}^{n}\left(a_0 + a_1 H_{P,i} + a_2 H_{P,i}^{\ 2}\right) \tag{9}$$

Constraints:

$$H_{P,i} - \min(H_{P,i}) \geq 0 \ \ (i = 0,1,\cdots n) \tag{9a}$$

$$\max(H_{P,i}) - H_{P,i} \geq 0 \ \ (i = 1,2,\cdots n) \tag{9b}$$

$$H_{P,0} = H_{VL,0} \tag{9c}$$

$$\left(K1_i m_{w,i}^2\right) - H_{P,i} + 2\cdot K2_i\left(\sum_{j=i}^{n} m_{w,j}\right)^2 - \left(K1_{i-1} m_{w,i-1}^2\right) + H_{P,i-1} = 0 \quad \left(i = 1,2,\cdots n\right) \tag{9d}$$

The head loss across the control valve in the common pipe is $H_{VL,0}$. If there is no control valve, $H_{VL,0}$ must equal to the pressure difference of the common pipe.

For the primary-secondary-tertiary pumping system, it is a very complex system and its configuration is shown in Figure 4. This kind of pumping system can be classified into two cases to consider respectively according to the type of tertiary pump.

If the tertiary pumps have a constant speed, the pump head of the *i*th subsystem can be considered as a constant, after the cooling load of each coil in *i*th subsystem is determined. According to measured data and Equation (2-5), the water flow rate of the *i*th subsystem ($m_{w,i}$), the pressure difference between inlet and outlet of the *i*th subsystem ($H_{BL,i}$), and water temperature leaving subsystem are easy to calculated. With all these data, the *i*th subsystem can be treated as an equivalent cooling coil. When all of subsystems transform into cooling coils, the optimization of whole system is same as the primary-secondary pumping system. When all the set points in the distribution loop are determined, they also need to be used to determine the set points of control valves in subsystems.

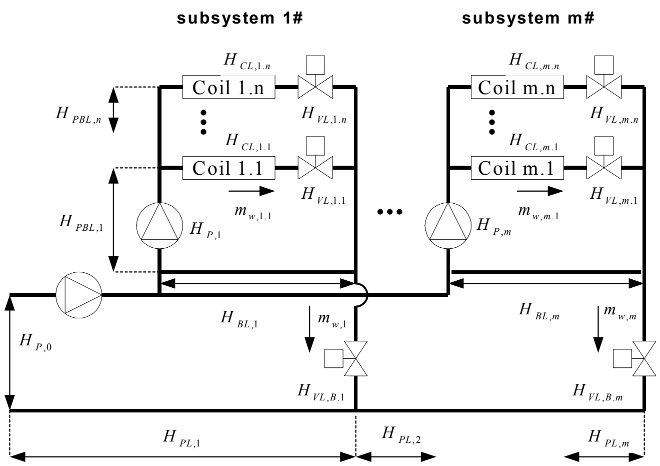

Figure 4. Primary-secondary-tertiary pumping system

If the tertiary pumps are all controlled by VSD, the optimization problem will be more complicated. It can be expressed in equation form like following.

Objective function:

$$\text{Min } P_{pump} = \sum_{i=0}^{m} f\left(H_{P,n}\right) = \sum_{i=0}^{m}\left(a_0 + a_1 H_{P,i} + a_2 H_{P,i}{}^2\right) \tag{10}$$

Constraints:

$$H_{P,i} - \min\left(H_{P,i}\right) \geq 0 \ \left(i = 0,1,\cdots m\right) \tag{10a}$$

$$\max\left(H_{P,i}\right) - H_{P,i} \geq 0 \ \left(i = 0,1,\cdots m\right) \tag{10b}$$

$$H_{VL,j,i} \geq 0,\ H_{VL,B,j} \geq 0 \ \left(i = 1,2,\cdots n;\ j = 1,2,\cdots m\right) \tag{10c}$$

$$H_{BL,i} + H_{VL,B.i} + 2\cdot K2_i\left(\sum_{j=i}^{n} m_{w,j}\right)^2 - H_{BL,i-1} - H_{VL,B.i-1} = 0 \quad \left(i = 1,2,\cdots m\right) \tag{10d}$$

$$H_{CL,j.i} + H_{VL,j.i} + \sum_{l=1}^{i}\left(2\cdot K2_{j.l}\left(\sum_{k=i}^{n} m_{w,j.k}\right)^2\right) - H_{P,j} - H_{BL,j} = 0 \ \left(i = 1\cdots n;\ j = 1\cdots m\right) \tag{10e}$$

$$K1_{j,i}\cdot m_{w,j,i}^2 - H_{CL,j,i} = 0 \ \left(i = 1\cdots n;\ j = 1\cdots m\right) \tag{10f}$$

There are totally $m(n+1)+1$ variables in the problem. After optimization, the set points of $m+1$ pump heads and $m\times n$ head loss of control valve will be given simultaneously and all of them should be used to control the system at the same time. The other alternative is to optimize each subsystem firstly, then to the main system. However, the procedure of this method is even more complicated.

AN APPLICATION EXAMPLE

To investigate the possibility of reducing energy requirements by new approach to set the pump head and control valve positions, a computer simulation has been done for arbitrarily selected load profile and the common used direct-return hydronic system.

There are totally four coils in the system and they have the same size. It is suppose that the water flow rate of each coil is 200 gpm (12.6 liter/s) at the full load condition and the differential pressure drop across the coil is 8 ft (23.9 kPa). At the full load condition, $H_{VL,1}$=4 ft (12.0 kPa); $H_{VL,2}$=$H_{VL,3}$=$H_{VL,4}$=1 ft (3.0 kPa). The cooling load profile is selected arbitrarily and shown in Figure 5. All the cooling coils are at very light load condition during the off-peak time. During the working time, the cooling loads are between 30% and 100% of the full load. At the early afternoon, the total cooling load reaches the maximum of the whole day.

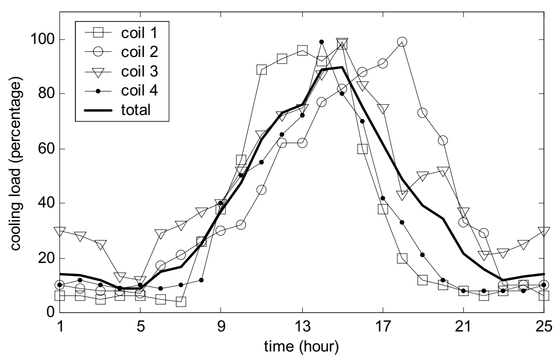

Figure 5. Cooing load profile

The required pump heads of two different methods are shown in Figure 6.

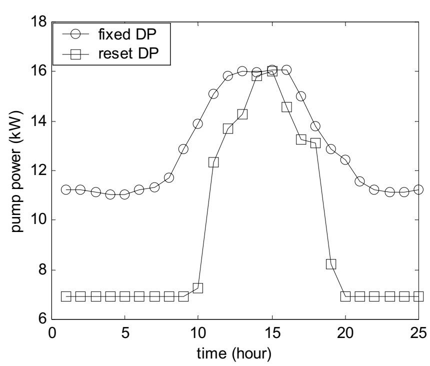

Figure 6. comparison of required head of two methods

From this figure, it is clearly that the approach with reset set points of the differential pressure can save a lot of energy compared with the conventional approach with the fixed differential pressure set point. During the peak time of operation, two approaches need consume almost same energy to deliver the appropriate water flow for each coil. The energy savings are mainly at the off-peak time. When the cooling loads of all four coils are very light, the chilled water pump is working at the lower bound of operating limits and the energy savings under this condition is significant.

CONCLUSIONS

Based on the discussion above, a lot of energy consumption of the distribution pumps can be saved by properly implementation the VSD and DDC techniques into chilled water loops. Compared with the conventional control strategy, fixed differential pressure set point control, the proposed approach can find the optimal set points of the chilled water system in real time. With the optimal set points of VSD pump speed and control valve positions, the minimal required chilled water flow rate can be maintained in water distribution systems. At the same time, designed chilled water return temperature can be achieved by the appropriate coil water flow rates determined by control valves.

REFERENCES

Ahmed, O., “DDC applications in variable-water-volume systems,” ASHRAE Transactions, Vol. 97(1), pp. 751-758, 1991.

ASHRAE, ASHRAE Handbook-Applications, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1999.

Avery, G., “Controlling chillers in variable flow systems,” ASHRAE Journal, Vol. 40, No.2, pp. 42-45, 1998.

Bynum, H. and Merwin, E., “Variable flow—a control engineer’s perspective,” ASHRAE Journal, Vol. 41, No. 1, pp. 26-30, 1999.

Eppelheimer, D.M., “Variable flow—the quest for system energy efficiency,” ASHRAE Transactions, Vol. 102(2), pp. 673-678, 1996.

Green, R.H., “An air-conditioning control system using variable-speed water pumps,” ASHRAE Transactions, Vol. 100(1), pp. 463-470, 1994.

Hartman, T., “Design issues of variable chilled-water flow through chillers,” ASHRAE Transactions, Vol. 102(2), pp. 679-683, 1996.

Hartman, T., “Packing DDC networks with variable speed drives,” HPAC, Vol. 70, No. 11, pp. 80-84, 1998.

Hegberg, R.A., “Converting constant-speed hydronic pumping systems to variable-speed pumping,” ASHRAE Transactions, Vol. 97(1), pp. 739-745, 1991.

Hodge, B.K. and Taylor, R.P., Analysis and Design of Energy Systems, Prentice Hall: New Jersey, 1999.

Luther, K.R., “Variable volume pumping fundamentals,” HPAC, Vol. 70, No.8, pp. 69-75, 1998.

Luther, K.R., “Applying variable volume pumping,” HPAC, Vol. 70, No. 10, pp. 53-58, 1998.

Nocedal, N. and Wright, S.J., Numerical Optimization, Springer-Verlag: New York, 1999.

Rishel, J.B., “Control of variable-speed pumps on hot- and chilled-water systems,” ASHRAE Transactions, Vol. 97(1), pp. 746-750, 1991.

Tillack, L. and Rishel, J.B., “Proper control of HVAC variable speed pumps,” ASHRAE Journal, Vol. 40, No. 11, pp. 41-47, 1998.

Wang, S.K., Handbook of Air Conditioning and Refrigeration, McGraw-Hill: New York, 2001.