2010

Control strategies for variable speed pumps in super high-rise building

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Control Strategies for Variable Speed Pumps in Super High-Rise Building

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Two control strategies for variable speed pumps were tested in a super high-rise building in Hong Kong.[1] The 1,608 ft (490 m) high building has a floor area of 3.4 million ft^2 (321 000 m^2). The basement is four floors, a block building is six floors and a tower building is 98 floors.

Figure 1 shows the central chilling system with six identical high voltage centrifugal chillers located on the sixth floor (2,056 ton [7230 kW] each). The design chilled water supply is 5.5°C (41.9°F) and return temperature is 10.5°C (50.9°F). Each chiller is associated with one constant condenser water pump and one constant primary chilled water pump. The heat dissipated from the chiller condensers is rejected by means of 11 evaporative water cooling towers with a total design capacity of 14,703 tons (51 709 kW).

To avoid extremely high pressure in the chilled water pipelines and terminal units, the secondary chilled water system is divided into four zones. Only Zone 2 is supplied with the secondary chilled water directly. For the other three zones, the heat exchangers are used to transfer the cooling energy from low zones to high zones to avoid the high water static pressure.

Zone 1 is supplied with the secondary chilled water through the heat exchangers (HX-06) located on the sixth floor, while the chilled water from chillers serves as the cooling source for the

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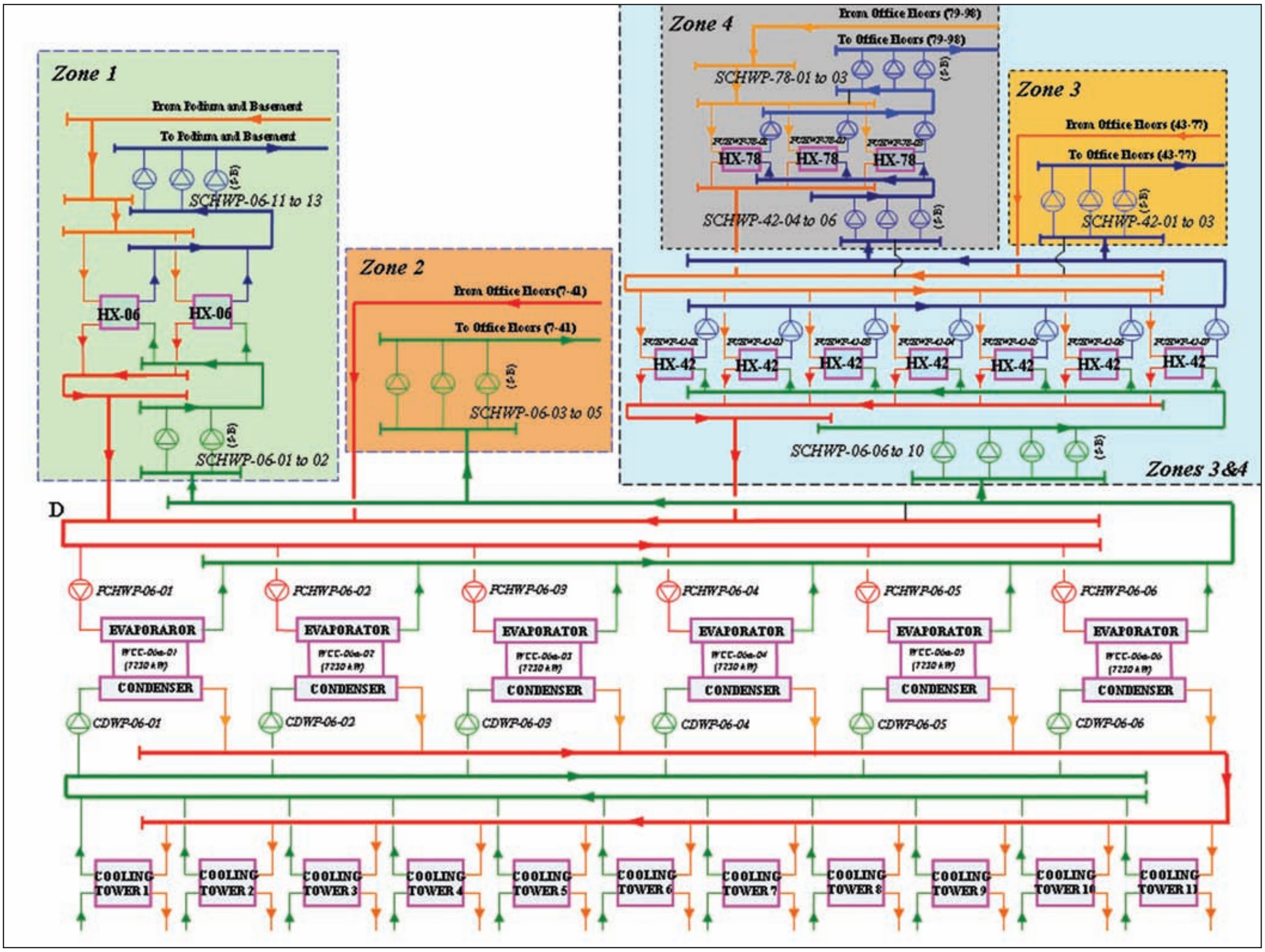

Figure 1: Schematics of the central chilling system.

heat exchangers. Zones 3 and 4 are supplied with the secondary chilled water through the first stage heat exchangers (HX-42) located on the 42nd floor. Some of the chilled water after the first stage heat exchangers is delivered to Zone 3, and some is delivered to the second stage heat exchangers (HX-78) located on the 78th floor.

The design chilled water supply is 6.3°C (43.3°F), and the return temperature is 11.3°C (52.3°F) at the secondary sides of the heat exchangers in Zone 1, and the first stage heat exchangers in Zones 3 and 4. Both design supply and return temperatures at the secondary sides of the second stage heat exchangers in Zone 4 are 7.1°C (44.8°F) and 12.1°C (53.8°F), respectively.

All pumps in the secondary water system are equipped with variable-frequency drives (VFDs) except the primary chilled water pumps dedicated to the heat exchangers in Zones 3 and 4, which are constant speed pumps. An alternative design configuration for these heat exchangers without using the dedicated primary chilled water pumps is proposed in a previous *ASHRAE Journal* article.[1]

All variable speed pumps in the secondary system can be categorized into two groups: the pumps distributing water to terminal units and the pumps distributing water to heat exchangers. This article focuses on the speed control of variable speed pumps distributing water to heat exchangers. The speed control of variable speed pumps distributing water to terminal units has been extensively studied elsewhere.[2–6]

Original Control Strategy

Figure 2 shows the original control strategy used in this building for controlling the operating speed of variable speed pumps distributing water to heat exchangers. In this strategy, the measured temperatures at the secondary side and the measured differential pressure at the primary side of heat exchangers are used. The operating speed of the pump in the primary side is controlled by maintaining the differential pressure at the primary side at a predetermined constant value (*Figure 2*).

To maintain the supply water temperature at the secondary side, a modulating valve is installed at the primary side of each heat exchanger loop. The opening of the modulating valve is controlled to maintain the supply water temperature at the secondary side at its setpoint. When the load of the terminal units changes, the opening of the modulating valves is adjusted to meet the preset chilled water temperature setpoint and, therefore, meet the load change of the terminal units.

The speed of the pumps at the primary side is modulated to maintain a constant differential pressure at the primary side. The control algorithm used in this method is similar to the conventional method used to control the operating speed of pumps distributing water to terminal units.

In this original control strategy, additional pump energy will be consumed at part-load conditions, especially at off-design conditions. When the load of the terminal units reduces, the modulating valves need to close down to reduce the water flow rate at the primary side of heat exchangers. This causes the head of the pumps to be consumed by the modulating valves. Therefore, unnecessary pump power will be consumed.

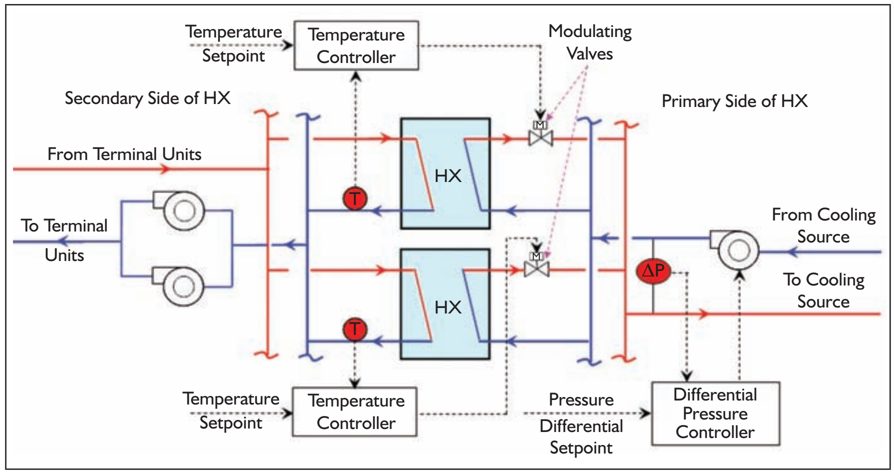

Figure 2: Schematic of original control strategy.

Alternative Control Strategy

To achieve energy-efficient control of the pumps distributing water to heat exchangers, an alternative control strategy (*Figure 3*) is proposed. In this strategy, the water flow measurements at primary and secondary sides of the heat exchangers, and measurements of the water temperatures at the secondary side of heat exchangers are used. The supply temperature at the secondary side is controlled by adjusting the water flow rate (pump speed) at the primary side while the original modulating valves are set fully open. A cascade control scheme is used to provide stability and ensure that the difference between the water flow rates on both sides is controlled within an acceptable range.

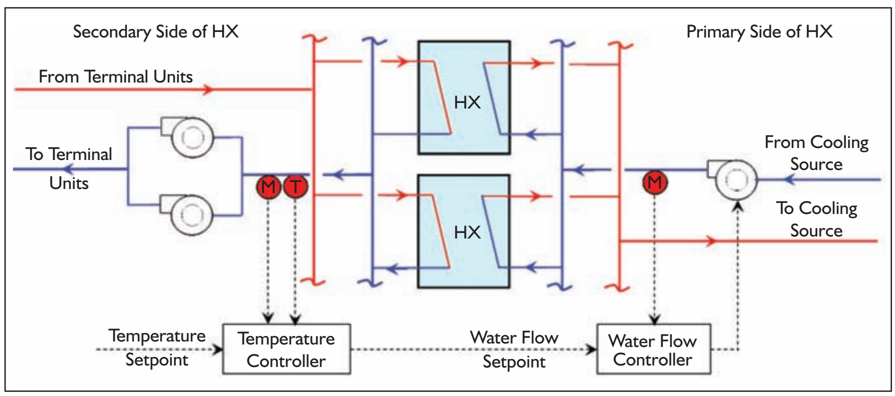

Figure 3: Schematic of alternative control strategy.

The PI temperature controller (*Figure 3*) generates the water flow setpoint of the primary loop. A simple linear relationship with the PI output is adopted to determine the water flow setpoint (M_{set}) (*Figure 4*). M_{low} and M_{up} are the lower and upper limits of the water flow rates at primary side and are set as the parameters. The water flow setpoint is set within a range near the actual water flow at the secondary side (many experts have suggested that the water flow rates at both sides must be balanced). Their selection is described in Equations 1 through 3. M_{sec} is the measured water flow rate at the secondary side. M_{max} and M_{min} are the maximum and minimum limits of primary flow rate, which are set according to the water loop design parameters.

In this building, the value of α selected is 0.25. M_{max} and M_{min} are the design flow rate and 20% of the design water flow rate at the primary side. The value of 0.25 was selected based on site experiences and discussion with system designer and operators, ensuring that the difference between flow rates at both sides is not large. The value of 20% was selected to ensure that the system operation can cover the possible allowed operation range.

$$M_{low} \le M_{set} \le M_{up} \tag{1}$$

$$M_{up} = \min\left(\left(1+\alpha\right)M_{sec}, M_{max}\right) \tag{2}$$

$$M_{low} = \max\left(\left(1-\alpha\right)M_{sec}, M_{min}\right) \tag{3}$$

Compared with the original control strategy, there is no need to use the modulating valves at the primary side when using this alternative control strategy.

Performance Tests and Evaluation

Both control strategies have been implemented in the BAS of the building. Zones 1 and 2 have been occupied for more than one year, while the other upper zones are still under construction. Upon initial operation, the original control strategy was used to provide the control function for the pumps in the primary side of the heat exchangers.

The alternative control strategy replaced the original strategy last summer. Since both low zones were not fully occupied, the cooling loads in the zones were still very small, and the pump constantly operated near the lowest allowable operating frequency when both strategies were used. Therefore, the performances of both strategies only can be tested and compared on site with low pump operating frequency. An extensive simulation study also was conducted to further compare the energy performances of both strategies with normal cooling loads in the zone (fully occupied) and evaluate their annual energy performance (which cannot be tested on site).

Site Tests

The energy performance of both strategies was tested on site. During the tests, the temperature setpoint at the secondary side, chiller supplied water temperature setpoint and the operating numbers of all related components (e.g., heat exchangers and water pumps) were set the same for both strategies. Taking into account the light load in the zone at present, the pressure differential setpoint in the original strategy was set at 20 kPa (2.9 psi) instead of at the design value of 60 kPa (8.7 psi).

When using the original strategy, the power consumption of the pump was 14.1 kW. When using the alternative strategy, the actual pressure drop across the heat exchanger loop was 8.3 kPa (1.2 psi) and the power consumption of the pump was reduced to 11.0 kW. The energy saving due to the use of the alternative control strategy was 22%. This demonstrated that the alternative strategy has better energy performance than that of the original.

The site application of the alternative control strategy over several months also demonstrated that this strategy can provide stable and reliable control in practical applications.

Simulation Tests

The simulation tests used a virtual building system that was constructed for previous studies.[7,8] Detailed operational performance of the variable speed pumps distributing water to the heat exchangers of only Zone 1 is presented in this article as the example. The cooling loads used in the tests were predetermined using EnergyPlus[9] based on the design data and hourly based weather data of the typical year in Hong Kong.

Since the operation of variable speed pumps distributing water to heat exchangers has an impact on the operation of the pumps on the secondary side and chillers, the control strategies used for these components are introduced here briefly. The chillers were sequenced based on their design cooling capacities due to one constant primary chilled water pump and one constant condenser water pump dedicated to one chiller in this system. A threshold of 10% of the design cooling capacity was used for bringing chillers online and offline.

For variable speed pumps and heat exchangers, they were sequenced based on 85% of their design water flow rates. A threshold of 10% of the design flow rates was used for switching the pumps and heat exchangers on and off. For stable control, a minimal time interval was introduced in these sequence strategies to avoid frequent switching.

The speed of pumps distributing water to terminal units were controlled through resetting the pressure differential setpoint at the critical loop while the setpoint was optimized using the optimization strategy presented in *ASHRAE Handbook*,[10] in which the water valve positions of terminal units were used to determine the increase or decrease of the pressure differential

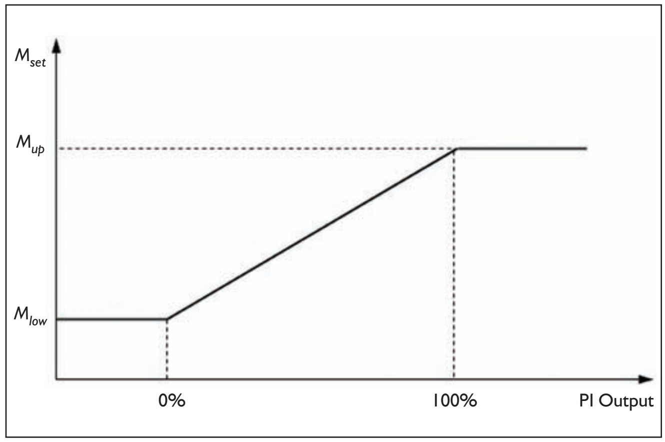

Figure 4: Relationship between the PI temperature output and the water flow setpoint.

setpoint based on the last setting, by a fixed increment.[10] Since the chilled water supply temperature setpoints at the chiller side and secondary side of heat exchangers have significant impacts on the energy consumptions of chillers and pumps, the design setpoints were used and remained unchanged during the tests.

The operational data of the tests during three typical working days using the previous control strategies are presented to compare the energy performances of both strategies. These typical working days represent the typical operating conditions of the air-conditioning system in the spring, mild summer and sunny summer. As shown in *Table 1*, compared with the original strategy, the alternative strategy saved 182.3 kWh (16.26%), 202.9 kWh (15.28%) and 186.2 kWh (11.25%) of energy of the primary pumps in the three typical days.

Table 1 shows that when using the alternative strategy, the energy consumption of the secondary pumps was reduced slightly while the energy consumption of the chillers was increased slightly in the three test days. The pump energy reduction is attributed to, at very light load conditions, a lower (i.e., lower than its setpoint) supply water temperature at the secondary side of heat exchangers was provided by the alternative strategy. This was due to the introduction of the lower limit of 20 Hz for the pump operating frequency. This lower water supply temperature resulted in saving some energy of the secondary pumps.

The increase of the chiller energy is probably due to the increase of the latent load resulted by the lower supply water temperature to terminal units, or the effects of system strong dynamics in the simulation, etc. During the tests, the pressure differential setpoint used in the original control strategy was the design value of 60 kPa (8.7 psi).

The monthly energy consumption of the primary and secondary pumps in Zone 1 and the total monthly energy consumption of the chilled water system (including the energy consumptions of chillers, all variable speed pumps in Zone 1 and constant speed pumps dedicated with chillers) using both strategies was simulated, and the results are presented in *Table 2*.

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Test Cases			Spring		Mild Summer		Sunny Summer	
Control Strategies			Original	Alternative	Original	Alternative	Original	Alternative
Energy Consumption	Primary Pumps	kWh	1,121.0	938.7	1,328.0	1,125.1	1,654.5	1,468.3
	Secondary Pumps	kWh	2,096.3	2,065.5	2,348.8	2,343.3	2,772.7	2,772.3
	Chillers	kWh	18,271.5	18,338.0	20,946.9	21,006.6	28,873.3	28,878.2
Savings	Primary Pumps	kWh	–	182.3	–	202.9	–	186.2
		%	–	16.26	–	15.28	–	11.25
	Secondary Pumps	kWh	–	30.8	–	5.5	–	0.4
		%	–	1.47	–	0.23	–	0.01
	Chillers	kWh	–	–66.5	–	–59.7	–	–4.9
		%	–	–0.36	–	–0.29	–	–0.02

Table 1: Comparison of daily energy consumptions using different control strategies for variable speed pumps distributing water to heat exchangers in Zone 1 of the building.

Month	Original Strategy			Alternative Strategy			Savings		
	Energy of Primary Pumps (kWh)	Total Energy of Primary and Secondary Pumps (kWh)	Total Energy of the Chilled Water System (kWh)	Energy of Primary Pumps (kWh)	Total Energy of Primary and Secondary Pumps (kWh)	Total Energy of the Chilled Water System (kWh)	Energy of Primary Pumps (%)	Total Energy of Primary and Secondary Pumps (%)	Total Energy of the Chilled Water System (%)
Jan.	38,588	101,965	937,239	32,410	95,560	932,680	16.01	6.28	0.49
Feb.	35,376	93,477	853,173	29,979	87,854	849,149	15.26	6.02	0.47
Mar.	40,491	106,995	965,196	34,652	101,210	960,759	14.42	5.41	0.46
Apr.	39,752	105,041	941,798	34,131	99,279	936,831	14.14	5.49	0.53
May	45,664	120,664	1,070,255	39,266	114,050	1,063,014	14.01	5.48	0.68
Jun.	49,166	129,916	1,165,198	42,978	123,473	1,158,492	12.59	4.96	0.58
Jul.	51,781	137,500	1,285,213	45,473	130,931	1,278,506	12.18	4.78	0.52
Aug.	51,345	135,823	1,273,334	45,174	129,580	1,267,233	12.02	4.60	0.48
Sep.	49,247	130,105	1,183,541	43,039	123,785	1,177,319	12.61	4.86	0.53
Oct.	45,091	119,148	1,057,604	38,809	112,629	1,050,841	13.93	5.47	0.64
Nov.	41,819	110,502	992,036	36,225	104,972	986,873	13.38	5.00	0.52
Dec.	39,688	104,871	956,998	33,996	99,324	953,129	14.34	5.29	0.40
Annual Saving of the Chilled Water System (Zone 1)									66,759 kWh

Table 2: Comparison of monthly energy consumptions using different control strategies for variable speed pumps distributing water to heat exchangers in Zone 1 of the building.

Compared with the original strategy, about 12.02% to 16.01% of the monthly energy of the primary pumps and 4.60% to 6.28% of the total monthly energy of the primary and secondary pumps in Zone 1 were saved when the alternative strategy was used. The total monthly energy saving in the chilled water system were ranged from 0.40% to 0.68%. The annual energy saving due to the use of the alternative control strategy for primary pumps in Zone 1 of the building was 66,759 kWh.

Table 3 presents a summary of the annual energy consumption of all variable speed primary pumps (primary to heat exchangers) in the building by using two control strategies. This comparison was based on the assumptions that Zone 1 was in 24-hour operation and the other zones were only operated during office hours. Annual energy of 251,869 kWh can be saved if the alternative strategy is used to control all variable speed primary pumps in the building, as compared to using the original.

Conclusions

The proposed alternative strategy has been implemented on site and compared with the original strategy. The results from the site tests and simulation study showed that significant chilled water pumps energy can be saved when using this alternative control strategy. Compared with the original control strategy, about 12.02% to 16.01% of the energy of the pumps distributing water to heat exchangers (i.e., primary pumps) can be saved when the alternative control strategy is used. The annual energy saving of all variable speed primary pumps in the building by using the alternative control strategy was about 251,869 kWh. Such energy saving was achieved by using only the improved control strategy and without adding any additional cost.

Over several months, the site application of the alternative control strategy demonstrated that it can provide stable and reliable control in practical applications.

Pumps	Number (Standby)	Energy Consumption (kWh)		Saving (kWh)
		Original Strategy	Alternative Strategy	
Primary Pumps in Zone 1	1(1)	528,008	456,132	71,876
Primary Pumps in Zones 3 & 4	3(1)	921,235	795,830	125,405
Primary Pumps in Zone 4	2(1)	401,008	346,420	54,588
			Total Saving of the Primary Pumps	251,869

***Table 3:** Comparison of annual energy consumption of all variable speed (primary) pumps distributing water to heat exchangers in the building using different strategies.*

Acknowledgments

Research work was financially supported by the grants of the National 11-5 program and The Hong Kong Polytechnic University, as well as the support from Sun Hung Kai Real Properties Limited. The authors would like to thank Mr. W. K. Pau, Sun Hung Kai Real Properties Limited for his essential support to the research work.

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