ENERGY COST SAVINGS BY INTELLIGENT UTILIZATION OF LOAD SHIFTING AND OF A HEAT PUMP

Renato Lepore, Senior Researcher, University of Mons, Mons, Belgium
Christine Renotte, Professor, University of Mons, Mons, Belgium
Eric Dumont, Senior Researcher, University of Mons, Mons, Belgium
Marcel Remy, Professor, University of Mons, Mons, Belgium
Marc Frère, Professor, University of Mons, Mons, Belgium

Abstract: The study is concerned with a 14-storey office building, which requires comfort in temperature and in humidity during the working hours only. The Heating, Ventilation and Air-Conditioning (HVAC) system is of the forced-air type with energy recovery capabilities from the exhaust air. The heating and refrigeration systems comprise not only standard gas boilers and cooling machines, respectively, but also geothermal heat pumps running in both heating and cooling modes.
From earlier evaluations of the total system over one year, the authors have observed that the thermal inertia of the building (due to the concrete structure) plays an important role as for the daily profile of the energy required.
In this study the benefit from shifting the energy load is evaluated, i.e., from storing into the building some energy produced by the heat pump only, preferably at higher Coefficient Of Performance (COP) values, and leaving the building deliver the energy when needed. Due to the slow dynamics of the system, one must account for the future evolution of the weather conditions and of the building occupancy (which yields important heat gains); for this reason a Model-based Predictive Control (MPC) scheme is developed in Matlab®. The algorithm calculates every evening with the rest of the week as a horizon the control variable values allowing for the minimal energy cost while satisfying the comfort constraints. The strategy applies with good results to both heating and cooling periods. In the future the authors will first evaluate the robustness to inaccuracies in the model structure/parameters and in the inputs, then implement the control scheme on site.

Key Words: HVAC applications, heat pumps, advanced control, simulation

1 INTRODUCTION

A scientific collaboration has been taking place for a couple of years between the *University of Mons* and the company *Electrabel (GDF Suez group)*. The projects focus on various energy aspects, including the energetic optimization of the Heating, Ventilation and Air-Conditioning (HVAC) systems of the company's buildings. The office block under study is a concrete-and-glass structure which has been recently built in Brussels, Belgium. The HVAC systems are of the *forced-air type* and consist of

- centralized Air-Conditioning (AC) units, with enthalpy wheels in order to recover energy from the exhaust air and appropriate end-user AC units;
- to some extent, free cooling capabilities from the outdoor air and/or from the ground.

The energy production systems consist of

- standard gas boilers and cooling machines;
- geothermal heat pumps, which can be operated in heating or cooling mode;

- pipe switching systems allowing for different utilizations of the borehole field: heat extraction to or supply from the heat pump, geothermal cooling of the building and, to an extent, regeneration of the ground when successive heat extraction and supply stages do not compensate.

At the first stage of the study, the commercial software Trnsys® was used to evaluate the total system, i.e., the building and the HVAC components, when equipped with an on-off control system to maintain the indoor comfort (as of temperature and humidity) during the working hours. The simulation results revealed a high thermal inertia of the building and therefore higher heating needs in the early morning than during the rest of the day, especially on Monday due to the week-end inactivity.

This observation has led the authors to investigate the potential of load shifting, i.e., to use the concrete structure as a thermal storage for later delivery. The storage can be loaded during non-working hours at better cost by the heat pumps. Provided that a dynamic model of the system is available together with the future evolution of the inputs (weather forecast and prediction of the occupancy schedule), Optimal Control (OC) can advantageously replace the traditional control strategies. If regular measurements of the state variables are available, the OC procedure is repeated and the algorithm is named Model-based Predictive Control (MPC).

Load shifting has been dealt with in the scientific literature for more than twenty years. Early studies appear with solar energy applications, such as the production of Domestic Hot Water (DHW) (Gruenenfelder and Toedtli 1985) or the residential heating equipped with more or less complex optimization schemes (Prud'homme and Gillet 2001). The utilization of the building itself as the thermal storage appeared initially in cooling applications where the electricity cost depends heavily on day-night prices and/or peak-load penalties. Among many researchers having dealt with the topic, Braun has been a major contributor and most of his work is reported in (Braun 2003) and the references therein. Strategies involving the building and/or a separate storage tank (passive and active storage) have also been evaluated (Henze et al. 2004).

In the more recent times a team from the University of California, Berkeley has achieved an important work from design to on-site implementation. Their work is reported in (Ma et al. 2012), with many references to the whole scientific literature on the topic. The same concept has also been developed and implemented on site in heating mode by a Czech team in Prague (Siroky et al. 2011, Privara et al. 2011). Finally let us note that the subject is also treated among many optimization problems related to building energy in the OptiControl Group (http://www.opticontrol.ethz.ch).

The real challenge of applying load shifting in our case lies in the energy transfer mode (forced-air type), i.e., loading the concrete storage is achieved through the indoor air with inevitable losses to outdoor. In a preliminary study developed in Matlab® (Lepore et al. 2013), the authors have evaluated load shifting together with MPC by using a simple set-up: the cost function to minimize combines the amount of primary energy and the deviation from the comfort target, the selected sampling time and prediction horizon are 1 h and 24 h, respectively. The results for a few winter days (heating mode) have been very encouraging.

Here the authors have enhanced the Matlab® program with new important MPC features:

- it is accounted for the major behaviour of the system: typically the concrete temperature falls during the week-end then increases until Friday evening, thus exhibiting some one-week frequency; this means that, with perfectly known inputs, an Optimal Control (OC) procedure with a one-week horizon could be applied

repeatedly; however, mainly because of the inaccuracy in the weather forecast, the OC procedure is performed every day with a shrinking horizon;
- the MPC scheme acts at a higher level with respect to the comfort control during the working hours (low-level control); the procedure calculates the amount of energy to deliver during the night in order to minimize the cost over the rest of the week, all things remaining unchanged during the working hours included the comfort control;
- the same strategy applies to the heating and cooling modes, however with slight differences in the results.

Let us note that:
- the model of the building and of the HVAC components is full-scale;
- the model describes the system in temperature and humidity. However, as the humidity control can be decoupled due to the different dynamics, it is not dealt with in this study.

The rest of the article is as follows. Section 2 describes the building, the low-level control and the HVAC systems. Section 3 describes briefly the principles of MPC and its implementation in more detail. The simulation results and the corresponding analysis are found in Section 4. Section 5 is devoted to the conclusions and perspectives.

2 MODELLING

2.1 Physical description

Figure 1 represents the twin block of the 14-storey building under study, which is 60 m high and has an H-shaped base of 2300 m^2. The total facade wall has a surface area of 18000 m^2, most of it is glassy with a heat transfer coefficient of 1.1 $W.K^{-1}m^{-2}$ and a solar factor of 0.25. The infiltration rate is estimated at 0.3 h^{-1}. The rest of the facade and the structure are made of concrete, the inter-floor separation is 0.33 m thick.



Figure 1: Building under study (Source: Officine Tosoni)

The operating data of the building are from the design specifications. The building is occupied during the working days from 8 a.m. to 6 p.m. This activity (equipment and humans) causes an average internal gain heat flow rate of 900 kW and an average vapour mass flow rate of 80 $kg.h^{-1}$. The indoor air quality is controlled from 7 a.m. to 7 p.m. during the working days by a forced-air system mainly based on central air handling units. The fresh air flow rate is of 42 $m^3.s^{-1}$; however an enthalpy wheel allows for recovering part of the

exhaust air heat. The energy production systems are properly sized: they consist of boilers and cooling machines and, additionally, of two geothermal heat pumps running in heating or cooling mode. In standard heating (cooling) mode conditions, the heat pumps deliver a heat (cool) flow rate of 800 kW (840 kW).

2.2 Model equations

2.2.1 Building model

The behaviour of the building in temperature and humidity is described by equations (1)

$$C_i \, dT_i/dt = (UA_t + UA_{inf})(T_o(t) - T_i) + UA_{ci}\,(T_c - T_i) + \Phi_{vent}(t) + \Phi_{ig}(t) + \Phi_{sg}(t) + \Phi_{hvac}(t) \quad (1.a)$$

$$C_c \, dT_c/dt = UA_{ci}\,(T_i - T_c) \quad (1.b)$$

$$dx_i/dt = \alpha_{inf}\,(x_o(t) - x_i) + \alpha_{vent}(t)(x_o(t) - x_i) + q_{lg}(t)/(V_i\,\rho_{da}) \quad (1.c)$$

where

- T_i, T_c and T_o are the temperatures of the indoor air, of the concrete structure and of the outdoor air, respectively (K); x_i and x_o are the humidity ratios of the indoor and outdoor air, respectively ($kg_{water}.kg^{-1}_{dry\ air}$);
- C_i and C_c are the total heat capacities of the indoor air and of the concrete structure, respectively ($J.K^{-1}$); UA_t and UA_{inf} are the total heat transfer conductances between the indoor and outdoor air nodes, due to transmission and infiltration, respectively ($W.K^{-1}$); UA_{ci} is the total heat transfer conductance between the indoor air and concrete structure nodes ($W.K^{-1}$);
- α_{inf} and $\alpha_{vent}(.)$ are the air change rates due to infiltration and ventilation, respectively (s^{-1}); V_i is the indoor air volume (m^3), ρ_{da} is the density of dry air ($kg.m^{-3}$);
- $\Phi_{vent}(t)$, $\Phi_{ig}(t)$ and $\Phi_{sg}(t)$ are the heat flow rates due to ventilation, internal gains, solar gains, respectively (W); $\Phi_{hvac}(t)$ is the heat flow rate delivered by the HVAC systems (W), which can be positive (heating mode) or negative (cooling mode); $q_{lg}(t)$ is the mass flow rate of water due to the latent gains ($kg.s^{-1}$).

It should be noted that:

- $UA_{inf} = \alpha_{inf}.C_i$;
- equations (1) are kept simple for the sake of comprehension, in reality some parameters (e.g. C_i, UA_{inf}) are not constant values but rather slightly depend on the indoor and/or outdoor air humidity ratios x_i and x_o;
- in equations (1), the explicit dependence on time, noted *(t)*, is due to the varying outdoor conditions (T_o, x_o, Φ_{sg}), to the occupancy schedules (Φ_{ig}, q_{lg}) or to the HVAC schedules (Φ_{vent}, Φ_{hvac});

The ventilation heat flow rate $\Phi_{vent}(t)$ is the net heat flow rate exchanged with the outdoor air, i.e., it accounts for the energy recovery due to the enthalpy wheel. So $\Phi_{vent}(t)$ is expressed by

$$\Phi_{vent}(t) = q_{vent}\, c_{air}\,(1 - \varepsilon_{rec})\,(T_o(t) - T_i) \quad (2)$$

where q_{vent} is the ventilation mass flow rate ($kg.s^{-1}$), c_{air} is the specific heat of humid air and ε_{rec} is the efficiency of the recovery wheel when viewed as a heat exchanger.

2.2.2 Low-level control

The low-level control aims at controlling the indoor air temperature T_i by manipulating the heat flow rate Φ_{hvac}. A Proportional Integral (PI) controller in continuous mode is used, which leads to the additional state equation

$$dv/dt = e_T \quad (3)$$

where $e_T = T^*_i - T_i$, T^*_i being the indoor air temperature set-point,

$$\Delta\Phi_{hvac}(t) = G_{PI} (e_T + v/T_{PI}) \quad (4)$$

where $\Delta\Phi_{hvac}(t) = \Phi_{hvac}(t) - \Phi_{hvac}(t_0)$, t_0 being the triggering instant of the controller, G_{PI} and T_{PI} are the gain and time constant of the PI controller, respectively.

2.2.3 Energy production systems

The model of the energy production systems relies on the following assumptions:

- the components are properly sized, so that no saturation applies on input $\Phi_{hvac}(t)$;
- in heating (cooling) mode, the boilers (cooling machines) and the heat pumps work in parallel (the heat flow rates are added);
- at any time, the heat pump is considered as the most efficient system, thus it has priority to deliver the maximum heat/cold flow rate within $\Phi_{hvac}(t)$.

The heat/cold flow rate allocation, i.e., the calculation of the heat pump flow rate $\Phi_{HP}(t)$ and of the traditional system flow rate $\Phi_{trad}(t)$ are as follows (|x| stands for absolute value of x):

$$|\Phi_{HP}(t)| = \min (|\Phi_{hvac}(t)|, \Phi^m_{HP}(T_{i;ev},T_{i;co})) \quad (5.a)$$
$$|\Phi_{trad}(t)| = |\Phi_{hvac}(t)| - |\Phi_{HP}(t)| \quad (5.b)$$

where $\Phi^m_{HP}(T_{i;ev},T_{i;co})$ is positive and is the maximum heat/cold flow rate that the heat pump can deliver, it depends on the input temperature of the secondary fluids at the evaporator ($T_{i;ev}$) and at the condenser ($T_{i;co}$).

The quantity Φ^m_{HP} (noted $\Phi_{HP;H}$ in heating mode and $\Phi_{HP;C}$ in cooling mode) is reasonably approximated by linear laws (6)

$$\Phi_{HP;H} = a_{0;H} + a_{1;H} T_{i;ev} + a_{2;H} T_{i;co} \quad (6.a)$$
$$\Phi_{HP;C} = a_{0;C} + a_{1;C} T_{i;ev} + a_{2;C} T_{i;co} \quad (6.b)$$

The detailed calculation of these temperatures is beyond the scope of the article, however it is worth describing briefly the connection schemes regarding the heat pump:

- in heating mode, the condenser is connected to the heating coil of the HVAC system, the evaporator is connected to the borehole field through a heat exchanger;
- in cooling mode, the evaporator is connected to the cooling coil of the HVAC system, the condenser is connected to the borehole field through a heat exchanger.

The ground temperature T_{ground} is assumed constant over the year.

3 MODEL-BASED PREDICTIVE CONTROL

3.1 Principle

This section summarizes the basic principles and definitions of the MPC concept. More comprehensive information can be found in the dedicated literature, see for example (Camacho and Bordons 1999, Maciejowski 2001).

As in any control scheme, a control variable u(t) has to be computed at regular sampling times $t_k = k.\Delta T$, where k = 0,1,… and ΔT is the sampling period.

The first feature of MPC is that the computation is performed by solving an optimization problem, i.e.,

$$u(t_k) = \min_{\hat{u}(\tau)} \int_{t_k}^{t_k+H} CF(\hat{x}(\tau), \hat{u}(\tau))d\tau \quad (7.a)$$

given a model and constraints as follows

$$\frac{dx}{d\tau} = f(\hat{x}, \hat{u}(\tau)) \quad (7.b)$$

$$\hat{x}_0 = x(t_k) \quad (7.c)$$

$$g(\hat{x}, \hat{u}) \leq 0 \quad (7.d)$$

Equation (7.a) expresses that

- some cost function CF(.) depending on the predicted state variables $\hat{x}$ and on the control variable $\hat{u}$ is integrated over a time period H called *the prediction horizon*;
- the integral quantity is minimized in order to find the *best* $\hat{u}(\tau)$ profile.

For each $\hat{u}(\tau)$ candidate, variable $\hat{x}(\tau)$ is obtained by solving problem (7.b-7.c). The profiles of the state and control variables must verify constraints (7.d) (typical constraints come from the saturation of the control variables, from the safety/comfort limits regarding the state variables).

The second feature is that the control variable is applied *on the first sampling period only* (from t_k to t_{k+1}). Then, based on a new measurement $x(t_{k+1})$ as the initial condition, a new problem is solved over time horizon H, however from time t_{k+1}.

3.2 Implementation

The MPC control scheme is implemented here according to the following features:

- the MPC control scheme is a high-level control whereas the PI control is a low-level control. For this reason we call conveniently *open-loop system* (described by f(.) in equation (7.b)) the one including the building, the HVAC and the low-level control;
- the cost function CF(.) includes only the billing cost for the company;
- the constraint g(.) ensures that the comfort with MPC is not better than in open loop (let us note that it does not specify any saturation of the control variables);
- the control variables adjust the heating/cooling run of the heat pump during the night and attenuate the low-level control in order to verify the comfort constraint;
- in heating (cooling) mode, 100 % (0 %) of indoor air is used for pre-heating (pre-cooling).

3.2.1 Cost function and constraints

The cost function CF(.) divides into the billing costs of the heat pump $CO_{HP}(t)$ and of the traditional system, $CO_{BO}(t)$ (boiler) or $CO_{CM}(t)$ (cooling machine).

$$CO_{BO}(t) = (\Phi_{BO}(t)/\eta_{BO}).co_{gas}(t) \quad (8.a)$$

$$CO_{CM}(t) = (\Phi_{CM}(t)/COP_{CM}(t)).co_{elec}(t) \quad (8.b)$$

$$CO_{HP}(t) = (|\Phi_{HP}(t)|/COP_{HP}(t)).co_{elec}(t) \quad (8.c)$$

where

- $\Phi_{BO}(t)$, $\Phi_{CM}(t)$ and $\Phi_{HP}(t)$, are the heat flow rates of the boiler, of the cooling machine and of the heat pump, respectively ;
- $co_{gas}(t)$ and $co_{elec}(t)$ are the *possibly time-varying* prices of gas and electricity, respectively, η_{BO} is the boiler efficiency;
- depending on the heat pump mode, care must be taken regarding relation (8.c)
  - in heating mode, $\Phi_{HP}(t)$ is positive and $COP_{HP}(t)$ corresponds to the heat pump operation mode;
  - in cooling mode, $\Phi_{HP}(t)$ is negative and $COP_{HP}(t)$ corresponds to the cooling machine operation mode.
- $COP_{CM}(t)$ is the COP of the cooling machine; in this study it is calculated based on the heat pump characteristics, i.e., equations (6); however, it is considered that the evaporator is directly connected to the cooling coil and that the condenser is connected to outdoor through a heat exchanger.

The constraint function g(.) is presented in two steps:

- one defines some *comfort indicator over some horizon H,* Comf(H), as follows. Let T^{lsp} and T^{usp} be the temperature set-points for the heating (lower set-point) and cooling (upper set-point) mode, respectively, and Tol^T some tolerance parameter then
  - the indoor temperature during the working hours is said *comfortable* if it lies in the temperature interval $[T^{lsp} - Tol^T, T^{usp} + Tol^T]$;
  - Comf(H) is defined as the percentage of comfortable hours over all working hours on time horizon H;
- at each sampling time t_k, one evaluates this comfort quantity in open loop $Comf^{OL}(t_k, H)$ and aims at maintaining with MPC some comfort level close to it; actually, if we define $Comf^{MPC}(H, \hat{x})$ as the comfort quantity obtained with some candidate $\hat{u}(\tau)$ and Tol^{COMF} as some *comfort tolerance allowed to MPC with respect to open-loop (e.g., 10 %)*, the constraint becomes

$$g(.) = Comf^{MPC}(H) – (Comf^{OL}(H) - Tol^{COMF}) <= 0 \quad (9)$$

It must be noted that the comfort indicator is designed for the only purpose of comparing the open loop and MPC behaviours, not to indicate a real comfort in temperature. Besides it is very sensitive the tolerance Tol^T.

3.2.2 Control variables and MPC scheduling

In order to keep the problem tractable only two control variables are selected, as follows.
Firstly, it is assumed that the energy to deliver at night is proportional to the amount of energy delivered by the traditional machine during the day in open loop. The first control variable is that proportionality factor. From its value one calculates the run time of the heat pump at every night, each run ending at 7 a.m.
Secondly, it has been noted that, due to the additional energy delivery at night, running the open-loop system at day with no adaptation of the low-level control could lead to over-comfort. Thus the second control variable adapts the low-level control (here it decreases the gain of the PI controller) in order to verify the comfort constraint.

Every week, the optimization procedure is performed as follows:

- it takes place every day at 7 p.m., from Sunday to Thursday;

- the prediction horizon is equal to 5 days on Sunday evening, then shrinks by 1 day at each sampling time so that the horizon ends always on next Friday at 7 p.m.;
- An extra-computation takes place first, during the week-end, allowing for a long run time of the heat pump during the week-end, i.e., a run time greater than 12 h.

4 RESULTS

4.1 Experimental set-up

The simulation experiments aim at comparing over a one-week (7-day) run the MPC scheme with the traditional, open-loop scheme given that:

- the MPC model fits exactly the system model and the model inputs (weather forecast and internal gains) are known perfectly;
- at every sampling time, the system state is known.

The time evolution of the building temperatures, of the energy flow rates and of the COPs are shown and analyzed. One must note that the week starts on Friday at 7 p.m. (time 0 d in the figures) and ends on next Friday at 7 p.m. (time 7 d in the figures).
Additionally, four main indicators are evaluated: two of them are the *comfort* and the *billing cost*, other two are the *primary energy over the week* and the *accumulated energy indicator.*

The primary energy used by the production systems is defined by equation (10)

$$\mathrm{PE_{week}} = \int_{t_{0w}}^{t_{fw}} \left[\frac{\Phi_{\mathrm{bo}}(\tau)}{\eta_{\mathrm{bo}}} + \frac{\Phi_{\mathrm{HP}}(\tau)}{\mathrm{COP_{HP}}(\tau)\eta_{\mathrm{ps}}} + \frac{\Phi_{\mathrm{CM}}(\tau)}{\mathrm{COP_{CM}}(\tau)\eta_{\mathrm{pp}}}\right] \mathrm{d}\tau \qquad (10)$$

where t_{0w} and t_{fw} are the initial and final time of the week, respectively, η_{pp} is the fixed value of the power plant efficiency. The accumulated energy indicator AEI_{week} allows for verifying that the energy accumulation by the concrete structure over the week does not impact on the results (e.g., that the heat savings in winter are not mainly due to heat delivery by the concrete structure). It is defined by equation (11) as the percentage of the energy accumulated by the concrete structure on the total energy delivered to the system.

$$\mathrm{AEI_{week}} = \frac{\mathrm{C_c}(\mathrm{T_{c;tfw}} - \mathrm{T_{c;t0w}})}{\int_{\mathrm{t0w}}^{\mathrm{tfw}} (\Phi_{\mathrm{bo}}(\tau) + \Phi_{\mathrm{HP}}(\tau) + \Phi_{\mathrm{CM}}(\tau))\mathrm{d}\tau} . 100 \qquad (11)$$

Table 1 contains all the parameter values. The weather inputs (temperature, humidity and solar radiation) are from a weather type of Trnsys (source Meteonorm).

Table 1: Parameter values of the building, of the HVAC system and of the control schemes

Parameter	Value	Unit	Parameter	Value	Unit
UA_t	1.69 10^4	$W.K^{-1}$	T^{lsp}	22	°C
UA_{inf}	1.28 10^4	$W.K^{-1}$	T^{usp}	25	°C
UA_{ci}	3.27 10^5	$W.K^{-1}$	$a_{0;H}$; $a_{0;C}$	884 ; 789	kW
C_i	1.54 10^8	$J.K^{-1}$	$a_{1;H}$; $a_{1;C}$	22.5 ; 22.3	$kW.K^{-1}$
C_c	2.79 10^{10}	$J.K^{-1}$	$a_{2;H}$; $a_{2;C}$	-4.4 ; -7.8	$kW.K^{-1}$
V_i	126.7 10^3	m^3	T_{ground}	8	°C
α_{inf}	0.3	h^{-1}	η_{BO}	0.9	-
α_{vent}	1.18	h^{-1}	co_{gas}	0.030	$€.kWh^{-1}$
ε_{rec}	0.6	-	co_{elec}	0.092	$€.kWh^{-1}$
Φ_{ig}^{nom}	911.3	kW	η_{pp}	0.4	-
q_{lg}^{nom}	78.7	$kg.h^{-1}$	Tol^T	0	°C

G_{PI}	106 10^3	$W.K^{-1}$	Tol^{COMF}	10	%
T_{PI}	0.066	h			

4.2 Winter week

Week 6 has been selected (around the 10th of February) where the outdoor temperature is around 0 °C with a mean deviation of 4 °C. Figure 2 (temperatures) shows that comfort is guaranteed with a sufficient margin. Figure 2 and Figure 3 (heat flow rates) illustrate that the MPC scheme forces pre-heating for around 12 h during Sunday night, which increases the concrete temperature and thus allows for reducing the heating peaks. Despite the increase of heat losses at night, this strategy has a double positive effect: firstly, the heating peaks delivered by the more expensive gas boilers are reduced (at least during the first three days), secondly, the COP is higher with MPC when these peaks occur (see day times 2.5, 3.5 and 4.5 in Figure 4). The main indicators in Table 2 validate these observations: gains of 13 % and 17 % are obtained with MPC in terms of billing cost and of primary energy, respectively, comfort is even better and the energy accumulated is small.

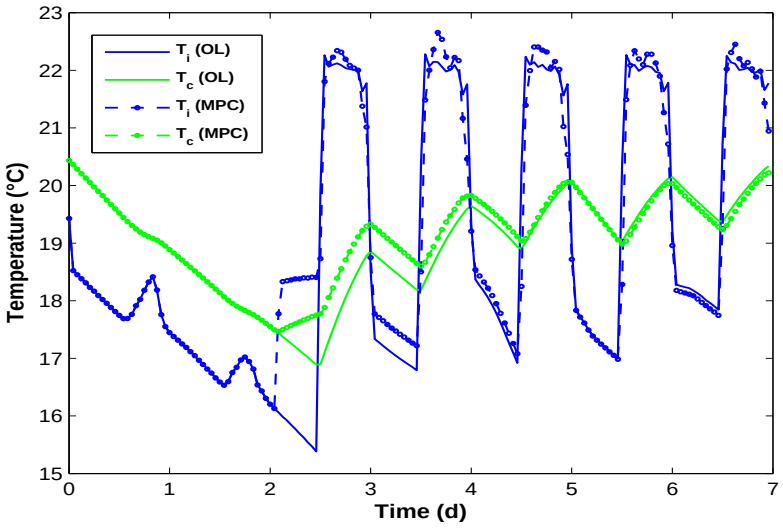


Figure 2: OL and MPC strategies in a winter week – Building temperatures

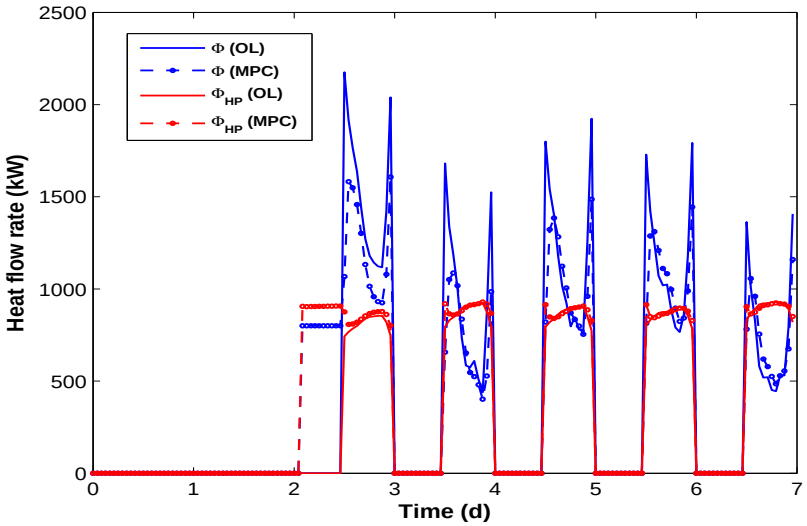


Figure 3: OL and MPC strategies in a winter week – Heat flow rates

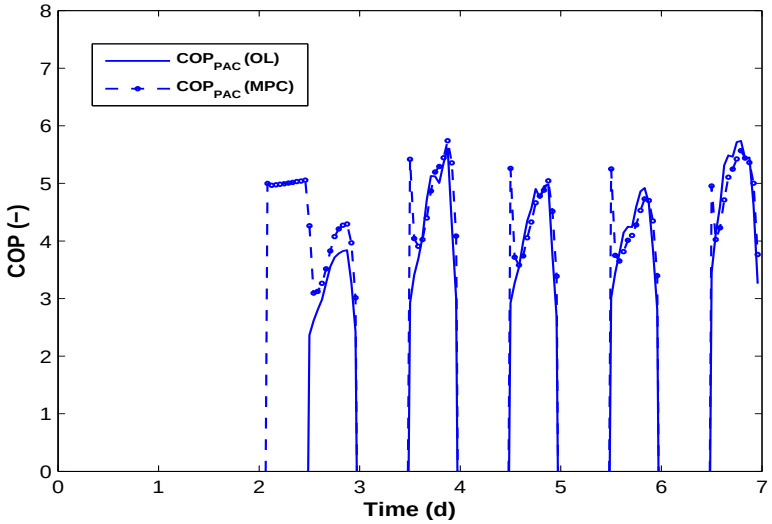


Figure 4: OL and MPC strategies in a winter week – Heat pump COP

4.3 Summer week

Week 34 has been selected (around the 15th of August) where the outdoor temperature is around 20 °C with a mean deviation of 4 °C. Figure 5 (temperatures) shows that comfort is reasonably well maintained. Figure 5 and Figure 6 (heat flow rates) illustrate that the MPC scheme forces pre-cooling mainly on Sunday night, but not only, which decreases the concrete temperature. Compared to the heating case, the decrease of the peaks is less drastic, however, the strategy is interesting again for two reasons: firstly, the effect on the COP of the heat pump but also of the cooling machine is noticeable (Figure 7), secondly, the pre-cooling is achieved with reduced losses (compared to pre-heating). The global indicators in Table 2 validate these observations: gains of 21 % are obtained with MPC in terms of billing cost and of primary energy, comfort is even better and the energy accumulated is small.

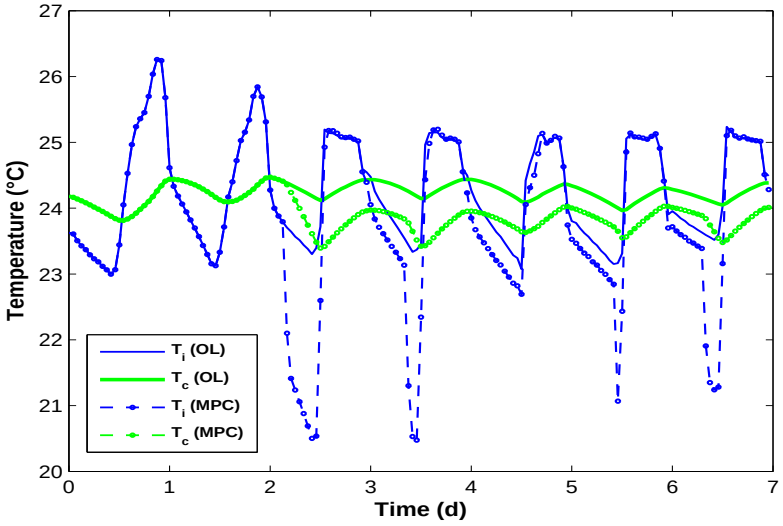


Figure 5: OL and MPC strategies in a summer week – Building temperatures

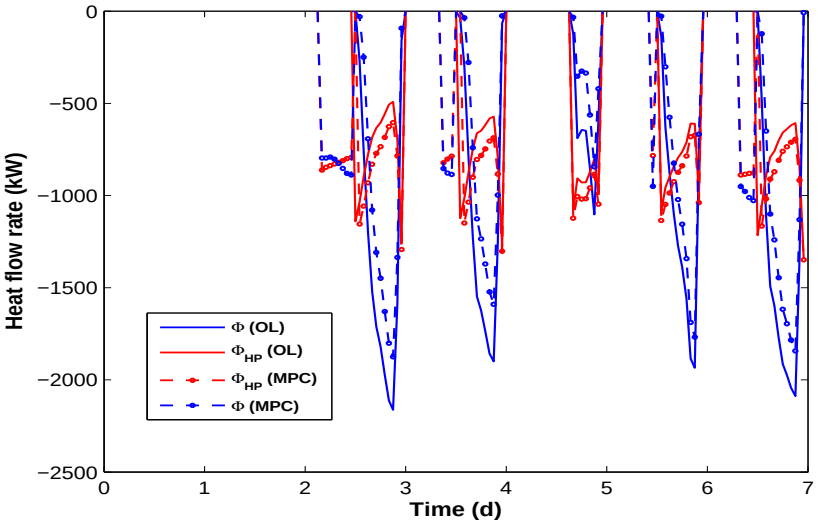


Figure 6: OL and MPC strategies in a summer week – Heat flow rates

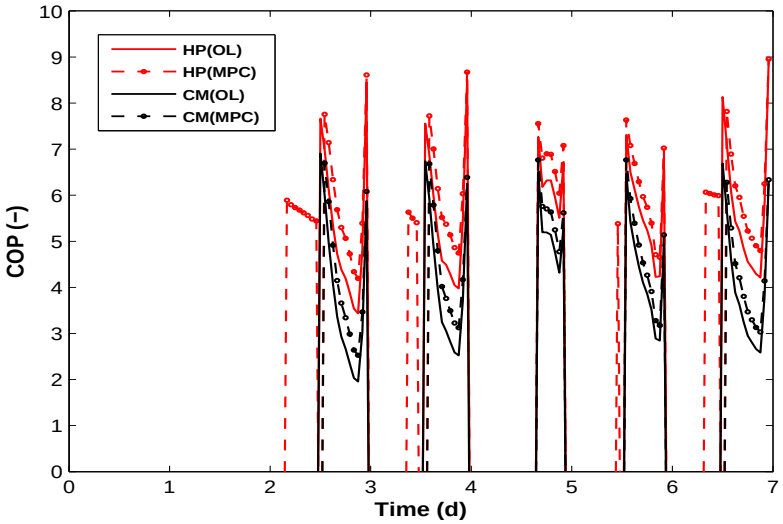


Figure 7: OL and MPC strategies in a summer week – COP

Table 2: OL and MPC strategies – Global indicators

		Billing cost (euros)	Comfort (%)	PE (kWh)	AEI (%)
Winter week	**OL**	1717	72	52720	-1.2
	MPC	1492	74	43750	-2.6
Summer week	**OL**	1501	74	40800	-2.6
	MPC	1180	78	32070	+2.3

5 CONCLUSION

The authors have evaluated with simulation a combined load shifting strategy-MPC control strategy in the special case where the HVAC system is of the forced-air type and where geothermal heat pumps are available beside traditional systems. The MPC algorithm adjusts the standard run of the HVAC system by delivering heat or cold during the night and by tuning the comfort (temperature only) controller in order to decrease the billing cost of the energy while maintaining the indoor temperature of the traditional system. As the MPC operates at a higher level, the comfort control is always assured in case of a MPC failure, which is desirable in practice. The MPC model and the inputs are assumed perfectly known. The simulation runs are performed for winter (heating mode) and summer (cooling mode) weeks. The results show that, with this particular set-up, the gain in either the billing cost or the primary energy is important (more than 15 %) while maintaining the comfort level. In the future, robustness will be evaluated in terms of model and input errors and the estimation of the concrete temperature (state observer) will be studied. Then the algorithm should be implemented on real site.

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