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In-situ Implementation and Evaluation of An Online Robust Pump Speed Control Strategy for Avoiding Low Delta-T Syndrome in Complex Chilled Water Systems of High-rise Buildings

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Abstract:

The low delta-T syndrome is one of the major faults that affect the operation and energy performance of the chilled water systems in practice, particularly for the complex chilled water systems. Low delta-T syndrome refers to the situation where the measured mean temperature difference of the overall terminal air-handling units is much lower than the expected normal value. The conventional pump speed control strategies lack the ability to handle the low delta-T syndrome. This paper presents an online robust control strategy for practical applications to avoid the low delta-T syndrome for chilled water systems including complex systems. On top of the conventional control strategies, a temperature set-point reset scheme is developed aiming at providing the reliable temperature set-point for enhancing the operation reliability of chilled water pumps. In addition, a flow-limiting control scheme is employed to perform the function of actively eliminating the deficit flow in the bypass line by a feedback mechanism. This robust pump speed control strategy has been implemented and evaluated on a real complex chilled water system in a high-rise building. The site test results show that the temperature set-point given by the proposed strategy is reliable and the system temperature difference is significantly raised by eliminating the deficit flow problem. When compared to the conventional control strategies, 78% of the total chilled water pump energy was saved in the test period. The actual pump energy saving percentage could be 39% in a year after implementing the robust control strategy in the studied system.

Keywords: *Low delta-T syndrome; chilled water system; pump speed control; building energy.*

1. Introduction

Heating, ventilating and air-conditioning (HVAC) systems are the major energy consumers in commercial buildings. Statistical data shows that in the U.S, nearly 47% of building energy was consumed by space heating and cooling [1]. During the last two decades, a large number of studies have been conducted to enhance the energy performance of buildings and building HVAC systems [2-7]. Currently, the primary constant-secondary variable chilled water system is still the popular configuration of the HVAC system in commercial buildings, especially in the high-rise buildings, due to its higher energy efficiency than the traditional constant flow system. A typical primary-secondary chilled water distribution system consists of two loops: the primary loop and the secondary loop. In the primary loop, each chiller is associated with a constant speed primary pump to ensure the constant flow through the individual chiller. In the secondary loop, variable speed pumps are employed to supply the chilled water according with the cooling demands of the terminal Air-Handling Units (AHUs). A bypass pipeline decouples the two loops.

While in industrial practice, most of the primary-secondary systems, from time to time, do not work as efficiently as anticipated due to the low temperature difference syndrome (i.e. low delta-T syndrome). Low delta-T syndrome is a serious operation problem that is described in detail in [8]. It refers to the situation where the mean water temperature difference produced by the overall AHUs is much lower than its normal value expected. Since the cooling coils of the AHUs are selected to produce a temperature rise at full load that is equal to the temperature differential selected for the chillers. The flow rate of secondary loop should be therefore equal to that of the primary loop under full load condition and should be less than that of primary loop under part load conditions. When a serious low delta-T syndrome occurs, the chilled water of the secondary loop is significantly over-supplied when compared to the normal demand. Once the secondary loop water flow rate exceeds that of the primary loop, the water flow in the bypass line would flow in a reverse direction (i.e. from return side to supply side), which is called the deficit flow.

A series of operational problems would be caused by the low delta-T syndrome and the deficit flow problem, such as the high supply water temperature, the over-supplied chilled water, and the increased energy use of the secondary pumps. If such phenomenon cannot be eliminated, a vicious circle in the secondary loop may be caused. It means that, when the deficit flow occurs, the main supply water temperature is increased because the return water mixes with the supply water. The higher supply water temperature will consequently leads to an increased chilled water demand of the secondary loop, which further worsens the deficit flow.

The low delta-T syndrome not only has significant effects on the pumps but also may affect the energy performance of chillers if chillers are sequenced based on the flow rate in the bypass line. When serious low delta-T syndrome causes the deficit flow in the bypass line, an additional chiller has to be switched on although it is not necessary. The chillers would operate under low load ratio, which significantly degrades the chiller COP (coefficient of performance). Ma [9] conducted simulation studies to test the effect of low delta-T syndrome on the system energy performance by introducing air-side fouling. The results showed that the energy penalty for chillers and the overall chilling system could be 12.7% and 21.9% respectively when low delta-T syndrome was caused under 30% air-side fouling conditions. Wang [10] presented an approach that experimentally validated the feasibility of using a check valve in the chilled water bypass line to solve the low delta-T syndrome. Results showed that about 9.2% of total energy consumption of the chillers and secondary water pumps was saved due to the elimination of the deficit flow. In the last two decades, the low delta-T syndrome was frequently studied. Kirsner [11] pointed out that the low delta-T chilled water syndrome exists in almost all large distributed chilled water systems. Possible reasons of the low delta-T syndrome have been studied. In [12, 13], the possible reasons for the low delta-T syndrome are presented, which include use of tree-way valves, improper set-point of the supply air temperature for AHUs, mismatched design conditions, improper coil selection, improper control valve selection, coil fouling, laminar coil flow, improper sensor calibration, system pressure difference above the valve shutoff head and the occurrence of the deficit flow, etc. Taylor [13] stated that some causes cannot be avoided, such as reduced coil effectiveness, outdoor air economizers and 100% outdoor air systems. While some causes can be avoided through careful calibrations and commissioning, such as improper set-point or controls calibration, the use of three-way valves, improper coil and control valve selection, no control valve interlock, and uncontrolled process load, etc.

Some studies focused on dealing with the low delta-T syndrome and deficit flow problem. Among the studies, Fiorino [14] indicated strongly that a higher delta-T could be achieved by proper design, such as proper application of cooling coils, control systems, distribution pumps, and piping systems. Up to 25 methods concerning design are recommended to achieve high chilled water delta-T ranging covering component selection, sensor calibration, and configurations of distribution systems, etc. In [15], use of pressure independent control valves was proposed to replace the conventional pressure dependent control valves to overcome the low delta-T problem in primary-secondary systems. Aevery [16, 17] recommended that a check valve can be installed in the plant bypass line to deal with the low delta-T syndrome. A check valve was installed in a real cooling plant for system retrofits and upgrading. The actual operation results showed that as much as 20% chiller plant energy and 28% annual chiller utilization hours were reduced due to the inclusion of the check valve as compared to that without using the check valve. It can be found from the above existing studies that the low delta-T syndrome widely existed in the primary-secondary chilled water system and the elimination of this problem can improve the energy efficiency of the chilled water system. However, most of the studies focused on analyzing the possible causes and solutions of this problem from the viewpoint of design and commissioning. In practice, even the HVAC systems were properly designed and well commissioned, the low delta-T syndrome still could not be completely avoided in the operation period due to the improper control strategies or unreliable control settings. Particularly, the low delta-T syndrome occurs more easily in the high-rise buildings due to the complex configurations of the secondary chilled water loops. In order to deal with the high static pressure, the secondary chilled water loop in the high-rise buildings is often divided into several sub-zones vertically. Plate heat exchangers are employed to transfer the cooling energy from chillers or the lower levels to higher levels. In the previous studies, the authors [18] presented a case study on diagnosing the low delta-T problem resulted from the deficit flow that frequently occurred in the chilled water system of a newly-built super high-rise building. The on-site test showed that 75% of the pump energy could be wasted when serious deficit flow occurred. The improper set-point reset of outlet water temperature on the secondary sides of heat exchangers were finally to be determined as the major fault that caused the deficit flow problem.

Conventionally, the set-point of the outlet water temperature at the secondary sides of heat exchangers is a fixed value or is set with a fixed rise above the chiller supply water temperature, shown as Fig. 1. This conventional temperature set-point reset scheme is not robust for the speed control of pumps distributing water from chillers to the heat exchangers. When the system is suffering from significant disturbances such as sudden cooling load increase (e.g. during the morning start period) or degraded system (e.g. fouling), the measured temperature often could not reach this temperature set-point. The pumps are then over-speeded and the deficit flow problem is easily triggered.

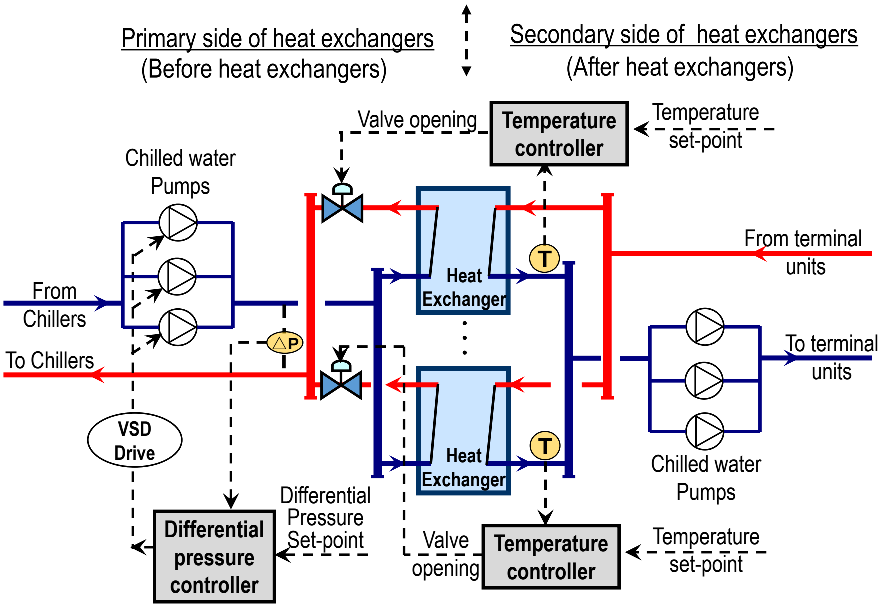

Fig. 1 Conventional speed control for chilled water pumps (pressure-based)

Therefore, for the chilled water system in high-rise buildings, proper measures are needed to enhance the operation reliability of the chilled water pumps with consideration of handling the deficit flow problem and the low delta-T syndrome. A previous study of the authors [19] developed a fault-tolerant control strategy for a typical primary-secondary chilled water pump system to mitigate the low delta-T problem. The strategy performs the real-time modulation of the pressure set-point for secondary pump speed control, aiming at ensuring the water flow of secondary loop not exceed that of the primary loop while still maintaining highest possible delivery capacity of cooling to terminals. However, this work only focuses on the typical and simple primary-secondary system excluding plate heat exchangers. The another study of the authors [20] proposed a model-based optimal control strategy for real-time control of complex chilled water systems involving intermediate heat exchangers to enhance operation and energy performances. Although the simulation test results show that this strategy could enhance the control reliability and help to avoid the occurrence of the deficit flow problem, the main point of this work focuses on searching the optimal control settings for energy efficient of the pumps. Due to the heavy computational burden of the complex models involved and need of a lot of real-time measurements, this optimal control strategy was only evaluated in a simulation platform and has not been fully tested and evaluated on real systems.

It can be concluded that the conventional speed control strategies for the chilled water pumps associated with heat exchangers are not reliable and the low delta-T syndrome and deficit flow problem are easily caused. Although the author's previous works presented a fault-tolerant control strategy and a model-based optimal control strategy for pump speed control, the proposed control strategies were relatively complex and a lot of real-time measurements are required as inputs, which prevent the wide implementations in practice (e.g. information-poor buildings). For the majority of existing high-rise buildings, there is a lack of simple solutions, which have been practically validated and require as few real-time measurements as possible, to improve the performance of the conventional control strategies for avoiding the low delta-T syndrome and the deficit flow problem. This paper presents a simplified robust control strategy for real-time control of the chilled water distribution system, aiming at the convenience of practical implementations in high-rise building HVAC systems. On top of the conventional control strategies, the proposed control strategy employs a temperature set-point reset scheme and a flow-limiting control scheme to enhance the operation reliability of the secondary chilled water pumps for avoiding the low delta-T syndrome and the deficit flow problem. The proposed control strategy has been tested and evaluated by field tests in a real complex chilled water system of a high-rise building in Hong Kong. Detailed field test results are presented to show how the proposed robust control strategy can handle the low delta-T syndrome and to what extant the pump energy performance can be enhanced.

Three major advances are involved in this strategy. First, a temperature set-point reset scheme is developed to enhance the operation reliability and the energy performance of the chilled water pumps, which is a passive solution to avoid the occurrence of the deficit flow when the system is suffered from the significant disturbances and faults. Second, a flow-limiting scheme is developed to reset the pump speed by means of a feedback mechanism, which is a proactive solution to actively eliminate the measured deficit flow once detected. Third, the overall robust control strategy is designed to be simple with a need of less real-time measurements, aiming at convenient implementations in the real applications. Particularly, it can work together with the conventional pump speed control and enhance their performances.

This paper is organized as follows: Section 2 presents the methodology involving the descriptions of the conventional pumps speed control strategy and the proposed robust control strategy. Section 3 describes the studied platform for in-situ test and setup of the field tests. Section 4 show the in-situ test and validation results. Conclusion is given in the last section.

2. Methodology

A robust control strategy for the speed control of chilled water pumps is developed, aiming at improving the conventional control strategy for avoiding the deficit flow problem and enhancing the system temperature difference. The robust control strategy consists of two control schemes developed on top of the conventional control strategies to enhance the operation performance.

In the following parts, the conventional control strategies will be firstly presented and the related operation problems will be discussed simultaneously. Then, the proposed robust control strategy is presented with detailed working principles.

2.1 Conventional pump speed control strategy and the related operation problem

In practice, there are normally two typical conventional strategies for the speed control of chilled water pumps distributing chilled water from chillers to the plate heat exchangers. The main objective is to modulate the pump speed to maintain the outlet water temperature at the secondary side of heat exchanger group at a predefined set-point.

Fig. 1 shows the first typical conventional pump speed control strategy (called 'pressure-based' in this paper). The pump speed is controlled to maintain the measured differential pressure between the main supply and return water pipes before (i.e., at the primary side) the heat exchanger group at a predefined set-point. A temperature controller is used to keep the outlet temperature ($T_{out,ahx,mes}$) after (i.e., at the secondary side) heat exchangers at its set-point by modulating the valve openings before heat exchangers. In this control strategy, the differential pressure set-point is constant, while the temperature set-point after heat exchanger keeps a fixed temperature difference above the chiller supply water temperature. When $T_{out,ahx,mes}$ is higher than its set-point, the modulating valves before heat exchangers will widely open to obtain more chilled water before heat exchangers.

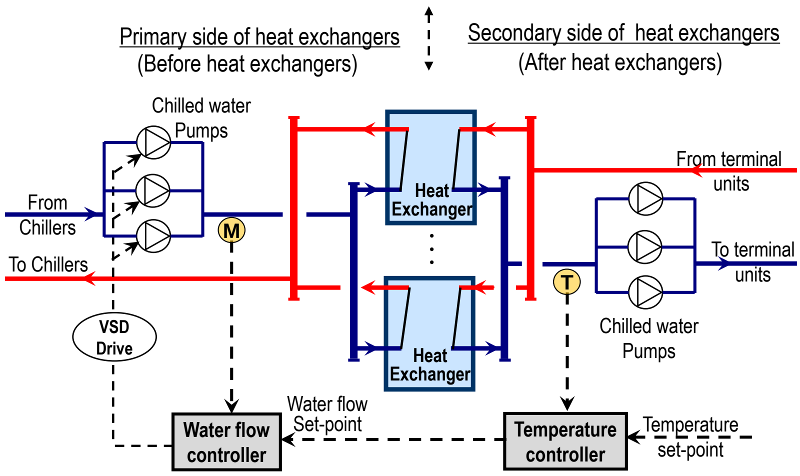

Fig. 2 Conventional speed control for chilled water pumps (flow-based)

Fig. 2 illustrates the second typical conventional pump speed control strategy (called 'flow-based' in this paper) [21]. It is a cascade control involving two loops: inner loop and outer loop. In the outer loop, a temperature controller is utilized to generate a water flow set-point for the inner loop by comparing the measured water temperature after heat exchangers with the preset temperature set-point. In the inner loop, a water flow controller is employed to control the pump speed before heat exchangers by comparing the measured water flow rate before the heat exchanger group with the water flow set-point generated in the outer loop.

It is worthy noticing that temperature set-point of the outlet water after heat exchangers ($T_{out,ahx,set}$) used in the two conventional pump speed control strategies are normally determined based on the chiller supply water temperature set-point ($T_{ch,sup,set}$). Practically, there is a fixed difference between the two temperature set-points, as shown in Eq.(1).

$$T_{out,ahx,set} = T_{ch,sup,set} + \Delta T \tag{1}$$

where, $T_{out,ahx,set}$ is the temperature set-point of the outlet water after heat exchangers. $T_{ch,sup,set}$ is the chiller supply water temperature set-point. ΔT is a constant value, which is normally predefined to equal the rated differential temperature (i.e. temperature difference between the inlet water before heat exchangers and the outlet water after heat exchangers) for sizing the plate heat exchangers at the design stage.

However, the practical case study [18] illustrates that the conventional temperature set-point is not reliable and often cannot be always reached. For instance, during the morning start-up period with cooling peak demand, , the measured supply water temperature ($T_{out,ahx,mes}$) at secondary side of heat exchangers would fail to reach the predefined set-point. In this case, the pumps at primary side have to increase pumping speed to deliver more water aiming at allowing $T_{out,ahx,mes}$ to approach the set-point. The deficit flow may be triggered when the water at primary side of heat exchangers exceeds the water of the chiller loop. When deficit flow exists, the supply water temperature at the primary side of heat exchangers will be increased, which further allows $T_{out,ahx,mes}$ to keep away from the set-point. A higher $T_{out,ahx,mes}$ will reduce the cooling capacity of AHUs at the same water flow conditions. When the phenomenon occurs, that means the chilled water system is out of control and cannot automatically return to the normal operation status. In other words, there is lack of fault-tolerant ability in the conventional control strategy. The fault-tolerant ability, here, refers to the situation that the control strategy can help the system to avoid the deficit flow as far as possible and allow the pumping system hold the accepted performance when the system suffers from the deficit flow problem.

2.2 The proposed robust control strategy

The robust control strategy, as shown in Fig. 3, is developed on top of the conventional control strategy. Here, the flow-based cascade control strategy earlier mentioned is selected as an example of conventional control strategies for developing the robust control strategy.

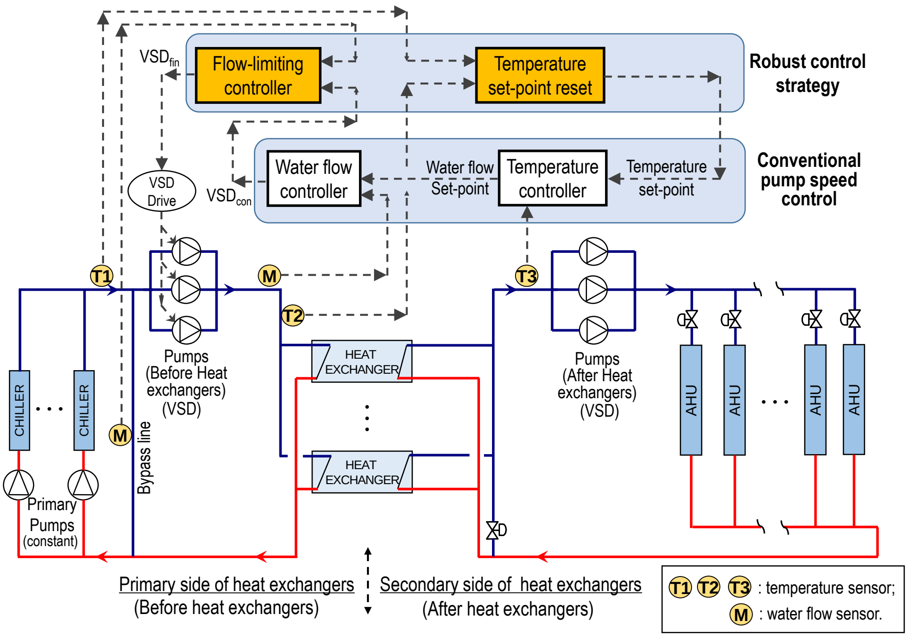

Fig. 3 Proposed robust speed control for chilled water pumps

The robust control strategy consists of two schemes: one is the temperature set-point reset scheme and the other is the flow-limiting control scheme. The temperature set-point reset scheme provides the time-varying set-point considering both the measured chiller supply water temperature (T_1 sensor in Fig. 3) and the measured inlet water temperature before heat exchanger group (T_2 sensor in Fig. 3). The main objective of the temperature set-point reset scheme is to ensure the temperature set-point after heat exchangers can be easily reached under most of working conditions and to prevent pumps from being over-speeded. Moreover, the flow-limiting control scheme is developed that employs a feed-back mechanism to actively eliminate the deficit flow once detected. It continuously re-regulates the pump speed control signal from the conventional control strategy. When detecting the deficit flow (i.e., water flows from return side to supply side) in the bypass line, a rescale mechanism is employed to produce a final speed signal (VSD_{fin}) by reducing the conventional speed signal (VSD_{con}) until the deficit flow is eliminated. The detailed descriptions of the two schemes are shown in Fig. 4, which will be introduced as follows.

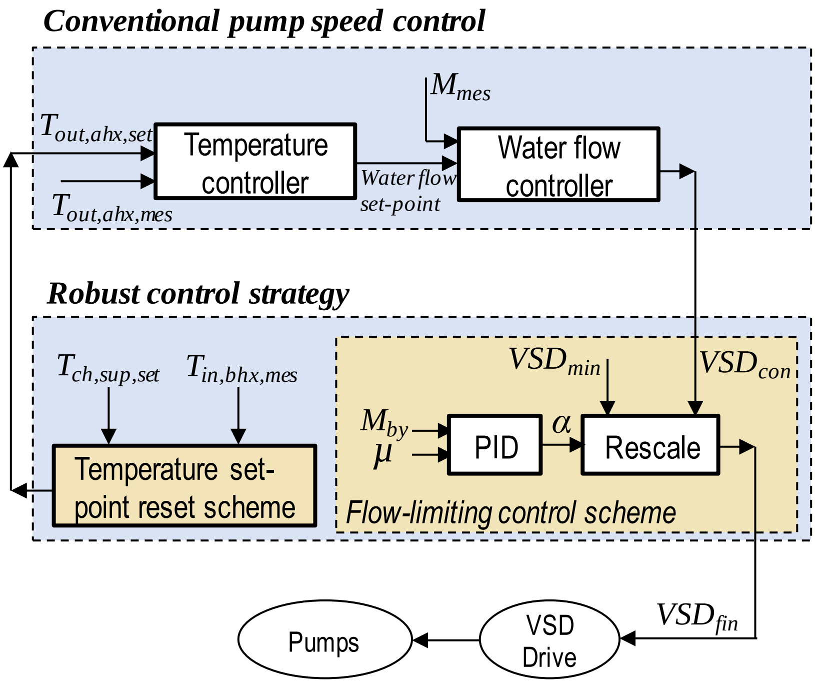

Fig. 4 Basic working principle of the robust control strategy

Temperature set-point reset scheme

The temperature set-point reset scheme provides the time-varying reliable temperature set-point ($T_{out,ahx,set}$) of the outlet water after heat exchangers, which is used to regulate the speed of pumps before the heat exchanger group. Detailed descriptions of the temperature set-point reset are expressed as Eq. (2) and (3).

$$T_{out,ahx,set} = \begin{cases} T_1 \text{ ,} if \ T_1 \le T_{max} \\ T_{max} \text{,} \ otherwise \end{cases} \tag{2}$$

$$T_1 = \max(T_{in,bhx,mes} + \Delta T_1, T_{ch,sup,set} + \Delta T_2) \tag{3}$$

where, $T_{out,ahx,set}$ is the temperature set-point of the outlet water after heat exchangers, °C. T_{max} is the allowed upper limit of the set-point, °C. $T_{ch,sup,set}$ is the chiller supply water temperature set-point, °C. $T_{in,bhx,mes}$ is the measured inlet water temperature before the heat exchanger group, °C. ΔT_1 and ΔT_2 are two constants respectively, °C. ΔT_1 is assigned as the rated temperature difference used for sizing the heat exchangers during design stage. ΔT_2 can be set a little higher than ΔT_1 because studies show that properly increasing $T_{out,ahx,set}$ could enhance the operation reliability while not significantly affecting the pump energy [20].

Different from the conventional methods only based on the chiller supply water temperature set-point, this improved scheme has two advantages. First, it is compatible with the conventional method because $T_{ch,sup,set}$ is also one of the inputs. Second, the measured $T_{in,bhx,mes}$ is selected as another key input. When $T_{ch,sup,set} + \Delta T_2$ is higher than $T_{in,bhx,mes}+ \Delta T_1$, the conventional set-point is used. When $T_{ch,sup,set} + \Delta T_2$ is lower than $T_{in,bhx,mes}+ \Delta T_1$, $T_{in,bhx,mes}+ \Delta T_1$ will be selected as the final set-point. This could ensure a minimum difference of ΔT_2 between the temperature set-point ($T_{out,ahx,set}$) and $T_{in,bhx,mes}$ at all times. The temperature set-point ($T_{out,ahx,set}$) thus can be easily reached. The temperature rest scheme avoids the situation that the temperature set-point cannot be reached under the conventional control method when $T_{in,bhx,mes}$ is higher than $T_{ch,sup,set}$ due to the deficit flow.

It is noted that an upper limit (T_{max}) is set for limiting $T_{out,ahx,set}$ to guarantee the temperature of chilled water supplied to terminal units within a safe range for humidity control in occupied zones.

Flow-limiting control scheme

However, the deficit flow problem would not be fully avoided only using the proposed temperature set-point reset scheme because too many disturbances and faults may affect the practical operations. For instance, the measured outlet water temperature ($T_{out,ahx,mes}$) after heat exchangers would also have the risk of not reaching the set-point ($T_{out,ahx,set}$) when serious water fouling exists in the heat exchangers [22]. Or during the morning start period when the AHUs start to work while the thermal zones' temperatures have not reached the comfort level, the over-demanded cooling by AHUs would make $T_{out,ahx,mes}$ higher than $T_{out,ahx,set}$. Once the above situations occur, the chilled water pumps would have the risk of out of control and lead to the deficit flow problem.

Aiming at dealing with the above issues, a flow-limiting control scheme is developed to provide the active solution to actively eliminate the deficit flow. Its basic principle is to re-regulate the pump speed signal generated from the conventional control strategy by employing a feedback mechanism, as shown in Fig. 4. A proportional-integral-derivative (PID) controller is utilized to generate a control signal $\boldsymbol{\alpha}$ (between 0 and 1) by comparing the measured water flow rate (M_{by}) in the bypass line with a predefined threshold $\boldsymbol{\mu}$. In the function box "Rescale", $\boldsymbol{\alpha}$ is used to rescale the speed control signal (VSD_{con}) provided by the conventional pump speed control. The final speed control signal (VSD_{fin}) can be determined by Eq. (4).

$$VSD_{fin} = \left(VSD_{con} - VSD_{\min}\right) \cdot \alpha + VSD_{\min} \ , \alpha \in [0,1] \tag{4}$$

where, VSD_{final} is the final speed control signal used for pump speed control, Hz. VSD_{con} is the speed control signal from the conventional pump speed control, Hz; VSD_{min} is a constant parameter representing the allowed minimum speed, Hz. $\boldsymbol{\alpha}$ is the time-varying control signal from the PID controller and varies between 0 and 1.

The PID controller used in this study is a typical feedback controller commonly used in industry. It continuously attempts to minimize an error (i.e., difference between a measured process variable and a desired set-point) over time by adjustment of a control variable. The control signal is thus a sum of three terms: the P-term (which is proportional to the error), the I-term (which is proportional to the integral of the error), and the D-term (which is proportional to the derivative of the error). The control signal (i.e., the output of PID) can be calculated by the classical mathematical description as Eq.(5). Nowadays, the PID controller as a standard function module is integrated in the DDC controller for convenient use. In this study, a lower limit (i.e., 0) and a higher limit (i.e., 1) should be set to the output of the PID. The three tuning parameters (K_p, K_i and K_d) should be properly tuned in advance.

$$u(t) = K_p e(t) + K_i \int e(\tau) d\tau + K_d \frac{de(t)}{dt} \tag{5}$$

where, K_p, K_i and K_d are the tuning parameters represent proportional gain, integral gain and derivative gain, respectively; e is the error, which indicates the difference between the measured water flow rate and the desired set-point in this study; t is instantaneous time; τ is the variable of integration which takes on values from time 0 to the present t.

The threshold $\boldsymbol{\mu}$ represents the expected minimal and positive (negative value means deficit flow) flow in the bypass line. In this study, μ is set to be about 3% of the design water flow rate of a single chiller. In the case when the measured M_{by} is larger than $\boldsymbol{\mu}$, which means there is no deficit flow, the output (i.e. $\boldsymbol{\alpha}$) of PID will be increased rapidly until reaching 100%. In that way, the final speed control signal (VSD_{fin}) is equal to VSD_{con}. In another case when the deficit flow is detected, the measured M_{by} is a negative value and is less than $\boldsymbol{\mu}$. The PID output (i.e. $\boldsymbol{\alpha}$) will be reduced gradually. VSD_{fin} will be reduced consequently, resulting in a reduced pump speed. The pump speed will be reduced continuously until the deficit flow is eliminated or the pump speed reaches the minimum value (VSD_{min}).

3. In-situ validation platform: a real chilled water system in a high-rise building

3.1 Description of the studied system

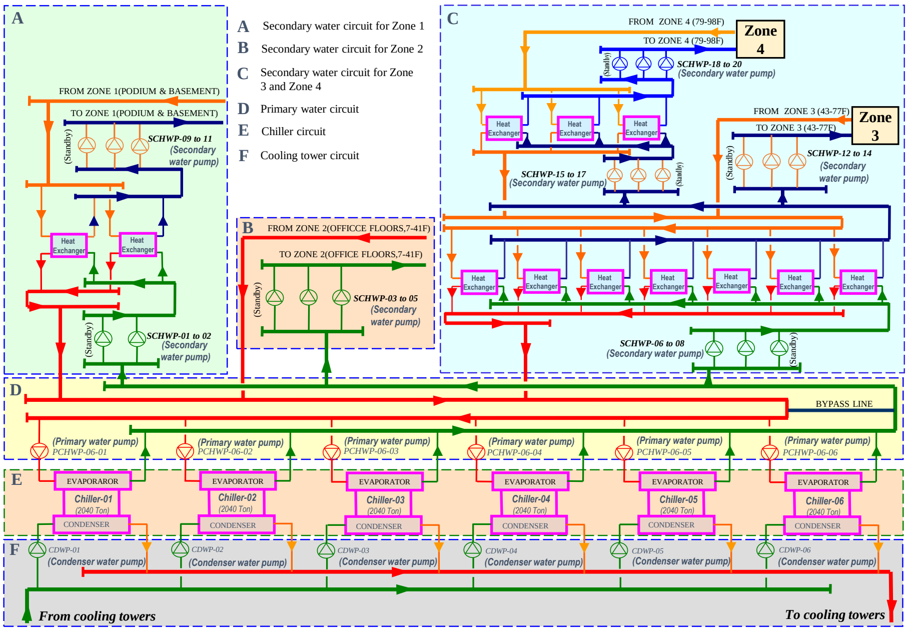

Fig. 5 Schematic of the chilled water system

The central cooling system concerned in this study is a complex primary-secondary chilled water system in a super high-rise building in Hong Kong [23]. The building is about 490 meters high with approximately 321,000 m^2 floor areas, consisting of a basement of four floors, a block building of six floors and a tower building of 98 floors. As shown in Fig. 5, this central chilled water plant employs six identical constant speed centrifugal chillers to provide chilled water for air handling units in the building. The rated cooling capacity and power consumption of each chiller are 7230 kW and 1270 kW with the rated COP (coefficient of performance) of 5.69. The design chilled water supply and return temperatures for chillers are 5.5°C and 10.5°C respectively. Each chiller is associated with one constant condenser water pump with the rated flow rate of 410 L/s. The heat generated from the chiller condensers is rejected by eleven evaporative water cooling towers with a total design capacity of 51,709 kW. A typical primary-secondary chilled water system is employed in this chilled water system. In the primary loop, each chiller is associated with a constant speed primary water pump to guarantee the fixed water flow through the chiller. The primary loop is decoupled with the secondary loop through the bypass line. The secondary chilled water loop is divided into four zones, in which Zone 2 is supplied with the chilled water from chillers directly. The plate heat exchangers are employed to transfer cooling energy from chillers to terminal air-handling units in Zone 1, Zone 3 and Zone 4 to avoid chilled water pipelines and terminal units from suffering extremely high static pressure. The design inlet and outlet water temperatures at the secondary side of the heat exchangers are 11.3°C and 6.3°C, respectively. All the secondary pumps are equipped with variable speed drivers and all the primary pumps are constant speed pumps. The specifications of chilled water pumps are listed in Table 1.

Table 1 Specifications of chilled water pumps

Pumps	Number*	Flow (*L/s*)	Head (*kPa*)	Power (*kW*)	Remarks
Primary water pumps					
PCHWP-01 to 06	6	345	310	126	Constant speed
Secondary pumps for Zone1					
SCHWP-01 to 02	1(1)	345	241	101	Variable speed
SCHWP-09 to 11	2(1)	155	391	76.9	Variable speed
Secondary pumps for Zone 2					
SCHWP-03 to 05	2(1)	345	406	163	Variable speed
Pumps for Zone 3 and Zone 4					
SCHWP-06 to 08	2(1)	345	297	122	Variable speed
SCHWP-12 to 14	2(1)	294	358	120	Variable speed
SCHWP-15 to 17	2(1)	227	257	69.1	Variable speed
SCHWP-18 to 20	2(1)	227	384	102	Variable speed

*Value in parentheses indicates number of stand by pumps

This central chilled water plant frequently suffered from the deficit flow problem and the low chilled water temperature difference syndrome after its first use since the middle of 2008. An on-site study [18] were conducted to diagnose and find out the major reason for the deficit flow problem. The improper set-point of outlet water temperature on the secondary side of heat exchangers was finally diagnosed as the main reason that resulted in the deficit flow and low delta-T syndrome. Therefore, the proposed robust control strategy was applied and evaluated in this chilled water system.

The studied chilled water system is a representative primary-secondary system that is widely used in the high-rise commercial buildings in Hong Kong. The major feature is that the plate heat exchanges are employed to transfer the cooling energy from low levels to high levels to prevent the extremely high static pressure on AHUs. Similar configurations involving plate heat exchangers are popular in the high-rise buildings worldwide. According to the pressure bearing capacity of the HVAC equipment (e.g. AHUs, pipelines), the chilled water system could be divided into different sub-systems every certain meters (e.g. 100m or 160m) vertically.

3.2 Setup of the test

In the studied system, the deficit flow is mainly contributed from the abnormal operation of the chilled water pumps at the primary side of the heat exchangers group serving Zone 3, as shown in Fig. 5. The pumps associated with the heat exchangers serving Zone 3 are selected as the experimental object, which is extracted and shown in a simplified schematic, as shown in Fig. 6. Secondary pumps (SCHWP-6 to 8), at the primary side of heat exchangers, are controlled with the proposed robust control strategy, which distribute the chilled water from the chillers to the plate heat exchangers. At the secondary side of heat exchangers, chilled water pumps (SCHWP-12 to 14) deliver the water from the heat exchangers to the terminal AHUs.

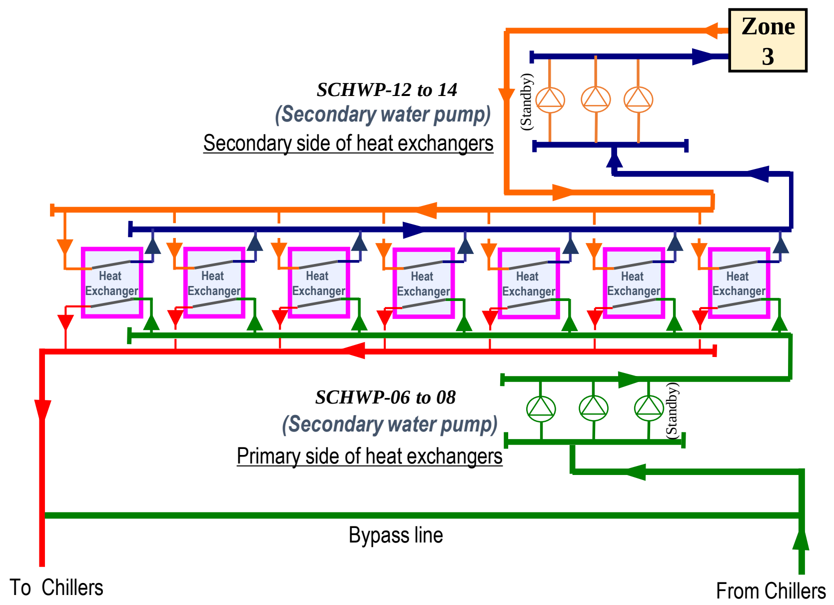

Fig. 6 Schematic of the test platform

The local speed control strategy for pumps (SCHWP-06 to 08) at the primary side of heat exchanges used originally is the flow-based conventional control strategy as shown in Fig. 2, which was detailed introduced earlier. The temperature set-point of the outlet water after heat exchangers used originally is a fixed-differential-based set-point that uses a fixed difference above the chiller supply water temperature set-point.

able 2 Description of the control strategies in tests

Strategy	Description	Parameters used in Eq. (1) or (2)
Case #1 (conventional control strategy)	Fixed-differential-based temperature set-point	ΔT=0.8 ºC
Case #2 (proposed robust control strategy)	Proposed temperature set-point reset scheme and flow-limiting control scheme	ΔT_1=0.8 ºC ΔT_2=1.2 ºC

When in the field tests, on top of the conventional pump speed control, the proposed temperature set-point reset scheme and the flow-limiting control scheme in the proposed robust control strategy were implemented. In order to evaluate the performance of the proposed robust control strategy, the conventional control strategy using the fixed-differential-based set-point is selected as the reference case for comparison. The strategies and the related parameters used in Eq. (1) and (2) are presented in Table 2.

3.3 Metering system

A building management system (BMS) is employed in the studied building, which monitors and controls the building's mechanical and electrical equipment such as air-conditioning system, lighting, power systems, fire systems, and security systems. As for the air-conditioning system, the key operation parameters are measured by sensors, such as the sensors of temperature, water flow rate, pressure and power. The real-time measurements are collected and stored in a database. Some key real-time measurements are sent to the control platform as the inputs of control strategies. All the sensors have been calibrated. The sampling time step is one minute.

The measurement devices related to this study are listed in Table 3. The bi-directional insertion flow meter is used to measure the water flow rate in both directions of the bypass line. A negative value means the deficit flow in this study. The heat meter is used for measuring the cooling load, which actually consists of two temperature sensors, a flow meter and a calculator. The pump power is measured by the power meter.

Table 3 Measurement devices and parameters

Meter type	Range	Repeatability	Accuracy
Bi-directional insertion flow meter	(0.15 to 6 m/s) Connection Size	±0.5%	-
Insertion flow meter	(0.08 to 6.09 m/s) Connection Size	±0.5%	-
Temperature sensor	-46 -104 ℃	-	±0.19 ℃
Power meter	Voltage: 80- 480V Current: 1-600A	-	Voltage/current: ±0.5% Power: ±1.0% Energy: ±1.0%
Heat meter	Water velocity: (0.2 to 6.1 m/s)	-	Temperature: ±0.5% Flow: ±1% Calculator：±0.5% Complete instrument：±2%

3.4 Validation method

The proposed robust control strategy has been programmed into a control module and implemented on the control platform of the building. Field tests have been conducted to evaluate the performance of the proposed control strategy. Different from the laboratory tests in which the same working conditions can be repeated for different tests at the same time, the in-situ test only allow to test one control strategy at a time. Therefore, it is hard to compare the two control strategies under almost the same working conditions. In order to obtain reasonable comparisons, building cooling load was selected as the key working condition for comparing the performances of different control strategies. For a building, there are many factors that affect the operation of the air-conditioning system, such as the weather conditions, the occupancies, the operating schedules, etc. However, the effects of these factors on a building eventually can be reflected by the cooling load. For an air-conditioning system, its energy performance can be evaluated based on the energy consumed for handling unit cooling load.

In this study, the performances of the two control strategies were compared in terms of two aspects: dynamic operation performance and energy performance. For dynamic operation performance comparison, the cooling load profile is the important factor. Two days with the similar cooling load profile were selected, in which different control strategies were employed respectively. Under the similar cooling load profile, the operating parameters of chilled water system, such as water flow rate of the bypass, pump speed, the measured water temperature at secondary side of the heat exchangers, were compared. For energy performance comparison, particularly the yearly energy use, the annual energy use of the related pumps is the essential indicator. The two control strategies were implemented respectively in two different years with similar accumulated cooling load. The difference between the pump energy uses of the two years could be determined as the energy savings due to the implementation of the proposed control strategies. Although there would be some different weather conditions or operation schedules in the two years, it is reasonable to confirm the energy savings as long as the accumulated cooling loads of the two years are similar and there are no other measures implemented. In order to prove the energy saving, the yearly cooling load using the conventional control strategy had better to be not more than that using the proposed control strategy.

4. Results and discussions

In this part, the actual performance of the chilled water system under the conventional control strategy (Case #1) is firstly outlined. Then, the filed test results using the proposed robust control strategy (Case #2) is presented and compared with the performance under the conventional control strategy in a selected similar day.

4.1 Operation performance under the conventional control strategy (Case #1)

Fig. 7 shows the measured water flow rate in the bypass line under the conventional control strategy (Case #1) using the fixed-differential-based temperature set-point. The negative flow values refer to the occurrence of deficit flow that means the water flow rate of the secondary loop (i.e., main loop before heat exchangers) exceeds that circulated in the chiller loop. It can be observed that the deficit flow occurred in nearly every day in the eight consecutive days, particularly during night time. The maximum duration of the consecutive deficit flow was up to 31 hours and the most serious deficit flow was up to 400 L/s that accounts for about 58% of the rated water flow of this subsystem under full load conditions. It is worth noting that the deficit flow was hard to be eliminated automatically in short period once it was triggered. It usually disappeared after a long time when additional chiller was switched on.

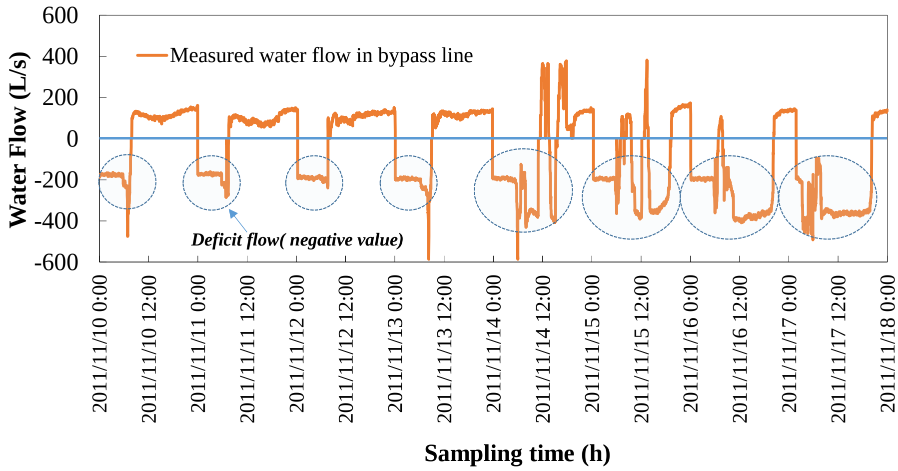

Fig. 7 The measured water flow rate of the bypass (Conventional control)

Fig. 8 describes the relationship between the measured outlet water temperature ($T_{out,ahx,mes}$) after the heat exchanger group and the corresponding set-point when conventional control strategy was used. Obviously, in most of the time, the measured temperature could not be maintained at the set-point. Particularly, under the situations when the deficit flow occurred, the measured temperatures significantly deviated from the set-points. The maximum temperature deviation was up to 3°C. The reason why $T_{out,ahx,mes}$ could not be maintained at the set-point is that the deficit flow resulted in increased inlet water temperature ($T_{in,bhx,mes}$) before heat exchangers. The increased $T_{in,bhx,mes}$ would in turn cause the rise of $T_{out,ahx,mes}$. The measured $T_{out,ahx,mes}$ is thus higher than the preset conventional set-point, which is set as a fixed value higher than the chiller supply water temperature. According to the conventional control strategy, when the measured $T_{out,ahx,mes}$ cannot be maintained at the set-point for a while, the pump speed would be continually increased to supply more water. Therefore, the operation stability and the efficiency of the chilled water system were significantly degraded.

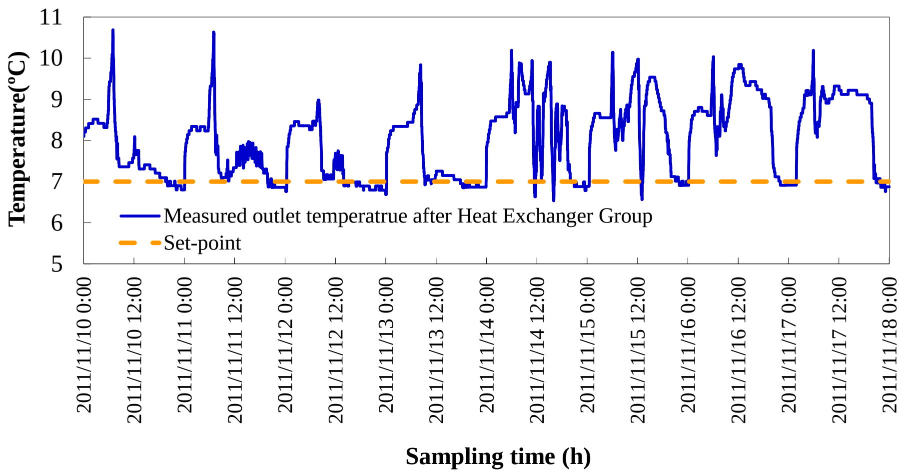

Fig. 8 The measured outlet water temperature after heat exchanger group (Conventional control)

<u>4.2 Test results of the proposed robust control strategy (Case #2)</u>

Some repeated experiments were conducted to test the dynamic performance of the proposed robust control strategy during the night time because the deficit flow was frequently observed during this period in the past. The results from the test during the night of Nov. 17 2013 were presented (i.e., namely Case # 2). In order to evaluate the dynamic operation performance and the energy performance of the robust control strategy, a reference case using the conventional control strategy was selected for comparisons. A night in November of 2011 with similar cooling load profile was selected as the reference case (i.e., namely Case # 1) when the conventional control strategy was used.

The cooling load profiles of the same period in the two nights are shown in Fig. 9. It can be found that the cooling loads of the two nights are very similar during most of the time except for the morning start period. In case #1, most of the terminal air handling units (AHUs) started to work from 6:00am in the morning, which results in a sudden increase of the cooling demand. In case #2, there were two start times in the morning: 7:00am and 8:00am, which corresponds to two smaller peak demand of cooling. That means the AHUs of this zone were divided into two groups and started to work at the two times, respectively.

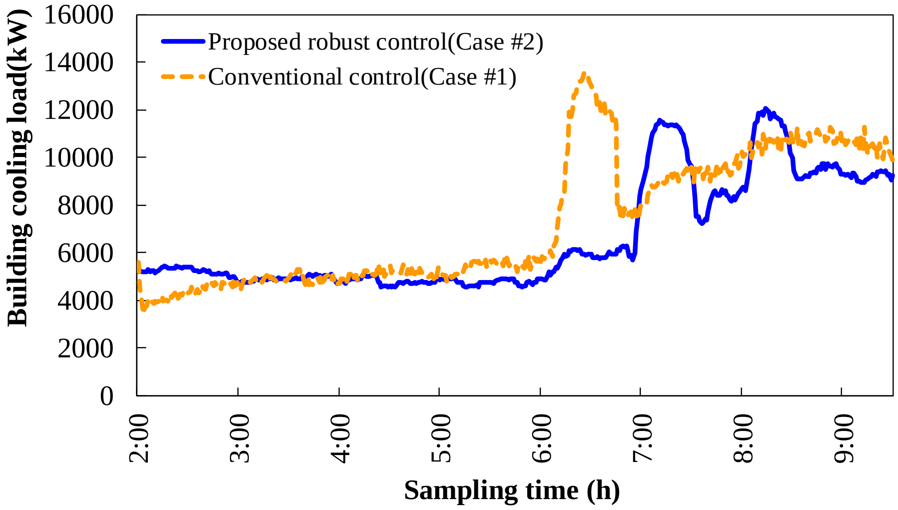

Fig. 9 Measured cooling load profiles

Operation performance evaluation

Fig. 10 depicts the measured water flow rate in the bypass line in the two similar nights. In case #1 using the conventional control strategy, the system experienced serious deficit flow (up to 500 L/s) during the entire night. During the early morning, even additional chiller and the associated constant pump were switched on, the deficit flow was still not eliminated. While in case #2 using the proposed robust control strategy, there was nearly no deficit flow except the morning start period (6:00 to 7：00am).Actually, there is a normal phenomenon when small amount of deficit flow occurs during the morning start period. That is because that there are excessive cooling demands from the occupied zones before their indoor thermal comforts reach the predefined level during the start period. It also can be observed that, in case #2, the flow rate in bypass line approached zero twice between 7:00 and 8:30am, which means the deficit flow was going to arise but was stopped. This benefits from the flow-limiting controller, which timely reset the pump speed to reduce the water flow rate circulated before heat exchangers, preventing the water flow rate of the bypass line further from dropping to the negative value (i.e., deficit flow).

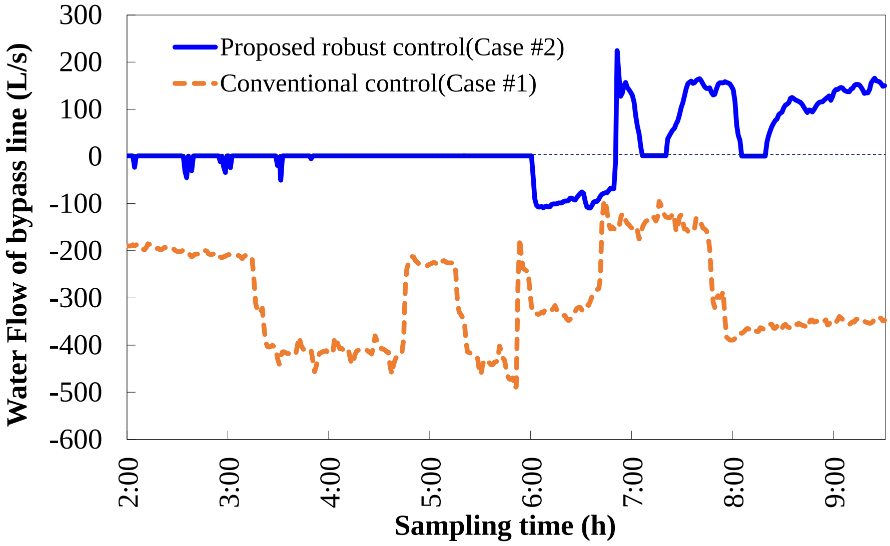

Fig. 10 Measured water flow rate of the bypass line

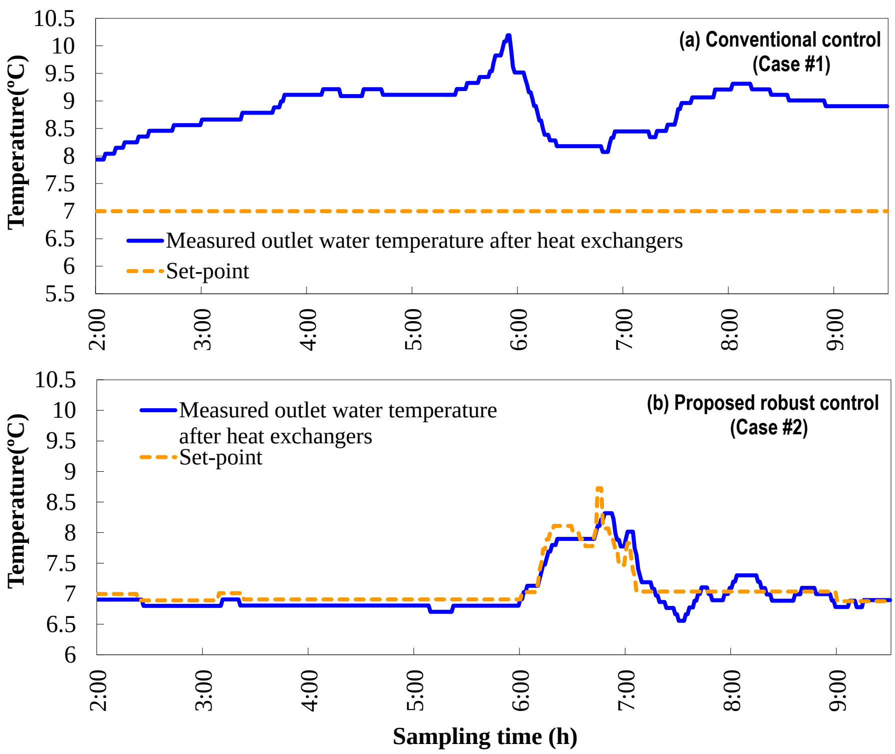

Fig. 11 Measured outlet water temperature after heat exchangers

Fig. 11 shows the measured outlet water temperature ($T_{out,ahx,mes}$) after the heat exchanger group in the two cases. In case #1, $T_{out,ahx,mes}$ is significantly higher than the set-point within all the time. Particularly when the deficit flow occurred, the deviation is enlarged. Although more water was supplied before heat exchanger group when deficit flow existed, $T_{out,ahx,mes}$ was still not effectively lowered down. This is mainly due to the increased inlet water temperature before heat exchangers ($T_{in,bhx,mes}$). In this situation, however, the conventional temperature set-point reset method lacked the corresponding adjustment in response to the time-varying working conditions. In case #2, $T_{out,ahx,mes}$ can timely follow the time-varying set-point and can be nearly always maintained at the set-points (except for the one-hour morning start period). That is because the proposed temperature set-point is determined based on not only the chiller supply water temperature but also the measured $T_{in,bhx,mes}$. When $T_{in,bhx,mes}$ increased, the set-point would be reset and increased accordingly, which can be observed between 6:00 and 7:00am in Fig. 11. Thus, the proposed set-point were continuously adjusted with the time-varying working conditions, aiming at allowing the measured $T_{out,ahx,mes}$ reach the set-point at most of the time.

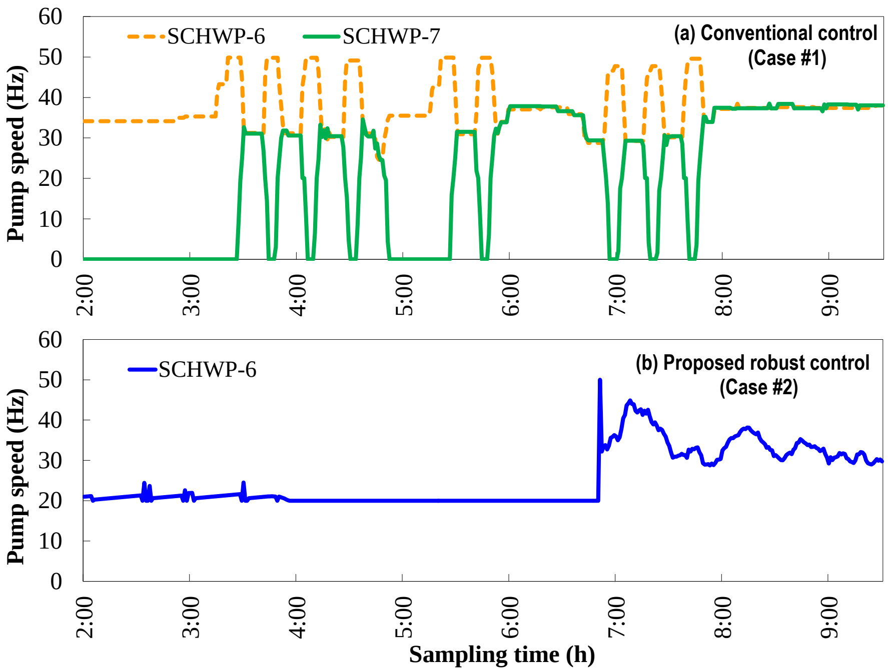

Fig. 12 Measured operating frequency of the chilled water pumps

Fig. 12 illustrates the measured operating frequency of the chilled water pumps before the heat exchangers. In case #1, the operating number of pumps changed frequently, which indicates the significant instability of the pump operations. Actually, two pumps were not necessary under the low cooling load condition of nights. However, due to the fact that $T_{out,ahx,mes}$ cannot reach the set-point, the pump speed was continuously increased. When one pump reached the maximum speed (i.e. 50 Hz), additional pump was switched on automatically based on the predefined control logic. In case #2, there was only one pump (i.e., SCHWP-6) working during the whole test period. The pump worked stably. The frequency of the chilled water pump was maintained at the minimum speed (i.e. 20 Hz) during most of the night time. That is because the flow-limiting scheme was in effect. When detecting the occurrence of the deficit flow, the flow-limiting scheme automatically re-regulated and reduced the pump speed. As a result, the water flow of the secondary loop dropped until the deficit flow was eliminated. The result in case #2 illustrates that only one pump is enough for delivering water under the cooling load condition of the selected day, which also illustrates the additional pump should be not required in case #1.Fig. 13 compares the measured chiller supply water temperature ($T_{ch,sup}$) and the supply water temperature ($T_{in,bhx,mes}$) before the heat exchanger group. At normal conditions when there is no deficit flow, the two temperatures shall be nearly equal. In case #1 of Fig. 12(a), $T_{in,bhx,mes}$ is significantly higher than $T_{ch,sup}$ due to the large amount of deficit flow flowing from the return side to the supply side of the secondary chilled water loop. The average difference is 1.6ºC and the maximum difference is 3 ºC. In case #2, $T_{in,bhx,mes}$ was a little higher than $T_{ch,sup}$ during night time (before 6:00am) when there was very small amount of deficit flow, and was almost equal to $T_{ch,sup}$ in the morning when there was no deficit flow. The average difference is 0.3 ºC and the maximum difference is 1.5 ºC.

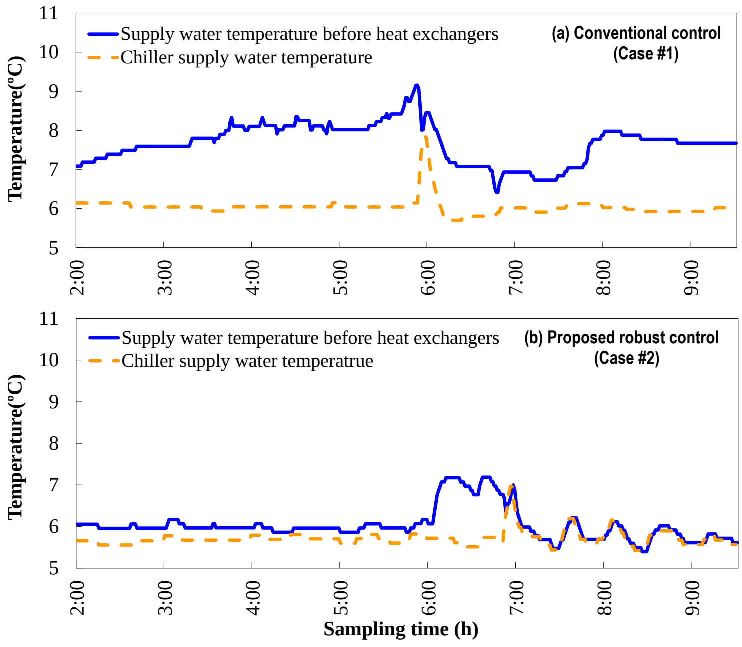

Fig. 13 Measured supply water temperature before heat exchangers

The measured system temperature difference of the overall secondary water loop in the two cases was compared in Fig. 14. The temperature difference in case #1 was about 1.5ºC during night and was about 2.4 ºC in the morning. In case #2, due to the robust control strategy, the measured system temperature difference was highly increased: an average of 4.5 ºC during night and 5.5 ºC in the morning.

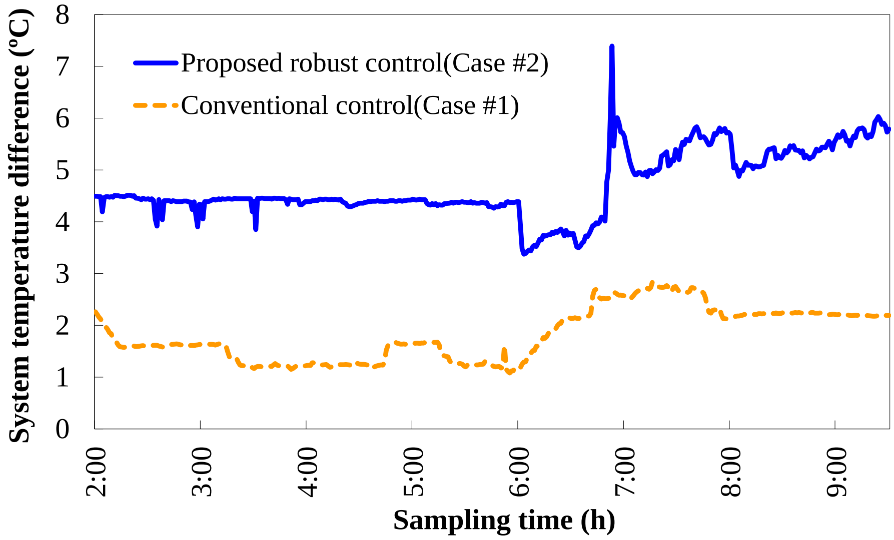

Fig. 14 Measured system temperature difference of the secondary loop

Energy performance evaluation

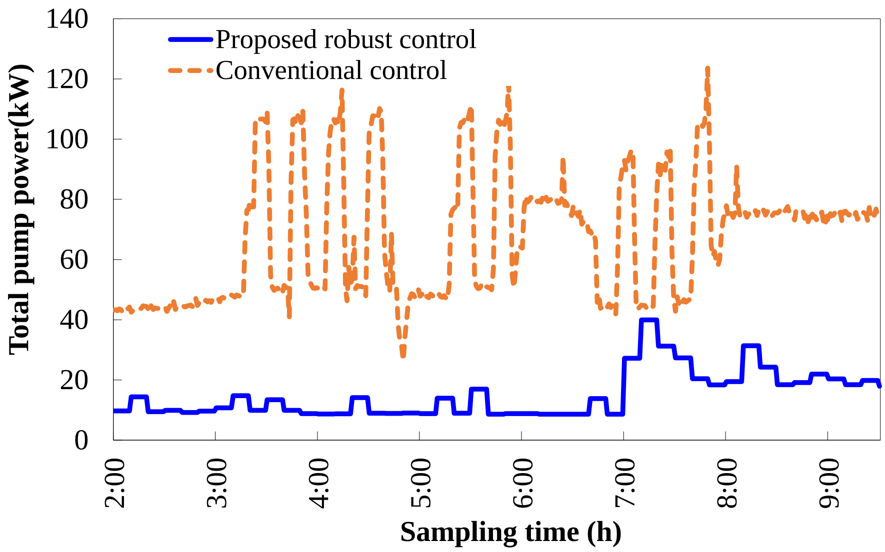

Fig. 15 Comparison of pump energy profile in two cases

Fig. 15 presents the comparison of the total energy use of the chilled water pumps before heat exchangers of the two cases. Although the cooling load profiles of the two cases were similar, the actual energy consumed by chilled water pumps was quite different. It can be found that the total energy consumed by the chilled water pumps before the heat exchanger group in case #2 was significantly lower than that in case #1. It is observed that the total pump power fluctuated greatly in case #1 because of the frequent on-off of the additional pump.

Table 4 Summary of the pump energy savings during the test period

Strategies	Total pump energy use during test period (kWh)	Energy Saving during test	
		(kWh)	(%)
Case #1 (conventional control strategy)	506	-	-
Case #2 (proposed robust control strategy)	112	394	77.8

Table 5 Summary of the pump energy savings in a whole year

Strategies	Annual building cooling load (MWh)	Annual pump energy use (kWh)	Energy saving	
			(kWh)	(%)
Case #1 (conventional control strategy)	78,633	356,877	-	-
Case #2 (proposed robust control strategy)	68,686	216,101	140,776	39.4

According to Table 4, the total energy saving of the chilled water pumps during the test was 394 kWh, which accounts for 78% of the energy consumption under the conventional control strategy. The energy saving is mainly contributed from the elimination of the deficit flow. This robust control strategy has been applied in the studied system since 2014. The measured yearly energy use of the three chilled water pumps in 2014 (robust control) was compared with that in 2011 (conventional control), as shown in Table 5. 140,776 kWh pump energy can be saved in the whole year, which occupies 39% of the pump energy use in 2011. The detailed comparison of each month is presented in Fig. 16. It can be observed the maximum monthly energy saving appeared in November. About 24700 kWh (66%) pump energy is saved in this month. It is also worthy of pointing out that the measured cooling load of 2014 (Case #2) is basically a little lower than that in 2011 (Case #1), as shown in Fig. 17. That means the pump energy saving is mainly obtained from the robust control strategy.

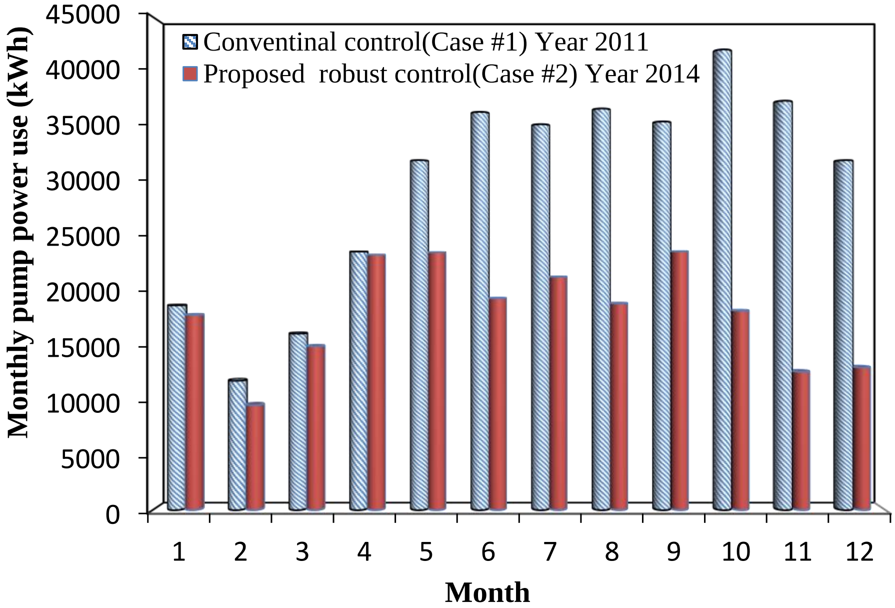

Fig. 16 Monthly pump energy use in two years

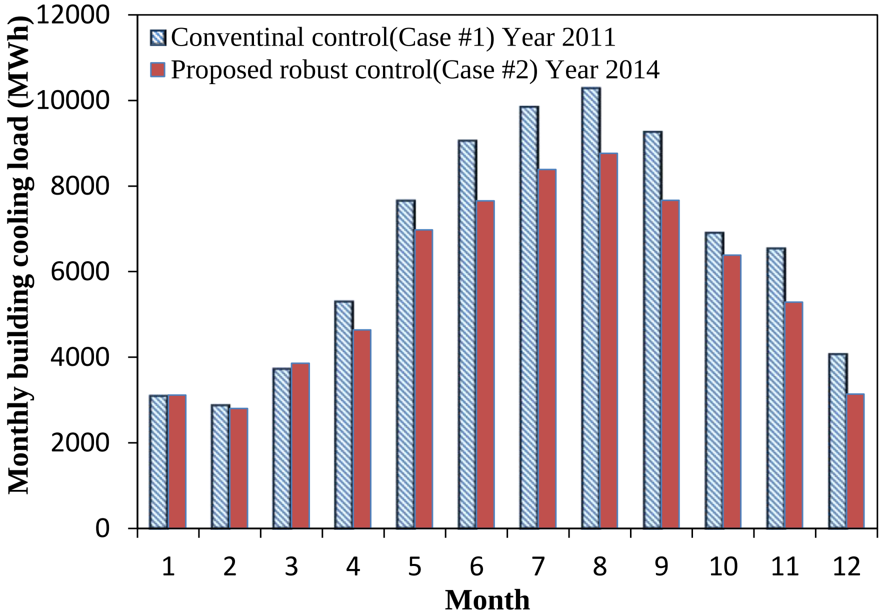

Fig. 17 Monthly building cooling load in two years

4. 3 Discussions

The results from the above field tests show that the temperature set-point at the secondary side of heat exchangers has significant effect on the operation and energy performances of chilled water pumps. Under the conventional control strategy with the set-point predefined using a fixed differential temperature above the chiller supply water temperature set-point, the deficit flow problem and the low delta-T syndrome are easily caused. A series of operational problems are consequently triggered, such as the higher supply water temperature before heat exchangers, the low delta-T syndrome, the over-speeded pumps and the highly degraded energy performance. The main reason is that the fixed-differential-based temperature set-point is not robust or reliable in practical applications. Heat exchanges are normally sized to achieve a fixed differential temperature between the supply water temperature ($T_{in,bhx,mes}$) before heat exchangers and the outlet water temperature ($T_{out,ahx,mes}$) after heat exchangers under the rated water flow rate. When in normal operations, $T_{in,bhx,mes}$ is nearly equal to the chiller supply water temperature. But, $T_{in,bhx,mes}$ is easily affected by the disturbances in the practical operations, such as the deficit flow. Once $T_{in,bhx,mes}$ is higher than the chiller supply water temperature (e.g. deficit flow occurs in the bypass line), more chilled water before heat exchangers is required in order to ensure $T_{out,ahx,mes}$ reach the set-point. Chilled water pumps would be continuously speeded up until the $T_{out,ahx,mes}$ reaches the set-point or the pumps reach their full speeds. The deficit flow would be further worsened and considerable pump energy will be wasted.

The above field test results prove that the proposed robust control strategy is feasible for eliminating the deficit flow problem and is able to enhance the operation and energy performances of pumps. First, the proposed temperature set-point could timely vary according to the measured supply water temperature ($T_{in,bhx,mes}$) before the heat exchanger group. Since a minimum temperature difference is set between the temperature set-point and $T_{in,bhx,mes}$, this allows the actual outlet water temperature can easily reach the set-point. This prevents the pumps from being over-speeded. Second, it is proved that the proposed flow-limiting scheme can actively eliminate the deficit flow by employing a feed-back mechanism. When detecting the occurrence of the deficit flow, the pump speed is automatically reduced until the deficit flow is fully eliminated.

It should be noted that there was still small amount of deficit flow observed during the morning start period when the proposed robust control strategy was used in the test. That is because the secondary loop of this studied system consists of three risers, each of which is associated with one group of secondary pumps serving related zones. The proposed robust control strategy was only implemented for the chilled water pump group of one riser serving Zone 3. During morning start period, some of the air handling units of the other two risers started to work. Excessive water was supplied in the other two risers and resulted in the fact that the total water flow rate of the secondary loop exceeded that of the primary loop. This situation is supposed to be improved if the proposed robust control strategy will be implemented for the chilled water pumps of the other two risers in the future.

Actually, switching on an additional chiller and the associated primary pump to increase the water flow of the primary loop is also an alternative solution for avoiding the deficit flow. However, this solution also could lead to increased energy use. When handling the same cooling load, excessive operating chillers with lower load ratios may degrade the operating COP (coefficient of performance) of individual chillers. Besides, when adding an additional operating chiller, the associated primary chilled water pump, the condensing water pump and the cooling tower also should be switched on. Therefore, adding an additional chiller would be a solution but not the economic one.

4. 4 Application issues

The proposed robust control strategy is developed to improve the reliability of the conventional pump speed control strategies by employing a temperature set-point reset scheme and a flow-limiting control scheme. The two schemes can be easily integrated into the existing controllers of the conventional control strategy. For instance, Eq. (2) ,Eq.(3) and (4) of the two schemes can be programmed into separated modules and implemented in the existing DDC (direct digital control) controllers that are responsible for the pump speed control. Therefore, it is not only applicable for the new system but also for a large number of existing systems. The connections between the proposed control strategy and the conventional control strategy are shown in Fig. 4. In addition, the proposed robust control strategy is developed for the chilled water pumps distributing water to heat exchangers. But for the chilled water system without plate heat exchangers, only the flow-limiting control scheme in the robust control strategy can be applied for the chilled water pumps distributing water to the terminal units.

5. Conclusions

This paper presents an online robust control strategy for real-time control of chilled water pumps to deal with the low delta-T syndrome for practical applications in the complex chilled water system of high-rise buildings. On top of the conventional control strategies, a temperature set-point reset scheme and a flow-limiting control scheme are developed to enhance the pump operation and energy performances. This robust control strategy was tested and evaluated in a real chilled water system of a high-rise building. The results show that the robust control strategy can enhance the reliability of the chilled water pumps and effectively avoid the deficit flow problem, leading to increased overall system temperature difference. Some detailed conclusions during the tests are summarized as follows.

- When using the proposed temperature set-point reset scheme, the test results proved the set-point could automatically vary in response to the time-varying inlet water temperature at primary side of heat exchangers, which make the set-point could be easily reached.
- When using the proposed flow-limiting scheme, the test results proved that the flow rate of the bypass could be actively limited to not lower than zero when the measured flow rate touches the predefined threshold. The pump speed signal was re-regulated and to reduce the water flow rate at primary side of heat exchangers.
- The combined effect of the two schemes ensure a more reliable operation performance of chilled water pumps. The improper operation of pumps, such as frequent on-off or over-speeds, under conventional control strategies was avoided. The deficit flow problem and low delta-T syndrome were effectively avoided.
- Compared with the conventional pump control strategy, about 78% of the total chilled water pump energy was saved in the test period when using the robust control strategy. 140,776 kWh (39%) of pump energy was actually saved in the whole year after the robust control strategy was implemented for eliminating the deficit flow in the studied system.

This robust control strategy is developed without complex models, which avoids heavy computations and is feasible for practical online applications. This robust control strategy is not only applicable for the speed control of chilled water pumps distributing water to plate heat exchangers, but also for the chilled water pumps distributing water to the terminal units by means of the proposed flow-limiting control scheme only.

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Nomenclature

$T_{out,ahx,mes}$ outlet water temperature at the secondary side of heat exchange group, °C

$T_{out,ahx,set}$ temperature set-point of the outlet water after heat exchangers, °C

$T_{ch,sup,set}$ chiller supply water temperature set-point, °C

$T_{in,bhx,mes}$ measured inlet water temperature before the heat exchanger group, °C

ΔT differential temperature, °C

T_{max} upper limit of the temperature set-point, °C

ΔT_1 the rated temperature difference used for sizing the heat exchangers, °C

ΔT_2 a fixed temperature difference, °C

M_{by} water flow rate in the bypass line, kg/s

VSD_{final} final speed control signal used for pump speed control, Hz

VSD_{con} speed control signal from the conventional pump speed control, Hz

VSD_{min} a constant parameter representing the allowed minimum speed, Hz

K_p proportional gain

K_i integral gain

K_d derivative gain

e error

t instantaneous time

Greek symbols

α time-varying control signal from the PID controller and varies between 0 and 1

τ the variable of integration which takes on values from time 0 to the present t, s

μ a threshold representing the expected minimal and positive flow in the bypass line, kg/s