Energy efficient control of variable speed pumps in complex building central air-conditioning systems

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ABSTRACT

This paper presents the optimal control strategies for variable speed pumps with different configurations in complex building air-conditioning systems to enhance their energy efficiencies. Through a detailed analysis of the system characteristics, the pressure drop models for different water networks in complex air-conditioning systems are developed and then used to formulate an optimal pump sequence control strategy. This sequence control strategy determines the optimal number of pumps in operation taking into account their power consumptions and maintenance costs. The variable speed pumps in complex air-conditioning systems can be classified into two groups: the pumps distributing water to terminal units and pumps distributing water to heat exchanges. The speeds of pumps distributing water to terminal units are controlled by resetting the pressure differential set-point using the online opening signals of water control valves. The speeds of pumps distributing water to heat exchanges are controlled using a water flow controller. The performances of these strategies are tested and evaluated in a simulated virtual environment representing the complex air-conditioning system in a super high-rise building by comparing with that of other reference strategies. The results showed that about 12–32% of pump energy could be saved by using these optimal control strategies.

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1. Introduction

During the past several decades, the costs of variable speed devices have come down significantly due to the advances in technology, which allowed widespread applications of variable speed pumps in building air-conditioning systems. An increased issue associated with the use of variable speed pumps is the control and optimization of their operation.

In terms of enhancing the operating efficiency, improving the control robustness and prolonging the service life, many practitioners in the control engineering have paid great efforts on energy efficient control and operation of variable speed pumps during the last two decades [1–4]. A number of researchers and experts in the heating, ventilation and air-conditioning (HVAC) field have also devoted considerable efforts on developing and applying proper and optimal control algorithms for variable speed pumps to enhance their energy efficiencies [5–18].

Among the existing studies, Burke [5] pointed out that when pumps operate within ±20% of their best efficiency points, there would seldom have any operation problem. Bernier and Bourret [6]

examined the cumulative effects of the deteriorating values of the motor efficiency and variable frequency driver (VFD) efficiency as the speed of a pump is reduced. The results showed that the power required at the inlet of a pump-motor-VFD set was significantly higher, especially for oversized motors, than the power predicted by using the classic pump affinity laws. The parallel operation of variable speed pumps in chilled water systems was studied by Hansen [7]. The results showed that the benefits gained from the unequal speed operation of parallel pumps were minimal, and all pumps in a given installation were not necessary to be equipped with the speed control devices.

To optimize the speeds of variable speed condenser cooling water pumps, an online optimal control strategy was proposed by Wang and Burnett [8], in which an adaptive and derivative method was used to set the pressure set-point according to the estimated derivative of the total power consumption of chillers and water pumps with respect to the pressure to control the pump speed. Rishel [9–12] and Tillack and Rishel [13] stated that pumps can be sequenced properly based on the wire-to-water efficiency or kW input to the pumping system and the pump speed can be controlled through the use of pressure differential (PD) transmitters located at the critical loops and continuous interrogation of them. In the optimization strategy for air-conditioning systems proposed by Lu et al. [14], an adaptive neuron-fuzzy inference

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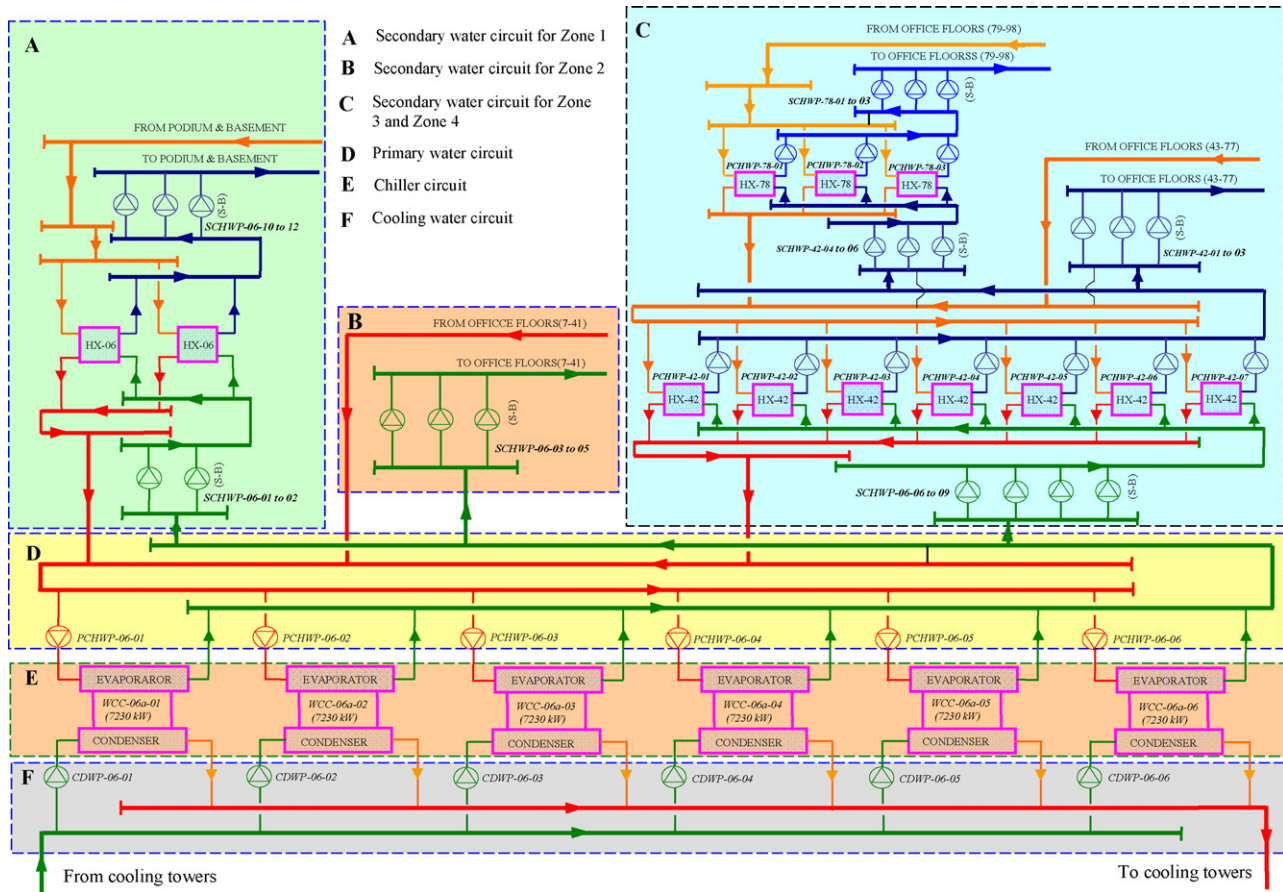E-mail address: beswwang@polyu.edu.hk (S. Wang).

Fig. 1. Schematics of the central chilled water system.

system was used to obtain the optimal pressure differential set-point and model the duct and pipe networks based on the limited sensor information. To evaluate the energy use and economic feasibilities of alternative designs of chilled water pumping systems, a series of polynomials were used by Bahnfleth and Peyer [15,16] to model the performances of variable speed pumps.

Several studies have presented that the valve opening signal can be a valuable tool to optimize the pressure differential set-point to control the pump speed [13,17–19]. In Chapter 41 of the *2007 ASHRAE Handbook—HVAC Applications* [19], it was stated that the speed of the variable speed pump is practically controlled to maintain a constant pressure differential between the main chilled water supply and return pipelines although this approach is far from optimal. The pumps can be sequenced based on continuous monitoring of the outputs of the pump controllers.

It is noted that the pump speed control has been addressed in most of the above studies, while the studies on the pump sequence control still seem inadequate. This is probably due to one pump installation designed in many applications or simple sequence control strategies used in practice. In addition, the research associated with proper control of variable speed pumps in complex air-conditioning systems still seems missing.

This paper therefore aims at developing and addressing optimal control strategies, including the speed control strategy and the sequence control strategy, for variable speed pumps with different configurations in complex building air-conditioning systems for real-time applications. The performances of these strategies are tested and evaluated in a simulated virtual environment representing the complex air-conditioning system in a super high-rise building being constructed in Hong Kong.

2. Building and system descriptions

The building concerned in this study is a super high-rise building of approximately 490 m height and 321,000 m^2 of floor area. This building has a basement of four floors, a block building of six floors and a tower building of 98 floors. Fig. 1 is the schematic of the central chilled water system in this building, in which six identical centrifugal chillers with the capacity of 7230 kW each and the nominal power consumption of 1270 kW each at the full load condition are used to supply the chilled water at 5.5 °C. Each chiller is associated with one constant condenser water pump and one constant primary chilled water pump.

To avoid the chilled water pipelines and terminal units from suffering extremely high pressure, the secondary chilled water system is divided into four zones and only Zone 2 (indicated as B in Fig. 1) is supplied with the secondary chilled water directly. Zone 1 (indicated as A in Fig. 1) is supplied with the secondary chilled water through the heat exchangers located on the sixth floor (i.e., mechanical floor), while the chilled water from chillers serves as the cooling source of the heat exchangers. Zones 3 and 4 (indicated as C in Fig. 1) are supplied with the secondary chilled water through the first stage heat exchangers (HX-42 in Fig. 1) located on the 42nd floor (i.e., mechanical floor). Some of the chilled water after the first stage heat exchangers is delivered to Zone 3 by the secondary chilled water pumps (SCHWP-42-01 to 03) located on the 42nd floor. Some water is delivered to the second stage heat exchangers (HX-78 in Fig. 1) located on the 78th floor (i.e., mechanical floor) by the secondary chilled water pumps (SCHWP-42-04 to 06) located on the 42nd floor. The water system after the second stage heat exchangers is the conventional primary–

Table 1
Specifications of all pumps in the air-conditioning system.

Pumps	Number[a]	M_w (L/s)	H (m)	η (%)	W (kW)	W_{tot} (kW)
CDWP-06-01 to 06	6	410.1	41.60	83.6	202.0	1212
PCHWP-06-01 to 06	6	345.0	31.60	84.5	126.0	756.0
SCHWP-06-01 to 02	1 (1)	345.0	24.60	82.2	101.0	101.0
SCHWP-06-03 to 05	2 (1)	345.0	41.40	85.7	163.0	326.0
SCHWP-06-06 to 09	3 (1)	345.0	30.30	84.2	122.0	366.0
SCHWP-06-10 to 12	2 (1)	155.0	39.90	78.8	76.9	153.8
PCHWP-42-01 to 07	7	149.0	26.00	84.9	44.7	312.9
SCHWP-42-01 to 03	2 (1)	294.0	36.50	87.8	120.0	240.0
SCHWP-42-04 to 06	2 (1)	227.0	26.20	84.3	69.1	138.2
PCHWP-78-01 to 03	3	151.0	20.60	84.3	36.1	108.3
SCHWP-78-01 to 03	2 (1)	227.0	39.20	85.8	102.0	204
Total design power of all pumps (kW)						3,918.2
Total design power of all variable speed pumps (kW)						1529.0

[a] Value in parentheses indicates the number of standby pumps, and W_{tot} did not include the power of standby pumps.

secondary chilled water system. The configuration of the water piping system of this building is the reverse-return system.

All pumps in this air-conditioning system are equipped with VFDs to allow energy efficiency except that the primary chilled water pumps dedicated to the chillers and heat exchangers in Zones 3 and 4 are constant speed pumps. The major specifications of all pumps in this air-conditioning system are summarized in Table 1. The total design power of all pumps is 3918.2 kW, which occupies about 21.04% of the total power load of the overall air-conditioning system. The total design power load of all variable speed pumps is 1529.0 kW, which accounts for about 39.0% of the total power load of all pumps in the air-conditioning system. Therefore, proper control and operation of these variable speed pumps have significant impacts on energy efficiency of buildings besides their designs.

3. Optimal pump speed control strategies

In this complex air-conditioning system, all variable speed pumps can be classified into the pumps distributing water to terminal units and pumps distributing water to heat exchangers. Although the major functions of both groups of pumps are same, i.e., to deliver the adequate chilled water to satisfy the indoor cooling demands, their speed control methods are significantly different.

3.1. Speed control of the pump distributing water to terminal units

In practice, the speeds of variable speed pumps distributing water to terminal units are often controlled to maintain a constant pressure differential between the main chilled water supply and return pipelines or maintain a constant pressure differential at the critical loops. However, both strategies are not optimal and a significant amount of energy might still be wasted, which will be demonstrated later. In this study, an optimal strategy, as illustrated in Fig. 2, was used to control the speeds of pumps distributing water to terminal units. In this strategy, the pressure differential set-point is optimized using a pressure differential set-point optimizer. The detailed optimization process of this optimizer is shown in Fig. 3, which is modified based on the strategy presented in Chapter 41 of the 2007 ASHRAE Handbook–HVAC Applications [19]. At time K, this strategy firstly checks whether the supply air temperature set-points ($T_{set,i}$) of all terminal units of concern are reached or not. Then it will use the maximal opening signal (OS_{max}) among all water control valves and the number of valves (Num) with this maximal opening signal to determine an optimal pressure differential set-point ($PD_{set,k}$) based on the previous setting ($PD_{set,k-1}$) by increasing/decreasing a fixed value ΔP (e.g., 5% of the design value). The pressure differential set-point

determined by this approach is enough and just enough for the most heavily loaded loops. At this situation, one of control valves keeps almost fully open at all times.

For reverse-return systems, as shown in Fig. 2, a pressure differential sensor is installed at each end of the loop. A representative measured pressure differential is selected based on the larger deviation of the measurements of both pressure differential sensors from the set-point optimized, and then used to perform the pump speed control [12,13].

3.2. Speed control of the pump distributing water to heat exchangers

For variable speed pumps distributing water to heat exchangers, their operating speeds can be controlled using a water flow controller, as illustrated in Fig. 4. In this controller, a water flow meter is installed at the primary side and secondary side of heat exchangers, respectively. The pump can then be controlled to maintain the water flow rate in the primary side of heat exchangers equal to the actual water flow rate in the secondary side of heat exchangers. This controller is straightforward and easy to tune in practice.

4. An optimal pump sequence control strategy

The pump sequence control is to determine both the order and point that pumps should be brought online and offline [19]. For small systems where only one pump operates at all times, there is therefore no problem with the pump sequence control. However, for large systems, multiple variable speed pumps with parallel
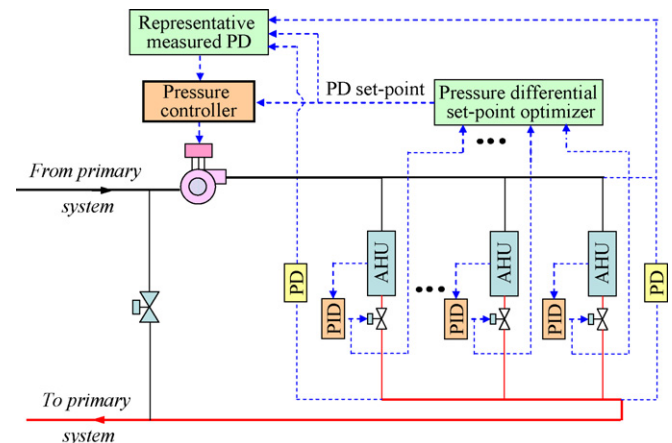

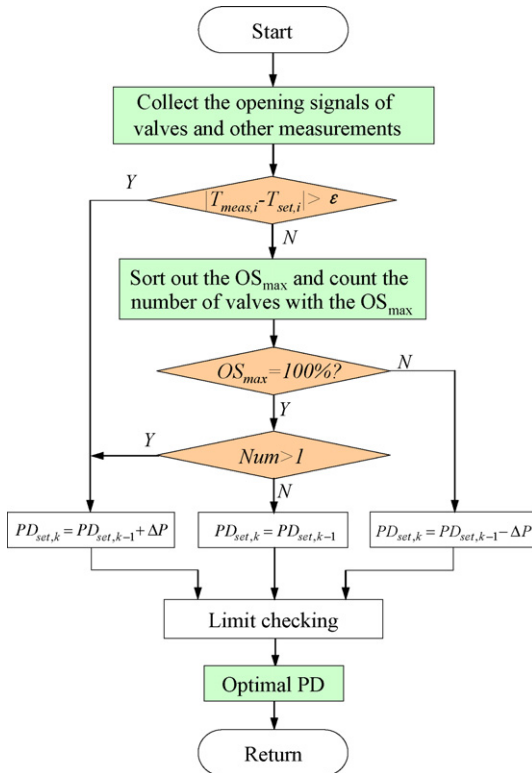**Fig. 2.** The speed control strategy for variable speed pumps distributing water to terminal units.

Fig. 3. The pressure differential set-point optimizer.

Fig. 5. The power consumptions of pumps under various operating combinations.

installations are often designed. These pumps should be sequenced properly to deliver the adequate chilled water to all terminal units with least energy input.

Due to the use of heat exchangers in the complex air-conditioning system to transfer the cooling energy from low zones to high zones to avoid the high water static pressure, the complexity and difficulty associated with the sequence control of variable speed pumps in this system are therefore increased accordingly. To develop an optimal sequence control strategy, the variables affecting the pump sequence control are studied firstly.

4.1. Study on the variables affecting the pump sequence control

In the available literature, the best strategy to sequence control the operation of variable speed pumps is to use the wire-to-water efficiency or kW input [9,12]. In this method, variable speed pumps were finally sequenced as a form of a performance map in terms of
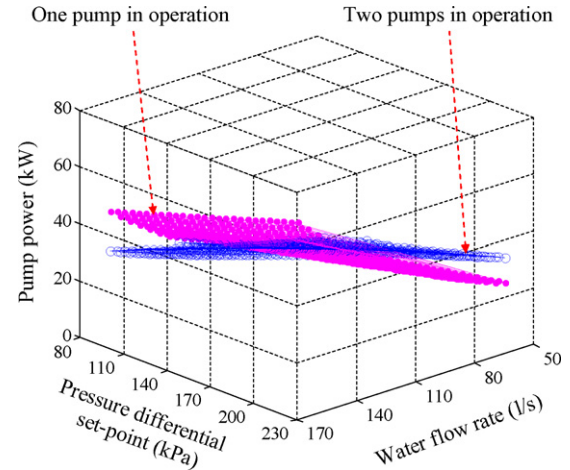
the system water flow rate. The performance map can be generated through complete simulations or by testing the system over significant ranges of settings and operating conditions. However, in this method, the effect of the variation of the pressure differential set-point (if optimized) on the pump sequence control was not taken into account.

In chilled water systems, there are two variables (i.e., the pressure differential set-point and system water flow rate) that might affect the pump sequence control. To study the effect of both variables on the pump sequence control, a simulation study based on one of subsystems with two pumps with a parallel installation in the building under study was conducted. In the simulation, the power consumptions of the pumps under various operating combinations (i.e., different system water flow rates combined with different pressure differential set-points) for different operation modes (one pump in operation and two pumps in operation) were simulated using a pump model and a water network pressure drop model presented in Section 4.4. Fig. 5 gives the simulation results. It can be seen that both the system water flow rate and the value of the pressure differential set-point have impacts on the power consumption of pumps. At a given system water flow rate, the value of the pressure differential set-point directly affects the number of pumps in operation and, therefore affects the power consumption of pumps. It also can be observed that for a given operating combination (i.e., a given pressure differential set-point and a given system water flow rate), the power consumptions of the pumps with different operation modes were different. The optimal operating number of pumps for a given
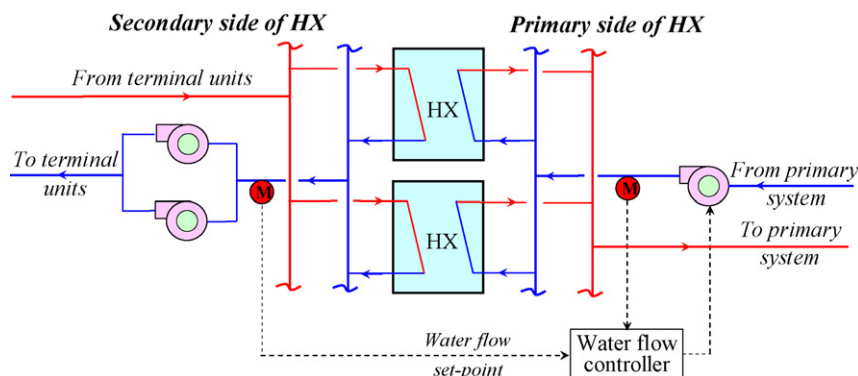

Fig. 4. The speed control strategy for pumps distributing water to heat exchangers.

condition is the one with the minimum total power consumption while still satisfying the system head and water flow rate requirements. Therefore, to develop an optimal pump sequence control strategy, the effect of both the pressure differential set-point (optimized) and system water flow rate on the pump sequence control should be carefully considered.

4.2. Outline of the optimal pump sequence control strategy

Based on the above simulation results, an optimal pump sequence control strategy is developed in the following using a model-based approach by considering the requirements and constraints of practical applications, i.e., control accuracy, control robustness, computational cost, etc.

Since the general sequence control strategy for variable speed pumps at the secondary side of heat exchangers (e.g., the pumps of SCHWP-06-10 to 12 in Fig. 1) could provide some coverage for most of possible sequence control problems for variable speed pumps encountered in the complex air-conditioning system, the sequence control strategy for these pumps is therefore presented in the following in detail, and the sequence control strategies for variable speed pumps with other installations could be simplified based on it. Fig. 6 outlines the detailed sequence control logic, which consists of model-based performance predictors, an energy estimator, a decision maker, a supervisory strategy and a number of local control strategies. The central idea covered in this strategy is to operate pumps as economically as possible considering their power consumptions and related maintenance costs.
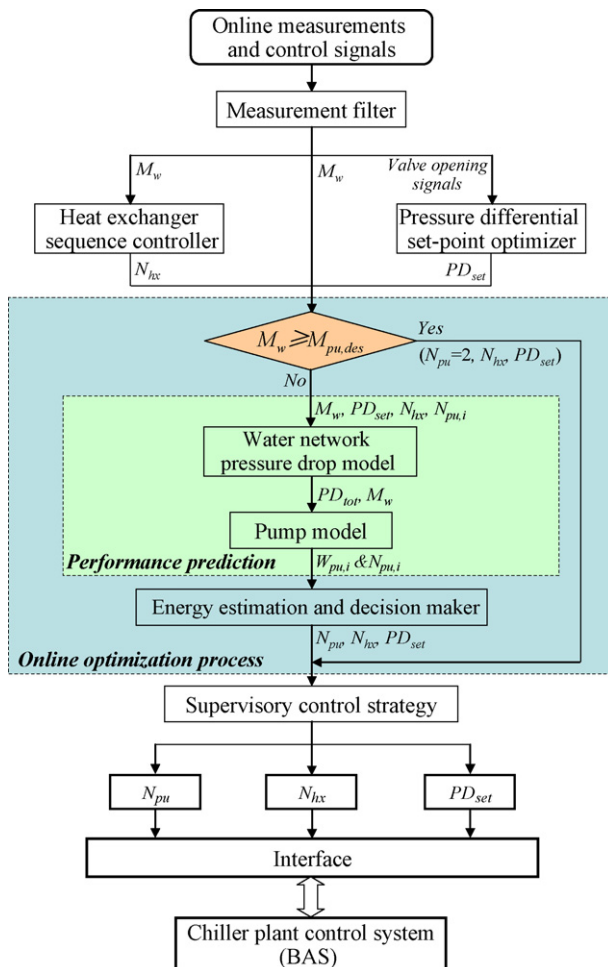

Fig. 6. Outline of the model-based pump sequence control strategy.

In this strategy, a pressure differential set-point optimizer as described in Section 3.1 was used to determine the optimal pressure differential set-point at a given chilled water supply temperature set-point. A heat exchanger sequence control strategy was applied to determine the number of heat exchangers operating. The performance predictor utilized the water network pressure drop models and the pump model to predict the power consumptions of pumps under different possible operating combinations. The detailed descriptions of the performance models and heat exchanger sequence control strategy as well as the optimization procedures of this optimal sequence control strategy are provided in the following along with a brief description of the major functions of the energy estimator, decision maker and supervisory strategy.

4.3. Heat exchanger sequence control strategy

In this complex air-conditioning system, the heat exchangers used can be classified into two groups: one is the heat exchangers in Zone 1 and the other is the heat exchangers in Zones 3 and 4. Since each heat exchanger in Zones 3 and 4 is associated with one constant primary chilled water pump, the operation of these heat exchangers can be sequenced using a simple strategy as follows. Using this strategy, 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 rate.

For the heat exchangers in Zone 1, the sequence control strategy is somewhat different since the water systems in both sides of heat exchangers are equipped with variable speed pumps. Although operating more heat exchangers can reduce the power consumptions of pumps in both sides of heat exchangers to some extent due to the reduction of the total pressure drop on the heat exchangers, however, operating more heat exchangers will increase their maintenance costs and shorten their service life. In addition, the minimal water flow rates in both sides of heat exchangers are inherently constrained by the low bound of the pump operating frequency (e.g., 20 Hz). By taking into account all these factors, a simple sequence strategy was designed for these heat exchangers. Using this strategy, another heat exchanger is activated when the water flow rate of each operating heat exchanger exceeds 80% of its design value. One of the operating heat exchangers is deactivated if the water flow rate in the system can be handled by the remaining heat exchangers at 80% or below 80% of their design values.

4.4. Simplified models

To formulate the optimal pump sequence control strategy, the pressure drop models that characterize the pressure drops on each individual component in the system of concern for different water networks in the complex air-conditioning system are developed. In the complex air-conditioning system, the chilled water network can be sub-divided into the water networks with terminal units and water networks at the primary side of heat exchangers. The pressure drop models for both groups of water networks are different and they are presented in the following in detail along with a brief description of a simple pump model.

4.4.1. Pressure drop model for the water network with terminals units

In the complex air-conditioning system, some of water networks with terminals units are supplied with the secondary chiller water through heat exchangers (i.e., the water network at the secondary side of heat exchangers in Zone 1), while some of them are supplied with the secondary chilled water from chillers directly (i.e., the water network in Zone 2). Since the water
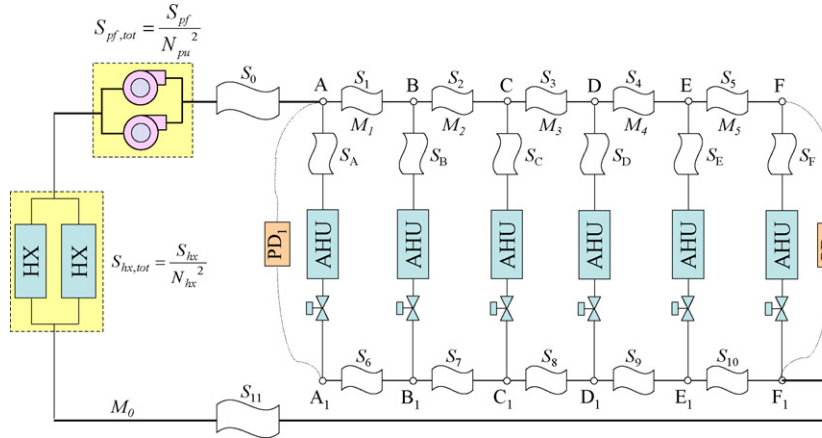
Fig. 7. Structure of the pressure drop model for the water network with terminal units.

networks at the secondary side of heat exchangers could be the most complicated systems, other water networks with terminal units can be considered as the simplifications of such networks. Therefore, a pressure drop model for such networks is presented in this study in detail.

Fig. 7 presents the simplified structure of the pressure drop model for the water network at the secondary side of heat exchangers in Zone 1, in which only six 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 Eq. (1), which includes the pressure drop on the heat exchangers (including the pressure drops on the heat exchangers and on the headers that direct the flow into and from each heat exchanger), the pressure drop on the fittings around pumps (including the pressure drop on the headers that direct the flow into and from each pump and the pressure drop on the valves in the pump headers), the pressure drops on main supply and return pipelines, the pressure drop across the sub-branch (i.e., F–F_1) and the pressure drops on the pipeline sections of A–B, B–C, C–D, D–E and E–F.

Eq. (1) can be further re-written as in Eq. (2) by representing the water flow rate in each pipeline section (i.e., A–B, B–C, C–D, D–E and E–F) as a function of the total system water flow rate by multiplying a converting factor (i.e., φ_1–φ_5). For simplifying the calculations, the factors (φ_1–φ_5) in the same zone can be considered as constants under various load conditions since the usage characteristics and load profiles of each floor in the same zone are similar in the building under study. The overall system pressure drop can therefore be finally expressed as in Eq. (3).

$$PD = \frac{S_{hx}}{N_{hx}^2} M_0^2 + \frac{S_{pf}}{N_{pu}^2} M_0^2 + (S_0 + S_{11})M_0^2 + S_1 M_1^2 + S_2 M_2^2 + S_3 M_3^2$$
$$+ S_4 M_4^2 + S_5 M_5^2 + PD_{FF_1} \tag{1}$$

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

$$PD = \frac{S_{hx}}{N_{hx}^2} M_0^2 + \frac{S_{pf}}{N_{pu}^2} M_0^2 + \{(S_0 + S_{11})M_0^2 + S_{fic}M_0^2\} + PD_{FF_1} \tag{3}$$

where PD is the pressure drop, M is the water flow rate, N is the operating number of equipment, S is the flow resistance, and subscripts hx, pu, pf and fic represent heat exchanger, pump, pump fitting and fictitious, respectively.

The last term on the right-hand side of Eq. (3) is the optimal pressure differential set-point (named PD_{set}) at a given chilled water supply temperature set-point. The other three terms are the pressure drops across the heat exchangers, the fittings around pumps and the distribution pipelines, respectively. These three pressure drops are named piping pressure drop (PD_{piping}). The overall system pressure drop can therefore be divided into two parts, i.e., PD_{set} and PD_{piping}, as shown in Fig. 8. For a given condition, the pump operation point is the intersection of both the pump curve and control curve. It is worth-noticing that both PD_{set} and PD_{piping} in Fig. 8 vary with the changes of working conditions. The piping head loss curve is not constant. It varies with the changes of the number of pumps in operation and/or the number of heat exchangers in operation.

For online applications, there are five parameters (S_{hx}, S_{pf}, S_0, S_{11} and S_{fic}) required to be identified. In fact, the parameters of S_0, S_{11} and S_{fic} can be integrated as a single parameter ($\bar{S}_{fic}$) to represent the flow resistance on the distribution pipelines, including the main supply and return pipelines and the pipeline sections of A–B, B–C, C–D, D–E and E–F. Eq. (3) can therefore be further expressed as in Eq. (4). The parameters of S_{hx} and S_{pf} can be determined using the measured pressure drops across the heat exchanger sub-circuit and pump fitting sub-circuit together with the measured system water flow rate. The parameter of $\bar{S}_{fic}$ can be determined using the system water flow rate, pump head and the pressure differential set-point at the design condition as well as the identified
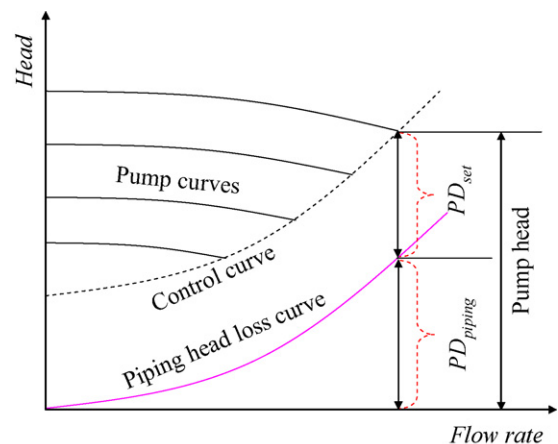

Fig. 8. Schematics of the system control curve and piping head loss curve as well as pump curves for the water network with terminal units.
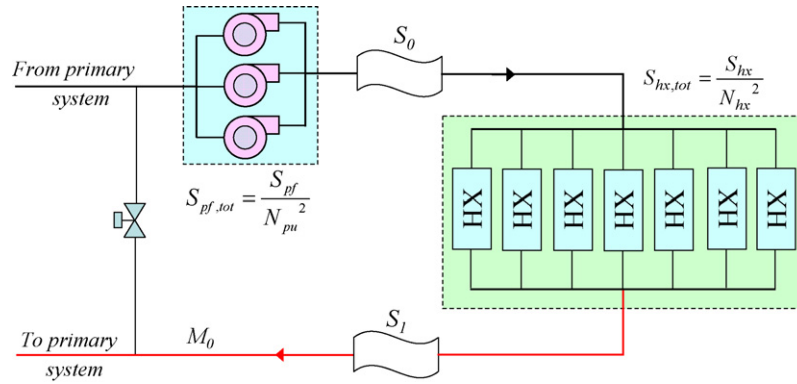
Fig. 9. Structure of the pressure drop model for the water network at the primary side of heat exchangers.

parameters of S_{hx} and S_{pf}.

$$PD = \frac{S_{hx}}{N_{hx}^2} M_0^2 + \frac{S_{pf}}{N_{pu}^2} M_0^2 + \bar{S}_{fic} M_0^2 + PD_{FF_1} \quad (4)$$

It is worthwhile to point out that the above water network pressure drop model is only feasible for reverse-return piping systems. The water network pressure drop model for direct-return piping systems is somewhat different and could be more complex.

4.4.2. Pressure drop model for the water network at the primary side of heat exchangers

The pressure drop model for the water network at the primary side of heat exchangers is relatively simple as compared with that for the water networks with terminal units. As the water network at the primary side of the first stage heat exchangers in Zones 3 and 4 is an example, the pressure drop model for this water network can be simplified as in Fig. 9. The overall pressure drop for this network can be described as in Eq. (5), which consists of the pressure drop on the heat exchangers, the pressure drop on the fittings around pumps and the pressure drops on main supply and return pipelines.

The schematics of the system control curve and pump curves for this network are illustrated in Fig. 10. It is noted that the control curve in Fig. 10 also varies with the changes of the numbers of pumps and/or heat exchangers operating. For online applications, there are four parameters (S_{hx}, S_{pf}, S_0, and S_1) required to be identified. The methods used to determine these parameters are similar to that used to determine the parameters

in Eq. (4).

$$PD = \frac{S_{hx}}{N_{hx}^2} M_0^2 + \frac{S_{pf}}{N_{pu}^2} M_0^2 + (S_0 + S_1) M_0^2 \quad (5)$$

4.4.3. Pump model

For variable speed pumps, the equipment efficiency, namely wire-to-water efficiency, is often used to characterize how much energy applied to a pump-motor-VFD set results in useful energy to deliver the water. A typical pump-motor-VFD set consists of three sub-efficiencies, including the pump efficiency (η_{pu}), motor efficiency (η_m) and VFD efficiency (η_v). These three sub-efficiencies should be involved in the model of variable speed pumps.

In this study, the performance of variable speed pumps was modeled using a series of polynomial approximations [15,16]. They are comprised of polynomials representing head versus flow and speed, and efficiency versus flow and speed. The head and efficiency characteristics are based on the manufacturers' data at the full speed operation and extended to the variable speed operation using the pump affinity laws. The motor efficiency (η_m) is modeled using Eq. (6), which is a function of the fraction of the nameplate brake horsepower, and the VFD efficiency is modeled using Eq. (7), which is a function of the fraction of the nominal speed [6]. The power input to a pump-motor-VFD set is computed using Eq. (8). The coefficients in these polynomials can be regressed using the pump performance data or performance curves, the motor efficiency curve and VFD efficiency curve provided by the manufacturers.

$$\eta_m = c_0(1 - e^{c_1 x}) \quad (6)$$

$$\eta_v = d_0 + d_1 x + d_2 x^2 + d_3 x^3 \quad (7)$$

$$W_{pu,in} = \frac{M_w \times H_{pu} \times SG}{102 \times \eta_v \times \eta_m \times \eta_{pu}} \quad (8)$$

where H is the pump head, η is efficiency, SG is the specific gravity of the fluid being pumped, x is the fraction of the nameplate brake horsepower or the nominal speed, c_0–c_1 and d_0–d_3 are coefficients, and subscripts m, v and in represent motor, VFD and input, respectively.

4.5. Description of the detailed optimization procedures

At each given condition, this model-based sequence control strategy seeks the optimal number of pumps operating with the following procedures (also illustrated in Fig. 6).
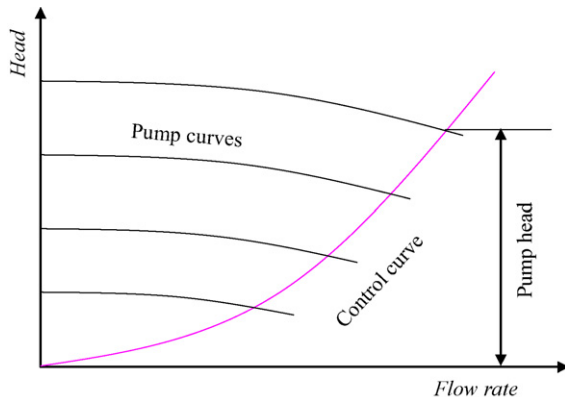

Fig. 10. Schematics of the system control curve and pump curves for the water network at the primary side of heat exchangers.

(1) Validate the measurements and control signals using a filter,
(2) Determine the number of heat exchangers operating and the optimal pressure differential set-point at the given chilled water supply temperature set-point,
(3) If the monitored water flow rate in the system is larger than the design value of a single pump, two pumps are set to operate, and go to Step 6. Otherwise, go to Step 4,
(4) Predict the power consumptions of pumps under different operation modes at the given water flow rate and the pressure differential set-point optimized,
(5) Determine the optimal number of pumps operating using the energy estimator and decision maker, based on the prediction results of two possible operating modes considering their power consumptions and associated maintenance costs. If the difference of the power consumptions between two operation modes is less than a value (e.g., 5% of the design value), one pump is set to operate to reduce the maintenance costs of pumps in the long run. Otherwise, two pumps are set to operate to reduce the power input,
(6) The supervisory strategy then provides the final decision, including the number of pumps operating, the number of heat exchangers operating and the optimal pressure differential set-point taking into account the operating constraints for practical applications.

5. Performance tests and evaluation of the optimal control strategies

Since the building under study is still being constructed, the performances of these optimal control strategies were tested and evaluated in a simulated virtual environment representing the complex air-conditioning system in the super high-rise building studied [20].

5.1. Set up the tests

During the tests, the predetermined cooling loads of each individual zone and weather data were provided to the simulated virtual environment by a data file. The hourly based annual cooling load profiles of each individual zone of this building were simulated using EnergyPlus based on the design data and hourly based weather data of the typical year in Hong Kong. The input frequency to variable speed pumps was bounded between 20 Hz and 50 Hz. The pressure differential set-points for Zones 1 and 2 were bounded between 80 kPa and 215 kPa and between 90 kPa and 230 kPa respectively, while the pressure differential set-points for Zones 3 and 4 were bounded between 80 kPa and 200 kPa considering the design cooling loads of each individual zone and the pump heads at the design condition. The operation of the constant primary chilled water pumps associated with the chillers and heat exchangers in Zones 3 and 4 was dedicated to the operation of their correlated chillers or heat exchangers that they serve. The chillers were sequenced using the conventional sequence control strategy based on their design cooling capacities.
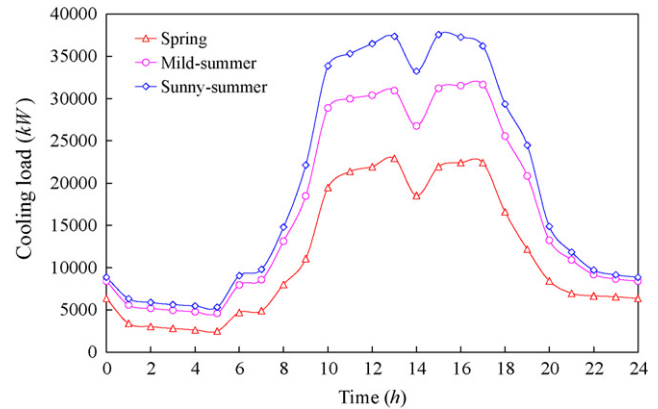

Fig. 11. Building cooling load profiles in the three selected typical days.

Three typical days, which represent the typical operating conditions of the air-conditioning system in the typical spring, mild-summer and sunny-summer days respectively, were selected to test and evaluate the energy performances of the proposed optimal control strategies. Fig. 11 presents the building cooling load profiles in the three selected typical days.

5.2. Results of tests and evaluation

To demonstrate the energy saving potentials associated with the use of the optimal control strategies for variable speed pumps, four different strategies were tested and compared in the following studies. They include the strategy using the fixed pressure differential set-point at the critical loops and the simple pump sequence control strategy (namely Strategy #1), the strategy using the fixed pressure differential set-point at the critical loops and the optimal pump sequence control strategy (namely Strategy #2), the strategy using the optimal pressure differential set-points at the critical loops and the simple pump sequence control strategy (namely Strategy #3), and the strategy using the optimal pressure differential set-points at the critical loops and the optimal pump sequence control strategy (namely Strategy #4). The simple pump sequence control strategy used is a conventional strategy as follows: bring another pump online when the frequencies of the operating pumps exceed 45 Hz. One of the operating pumps is switched off if the remaining pumps can handle the system water flow rate and head requirements at the frequency of 45 Hz or below 45 Hz. The fixed pressure differential set-points used for each individual zone were the upper limits of the pressure differential set-points constrained.

Table 2 summarizes the daily power consumptions of all variable speed pumps in the secondary chilled water system in that building in the three typical days under different control and operation strategies. Among these four strategies, Strategy #4 using the optimal pump speed control and optimal pump sequence control is more energy efficient and cost effective. Compared with

Table 2
Comparison of the daily power consumptions of variable speed pumps under different control and operation 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	–	–	15948.2	–	–
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

Strategy #1 using the conventional control strategies, Strategy #4 using optimal control strategies 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.

From Table 2, it also can be observed that the strategies using the optimal pressure differential set-points (Strategies #3 and #4) can save significantly more energy as compared with the strategies using the fixed pressure differential set-points (Strategies #1 and #2). Compared with Strategy #1 using the fixed pressure differential set-point, Strategy #3 using optimal pressure differential set-points 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. In the strategies using the fixed pressure differential set-point, to maintain a constant pressure differential set-point with the changing flow, the control valves for terminal units must be closed as the load is reduced, resulting in an increase in the flow resistance. Therefore, an additional energy is wasted at part-load conditions, especially at light-load conditions. However, in the strategies using the optimal pressure differential set-points, the pressure differential set-point can be lowered when the load is reduced, which therefore minimizes the system flow resistance and reduces the power consumption of pumps.

From Table 2, it also can be found that the energy saving potential related to the use of the optimal pump sequence control strategy was relatively small as compared with that of using the simple pump sequence control strategy. Compared with Strategy #1 using the simple pump sequence control strategy, Strategy #2 using the optimal pump sequence control strategy saved about 504.4 kWh (5.07%), 231.5 kWh (1.84%) and 111.3 kWh (0.70%) energy in the typical spring day, mild-summer day and sunny-summer day, respectively. This part of energy savings is achieved due to the consideration of the effects of the variations of the pressure differential set-point and system water flow rate on the pump sequence control simultaneously.

Based on the above studies, it can be concluded that the qualities of both the pump speed control and pump sequence control significantly affect the actual energy consumption of variable speed pumps. Strategy #4 using the optimal pump speed control and optimal pump sequence control simultaneously can save more energy than the other three strategies. It is worthy noticing that, in these four control strategies (Strategies #1 to #4), the speeds of variable speed pumps at the primary side of heat exchangers were all controlled using the same water flow controller. Therefore, these pumps in Strategy #1 used as the baselines in this study have already been optimized. The actual energy savings associated with the use of Strategy #4 could be more than the energy savings presented above since the control strategies utilized in practice might be simpler and more inadequate.

6. Conclusions

This paper presents the optimal control strategies, including the speed control strategy and the sequence control strategy, for variable speed pumps with different configurations in complex building air-conditioning systems to enhance their energy efficiencies. The results obtained from the performance tests and evaluation showed that a significant amount of energy in the complex air-conditioning system can be saved when applying the control strategy (Strategy #4) using the optimal pressure differential set-points at the critical loops and optimal pump sequence control as compared with the other three reference control strategies, i.e., the strategy using the fixed pressure differential set-point at the critical loops and the simple pump sequence control (Strategy #1), the strategy using the fixed pressure differential set-point at the critical loops and the optimal pump sequence control (Strategy #2), and the strategy using the optimal pressure differential set-points at the critical loops and the simple pump sequence control (Strategy #3). Compared with Strategy #1, Strategies #4, #3 and #2 saved about 12.69–32.43%, 11.20–24.17% and 0.70–5.07% of the total energy consumption of all variable speed pumps in the complex air-conditioning system, respectively.

It is worth-noticing that the optimal control strategies presented in this paper can be easily adapted to, but not limited to, control of variable speed pumps in most of HVAC water systems. These optimal control strategies are still simple and easy to implement in practice. They are being implemented in the super high-rise building for field validation.

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