Optimal Strategy for Participation of Commercial HVAC Systems in Frequency Regulation

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***Abstract*—In this article, an optimal strategy is proposed for heating, ventilation, and air conditioning (HVAC) systems in commercial buildings to fairly ensure the occupants' comfort, while participating in frequency regulation. With the nonlinear relationship between temperature setpoint and the fan power, the differences between indoor temperatures and temperature setpoints are considered as the objective to fairly ensure the occupants' comfort. By the proposed strategy, the task for frequency regulation is optimally distributed according to the capacity of HVAC systems in a load aggregator. Therefore, performing frequency regulation and fairly ensuring occupants' comfort are achieved simultaneously. Simulations on a two-area interconnected power system show that the proposed optimal strategy can improve the quality of frequency, reduce traditional generator regulation, and ensure the occupants' comfort fairly.**

***Index Terms*—Area control error (ACE), frequency regulation, heating, ventilation, and air conditioning (HVAC) system, occupants' comfort, optimal dispatch (OD).**

I. Introduction

Renewable energy sources (RESs) have been increasing significantly and are expected to account for approximately 24% of the generation capacity by 2040 [1]. The large-scale RES integration into the power grid will impact the generation-load balance and result in higher frequency regulation procurements due to the randomness and uncertainty of RESs [2], which will challenge the frequency stability of power systems. Traditionally, the fossil-fueled generators are used to ensure frequency stability by regulation. However, due to slow ramping rates, traditional generators cannot balance the fast generation-load mismatch caused by RESs. Therefore, many fast-responding and expensive generators must be added. As an alternative, utilizing flexible demand-side resources, such as electric vehicles, air conditioners, water heaters, and refrigerators, is a more cost-effective approach [3]–[15]. In particular, the heating, ventilation, and air conditioning (HVAC) system as an attractive resource at demand side has been paid much attention for ensuring frequency stability in recent years.

HVAC systems can be divided into residential and commercial types, which are motivated by the inherent differences in residential and commercial HVAC units. The former is typically an on/off type unit [16]–[19], while the latter can be continuously controlled to track the frequency signal closely. Therefore, HVAC systems in commercial buildings can show the better potential for frequency regulation. On the other hand, in the United States, commercial buildings' consumption accounts for 35.5% of the total electricity [20], while HVAC systems are about 45% of the total electricity in commercial buildings [21]. With a large proportion of both energy and electricity consumption, commercial HVAC systems can provide frequency regulation service with little compromise on occupants' comfort.

For commercial HVAC systems, heat pumps, chillers, and the fan power all can be controlled for frequency regulation. Heat pumps with variable speed drive have shown the potential for frequency regulation in [22]. As shown in [23], the direct load control enabled by variable speed heat pump reduces grid frequency deviations. For chillers, with the frequency range of [1/(60 min), 1/(3 min)], the chillers can respond to frequency regulation with a small impact on the occupants' comfort [12]. Su and Norford [24], [25] evaluated the chillers control for frequency regulation, which shows that the chillers control has better performance than the traditional generator. Compared with the chillers control, the fan power control is more favorable to provide frequency regulation, because the commercial HVAC system is always equipped with variable frequency drives (VFDs) that can quickly and continuously regulate the fan power to follow frequency signals [26]. Up to 15% of the fan power capacity can be deployed for frequency regulation [27], [28]. In [27], a simplified model of HVAC system is used to evaluate frequency regulation by the fan power control. In [26], the fan power control for frequency regulation is investigated through experiments in a closed loop. In [29] and [30], HVAC systems in commercial buildings can track a frequency regulation signal with high accuracy and minimal occupant discomfort by fan power control. To have a better understanding of the economic values of potential resources for frequency regulation, the potential capacities

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provided by the fan of HVAC systems of all commercial buildings in Hong Kong are quantified [31].

In general, temperature adjustment, duct static pressure control, and fan speed control are the three ways to consider the fan power control to track a frequency regulation signal. Temperature adjustment can control the fan power of commercial HVAC systems by changing the thermostat setpoint. Zhao *et al.* [32], [33] investigated the technical potential for converting HVAC systems into the providers for frequency regulation by adjusting the indoor temperature setpoint. In [34], a predictive model of the fan power is developed, where temperature adjustment is used to change the HVAC systems' power in open loop. Changing the setpoint value of duct static pressure can control the fan speed to regulate fan power for frequency regulation. A simplified HVAC system in [33] is modeled to explore the potential of commercial HVAC systems for frequency regulation by controlling the duct static pressure. In [35] and [36], for a typical HVAC system in commercial buildings, the amount of ancillary power that can be provided by controlling the duct static pressure is investigated. As for the fan speed control, a control strategy is proposed for frequency regulation by considering the regulation signal and the feedback of a fan speed signal in [28]. In [37], a feedforward architecture is proposed to control the fan power to track regulation signals, and the proposed strategy is tested based on a calibrated building model with high-fidelity nonlinearity. Compared with temperature adjustment and duct static pressure control, the fan speed control is the fastest method to regulate fan power for frequency regulation [38].

The occupants' comfort is another problem that cannot be ignored in the participation of HVAC systems for frequency regulation. In [16], the potential of providing intrahour load balance is investigated by using aggregated HVAC loads, and a temperature-priority method is proposed to maintain occupant-desired indoor temperatures. In [39], a distributed control strategy is proposed for HVAC systems in commercial buildings to minimize the weighted sum of energy consumption and the thermal discomfort. While ensuring the occupants' comfort, different strategies have also been proposed to address the participation of HVAC systems in frequency regulation by controlling heat pumps in [22] and [23], chillers in [12], [24], and [25], and the fan power in [28] and [32]–[38].

However, the existing research is mainly devoted to evaluate the capability of HVAC systems for regulation [12], [22]–[27], or track a regulation signal [28], [29]–[38], or mention Internet of Things-based thermal model for buildings [40], and the participation of commercial HVAC systems in regulation as a closed-loop dispatch is rarely discussed [26], [41], [42]. Besides, fairly ensuring occupants' comfort is also neglected, while performing frequency regulation. Therefore, we develop a closed-loop optimal strategy to cope with the issues. The major contributions of this article are summarized as follows.

1) A hierarchical framework is proposed to enable the coordination between the power grid and HVAC systems, in which the load aggregator can manage lots of HVAC systems to participate in frequency regulation.
2) An optimal closed-loop strategy is proposed in the load aggregator to control HVAC systems to undertake the real-time dispatch for frequency regulation within the allowed temperature range.
3) The differences between indoor temperatures and their temperature setpoints are considered as the objective to fairly ensure the occupants' comfort.

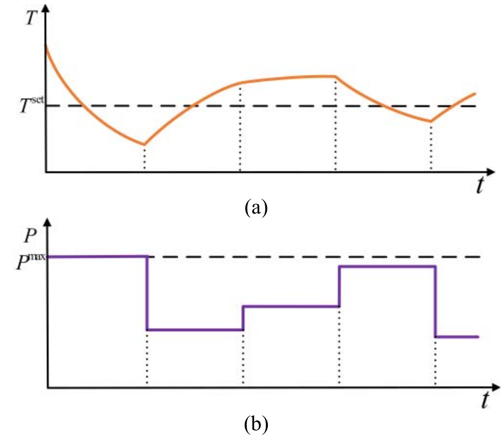


Fig. 1. Thermal behaviors of HVAC systems. (a) Indoor temperature curve. (b) Fan power for ensuring comfort degree.

The remainder of this article is organized as follows. In Section II, the problem on the participation of HVAC systems in frequency regulation is discussed. In Section III, the model of HVAC systems in commercial buildings is introduced. In Section IV, the hierarchical framework and capacity for regulation are addressed. In Section V, an optimal strategy is proposed for HVAC systems to participate in frequency regulation while ensuring the comfort of occupants. In Section VI, simulation results are presented to validate the effectiveness of the proposed strategy. Finally, the conclusions are drawn in Section VII.

II. Problem Description

A. HVAC System for Frequency Regulation

A typical HVAC system in commercial buildings, called the variable air volume (VAV) system [27], [34], [38], is equipped with the VFD for driving the fan of the HVAC system. The fan power of an HVAC system can be continuously regulated by VFD to ensure the occupant's comfort, while performing frequency regulation.

1) Ensuring the Occupants' Comfort: The occupant generally prefers a temperature setpoint to guarantee indoor comfort. For instance, in winter, the setpoint could be higher, and in summer, it could be lower. The primary objective of an HVAC system is to ensure the occupant's temperature setpoint. In this condition, the fan power has to be controlled according to the difference between indoor temperature and the temperature setpoint, as shown in Fig. 1. If the indoor temperature is above the temperature setpoint, the fan power will increase to supplement supply air flow rate for ensuring the temperature

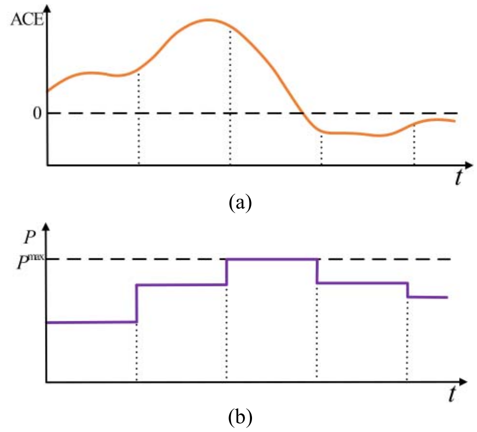


Fig. 2. Response of HVAC systems for frequency regulation. (a) ACE signal. (b) Fan power for performing frequency regulation.

setpoint. If not, the fan power should decrease so that supply air flow rate is reduced.

2) Performing Frequency Regulation: While HVAC systems participate in frequency regulation, the fan power will be regulated according to the area control error (ACE) that is used to indicate the generation-load mismatch of the power grid. As shown in Fig. 2, when ACE is above zero, the fan power will be increased, because there is more generation than load. If ACE is below zero, the fan power has to be reduced to make the ACE approach to zero.

However, while responding to ACE signals, the fan power decided by the control center can aggravate deviations of the indoor temperatures from the setpoints. This is because ensuring temperature setpoints and performing frequency regulation can lead to the control of the fan power in opposite directions, as shown in Figs. 1(b) and 2(b). Besides, different fan powers can result in different temperature deviations. Therefore, there must be a coordination among HVAC systems for fairly ensuring the comfort of occupants, while performing frequency regulation.

B. Optimal Problem Descriptions

Within the capacity of HVAC systems, the regulation task can be dispatched from the control center to HVAC systems. In order to fairly ensure the comfort of occupants, we consider the differences between indoor temperatures and the temperature setpoints as the objective of HVAC systems for performing frequency regulation. With the objective, the regulation task can be optimally distributed among HVAC systems with different thermal behaviors. As illustrated in Fig. 3, the fan power change can be different from occupant to occupant, even if the same temperature deviation is maintained. This is because the temperature setpoint is a nonlinear function of the fan power. For instance, we assume two occupants with two different setpoints illustrated by points a and b in Fig. 3. When the states, respectively, move from points a and b to points a' and b', the fan power changes (ΔP_1 and ΔP_2) are different. Therefore, the power change based on ACE signals have to be optimally distributed to HVAC systems to fairly ensure the comfort of occupants.

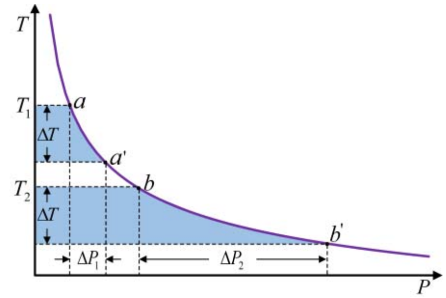


Fig. 3. Relationship between temperature setpoint and the fan power.

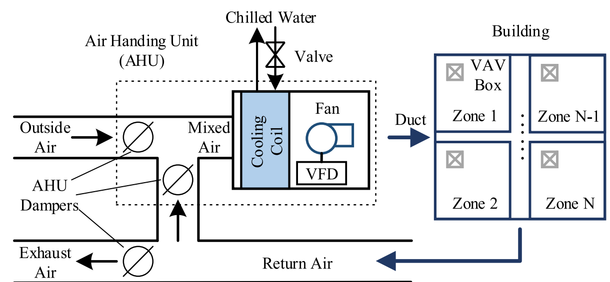


Fig. 4. Typical HVAC system in commercial buildings.

III. Model of Commercial Buildings HVAC Systems

A. Configuration of HVAC Systems

A typical commercial building HVAC system has multiple zones as illustrated in Fig. 4 [43]. The mixed air, which is consisted of the outside air and the part of return air, can be cooled down to ensure temperature setpoints, while passing through the cooling coil inside the air handing unit (AHU). The mixed air is delivered by the fan into multiple zones, which is called supply air. Therefore, the supply air flow rate can be regulated based on the fan power to ensure the comfort of occupants.

B. Commercial Buildings Thermal Model

The simplified thermal model of commercial buildings can be expressed in the following form [27], [28]:

$$C_{i,l}\frac{dT_{i,l}^t}{dt} = -\frac{1}{R_{i,l}}\left(T_{i,l}^t - T_{\text{out}}^t\right) + c^p m_{i,l}^t\left(T_i^s - T_{i,l}^t\right) + Q_{i,l}^t \quad (1)$$

where t is the time; $T_{i,l}^t$ is the indoor temperature of the lth zone in the ith building; T_{out}^t is the outside air temperature; T_i^s is the supply air temperature of the ith building; $C_{i,l}$, $R_{i,l}$, $Q_{i,l}^t$, and $m_{i,l}^t$ are the thermal capacitance, thermal resistance, external disturbance, and the supply air flow rate of the lth zone in the$i$th building, respectively; and c^p is the specific heat of the supply air.

C. Fan Power Consumption Model

In (1), the supply air flow rate, which determines the building indoor temperature, can be expressed with the

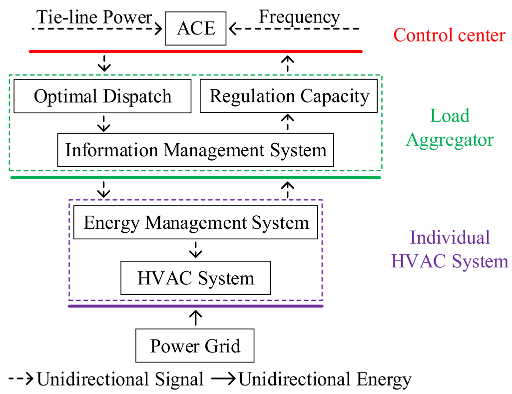


Fig. 5. Hierarchical framework of HVAC systems.

fan speed [27], [28]

$$m_i^t = c_1 v_i^t \tag{2}$$

where c_1 is a constant, and v_i^t is the fan speed, which is measured in the unit of percentage of the maximum speed.

The supply air flow rate of the lth zone in the ith building can be calculated as

$$m_{i,l}^t = \alpha_{i,l}^t m_i^t \tag{3}$$

where $\alpha_{i,l}^t$ is the distribution coefficient of supply air flow rate for the lth zone.

As shown in (2), the supply air flow rate is related to the fan speed. Therefore, the fan speed controlled by VFD and decided by the fan power has to be regulated to ensure building indoor temperature. The fan power can be expressed with the cube of the fan speed as [28]

$$P_i^t = c_2 \left(v_i^t\right)^3 \tag{4}$$

where c_2 is a constant, and P_i^t is the fan power.

IV. Hierarchical Framework and Regulation Capacity

A. Framework for Frequency Regulation

As illustrated in Fig. 5, the hierarchical control of HVAC systems consists of the control center, load aggregators, and HVAC systems.

In the control center, ACE is calculated according to the tie-line power and frequency deviations, and dispatched to the load aggregators within the capacity of HVAC systems for frequency regulation. In the load aggregator, the ACE from the control center is optimally allocated and sent to the energy management system in an individual HVAC system. At the same time, the capacity of HVAC systems is calculated and uploaded to the control center. In an individual HVAC system, the regulation task from the load aggregator is executed to regulate the fan power for regulation. Besides, the information of HVAC systems, such as real-time fan power, indoor temperature, and the maximum/ minimum fan power, is uploaded to the information management system in the load aggregator.

B. Capacity for Frequency Regulation

1) Capacity of HVAC Systems: The capacity for regulation at time t related to the fan power can be calculated as

$$\begin{cases} C_{i,t}^{\text{up}} = P_i^{t-1} - P_i^{\min} \\ C_{i,t}^{\text{down}} = P_i^{\max} - P_i^{t-1} \end{cases} \tag{5}$$

where $C_{i,t}^{\text{up}}$ and $C_{i,t}^{\text{down}}$ are the capacities of the ith HVAC system for regulation up and down, respectively; P_i^{t-1} is the ith fan power; and $P_i^{\max}$ and $P_i^{\min}$ are the allowable maximum/minimum fan power. Note that the HVAC system will not participate in regulation, while indoor temperature is not within allowable limits.

2) Capacity for Regulation in Load Aggregators: In a load aggregator, the capacity for frequency regulation at time t can be calculated in the following form:

$$\begin{cases} C_{j,t}^{\text{agg\_up}} = \sum_{i=1}^{n} C_{i,t}^{\text{up}} \\ C_{j,t}^{\text{agg\_down}} = \sum_{i=1}^{n} C_{i,t}^{\text{down}} \end{cases} \tag{6}$$

where n denotes the number of HVAC systems in a load aggregator, $C_{j,t}^{\text{agg\_up}}$ is the capacity of the jth load aggregator for regulation up, and $C_{j,t}^{\text{agg\_down}}$ is the capacity of the jth load aggregator for regulation down.

Therefore, in the control center, the capacity of load aggregators for frequency regulation can be calculated as

$$\begin{cases} C_t^{\text{up}} = \sum_{j=1}^{k} C_{j,t}^{\text{agg\_up}} \\ C_t^{\text{down}} = \sum_{j=1}^{k} C_{j,t}^{\text{agg\_down}} \end{cases} \tag{7}$$

where k is the number of load aggregators, C_t^{up} is the capacity of load aggregators for regulation up, and C_t^{down} is the capacity of load aggregators for regulation down.

V. Optimal Dispatch of HVAC Systems for Frequency Regulation

A. Dispatch in the Control Center

In the control center, ACE can be used as the signal for frequency regulation and dispatched from the control center to HVAC systems in the following form:

$$P_t^{\text{dis}} = \begin{cases} \text{ACE}_t, & \left(-C_t^{\text{up}} < \text{ACE}_t < C_t^{\text{down}}\right) \\ C_t^{\text{down}}, & \left(\text{ACE}_t \geq C_t^{\text{down}}\right) \\ -C_t^{\text{up}}, & \left(\text{ACE}_t \leq -C_t^{\text{up}}\right) \end{cases} \tag{8}$$

where P_t^{dis} is the regulation task from the control center, and ACE_t is the regulation signal at time t. If ACE_t is within the scope of the capacity, ACE_t is only undertaken by HVAC systems. Otherwise, ACE_t is undertaken by both HVAC systems and generators.

B. Distribution Among Load Aggregators

The regulation task from the control center is distributed in proportion among load aggregators according to the capacity as

$$P_{j,t}^{\text{dis}} = \begin{cases} P_t^{\text{dis}} \dfrac{C_{j,t}^{\text{agg\_down}}}{C_t^{\text{down}}}, & \left(P_t^{\text{dis}} > 0\right) \\ P_t^{\text{dis}} \dfrac{C_{j,t}^{\text{agg\_up}}}{C_t^{\text{up}}}, & \left(P_t^{\text{dis}} < 0\right) \end{cases} \tag{9}$$

where $P_{j,t}^{\text{dis}}$ is the regulation task of the jth load aggregator.

C. Optimal Regulation in Load Aggregator

In a load aggregator, the regulation task can be optimally distributed according to the capacity of HVAC systems in (6) with the objective function in (10) and the constraints in (11)–(18).

1) Objective Function: In a load aggregator, the objective is to minimize the differences between indoor temperatures and the temperature setpoints, as follows:

$$\min \ F_j^t\left(\Delta P_i^t, \alpha_{i,l}^t\right) = \sum_{i=1}^{n}\sum_{l=1}^{N}\left(T_{i,l}^t - T_{i,l}^{\text{set}}\right)^2 \tag{10}$$

where n is the number of HVAC systems, N is the number of zone in a building, ΔP_i^t is the ith fan power regulation, $\alpha_{i,l}^t$ is the distribution coefficient of the supply air flow rate of the lth zone in the ith building, and $T_{i,l}^t$ and $T_{i,l}^{\text{set}}$ are the indoor temperature and temperature setpoint of the lth zone in the ith building, respectively.

In general, the occupants expect the indoor temperature to be close to their setpoints for comfort. When the fan power is changed due to the performance of frequency regulation, indoor temperatures will deviate from the occupants' expected setpoints. If the coordination among HVAC systems is not considered, it would be difficult to keep all occupants comfort. In other words, some occupants' comforts are ensured, while some may feel uncomfortable due to the great deviation of indoor temperatures from the expected. In order to fairly ensure the comfort of occupants while implementing frequency regulation, we consider (10) as the objective. Note that in (10), the closer F_j^t is to zero, the better the thermal comfort of occupants is.

2) Equality Constraints: From (1), the indoor temperature can be expressed by

$$T_{i,l}^t - T_{i,l}^{t-1} = \frac{h}{C_{i,l}}\left(-\frac{1}{R_{i,l}}\left(T_{i,l}^t - T_{\text{out}}^t\right) + c^p m_{i,l}^t\left(T_{i,l}^s - T_{i,l}^t\right) + Q_{i,l}^t\right) \tag{11}$$

where h is the difference step.

From (2) and (4), the air flow rate can be expressed with the fan power as

$$m_{i,l}^t = c_1\alpha_{i,l}^t\sqrt[3]{\frac{P_i^t}{c_2}} \tag{12}$$

where

$$\sum_{l=1}^{N}\alpha_{i,l}^t = 1. \tag{13}$$

Therefore, the ith fan power P_i^t at time t can be calculated as

$$P_i^t = P_i^{t-1} + \Delta P_i^t. \tag{14}$$

As the regulation of a fan is determined by the control center, in a load aggregator, the regulation of all fans should be equal to the task from the control center. Therefore

$$\sum_{i=1}^{n}\Delta P_i^t = P_{j,t}^{\text{dis}}. \tag{15}$$

Algorithm 1 OD for Frequency Regulation

a. Initialization: Upload system parameters $C_{i,l}$, $R_{i,l}$, $T_{i,l}^s$, $T_{i,l}^{set}$, $P_i^{\max}$, $P_i^{\min}$.
b. Update state parameters ACE_t, $T_{i,l}^{t-1}$, P_i^{t-1}, $m_{i,l}^{t-1}$, T_{out}^{t-1}, $Q_{i,l}^{t-1}$.
c. Calculate the capacity $C_{i,t}^{\text{up}}$, $C_{i,t}^{\text{down}}$, $C_{j,t}^{\text{agg\_up}}$, $C_{j,t}^{\text{agg\_down}}$, according to (5) and (6), respectively;
d. Upload and sum the capacity in load aggregators (C_t^{up}, C_t^{down}) according to (7).
e. Perform the dispatch according to (8) in a load aggregator.
f. Solve the optimization model with the objective function in (10) and constraints in (11)-(18).
g. Regulate the fan power of HVAC systems following the optimal solution.
h. Time T+1, return to step b.

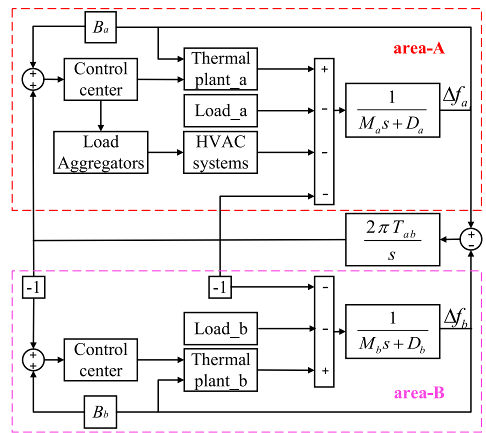


Fig. 6. Two-area interconnected power system.

3) Inequality Constraints: The regulation of a fan has to be constrained by its capacity

$$\begin{cases} -C_{i,t}^{\text{up}} \le \Delta P_i^t < 0, & \left(P_{j,t}^{\text{dis}} < 0\right) \\ 0 < \Delta P_i^t \le C_{i,t}^{\text{down}}, & \left(P_{j,t}^{\text{dis}} > 0\right). \end{cases} \tag{16}$$

Besides, the limits of a fan power must be considered

$$P_i^{\min} \le P_i^t \le P_i^{\max}. \tag{17}$$

The distribution coefficient of the supply air flow rate of the lth zone in the ith building is limited by

$$0 \le \alpha_{i,l}^t \le 1. \tag{18}$$

D. Optimal Algorithm for Regulation in Load Aggregator

The closed-loop algorithm used to perform the optimal dispatch (OD) is explained in Algorithm 1.

VI. Simulation and Discussion

A. Simulation Systems

The simulation on a two-area interconnected power system illustrated in Fig. 6 is implemented under the MATLAB

TABLE I
PARAMETERS OF POWER GRID

Parameters\area	Area-A	Area-B
Maximum load capacity (MW)	20000	10000
Proportional and integral gains	1, 0.01	1, 0.01
Time constant for LFC(s)	4	4
Frequency bias factor B (pu/Hz)	0.15	0.075
Inertia constant M (pu·s)	0.32	0.16
Load damping coefficient D (pu/Hz)	0.04	0.02
Synchronizing torque coefficient T_{ab} (pu/Hz)	0.04	
Dead band of primary frequency control (Hz)	0.033	
Time constant for frequency detection (s)	0.1	0.1
Communication delay (s)	1	1
Dead band of area control error (MW)	20	20
Ramp speed (MW/min)	400	200

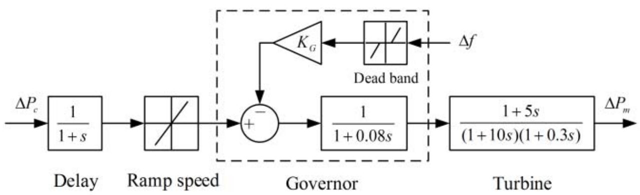


Fig. 7. Thermal power plant for frequency regulation.

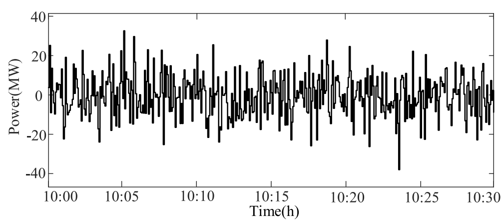


Fig. 8. Load fluctuation in a time series with 4-s intervals.

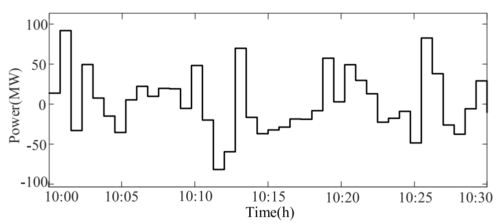


Fig. 9. Load fluctuation in a time series with 45-s intervals.

TABLE II
BUILDING AND SIMULATION PARAMETERS

Parameters	Value
The number of load aggregators, k	300
The number of HVAC systems in a load aggregator, n	100
The number of zones in a HVAC system, N	4
Building thermal capacitance, $C_{i,l}(J/^{\circ}C)$	$C_{i,l} \sim N(1\times10^{7}, 1\times10^{6})$
Wall thermal resistance, $R_{i,l}(^{\circ}C/W)$	$R_{i,l} \sim N(4\times10^{-3}, 5\times10^{-4})$
Occupant temperature setpoint, $T^{set}(^{\circ}C)$	$T_{i,l}^{set} \sim N(21.5, 0.5)$
Supply air temperature, $T^{s}(^{\circ}C)$	16
Specific heat of air, $c^{p}(J/g/^{\circ}C)$	1006
Allowable maximum/minimum temperature deviation $(^{\circ}C)$	0.5
Allowable maximum/minimum fan power P^{max}/P^{min} (kW)	33/0
Constant, $c_1(kg/s)$	0.0964
Constant, $c_2(kW)$	3.3×10^{-5}

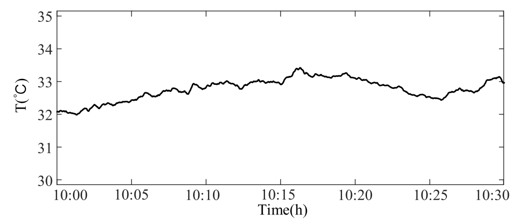


Fig. 10. Outside air temperature.

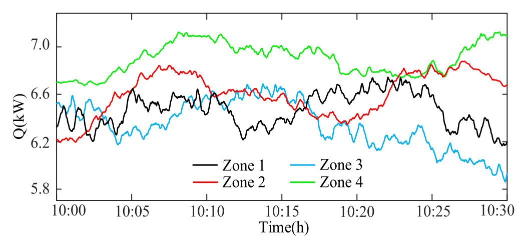


Fig. 11. External disturbances of four zones in an HVAC system.

simulink environment, where an equivalent power plant with parameters listed in Table I is considered [6], as illustrated in Fig. 7. Note that the equivalent model illustrated in Fig. 7 can be aggregated by multiple power plants [44]. The load fluctuations, which follow normal distributions with zero mean, are illustrated in Figs. 8 and 9. The outside air temperature is illustrated in Fig. 10. We assume that HVAC systems are integrated into area A.

B. Parameters of HVAC Systems

We assumed that the control center manages 300 load aggregators, where each aggregator controls 100 HVAC systems, and each HVAC system includes four zones with the random parameters and external disturbances shown in Table II and Fig. 11, respectively.

C. Simulations for Multiple Zones

In order to show the effectiveness of the proposed OD, we compare it with another strategy with proportional dispatch (PD) in a load aggregator, as follows:

$$\Delta P_i^t = \begin{cases} P_{j,t}^{\text{dis}} \frac{C_{i,t}^{\text{up}}}{C_{j,t}^{\text{agg\_up}}}, & \left(P_{j,t}^{\text{dis}} < 0, T_{i,l}^{t} < T_{i,t}^{\max}\right) \\ P_{j,t}^{\text{dis}} \frac{C_{i,t}^{\text{down}}}{C_{j,t}^{\text{agg\_down}}}, & \left(P_{j,t}^{\text{dis}} > 0, T_{i,l}^{t} > T_{i,t}^{\min}\right) \\ 0, & \text{else.} \end{cases} \tag{19}$$

Note that for PD, the supply air flow rate is distributed in proportion to an HVAC system [45] and assigned to four zones by the PI strategy [37]. The case that HVAC systems do not participate in frequency regulation is called as “without dispatch (WD).”

1) Effect on System Frequency: As illustrated in Fig. 12, the participation of HVAC systems in frequency regulation can suppress frequency fluctuation due to the fast response and

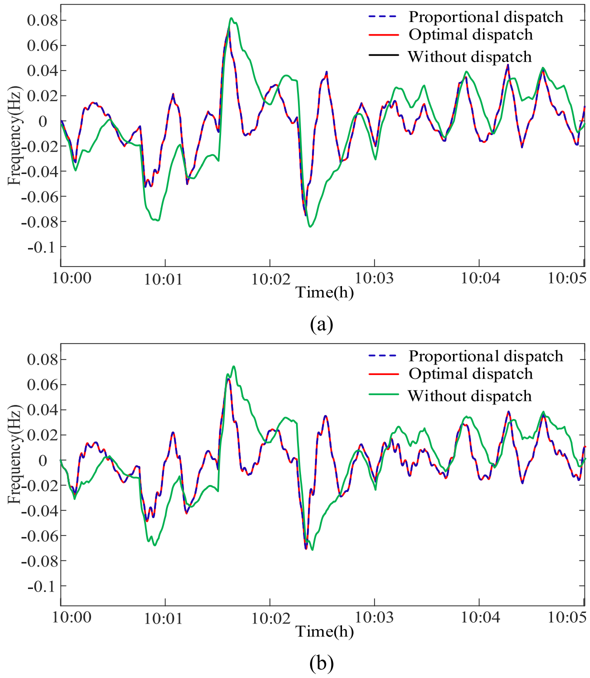


Fig. 12. Frequency responses of the two-area interconnected system. (a) Frequency response in area-A. (b) Frequency response in area-B.

TABLE III
COMPARISONS ON FREQUENCY DEVIATION AND ACE IN AREA-A

Parameters	WD	PD	OD
The RMS of frequency deviation (Hz)	0.0252	0.0165	0.0165
The RMS of area control error (MW)	51.47	30.18	30.18

TABLE IV
COMPARISONS ON MV OF ATD IN A LOAD AGGREGATOR

Parameters	WD	PD	OD
The MV of Zone 1's ATD (°C)	0.1053	0.1110	0.0763
The MV of Zone 2's ATD (°C)	0.0899	0.0970	0.0722
The MV of Zone 3's ATD (°C)	0.0927	0.1008	0.0741
The MV of Zone 4's ATD (°C)	0.0918	0.1024	0.0725

regulation, compared with WD. From Table III, OD has the same root-mean-square (RMS) values of frequency deviation and ACE as PD. Compared with not considering HVAC, PD and OD achieve the better quality of frequency and ACE.

2) Influence on Occupants' Comfort: For clarity, we take a load aggregator with 100 HVAC systems as an example. Note that each HVAC system includes four zones as illustrated in Table II. From Table IV, the mean values (MV) of the absolute temperature deviations (ATD) are larger for PD than for WD. This is because for PD, the occupants' comfort is neglected while participating in frequency regulation. However, compared with WD and PD, the MV of ATD is smallest for OD. Therefore, OD has the advantage over WD and PD in ensuring occupants' comfort from a load aggregator.

As shown in Fig. 13, PD cannot fairly ensure the occupants' comfort due to different indoor temperature deviations. For instance, in zone 1 illustrated in Fig. 13(a), indoor temperature deviations of some occupants are small, while the others have larger indoor temperature deviations. As shown in Fig. 14, indoor temperature deviations of all occupants are close to zero. Therefore, OD can fairly ensure the occupants' comfort. This is because both fairly ensuring occupants' comfort and performing frequency regulation are considered by OD simultaneously.

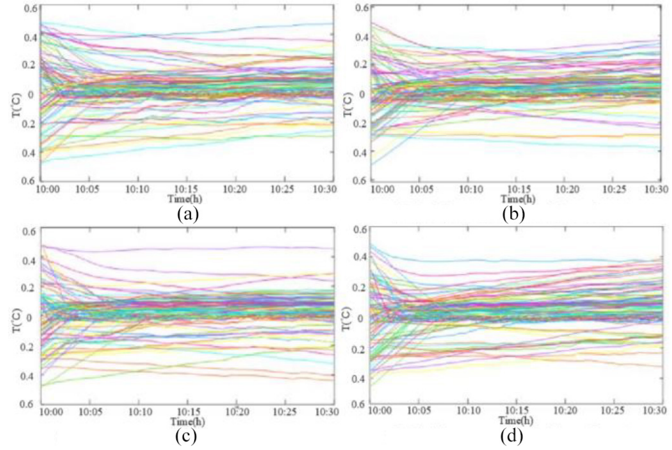


Fig. 13. Indoor temperature deviations in a load aggregator with PD. (a) The indoor temperature deviationsof zone 1. (b) The indoor temperature deviationsof zone 2. (c) The indoor temperature deviationsof zone 3. (d) The indoor temperature deviationsof zone 4

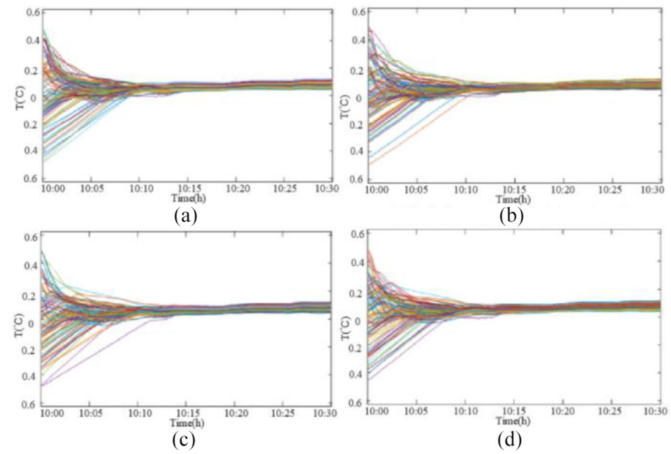


Fig. 14. Indoor temperature deviations in a load aggregator with OD. (a) The indoor temperature deviationsof zone 1. (b) The indoor temperature deviationsof zone 2. (c) The indoor temperature deviationsof zone 3. (d) The indoor temperature deviationsof zone 4

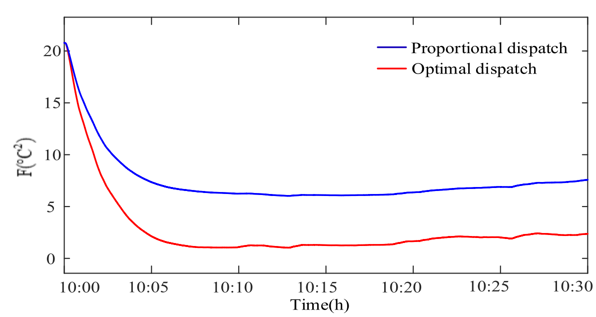


Fig. 15. Change of the objective function in (10) over time.

We can also evaluate the occupants' comfort from the point of view of the objective function, as shown in Fig. 15. Compared with PD, the objective function, i.e., the temperature deviation changes over time, decreases faster for OD. This

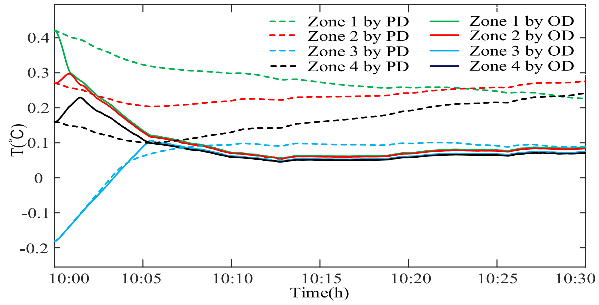


Fig. 16. Indoor temperature deviations of four zones in an HVAC system.

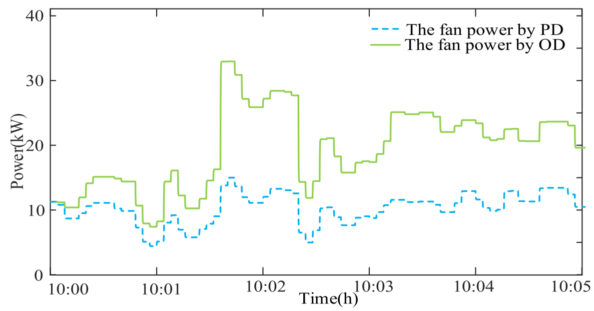


Fig. 17. Fan power of an HVAC system.

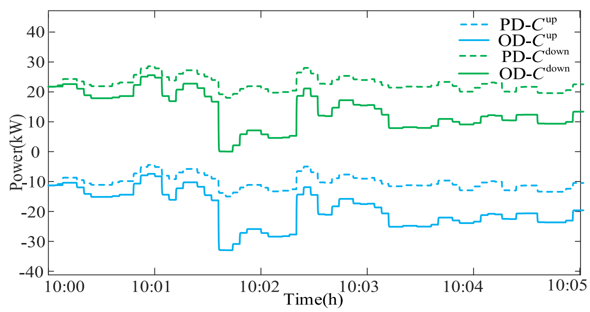


Fig. 18. Capacity of an HVAC system for frequency regulation.

is because PD does not consider temperature constraints, while OD fairly ensures the occupants' comfort. To clearly demonstrate the advantage of OD over PD in ensuring the occupants' comfort, we randomly choose an HVAC system, as illustrated in Fig. 16. According to PD, zones 1 and 3 approach to zero, while the others deviate from zero. However, OD can maintain indoor temperature deviations of four zones close to zero and achieve the better comfort of occupants in comparison to PD.

Compared with PD, for OD, the fan power must be greatly changed to ensure the occupants' comfort, as illustrated in Fig. 17. Therefore, the capacity for frequency regulation also varies more greatly for OD than for PD, as shown in Fig. 18.

3) Influence on Traditional Generator Regulation: Because of the participation of HVAC systems in frequency regulation, the regulation from traditional generating units is greatly reduced, as shown in Fig. 19. This means that traditional generators are released from frequent regulation.

D. Simulations for Single Zone

We here consider an HVAC system with a single zone for further demonstration. Besides the parameters illustrated

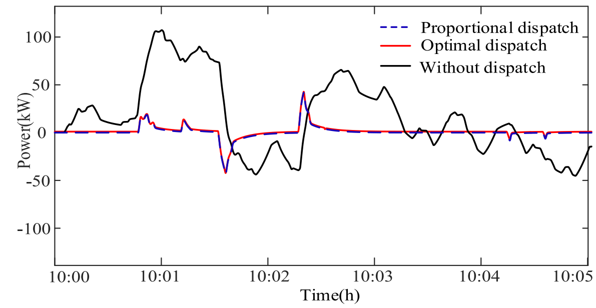


Fig. 19. Influence on traditional generator regulation.

TABLE V
PARAMETERS FOR HVAC SYSTEM WITH SINGLE ZONE

Parameters	Value
Building thermal capacitance, $C(J/{}^{\circ}C)$	3.4×10^{4}
Wall thermal resistance of building, $R({}^{\circ}C/W)$	0.0013
Supply air temperature, $T^{s}({}^{\circ}C)$	16
Occupant temperature setpoint, $T^{set}({}^{\circ}C)$	21.5

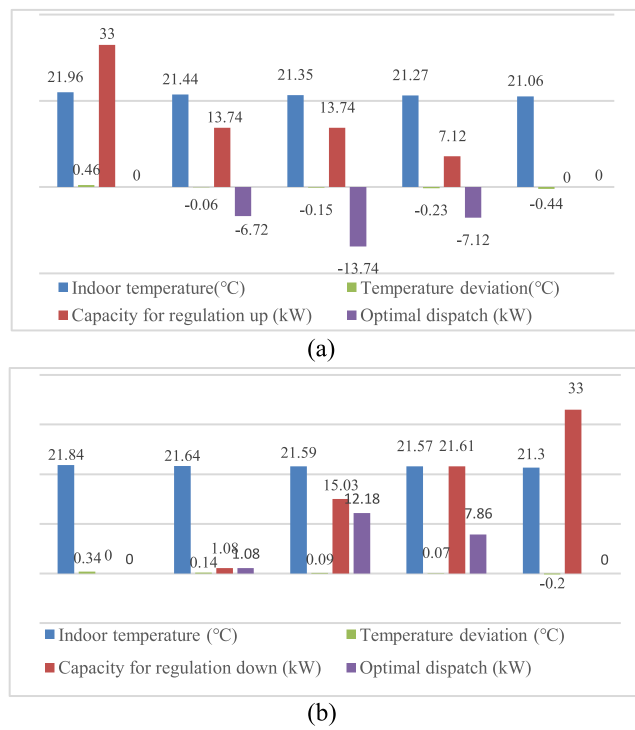
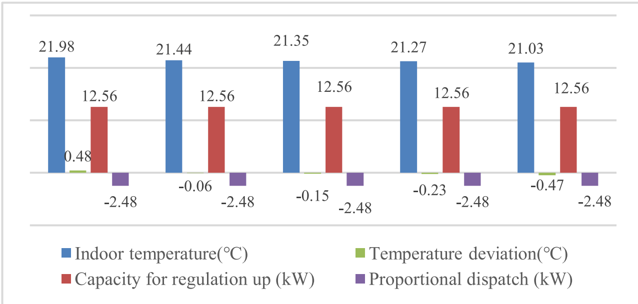


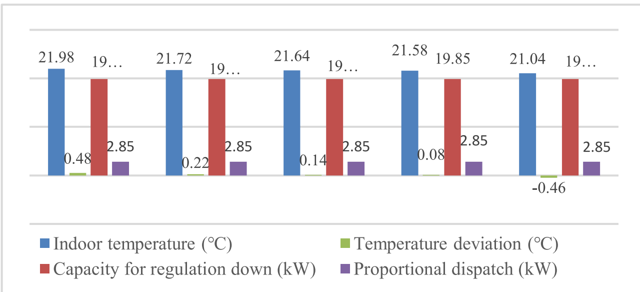
Fig. 20. OD of HVAC systems. (a) Regulation up. (b) Regulation down.

in Tables I and II, the other parameters are shown in Table V.

1) Optimal Dispatch of HVAC Systems for Regulation: For clarity, we randomly take five HVAC systems as an example. As illustrated in Fig. 20, the dispatch for regulation is optimally distributed among HVAC systems. The bigger the negative temperature deviation is, the more the fan power should be reduced, which means that for regulation up shown in Fig. 20(a), the more the dispatch for regulation up is. Note that in Fig. 20(a), the dispatches of two HVAC systems are limited by the capacity for regulation up. Similarly, the bigger the positive temperature deviation is, the more the fan



(a)



(b)

Fig. 21. PD of HVAC systems. (a) Regulation up. (b) Regulation down.

TABLE VI
COMPARISONS ON FREQUENCY DEVIATION, ACE, AND ATD

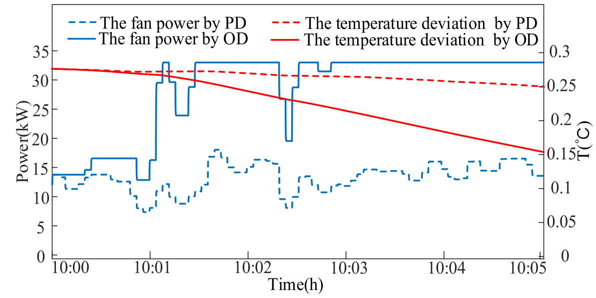
Parameters	WD	PD	OD
The RMS of frequency deviation (Hz)	0.0252	0.0165	0.0165
The RMS of ACE (MW)	51.47	30.18	30.18
The MV of ATD (°C)	0.198	0.198	0.097

power should be increased, and the more the dispatch for regulation down is as illustrated in Fig. 20(b). In this way, indoor temperatures can be fairly controlled to approach to the occupants' setpoints by the proposed OD strategy.

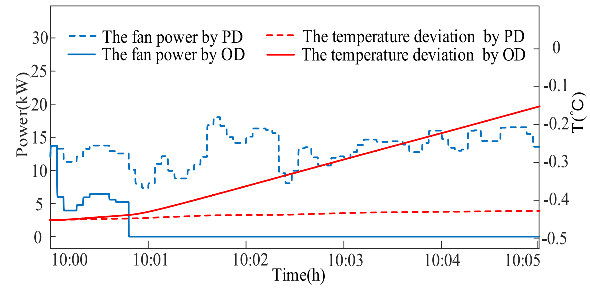
However, for the case that PD is used, the dispatch from the control center is distributed in proportion according to the capacity of HVAC systems. If the capacity is the same for all of HVAC systems, the dispatch will be kept as the same for regulation. As shown in Fig. 21, this dispatch for regulation is the same, although the temperature deviations are different from HVAC to HVAC. For instance, as shown in Fig. 21(a), the same dispatch task for regulation up is assigned to each HVAC system, as the capacity for regulation up is the same for five HVAC systems. Therefore, compared with PD, the proposed OD can show the better advantage in fairly ensuring different occupants' comfort.

2) Effect on Frequency and Indoor Temperature: As shown in Table VI, the RMSs of frequency deviation and ACE in area-A are the same for PD and OD. Compared with WD, the participation of HVAC systems in frequency regulation can improve the quality of frequency and ACE due to the fast response and regulation. From the point of view of a load aggregator, OD has the advantage over WD and PD in ensuring occupants' comfort due to the smaller MV of ATD.

3) Energy Consumption Analysis: We randomly choose two HVAC systems to analyze the energy consumptions of PD and OD for positive and negative temperature deviation scenarios.



(a)



(b)

Fig. 22. Fan power and temperature deviation of HVAC systems. (a) Positive temperature deviations. (b) Negative temperature deviations.

As shown in Fig. 22(a), compared to that PD does not consider the occupant' comfort, OD must increase the fan power to decrease indoor temperature. In this condition, OD has more energy consumption than PD does. When the indoor temperature deviation is below zero, OD reduces the fan power to elevate indoor temperature, as shown in Fig. 22(b). However, PD remains keeping the large fan power even if indoor temperature is low and leads to excessive energy consumption.

VII. CONCLUSION

In this article, an OD strategy is proposed to allow commercial HVAC systems to engage in frequency regulation, while maintaining equal comfort for the occupants. The objective of the proposed optimal strategy is to minimize the indoor temperature deviations of occupants with the constraints, such as power limits, the capacity for regulation, and regulation task. The simulation results on a two-area interconnected power system have demonstrated better performances of the proposed optimal strategy in improving the quality of frequency, reducing traditional generator regulation, and fairly ensuring the occupants' comfort, compared with the PD.

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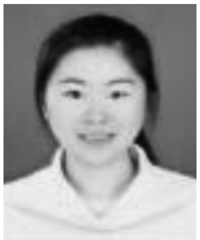

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