Using flexible HVAC power consumption of commercial buildings for regulation service to the grid

Technical Report

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***Abstract*— We study providing ancillary service to the power grid by using the flexibility of HVAC (Heating, Ventilation, Air Conditioning) systems in commercial buildings. The building HVAC system's thermal and power consumption models are constructed and key model parameters are estimated from measure data from Pugh Hall of University of Florida. We develop a method for optimal output tracking of a time-varying regulation signal by manipulating the fan speed of the HVAC system. Due to the large thermal capacity of commercial building and its flexibility of HVAC systems, we show that it is highly feasible for commercial buildings to provide ancillary service to the grid by dynamically adjust its fan power consumption without noticeably impacting the building inside environment and the comfort of occupants. In addition, numerical experiments are conducted to show that for a typical HVAC system, it is able to provide about 10% of its fan power capacity for frequency regulation, with the building inside temperature deviation less than $0.2\ ^{\circ}C$.**

I. Introduction

Due to growing environmental concerns as well as economic and political requirements, the envisioned future power grid will increasingly rely on renewable sources such as wind and solar energies. For example, President Obama's "New Energy for America" calls for renewable energy to supply 10% of the nation's electricity by 2012, rising to 25% by 2025 [1]. Figure 1 shows the historical and projected curve of U.S. renewable energy consumption [2]. We see from the figure that in the next 20 years, the renewable resources are going to increased 60% from 9 Quads in 2012 to 14.4 Quads in 2032.

However, these variable generations are volatile, intermittent and uncontrollable. With deep penetration of renewable resources, it is challenging to maintain the reliability of the current electricity grid infrastructure, since the existing regulation services do not have enough capacity to compensate the volatility of renewables. Moreover, most current ancillary service comes from fossil fuels, which produces green-house gas. To find other clean sources of flexibility, responsive load remains the largest underutilized reliability resources available to the North American Power system today [3]. Buildings in the United States consume 70% of its electricity [4], [5]. Furthermore, the building HVAC (Heating, Ventilating, and Air Conditioning) systems account for 40% of the electricity consumption.

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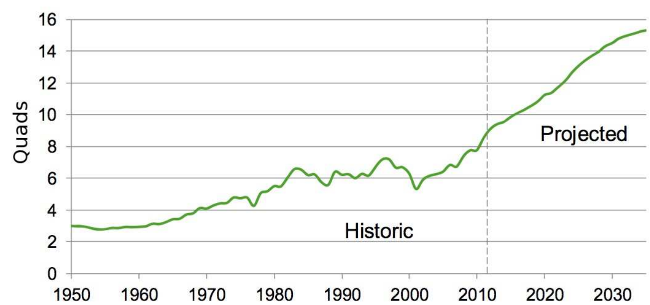

Fig. 1. History and projection of U.S. renewable energy consumption.

The idea of using buildings to provides ancillary services to improve grid reliability is novel. Callaway and Hiskens *et. al.* proposed to using aggregated thermostatically controlled loads such as refrigerators, air conditioner and water heaters in the residential buildings for demand response [6]–[8]. Besides buildings, some aluminum manufacturing companies and agriculture farms provides demand responses by ramping up and down their energy use in response to the uncertainty in the grid [3], [9].

In this paper, we focus on commercial buildings. The reasons for choosing commercial buildings have several manifolds. First, Building Energy Management Systems (BEMS) serve about one-third of the overall commercial buildings in the United States [10]. These systems can receive the regulation signal from the ISO without any requirements on the smart meter infrastructures that are not currently mature. Second, the BEMS already integrates the control system of the building which can easily manipulate the control variables such as fan speed, valve position etc. Third, the commercial buildings are equipped with fast-responding VFDs (Variable Frequency Drive). These power electronic devices can be controlled by BEMS. Fourth, some studies have shown that the owners of residential buildings are reluctant to participate in the demand response because of unguaranteed cost reduction and they feel uncomfortable when the agents are tampering with their appliances and interrupt their lifestyles []. Last but not the least, commercial

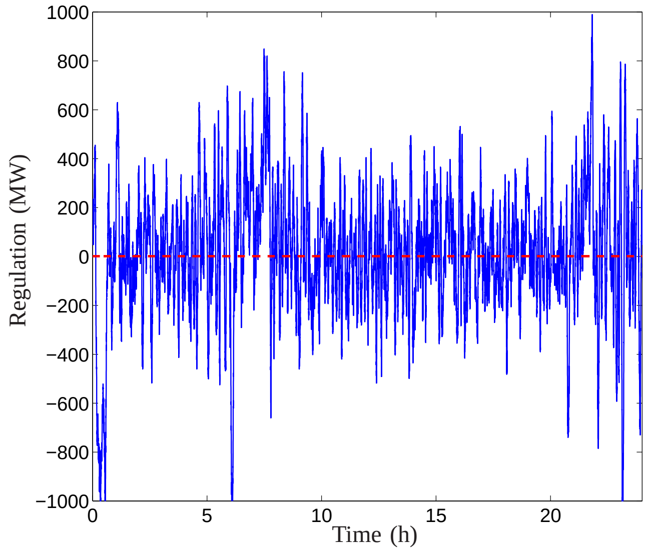

Fig. 2. Regulation signal of PJM for a typical day.

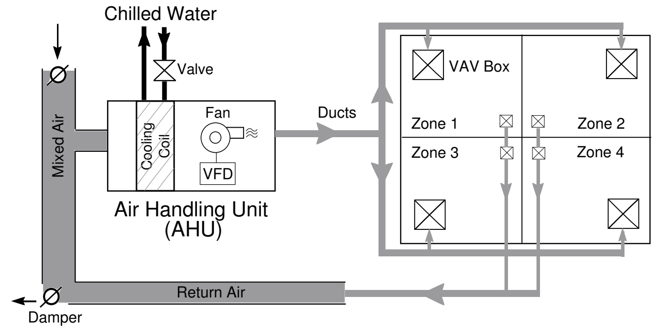

Fig. 3. The HVAC system of a four-zone building

buildings have large thermal capacitances, small variations in mass flow rate (thus fan speed) do not change the indoor environment.

Ancillary services provide the resources the ISO requires to reliably maintain power balance between generation and load for all kinds of time-scales. One of the ancillary service is frequency regulation, which aims to compensates for minute-to-minute fluctuations in total system load and uncontrolled generation and maintain a constant system frequency. In this work, we focus frequency regulation by using the flexibility of commercial building HVAC systems. Figure 2 depicts the regulation signal of PJM for a typical day. We see from the figure that regulation is a *zero-energy* service [11]. The regulation signal fluctuates around zero and has an mean of about zero. It was shown that the regulation deployed in ERCOT remains in one direction for 20 minutes on average, with a peak of 94 minutes in one direction [12]. This feature of regulation signal is suitable for buildings, who have large thermal capacitance. By dynamically ramping up and down the supply fan speed (thus power consumption) of the HVAC system, it is feasible to tracking a regulation signal without much change to the building inside environment.

In this paper, we first construct a thermal model for the commercial building and a power consumption model for its HVAC system. The key parameters of these models are estimated based on the measurement date from Pugh Hall of University of Florida. We developed a fast control method for tracking of a time-varying regulation signal by manipulating the supply fan speed of the air handling unit in the HVAC system. Numerical experiments show that for a typical HVAC system, it is feasible to provide 10% of the total fan power to the grid without noticeably impacting the building inside environment and the occupants' comfort.

The rest of this paper is organized as follows. Section II presents thermal and power consumption model of the commercial building HVAC system. The linearized model of thermal and power consumption dynamics is derived in Section III and its model parameters are estimated in Section V. Numerical experiments are conducted in Section VI. The paper ends with conclusions and future work in Section VII.

II. Building HVAC thermal and power consumption models

A. Configuration of HVAC system in commercial buildings

A typical multi-zone HVAC system in a modern commercial building is depicted in Figure 3. Its main components consist of AHU (air handling unit), supply fan, and VAV (variable air volume) boxes. The AHU recirculate the return air from each zone and mixes it with fresh outside air. The ratio of the fresh outside air to the return air is controlled by dampers. The mixed air is drawn through the cooling coil in the AHU by centrifugal fans, and it is cooled down to a constant setpoint temperature (usually $13°C$) by adjusting the valve position of the supplied chilled water. The leaving (cooled) air is then distributed to each zone through ducts. At the entrance of each zone, a VAV box determines the air flow into the zone by controlling the damper position. In addition, the VAV box also controls the supply air temperature into each zone by appropriate reheating.

B. Thermal model of commercial buildings

We consider the following physics-based thermal model of a commercial building

$$C\frac{dT}{dt} = -\frac{1}{R_w}(T - T_{oa}) + c_p\dot{m}(T_{la} - T) + Q_x, \qquad (1)$$

where the parameters and their units are described in Table I. The first term $-\frac{1}{R_w}(T - T_{oa})$ on the RHS of (1) represents the heat loss due to heat conduction through the walls, and the second term $c_p\dot{m}(T_{la} - T)$ denotes the heat gain from the air conditioner. The last term Q_x on the RHS of (1) is the heat gain from reheating, solar, occupants, lights, etc.

TABLE I

DESCRIPTION OF PARAMETERS OF A BUILDING THERMAL DYNAMICS

Parameter	Description	Unit
T	building inside temperature	$^\circ C$
C	thermal capacitance of building	$J/ ^\circ C$
R_w	wall thermal resistance of building	$^\circ C/W$
T_{oa}	outside air temperature	$^\circ C$
T_{la}	leaving air temperature	$^\circ C$
c_p	specific heat of air	$J/g/^\circ C$
$\dot{m}$	supply air flow rate	kg/s
Q_x	heat gain from reheating, solar, etc.	W

The overall air flow rate $\dot{m}$ supplied into the building is determined by the fan speed V^{fan}. We assume the air flow rate is in a linear relationship to the fan speed,

$$\dot{m}(t) = c_1 V^{fan}(t), \tag{2}$$

where c_1 is a constant, and V^{fan} is the rated fan speed in percentage. For example, 100% represents the fan is running at full speed and 50% means it is running at half speed.

In practice, the fan speed is controlled by a variable frequency drive (VFD), see Figure 3. The VFD is a fast-responding and programmable power electronic device that changes the AC motor speed by varying motor input frequency and voltage. Besides the function of adjusting motor speed, the VFD also ramps the fan gradually up and down to prevent sudden jump which can trip on an over current alarm or excessive stress on the gear boxes. Because of this ramping feature of VFD, we assume the transfer function from the control command to the fan speed is of first-order

$$\tau \frac{dV^{fan}(t)}{dt} + V^{fan}(t) = u^{fan}(t), \tag{3}$$

where τ is the time-constant, and $u^{fan}(t)$ is the fan control command (in percentage) sent from the building energy management system (BEMS).

C. Power consumption model of HVAC system in commercial buildings

The major electricity power consumer of a building HVAC system is the supply fan. The supply fan is the heart of the HVAC system. Like a heart that pumps blood through a human body, the fan distributes the leaving (cooled) air throughout the buildings over ducts. The power consumption of a fan is proportional to the cubic of its speed

$$P^{fan} = c_2 (V^{fan})^3, \tag{4}$$

where c_2 is a constant, and V^{fan} is the rated fan speed in percentage.

Remark 1: In this paper, we assume the electricity power consumptions of 1) the furnace that supplies the hot water to the VAV boxes (for reheating), 2) the chiller and cooling tower that provide the chilled water to the cooling coil are independent of the fan power. In many HVAC systems, the furnaces consume natural gas instead of electricity. In addition, for some HVAC systems, the chilled water is supplied from a water storage tank. The purpose of using water storage is to shift the peak colling demand in respond to dynamics utility prices, ambient temperature, etc. [13]. For those HVAC systems that the power consumptions of the furnaces and chillers/cooling towers are dependent of the fan power, dynamics models that relate the fan action to the furnace and chiller powers can be identified from sensor data. In that case, those HVAC systems can provide more capacity of ancillary service to the grid. □

III. LINEARIZED MODELS OF BUILDING HVAC THERMAL AND POWER DYNAMICS

During occupied operation mode (08:00 am - 22:00 pm), the building HVAC system is operating near a steady-state status. The building inside temperature is controlled to a fixed setpoint (about $22.4^\circ C$, see Figure 4), and the supply fan is also running at relative constant speeds. For control purpose, we linearize the building thermal dynamics (1)-(3) and the power consumption dynamics (4) at its operation equilibrium. At steady-state, the thermal model (1) satisfies

$$0 = -\frac{1}{R}(T^* - T_{out}) + c_p \dot{m}^* (T_{la} - T^*) + Q_x, \tag{5}$$

where T^* and $\dot{m}^*$ are the steady-state temperature and supply air flow rate. In addition, we have assumed the outside air temperature T_{oa} and the heat gain Q_x are constant. We now consider the following perturbation profiles

$$T = T^* + \tilde{T}, \quad \dot{m} = \dot{m}^* + \tilde{\dot{m}}, \tag{6}$$

where $\tilde{T}$ and $\tilde{\dot{m}}$ are respectively the deviations of the building inside temperature and supply air flow rate from their nominal values T^* and $\dot{m}^*$. Substituting (6) into (1), and using (5), we obtain the linearized model of building thermal dynamics:

$$\frac{d\tilde{T}}{dt} = -\frac{1 + c_p R_w \dot{m}^*}{C R_w} \tilde{T} + \frac{c_p (T_{la} - T^*)}{C} \tilde{\dot{m}}. \tag{7}$$

Based on (2) and (3), the linearized dynamics of the supply air flow is given by

$$\frac{d\tilde{\dot{m}}}{dt} = -\frac{1}{\tau}\tilde{\dot{m}} + c_1 \tilde{u}^{fan}, \tag{8}$$

where $\tilde{u}^{fan}$ is the deviation of the actual fan speed control command from the nominal fan speed control command. In addition, we let

$$P^{fan} = P^{fan*} + \tilde{P}^{fan}, \quad V^{fan} = V^{fan*} + \tilde{V}^{fan},$$

where P^{fan*} is the nominal fan power consumption and $\tilde{P}^{fan}$ is the power consumption deviation from the nominal value. The other variable can be interpreted in a similar way. Substituting the above equations into (4) as well as using (3), we obtain the following linearized model for fan power consumption

$$\frac{d\tilde{P}^{fan}}{dt} = -\frac{1}{\tau}\tilde{P}^{fan} + \frac{3c_2 (V^{fan*})^2}{\tau} \tilde{u}^{fan}. \tag{9}$$

Combining the above equations (7)-(9), we represent the linearized system in the state-space form

$$\dot{x} = Ax + Bu, \qquad y = Cx, \tag{10}$$

where the state, input and output are defined as $x := [\tilde{T}, \tilde{\dot{m}}, \tilde{P}^{fan}]^T$, $u := \tilde{u}^{fan}$, $y := \tilde{P}^{fan}$, and the state matrix, input matrix and output matrix are respectively given by

$$
\begin{aligned}
A &= \begin{bmatrix} -\frac{1+c_p R_w \dot{m}^*}{C R_w} & \frac{c_p (T_{la} - T^*)}{C} & \\ & -\frac{1}{\tau} & \\ & & -\frac{1}{\tau} \end{bmatrix}, \\
B &= \begin{bmatrix} 0 \\ \frac{c_1}{\tau} \\ \frac{3 c_2 (V^{fan*})^2}{\tau} \end{bmatrix}, \\
C &= \begin{bmatrix} 0, & 0, & 1 \end{bmatrix}.
\end{aligned}
$$

Remark 2: In practice, although the outside air temperature T_{oa} and the heat gain from occupants, solar, etc. are time-varying, these changes are compensated by the reheat in the VAV boxes to keep the overall heat gain Q constant in the occupied operation. In addition, the rate of change of these parameters are very slow when compared to the dynamics of thermal and power consumption models. Hence, in real applications, the above linearization process can be repeated every few minutes to mitigate these time-varying effects. □

IV. Optimal Output Tracking of Regulation Signal

With the linearized model given in (10), the control objective is to output track of a time-varying regulation signal by manipulating the supply fan speed of the HVAC system. We pose the following optimal control problem

$$
\min_u J(x_0) = \min_u \int_0^{T_r} Q(y(t) - r(t))^2 + R u(t)^2 \, dt,
$$

where $r(t)$ is the regulation signal provided by the independent system operator (ISO), and T_r is the regulation time period. The parameters $Q, R \in \mathbb{R}$ are positive weights. The first term of the integrand in the cost function $Q(y(t) - r(t))^2$ penalizes the tracking error, and the second term $Ru(t)^2$ penalizes violation of linearization.

From standard optimal control theory [14], the optimal control solution is given by the following form

$$
u(t) = -R^{-1} B^T (P(t) x(t) + b(t)),
$$

where $P(t)$ solve the following finite-time differential Riccati equation

$$
\begin{aligned}
\frac{dP(t)}{dt} =& - P(t)A - A^T P(t) + P(t) B R^{-1} B^T P(t) \\
& + C^T Q C, \\
P(T_r) =& 0,
\end{aligned}
$$

and $b(t)$ satisfies the following differential equation

$$
\begin{aligned}
\frac{db(t)}{dt} &= -(A - BR^{-1} B^T P(t))^T b(t) - C^T Q r(t), \\
b(T_r) &= 0.
\end{aligned}
$$

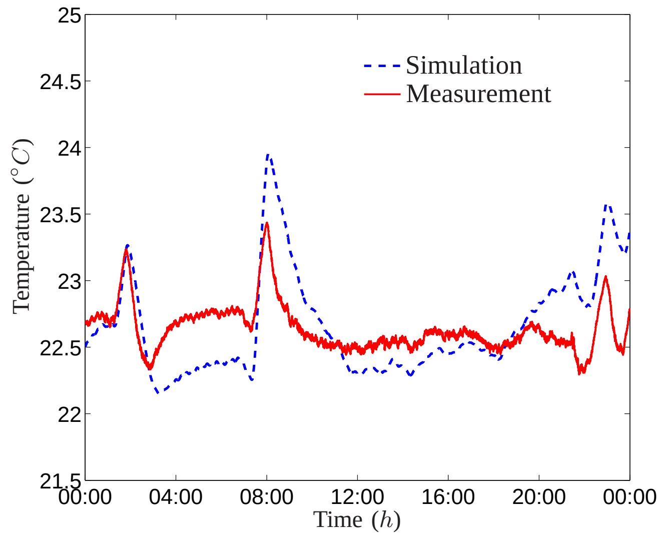

Fig. 4. Comparison of building inside temperature between simulation and measurement.

V. Parameters Estimation

In this section, we aim to identify the constant parameters (such as thermal capacitance C) given in the previous sections from measurement data of Pugh Hall in University of Florida. The Pugh Hall, a LEED (Leadership in Energy and Environmental Design) certified building, was constructed in 2008 and has an area of $45,690$ square feet. There is three AHUs in the building. The parameters in this paper are identified from the measure data for AHU1.

Figure 4 depicts the comparison between the measured temperature and the simulated temperature using model (1). The constant parameters selected are $C = 7 \times 10^5 \ J/^\circ C, R_w = 5 \times 10^{-3} \ ^\circ C/W, T_{la} = 13 \ ^\circ C, c_p = 1006 \ J/g/^\circ C$. The outside air temperature T_{oa} is obtained from historical data [15]. In simulation, the external heat gain are divided into two parts $Q_x = Q_r + Q_o$, where reheating part Q_r is measured from sensor, and for simplicity, the other heat gain Q_o is assume to be a constant $Q_o = 2.3 \times 10^4 \ W$.

The constant c_1 in model (2) is estimated as $c_1 = 0.0964 \ kg/s$. Figure 5 depicts the comparison of supply air flow rate between the measurement and the simulation using model (2).

The constant c_2 in model (4) is estimated as $c_2 = 3.3 \times 10^{-5} \ kW$. Figure 6 depicts the comparison of supply fan power between the measurement and the simulation using model (4).

In addition, the time constant τ given in (3) depends on the setting of VFD, this parameter is programmable. In this paper, its value is picked as $\tau = 0.1 \ s$.

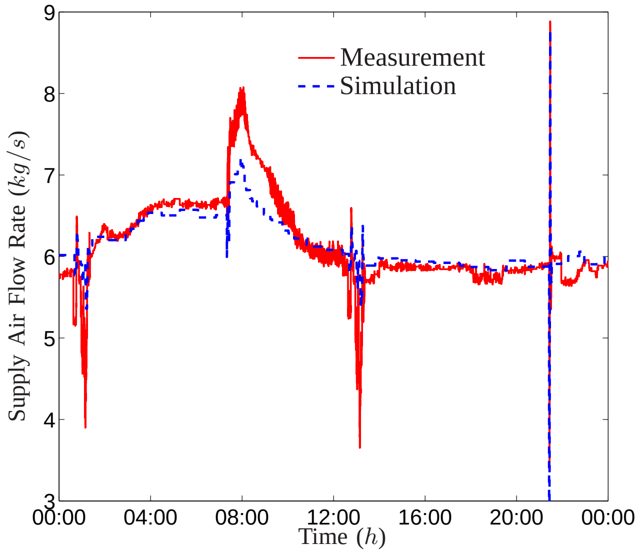

Fig. 5. Comparison of supply air flow rate between simulation and measurement.

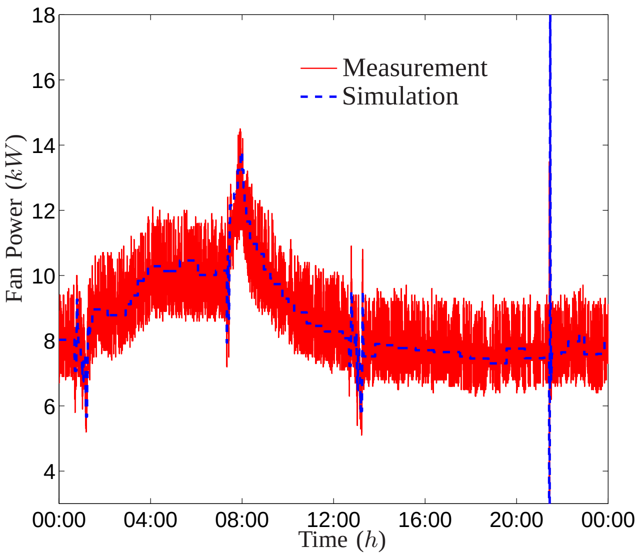

Fig. 6. Comparison of fan power between simulation and measurement.

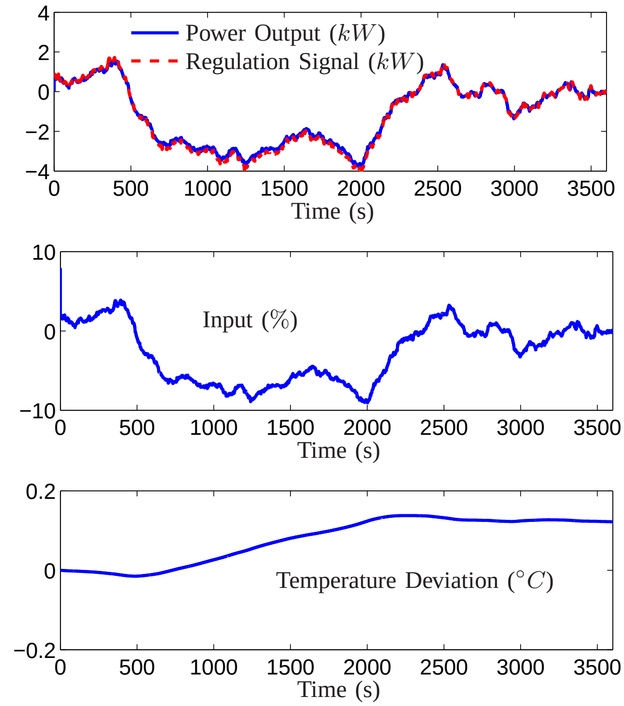

Fig. 7. Numerical experiment of output tracking a regulation signal for a single building.

VI. Numerical Simulations

In this section, we conduct numerical experiment of optimal control of a single building HVAC system to tracking a *scaled* time-varying regulation signal. The regulation signal is obtained from PJM [16]. For the purpose of simulation for a single building, the magnitude of the regulation signal is scaled to be less or equal to $4\ kW$. The weights used in the cost function J are $Q = 100, R = 1$.

Figure 7 shows the simulation result of output tracking a scaled regulation signal with duration of one hour. The time took to solve the optimal control problem is a typical desktop computer is about three minutes. The top plots depicts the regulation signal and the power output of the HVAC system. We see that the output perfectly tracks the regulation signal. The optimal control solution is shown in the middle plot. Its magnitude is less than 10%. Recall the control input is in percentage, this means the actual control is within the bound $[u^{fan*} - 10, u^{fan*} + 10]$, where u^{fan*} is nominal control value ($u^{fan*} \approx 65\%$). This justified the linearization assumption. The bottom plot depicts the temperature deviation of the building inside temperature. We observe that the maximum deviation is less than 0.2°, this means the building inside environment rarely changes and the occupants inside the building would not notice any change in the climate.

Additional simulations have been conducted for the cases where the regulation signal durations are shorter than one hour. It was shown that the time to solve the optimal control solution is only $1/30$ of the duration of the regulation signal. This implies the proposed method is highly feasible for real-time frequency regulation. In addition, for shorter duration of regulation and/or for a typical regulation signal with average positive and negative period of 20 minutes [12], the temperature deviation of the building inside temperature is much smaller than 0.2°, which is unnoticeable for the occupants inside the building.

VII. Conclusions and Future work

We studied providing ancillary service to the power grid by using the flexibility of HVAC systems in commercial buildings. We constructed models for the buildings thermal and power consumption dynamics of the HVAC system. Key models parameters are estimated by using the measurement data of the Pugh Hall in University of Florida. Thanks to the large thermal capacity of commercial buildings and the flexibility of the HVAC systems, we show that commercial buildings could dynamically adjust it power consumption to provide ancillary service without noticeably impacting the building inside environment and the occupants' comfort.

Based on the simulation results, we show that for a single HVAC system with $35\ kW$ supply fan, it is able to provide about $4kW$ capacity of ancillary service to the grid. In Pugh Hall of University of Florida, which serves about $5,000$ square feet, there are three HVAC systems. This means for a typical commercial building, it could provide about $10kW$ regulation capacity to the grid. In an aggregated manner, for a group of thousands of building, it is expected that they can provide the whole regulation service for an ISO such as PJM.

In this paper, we studied providing ancillary service using flexible HVAC systems in commercial buildings. The next step is to study optimal dispatch of distributed energy resources by commercial buildings. Another task is to implement the proposed method in a commercial building for on-site test.

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