Commercial building HVAC system in power grid ancillary services

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***Abstract*—With the introduction of volatile renewable energy sources into the grid, the need for inexpensive ancillary service increases. We propose a method to provide ancillary service by using the flexibility of demand in commercial building HVAC (heating, ventilation, air-conditioning) systems. In particular, we show how a regulation command transmitted by a balancing authority can be tracked by varying the cooling demand in commercial buildings in real-time. A key idea here is the bandwidth limitation of the regulation signal, which allows the building's HVAC system to provide this service with little effect on the indoor climate. The proposed control scheme can be applied on any building with a VAV (Variable Air Volume) system and on-site chiller(s). Simple calculations show that the commercial buildings in the U.S. can provide 12 GW of regulation reserves in the frequency band $f \in [1/(60\,\textbf{min}), 1/(3\,\textbf{min})]$ with virtually no change in the indoor climate, while meeting the standards of PJM for regulation following exceedingly well.**

I. Introduction

To ensure the functionality and reliability of a power grid, supply and demand must be balanced at all time scales. Correcting the mismatch between them, which occurs due to many sources of uncertainty, requires ancillary service. A large amount of ancillary service will be required in the future if a large fraction of our energy needs is to be met from renewable energy sources with their associated unpredictability and volatility.

Traditionally, ancillary service to balance demand and supply in fast time-scales is provided by fast ramping generation. Another way to achieve balance is to change demand. The inherent flexibility of many electric loads, when harnessed without impacting consumer comfort, can be an inexpensive source of ancillary service. Recent research has shown that thermostatic loads in particular, can provide ancillary service with the help of appropriate control algorithms; see [1], [2], [3] and references therein. These works focus on residential loads such as A/C and refrigerators. One drawback of residential loads is that often only on/off control is available, which greatly reduces the flexibility of control strategy that can be applied.

Commercial buildings have a number of advantages over residential buildings in providing ancillary service by exploiting load flexibility. About 30% of all commercial building floorspace in the U.S. is serviced by VAV systems in which the air flow rate can be varied continuously between a low and high value [4]. This makes them particularly well-suited for sophisticated control. Moreover, many buildings are equipped with building automation systems, making the task of implementing additional control algorithms easy and inexpensive. Finally, commercial buildings have high thermal inertia, so that small and high-frequency changes to airflow rates cause little change to the indoor climate. Commercial buildings have been used for demand response programs (see for example [5], [6], [7]), which typically involve reduction of peak power in emergency situations. But providing ancillary service that involve both increase and decrease of power is not common.

The use of commercial building HVAC systems for providing high frequency ancillary services was examined in [8]. In particular, it was shown that the fans in commercial buildings that have variable air volume HVAC systems in the U.S. can provide 70% of the regulation reserves in the frequency range $f \in [1/(3\text{ min})\;\; 1/(8\text{ sec})]$.

In this paper, we extend the time scale of ancillary service from commercial building HVAC system to the range 3 minutes to an hour by using the flexibility in the power demand from chillers. While [8] considers fans as the only source of flexible demand in commercial buildings, chillers are a much larger consumer of electricity and are therefore a natural choice as a load that can provide ancillary service. We propose a control architecture in which the building receives a regulation reference signal from a balancing authority. The regulation controller in the building then changes the rate of airflow through the building from the baseline, so that the deviation of the cooling power consumed by the chiller from the baseline tracks the regulation reference. "Baseline" refers to the counterfactual - the value of a variable due to the actions of the closed loop control system that operates the HVAC system, in the absence of the regulation controller.

A key requirement is that the resulting change in the indoor climate from the baseline cannot be large. This puts a limitation on the frequency band at which ancillary service can be provided. If rate of airflow is changed for a long time – even if the change is small in magnitude – the indoor temperature will vary significantly. Additionally, such a change will instigate the existing climate controller that operates the HVAC system to reject the changes made by the regulation controller: the commands of the regulation controller is seen as a disturbance. On the other hand, if the variation in the regulation reference is extremely fast, it is not likely to be able to achieve the desired variation in power consumption since the dynamics of the HVAC system is slower. By bandpass filtering the regulation command, we ensure that such a situation is not encountered. Simulations with a calibrated dynamic model of a building HVAC system indicates that the frequency range $\mathscr{F}$ in which the proposed controller can provide ancillary service is $\mathscr{F} \triangleq [f_1,\; f_2]$, where $f_2 \leq 1/(3\,\text{min})$ and $f_1 \geq 1/(60\,\text{min})$. This range crosses both secondary control (1-10 min) and tertiary control (10 min - hours) [9].

There are a few challenges in designing control algorithms for harnessing ancillary service from commercial building

HVAC systems, especially in the time scales we are considering. The first challenge is that the relevant dynamics such as of the cooling coil and air flow through ducts are complex. Lack of good model is a hurdle for controller design. Tashtoush et al. [10] and Huang et al. [11] each propose a model for VAV HVAC system. However, there are large number of parameters that are hard to obtain accurately, also model validation is not presented in their work. In this paper, we develop simplified model for each component in a VAV HVAC system and then integrate them together. The model is calibrated and validated with field data collected from Pugh Hall in the University of Florida campus.

The second challenge is that there is a transport delay between the change in air flow and the change in power consumption in the chiller, due to the time required for the chilled water to flow from the cooling coil back to the chiller. To achieve reference tracking despite the time delay, a Kalman predictor is used to predict the future reference and a Smith predictor is used to address closed loop stability issues. The controller is design on a linearized version of the plant, which is a complex hybrid nonlinear system.

Simulations show that the proposed control architecture provides high-quality ancillary service according to the criteria established by PJM [12] while having little impact on indoor climate. Parametric variation studies show that the controller is also robust to potential mismatch between the true value of the transport delay and that used in the design.

The rest of the paper is organized as follows. The overall control architecture is presented in Section II. For the purpose of controller design, we need dynamic models of the components involved; which are described in Section III. Controller design is explained in Section IV. The performance of the controller is then tested through simulations on the original non-linear model of the system. Results are described in Section V.

II. Proposed control architecture

A schematic of a typical single-zone variable air volume (VAV) HVAC system in commercial building is shown in Figure 1. Part of the return air is mixed with outdoor air and

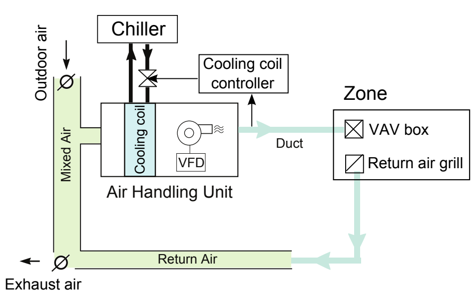

Fig. 1. A typical commercial building VAV HVAC system with a single zone.

sent into the AHU (Air Handling Unit), where it is cooled and dehumidified. The conditioned air is then supplied to the zone by a supply air fan. A control system maintains the discharge air temperature at a pre-specified set-point, usually 55^oF, by varying the flow rate of chilled water passing through the cooling coil. The inlet temperature of the chilled water into the cooling coil is usually constant, at around 44^oF. An indoor climate controller varies the rate of supply airflow to maintain the temperature of the space at a pre-determined set-point. The power consumption of the chiller is directly affected by variation in the airflow rate since conditioning more air requires more cooling energy.

Our objective is to use chillers and fans to vary their instantaneous power consumption to provide ancillary service. In the proposed control architecture, a *regulation reference signal*, denoted by $\tilde{P}_r$, will be transmitted to each participating building. This reference signal has to be of appropriate magnitude and frequency for the capacity of individual participant. In this paper, we take the area control error (ACE), which indicates the imbalance in the grid that needs to be fixed [9], scale it down by a scaling factor and feed it through a bandpass filter to get the proper $\tilde{P}_r$. A local controller at a building - that we call *the regulation controller* - will manipulate the supply air flow rate in the building so that the deviation of the instantaneous power consumption from the baseline power tracks the regulation reference signal. The reasons for choosing the flow rate of air as the control command are that this variable has a large influence on the power consumption, and it can be easily commanded using the building automation system.

Figure 2 shows a schematic representation of the signal flow in the proposed control architecture. It should be emphasized that the proposed architecture does not replace the existing building climate controller. It merely modifies the commanded rate of airflow. The desired power consumption

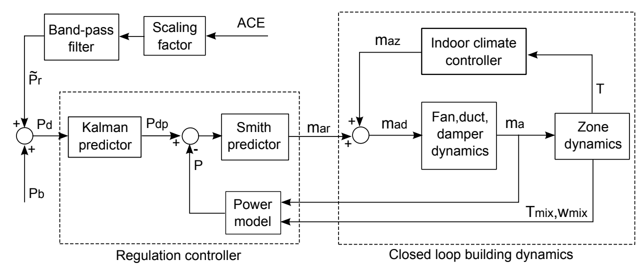

Fig. 2. Proposed control architecture.

$P_d(t)$ is the sum of reference signal $\tilde{P}_r(t)$ and baseline power $P_b(t)$. It is fed into a Kalman predictor to predict the future reference signal $P_{dp}(t) = \hat{P}_d(t+\tau)$, where τ is the transport delay in the chiller power. More detail can be found in Section IV. Let $\tilde{P}(t) = P(t) - P_b(t)$ be the deviation of the measured power consumption from the baseline. The goal of our regulation controller is to compute the desired additional supply air flow rate $m_{ar}(t)$ which drives $\tilde{P}(t) - \tilde{P}_r(t)$ to 0. The building's existing indoor climate controller computes the desired supply air flow rate $m_{az}(t)$ based on zone temperature. The sum of $m_{ar}(t)$ and $m_{az}(t)$, denoted by $m_{ad}(t)$, is the desired supply flow rate, which is commanded though the BAS. The building's HVAC control system commands the fan and dampers to produce this air flow rate. The actual airflow flow rate $m_a(t)$ is the output of a closed loop control system that depends on the dynamics of the fan controller, damper, and airflow in ducts. The exhaust air from the zone will be mixed with outdoor air in a mixing box. The mixed air temperature and humidity ratio are denoted by $T_{mix}(t)$ and $w_{mix}(t)$. The mixed air temperature, humidity, and mass flow rate $m_a(t)$ determine power consumed by the chiller $P(t)$.

III. BUILDING HVAC SYSTEM MODELING

This section presents the dynamic model for each part of the HVAC system shown in Figure 1. Parameter estimation will be discussed in section V-A.

A. Zone thermal dynamics

Zone temperature and humidity, the two main state variables, are affect by outdoor temperature, solar and internal heat gains, and the heat/moisture exchanged with supply air from HVAC system. Temperature dynamics can be captured by a resistor-capacitor (RC) analogy [**?**], [**?**]. In this paper, we adopt the $2R-2C$ model suggested in [13]:

$$\begin{aligned} C_r\dot{T} &= \frac{1}{R_w}(T_w - T) + Q_{AC} + Q_s + Q_i \\ C_w\dot{T}_w &= \frac{1}{R_w}(T - T_w) + \frac{1}{R_w}(T_o - T_w) \end{aligned} \tag{1}$$

where T is the zone temperature, T_w is the wall temperature, T_o is the outdoor temperature, R,C's are the resistances and capacitances of the walls, Q_s the solar heat gain, and Q_i is the internal heat gain. The term Q_{AC} is the heat exchange due to HVAC system that can be modeled as:

$$Q_{AC} = m_aC_a(T_s - T) + m_ah_{we}(w_s - w)$$

where $ma_,T_s,w_s$ are the flow rate, temperature and humidity ratio of supply air, while C_a,h_{we} are constants. Dynamics of the zone humidity ratio is one of mass transfer, and is governed by a first order differential equation [14].

Certain amount of outdoor air is mixed with exhaust air recirculated from the zone in a mixing box before entering the AHU to maintain indoor air quality. Let α be the ratio of outdoor air flow rate to the total supply air flow rate, then we have $T_{mix} = \alpha T_o + (1-\alpha)T$ and $w_{mix} = \alpha w_o + (1-\alpha)w$. These equations together define the dynamic model of the zone, a system of coupled non-linear differential equations with 3 states, 8 inputs $(m_a, w_s, T_s, Q_s, Q_i, Q_{rh}, T_o, w_o)$, and 4 outputs (T, w, T_{mix}, w_{mix}).

B. Indoor climate controller and airflow dynamics

The so-called single-maximum logic is commonly used in commercial buildings to command the airflow so as to maintain the indoor temperature at a pre-determined set point and ensure adequate ventilation. It is a hybrid control logic that includes if-else conditions that determines control "modes" (i.e., when to blow cold air and when to reheat), along with proportional controllers that decide on the amount of airflow in each mode. Due to lack of space, we refer the interested reader to [15], [16] for details.

Once the climate controller computes the desired supply air flow rate, it is transmitted to the fan controller. The fan controller varies the fan speed to deliver the desired air flow rate. We model the closed loop system from the desired supply flow rate m_a^{ref} (input) to actual supply air flow rate m_a (output) to be a first order system, i.e.,

$$m_a(s) = \frac{1/\tau_f}{s + 1/\tau_f} m_a^{ref}(s) \tag{2}$$

where τ_f is the time constant of the system. This time constant aggregates the dynamic effect of the inertia of the fan and dynamics of airflow through ducts.

C. Cooling coil dynamics

Heat and moisture is removed from air at the cooling coil at the AHU. The dynamics of a cooling and dehumidifying coil are complex with many unknown parameters [17], [18]. In this paper, we adopt the idea of adding a time constant to a steady state model – as done in [19] – to get a first order dynamical model for the cooling coil. We use the subscript a for air side, w for water side, 1 for inlet conditions, and 2 for outlet conditions. The inlet and outlet water and air mass flow rate are the same, i.e., $m_{w1} = m_{w2} = m_w$. The inlet and outlet air mass flow rates are also assumed to be equal since the difference due to water vapor condensation is small, i.e., $m_{a1} = m_{a2} = m_a$. The inputs of the cooling coil are the inlet air and water conditions: $u_{cc} = [T_{a1}, w_{a1}, m_a, T_{w1}, m_w]^T$, outputs are the outlet air and water conditions: $y_{cc} = [T_{a2}, w_{a2}, T_{w2}]^T$. Suppose the steady state input-output relations are given by $y_{cc} = g(u_{cc})$, $g : \mathbb{R}^5 \to \mathbb{R}^3$, which is determined by the design parameters of the cooling coil. We then linearize it around the design conditions, which are denoted by u_{cc}^* and y_{cc}^*. By defining $\tilde{u}_{cc} = u_{cc} - u_{cc}^*$ and $\tilde{y}_{cc} = y_{cc} - y_{cc}^*$, we get

$$\tilde{y}_{cc} \approx J\tilde{u}_{cc},\ J = \frac{\partial g}{\partial u_{cc}}\,|_{u_{cc}^*} \tag{3}$$

Adding a single time constant to the steady state model (3), the cooling coil dynamics can be written as:

$$\tilde{y}_{cc}(s) = \frac{1/\tau_{cc}}{s + 1/\tau_{cc}} J\tilde{u}_{cc}(s) \tag{4}$$

where τ_{cc} is the time constant of the open-loop cooling coil dynamics. Note that the Jacobian J defines the DC gains of the transfer functions between $\tilde{u}_{cc}$ and $\tilde{y}_{cc}$.

In practice, the cooling coil is under closed loop operation; see Figure 1. The closed loop cooling coil model is obtained by using a PID controller which commands the chilled water flow rate to achieve desired conditioned air temperature. The closed loop cooling coil model is an LTI system with 3 states, 5 inputs and 3 outputs.

D. Power consumption

We confine ourselves to systems where reheat is powered by steam, so that it does not contribute to the electric power. The total power consumed $P(t)$ is the sum of fan power and chiller power: $P(t) = P_f(t) + P_c(t)$. The fan power is related to mass flow rate of air as $P_f = c_f m_a^3$, where c_f is a constant coefficient which can be estimated from data [8].

The cooling and dehumidification of air occurs at the cooling coil, where the chilled water gains heat $Q(t)$ from the air: $Q(t) = m_w(t)C_{pw}(T_{w2}(t) - T_{w1}(t))$. The return water is cooled in the chiller where power is consumed. Due to the transport delay caused by the speed of water flow from the cooling coil to the chiller, which may be located far from the air handling unit, the power consumed by the chiller is

$$P_c(t) = \frac{1}{\eta_C} Q_c(t - t_d)$$

where t_d is the delay and η_C is the chiller efficiency.

IV. Control design for ancillary service

The two major challenges in this regulation controller design are: (i) complex nonlinear hybrid dynamics of the HVAC system; (ii) transport delay in chiller power. We will linearize the system in design phase to tackle (i), and use Smith predictor [20] and Kalman predictor [21] to deal with (ii) so that the controller can be designed based on non-delayed system.

Consider the delay free case: $P_{nd}(t) = P_f(t) + Q_c(t)$. We combine dynamics of all the components of the HAVC system with m_{ad} as input and P_{nd} as the output. An equilibrium point (x^*, m^*_{ad}, w^*) is chosen, where m^*_{ad} is the nominal mass flow that is observed in normal operation and w^* is the nominal value of all external signals including zone temperature set points, outside weather conditions, and etc. The LTI approximation is then obtained by linearization around this equilibrium point:

$$\begin{aligned} \delta\dot{x} &= A\delta x + B\delta m_{ad} + E\delta w \\ \delta P_{nd} &= C\delta x + D\delta m_{ad} \end{aligned} \tag{5}$$

where $\delta x = x - x^*, \delta m_{ad} = m_{ad} - m^*_{ad}, \delta w = w - w^*, \delta P_{nd} = P_{nd} - P^*_{nd}$, and P^*_{nd} is the equilibrium power consumption when $m_{ad}(t) \equiv m^*_{ad}, w(t) \equiv w^*$. The regulation controller is then designed as a compensator so that the closed loop sensitivity function $S(j\omega)$ is close to 1 in the frequency of interest, and small otherwise, so that both reference tracking and disturbance rejection can be achieved. Note that ambient environment, solar heat gain, and internal heat gain are all uncontrollable inputs, which can be viewed as disturbance, so it is important for the controller to be robust to disturbances.

Now we will address the transport delay discussed in III-D. The delay between mass flow rate change and chiller power consumption can be estimated from the flow rate of chilled water and the geometry and length of the pipe, which makes Smith predictor applicable. However, the smith predictor does not achieve real time reference tracking since the delay remains in the path from the reference to the output. To be able to get reference tracking, we use a Kalman predictor to predict the reference signal $\hat{\tau}$ time units into the future, where $\hat{\tau}$ is the estimated delay in the plant, and this predicted reference will be used by the regulation controller.

The Kalman predictor uses a double integrator model of the process, with the first state being the reference signal, and the output being the reference signal corrupted by noise. The idea behind the model is that since the reference signal is smooth, it changes at an approximately constant rate in short time intervals. The continuous dynamics is first discretized, a standard Kalman predictor is then used to calculate the reference signal n -steps into the future [21]:

$$\hat{P}_d(k+n) = C_o A^n \hat{x}(k \mid k) \tag{6}$$

where $C_o = [1, 0]$ is the output matrix, $\hat{x}(k \mid k)$ is the state estimated at time k by the Kalman filter.

The accuracy of prediction depends on the bandwidth of the input and the delay. The reference signal $\tilde{P}_r$ is assumed to have frequency range $\mathscr{F}$ in this part, since we envision it to be the general case. Assuming on-site chiller is used, the delay is estimated to be $30s$ for Pugh Hall. We ran the simulation with different delays, and it turns out the prediction error is reasonable up to $90s$ of delay. In reality, accurate knowledge of the delay may not be available. We study the effect of this uncertainty on prediction accuracy by performing simulations in which the true delay is $30s$ but the Kalman predictor uses a delay estimate of $20s$ and $40s$, respectively. The results are shown in Figure 3, where the *error ratio* in the figure is defined as the ratio of prediction error to root mean square of the reference signal. The result shows that delay mismatch increases the prediction error, as expected, but not by a lot. Although the error appears to be large at some

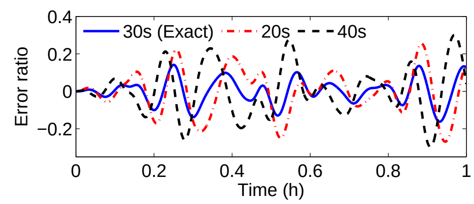

Fig. 3. Comparison of prediction errors of regulation reference signal when there is a mismatch between the true delay and delay used in controller design.

instances, it occurs when the magnitude of the reference signal is small. The effect of this error on reference tracking is further discussed in Section V-C, which shows that the resulting error in reference prediction is acceptable.

V. Simulation study

A. Simulation setup

The subsystems described in III are integrated together and implemented in *Simulink*. Field data is collected from Pugh Hall on University of Florida campus to estimate parameters in the model. We use data from from AHU-2 in the building, which is used as a dedicated AHU for a large auditorium that is $22ft$. high with floor area $6000ft^2$, and can hold more than 200 occupants.

Zone parameters are estimated using the method in [13]. The measured zone temperature and the temperature predicted by the model are shown in Figure 4 (left).

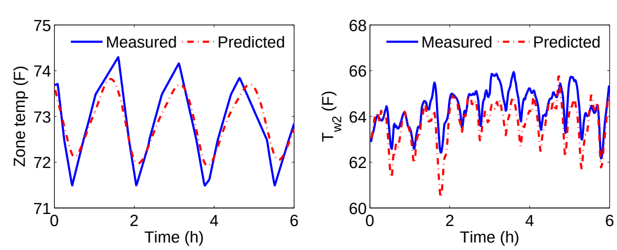

Fig. 4. Zone climate model (left) and cooling coil model (right) validation.

To estimate J in the cooling coil model, we first pick a particular coil model that resembles the coil in AHU2 of Pugh Hall. For a given inlet conditions, the outlet conditions are obtained from *Daikin McQuay Tools Suite* [22]. The Jacobian is then estimated numerically. The outlet conditions predicted by the model and the measured outlet conditions are shown in Figure 4. Due to lack of space, only return chilled water temperature T_{w2} is shown. It can be seen from the figure that our model predicts T_{w2} well with a maximum prediction error less than 2^oF.

Other model parameters are listed in Table I. $m_{a,min}$ is computed from the floor area and occupants of the zone to meet ASHRAE standard for indoor air quality. The velocity of chilled water is computed to be around $1.5m/s$. We assume that the pipe is $45m$ long for a on-site chiller, which gives $\tau = 30s$. The fan constant c_f is taken from [23], where it was calibrated using data from the same building. The other parameters are observed or estimated from real data collected from Pugh Hall. The baseline power P_b is obtained by simulating the system without the frequency regulation controller.

TABLE I. MODEL PARAMETERS

Parameter	Value	Description
C_r	$3.4\times10^7 J/K$	Zone capacitance
C_w	$5.1\times10^7 J/K$	Wall capacitance
R	$1.3\times10^{-3} K/W$	Wall resistance
$m_{a,min}$	$4000CFM$	Minimum supply air flow rate
$m_{a,max}$	$8000CFM$	Maximum supply air flow rate
T_{CLG}	73^oF	Zone cooling set point
T_{HTG}	71^oF	Zone heating set point
α	0.4	Outdoor air ratio
T_{w1}	44^oF	Chilled water inlet temperature
$T_{a2,r}$	55^oF	Discharge air temperature setpoint
τ_f	$30s$	Fan, damper, duct time constant
τ_{cc}	$100s$	Cooling coil time constant
τ	$30s$	Delay in chiller power
c_f	$36.84W/(kg/s)$	Fan power coefficient

Outdoor environment and loads in the zone are the exogenous inputs need to be specified in the simulation. Hourly outdoor temperature T_o and humidity w_o information is taken from the website *www.wunderground.com* for Gainesville, FL on the same day as the ACE signal (2009-05-04). The solar load Q_s are computed based on the location and orientation of the building and the area and the shading coefficient of the windows, using the method described in [24]. The internal load Q_i is generated from normal schedule of occupants and equipments.

B. Performance metrics

Performance of the control architecture depends on (i) how much ancillary service is provided through regulation reference tracking, and (ii) how much deviation of the indoor climate from the baseline conditions occur as a result of the controller's actions.

Measuring regulation reference tracking is somewhat involved because of the way ancillary service is evaluated by ISOs. Traditionally, once certified, the frequency regulation service providers are usually compensated by capacity, not performance. However, this is unfair to those who provide faster or more accurate response. FERC order No.755 [25] stressed this problem and asked RTOs and ISOs to design performance-based compensation in their tariff. In this paper, we will use the performance score defined by PJM [12].

The score contains three parts: S_c - the correlation score, S_d - the delay score, and S_p - the precision score. S_c and S_d are used to quantify the delay between the regulation signal and the response of the resource. Define the correlation coefficient to be:

$$R_P(\tau) = \frac{cov(\tilde{P}_r(t), \tilde{P}(t+\tau))}{\sigma_{\tilde{P}_r(t)}\sigma_{\tilde{P}(t+\tau)}} \tag{7}$$

where σ is the standard deviation of the signal. The parameter τ^* is defined as the time shift with which the response has the highest correlation with the reference signal:

$$\tau^* = \underset{\tau\in[0,5mins]}{\arg\max}\ R_P(\tau) \tag{8}$$

The scores S_c and S_d are then determined as:

$$S_c = R_P(\tau^*),\ S_d = \left|\frac{\tau^* - 5mins}{5mins}\right| \tag{9}$$

The precision score S_p is defined as:

$$S_p = 1 - \frac{1}{n}\sum_{i=1}^{n}\frac{|P(i) - P_r(i)|}{|P_{r,a}|} \tag{10}$$

where $P_{r,a}$ is the hourly average of the reference signal, n is the number of samples. The total performance score S_t is the average of the three parts, i.e., $S_t = \frac{1}{3}S_c + \frac{1}{3}S_d + \frac{1}{3}S_p$.

C. Results

ACE data from PJM is used as the regulation signal. The scaling factor need to be determined first. If the scaling factor is too large, the supply air flow rate has large oscillation, which is undesirable. First, it will violate the outdoor air requirement for indoor air quality when the supply flow rate becomes too low. Second, the oscillation increases wear and tear of the equipments. We evaluate the oscillation by comparing the variation from the baseline supply flow rate. More precisely,

$$v = \frac{1}{n}\sum_{i=1}^{n}\frac{|m_a(i) - m_{a,b}(i)|}{m_{a,b}(i)} \tag{11}$$

where m_a is the supply flow rate with the regulation controller, $m_{a,b}$ it the supply air flow rate of the baseline case. We chose the scaling factor to be 4×10^{-5}, in which case $v = 15\%$.

The reference tracking result is shown in Figure 5. In the figure, two simulations with difference time scale in reference signal are shown: 3 to 30 minutes for Case 1 and 3 to 60 minutes for Case 2. This is achieved by changing the bandpass filter for the ACE. From the figure, we see that in both cases we are able to track the reference signal with maximum power of about $20KW$. The temperature deviation from baseline case is larger in Case 2 than in Case 1, but both are less than 1^oF. The maximum temperature deviation ΔT_{max} (^oF) is shown in Table II. The performance score is evaluated, see also Table II. The performance score is computed for each hour in a 12-hour duration, and then averaged. PJM require the provider to reach a score of 0.75 to be qualified for frequency regulation market. Our controller performs well above the requirement.

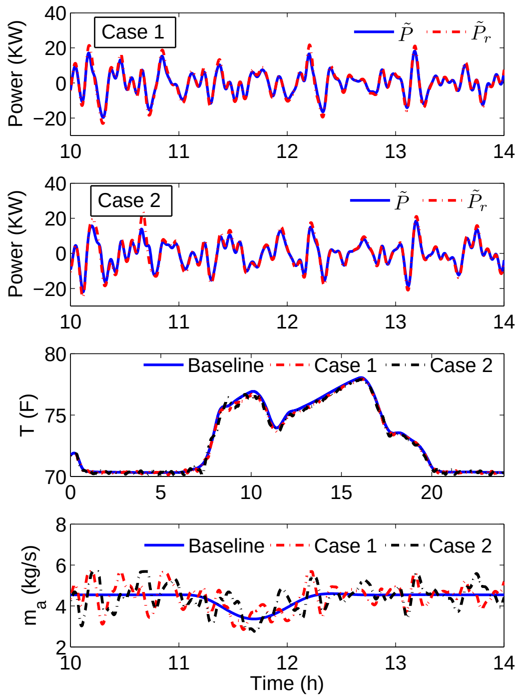

Fig. 5. Performance of the regulation controller, and comparison with baseline.

TABLE II. PERFORMANCE WITH REFERENCE SIGNALS OF DIFFERENT FREQUENCY.

Time scale (min)	S_c	S_d	S_p	S_t	ΔT_{max} (^oF)
1 to 3	0.67	0.85	-0.95	0.50	0.24
3 to 10	0.96	1	0.82	0.93	0.10
10 to 30	0.96	1	0.81	0.92	0.44
30 to 60	0.95	0.95	0.80	0.90	1.69
3 to 30 (Case 1)	0.96	1	0.81	0.92	0.45
3 to 60 (Case 2)	0.96	0.98	0.79	0.91	0.87

We also tested our algorithm with reference signals of different frequency ranges by varying the passband $\mathscr{F}$ of the bandpass filter. The simulation result is shown in Table II. The controller does not work well for high frequency range (1 to 3 minutes). Since the time constant of the chiller falls in that range, it cannot react fast enough to track the reference signal. The tracking performance of the middle and low frequency ranges are both good. However, the zone temperature variation from the baseline case grows as the reference signal becomes slower. That is because we are trying to over-condition or under-condition the zone for longer times, which drives the temperature away from the desired value.

Effect of chiller power delay mismatch on tracking performance is also studied in simulation. The true delay of the system is $30s$. Table III shows the performance of the controller when it is designed assuming the delay is $15s$ and $45s$, respectively. As shown in the table, in both cases, the effect of chiller delay mismatch on the delay score S_d is small. In the $45s$ case, the precision score S_p is worse compared to the case when accurate delay knowledge is used. However, it is still acceptable. We also observed that larger delay mismatch may drive the system unstable. Thus, with this given model and reference signal, our control strategy is able to handle 50% of delay mismatch in both directions.

TABLE III. EFFECT OF DELAY MISMATCH IN CHILLER POWER.

True Delay (s)	Delay in design (s)	S_c	S_d	S_p	S_t
30	15	0.95	1	0.84	0.93
30	45	0.97	0.99	0.79	0.92

The simulation results show that AHU2 in Pugh Hall, which has a rated cooling capacity of 97.5 kW, could provide 20; KW of ancillary service. The total cooling capacity of Pugh Hall (40,000 ft^2) that has 2 other AHUs is estimated to be 100 kW. In the U.S., the total floor area of commercial buildings is about 72,000 million square feet, about 30% of which is served by VAV systems [4]. Assuming that the cooling power density (kW per sq. ft.) is similar among these buildings, the commercial building sector could provide 12GW of regulation service, which is more than the total regulation capacity required in the U.S. (about 10GW) [26].

VI. CONCLUSION AND FUTURE WORK

High thermal inertia of commercial buildings allow small variations in the airflow to have negligible effect on indoor temperature as long as the variation in airflow rate is fast enough. By integrating proper control strategy to existing HVAC system, commercial buildings could provide ancillary service to power grid without affecting the indoor climate significantly. Simulation results presented here shows the possibility of satisfying the entire regulation demand in the U.S from commercial buildings. Compared to the work in [8], we are able to extend the band of regulation signal that can be tracked to $f \in [1/(60\text{ min}), 1/(3\text{ min})]$. We also increase the magnitude of ancillary service that can be provided by utilizing chillers.

In this work, we do not consider the reheat power consumption since reheat is often powered by steam. For the systems with electric reheat, integrating reheat power consumption will be an avenue of future work. In addition, our control strategy only works for on-site chillers whose time delay is relatively small. For off-site chillers with larger delay, better method for predicting reference signal or another way to deal with the delay need to be studied.

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