# Ancillary Service for the Grid via Control of Commercial Building HVAC Systems

He Hao, Anupama Kowli, Yashen Lin, Prabir Barooah, and Sean Meyn

***Abstract*— The thermal storage potential in buildings is an enormous untapped resource for providing various services to the power grid. The large thermal capacities of commercial buildings in particular make the power demands of their Heating, Ventilation, and Air Conditioning (HVAC) systems inherently flexible. In this paper, we show how fans in air handing units (AHUs) of commercial buildings alone can provide substantial regulation service, with little change in their indoor environments. A feedforward architecture is proposed to control the fan power consumption to track a regulation signal. The proposed control scheme is then tested through simulations based on a high fidelity commercial building model constructed based on Pugh Hall located on the University of Florida campus. For the HVAC system under consideration, numerical experiments demonstrate how up to $15\%$ of fan power capacity can be deployed for regulation purposes while having little effect on the building inside temperature. The regulation signal can be successfully tracked in the frequency band $[1/\tau_0,\ 1/\tau_1]$, where $\tau_0 \approx 3$ minutes and $\tau_1 \approx 8$ seconds. Our results indicate that fans in existing commercial buildings in the U.S. can provide about $70\%$ of the current national regulation reserve in the aforementioned frequency band.**

## I. Introduction

The proper functioning of a power grid requires continuous matching of supply and demand, in spite of the randomness of electric loads and the uncertainty of generation. A direct consequence of a supply-demand mismatch is a deviation in the system frequency. One of the most important ancillary services is frequency regulation: it is deployed on the fastest time-scale (seconds to minute) to correct the short-term power imbalance in load and generation to maintain system frequency within prescribed limits. This service has been traditionally provided by generators by tracking a regulation signal sent by the grid operator that dictates changes in the generators' outputs. In this paper, we argue that (a) commercial buildings can be tapped for ancillary services, (b) HVAC systems can be manipulated for regulation service on faster timescales more effectively than generators, and, (c) commercial buildings can provide this service at a very low cost.

Studies indicate that increased reliance on renewable generation introduces greater volatility and uncertainty in power system dynamics and imposes additional regulation requirements on the grid [1], [2]. The regulation requirements can be lowered if faster responding resources are available [3], [4]. These factors coupled with the search for cleaner sources of flexibility as well as regulatory developments such as Federal Energy Regulatory Commission (FERC) order 755 have garnered a growing interest in tapping the fast response potential of storage and demand-side resources. In the absence of utility-scale storage alternatives, loads with virtual storage capabilities such as heating and cooling loads, water pumps and refrigerators are fast-becoming popular choices to fulfill ancillary service requirements of the grid [5], [6]. Additionally, manufacturing companies and agriculture farms have been successfully engaged by ramping up and down their energy use in response to the requirements of the grid [7], [8].

The flexibility potential of demand-side resources was recognized as early as the 1980s by Schweppe *et al.* in [9] wherein a frequency responsive mechanism was proposed to control thermal loads. More recent approaches have proposed aggregating residential loads such as refrigerators, air conditioner and water heaters for ancillary service provision [10]–[12]. Pre-cooling of buildings to reduce peak load has been extensively studied; see for example references [13], [14]. Many load control mechanisms implemented in utilities and explored in the literature are primarily concerned with low frequency changes in demand, i.e., the changes occurs over minutes/hours timescale. In this paper, we focus high frequency load changes in commercial buildings, so as to provide regulation service to the grid.

Buildings account for $75\%$ of total electricity consumption in the U.S., with roughly equal share between commercial and residential buildings [15]. Buildings are, therefore, a natural candidate when seeking demand-side flexibility. The choice of commercial buildings is motivated by several important factors. First, a commercial building can provide a larger amount of demand response (compared to a residential building) due to its much larger thermal inertia. Second, approximately one third of the commercial building floor space is equipped with variable frequency drives (VFDs) that operate the supply fans and pumps. Their speeds and power consumptions can be commanded to vary quickly and continuously, instead of in an on/off manner. This is a crucial advantage for providing regulation service, since the regulation signal to be tracked changes in the order of seconds. Third, a large fraction of commercial buildings in the United States are equipped with Building Automation Systems. These systems can receive regulation signals from grid operators and manipulate the control variables needed for providing regulation service, without requiring additional equipments such as smart meters. Ancillary services can thus be provided at virtually no cost; these are obtained as a

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simple add-on to the current HVAC control system.

In this paper we present preliminary results that showcase the feasibility of extracting regulation service from commercial buildings. We consider power consumption of the fan in the building's HVAC system as the only source of flexibility. A feedforward control architecture is proposed, wherein the fan speed commanded by the building's existing control system is modified so that the change in the fan's power consumption tracks the regulation signal. A simplified thermal model of a building is used for control design. The model parameters are identified from data collected from a commercial building in the University of Florida campus (Pugh Hall). The controller is then tested on a high fidelity non-linear model constructed from the same building. The results show that the simplified model is adequate for the purpose of control; the controller performs on the complex model as predicted by the simplified model. Numerical experiments show that it is feasible to use up to $15\%$ of the total fan power for regulation service to the grid, without noticeably impacting the building's indoor environment and occupants' comfort, provided the bandwidth of regulation service is suitably constrained. To ensure the comfort of occupants, and to manage stress on HVAC equipment, both upper and lower bounds on bandwidth are necessary. Based on simulation experiments, this bandwidth is estimated to be $[1/\tau_0, 1/\tau_1]$, where $\tau_0 \approx 3$ minutes, and $\tau_1 \approx 8$ seconds.

## II. Control Architecture

The regulation signal sent by the grid operator is typically a sequence of pulses at $4$ second intervals. In the case of loads engaged in regulation, the magnitude of the pulse is the amount of deviation in their power consumption asked by the grid operator. Suppose the building is required to provide $r(t)$ (in kW) amount of regulation service at time $t$. This signal is referred to as the *(building-level) regulation reference*. The job of a *(building-level) regulation controller* is to change the power consumption of the building so that the change tracks the regulation reference.

We propose to achieve this through a feedforward controller. The controller changes the command to the fan so that the fan's power consumption is changed in such a way that the deviation in consumption – both positive and negative – tracks the regulation reference $r(t)$. The architecture of the control system is shown in Fig. 1. The regulation signal $r$ is transformed to a *regulation command* $u^r$ by the regulation controller. This command is then added to the nominal fan speed command $u^b$ produced by the building's fan controller. Suppose $p^b(t)$ is the nominal power consumption of the fan due to the thermal load on the building, and $p^{b+r}(t)$ is the fan power consumption with the additional regulation command. Then the deviation in power consumed by the fan is $\Delta p(t) := p^{b+r}(t) - p^b(t)$. Clearly, changing the fan speed from the nominal value determined by the building's existing control system will change the air flow to the building. The goal is to design the regulation controller so that $\Delta p(t)$ tracks $r(t)$ while causing little change in the building's indoor environment (measured by the deviation of the zonal temperatures from their set points).

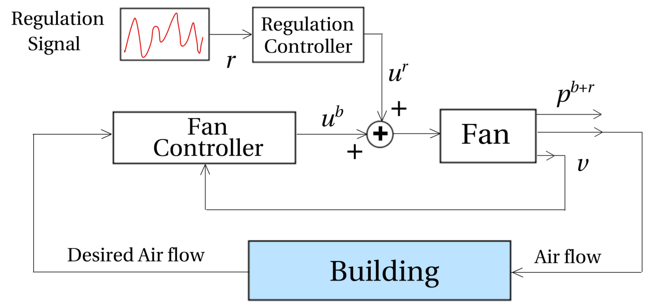


Fig. 1. The proposed control architecture. A transformed regulation signal will be used to compute the additional fan speed command $u^r(t)$ so that the resulting deviation of the fan power $p^{b+r}(t)$ from the nominal value $p^b(t)$ tracks the regulation signal $r(t)$, while having little effect on the indoor temperatures.

*Remark 1:* In this paper, we assume that the power consumed by the furnace supplying hot water to the variable air volume (VAV) boxes (for reheating) and the chiller/cooling tower providing chilled water to the cooling coil of the AHU are independent of the power consumed by the fan. 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 may in fact change if the fan power and, consequently, 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 decoupling assumption – that fan power variations do not change chiller power consumption – holds as long as the variations are fast and of small magnitude. In addition, in some HVAC systems chilled water is supplied from a water storage tank. For such systems, the decoupling assumption holds naturally. □

## III. Data-driven Control Oriented Modeling

The dynamics of the complete closed loop system of a building relating zone temperatures to fan speed command are quite complex due to the interconnection of the zone-level controlled dynamics, dynamics of pressure distribution in the ducts, and building-level fan controller. For the purpose of control design, we derive simplified models of some of these components using data collected from Pugh Hall, a typical example of a modern commercial building.

### A. HVAC power consumption model

The power consumption of a fan is proportional to the cube of its speed [16]

$$p(t) = c_1 (v(t))^3, \tag{1}$$

where $c_1$ is a constant, and $v$ is the normalized fan speed in percentage. The fan speed is controlled by a fan controller so that the total mass flow rate tracks a desired total mass flow rate, denoted by $m^d(t)$. In practice the desired mass flow rate,

$m^d(t)$, is communicated to the fan speed indirectly through a change in the duct pressure caused by the actions of the zonal controllers. Here we make the simplifying assumption that the fan controller senses the desired value directly and changes the fan speed to make the actual mass flow rate through the AHU, $m(t)$, track $m^d(t)$. This allows us to sidestep the very challenging problem of modeling the duct pressure dynamics. Yet, the assumption is justified since that is what the fan control loop does, albeit indirectly. The mass flow rate has a linear relationship with the fan speed,

$$m(t) = c_2 v(t), \tag{2}$$

where $c_2$ is a constant. Similarly, given a desired air flow rate $m^d(t)$, the corresponding desired fan speed that the fan controller tries to maintain is $v^d(t) = m^d(t)/c_2$. In practice, the fan speed is controlled by the VFD which also accelerates or decelerates the fan motor slowly in the interest of equipment life. Because of this ramping feature, we assume the transfer function from the control command to the fan speed is of first-order

$$\tau \frac{dv(t)}{dt} + v(t) = u(t), \tag{3}$$

where $\tau$ is the time-constant, and $u(t)$ is the fan speed command sent by the fan controller. The fan speed controller is typically a PI (proportional integral) controller. Note that $v, v^d$ and $u$ are all measured in percentage. The parameters $c_1, c_2$ and $\tau$ in the models (1)-(3) are estimated from data collected from Pugh Hall. The data used is from one of the three AHUs in the building with a 35-kW rated fan motor which supplies air to 41 zones. Using a randomly chosen 24 hour long data set, the parameters are estimated to be $c_1 = 3.3 \times 10^{-5}\ kW$, $c_2 = 0.0964\ kg/s$, and $\tau = 0.1\ s$.

### B. Aggregated Building Thermal Model

In what follows, a simplified thermal model of the building based on the aggregate *building temperature* $T(t)$, which can be thought of as the average temperature of all zones, is discussed. This simple but non-linear thermal model relates the total mass flow rate to the building temperature.

Consider the following physics-based lumped thermal model of the building

$$C\frac{dT}{dt} = -\frac{1}{R}(T - T_{oa}) + c_p m(T_{la} - T) + Q, \tag{4}$$

where $C, R$ are the thermal capacitance of the building and the resistance that the building envelope provides to heat flow between the building and the outside. $T_{oa}$ is the outside air temperature, $c_p$ is the specific heat of air, $m$ is the supply air flow rate, and the leaving air temperature $T_{la}$ is the temperature of the air immediately downstream of the AHU. The variable $Q$ denotes the heat gain from reheating, solar radiation, occupants, lights and so on.

During normal business hours, the building's HVAC system operates near a steady-state status and the indoor temperature is maintained at a fixed setpoint. This allows us to linearize the dynamics. Following straightforward algebra, we obtain

$$\frac{d\tilde{T}}{dt} = -\frac{1 + c_p R m^*}{CR}\tilde{T} + \frac{c_p(T_{la} - T^*)}{C}\tilde{m}, \tag{5}$$

where $\tilde{T} = T - T^*$, $\tilde{m} = m - m^*$, in which $T^*$ and $m^*$ are the steady-state temperature and supply air flow rate. In addition, we have assumed $T_{oa}$ and $Q$ to be constants for the time durations under consideration. We use this assumption and linearization only for design and test the design through simulations with time varying signals and high fidelity nonlinear model.

We next aggregate the effect of all the zonal controllers into one controller that we call the *building temperature controller*, by imagining that it computes the desired total mass flow rate $m^d(t)$ based on the difference between the desired building temperature $T^d$ and actual building temperature $T(t)$, and then signals the fan controller to provide this mass flow rate. Since each of the zonal controllers in commercial buildings are usually PI controllers, we choose the building temperature controller to be a PI controller as well.

## IV. Regulation by Fan Command Manipulation

We claim that buildings can provide regulation service to the grid without causing discomfort to occupants or damaging the HVAC equipment so long as the bandwidth of the regulation signal is suitably constrained. The considerations in determining this bandwidth are discussed here along with the control strategy implemented to extract regulation service.

The bandwidth of the regulation signal sent to buildings should be chosen based on the following factors. First, high frequency content in resulting regulation command $u^r$ (see Fig. 2) is desirable up to a certain upper limit. Since the thermal dynamics of a commercial building have low-pass characteristics due to its large thermal capacitance, high frequency changes in the air flow cause little change in its indoor temperature. The statement is also true for individual zones of the building. Additionally, the VFD and fan motor have large bandwidth so that high frequency changes in the signal $u^r$ lead to noticeable change in the fan speed and, consequently, fan power. Both effects are desirable, since we want to affect the fan power consumption without affecting the building's temperature. However, an extremely high frequency content in $u^r(t)$ is not desirable as it might cause wear and tear of the fan motor. Likewise, $u^r$ should not have very low frequency content. Otherwise, even if the magnitude of $u^r$ is small, it may cause a noticeable change in the temperature of the building, which in turn will cause the zonal controllers to try to change air flow rate to reverse the temperature change. In effect, the building's existing control system will try to reject the disturbance caused by $u^r$. Being a feedback loop, this disturbance rejection property is already present in the building's control system. However, if fan and zonal controllers do not have high bandwidth, they would not reject high frequency disturbance. In short, the frequency content of the disturbance $u^r(t)$ should lie in a particular band $[f_{\text{low}},\ f_{\text{high}}]$, where the gain of the closed loop transfer function from $u^r$ to fan speed $v$ is sufficiently large while that of the transfer function from $u^r$ to temperature $T$ is sufficiently small.

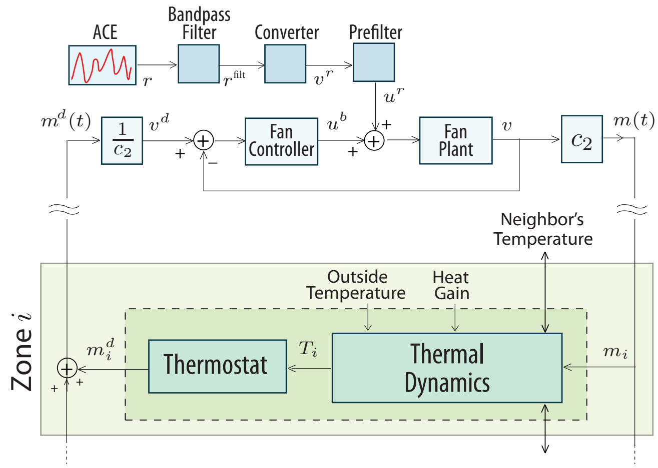


Fig. 2. Schematic representation of the interconnection between zone supply air flow request and the fan speed control architecture integrated with regulation.

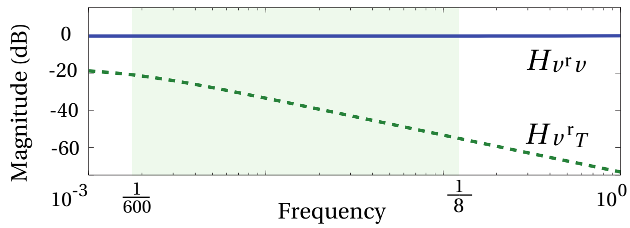


Fig. 3. Magnitude vs. frequency of the closed loop transfer functions from disturbance to fan speed $H_{v^r v}$, and from disturbance to temperature $H_{v^r T}$.

The parameters $f_{\text{low}}, f_{\text{high}}$ are design variables to compute a suitable regulation signal for the buildings. We assume that these variables describing the bandwidth along with the total capacity of regulation that the building can provide is communicated to the grid operator and used in constructing an appropriate regulation signal for the building. In the simulation studies described in Section VI, this is achieved in two stages. First, we pass the ACE (area control error) data $r(t)$ through a bandpass filter with a passband $[f_{\text{low}},\ f_{\text{high}}]$. Second, we design the PI gains of the fan controller and zonal controllers so that the closed loop gain criteria described above are met. This in practice may be an iterative design process.

Suppose the regulation signal to be tracked by the building is $r^{\text{filt}}(t)$. This signal is then converted into speed deviation command using linearization $\tilde{p}(t) = 3c_1(v^*)^2\tilde{v}(t)$, where $\tilde{p} := p - p^*, \tilde{v} := v - v^*$, and $p^*, v^*$ are the nominal power consumption and speed of the fan. Specifically, converter block in Fig. 2 is a static function that computes the command $v^r$

$$v^r = r^{\text{filt}}(t)/(3c_1(v^b)^2), \tag{6}$$

where $v^b$ is the nominal fan speed due to thermal load on building. The command $v^r$ is passed through a prefilter to produce the command $u^r$. The fan speed command that is sent to the VFD is $u^b + u^r$. The prefilter is needed so that the gain of the transfer function from $v^r$ to $v$ in the band $[f_{\text{low}},\ f_{\text{high}}]$ is close to 1, see Fig. 3. In this figure, as well as in simulation studies, we take $[f_{\text{low}},\ f_{\text{high}}]$ to be $[1/600,\ 1/8]$. The prefilter is designed by computing an approximate inverse of the transfer function from $u^r$ to $v$. The magnitude responses of two crucial transfer functions are shown in Fig. 3. We see from the figure that within the prespecified band, the transfer function from regulation command $v^r$ to fan speed $v$ has a relatively high gain while to the temperature $T$ has a extremely low gain.

## V. High Fidelity Non-linear Model for Simulation Studies

Although a simplified model is used for control design presented in Section III, we use a complex physics-based model in the simulation studies aimed at testing the controller performance. This model is briefly described next; see [17] for details.

To cope with the difficulty of modeling duct pressure dynamics that couple zone level dynamics to the fan dynamics, we make the following simplification. We assume that each zonal controller asks for a certain amount of air flow rate, by generating a *desired* air flow rate command $m_i^d(t)$ in response to the measured temperature deviation from the set point: $T_i^d(t) - T_i(t)$. The total desired supply air flow rate, $m^d(t)$, is the sum of the desired supply air flow rate into each zone $m_i^d(t)$:

$$m^d(t) = \sum_{i=1}^{n} m_i^d(t). \tag{7}$$

The signal $m^d(t)$ is the input to the fan speed controller: the desired fan speed is computed as $v^d(t) = m^d(t)/c_2$, cf. (2). The *actual* total mass flow rate is $m(t) = c_2 v(t)$, where $v(t)$ is the actual fan speed. It is divided among the zones in the same proportion as the air flow rate demands:

$$m_i(t) = \alpha_i m(t), \quad \alpha_i = m_i^d / \sum_j m_j^d. \tag{8}$$

The building's control system effectively performs this function, although signaling is performed through physical interaction and not through the exchange of electronic signals.

The thermal dynamic model of a multi-zone building is constructed by interconnection of RC-network models of individual zones and the corresponding zonal controllers. We consider the following RC-network thermal model for each zone in the building:

$$\begin{aligned} C_i \frac{dT_i}{dt} =& \frac{T_{oa} - T_i}{R_i} + \sum_{j \in \mathcal{N}_i} \frac{T_{(i,j)} - T_i}{R_{i,j}} \\ &+ c_p m_i (T_{la} - T) + Q_i, \end{aligned} \tag{9}$$

$$C_{(i,j)} \frac{dT_{(i,j)}}{dt} = \frac{T_i - T_{(i,j)}}{R_{(i,j)}} + \frac{T_j - T_{(i,j)}}{R_{(i,j)}}, \tag{10}$$

The above equation is similar to (4). The differences are that the second term on the right hand side of (9) represents the heat exchange between zone $i$ and its surrounding walls that separate itself from neighboring zones, and (10) models the heat exchange between zone $i$, zone $j$, and the wall separating them.

A widely used control scheme for zonal controllers in commercial buildings is the so-called "single maximum". There are three operating modes in this control scheme: cooling mode, heating mode, and deadband mode. In this paper, we assume all the zones are in the *Cooling Mode*. In this mode, there is no reheating, and the supply air flow rate is varied to maintain the desired temperature in the zone. Typically a PI controller is used that takes temperature tracking error $T_i^d - T_i$ as input and desired air flow rate $m_i^d$ as output. The high fidelity model of a multi-zone building's thermal dynamics is constructed by coupling the dynamics of all the zones and zonal controllers, with $m_i$'s as controllable inputs, $T_{oa}, Q_i, T_{la}$ as exogenous inputs, and $T_i$'s and $m_i^d$'s as outputs. A schematic of the complete closed loop dynamics with the high fidelity model, along with all the components of the regulation controller, is shown in Fig. 2.

*F-building test case:* For our studies, we imagine a fictitious building with $4$ stories and $44$ zones, which we name *F-building*. Each story has $11$ zones constructed by cutting away a section of Pugh Hall. The HVAC system of the F-building consists of a single AHU and zonal controllers for each of its zones. The F-building is meant to mimic the section of Pugh Hall serviced by one of the three AHUs that services 41 zones. The zones serviced by each of the AHUs in Pugh Hall are not contiguous, which necessitates such a fictitious construction. We identify the model of each of these 11 zones from data collected in Pugh Hall. Model identification consists of determining the $R$ and $C$ (resistance/capacitance) parameters in the model (9)-(10) for the zone. The least-squares approach with direct search method described in [17] is used to fit the model parameters.

## VI. Regulation reference tracking by F-building fan

We describe here simulation experiments which test the performance of the developed regulation controller for tracking regulation signal by varying the fan power. The F-building described in the previous section is used for simulations. The ACE signal $r$ used for constructing the regulation reference $r^{\text{filt}}$ for the F-building is taken from a randomly chosen $5$-hr long sample from PJM (Pennsylvania-New Jersey-Maryland Interconnection) [18]. It is then scaled so that its magnitude is less than or equal to $5$ kW – a conservative estimate of the regulation capacity of F-building. A fifth-order Butterworth filter with passband $[1/600, 1/8]$ Hz is used as the bandpass filter while constructing $r^{\text{filt}}$. The choice of the passband will be explained soon.

To unambiguously determine performance of the control scheme, we perform two simulations. First, a benchmark simulation is carried out with the regulation controller turned off so that $u^r(t) \equiv 0$. The fan speed is varied only by the building's closed loop control system to cope with the time-varying thermal loads. Then, a second simulation is conducted with the regulation controller turned on and all the exogenous signals (heat gains of the building, outside temperature) are identical to those in the benchmark simulation. The fan power deviation, $\Delta p(t)$, is the difference between the fan power consumption observed in the second simulation and that in the first. Fig. 4 (Top) shows the regulation reference $r^{\text{filt}}(t)$ and the actual regulation provided: $\Delta p(t)$. We see that the fan power deviation tracks the regulation signal extremely well. The deviation in the fan speed caused by tracking the regulation signal is depicted in the middle plot. Although the baseline fan speed is time-varying, the regulation controller designed with a constant baseline speed assumption performs well. Finally, the bottom plot depicts the deviation of the temperatures of the individual zones from their set points. We observe that the maximum deviations are at the beginning of the simulation, this is because of i) initial conditions, and, ii) to provide the same amount of regulation, the fan speed deviation from nominal speed at lower speed is larger than that at higher speed (cf. Eq. (6)). Nevertheless, the temperature deviations after transient are very small, which will most likely be unnoticed by the occupants.

The passband of the bandpass filter is designed based on additional simulations not reported here. These simulations show that if the regulation signal contains frequencies lower than $1/600$ Hz (corresponding to period of $10$ minutes), the zonal controllers compensate for the indoor temperature deviations in the zones by modifying air supply requirements, thus nullifying the speed deviation command of the regulation controller. This results in a poor regulation tracking performance. We estimate the upper band limit to be $1/8$ Hz to avoid stress on the mechanical parts of the supply fan. In addition, since the ACE data from PJM is sampled every $4$ seconds, the detectable frequency content in this data is limited to $1/8$ Hz. Thus, the passband of the bandpass filter is chosen as $[1/600, 1/8]$ Hz; cf. Fig. 3. Since we have neglected chiller dynamics, which has a time constant typically larger than $200$ seconds [19], we expect that in practice the proposed control architecture will be able to successfully track regulation signal in the band $[1/\tau_0, 1/8]$, where $\tau_0 \leq 200$. At frequencies lower than that, unmodeled dynamics of chillers and cooling coils may affect performance.

### *Regulation potential of commercial buildings in the U.S.*

The simulation results show that a single $35$ kW fan can easily provide about $5$ kW regulation capacity, which is about $15\%$ of the total fan power. The regulation capacity is estimated in a conservative way. In the simulation, we consider a hot summer and high load, in which case the peak fan speed almost reaches the maximum, see the middle plot of Fig. 4. The problem of quantifying the maximum regulation capacity as a function of weather, internal load, building temperature is under current investigation.

In Pugh Hall of University of Florida, there are two other AHUs, whose fans are $25$ kW and $15$ kW respectively. This means Pugh Hall by itself could provide about $11$ kW regulation capacity to the grid. 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

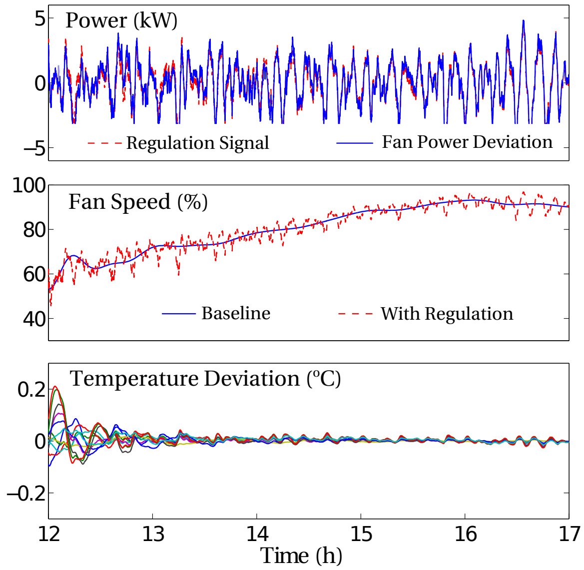


Fig. 4. Numerical experiment of tracking a regulation signal for a single building. The plots show the regulation signal $r^{\text{filt}}$ and fan power deviation $\Delta p$ (top), fan speed with and without regulation (middle), and temperature deviation $\tilde{T}_i$ for each zone (bottom).

with VFDs [20]. Assuming fan power density per sq. ft. of all these buildings to be the same as that of Pugh Hall which has an area of $40,000$ sq. ft., 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 [21].

## VII. Conclusions and Future work

Thanks to the large thermal capacity, commercial buildings can provide significant ancillary service for reliable grid operations, without noticeably impacting the building's indoor environment. We showed that fans in commercial buildings alone can provide a large fraction of the current regulation requirement of the U.S. national grid without requiring additional investment. A feedforward control architecture was proposed to provide this service that was designed on simplified models of building and HVAC dynamics. The architecture was tested through simulations on a calibrated high-fidelity model of a commercial building where these assumptions were violated. Still, good regulation reference tracking was achieved with negligible change in the building's temperature.

In this work we have neglected the effect of chillers and cooling coils on power consumption and indoor climate since their dynamics are slow. A more accurate characterization of the low frequency range at which regulation signal can be tracked with the proposed architecture requires incorporating these dynamics. In addition, 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. Another avenue for further work is optimal dispatch of distributed energy resources by commercial buildings that have on-site distributed generation capability. We are considering various other sources of ancillary service: In Florida, pool pumps are a natural source of ancillary service at lower frequency bands. It is likely that flywheels, batteries and other sources must be used to address regulation at very high frequencies. At ultra-low frequencies, we envision flexible manufacturing (e.g., desalination and aluminum manufacturing) will play an increasingly important role for ancillary service.

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