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A disturbance compensation enhanced control strategy of HVAC systems for improved building indoor environment control when providing power grid frequency regulation

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Abstract: Renewable electricity generations are promising to address the global energy issue while they also place great pressure on the reliability of power grids due to their intermittent nature. In recent years, existing heating, ventilation and air-conditioning (HVAC) systems in buildings have attracted increasing attention to implement continuous demand response in providing frequency regulation service, which can enhance instantaneous power balance and reliability of power grids without extra huge investment. When providing frequency regulation service, the power use of HVAC systems would follow the regulation signals. On the other hand, these signals, acting as continuous disturbances, affect naturally the building indoor environment control at the demand side. In this paper, a novel control strategy is proposed, which can prevent the sacrifice of the building indoor environment when providing the service. The core element of this control strategy is a frequency disturbance compensation scheme, which is developed based on the concept of "disturbance-observer-based control". Experimental results show that the use of the proposed strategy can achieve significant improvement in the building indoor environment control without sacrificing the quality of frequency regulation service. In addition, the wear level of the valve was not affected significantly when adopting the frequency disturbance compensation scheme.

Keywords: building demand response; HVAC systems; grid-responsive building; frequency regulation; smart grid; disturbance compensation.

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1. Introduction

Renewable electricity generations are promising to address the global energy issue while they also place great pressure on the reliability of power grids due to their intermittent nature [1]. The reliability and instantaneous balance of power grids (reflected in power grid frequency) is conventionally guaranteed through frequency regulation (one of the most important ancillary services [2]) provided at the supply side [3]. However, more frequency regulation capacity will be needed due to the increasing involvement of renewable power generations [4]. Huge investment, if solving this problem from the supply side as usual, cannot be avoided either by building energy/power storage [5] or by setting up more generators for reserve. In addition, due to the rapid increase of installed capacity, the annual working hours of generators will decrease dramatically [6].

Recently, more policies have been passed to encourage demand resources to provide frequency regulation service (through continuous demand response [7, 8]) with monetary incentives [9, 10]. The process and mechanism for the demand side to provide this service is elaborated as follows. The authorities of power grids calculate the "area control error" (ACE), the magnitude of the power imbalance between the supply side and the demand side. Then, the ACE is transformed and normalized to automatic generation control (AGC) signal (a frequency regulation signal from -1 to 1 for each signal point), and send to participants involved [11]. To provide this service to power grids, demand resources should implement continuous demand response [12], i.e., continuously manipulate their power use timely and accurately to follow the AGC signal. Note that demand resources can bid different regulation capacities according to their own flexibility for feasible financial rewards. Even small power consumers are encouraged to provide this service [13]. A large number of small power consumers can result in a large regulation capacity in total and can effectively help the power grids to relieve the power imbalance. In this way, the frequency of power grids can be maintained within an acceptable range. On the other hand, the authorities at the grid side test whether the power of participants at the demand side can follow the AGC signal properly. For example, an electric power organization, PJM (Pennsylvania-New Jersey Maryland Interconnection, regional transmission organization) uses performance scores to quantify the quality of frequency regulation service provided by the demand side participants. A participant is qualified only when its service can achieve a composite performance score of 0.75 or above [10].

Among various types of demand resources, such as electric vehicle [14] household appliances [15], heating, ventilation and air-conditioning (HVAC) systems in buildings are one of the most promising sources to provide this service. It is because they account for a large proportion of electric energy consumption [16, 17] and have great power use flexibility. Many researchers have explored the possibilities of HVAC systems in providing frequency regulation service to power grids. Chillers/heat pumps are attractive components in HVAC systems to be considered for providing this service, as they almost consume half of the energy consumption of the entire HVAC systems [18]. In the study of Cai and Braun [19, 20], a variable-speed packaged rooftop unit and a split heat pump were utilized for tests. The composite performance score reached 0.88, which could meet the requirement of PJM. The results also indicated that providing frequency regulation service has a negligible impact on indoor thermal comfort. In the study conducted by Su and Norford [21, 22], a chiller was utilized to provide frequency regulation service by resetting chilled water supply temperature setpoint. The achieved composite performance scores were 0.89, when following a 40-min RegA test signal, and 0.86 when following a 40-min RegD test signal. Here, RegA signal is a type of AGC signal with a relatively low frequency, and RegD signal is a type of AGC signal with a relatively high frequency. In addition, no appreciable impact on the average room temperature was observed. Some researchers paid their attention to fans due to their fast response. Lin et al. [23] conducted an experimental study of a variable air volume (VAV) system installed in a commercial building. The results showed that satisfactory frequency regulation service can be provided by the fans in HVAC systems (composite performance score: 0.89). In another experimental study conducted by Vrettos et al. [24, 25], the reported composite performance score of a fan reached up to 0.98 when following the RegD signal. In the study of Zhao et al. [26], the whole HAVC system (i.e., chillers, fans, and water pumps) was used to provide frequency regulation service by modulating the room temperature setpoint, the performance scores achieved were between 0.7991 and 0.8957 under different working conditions. The results indicated that frequency regulation control has a negligible impact on indoor comfort. In summary, previous studies mainly have shown: 1) HVAC systems in buildings can provide frequency regulation service with acceptable performance which can meet the requirement of power grids; 2) the indoor environment is not significantly affected when providing frequency regulation service.

One would ask: can the impacts of providing frequency regulation service on building indoor environment control be ignored in practice? In fact, previous studies mainly focused on the average room temperature when considering the impact of frequency regulation on building indoor environment. However, room temperature distribution is also a very important factor to the actual comfort of occupants [27]. According to our previous experimental study [28], although the average room temperature only had a 0.70 K offset at most, the temperature of the supply air to the air-conditioned space (i.e., air handling unit outlet air temperature) varied greatly, as much as 6.69 K, when providing frequency regulation service by variable-speed pumps in buildings. Such great fluctuation would deteriorate the thermal comfort of occupants seriously, particularly those near the air supply outlet. On the other hand, the fluctuation of the air handling unit (AHU) outlet temperature can result in a shortage of cooling/heating supply periodically. The studies [29, 30] reported that such shortage would cause the unbalanced distribution of cooling/heating among rooms and the thermal comfort of the rooms at an unfavorable position would suffer a lot from unbalanced distribution consequently. This problem could deteriorate when the fluctuation of AHU outlet temperature is larger and at low frequency.

This paper, therefore, proposes a novel control strategy, disturbance compensation enhanced control strategy, for building HVAC systems to prevent the sacrifice of indoor environment control quality when providing frequency regulation service to power grids. This strategy adopts a frequency disturbance compensation scheme developed based on the concept of "disturbance-observer-based control" [31], as the major original contribution, on top of a frequency regulation control scheme. An experimental test rig is constructed to validate the proposed control strategy. The test rig is a small-scale HVAC system in which a variable-speed pump is used for providing frequency regulation service. The proposed strategy is adopted to reduce the fluctuation of the AHU outlet temperature to prevent the sacrifice of the building indoor environment control quality.

The paper is organized as follows. The proposed disturbance compensation enhanced control strategy is presented and elaborated in Section 2. An outline of the test platform is given in Section 3. In Section 4, the experimental results of the control strategy validation are presented and analyzed. The conclusions are made in Section 5.

2. Development of the control strategy

This section presents the proposed disturbance compensation enhanced control strategy by elaborating its two control schemes, i.e., frequency regulation control scheme and frequency disturbance compensation scheme. The frequency regulation control scheme is used to modulate the power use of the pumps to provide frequency regulation service. The frequency disturbance compensation scheme is the core element of the proposed strategy, which aims to attenuate the impacts of providing frequency regulation service on indoor environment control.

2.1 Outline of the proposed control strategy

The proposed control strategy is developed on the basis of the conventional feedback control strategy with the intention to keep the feedback control mechanism as far as possible. For the ease of understanding the proposed new strategy and its innovation, a typical conventional control strategy of a typical constant air volume (CAV) system is shown in Fig. 1, which is used as the reference system of this study.

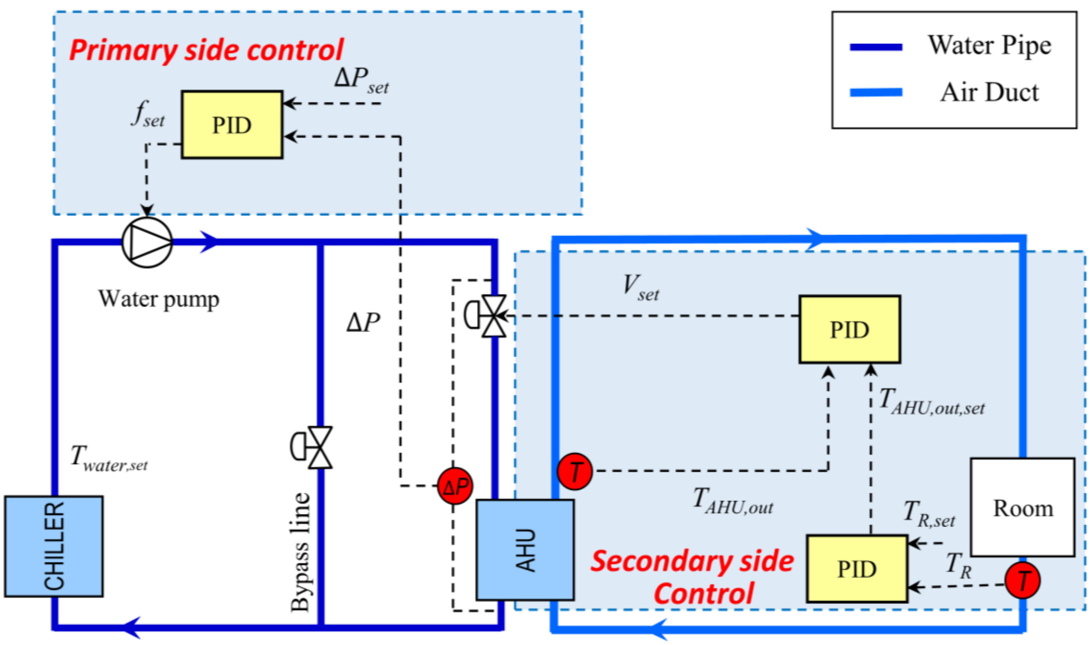

Fig. 1. The conventional control diagram of a constant air volume system.

The system control can be divided into the primary side control and the secondary side control. At the primary side, a proportional–integral–derivative (PID) controller is used to maintain the differential pressure of the most remote AHU at its setpoint (ΔP_{set}) by modulating the frequency setpoint (f_{set}) of the variable-speed pump. At the secondary side, a cascade control is adopted. Thus, a PID controller maintains the room temperature (T_R) at its setpoint ($T_{R,set}$) by resetting the setpoint of the AHU outlet air temperature ($T_{AHU,out,set}$). Another PID controller modulates the valve opening setpoint (V_{set}) to maintain the AHU outlet air temperature ($T_{AHU,out}$) at its setpoint.

The proposed new strategy is illustrated in Fig. 2, which includes two control schemes. One is the frequency regulation control (FRC) scheme at the primary side. Another one is the frequency disturbance compensation (FDC) scheme, at the secondary side.

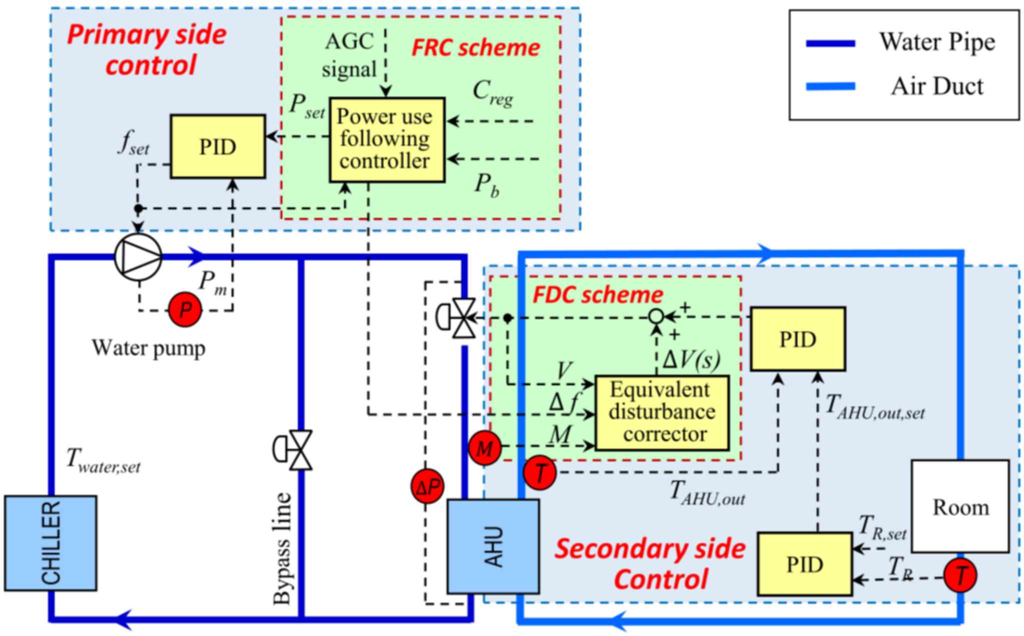

Fig. 2. Proposed control strategy of HVAC systems.

2.2 Frequency regulation control scheme

As mentioned above, the function of the frequency regulation control (FRC) scheme is to modulate the power use of the pumps to provide frequency regulation service. It can be observed that it includes a power use following controller which can determine the reference power use, as the power use setpoint (P_{set}) of the variable-speed pump, according to Eq. (1). The AGC signal is given by the power grid directly. P_b is the power use baseline, which refers to the power use of the pump under conventional control without providing frequency regulation service. It reflects the power use needed to meet the space heating/cooling demand. The regulation capacity (C_{reg}) is the capacity provided for frequency regulation service, i.e., the power use modulation magnitude around P_b. The P_b and C_{reg} can be obtained by assessing the heating/cooling demand and flexibility of the system. Related methods have been developed in previous studies [32, 33]. In addition, we have also developed a hierarchical optimal control strategy to determine P_b and C_{reg} [34]. In this study, a relatively large C_{reg} is selected directly (with P_b being determined accordingly), which has a large impact on indoor environment control. In this way, the proposed control strategy can be tested in a critical condition.

$$P_{set} = P_b + AGC\ signal \times C_{reg} \tag{1}$$

As mentioned in the Introduction, the AGC signal has been normalized to a range from -1 to 1. Therefore, the range of P_{set} is from P_b - C_{reg} to P_b+C_{reg}. In this study, a PID controller is used to modulate the power use of the pump P_m to follow this P_{set}, by adjusting the pump frequency setpoint, as shown in the primary side control in Fig. 2. There are two types of frequency mentioned in this paper, including the frequency of a power grid and the frequency of the HVAC devices (i.e., pumps). As illustrated in Fig. 3, the frequency of a power grid reflects the instantaneous balance of the power grid (i.e., the balance of power generation and power use) and should be strictly maintained within an acceptable range (e.g., 50$\pm$0.2 Hz in Hong Kong [35]). Normally, the increase of power generation or the decrease of power use can increase the frequency of the power grid. The decrease of power generation or the increase of power use can decrease the frequency of the power grid. Another type of frequency is the frequency of the pump. It is modulated to change the power use of the pump to provide frequency regulation service, which can be modulated in a large range (e.g., from 20 Hz to 50 Hz). The relationship between these two types of frequency can be illustrated by an example. A larger number of HVAC devices are controlled to increase/decrease their frequencies within the allowed range, which can collectively increase/decrease a great amount of power use of the power grid. This can then help to maintain the frequency of the power grid.

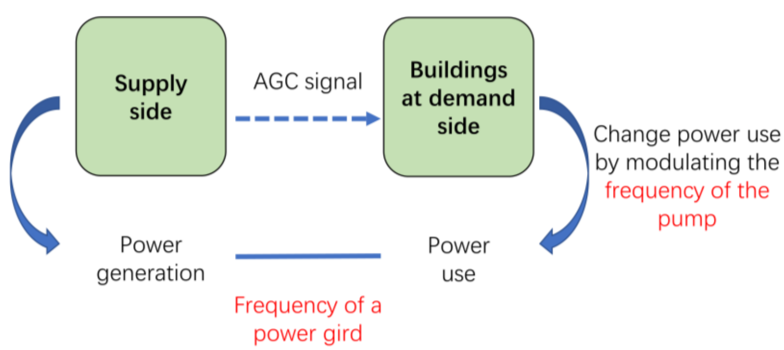

Fig. 3. Frequency control of HVAC devices *vs* frequency control of a power grid.

2.3 Frequency disturbance compensation scheme

As mentioned above, the function of the frequency disturbance compensation (FDC) scheme is to attenuate the impact of providing frequency regulation service on indoor environment control. Specifically, this scheme includes an equivalent disturbance corrector which adds an input $\Delta V(s)$, on the top of the PID controller originally used to control the valve opening, as shown in Fig. 2. Before elaborating its mechanism, the process of the impact of providing frequency regulation service on indoor environment control is first introduced, as shown in Fig. 4.

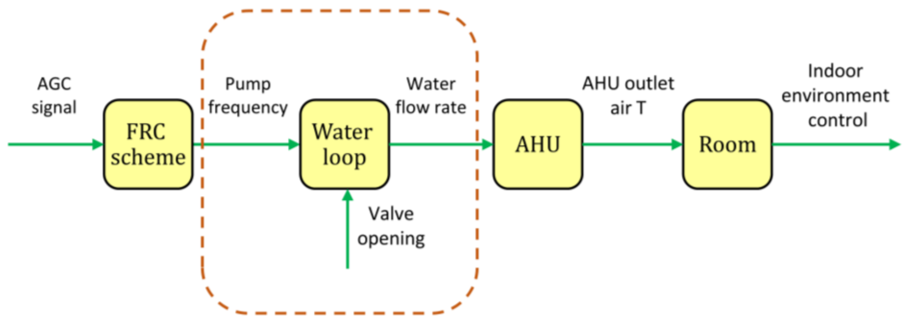

Fig. 4. Process of the impact of providing frequency regulation service on indoor environment control.

According to Eq. (1), the AGC signal, given by the power grid, can affect the power use setpoint of the pump. As mentioned in Section 2.2, to modulate the power use of the pump, its frequency is changed, which can further affect the water flow rate. Although there is a PID controller designed for maintaining the AHU outlet air temperature at its setpoint by modulating the valve opening setpoint (V_{set}) to control the water flow rate, our previous experimental results proved that this PID controller is unable to fulfill this task [28]. The reason is that the PID belongs to feedback control which can only respond when an offset occurs. This makes it naturally have a delay. When providing frequency regulation service, the frequency of the pump changes too rapidly, and the delay of the feedback control becomes more obvious, which finally results in a great fluctuation of the AHU outlet air temperature. As mentioned in the Introduction, the fluctuation of the AHU outlet air temperature can finally deteriorate the indoor environment control.

Actually, from the viewpoint of the original HVAC system control loop, the frequency of pump variation resulted from the AGC signals act as continuous and artificial disturbances [16]. To compensate the impact of this disturbance on the water flow rate, a frequency disturbance compensation scheme is proposed. This scheme adds an input $\Delta V(s)$ on the top of the PID controller originally used to control the valve opening. In this way, by enhancing the control performance of the valve on the water flow rate, this scheme can reduce the fluctuation of the AHU outlet temperature and prevent the sacrifice of the indoor environment control quality. As this scheme is developed on the basis of the "disturbance-observer-based control", a popular disturbance attenuation method, a brief overview of disturbance attenuation methods is first introduced.

2.3.1 An overview of disturbance attenuation methods

Disturbances widely exist in industrial systems which impose great challenges on the stable control of the systems [36]. In the last few decades, various disturbance attenuation methods have been developed. Some of them are widely accepted and applied, including disturbance observer, extended state observer, and unknown input observer [37]. The disturbance observer method was initially proposed by Ohnishi et.al [38] to improve torque and speed control by estimating load torque. The extended state observer method was developed to estimate comprehensive disturbances allowing for both unknown uncertainties and external disturbances [31]. The disturbance observer method is more relevant to the case of frequency regulation concerned in this study, which is elaborated in Fig. 5. y_r is the reference signal (i.e., setpoint). y is the actual output of the system $G(s)$. n is the measurement noise. $\bar{y}$ is the measured output. e_r is the offset between the measured output $\bar{y}$ and the setpoint y_r. To eliminate this offset, a feedback controller $C(s)$ is used. Normally, the controller output c is directly used as the input of the system $G(s)$. For disturbance-observer-based control, to compensate an external disturbance d, the inverse transfer function of the nominal model utilized for controller design $G_n^{-1}(s)$, is adopted to estimate the disturbance based on the $\bar{y}$ and u, which finally obtain the estimated disturbance (called lumped disturbance) d_l. Since the measurement noise and sensitivity of the identification algorithm. A stable filter $Q(s)$ is used, obtaining $\hat{d}_l$, which is then subtracted from the feedback controller output c. This $\hat{d}_l$ can compensate the external disturbance d [37]. As a result, the impact of the disturbance would be reduced, and system control stability and robustness are enhanced [31].

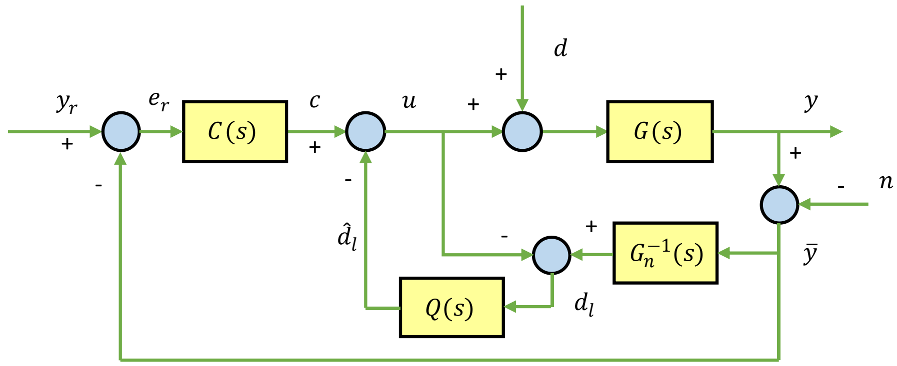

Fig. 5. Conceptual diagram of the disturbance-observer-based control.

2.3.2 Development of the frequency disturbance compensation scheme

Although the frequency disturbance compensation scheme is developed on the basis of disturbance-observer-based control, the situations and challenges in the application of frequency regulation control concerned this study have some differences with those in disturbance-observer-based control, as elaborated in Table 1.

Table 1 Comparison between disturbance-observer-based control and proposed frequency disturbance compensation control

	Disturbance-observer-based control	**Frequency disturbance compensation control**
Number of inputs	There is one input of the system, and the disturbance (*d*) is imposed on this control input. The solution is to subtract $\hat{d}_l$ from the original input of the system to compensate the external disturbance *d*.	For the system concerned (the water loop shown in Fig. 4.), there are two inputs, frequency of the pump and valve opening. The disturbance is imposed on one control input (frequency of the pump). The solution is to add Δ*V(s)* on the top of another input (valve opening) of the system to compensate the external disturbance. Here, Δ*V(s)* is added on rather than subtracted from the original valve opening. It is because Δ*V(s)* is directly the correction of the equivalent disturbance, corresponding to -$\hat{d}_l$.
Key challenge	The disturbance is unknown, and the key challenge is to “observe” or identify it.	The disturbance, pump frequency disturbance (Δ*f*), is known. The key challenge is, instead, to estimate the equivalent disturbance on another input (valve opening).

Use of a filter	Since the measurement noise and sensitivity of the identification algorithm, a stable filter is needed.	Since no measurement (noise) is involved and the corrector does not amplify the uncertainty/error of its inputs, no stable filter is needed.

For disturbance-observer-based control, the core element to obtain $\hat{d}_l$ is the inverse transfer function of the nominal model utilized for controller design $G_n^{-1}(s)$, which is developed based on the relationship between the control input (*u*) and output (*y*). Correspondingly, the core element in our case, equivalent disturbance corrector to generate $\Delta V(s)$ is developed based on the relationship between two inputs, valve opening (*V*) and frequency of the pump (*f*), and an output, water flow rate (*M*), as shown by Eq. (2) - Eq. (3). The head of the pump (*H*) can be described by Eq. (2) based on the pump characteristic curves. *H* can also be described by Eq. (3) according to the hydraulic model of the chilled water loop. *a*, *b*, c, *k*, and *m* are constant coefficients.

$$H = -a \cdot M^2 + b \cdot f \tag{2}$$

$$H = (c + k \cdot V^m) \cdot M^2 \tag{3}$$

By integrating Eq. (2) and Eq. (3), the relationship among valve opening, frequency, and water flow rate is obtained as shown by Eq. (4). In practice, the "power of *M"* may have some deviation from "2". Thus, it is also set as a coefficient, *e*, to be identified using experimental data. The $\Delta V(s)$ can then be obtained by taking the derivative of Eq. (5), as shown in Eq. (6), where Δf and *V* are available in the controllers. *M* can be measured by a water flow meter directly. Alternatively, *M* can also be estimated using the current differential pressure (ΔP) of the AHU and *V* when the water flow meter is not available. The coefficients, *n*, *m*, and *e* can be obtained by regression based on Eq. (5).

$$M^2 = \frac{f}{(a+c)/b + n \cdot V^m} \qquad n = \frac{k}{b} \tag{4}$$

$$M^e = \frac{f}{(a+c)/b + n \cdot V^m} \qquad n = \frac{k}{b} \tag{5}$$

$$\Delta V(\mathrm{s}) = \frac{\Delta f(s)}{(M^e \cdot n \cdot m \cdot V^{m-1})} \tag{6}$$

3. An outline of the test platform

A compact variable water flow HVAC system, which is used as the reference system and serves a laboratory space in this study, is constructed. A schematic of the system is shown in Fig. 6. This test platform mainly includes a variable-speed water pump, an electric heater, and an AHU. Here, the electric heater is installed to replace the function of a heat pump or boiler working in heating mode. The electric heater is constructed in a tank. The existing air-conditioning system (fan-coil unit) of this laboratory space, which acted as the cold source, generated the heating load for the test rig. The supply temperature setpoint of the existing fan-coil unit is set obviously lower than the room temperature during the tests so that the water valve of the fan-coil unit would remain open all the time to prevent the indoor temperature control of the test rig from being affected by the on/off control of the fan-coil unit. In this system, the AHU air loop represents the most remote loop of a CAV air-conditioning system. The bypass loop represents the other loops in the system. In Fig. 6, the controlled variables include the frequency setpoint of the water pump, the valve opening, and the heating rate of the electric heater. The measured variables are marked in circles, including the power use of the variable-speed water pump, the water flow rate, the differential pressures across the variable-speed pump and the remote pipeline, and the inlet/outlet air temperatures and inlet water temperature of the AHU.

The laboratory space served by the test rig is 6.8 m × 2.5 m × 2.7 m, which well matches the space served by an AHU of the same capacity in practice. This test rig is monitored and controlled by a computer station, which has a high-frequency data collection system and a free programmable control platform for implementing the strategies to be tested. In particular, the data were collected using the RS485 interface and Modbus Protocol. The photographs of the experimental platform are shown in Fig. 7.

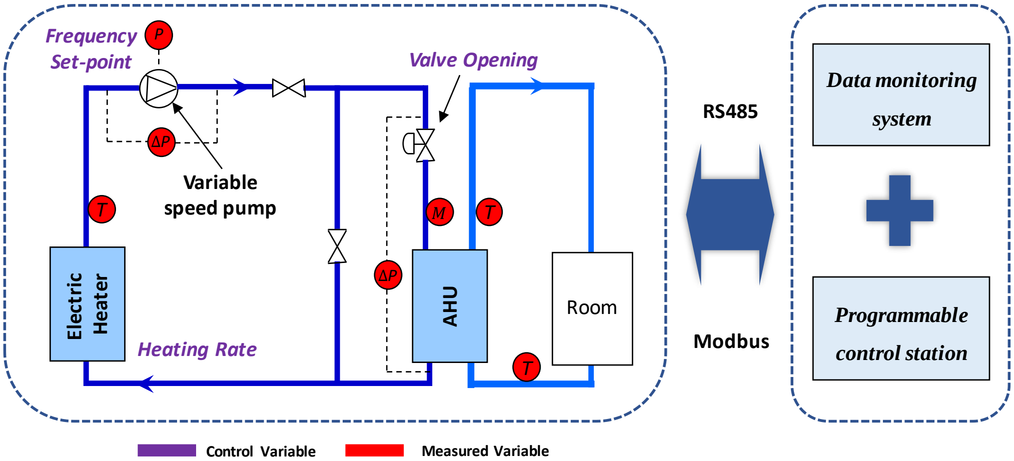

Fig. 6. Schematic of the test platform for providing frequency regulation service to power grids.

(A) An overview of the test platform

(B) The control cabinet

Fig. 7. HVAC system frequency regulation test rig for providing frequency regulation service to power grids.

4. Experimental tests and test results

This section presents the arrangement of the experimental tests and the test results. The proposed strategy is implemented in the automation system of the experimental test rig. The performance of the proposed strategy, particularly the frequency disturbance compensation scheme, is validated on the experimental test rig. Before that, the coefficients in the frequency disturbance compensation scheme are estimated.

4.1 Coefficients estimation in the frequency disturbance compensation scheme

As mentioned in Section 2.3.2, the coefficients (e, n, m) in the frequency disturbance compensation scheme can be identified according to Eq. (5), which represents the relationship among valve opening, frequency, and water flow rate. In the study, 382 scenarios covering all the normal working conditions are selected. The pump frequency is set from 20 to 45 Hz with an interval of 1 Hz. The valve opening is set from 15% to 50% with an interval of 1%. Note that the valve used in the test is a quick opening valve [39], therefore the valve opening has a much larger impact on water flow rate when the valve opening is small. Therefore, to achieve a better regression performance, only a certain range (15% to 50% decided according to our test rig) is concerned. This range can be appropriately adjusted for different systems.

By regression, coefficients in the frequency disturbance compensation scheme can be obtained eventually. The quantitative relationship among valve opening, pump frequency, and water flow rate is shown in Fig. 8. The coefficients in the frequency disturbance compensation scheme identified by regression are listed in Table 2.

Table 2 Identified coefficients in the frequency disturbance compensation scheme

Constant coefficient	***m***	***n***	***e***
Value	-0.903	471.1	1.09

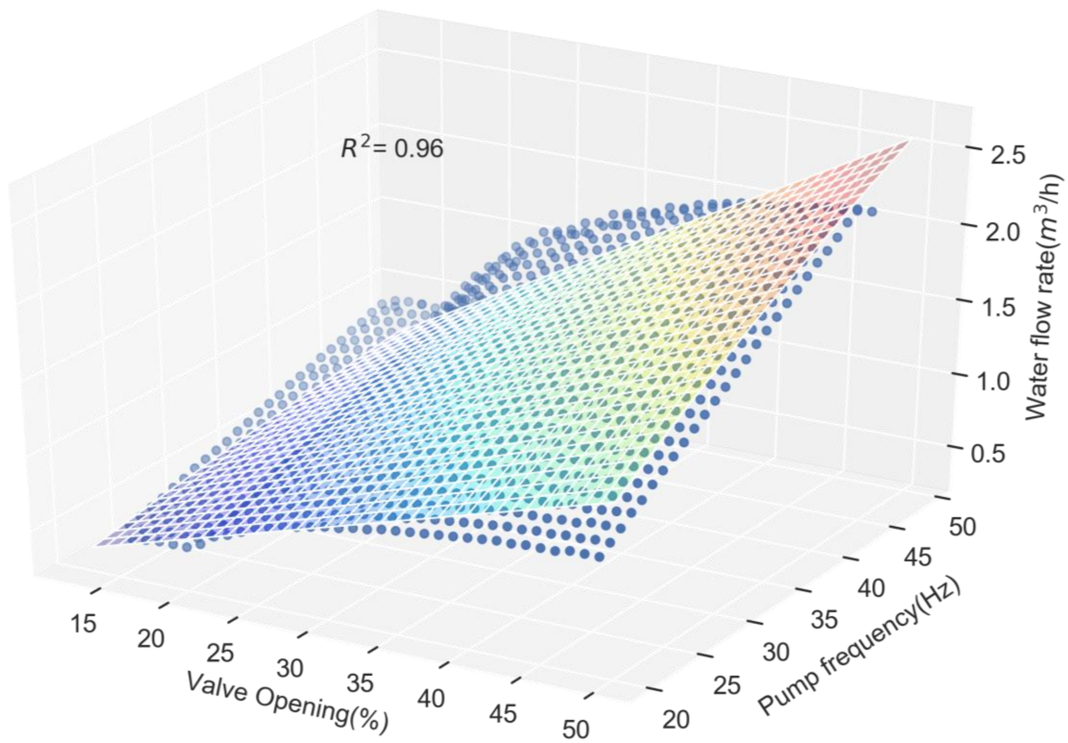

Fig. 8. The relationship among valve opening, pump frequency, and water flow rate.

4.2 Test and validation of the proposed control strategy

To evaluate the performance of the proposed control strategy, especially the frequency disturbance compensation scheme, control experiments are conducted on the test platform by using different control methods following the RegA and RegD test signals [40], respectively. These test signals are commonly used in frequency regulation service tests by previous studies, as mentioned in the Introduction. The scenarios of the control experiments include:

- Test Case 1: Following the RegA test signal with only the frequency regulation control (FRC) scheme while without the frequency disturbance compensation (FDC) scheme - RegA + control method 1;
- Test Case 2: Following the RegA test signal with both the FRC scheme and FDC scheme - RegA + control method 2;
- Test Case 3: Following the RegD test signal with only the FRC scheme while without the FDC scheme - RegD + control method 1;
- Test Case 4: Following the RegD test signal with both the FRC scheme and FDC scheme - RegD + control method 2;

Each test case was conducted at least two times to ensure the repeatability of experiments and the reliability of general conclusions, while only the result of one of them is presented in this paper. Figs 9, 11, 13, and 15 show the reference power use P_{set}, measured power use P_m, and frequency of the pump *f* (not the frequency of the power grid) in the above four test cases, respectively. Figs 10, 12, 14, and 16 show the valve opening and AHU outlet air temperature in the same test cases, respectively. It is worth noticing that, prior to the period providing frequency regulation service, the system has worked a period of time and reached a stable condition, while only the last 600s before providing frequency regulation service is shown in the figures. The setpoint of the AHU outlet air temperature is 42°C.

4.2.1 Test Case 1: RegA without using the frequency disturbance compensation scheme

It can be observed from Fig. 9 that the measured power use followed the reference power use of the pump very well. The composite score obtained is 0.988 (correlation score: 0.995, delay score: 0.999, precision score: 0.970). This score is much higher than the minimum requirement, 0.75, indicating that the pump can fulfill the requirements set in the grid regulation standard.

Fig. 10 shows that, without using the frequency disturbance compensation scheme, although the valve opening is controlled to maintain the AHU outlet air temperature, it is still unable to compensate the offset of the AHU outlet air temperature. The AHU outlet air temperature had a fluctuation of 5.00 K (from 39.65°C to 44.65°C). Since the AHU outlet air temperature equals the supply air temperature to space, such great fluctuation can directly deteriorate the thermal comfort of occupants significantly particularly those near the air supply outlet. On the other hand, this great fluctuation of AHU outlet temperature can deteriorate the unbalance distribution of cooling/heating among zones. The thermal environment of rooms at an unfavorable position would suffer a lot from unbalanced distribution consequently.

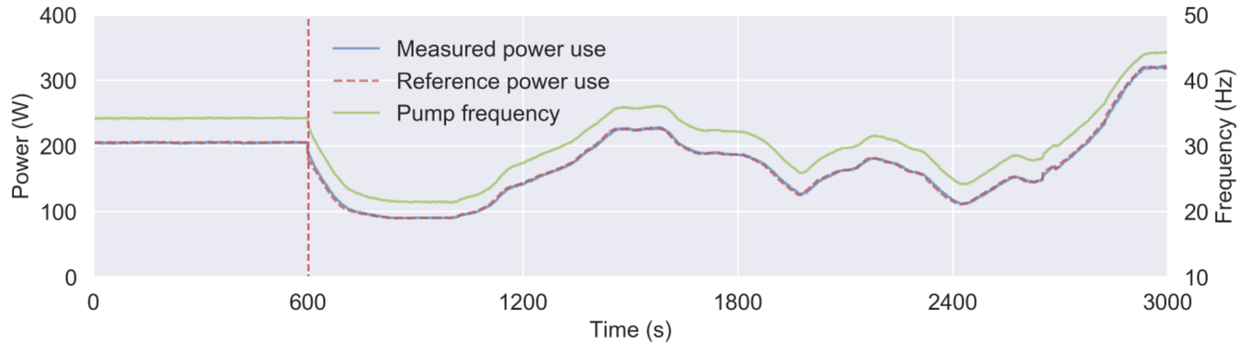

Fig. 9. Reference power use, measured power use, and pump frequency - Test Case 1.

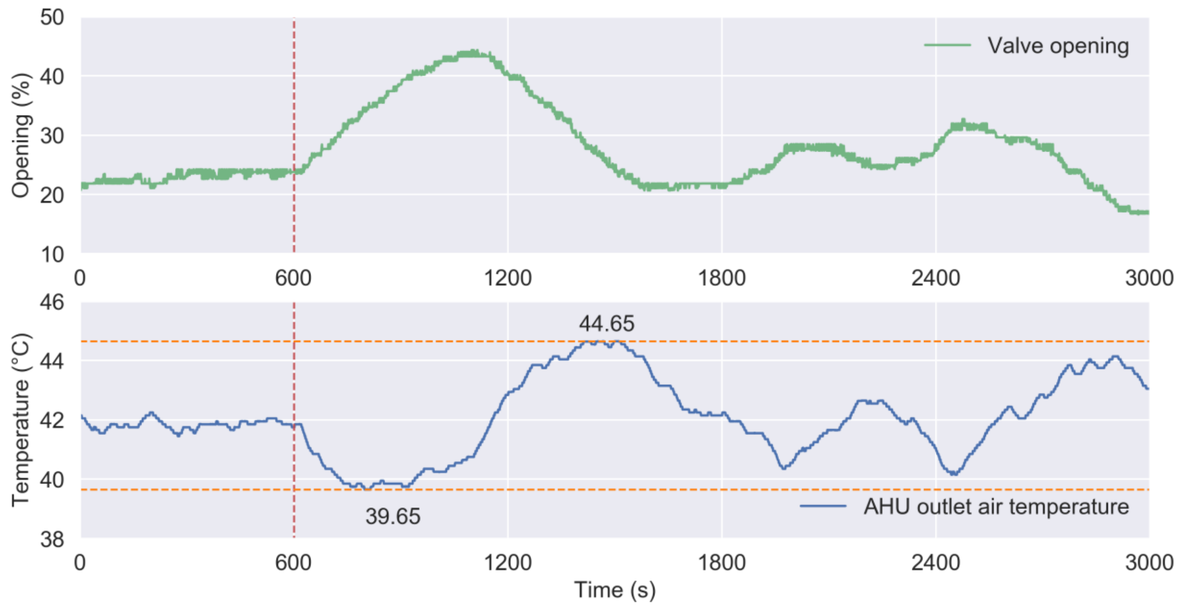

Fig. 10. Valve opening and AHU outlet air temperature - Test Case 1.

4.2.2 Test Case 2: RegA using the frequency disturbance compensation scheme

In Test Case 2, the experiment was conducted after adopting the developed frequency disturbance compensation scheme. In principle, after adopting this scheme, the valve opening is affected, which might affect the power use and then the quality of frequency regulation service in reverse. To check such effect is, therefore, one of the tasks of this study. Fortunately, it can be observed from Fig. 11 that the measured power use could also follow the reference power use accurately and timely when the frequency disturbance compensation scheme was adopted. The performance scores (i.e., composite score: 0.988; correlation score: 0.995; delay score: 0.999; precision score: 0.969) are also well meet the requirement. By comparison with Test Case 1, it can be found that after adopting the frequency disturbance compensation scheme, the performance of the frequency regulation service was almost unchanged. On the other hand, the fluctuation magnitude of the AHU outlet air temperature, shown in Fig. 12, decreased significantly, i.e., from 5.00 K to 2.40 K (between 40.65°C and 43.05°C). The decreased fluctuation magnitude of the AHU outlet air temperature means the improvement of the thermal comfort of the occupants (particularly those near the air supply outlet) and the improvement of the indoor environment of rooms at an unfavorable position.

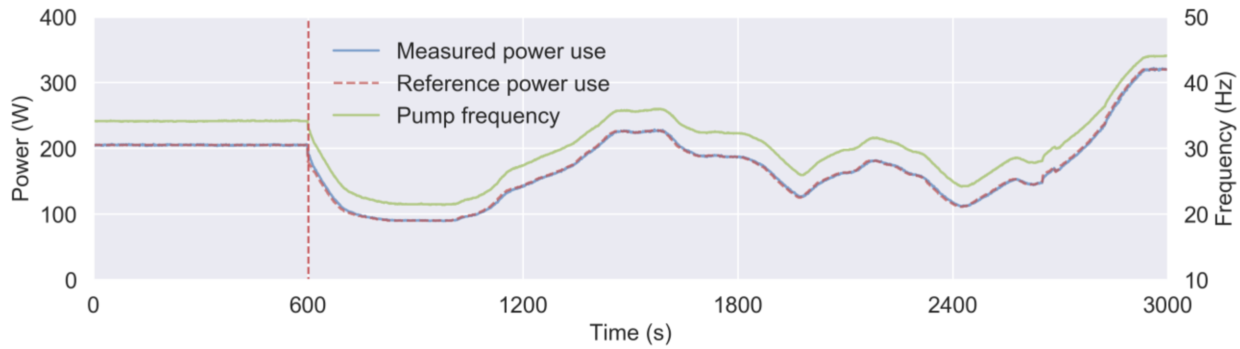

Fig. 11. Reference power use, measured power use, and pump frequency - Test Case 2.

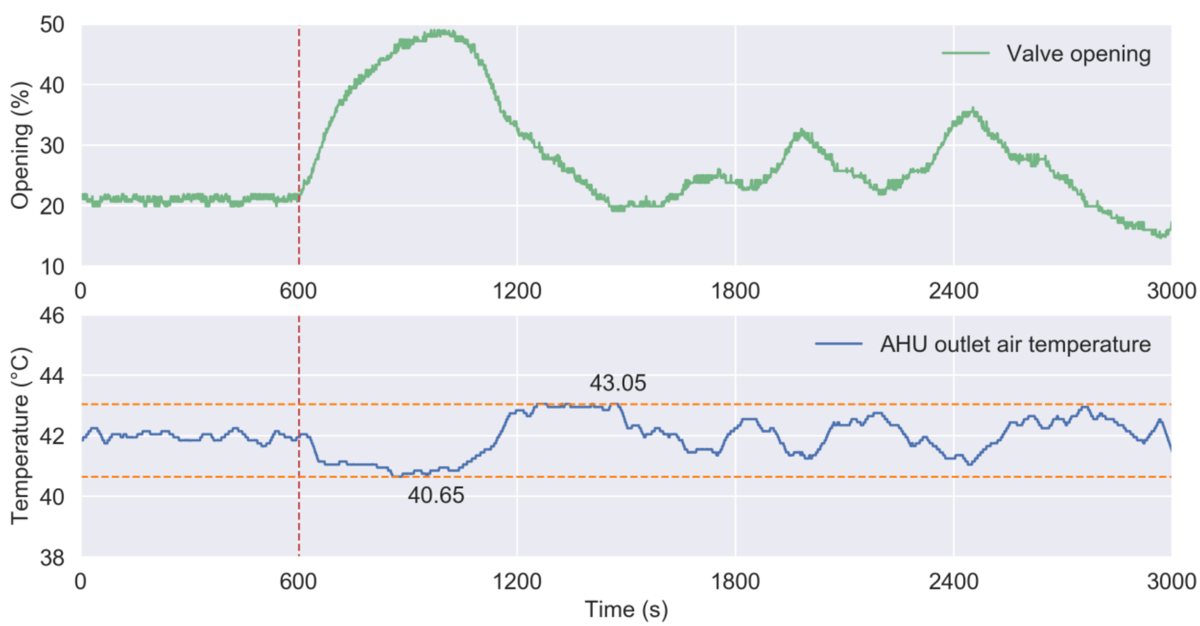

Fig. 12. Valve opening and AHU outlet air temperature - Test Case 2.

4.2.3 Test Case 3: RegD without using the frequency disturbance compensation scheme

Fig. 13 shows the reference power use, measured power use, and frequency in Test Case 3. The composite score obtained was 0.966 (correlation score: 0.986, delay score: 1.000, precision score: 0.911). This score was lower than that in Test Case 1, which result from the high requirement of RegD in terms of the response speed. Fig. 14 shows that, when following the RegD test signal, the AHU outlet air temperature had a fluctuation of 7.61 K (between 38.94°C and 46.55°C). This fluctuation was much larger than that when following the RegA signal. The reason is that the RegD signal is more volatile. Under such circumstances, the delay of the feedback control becomes more obvious.

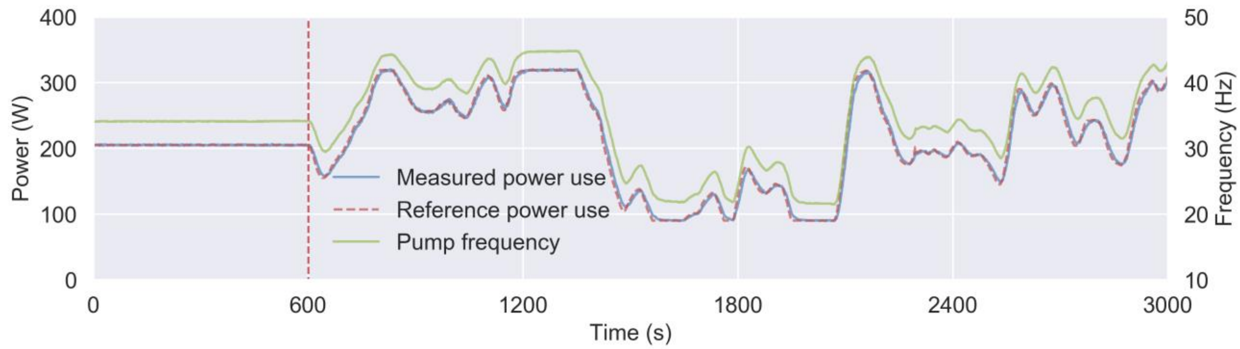

Fig. 13. Reference power use, measured power use, and pump frequency - Test Case 3.

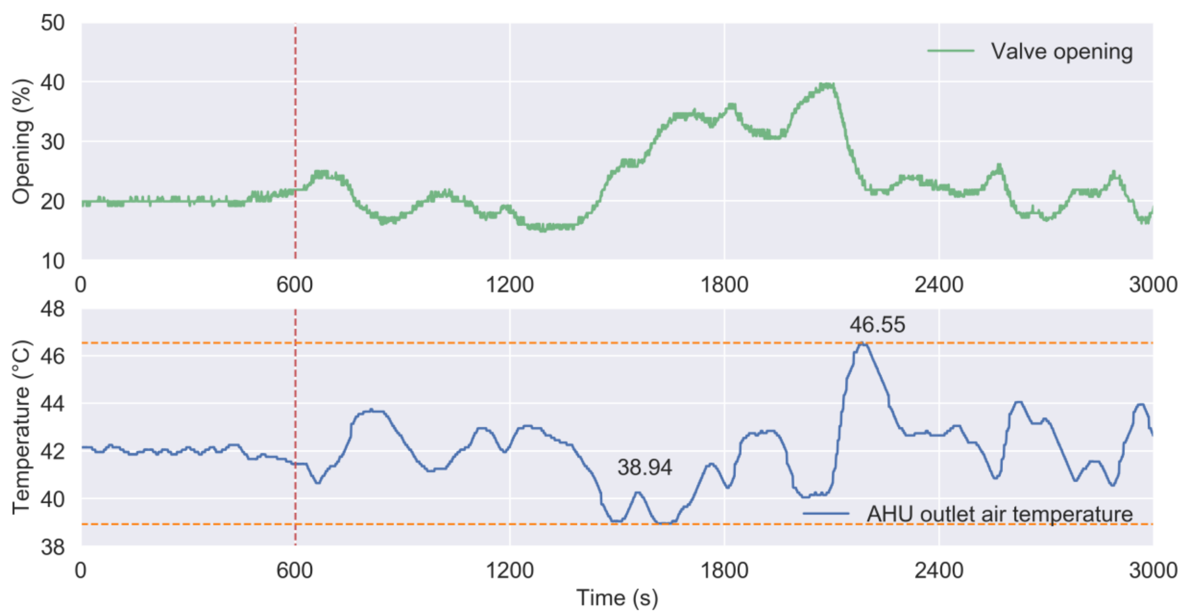

Fig. 14. Valve opening and AHU outlet air temperature - Test Case 3.

4.2.4 Test Case 4: RegD using the frequency disturbance compensation scheme

It can be observed from Fig. 15 that the measured power use could also follow the reference power use very well after adopting the frequency disturbance compensation scheme. The composite score obtained was 0.965 (correlation score: 0.985, delay score: 1.000, precision score:

0.910), which was almost the same as that in Test Case 3 without using the frequency disturbance compensation scheme. According to this comparison, we can draw the same conclusion as in Section 4.2.2 that the frequency disturbance compensation scheme has a neglectable impact on the quality of the frequency regulation service.

Fig. 16 shows that, although the fluctuation magnitude of AHU outlet air temperature was much larger when following RegD compared with the case following RegA signal, the frequency disturbance compensation scheme can also work very effectively to reduce this fluctuation, i.e., from 7.61K to 2.79 K. This result further verifies the effectiveness of the frequency disturbance compensation scheme developed.

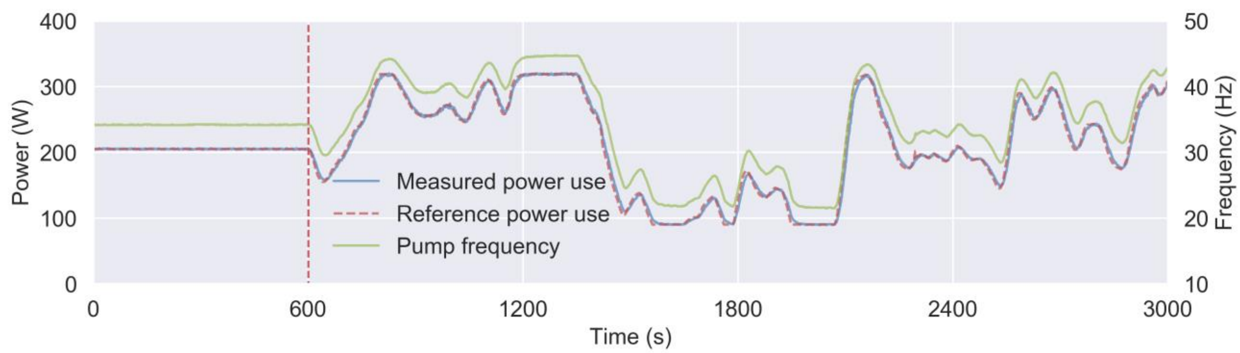

Fig. 15. Reference power use, measured power use, and pump frequency - Test Case 4.

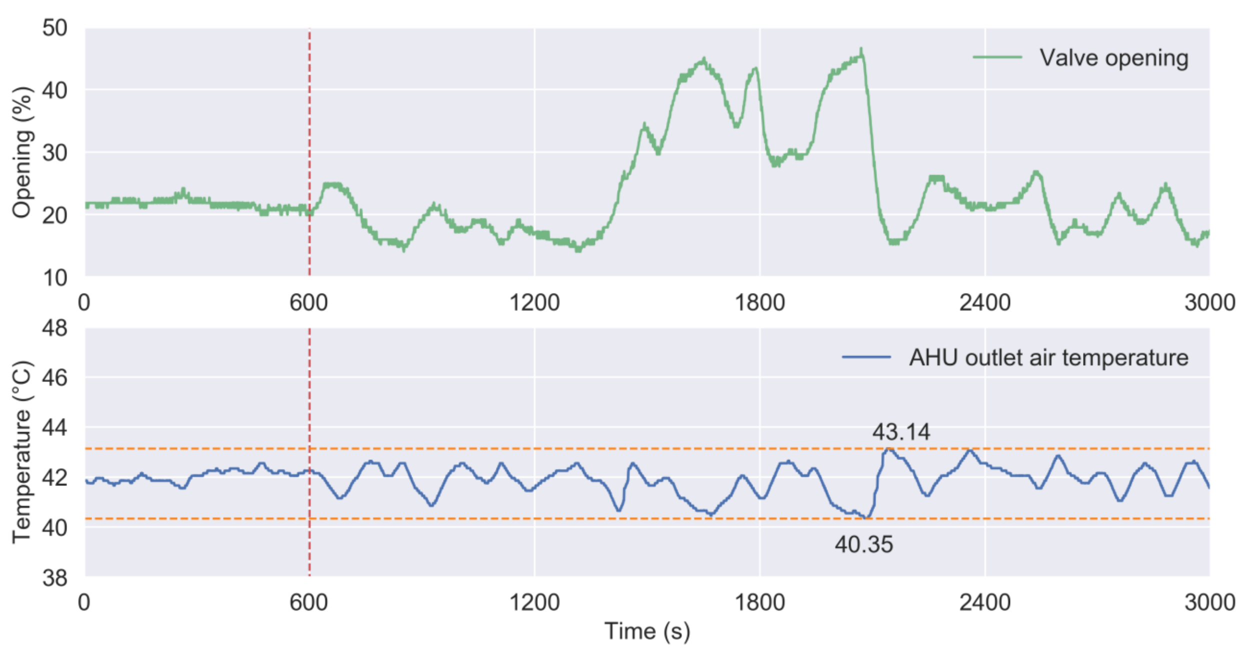

Fig. 16. Valve opening and AHU outlet air temperature - Test Case 4.

4.3. Experimental investigation on the mechanism of the frequency disturbance compensation scheme

In this section, the mechanism of the frequency disturbance compensation scheme is investigated experimentally and elaborated by comparing the results of Test Case 3 and Test Case 4. The detailed dynamic responses of the control loops during the period from 1300s to 1700s are selected for in-depth analysis, as shown in Fig. 17.

From 1300s to 1350s, the valve opening and AHU outlet air temperature in two test cases were nearly the same. After the 1350s, the pump frequencies in both of these two test cases experienced a significant decrease. In Test Case 3, the valve opening was only modulated by a PID controller according to AHU outlet air temperature $T_{AHU,out}$ and its setpoint $T_{AHU,out,set}$. Therefore, it just increased gradually after the $T_{AHU,out}$ deviated from its setpoint (i.e., 42°C). By comparison, in Test Case 4, apart from a PID controller, an extra output of the frequency disturbance compensation (FDC) scheme, $\Delta V(s)$ was added. It can be observed from Fig.17 that, after the frequency of the pump decreased from 1350s, the FDC scheme generated positive output on top of the PID controller to increase the valve opening. In particular, from 1350s to 1425s, the $T_{AHU,out}$ in two cases were almost the same due to delay. This means the outputs of two PID controllers in two test cases were almost the same during this period. However, at the 1425s, the valve opening in Test Case 4 was much larger than that in Test Case 3. This means that the larger valve opening in Test Case 4 actually all resulted from the output of the FDC scheme, $\Delta V(s)$. After the 1425s, the AHU outlet air temperature in Test Case 4 recovered quickly, which guaranteed stable control and less fluctuation of AHU outlet air temperature. In this case, it can be found that the frequency disturbance compensation scheme, as a proactive control scheme, can effectively remedy the defect of feedback (PID) control.

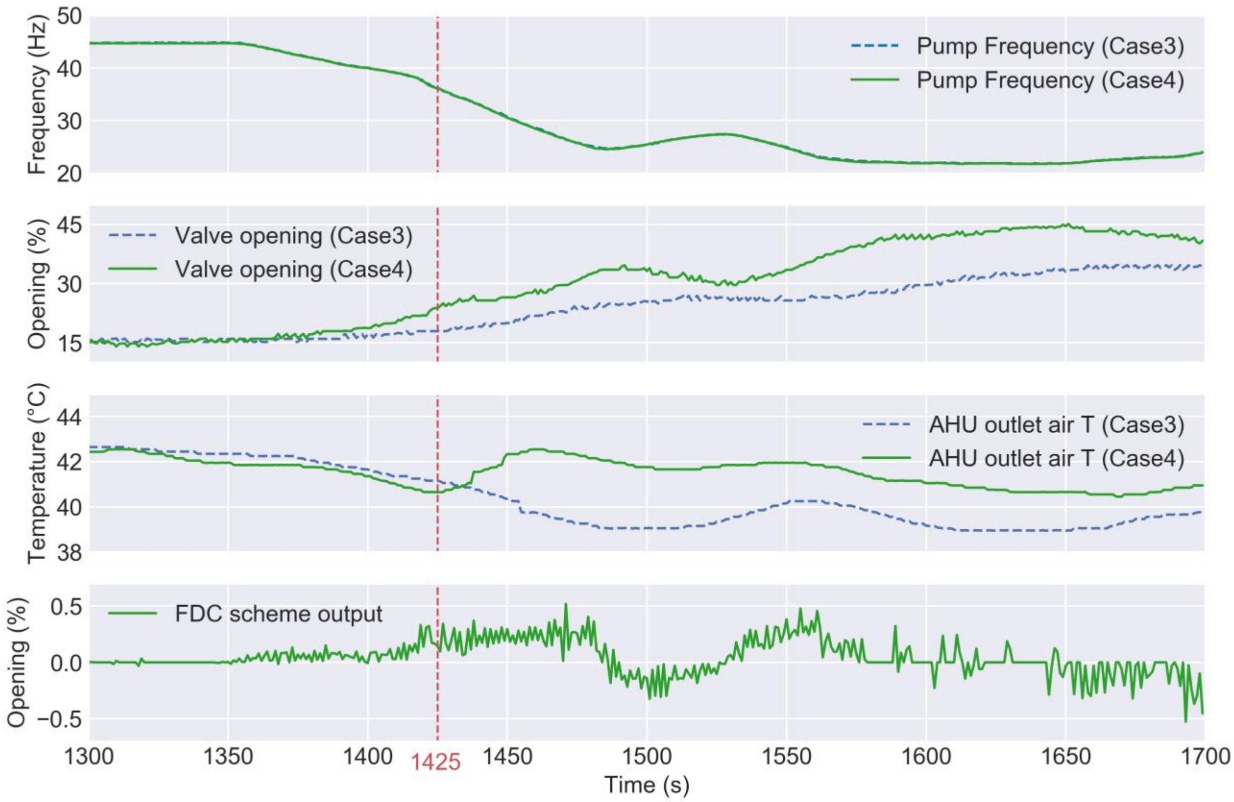

Fig. 17. Detailed dynamic effects of frequency disturbance compensation scheme on the frequency of the pump, valve opening, and AHU outlet air temperature - Comparison between Test Case 3 and 4.

4.4. Impacts on building indoor environment control, range of valve opening and valve wear level

As mentioned in the Introduction, previous studies only considered the average indoor air temperature (typically using the return air temperature of a room) when studying the impact of providing frequency regulation service on indoor environment control. For the comparison with these studies, the average indoor air temperature (i.e., the AHU inlet air temperature) in four Test Cases are summarized in Table 3. Especially, this study also studies the supply air temperature to the room (i.e., $T_{AHU,out}$) after considering its non-negligible impacts on indoor environment control. Thus, the supply air temperatures in four Test Cases are also summarized in Table 3.

The indoor air temperature setpoint was 26 °C. It can be observed from the table that under method 1 (without the frequency disturbance compensation scheme), the average indoor air temperature had a maximum deviation of 0.73 K (in the case when following RegD signal:

26.73°C). This indicates that the impact of providing frequency regulation service on average indoor air temperature is not significant, which is consistent with the conclusions of previous studies. On the other hand, since the fluctuation of supply air temperature is significant in this case, it can be inferred that the fluctuation of the supply air temperature in previous studies could also be significant. As the proposed frequency disturbance compensation scheme in this study has been proved to have a good performance to reduce the fluctuation of the supply air temperature, it could also be used as a reference for other studies to solve the same problem. In addition, it can also be found from the table that this scheme can further decrease the fluctuation range of the average indoor air temperature.

Table 3 Supply and average air temperature of the room in four Test Cases

Controlled variable	**Control method**	**RegA**			**RegD**		
		Min(°C)	*Max(°C)*	*Range(K)*	*Min(°C)*	*Max(°C)*	*Range(K)*
Average indoor T	*Method 1*	25.62	26.63	**1.01**	25.42	26.73	**1.31**
	Method 2	25.62	26.52	**0.90**	25.52	26.62	**1.10**
Supply air T to room	*Method 1*	39.65	44.65	**5.00**	38.94	46.55	**7.61**
	Method 2	40.65	43.05	**2.40**	40.35	43.14	**2.79**

In this study, the valve opening is also investigated because it is affected after adopting the frequency disturbance compensation scheme. It can be observed from Table 4 that the range of valve opening increased slightly after adopting the frequency disturbance compensation scheme. In another word, this scheme actually uses the energy stored in the valve (i.e., energy loss) as a kind of storage to provide the frequency regulation service. This energy loss is existing in practice because most pumps are oversize and work under a higher level (higher frequency or number) than that needed to guarantee a sufficient cooling supply in practice. In this study, the proposed frequency disturbance compensation scheme can just properly use this energy storage without cost.

Table 4 The range of valve opening in four Test Cases

Controlled variable	**Control method**	**RegA**			**RegD**		
		Min(°C)	*Max(°C)*	*Range(K)*	*Min(°C)*	*Max(°C)*	*Range(K)*

Valve opening	*Method 1*	16.63	44.35	27.72	14.83	39.7	24.87
	Method 2	14.65	48.99	34.34	14.06	46.64	32.58

As the range of valve opening is affected after adopting the frequency disturbance compensation scheme, we also investigate the valve wear level (related to its service life). The valve wear level can be assessed by the travel speed [41], as described by Eq. (7). It reflects the average speed a valve moves. Here, *V[i]* is the value of valve opening at the time *i*. *t* is the total time.

$$Travel\ speed = \frac{\sum_{i=0}^{n} |V[i+1]-V[i]|}{t} \tag{7}$$

According to Eq. (7), the travel speed of the valve under various working conditions can be obtained, as shown in Table 5. As mentioned in Section 4.2, each test case was conducted at least two times. The average valve travel speed of all tests in each test case is presented in Table 5. Here, 1 *round/min*, means with this travel speed, a valve can travel the whole round (from a fully open position to a fully closed position and then back to a fully open position) in one minute.

It can be observed from the table that the average valve travel speed under the stable condition without providing frequency regulation (FR) service was the shortest, which is quite reasonable. When providing frequency regulation service, the average valve travel speed increased obviously (about 26%). However, there was no obvious further increase (neither decrease) of the average valve travel speed after adopting the frequency disturbance compensation scheme. In summary, although the opening range increased after adopting the scheme, the average valve travel speed is not affected significantly.

Table 5 Average valve travel speed (round/min) under different working conditions

Normal operation without FR	RegA		RegD	
	Method 1	Method 2	Method 1	Method 2
0.084	0.107	0.106	0.105	0.103

5. Conclusion

In this paper, a disturbance compensation enhanced control strategy of HVAC systems is proposed, which adopts a frequency disturbance compensation scheme on top of a frequency regulation control scheme. This frequency disturbance compensation scheme is used to attenuate the impact of providing frequency regulation (FR) service on indoor environment control. The performance of the proposed control strategy is validated experimentally. The main conclusions can be made as follows.

- The impact of the conventional FR control (with only the frequency regulation control scheme) on the average indoor air temperature is not significant. This is consistent with the conclusions of previous studies. In the test cases, the average indoor air temperature only had a maximum deviation of 0.73 K when providing FR service.
- The indoor environment can be affected significantly under conventional FR control. In the experimental tests, the fluctuation magnitude of the supply air temperature to the room was as high as 5.00 K and 7.61 K under the conventional FR control when following RegA and RegD test signals, respectively. The great fluctuation magnitude of the supply air temperature affects the thermal comfort of occupants (particularly those near the air supply outlet). It also makes the room at an unfavorable position suffer a larger temperature fluctuation.
- The use of the frequency disturbance compensation scheme can achieve significant improvement in the building indoor environment control. When adopting this scheme, the fluctuation magnitude of the supply air temperature to the room was reduced significantly, i.e., from 5.00 K to 2.40 K when following the RegA test signal and from 7.61K to 2.79 K when following the RegD test signal. The decreased fluctuation magnitude of the supply air temperature means the improvement of the thermal comfort of the occupants (particularly those near the air supply outlet) and the improvement of the indoor environment of rooms at an unfavorable position.
- The quality of the FR service provided by the variable speed pump is not affected when the frequency disturbance compensation scheme is adopted. The composite performance scores in the tests were 0.988 and 0.966 under conventional control and 0.998 and 0.965 under the control with frequency disturbance compensation scheme when following RegA and RegD signals, respectively.

- The wear level of the valve is affected slightly when adopting the conventional FR control while adopting the frequency disturbance compensation scheme does not further affect the wear level of the valve. In the test without providing FR service, the average valve travel speed was 0.084 *round/min*. It increased to 0.107 and 0.105 *round/min* under conventional FR control, and to 0.106 and 0.103 *round/min* under FR control with frequency disturbance compensation scheme when following the RegA and RegD test signals, respectively.

The proposed control strategy with the frequency disturbance compensation scheme can be applied for different HVAC systems, such as CAV and variable air volume (VAV) air-conditioning systems. One chilled water loop often serves multiple zones (i.e., multiple AHUs) in applications. Although the AHUs in different zones make a similar response to the pump frequency changes, the interaction between a water loop and multiple AHUs should be further investigated to ensure the effectiveness of the frequency disturbance compensation scheme in practical applications. In further work, after quantitatively analyzing the impact of the fluctuation of supply air temperature on indoor environment control, a metric will be proposed which can also specifically quantify the improvement of indoor environment control when adopting the proposed strategy in this study.

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