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Online Optimal Control of The Central Chilling System in a Super High-Rise Commercial Building

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ABSTRACT

This paper presents the online optimal control strategies of the central chilling system in a super high-rise building in Hong Kong. The work presented aims at developing a package of optimal control strategies and evaluating their energy saving potentials and finally implementing these strategies in the building to enhance its energy efficiency. Based on testing the effects of interactions within the cooling water system and chilled water system on the system operation performance, the optimal control strategies for the cooling water system and variable speed pumps with different installations in the secondary chilled water system are developed. These strategies are tested and validated in a virtual environment representing the real building system and its central chilling system prior to site implementation. The implementation issues the online control strategies for practical application are also addressed.

1. INTRODUCTION

The growing concern on energy or cost efficiency has motivated society and building professionals to pay more attention on the overall system optimal control and provides incentives to develop the more extensive and robust control strategies for HVAC systems. Supervisory and optimal control, which addresses energy or cost efficient control of HVAC systems while providing desired indoor comfort and healthy environment under dynamic working conditions, is one of the main achievable approaches to minimize building energy consumption/operating costs and provide robust control performance.

Buildings nowadays are mostly equipped with comprehensive building automation systems (BASs) and building energy management and control systems (EMCSs) that allow the possibility of enhancing and optimizing the operation and control of HVAC systems. Over the last two decades or so, efforts have been made to develop supervisory and optimal control strategies for building HVAC systems thanks to the growing scale of BAS integration and the convenience of collecting a large amount of online operation data by the application of BASs. These efforts have resulted in many research papers and technical articles that specifically address the HVAC optimal control and operation available in literature (Wang and Jin, 2000; Sun and Reddy, 2005; etc.).

In cooling water systems, cooling towers are usually used for heat rejection purposes and the fans are equipped with variable frequency drivers (VFD) to control the condenser water supply temperature at its set-point intended. There are many existing control strategies for cooling towers. The simple strategy is to maintain a constant condenser water supply temperature set-point (i.e., fixed set-point) or to maintain a constant difference between the condenser water supply temperature set-point and the ambient air wet-bulb temperature (i.e., fixed approach) by varying the air flow rate through the cooling towers (Crowther and Furlong, 2004). Several near optimal control strategies have been also proposed to determine a near optimal condenser water supply temperature set-point, and then utilize this temperature set-point to approximately control the operation of the cooling towers (Braun and Diderrich, 1990; Sun and Reddy, 2005; etc.). These strategies are simple enough and easy to implement in practice. However, these strategies cannot provide the true optimal settings, which might provide the settings significantly different from the optimal values, and a significant amount of energy might still be wasted.

The energy costs of variable speed pumps form an important part of the total energy consumption of buildings. Aiming at enhancing the operating efficiency of pumps and prolonging their service life, a number of researchers and experts in the HVAC field have devoted considerable efforts on energy efficient control and operation of variable speed pumps in HVAC systems in the course of past years (Burke, 1995; Bahnfleth and Peyer, 2001; Rishel,

2003). Burke (1995) pointed out that when pumps operate within ±20% of their best efficiency points, there would seldom have any operation problems. Rishel (2003) presented that the pumps can be sequenced properly based on the wire-to-water efficiency or kW input to the pumping system and their speeds can be controlled through the use of pressure differential transmitters located at the critical loops and continuous interrogation of them. Several studies have presented that the valve position can be a valuable tool to optimize the pressure differential set-point to control the pump speed. To keep one control valve almost fully open at all times can minimize the pump energy (Moore and Fisher 2003; ASHRAE 2007; etc.). To evaluate energy use and economic feasibilities of alternative designs of chilled water pumping systems, simple polynomial models for pumps, motor efficiency and VFD efficiency were proposed by Bahnfleth and Peyer (2001).

This paper presents the online optimal control of the complex central chilling system in a super high-rise building in Hong Kong aiming at improving its energy efficiency and providing better operational performance. Based on testing of interactions within the cooling water system, an optimal control strategy for the cooling water system is developed using a systematic approach by considering the characteristics and interactions among all components concerned and their associated variables including climate variables. A HQS (hybrid quick search) method is developed and used to search for the best setting of the condenser water supply temperature set-point. Since the energy saving potential associated with the optimization of the chilled water supply temperature set-point in this building is very limited, the optimization of the chilled water system is mainly to optimize the operation of variable speed pumps. The optimal control strategies for variable speed pumps with different installations in the secondary chilled water system are developed in order to deliver the desired water flow rate and provide the required pressure head with least energy input and good control stability. These strategies are tested and validated in a virtual environment representing the real building system and its central chilling system prior to site implementation. The implementation guidelines for practical application of these strategies are also addressed.

2. BUILDING AND SYSTEM DESCRIPTIONS

The building concerned is a super high-rise building of approximately 490 m height (currently the tallest building in Hong Kong) and 321,000 m^2 floor areas. Figure 1 presents a rendering of this building, which involves a basement of 4 floors, a block building of 6 floors, and a tower building of 98 floors. The basement is mainly used for car parking with about 24,000 m^2. The block building from the ground floor to the 5th floor mainly serves as the commercial center involving hotel ballrooms, shopping arcades and arrival lobbies. The gross area is about 67,000 m^2. The tower building consists of 230,000 m^2 for commercial offices and a six star hotel on the upmost zone. Considering the usage characteristics of the hotel, separate air-cooled chillers located on the 99th floor are used to provide the cooling source for this part. For the remaining part, the cooling source is provided by the water-cooled chillers on the 6th floor. The schematic of the central chilling system of the remaining part is illustrated in Figure 2.

Figure 1: A rendering of the building concerned

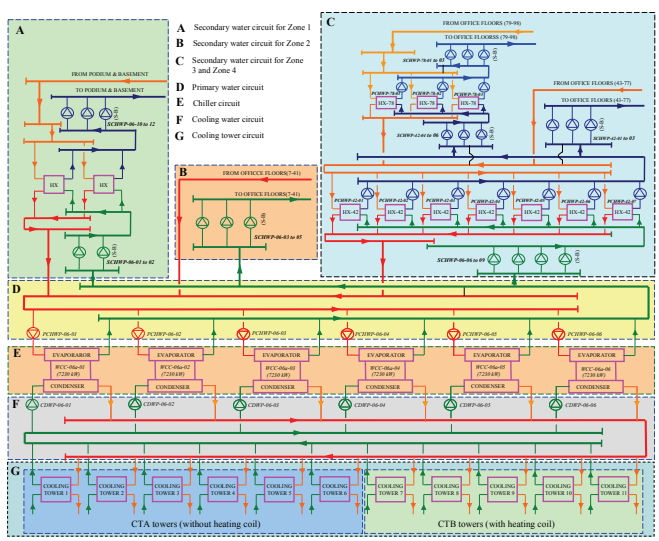

Figure 2: Schematic of the central chilling system

In this central chilling system, six identical high voltage (10,000V) centrifugal chillers with the cooling capacity of 7,230 kW each are used to supply the chiller water at 5.5°C. The nominal power consumption of each chiller is 1,270 kW at the full load condition. Each chiller is associated with one constant condenser water pump and one constant primary chilled water pump. The heat generated by the chiller motors is mostly taken away by the refrigerant. The heat dissipated from the chiller condensers is rejected by means of eleven evaporative water cooling towers with a total design capacity of 51,710 kW. All cooling towers are equipped with VFDs to allow energy efficiency. The chilled water distribution system is divided into four zones, and only Zone 2 (7th - 41st floor), indicated as B in Figure 2, is supplied with the secondary chilled water directly. For other three zones, the heat exchangers are used to transfer the cooling energy from low zones to high zones to avoid the chilled water pipelines and terminal units from suffering extremely high pressure. All pumps in the secondary chilled water system are equipped with VFDs to allow energy efficiency except that the primary chilled water pumps dedicated to the heat exchangers in Zone 3 and Zone 4 are constant speed pumps.

3. TESTING OF INTERACTIONS WITHIN THE COOLING WATER SYSTEM AND CHILLED WATER SYSTEM

In the central chilling system, the cooling water system and chilled water system are both highly interactive, and the fact is that the reduction of the power consumption of one subsystem will result in the increase of the power consumption of the other subsystem. Furthermore, the increase of the power consumption of one subsystem is normally different from the reduction of the power consumption of the other subsystem. They vary with changes of operation conditions. Therefore, testing the effects of interactions within the cooling water system and chilled water system on energy performance is essential for developing online optimal strategies for the central chilling system.

In the cooling water system, the chiller performance and cooling tower performance are affected by the condenser water supply temperature set-point. A lower temperature set-point can improve the COP of chillers, resulting in less power consumption of chillers, but the lower temperature set-point requires more air flow rate to achieve the higher heat rejection efficiency of cooling towers and, hence, more power is consumed by cooling towers. In contrast, a higher temperature set-point can save the power consumption of cooling towers, but it deteriorates the efficiency of chillers, resulting in more power consumption of chillers. Figure 3 illustrates the power consumption profiles of both chillers and cooling towers as well as the total power consumption of chillers and cooling towers using different condenser water supply temperature set-point control in the typical winter case. It can be found that the power consumption of chillers increased, but the power consumption of cooling towers decreased with the increase of the temperature set-point. The optimal temperature set-point is the trade-off between the power consumption of chillers and cooling towers. If taking 25°C as the benchmark of the operation condition of the winter case, about 131 kW energy can be saved when the system operated at the optimal temperature set-point of 19.6°C. Therefore, the energy saving potential related to the optimization of the condenser water supply temperature set-point is distinct for this building and this temperature set-point should be optimized in optimal control strategies.

In the chilled water system, the performances of chiller and chilled water pumps are affected by the chilled water supply temperature set-point. When the evaporating temperature rises as the result of increased chilled water supply temperature set-point, the power consumption of chiller compressors will be reduced as the suction pressure rises. However, as the chilled water supply temperature set-point increases, the heat transfer efficiency of the terminal units will de deteriorated and, hence, more chilled water will be required to compensate the efficiency deterioration, which will increase the power consumption of pumps. In contrast, a lower temperature set-point can save the power consumption of pumps, but it deteriorates the efficiency of chillers, resulting in more power consumption of chillers. Figure 4 shows the power consumption profiles of both chillers and pumps as well as the total power consumption of chillers and pumps using different chilled water supply temperature set-point control in the typical winter case. The results show that the power consumption of chillers was decreased, but the power consumption of pumps was increased with the increase of the chilled water supply temperature set-point. The minimum total power consumption (5131 kW) was achieved when the chilled water supply temperature set-point was about 7.0°C. However, the energy saving (23 kW) was relatively small as compared with the power consumption at the design temperature set-point of 5.5°C. The reasons are that the usage characteristics and load profiles of different zones in this building are different, and the inlet water temperatures of the terminal units in different zones are also significantly different due to the application of heat exchangers. Therefore, the optimal chilled water supply temperature set-point is the trade-off among the power consumption of chillers and pumps in different zones.

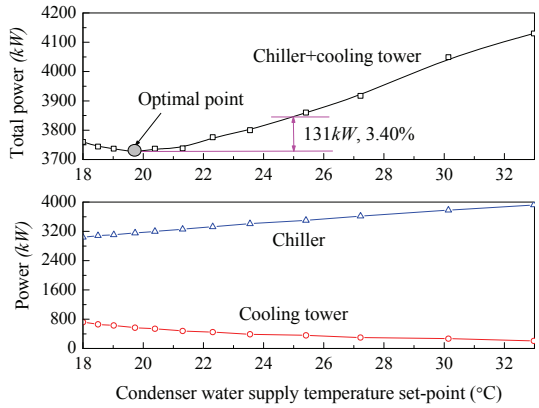

Figure 3: The interaction in the cooling water system.

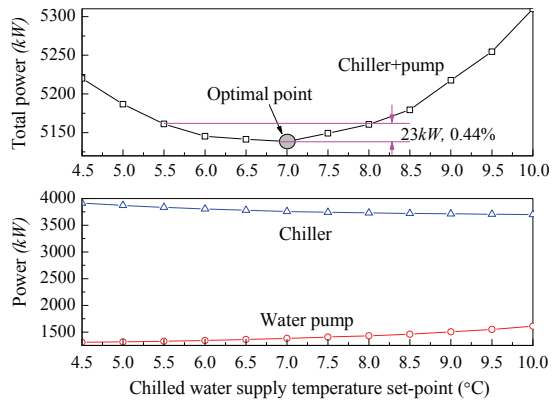

Figure 4: The interaction in the chilled water system.

4. ONLINE OPTIMAL CONTROL OF THE COOLING WATER SYSTEM

4.1 Formulation of the Optimal Control Strategy

Since the optimization of the condenser water supply temperature set-point can provide significant energy savings in this building, an optimal strategy is therefore designed to optimize the operation of the cooling water system. The objective function of this strategy is expressed in equation (1), in which the power consumption of condenser water pumps was not included due to the operation of these pumps dedicated to the operation of individual chillers that they serve. The simplified chiller and cooling tower models are used to predict the power consumption of chillers and cooling towers under any given condition. The input parameters of the chiller model are the chiller evaporator inlet water temperature, condenser water supply temperature, building cooling load, and water mass flow rates in the chiller evaporator and condenser. The output parameter is the chiller power consumption. The input parameters of the cooling tower model are the ambient air wet-bulb temperature, cooling tower inlet water temperature, and water and air mass flow rates through cooling towers. The output parameters are the heat rejection capacity and fan power consumption. The detailed description of both models can be found in Ma *et al*., (2008).

A HQS (hybrid quick search) method as an optimization tool is developed and used to seek the optimal condenser water supply temperature set-point. In this HQS optimization tool, a near optimal strategy, as shown in equation (2), is used to determine a near optimal temperature set-point. This near optimal control setting is then alternatively used as the search center to define a relatively narrow search range, as shown in equation (3). Based on this narrow search range defined, the exhaustive search method is then used to seek the global optimal setting for the given condition with a proper increment. The optimization tool determined by this manner can find the global optimal settings with a reliable manner while still satisfying the requirements and constraints of practical application.

$$\min_{T_{w,cd,\sup}} P_{tot} = P_{ch,tot} + P_{ct,tot} \tag{1}$$

$$T^{n,o}_{w,cd,\sup} = h_0 + h_1 T_{wb} + h_2 \left(\frac{Q_{ev}}{Q_{ev,des}} \right) \tag{2}$$

$$T^{n,o}_{w,cd,\sup} - \Delta t \le T_{w,cd,\sup} \le T^{n.o}_{w,cd,\sup} + \Delta t \tag{3}$$

4.2. Performance Testing and Evaluation

The performance of this strategy was evaluated by comparing with that of a near optimal strategy, as illustrated in equation (2), in terms of the condenser water supply temperature set-point and the entire power consumption of chillers and cooling towers. In this study, the performance of the fixed approach method (approach temperature is 5°C) is used as the benchmark. Figure 5 presents the profile of the condenser water supply temperature set-points obtained by using the proposed strategy and the near optimal strategy for the typical sunny-summer day. It is obvious that the optimal temperature set-points founded by using the proposed strategy were significantly different from the values determined by using the near optimal strategy. It also can be found that the optimal temperature set-points were all within the defined search ranges, which verifies the effectiveness of the HQS optimization tool.

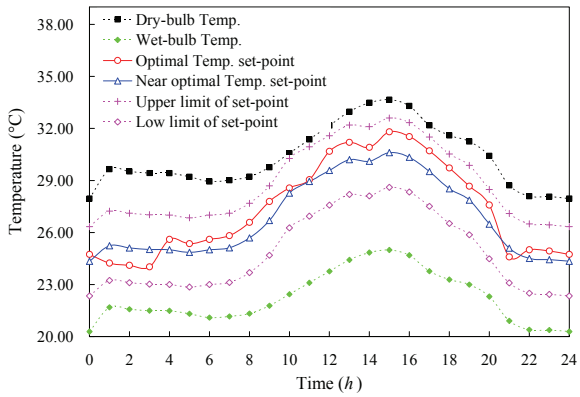

Figure 5: Optimal and near optimal temperature set-points as well as weather conditions

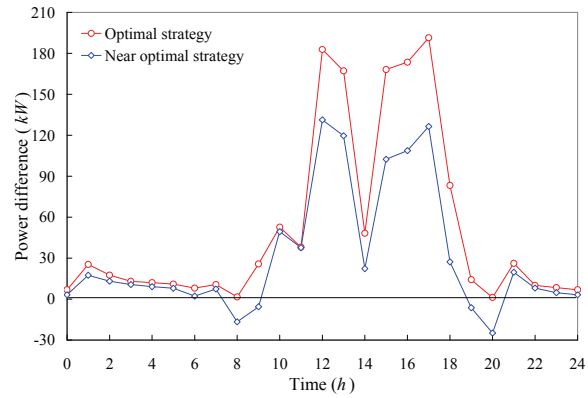

Figure 6: Savings in power consumption of optimal/near optimal strategies compared to the fixed approach method

Figure 6 presents the difference between the hourly-based power consumption using the fixed approach method and the proposed strategy (indicated as optimal strategy in Figure 6), and the difference between the hourly-based power consumption using the fixed approach method and the near optimal strategy in the typical sunny-summer day. It can be seen that the maximum difference between the hourly-based power consumption using the fixed approach method and the proposed strategy was about 191 kW while the maximum difference between the hourly-based power consumption using the fixed approach method and the near optimal strategy was about 126 kW. It also can be found that the near optimal strategy was not always better than the fixed approach method. All these results illustrate that this optimal strategy is much more energy efficient and cost effective than other strategies.

5. OPTIMAL CONTROL STRATEGIES FOR VARIABLE SPEED PUMPS

5.1 Formulation of Optimal Control Strategies

Since the energy saving potential associated with the optimization of the chilled water supply temperature set-point is very limited in this building, the optimization of the chilled water system is therefore to mainly optimize the operation of variable speed pumps in the secondary chilled water system. In this complex central chilling system, all variable speed pumps could be classified into two groups: the pumps distributing water to terminal units and the pumps distributing water to heat exchangers. Although the major functions of both groups of pumps are the same, i.e., to deliver adequate chilled water and provide the required pressure head, their speed control methods are significantly different. The pumps distributing water to terminal units can be controlled through resetting the pressure differential set-point using the positions of all water control valves. The set-point is reset enough and just enough for the most heavily loaded loops. At this situation, the maximum value among the positions of all valves of concern is near fully open. The interested readers can find the detailed reset procedures in ASHRAE (2007). The operating speed of variable speed pumps distributing water to heat exchangers can be controlled using a water flow controller, as shown in Figure 7. In this controller, a water flow meter is installed on the primary side and secondary side of heat exchangers, respectively. The pumps are controlled to maintain the water flow rate in the primary side of heat exchangers equal to the actual flow in the secondary side of heat exchangers.

To develop the sequence strategy for variable speed pumps, a water network pressure drop model that characters the pressure drops on each individual component in the system of concern is developed. Figure 8 presents the structures of this pressure drop model for a subsystem with heat exchangers and for the reverse-return piping system, in which 6 terminal units are illustrated for example. The overall pressure drop of this system, i.e. along the sub-branch F-F_1, can be mathematically described as in equation (4), in which the factors (i.e., φ_1, φ_2, φ_3, φ_4, φ_5) are used to convert the water flow rates in each pipeline section (i.e. A-B, B-C, C-D, D-E, and E-F) as a function of the total subsystem water flow rate. The overall pressure drop includes the pressure drops on the heat exchangers, the fittings around pumps, main supply and return pipelines, and the pressure drop across the sub-branch (i.e. the sub-branch F-F_1), as well as the pressure drops on the pipeline sections of A-B, B-C, C-D, D-E, and E-F. For simplifying the calculations, the factors (φ_1-φ_5) in the same zone can be considered as constant since the usage characteristics and load profiles of each floor in the same zone are similar in this building. The overall pressure drop can therefore be finally expressed as in equation (5). The pressure drop model for other subsystems can be simplified using the same way.

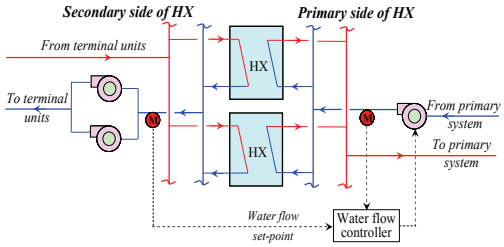

Figure 7: The speed control strategy for variable speed pumps distributing water to heat exchangers

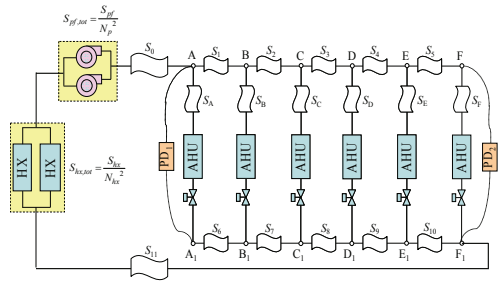

Figure 8: Structure of the water network pressure drop model

$$PD = \frac{S_{hx}}{N_{hx}^2} M_0^2 + \frac{S_{pf}}{N_{pu}^2} M_0^2 + (S_0 + S_{11}) M_0^2 + \left(S_1\phi_1^2 + S_2\phi_2^2 + S_3\phi_3^2 + S_4\phi_4^2 + S_5\phi_5^2\right) M_0^2 + PD_{FF_1} \tag{4}$$

$$PD = \frac{S_{hx}}{N_{hx}^2} M_0^2 + \frac{S_{pf}}{N_{pu}^2} M_0^2 + \left\{(S_0 + S_{11}) M_0^2 + S_{fic} M_0^2\right\} + PD_{FF_1} \tag{5}$$

To predict the power consumption of variable speed pumps under various operation conditions, proper pump models are needed. The pump models used were comprised of polynomials representing head vs. flow and speed, efficiency vs. flow and speed, motor efficiency vs. the fraction of the nameplate brake horsepower, and VFD efficiency vs. the fraction of the nominal speed. The head and efficiency characteristics are based on the manufacturers' data at full speed operation and extended to variable speed operation using pump affinity laws (Bahnfleth and Peyer, 2001).

Based on above water network pressure drop model and pump models, a sequence strategy for variable speed pumps can be developed. Figure 9 outlines the sequence strategy for pumps in the secondary side of heat exchangers in Zone 1. The detailed sequence procedures are as follows: 1) collect online measurements and control signals from the BAS, and send them to a measurement filter. Only validated measurements are used for control supervision; 2) determine the number of heat exchangers operating and the optimal pressure differential set-point at the given chilled water supply temperature set-point using the heat exchanger sequence controller and the pressure differential set-point optimizer (presented previously); 3) If monitored water flow rate in the system is larger than the design water flow of a single pump, two pumps are set to operate, and go to Step 6. Otherwise, go to Step 4; 4) using the pressure drop model and pump models to predict the power consumption of pumps under different operation modes (one pump and two pumps in operation) at the giving water flow and pressure differential set-point; 5) the energy estimation and decision maker then determine the operating number of pumps based on the prediction results of two possible operating modes; 6), the supervisory strategy then provides the final decision considering the operating constraints of practical application. The sequence strategies for other pumps can be formulated using the same way.

In this study, the heat exchangers in Zone 1 are sequenced using the following logic: another heat exchanger is brought online when the water flow rate of each operating heat exchanger exceeds 80% of its design water flow rate. One of the operating heat exchangers is brought offline when the system water flow can be handled by the remaining heat exchangers at 80% or below 80% of their design water flow rates. The heat exchangers in Zone 3&4 are sequenced using a different logic as follows since each heat exchanger is associated with one primary constant pump: another heat exchanger is switched on when the operating heat exchangers are fully loaded and one of the operating heat exchangers is switched off when the remaining heat exchangers can handle the system water flow.

5.2 Performance Testing and Evaluation

The performance of the control strategies for variable speed pumps was tested and evaluated in a virtual environment representing the real building system and its central chilling system. During the tests, the pressure differential set-points for Zone 1 and Zone 2 were bounded between 80 kPa and 215 kPa and between 90 kPa and 230 kPa, respectively, while the pressure differential set-points for Zone 3 and Zone 4 were bounded between 80 kPa and 200 kPa, considering the design cooling loads of each zone and the pump heads at the design condition.

To demonstrate energy saving potentials associated with the use of optimal control strategies, four different strategies were tested and compared. They include the strategy using the fixed pressure differential set-point at the critical loops and the simple sequence strategy (named Strategy #1), the strategy using the fixed pressure differential set-point at the critical loops and the optimal sequence strategy (named Strategy #2), the strategy using the optimal

pressure differential set-point at the critical loops and the simple sequence strategy (named Strategy #3), and the strategy using the optimal pressure differential set-point at the critical loops and the optimal sequence strategy (named Strategy #4). Here, the optimal sequence strategy is the strategy developed. The simple sequence strategy used is presented as follows: bring another pump online when the frequency of the operating pumps exceeds 40 Hz. One of the operating pumps is switched off if the water flow rate and head requirements in the system can be handled by the remaining pumps at the frequency of 40 Hz or below 40 Hz. The fixed pressure differential set-points used for each zone are the upper limits of the pressure differential set-point bounded.

Table 1 summarizes the test results under there typical days (spring day, mild-summer day, and sunny-summer day). It can be found that Strategy #4 using optimal pressure differential set-point and optimal sequence strategy can save a significant amount of energy as compared with other three strategies. Compared with Strategy #1, Strategy #4 saved about 3226.6 kWh (32.43%), 2835.3 kWh (22.49%) and 2023.5 kWh (12.69%) energy in the typical spring day, mild-summer day and sunny-summer day, respectively. It also can be found that Strategy #2 saved about 504.4 kWh (5.07%), 231.5 kWh (1.84%) and 111.3 kWh (0.70%) energy, while Strategy #3 saved about 2405.0 kWh (24.17%), 2169.9 kWh (17.21%), and 1786.4 kWh (11.20%) energy in the typical spring day, mild-summer day and sunny-summer day, respectively, as compared with Strategy #1.

Table 1 Comparison of daily power consumption of variable speed pumps under different control strategies

Control strategies	Spring			Mild-summer			Sunny-summer		
	Power (kWh)	Saving (kWh)	Saving (%)	Power (kWh)	Saving (kWh)	Saving (%)	Power (kWh)	Saving (kWh)	Saving (%)
Strategy #1	9948.7	--	--	12609.2	--	--	15948.6	--	--
Strategy #2	9444.3	504.4	5.07	12377.7	231.5	1.84	15836.9	111.3	0.70
Strategy #3	7543.7	2405.0	24.17	10439.3	2169.9	17.21	14161.8	1786.4	11.20
Strategy #4	6722.1	3226.6	32.43	9773.9	2835.3	22.49	13924.7	2023.5	12.69

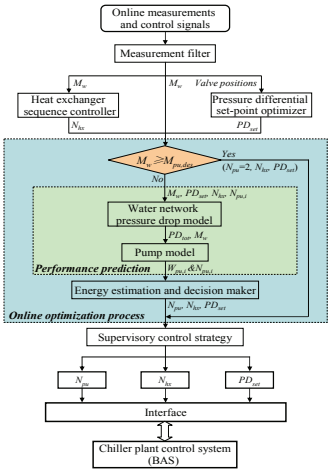

Figure 9: Outline of the sequence strategy

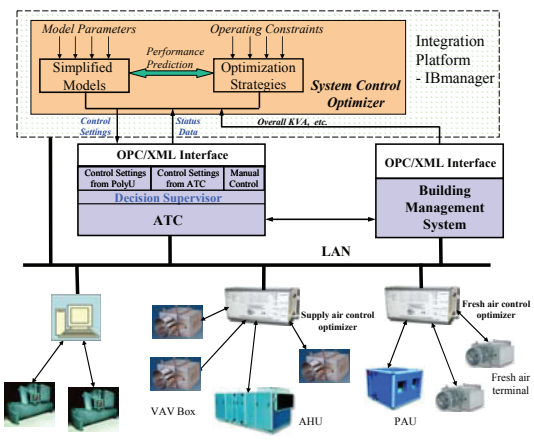

Figure 10: In-situ implementation architecture of the online control software.

6. IMPLEMENTATION PLATFORM

Figure 10 shows the in-situ implementation architecture of the online control strategies. The package of online control strategies developed will run on a separate PC station interfaced with the main station of the chiller control system (BMS). This standalone package will run in parallel with the chiller sequencing program provided by the MVAC&BMS contractors. The contractors provide the protocol or an interface for the communication between this package and the main station of the chiller control system. When the plant (chiller, cooling tower, pump, etc.) sequencing is of concern, this package will only provide the number of them to be operated and the chiller control system will determine which one is used. A decision supervisor in the chiller control system is designed for the operators to set whether the settings given by this package are used or ignored (not used). This implementation approach provides adequate spaces and freedoms for further improving the performance of these optimal control strategies developed to make them have satisfactory performance and be convenient used in practice.

7. CONCLUSIONS

The optimal control strategies for the cooling water system and variable speed pumps are presented in this paper. The optimal control strategy for the cooling water system is developed based on the performance prediction using simplified chiller and cooling tower models aiming at providing stable and accurate performance prediction. This strategy employs a HQS (hybrid quick search) method to seek the most energy efficient control settings within a defined search range determined by a near optimal strategy. The speed of pumps distributing water to terminal units can be controlled using the positions of water control valves and keeping one control valve almost fully open at all times. The speed of pumps distributing water to heat exchangers can be controlled using a water flow controller. The pumps can be sequenced based on the water network pressure drop models and pump models. The test results of these strategies showed that they are more energy efficient and cost effective than other strategies. These strategies are still simple and easy to implement in practice.

NOMENCLATURE

h_0-h_2	coefficients	(-)	des	design
M	mass flow rate	(kg/s)	ev	evaporator
N	number	(-)	fic	fictitious
P	power consumption	(kW)	hx	heat exchanger
PD	pressure differential	(kPa)	pf	pump fitting
Q	heat transfer rate	(kW)	pu	pump
S	flow resistance	(kPa×s^2/l^2)	sup	supply
T	temperature	(°C)	tot	total
Subscripts			w	water
cd	condenser		**Supscripts**	
ch	chiller		n	near
ct	cooling tower		o	optimal

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