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Nonlinear Model Predictive Control for Energy Efficient Cooling in Shopping Center HVAC

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***Abstract*—In this paper we present a novel approach to control a shopping center HVAC system which significantly reduces the amount of energy spent on cooling. The HVAC system considered is for a section of a Danish shopping center, including central ventilation, fan coil units and a chiller delivering cooling.**

The system is modeled using a grey-box RC-equivalent approach and identified parameters using measurement data extracted directly from the Building Management System from several days of live operation. From a comparison with measurements it has been concluded that the model is usable for the purpose of control design.

An optimal control problem to minimize total cooling effort by manipulating central ventilation supply temperature and chiller forward temperature has been posed. The intention being to shift cooling from the chiller to the ventilation unit when cooling is available through a low ambient temperature – avoiding both heating and cooling the same air. This optimal control problem has been used as the basis for a Model Predictive Controller. For prediction purposes, input signals from the previous days have been used, exploiting the fairly periodic behaviour of the system.

Simulation studies show that during heating seasons the Model Predictive Controller is capable of shifting the entire cooling load to the ventilation unit and still maintain the same performance as the nominal controller. This amounts to energy savings of $21\,\%$.

I. INTRODUCTION

Buildings are responsible for one third of the total energy consumption in Denmark [1]. Energy refurbishments of older buildings typically consider the building envelope itself which can be an expensive and cumbersome task. Instead, replacing or updating the heating and or cooling equipment can with less effort amount to larger energy savings [2] and can prove to be a more attractive investment in energy renovations from an owner's/operator's point of view.[3, 4]

One approach to reductions in energy consumption for buildings is through control applications. This has been extensively studied with the majority of recent work within predictive control [5, 2]. With many buildings featuring multi-zone characteristics and with the inherent scale of some commercial systems, both distributed [4], decentralized [3] and hierarchical [6] solutions have previously been investigated. Considering implementation costs, a centralized solution may still be more attractive for the owners/operators, which is a necessity for wider adoption and hence energy savings on a larger scale. In [7], a novel central Nonlinear Model Predictive Control (NMPC) is designed and implemented for a Constant Air Volume (CAV) Heating Ventilation and Air Conditioning (HVAC) system with large energy savings.

This paper presents work that is part of a project named Smart Energy Shopping Centers (SEBUT)[8] . SEBUT aims at developing control systems, knowledge services and tools for energy refurbishments of shopping centers in Denmark. SEBUT takes a holistic approach to both energy efficiency and flexibility [9]; touching upon indoor air quality, advanced control of indoor climate and lighting, energy storage, user requirements, behavior and potential barriers.[8]

The work presented in this paper is a continuation of the work done in [10], in which we presented a control-oriented multi-zone model suitable for modeling the temperature dynamics in a shopping center; the model was then used in a preliminary control design to increase energy efficiency of a chiller supplying shop-level cooling. In this paper we focus on the same system but extend it to include the central ventilation unit – this adds to the complexity of producing the required cooling capacity as efficiently as possible, given that both the ventilation unit and chiller can produce and deliver cooling. This complexity is not handled in the current control configuration, which amounts to energy wasted through a lack of coordination between production and consumption. We consider how we can manage the complexity through a control design that seeks to efficiently meet the shop-level cooling demands.

In **Section II**, the shopping center and HVAC system in question is accounted for, together with issues in the current control solution. Following, in **Section III**, the model equations are introduced together with parameterization. **Section IV** describes the control design with simulation results presented in **Section V**. Conclusions are given in **Section VI**.

Notation-wise, matrices are denoted in uppercase bold, e.g. $\mathbf{A}$, vectors are denoted in lowercase bold, e.g. $\mathbf{x}$. Dependence of variables on time t, $\mathbf{x}(t)$, is implied and will not necessarily be written explicitly.

II. SYSTEM OVERVIEW

In this paper we consider a typical HVAC layout exemplified by *Kolding Storcenter*, a mall in Denmark. The shops in Kolding Storcenter are ventilated using a CAV scheme, featuring fan coil units that enables heating and cooling at shop-level; allowing for local control of the supply temperature to each shop. Each shop is, depending on size, outfitted with several fan coil units – they are however controlled as a single unit. The shops are divided into clusters, where each cluster

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is supplied with ventilated air from a Central Ventilation Unit (CVU) and chilled water for shop-level cooling from a Central Cooling Unit (CCU). For heating, hot water from District Heating (DH) is supplied to all fan coils. This general HVAC layout is depicted in **Figure 2**. In Kolding Storcenter, a demo-area has been established for the SEBUT project. The demo-area consists of three shops supplied by the same CVU and the same CCU.

Each fan coil has two heat exchangers, one for cold and one for hot water, to which the flow is controllable with motor-controlled valves. Each shop features a temperature controller; manipulating valve openings to control shop supply temperature (T_{supply}) – the shop-local control is depicted in **Figure 1**. Reference signals for shop-local control are given by the Building Managment System (BMS); in which the central HVAC control is implemented.

The CCU, a heat pump, is controlled independently of the cooling requirements of the shops; the forward temperature ($T_{\mathrm{fwd,cool}}$) is typically kept constant around $10\,°\mathrm{C}$.

The CVU delivers ventilated air at a controlled supply temperature (T_{vent}) and an almost constant flow ($\dot{m}_{\mathrm{vent}}$). There is no recirculation in the CVU; air is drawn in at ambient temperature (T_{amb}) and then either cooled or heated (using its own heat pump, not the CCU), depending on the setpoint for the supply temperature. The setpoint for supply temperature is determined by a controller acting on extract temperature from the shops.

One issue with the current control architecture lies in the lack of coordination between the shop-local temperature control, the control of the CVU and the CCU. This lack of coordination shows as cases where energy is spent on e.g. heating air up in the CVU and then more energy is spent on cooling it down again in some of the fan coils. This specific issue is depicted in **Figure 3**.

In order to obtain measurements and manipulate with the HVAC system we 'piggyback' on the central control by interfacing with the existing BMS network through the use of a *gateway unit* [11]. This provides us with the same measurements and ways of actuation as the BMS. The gateway unit features an Internet connection, allowing new control algorithms to run on a device/platform that is not physically in the mall in question; e.g. in a cloud-environment.

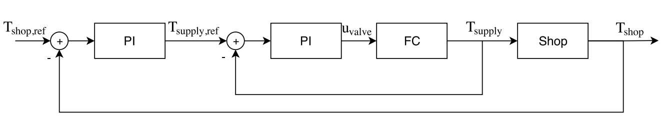

Fig. 1: The shop-local temperature control is implemented as two PI regulators in a cascade configuration. The FC block is a fan coil unit.

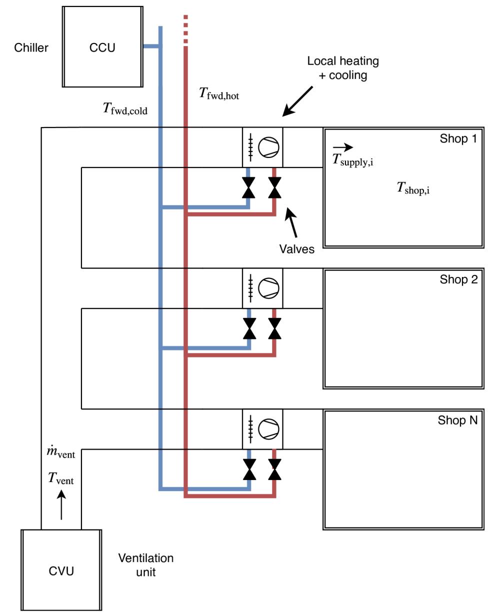

Fig. 2: The HVAC layout of Kolding Storcenter; depicting a cluster of shops supplied by shared central ventilation; CVU. A central chiller, CCU, supplies chilled water to each fan coil, for local cooling. Heating is through DH. No return pipes/ducts are depicted.

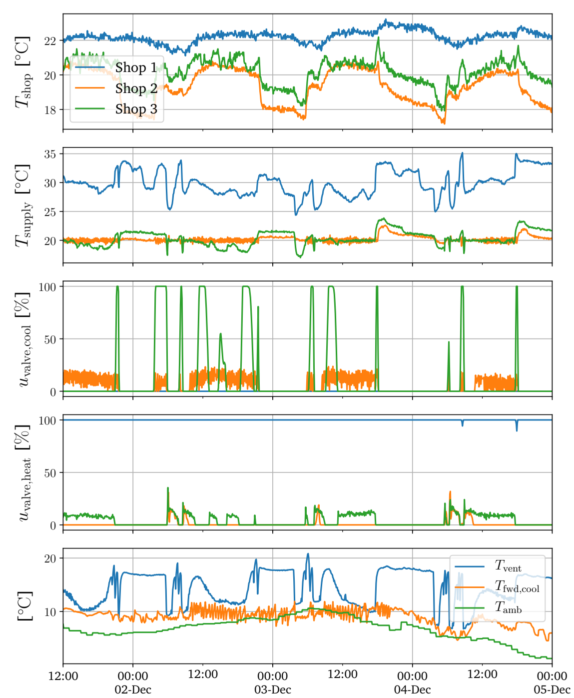

Fig. 3: Measurement data extracted from the BMS. The three shops in the demo-area behave differently; Shop 1 features large heating demand (heating valve saturating at $100\,\%$ all the time). Shop 2 and Shop 3 utilize both cooling and heating. $T_{\mathrm{vent}} > T_{\mathrm{amb}}$ at certain times through operational hours due to heating in the CVU.

III. MODEL

The model presented in this section builds upon the work in [10], where we employ a grey-box RC-equivalent modeling paradigm, treating each shop as a thermal zone with a lumped thermal capacitance.

A. Temperature dynamics

With N being the number of shops considered, the temperature dynamics of the i-th shop is given by:

$$C_{\mathrm{shop,i}}\,\dot{T}_{\mathrm{shop,i}} = \dot{Q}_{\mathrm{FC,i}} + \dot{Q}_{\mathrm{center,i}} + \dot{Q}_{\mathrm{int,i}} \tag{1}$$

where $C_{\mathrm{shop,i}}$ is the lumped thermal capacitance of shop i, $T_{\mathrm{shop,i}}$ is the shop temperature and $\dot{Q}_{\mathrm{center,i}}$ is the heat flow to/from the surroundings. $\dot{Q}_{\mathrm{FC,i}}$ is the heat flow supplied by fan coils and $\dot{Q}_{\mathrm{int,i}}$ models the internal heat gain, e.g. heat gain from occupancy, lighting and appliances. No heat exchange between the shops is considered as $\dot{Q}_{\mathrm{int,i}}$ dominates the energy balance, given the amount of display lighting. Also, no heat gain from solar load is included, as the shops are not exposed to direct sunlight.

The supply temperature dynamics for the i-th shop is given by:

$$\begin{aligned} C_{\mathrm{supply,i}}\,\dot{T}_{\mathrm{supply,i}} &= \dot{Q}_{\mathrm{vent,i}} + \dot{Q}_{\mathrm{recirc,i}} + \qquad (2)\\ &\quad \dot{Q}_{\mathrm{cool,i}} + \dot{Q}_{\mathrm{heat,i}} - \dot{Q}_{\mathrm{FC,i}} \qquad (3) \end{aligned}$$

where $C_{\mathrm{supply,i}}$ is the lumped thermal capacitance for the fan coils and $T_{\mathrm{supply,i}}$ is the temperature of the supply air to the shop. $\dot{Q}_{\mathrm{cool,i}}$ is heat flow from central cooling, $\dot{Q}_{\mathrm{heat,i}}$ is heat flow from heating and $\dot{Q}_{\mathrm{vent,i}}$ is heat flow from central ventilation. Also, for the fan coils, some air is recirculated from the shops giving the heat flow $\dot{Q}_{\mathrm{recirc,i}}$.

Collected as matrix/vector expressions, we consider the temperature dynamics in the following form:

$$\begin{aligned} \mathbf{C}_{\mathrm{shop}}\,\dot{\mathbf{T}}_{\mathrm{shop}} &= \dot{\mathbf{Q}}_{\mathrm{FC}} + \dot{\mathbf{Q}}_{\mathrm{center}} + \dot{\mathbf{Q}}_{\mathrm{int}} \qquad (4)\\ \mathbf{C}_{\mathrm{supply}}\,\dot{\mathbf{T}}_{\mathrm{supply}} &= \dot{\mathbf{Q}}_{\mathrm{vent}} + \dot{\mathbf{Q}}_{\mathrm{recirc}} + \qquad (5)\\ &\quad \dot{\mathbf{Q}}_{\mathrm{cool}} + \dot{\mathbf{Q}}_{\mathrm{heat}} - \dot{\mathbf{Q}}_{\mathrm{FC}} \end{aligned}$$

with all vectors belonging to $\mathbb{R}^N$ and the $\mathbf{C}$-matrices being square and invertible.

B. Heat flows

The heat flow supplied by the fan coils is given by:

$$\dot{\mathbf{Q}}_{\mathrm{FC}} = \dot{\mathbf{M}}_{\mathrm{FC}}\, c_{\mathrm{p,air}}\,(\mathbf{T}_{\mathrm{supply}} - \mathbf{T}_{\mathrm{shop}}) \tag{6}$$

where $\dot{\mathbf{M}}_{\mathrm{FC}} \in \mathbb{R}^{N\times N}$ is a diagonal matrix with $\dot{m}_{\mathrm{FC,i}}$ – the air flow to the i-th shop – in the diagonal. The specific heat capacity is denoted $c_{\mathrm{p,air}}$.

Each fan coil supplies air at a rate of μ kg/s constantly during operation (CAV), where β is from ventilation and $(1-\beta)$ is recirculated. Scaling with number of fan coils in a given shop:

$$\dot{\mathbf{M}}_{\mathrm{FC}} = \mu\,\mathbf{N}_{\mathrm{FC}}\ \mathrm{kg/s} \tag{7}$$

where $\mathbf{N}_{\mathrm{FC}} \in \mathbb{R}^{N\times N}$ is a diagonal matrix with number of fan coils, $N_{\mathrm{FC,i}}$, in the diagonal.

Heat flows from CVU and recirculation are given as:

$$\begin{aligned} \dot{\mathbf{Q}}_{\mathrm{vent}} &= \beta\,\dot{\mathbf{M}}_{\mathrm{FC}}\,c_{\mathrm{p,air}}\,(T_{\mathrm{vent}}\,\mathbf{1} - \mathbf{T}_{\mathrm{supply}}) \qquad (8)\\ \dot{\mathbf{Q}}_{\mathrm{recirc}} &= (1-\beta)\,\dot{\mathbf{M}}_{\mathrm{FC}}\,c_{\mathrm{p,air}}\,(\mathbf{T}_{\mathrm{shop}} - \mathbf{T}_{\mathrm{supply}}) \qquad (9) \end{aligned}$$

where $\mathbf{1}$ is a vector of all ones.

The heating/cooling heat flows are given as:

$$\begin{aligned} \dot{\mathbf{Q}}_{\mathrm{cool}} &= \alpha\,\mathbf{u}_{\mathrm{valve,cool}}\,c_{\mathrm{p,water}}\,(T_{\mathrm{fwd,cool}}\mathbf{1} - \mathbf{T}_{\mathrm{supply}}) \qquad (10)\\ \dot{\mathbf{Q}}_{\mathrm{heat}} &= \alpha\,\mathbf{u}_{\mathrm{valve,heat}}\,c_{\mathrm{p,water}}\,(T_{\mathrm{fwd,heat}}\mathbf{1} - \mathbf{T}_{\mathrm{supply}}) \qquad (11) \end{aligned}$$

where α is a combined term for coil efficiency and valve characteristics; it is assumed constant. The valve openings, $\mathbf{u}_{\mathrm{valve}}$, are determined by the governing PI regulators also included in the model.

The CVU is controlled through a setpoint for T_{vent}. This control is modelled with some first order dynamics – equivalently for the CCU:

$$\begin{aligned} \tau_{\mathrm{CVU}}\,\dot{T}_{\mathrm{vent}} &= T_{\mathrm{vent,r}} - T_{\mathrm{vent}} \qquad (12)\\ \tau_{\mathrm{CCU}}\,\dot{T}_{\mathrm{fwd,cool}} &= T_{\mathrm{fwd,cool,r}} - T_{\mathrm{fwd,cool}} \qquad (13) \end{aligned}$$

The total combined cooling capacity of the CVU and the CCU, $\dot{Q}_{\mathrm{cool,tot}}$, is modelled as:

$$\dot{Q}_{\mathrm{cool,CCU}} = \sum_{1}^{N} \dot{Q}_{\mathrm{cool,i}} \tag{14}$$

$$\dot{Q}_{\mathrm{vent,cap}} = \beta \sum_{1}^{N} \dot{m}_{\mathrm{FC,i}}\,c_{\mathrm{p,air}}\,(T_{\mathrm{vent}} - T_{\mathrm{amb}}) \tag{15}$$

$$\dot{Q}_{\mathrm{cool,tot}} = \dot{Q}_{\mathrm{cool,CCU}} + \dot{Q}^{-}_{\mathrm{vent,cap}} \tag{16}$$

where $\dot{Q}^{-}_{\mathrm{vent,cap}}$ is the negative part of $\dot{Q}_{\mathrm{vent,cap}}$, thus only taking cooling into account.

C. Parameterization

Model parameters have been identified using a combination of manual air flow measurements, measurements taken from the BMS and shop dimensions. Steady-state data has been used to determine magnitudes of heat flows and shop dimensions have been used to determine thermal capacitances. Internal heat gains are assumed constant throughout shop opening hours; this assumption is to a large extent valid given that display lighting dominates the term.

Parameters used are given in **Table I** and **Figure 4** compares model simulation with measurements obtained from the BMS in order to validate the use of the model for control purposes.

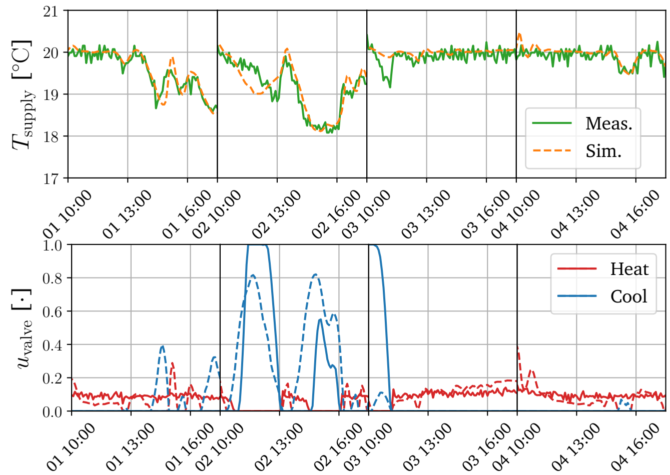

Fig. 4: Comparing simulated response (dashed) with measured (solid) for a single shop during hours where the HVAC was running. The fit is not perfect, but deemed good enough for the basis of a control design. Most notable shortcomings stem from the stochastic nature of $\dot{\mathbf{Q}}_{\text{int}}$.

TABLE I: Selected model parameters

Shop	Area $[\text{m}^2]$	C_{shop} [kJ/K]	C_{supply} [kJ/K]	$\dot{Q}_{\text{int}}$ [kW]
Shop 1	650	7.0×10^3	2.3×10^3	8.0
Shop 2	250	2.7×10^3	0.9×10^3	4.8
Shop 3	250	2.7×10^3	0.9×10^3	3.2
$\alpha = 0.05\,\text{kg/s}$, $\mu = 0.36\,\text{kg/s}$, $\beta = 1/3$				
$T_{\text{fwd,heat}} = 55\,°\text{C}$, $\tau_{\text{CVU}} = 15\,\text{min}$, $\tau_{\text{CCU}} = 5\,\text{min}$				

IV. MINIMIZING ENERGY SPENT ON COOLING THROUGH CONTROL

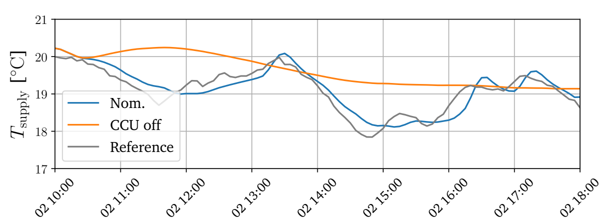

Fig. 5: Simulation where the CCU is turned off (*CCU off*) compared with a simulation with nominal/historical inputs, showing the supply temperature response for *Shop 3*. Turning the CCU off introduces a loss in regulation power for the shop-local supply temperature control, visible as as a degradation in tracking performance.

As described in **Section II**, there are two ways of heating the air and two ways of cooling it; either centrally at the CVU or locally at the fan coils. One issue is the apparent use of the CCU for cooling, even during the heating season. The issue is rooted in the control configuration lacking coordination – but just as much in the large internal heat gains in the shops. One attempt at minimizing energy spent on cooling, would be to simply turn off the CCU during heating season. However, given the current control architecture, this can pose problems with lack of regulation power for the control of shop-local supply temperature. This is exemplified with a simulation where the CCU is turned off. The supply temperature for *Shop 2* is for this simulation depicted in **Figure 5**.

A. Optimal Control Problem

In order to avoid the loss of regulation power by turning off the CCU, we consider Model Predictive Control (MPC) to shift the cooling capacity from the CCU to the CVU instead. This seems sensible, especially during the heating season, as the ambient temperature is typically lower than the desired supply temperature – giving a free source of cooling.

We consider $T_{\text{vent,r}}$ and $T_{\text{fwd,cool,r}}$ as our control inputs – and use historical input data for the exogenous inputs, including references for shop-local supply temperature control. No observer is necessary since measurements are available for all states. The state, control input, exogenous input and output is given as:

$$\mathbf{x} = \left[\mathbf{T}_{\text{shop}}^T, \mathbf{T}_{\text{supply}}^T, T_{\text{vent}}, T_{\text{fwd,cold}}, \mathbf{x}_{\text{aux}}^T\right]^T \tag{17}$$

$$\mathbf{u} = \left[T_{\text{vent,r}}, T_{\text{fwd,cold,r}}\right]^T \tag{18}$$

$$\mathbf{u}_{\text{ex}} = \left[\mathbf{T}_{\text{supply,r}}^T, T_{\text{amb}}\right]^T \tag{19}$$

$$\mathbf{y} = \left[\dot{Q}_{\text{cool,tot}}\right] \tag{20}$$

where $\mathbf{x}_{\text{aux}}$ denotes auxiliary states related to the supply temperature PI regulators. We then formulate our optimal control problem as:

$$\min_{\mathbf{u}} J = \int_{t_0}^{t_f} \mathbf{y}^T \mathbf{y}\, \mathrm{d}t \tag{21}$$

subject to:

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{f}(t, \mathbf{x}(t), \mathbf{u}(t), \mathbf{u}_{\text{ex}}(t)) \quad \text{(dynamics)} \\ \mathbf{y} &= \mathbf{h}(t, \mathbf{x}(t), \mathbf{u}(t), \mathbf{u}_{\text{ex}}(t)) \quad \text{(output)} \end{aligned}$$

and subject to state and input constraints:

$$\begin{aligned} \mathbf{T}_{\text{shop,min}} &\leq \mathbf{T}_{\text{shop}} &&\leq \mathbf{T}_{\text{shop,max}} \\ \mathbf{T}_{\text{supply,min}} &\leq \mathbf{T}_{\text{supply}} &&\leq \mathbf{T}_{\text{supply,max}} \\ T_{\text{vent,r,min}} &\leq T_{\text{vent,r}} &&\leq T_{\text{vent,r,max}} \\ T_{\text{fwd,cool,r,min}} &\leq T_{\text{fwd,cool,r}} &&\leq T_{\text{fwd,cool,r,max}} \end{aligned}$$

where $\leq$ should be taken element-wise in the vector case. This optimal control problem seeks to minimize $\|\dot{Q}_{\text{cool,tot}}\|^2$, which effectively means minimizing the total cooling effort described by (16). Note that the objective function does not directly penalize the control signal. Usually this would let MPC exhibit a 'bang-bang' behavior. In our case, however, the output contains a (practically) static contribution from the control signal via the expression for total cooling effort, as can be seen in (15). This prevents adverse control behavior consistent with the simulation response presented in **Section V**. It should also be noted, that constraints on the relationship between state and input are to be considered, e.g. $T_{\text{fwd,cool,r}} < T_{\text{supply,i}}$ when cooling – however, simulation results without (see **Section V**) give feasible solutions.

V. SIMULATION STUDIES

We have used *CasADi*[12] through Python to model the nonlinear system dynamics and to pose, discretize and solve the optimal control problem using a multiple-shooting approach.

We let the prediction horizon be equal to $t_{\mathrm{predict}} = 5\,\mathrm{h}$ and the sampling time of both measurements and our MPC is fixed to $5\,\mathrm{min}$. Given that we do not know the exogenous inputs $5\,\mathrm{h}$ in advance, we utilize the fact that the system is fairly periodic and employ inputs from the previous day (delayed $24\,\mathrm{h}$) for prediction purposes.

The state and inputs constraints have been set to:

$$\begin{aligned} 17\,^\circ\mathrm{C} \le T_{\mathrm{shop,i}} &\le 25\,^\circ\mathrm{C} \\ 10\,^\circ\mathrm{C} \le T_{\mathrm{supply,i}} &\le 35\,^\circ\mathrm{C} \\ 5\,^\circ\mathrm{C} \le T_{\mathrm{vent,r}} &\le 25\,^\circ\mathrm{C} \\ 5\,^\circ\mathrm{C} \le T_{\mathrm{fwd,cool,r}} &\le 25\,^\circ\mathrm{C} \end{aligned}$$

We have conducted simulation experiments of 4 consecutive days in December 2018. Given the operational hours of the HVAC system in Kolding Storcenter, the simulation has been limited to the hours between 08:00 and 18:00 during these days, for a total simulation time of $40\,\mathrm{h}$. The results are shown in **Figure 6**.

The results show noticeable less degradation of regulation power for the shop-local supply temperature control (for *Shop 2*), as compared to **Figure 5** where the CCU was simply turned off. This is to a large extent achieved by letting the CVU run with a supply temperature closer to the ambient temperature, hereby delivering more base cooling to the fan coils. This lowers the need to actuate the valves for cooling from the CCU. Also, $T_{\mathrm{fwd,cool}}$ is set significantly higher than for nominal control, which decreases cooling when exercising the cooling valves. From both input signals, it is possible to see the correlation with the previous day's cooling load by comparing with the shown T_{supply}.

Looking at the response for $\dot{Q}_{\mathrm{cool,CCU}}$ it is very clear that the reduction in cooling supplied by the CCU when using MPC is equivalent to simply turning off the CCU – as desired. This does not necessarily mean that net energy spent is lower, however, as it could simply be that energy spent on heating is equally higher.

Therefore we investigate the sum of heat flows responsible for both active heating and active cooling in the fan coils:

$$\dot{Q}_{\mathrm{tot,cap}} = \dot{Q}_{\mathrm{vent,cap}} + \dot{Q}_{\mathrm{cool,CCU}} + \sum_{1}^{N} \dot{Q}_{\mathrm{heat,i}} \qquad (22)$$

From **Figure 6** it is shown that $\dot{Q}_{\mathrm{tot,cap}}$ is generally lower when using the designed MPC than when using the nominal control; we calculate the difference in energy consumption:

$$\begin{aligned} E_{\mathrm{saved}} &= \int_{0}^{40\,\mathrm{h}} \dot{Q}_{\mathrm{tot,cap,nom}} - \dot{Q}_{\mathrm{tot,cap,MPC}}\,\mathrm{d}t \qquad (23) \\ &\approx 230\,\mathrm{kW\,h} \end{aligned}$$

which is equivalent to a $21\,\%$ decrease.

VI. CONCLUSIONS

This paper has through simulation studies demonstrated a control design which effectively minimizes energy spent on cooling during heating season, in the HVAC system of a Danish shopping center. A problem that stems from the control configuration which lacks coordination between supply and consumption of heating and cooling. By minimizing the energy spent on cooling in the heating season, it was found that energy savings of approximately $21\,\%$ are achievable. The design and simulations were carried out for a small section of the mall, but given the decentralized HVAC architecture described, it should be scalable to the entire mall.

From the simulation studies it can also be concluded, that it is probably not necessary to use MPC to achieve the same effect. To a large extent, the savings can be achieved by simply turning off the chiller and letting the ventilation unit run with a supply temperature closer to the ambient temperature. This can be achieved with a much simpler and less involved implementation, than for the case of MPC; hereby moving the solution from simple via complex to lucid[13] and avoiding stability and robustness considerations for a complex solution. This does not undermine the applications of MPC, but in this case MPC is used in an exploratory approach to first discover the desired behavior of a more simple solution.

As such, these results form the basis of a control design which will be implemented and tested through the SEBUT project.

ACKNOWLEDGMENT

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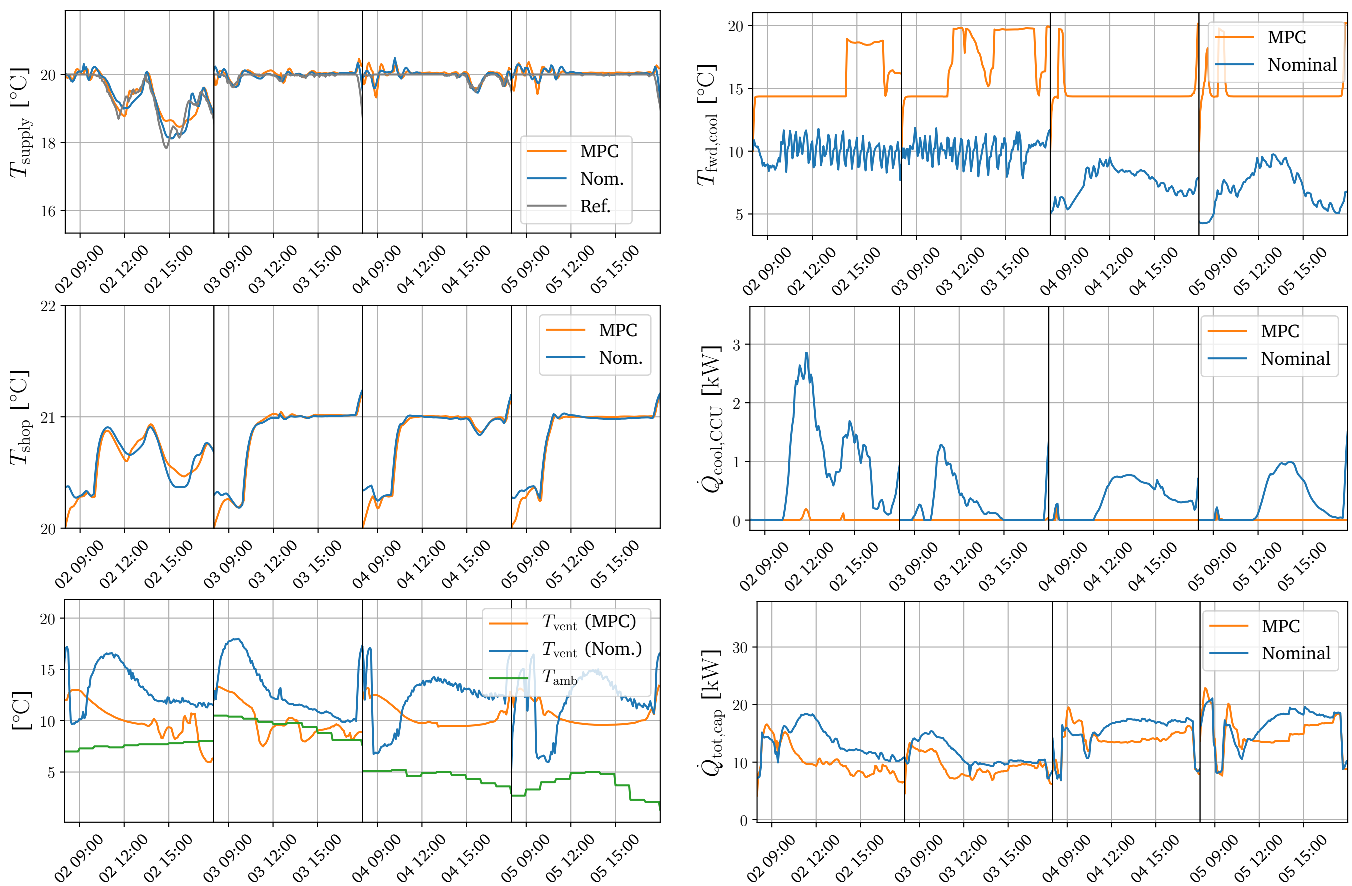

Fig. 6: Comparison of MPC and nominal control through simulation of 4 days. Simulation has been limited to between 8:00 and 18:00 due to operational hours of the HVAC. Simulations are based on data extracted from the BMS. The MPC attempts to lower energy spent on cooling by running with a lower T_{vent}.

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