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Thermal and Comfort Control for Radiant Heating/Cooling Systems

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***Abstract*—Water based, surface embedded heating and cooling systems, also referred to as radiant heating cooling systems (RHCS), are growing in popularity, due to their advantages in terms of low-noise, uniform temperature distribution, and energy saving potential. However, it is in general recognized that traditional control systems may deteriorate the energy performance of radiant systems, so that it is important to devise ad hoc strategies for such kind of systems. In this paper, a model-based approach is used to design an efficient control architecture for radiant heating/cooling systems coupled with fan-coil units with the main objective of increasing both thermal comfort for building occupants and energy saving. A building lumped parameter model for hygrothermal analysis coupled with a 2D discretization scheme for radiant heating/cooling systems is development in the Matlab/Simulink environment. The model simulation tool, together with a simple load forecasting strategy, allows to design a suitable controller, that we name *comfortstat*, which is based on the regulation of the Predicted Mean Vote (PMV) thermal comfort index. In this way, thermoigrometric conditions are kept within a range of acceptable comfort values, under performance constraints for reducing energy consumption and preventing floor surface condensation. The results show that the proposed thermal comfort control algorithm gives better satisfaction for the occupants and superior performance with respect to standard approaches.**

I. INTRODUCTION

The increasing effort towards the reduction of energy consumption associated to building heating and cooling processes and the growing attention to indoor environmental quality are supporting the use of water based, surface embedded heating and cooling systems also referred to as RHCS. Such systems allow to better exploit renewable and low-temperature thermal sources, and to achieve better indoor thermal comfort, with respect to traditional air or water based systems. However, it is in general recognized that traditional control systems may deteriorate the energy performance of radiant systems, so that it is important to devise ad hoc strategies for such kind of systems.

Conventional on-off control is widely used for regulation of radiant floor heating systems. However, the presence of large thermal storage in the structures yields delays in the control response causing relatively large deviations of the controlled temperature with from the set-point. A number of alternative control strategies have been proposed recently in the literature. In [1] a model to simulate the control strategies of a multi-zone radiant floor heating system is presented. Different control strategies, such as pulse-width-modulated zone valves with a constant-temperature boiler, supply water temperature outdoor reset with indoor temperature feedback, and outdoor reset plus pulse-width-modulated zone valves have been studied. The results of the simulations indicate that outdoor reset plus pulse-width-modulated zone valve is the best performing control strategy. In [2] a predictive control strategy for the improvement of the energy efficiency of intermittently heated radiant floor heating systems is explored. The results show that the adoption of the predictive control strategy could save energy during the cold winter months. In [3] the performances of different control methods for radiant floor cooling systems are analyzed and the applicability of each control method with regard to floor surface condensation and user comfort is considered through simulations and experiments. It is shown that in controlling the room air temperature, supply water temperature control gives better performance than mass flow control as far as comfort and condensation issues are concerned.

In Heating Ventilation Air Conditioning (HVAC) systems, the thermal comfort degree is described quantitatively in terms of the predicted mean vote (PMV). The control of HVAC system, based on PMV, not only improves thermal comfort but also reduces system energy consumption. In [4] and [5] algorithms have been developed in order to control PMV values and to save energy. The PMV index takes into account different physical parameters in evaluating indoor thermal comfort, among which, radiant temperature is a significant factor. Indoor air velocity can be a key factor to reduce the cooling need, thus allowing energy saving.

In this paper, a model-based approach is used to design an efficient control architecture for RHCS coupled with fan-coil (FC) units with the main objective of increasing both thermal comfort for building occupants and energy saving. Differently from others approaches, the algorithm has to be designed under some technological constraints, given by the fact that it has to be implemented as an upgrade of a traditional control unit. In particular, simple relay-based controllers have to be used to control both fan coils and the radiant surface via

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electro-thermal heads (ETH).

A building lumped parameter model for hygrothermal analysis coupled with a 2D discretization scheme for RHCS is used in Matlab/Simulink. The simulation tool, together with a simple load forecasting strategy based on an exponentially weighted moving average (EWMA) approach allows to design a controller, that we name *comfortstat*. The control strategy is based on the regulation of the PMV index. In this way, thermoigrometric conditions are kept within a range of acceptable comfort values, under performance constraints for reducing energy consumption and preventing floor surface condensation. To prevent conflicts between the PMV control system and vapour condensation control via fan coil operation a decoupling approach is proposed, that makes use of a load estimation scheme.

The results show that the proposed thermal comfort control algorithm gives better satisfaction for the occupants and superior performance with respect to standard approaches. The paper is organized as follows. In Section 2 the radiant system modeling and the simulation tool are outlined. In Section 3 the control architecture for radiant system is presented. In Section 4, simulation results are presented and discussed. Concluding remarks are given in Section 5.

II. The simulation environment

The simulation environment is developed by using IBPT (International Building Physics Toolbox) [6], in Simulink/Matlab. In state-of-the-art simulation tools for building energy evaluation, the calculation domain is subdivided into a wall domain, Ω_b which identifies all bounding surfaces, and an air domain, Ω_a which identifies all volumes containing air. A lumped parameter approach is commonly adopted for the air domain, by subdividing it into a number of nodes corresponding to the building rooms or groups of rooms. The physical model, defined over the whole computation domain, is expressed by the following equations, where the index k refers to the generic kth lumped air volume:

$$P_k \in \Omega_a, \quad V_k \frac{\partial(\rho_{a,k} c_{v,a,k} T_{a,k})}{\partial \tau} = \int \alpha_b (T_b - T_{a,k}) dA_{b,k} + G_k (h_{k,in} - h_{k,out}) + S_{h,k} V_k \,, \tag{1}$$

$$P \in \Omega_b, \quad \frac{\partial(\rho_b T_b)}{\partial \tau} = \nabla(K_b \nabla T_b) + S_h \,. \tag{2}$$

P is the generic bounding surface, V is the volume, ρ is the density and T is the temperature; τ is used as time, c_v is the specific heat at constant volume, α is the heat transfer coefficient, h is the enthalpy, S_h is volumetric heat generation or extraction rate, G is the mass flow rate and K is thermal conduct coefficient. Subscript a and b refers to air and building surface, respectively. For the solution of (1)-(2), a proper set of boundary conditions is needed. Usually, for (1), Dirichlet-type conditions at the air inlet enthalpy are adopted. On the other hand, for (2), a Neumann-type condition is imposed on the portion of the wall surface which is in contact with the external environment. The phenomenological coefficients which characterize the convection heat transfer coefficient (α_b) are evaluated according to common correlations reported in literature. The described physical model is characterized by strong coupling among equations, mainly due to (2), that depends on the temperature profile inside the constructions. In order to create a numerical code for simulating the physical model, a discretization technique is considered for the construction domain. In the IBPT tool, a Finite Control Volume (FCV) model is used for the modeling of the floor construction and floor heating system. The model is a two-dimensional section of the floor construction. It is assumed that the pipe temperature is uniform throughout the floor construction. A fully implicit scheme is adopted for the solution of the resulting ODE problem and a multi-scale integration is introduced to reduce computational time.

III. Control

Human thermal comfort is defined by ASHRAE as the state of mind that expresses satisfaction with the surrounding environment [7]. Maintaining thermal comfort for buildings occupants is one of the important goals of HVAC design. Thermal comfort is affected by body heat conduction, convection, radiation, and evaporative heat loss and it is maintained when the heat generated by human metabolism is allowed to dissipate, thus maintaining thermal equilibrium with the surroundings. Any heat gain or loss beyond this generates a sensation of discomfort. It has been long recognized that the sensation of feeling hot or cold is not just dependent on air temperature alone. The most outstanding work related to the prediction of comfort was done by Fanger [8], who merged physiological theory and statistical evidence of human response and developed a predictive mathematical model of thermal sensation. The benefit of the mathematical model developed by Fanger is that it includes six comfort variables (activity level, insulative clothing value (Clo), ambient air temperature, mean radiant temperature, air velocity, and relative humidity) producing a single comfort index. This statistical index, known as PMV, predicts how the average person would judge the indoor comfort conditions using the ASHRAE thermal sensation scale. Fanger's mathematical model is the basis for a comfort controller that we called *comfortstat*. Like a thermostat, the *comfortstat* would maintain comfort conditions within a range of acceptable PMV values. In a regular thermostat, the comfort index used is dry-bulb air temperature that accounts for only one of the six comfort parameters, whereas the PMV accounts ideally for all six comfort parameters. The thermostat can only effectively control devices that affect the ambient air temperature, whereas the *comfortstat* can also control additional devices that affect radiant temperature, air motion, and humidity [9]. Associated to thermal comfort, energy efficiency in buildings is nowadays an important issue due to the growth of energy costs, energy consumption, and global warming environmental impact. However, there is a trade-off between energy consumption and indoor thermal comfort, that has been progressively attracting the attention of industrial and academic researches. This present paper proposes a control strategy for reducing energy consumption while maintaining acceptable indoor thermal comfort conditions.

A. Standard control

In Figure 1 a basic scheme of multi-zone RHCS system coupled with fan-coil units is shown. The fan-coil system is introduced to supply latent cooling capacity in order to avoid vapour condensation on the cooled surfaces. The i, j indexes correspond to ith zone and jth hygrothermal sensor. Each zone radiant panel supply water is decoupled from the warmed/chilled one (coming from the heat pump/chiller) by a three-way valve. In standard applications the set-point value for the warmed/chilled water at the heat-pump/chiller output is set at 45 °C and 7 °C during winter and summer period respectively (no supply water temperature reset), whereas the panel supply water temperature, T_{3w}, is set at 35 °C and 15 °C, respectively. In cooling mode, to avoid surface condensation, the lower acceptable water temperature entering the radiant surface, which is controlled by the ith zone three-way valve, is set to the maximum jth dew point temperature. The indoor temperature is controlled by activating the radiant surface through an electro-thermal head (ETH) by using a relay logic. Condensation on the cold surface is avoided by controlling relative humidity with a fan-coil system. In this way the RHCS system is coupled with the fan-coil system in determining the indoor comfort, since the PMV depends on relative humidity. Fan-coils sensible cooling capacity also strengthens such coupling effect since they decrease room air temperature, and thus PMV, when they are activated. Moreover, if the radiant cooling system does not completely match the sensible load, fan-coils are activated to support it.

B. Comfortstat

In a traditional control prospective, the effect of operating fan coils on room temperature is seen as a disturbance, thus, there is a conflict between the two control actions. To overcome the drawbacks of traditional controllers associated with the coupling between radiant and fan-coil systems, a strategy based on PMV index and room dew point temperature control is proposed. The aim of the control strategy is to decouple comfort control from surface condensation prevention. The PMV is controlled by activating intermittently the radiant surface by using the ETH while each room dew point air temperature is regulated by fan-coils as a function of the surface temperature. The radiant surface activation, and therefore its electro-thermal head opening, is controlled by means of a relay logic with hysteresis on the PMV index set-point value:

$$\mathrm{ETH} = \begin{cases} \text{on}\,, & \text{if} \quad \mathrm{PMV} > \mathrm{PMV}_{\text{panel,high}}\,, \\ \text{off}\,, & \text{if} \quad \mathrm{PMV} < \mathrm{PMV}_{\text{panel,low}}\,. \end{cases} \tag{3}$$

The ith zone three-way outlet water temperature is varied as a function of the sensible load of each thermal zone. This reduces radiant surfaces intermittent operation while maximizing the ETH opening time. A load estimation algorithm is used for each room. Assume that, at a given time instant, the following relation between the jth room load, the PMV index, the three-way outlet water temperature, and the electro-thermal head opening time t_{ETH} holds:

$$Q_{i,j}(\tau) = \widetilde{\alpha}_{i,j}\left[\Gamma\left(\mathrm{PMV}_{i,j}(\tau)\right) - T_{3w,i}(\tau)\right] t_{\text{ETH},i,j}(\tau) \tag{4}$$

where $\widetilde{\alpha}_{i,j}$ is a transfer coefficient and the function Γ gives an equivalent temperature as function of the PMV index value:

$$\Gamma_H(\mathrm{PMV}) = 21.57 + 4.573\mathrm{PMV}\,, \tag{5a}$$

$$\Gamma_C(\mathrm{PMV}) = 24.71 + 3.317\mathrm{PMV}\,, \tag{5b}$$

where the (5a) refers to the heating mode and (5b) to the cooling mode. Equations (5) are obtained by polynomial regression from UNI-EN-ISO 7730 [10] curves around 20 °C and 26 °C room temperature, 60% air relative humidity, 1 and 0.5 Clo index in heating and cooling mode respectively. Mean radiant temperature is equal to air temperature and the other parameters contributing to PMV determination are kept constant. In the hypothesis of constant thermal load (slowly varying in time), the three-way valve outlet water temperature that gives the desired ETH opening time (e.g. $t_{\text{ETH,sp}}$ equals to 90%) is given by:

$$T^*_{3w,i,j}(\tau) = \Gamma(\mathrm{PMV}_{\text{sp},i,j}) - \widetilde{\alpha}^{-1}_{i,j} t^{-1}_{\text{ETH,sp}} Q_{i,j}(\tau)\,, \tag{6}$$

where $\mathrm{PMV}_{\text{sp},i,j}$ is the PMV set-point. To calculate (6), an estimation of each room heating/cooling load is required. An EWMA is used to this aim [11]. From a set of past observations, the following recursive relation gives the load forecasting:

$$\hat{Q}_{i,j}(\tau) = \lambda Q_{i,j}(\tau-1) + (1-\lambda)\hat{Q}_{i,j}(\tau-1)\,, \tag{7}$$

where $Q_{i,j}(\tau-1)$ is the prior observation included in the analysis. The initial value is an user defined value and λ is the value of the weighting factor (an optimal value of lambda can be determined that minimizes the prediction error). The outlet three-way valve temperature set-point, in heating and cooling mode, can be calculated as:

$$T_{3w,H,i}(\tau) = \max_j T^*_{3w,i,j}(\tau)\,, \tag{8a}$$

$$T_{3w,C,i}(\tau) = \max\left[\min_j T^*_{3w,i,j}(\tau), \max_j T_{dew,i,j}\right]\,, \tag{8b}$$

where the constraint on the maximum jth room dew point temperature is included for safety reasons. In cooling mode, fan-coils are controlled in order to prevent surface condensation by a relay logic based on dew point temperature measure of each jth room, which should be kept lower than the radiant surface temperature. The control variable is thus the difference between dew point and surface temperature. Differently from relative humidity, this variable directly measures the risk of condensation on cooled surfaces. This strategy does not completely decouple the radiant from the fan-coil system, given that fan-coils activation influences the PMV value. However, being more focused on the control target, which is avoidance of vapour condensation on cold surfaces, it appears more efficient than the standard one in controlling relative humidity.

Moreover, in cooling mode the three-way valve control rises the outlet water temperature at low cooling loads, thus reducing the risk of vapour condensation on the radiant surfaces and the fan-coil activation period, and consequently decreasing their effect on the PMV value associated with their sensible cooling capacity. Since the radiant surface temperature is not directly measured, three-way outlet water temperature is used for safety reasons:

$$\mathrm{FC} = \begin{cases} \mathrm{on}\,, & \text{if} \quad T_{dew,i,j} - T_{3w,i} > -dT_{sc}\,, \\ \mathrm{off}\,, & \text{if} \quad T_{dew,i,j} - T_{3w,i} < +dT_{sc}\,. \end{cases} \tag{9}$$

where $T_{dew,i,j}$ is the dew point temperature of ith zone and jth room and the safety coefficient dT_{sc} is set to 0.5. Additional fan-coils sensible load can be still supplied to support the RHCS system both in heating and cooling mode. Fan-coils are activated and controlled through a relay logic based on PMV index:

$$\mathrm{FC} = \begin{cases} \mathrm{on}\,, & \text{if} \quad \mathrm{PMV} > \mathrm{PMV}_{\mathrm{FC,high}}\,, \\ \mathrm{off}\,, & \text{if} \quad \mathrm{PMV} < \mathrm{PMV}_{\mathrm{FC,low}}\,. \end{cases} \tag{10}$$

Being the fan-coil activation limited to critical operating conditions, the value of $\mathrm{PMV}_{\mathrm{FC,high}}$ ($\mathrm{PMV}_{\mathrm{FC,low}}$) is larger (smaller) than that set in cooling (heating) mode. It is worth noticing that this solution is recommended in directional facilities characterized by high internal loads because of the relatively low radiant panel cooling capacity. The sensible load supplied by fan coil during the supporting period reduces the value of the PMV index thus affecting load forecasting. To avoid this, the added load can be estimated as a function of the fan coil run-time ratio calculated during the sensible load supporting period (see Figure 2):

$$\begin{aligned} \hat{Q}^{\mathrm{FC}}_{i,j}(\tau) &\propto \Delta\mathrm{PMV}_{\mathrm{FC},i,j}(\tau) \\ &= \left[\delta\mathrm{PMV}_{\mathrm{FC}}\left(1 + t_{\mathrm{FC,SLSP,on}} t^{-1}_{\mathrm{FC,SLSP,off}}\right)\right]_{i,j}\,, \end{aligned} \tag{11}$$

where:

$$\delta\mathrm{PMV}_{\mathrm{FC}} = \mathrm{PMV}_{\mathrm{FC,high}} - \mathrm{PMV}_{\mathrm{FC,low}}\,. \tag{12}$$

Equation (6) is then modified to account for such additional term as:

$$\begin{aligned} T^{*}_{3w,i,j}(\tau) &= \Gamma\left(\mathrm{PMV}_{\mathrm{sp},i,j}\right) + \Gamma\left(\Delta\mathrm{PMV}_{\mathrm{FC},i,j}(\tau)\right) \\ &\quad - \widetilde{\alpha}^{-1}_{i,j} t^{-1}_{\mathrm{ETH,sp}} \hat{Q}_{i,j}(\tau)\,. \end{aligned} \tag{13}$$

Moreover, to increase heat-pump/chiller efficiency a supply water temperature reset is adopted according to the minimum ith zone three-way outlet water temperature. By way of example, the chilled water temperature is calculated as:

$$T_{ch} = \min\left[\min\left(T^{*}_{3w,i}\right), \min\left(T_{r,i}\right)\right]\,, \tag{14}$$

where T_r is the minimum temperature ensuring vapour condensation on fan-coils surface and is defined as:

$$T_{r,i} = \min_j T_{dew,i,j}\left(T_{i,j}, RH_{i,j}\right) - dT_r\,. \tag{15}$$

In (15), $T_{r,i}$ corresponds to the dew-point temperature of ith zone and jth room with $RH_{i,j}$ relative humidity and dT_r is a safety coefficient.

IV. A Simulation Example

The simulation test case consists of a single-room environment with orientation and dimensions as shown in Table I. The system has been simulated using the IBTP tool in Venice (Italy) weather conditions. For the considered test case, room hygrothermal sensors measure room air dry-bulb temperature and relative humidity only. Dew point temperature is calculated from air properties. The PMV index is calculated considering the mean radiant temperature equal to the air temperature. Clo index is 0.5 and 1 for cooling and heating simulation, the metabolic index (met) is set to 1.2 corresponding to office activity, and the air velocity is set to 0.1 m/s. The room is climatised with an air-condensed packaged water heat pump; its power absorption (fans included), heating (H) and cooling (C) capacities are characterized by the following polynomial equations as a function of the external air temperature, $T_{a,ext}$, and inlet water temperature T_w:

$$P_{\{H,C\}} = a_1 T_w + a_2 T_{a,ext} + a_3\,, \tag{16}$$

$$P_{e\{H,C\}} = b_1 T_w + b_2 T_{a,ext} + b_3\,, \tag{17}$$

where the coefficients in (16) and (17) are obtained from manufacturer data (Table II).

Fan-coil sensible and cooling capacities are obtained by polynomial interpolation of manufacturer data as a function of room air temperature and relative humidity, water mass flow rate and air flow rate (as a function of three fan velocities). In Table III the heating mode parameters related to standard control and *comfortstat* are reported. It is worth noticing that the PMV set-point for the *comfortstat* corresponds to the temperature set-point for the standard control according to (5). In Table IV the results of simulation from 1st of January to 31th of March are shown in terms of average chilled water temperature, three-way outlet temperature, Energy Efficiency Ratio (EER defined as the ratio between the integrated values of cooling energy and absorbed power), ETH opening time and PMV. *Comfortstat* allows a heat pump (hp) and three-way outlet temperature reduction of 18.6 K and 10.6 K respectively.

This increases the ETH opening time, which rises from 43.8% to 66.5%. Decreasing of heat pump condensation temperatre allows significant energy saving, equals to 18.6%. This value is consistent with the calculated EER values which are equal to 3.67 and 4.18 for the standard control and comfortstat respectively. In Table V cooling mode parameters related are reported for standard and comfortstat control. It is worth noticing that the PMV set-point is given in accordance to (5). In Table VI the results of simulation from 15th of April to 15th of September are shown. With comfortstat adoption, chiller and three-way outlet temperature increase by 5.7 K and 1.9 K respectively. The ETH opening time is similar because of the radiant surfaces reduced capacity at low water inlet temperature. Significant energy saving, equals to 17.1% is obtained. The energy efficiency, EER, increases from 3.65 up to 3.86 with comfortstat.

In Figure 3, PMV index, indoor air temperature, relative humidity, fan coil operating time, ETH opening time, three-way outlet water temperature and internal loads are shown, respectively (*comforstat*, 3rd Week of July, from Monday to Friday). The time behavior of the controlled variables is that typical of relay-based control systems. PMV is maintained within the 0-1 comfort zone (set-point is equal to 0.4) to grant a mean temperature of 26 °C.

V. Conclusion

In this paper, the *comforstat*, a PVM-based control system for RHCS, has been proposed. Given existing technological and implementation constraints, control of both the radiant surfaces and fan-coil units is based on relay logic, as in traditional control systems. The control algorithm improves over traditional control schemes in that it reduces the conflicts between the needs of controlling user comfort (in terms of PVM) and those of preventing vapor condensation on the cooled surfaces by estimating the sensible load provided by the fan coils and consequently compensating for it when computing the opening of the ETH. The *comforstat* performance is evaluated by means of Matlab/Simulink simulations. Increase in the energy performance (as described by the EER) is achieved thanks to the increase (decrease) of the water chiller (heat pump) water temperature. User comfort is granted by keeping the PMV around an user supplied set-point.

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TABLE I
CASE STUDY: ROOM AND HVAC PARAMETERS.

Dimensions	4.0× 4.5×2.7	[m]
North-Sud wall surface area	10.8	[m^2]
East-West wall surface area	12.2	[m^2]
West windows surface area	3.0	[m^2]
Total internal loads	900 ± 150	[W]
Latent loads	180	[W]
Radiant panels	ceiling	[-]
Fan coil	815 (365 latent)	[W]
Air recirculation	0.5	[vol/h]

TABLE II
COEFFICIENTS: POWER ABSORPTION, HEATING AND COOLING CAPACITIES.

Coefficients	a_1	a_2	a_3	b_1	b_2	b_3
Heat pump	-0.11	1.05	57.27	0.40	0.07	2.13
Chiller	1.98	-0.67	42.59	0.14	0.39	5.61

TABLE III
HEATING MODE: CONTROL PARAMETERS.

	Standard control	Comfortstat	
T_{hp}	45	T^*_{3w}	[°C]
T_{3w}	35	T^*_{3w}	[°C]
T_{air}	20±1	-	[°C]
PMV	-	-0.34±0.22	[-]
Fan-coil	inactive	inactive	[-]

TABLE IV
HEATING MODE: RESULTS.

	Standard control	Comfortstat	
T_{hp}	45.0	26.4	[°C]
T_{3w}	36.9	26.3	[°C]
EER	3.67	4.18	[-]
ΔEER	0.0	13.9	[%]
Heating energy	2521	2339	[MWh]
Electric energy	687	559	[MWh]
ΔElectric energy	0.0	-18.6	[%]
ETH opening time	43.8	66.5	[%]
PMV	-0.44	-0.54	[-]

TABLE V
COOLING MODE: CONTROL PARAMETERS.

	Standard control	Comfortstat	
T_{ch}	7	$\min\left(T^*_{3w}, T_{dew} - 1\right)$	[°C]
T_{3w}	$\max(15, T_{dew} + 2)$	$\max\left(T^*_{3w}, T_{dew}\right)$	[°C]
T_{air}	26±1	-	[°C]
$T_{air,\mathrm{FC}}$	27.5±0.5	-	[°C]
RH_{air}	60±5	-	[%]
dT_{sc}	-	0.5	[K]
PMV	-	0.4±0.3	[-]
$\mathrm{PMV}_{\mathrm{FC}}$	-	0.85±0.15	[-]
T_{dew}	-	(T_{3w}-1)±1	[°C]

TABLE VI
COOLING MODE: RESULTS.

	Standard control	Comfortstat	
T_{ch}	7.0	12.7	[°C]
T_{3w}	16.7	18.6	[°C]
$T_{air} - T_{ch,in}$	10.9	7.7	[-]
EER	3.65	3.86	[-]
ΔEER	0.0	5.8	[%]
Cooling energy	4447	3908	[MWh]
FC Cooling energy	35.6	36.1	[%]
Electric energy	1219	1011	[MWh]
ΔElectric energy	0.0	-17.1	[%]
ETH opening time	71.2	72.8	[%]
FC operating time	48.9	58.0	[%]
PMV	0.23	0.52	[-]

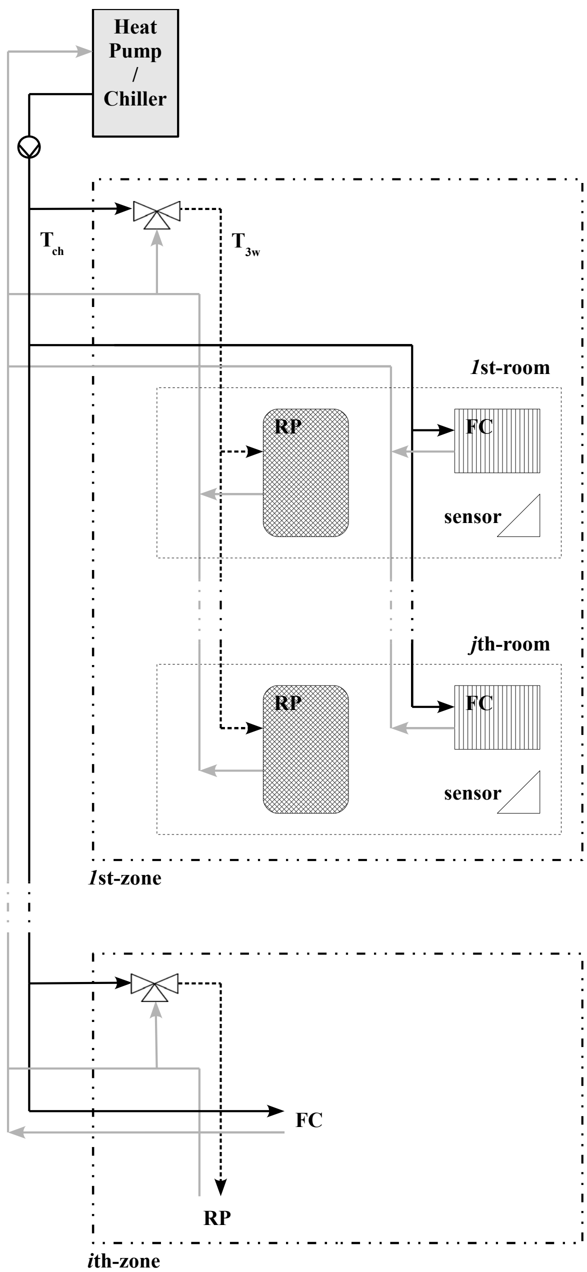

Fig. 1. Multiple zone radiant heating/cooling system: chiller, radiant panels (RP), fan coils (FC) and hygrothermal sensors.

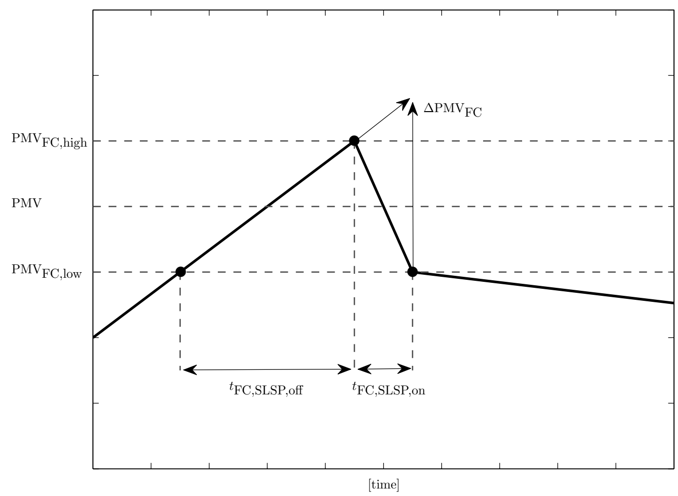

Fig. 2. Fan-coil: sensible load supporting period.

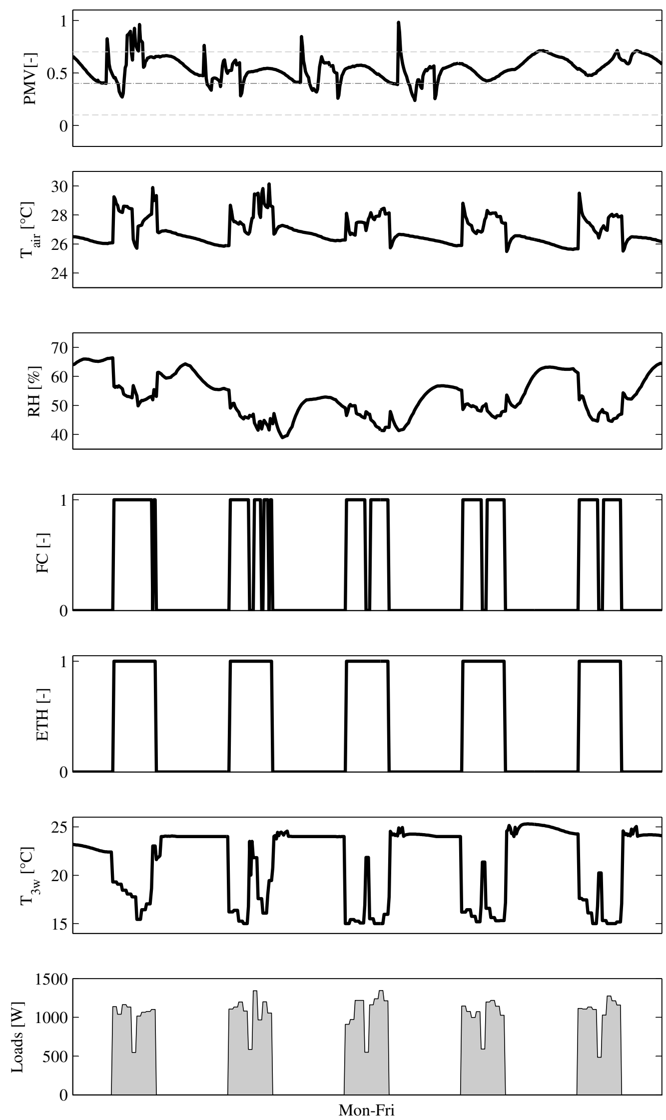

Fig. 3. *Comforstat*, cooling mode (3rd Week of July, Mon-Fri): PMV, T_{air}, RH, FC operating time, ETH opening time, T_{3w} and internal loads.