How demand response from commercial buildings will provide the regulation needs of the grid

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***Abstract*— The statement "energy is not storable" is heard in energy conference lectures around the world, even though each person in the audience is sitting in a vast energy storage device. The heat storage in buildings is an enormous untapped resource for providing regulation services. This will be especially important as the grid is subject to more and more volatility from the introduction of power from renewable energy sources.**

This paper describes how regulation services can be obtained by exploiting the inherent flexibility of HVAC (Heating, Ventilation, Air Conditioning) systems in commercial buildings. A particular simulation test case is considered - A large commercial building at the University of Florida. The conclusions of this research demonstrate that, 1) A simplified model of the building that is adequate for control can be obtained from input-output measurements. In this study, the only control input considered is the supply fan power. 2) Control synthesis to regulate the building air temperature while simultaneously providing regulation to the grid can be cast as an LQR problem that admits a simple closed form solution. 3) Numerical experiments show that for this HVAC system, 15% of fan power capacity can be provided for regulation, while maintaining indoor temperature deviation to no more than ± 0.2 °C.

Based on these results, we conclude that the HVAC systems in $90,000$ medium-sized commercial buildings can provide the entire regulation service needed by PJM today, without any noticeable change in indoor air quality. The total regulation services that can be potentially provided by all the commercial buildings in the U.S. that have the necessary equipment in place are much higher. Our results indicate that supply fans in existing commercial buildings can provide about 70% of the current regulation capacity needed in the United States.

I. INTRODUCTION

Due to growing environmental concerns as well as economic and political requirements, the envisioned future power grid will increasingly rely on renewable energy sources such as wind and solar. 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 increase 60%, from 9 Quads in 2012 to 14.4 Quads in 2032.

However, renewable energy sources are volatile, intermittent and uncontrollable. With deep penetration of renewable resources, it will be challenging to maintain the reliability of the current electric grid infrastructure, since existing regulation services do not have enough capacity to compensate the volatility of renewables. Moreover, most current ancillary service comes from fossil fuels, which produce green-house gases. 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 75% of total electricity generated. Furthermore, HVAC (Heating, Ventilating, and Air Conditioning) systems account for 50% of the electricity consumed in buildings [4]–[6]. This paper is on how to utilize the inherent flexibility of HVAC power demand in commercial buildings to provide ancillary services to the grid.

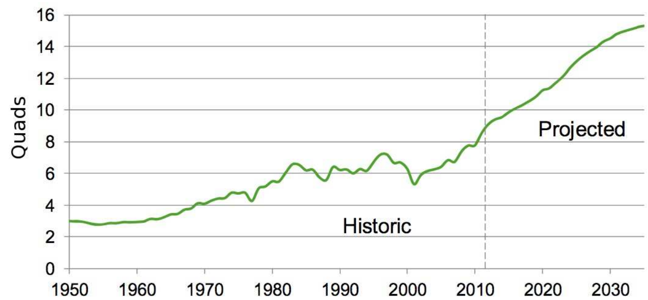

Fig. 1. History and projection of U.S. renewable energy consumption.

The reasons for choosing commercial buildings have several manifolds. First, Building Automation Systems (BAS) serve about one-third of the overall commercial buildings in the United States [7]. These systems can receive regulation signals from ISOs (Independent Service Operators) without requiring a smart meter infrastructure that is not currently mature. Second, since BAS already integrates the HVAC control systems of the building, manipulating the control variables needed for providing regulation services is easily implemented through a BAS. Third, a large fraction of commercial buildings are equipped with fast-responding VFDs (Variable Frequency Drives). These power electronic devices can be controlled through the BAS to change the supply fan power consumption of the building on the fly. Last but not the least, commercial buildings have large thermal capacitance, small variations in mass flow rate (thus fan speed) barely 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

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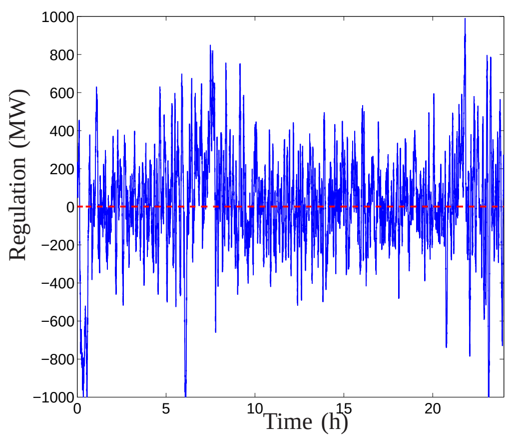

Fig. 2. Regulation signal of PJM for a typical day.

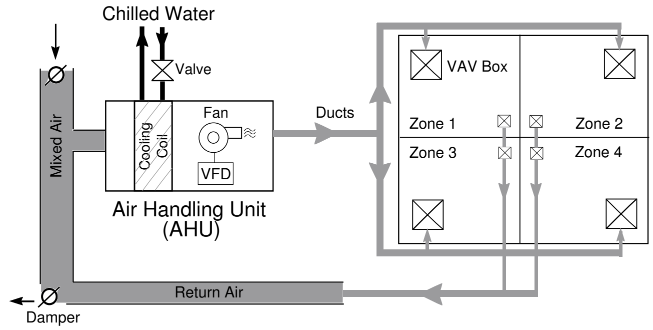

Fig. 3. The HVAC system of a four-zone building.

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 on frequency regulation by using the flexibility of commercial building HVAC systems. Figure 2 depicts the regulation signal[1] of PJM for a typical day. We see from the figure that regulation is a *zero-energy* service [8]. 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 [9]. 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.

The idea of using buildings to provide ancillary services to improve grid reliability is not 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 [10]–[13]. 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], [14]. In [15]–[17], the authors use pre-cooling of commercial buildings to reduce peak load demand. Strategies of using buildings to provide demand response have also been explored in [18]–[20]. To the best of our knowledge, though, using commercial buildings to provide frequency regulation services to the grid has not been explored so far.

In this paper we first construct a thermal model for a commercial building's indoor temperature and a power consumption model for its HVAC system. The key parameters of these models are estimated based on the measurements obtained from a $40,000$ sq. ft. building (Pugh Hall) at the University of Florida campus in Gainesville, FL. We then propose a control algorithm for tracking of a time-varying regulation signal by manipulating the supply fan speed of the air handling unit in the building. Numerical experiments show that it is feasible to use 15% of the total fan power as a flexible demand without noticeably impacting the building's indoor environment and occupants' comfort.

The rest of this paper is organized as follows. Section II presents the thermal and power consumption models of the commercial building HVAC system. The linearized models of thermal and power consumption dynamics are derived in Section III. The 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 the supply fan, and it is cooled down to a constant setpoint temperature (usually $12.8°C$) 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

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

where the variables and parameters, as well as 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 $c_p m(T_{la} - T)$ denotes the heat gain from the air conditioner. The last term Q_x

[1]In this paper, we refer ACE (area control error) as the regulation signal.

on the RHS of (1) is the heat gain from reheating, solar, occupants, lights, etc.

TABLE I
DESCRIPTION OF PARAMETERS OF 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	$J/g/^\circ C$
m	supply air flow rate	kg/s
Q_x	heat gain from reheating, solar, etc.	W

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

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

where c_1 is a constant, and V^{fan} is the fan speed expressed 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 progressively to its setpoint to protect the fan motor. 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 BAS.

C. Power consumption model of HVAC system in commercial buildings

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 [21]

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

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

Remark 1: In this paper, we assume the electricity power consumptions of (i) the furnace that supplies the hot water to the VAV boxes (for reheating) and (ii) chiller/cooling tower that provide chilled water to the cooling coil of the AHU are independent of the fan power. In many HVAC systems, the furnaces consume natural gas instead of electricity. The second assumption may appear strong - the power consumed by the chiller and cooling tower that provide the chilled water to the cooling coil in the AHU may in fact change if the fan speed changes. However, the dynamic interconnection between the AHU and the chiller can be thought of as a low pass filter due to the large mechanical inertia of the chiller/cooling tower equipment. Therefore, *high frequency* variations in the fan power will not change the power consumption of the chiller/cooling tower. Thus, the decoupled assumption - that fan speed variations do not change chiller power consumption - holds as long as fan speed variations are fast and of small magnitude. In addition, chilled water is supplied from a water storage tank in some HVAC systems. The purpose of using water storage is to shift the peak cooling demand in respond to dynamic utility prices, ambient temperature, etc. [22]. For systems with water tank, again the decoupled assumption holds.

For HVAC systems in which the power consumptions of the furnaces and chillers/cooling towers are dependent of the fan power, dynamic models that relate the fan action to the furnace and chiller powers need to be considered in the control design. Such HVAC systems can provide greater ancillary service to the grid, but the control design problem is more complex. □

III. LINEARIZED MODELS OF BUILDING HVAC THERMAL AND POWER DYNAMICS

During occupied operation mode (about 07:30 am - 22:30 pm), the building HVAC system is operating near a steady-state status. The indoor temperature is controlled to a fixed setpoint (about $22.4^\circ C$, see Figure 4), and the supply fan is also running at an approximately constant speed. For control design, we linearize the building thermal dynamics (1)-(3) and the power consumption dynamics (4) near their operation equilibrium. At steady-state, the thermal model (1) satisfies

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

where T^* and 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

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

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

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

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

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

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

$$A = \begin{bmatrix} -\frac{1+c_pR_wm^*}{CR_w} & \frac{c_p(T_{la}-T^*)}{C} & 0 \\ 0 & -\frac{1}{\tau} & 0 \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix},$$

$$B = \begin{bmatrix} 0 \\ \frac{c_1}{\tau} \\ \frac{3c_2(V^{fan*})^2}{\tau} \end{bmatrix},$$

$$C = \begin{bmatrix} 0, & 0, & 1 \end{bmatrix}.$$

Remark 2: In practice, although the outside air temperature T_{oa} and the heat gain from solar, lights and so on are time-varying, the changes in these parameters are much slower than the thermal and power consumption dynamics. 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 perform output-tracking of a time-varying regulation signal by manipulating the supply fan speed of the HVAC system. We pose the following optimal control problem

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

where $r(t)$ is the regulation signal provided by the independent system operator (ISO), and T_r is a user-specified 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 [23], the optimal control solution is given in 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

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

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

$$\begin{aligned}&\frac{db(t)}{dt} = -(A - BR^{-1}B^TP(t))^Tb(t) - C^TQr(t), \\ &b(T_r) = 0.\end{aligned}$$

V. Parameter Estimation

In this section, we aim to identify the parameters (such as thermal capacitance C) of the thermal and power models from measurements obtained from Pugh Hall in University of Florida. Pugh Hall, a LEED (Leadership in Energy and Environmental Design) certified building, was constructed in 2008 and has an area of about $40,000$ square feet. There are three AHUs in the building. The parameters in this paper are identified from AHU #1 data. In the estimation process, the building indoor temperature T is taken as the average temperature of all the zones served by AHU #1.

Figure 4 depicts the comparison between the measured temperature and the predicted temperature using model (1). The constant parameters selected are $C = 7 \times 10^5 \ J/°C$, $R_w = 5 \times 10^{-3} \ °C/W$, $T_{la} = 12.8 \ °C$, $c_p = 1006 \ J/g/°C$. The outside air temperature T_{oa} is obtained from historical data [24]. 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 sensors, 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 measurement and prediction 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 measurement and prediction using model (4). In addition, the time constant τ given in (3) is estimated as $\tau = 0.1 \ s$. Figure 7 depicts the comparison of supply fan speed between measurement and prediction using model (3).

VI. Numerical Simulations

In this section, we conduct numerical experiment of optimal control of a single building HVAC system to track a *scaled* time-varying regulation signal. The regulation signal is obtained from PJM [25]. For the purpose of simulation for a single building, the magnitude of the regulation signal is scaled down to be less or equal to $5 \ kW$. The weights used in the cost function J are $Q = 100, R = 1$.

Figure 8 shows the simulation result of output tracking the scaled regulation signal with duration of one hour. The time took to solve the optimal control problem in a typical desktop computer is about two minutes. The top plots depicts the scaled regulation signal and the fan power deviation. We see that the output (fan power deviation) tracks the regulation signal extremely well. The optimal control solution is shown in the middle plot. Its magnitude is less than 12. Recall the control input is in percentage, this means the actual control

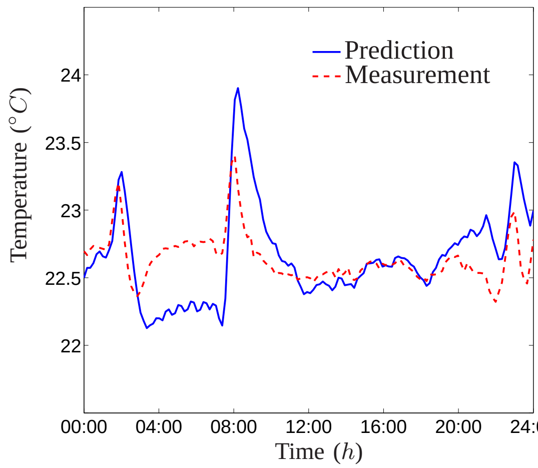

Fig. 4. Comparison of building indoor temperature between measurement (from Pugh Hall) and prediction using model (1).

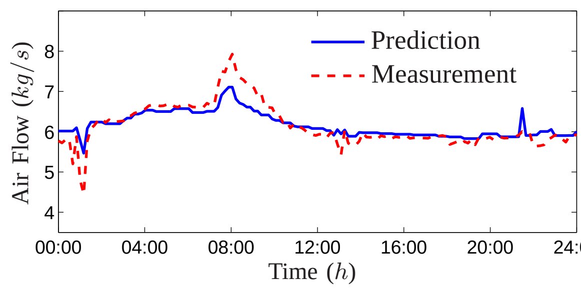

Fig. 5. Comparison of supply air flow rate between measurement (from Pugh Hall) and prediction from measured fan speed with estimated c_1 and model (2).

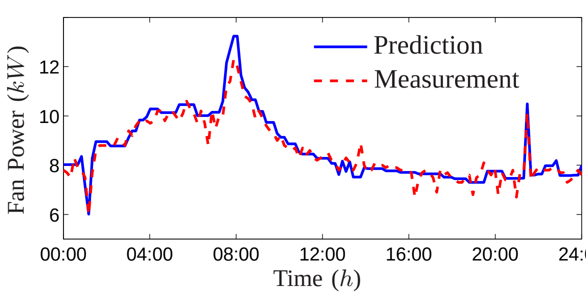

Fig. 6. Comparison of fan power between measurement (from Pugh Hall) and prediction from measured fan speed with estimated c_2 and model (4).

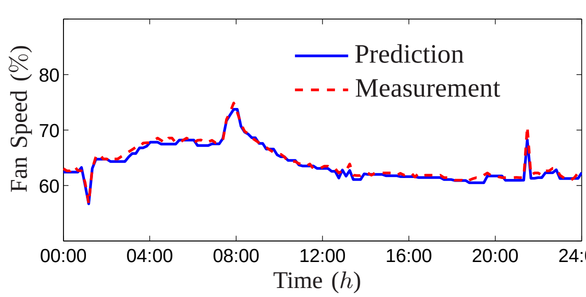

Fig. 7. Comparison of fan speed between measurement (from Pugh Hall) and prediction from measured fan input with estimated τ and model (3).

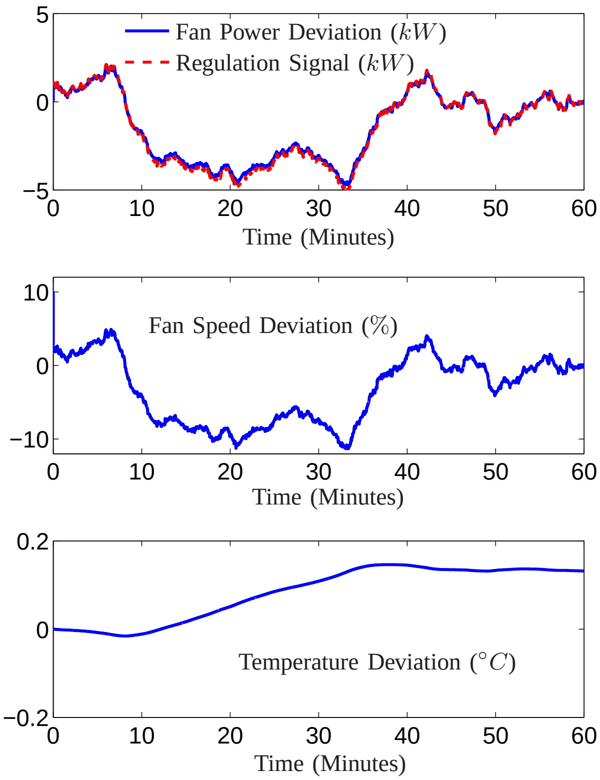

Fig. 8. Numerical experiment of output tracking a regulation signal for a single building. The plots show fan power deviation $\tilde{P}^{fan}$ and scaled regulation signal (top), fan speed deviation $\tilde{V}^{fan}$ (middle), and temperature deviation $\tilde{T}$ (bottom).

is within the bound $[u^{fan*} - 12, u^{fan*} + 12]$, where u^{fan*} is nominal control value ($u^{fan*} \approx 65$). The bottom plot depicts the deviation of the building indoor temperature from the desired set point. We observe that the maximum deviation is less than $0.2^\circ C$, this means that the change in the building's indoor environment is negligible, and the occupants would not notice any change.

Additional simulations have been conducted for the cases where the regulation signal durations are shorter than one hour. It was observed that the time to compute the optimal control solution is only about $1/30$ of the duration of the regulation signal. This implies the proposed method is 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 [9], the temperature deviation of the building indoor temperature is even smaller than the $0.2^\circ C$ change shown above.

VII. Discussion of Results 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 model parameters are estimated by using measurements from Pugh Hall at the University of Florida. Thanks to the large thermal capacity of commercial buildings, a building can dynamically adjust it power consumption to provide ancillary service to the grid without noticeably impacting the building's indoor environment.

The simulation results show that a single 35 kW supply fan can easily provide about 5 kW capacity of ancillary service to the grid. In Pugh Hall of University of Florida, there are two other AHUs, whose supply fan motors are 25 kW and 15 kW respectively. This means Pugh Hall by itself could provide about 11 kW regulation capacity to the grid. A group of $90,000$ such buildings can therefore provide 990 MW regulation reserves. This is enough to provide the entire regulation reserves to PJM (see Figure 2). The total available reserves are much higher. There are about 5 million commercial buildings in the U.S., with a combined floor space of approximately $72,000$ million sq. ft, of which approximately one third of the floor space is served by HVAC systems that are equipped with VFDs [26]. Assuming fan power density per sq. ft. of all these buildings to be the same as that of Pugh Hall who has an area of $40,000$ square feet, the total regulation reserves that are *potentially available* from all the VFD-equipped fans in commercial buildings in the U.S. are approximately 6.6 GW, which is about 70% of the total regulation capacity needed in the United States [27].

In this paper, we studied the problem of providing ancillary services utilizing power demand flexibility in commercial buildings' HVAC systems. Only fan power was used to provide flexible demand. Chillers consume much more power than fans and present a potentially larger opportunity. The problem of combined use of chillers and fans to provide regulation is the logical next step. There is one impediment in utilizing chillers to provide regulation reserves. It is not clear what fraction of commercial building chillers are equipped with VFDs; and VFDs are essential for the high frequency variations of their power consumption without putting stress on the machinery. Another interesting avenue for further work is optimal dispatch of distributed energy resources by commercial buildings that have on-site distributed generation capability. We are also working on implementing the method proposed in this paper in a commercial building (Pugh Hall) for feasibility demonstration.

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