NUMERICAL ANALYSIS OF THE THERMAL COMFORT IN A RETROFITTED FAMILY HOUSE USING A PCM/AIR HEAT EXCHANGER SYSTEM.

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ABSTRACT

This paper deals with the results of an investigation into the freecooling efficiency in a low energy building using a PCM/air heat exchanger coupled with the mechanical ventilation. The numerical model of the PCM system is coupled with the type 56 of TRNSYS in order to analyse thermal comfort conditions inside the building but also cooling energy savings due to such a system. Several air rate change, temperature of fusion, climate and convective heat transfer coefficient modes were analysed. It was found that the system provides more favourable thermal comfort and leads to reduce the cooling energy demand and the peak cooling demand. However, it was shown that the night ventilation is more effective than the PCM due to fusion/cristallization problems.

INTRODUCTION

In France, housing and office buildings are responsible for the consumption of approximatively 46% of all energies and approximatively 19% of the total CO_2 emissions. To reduce energy consumption in buildings, solutions have to be found both in new and retrofitted buildings. It is the goal of the thermal regulation RT2012 for instance. Regarding the rate of new buildings built per year, which is relatively low, it obviously has to focus on refurbishment. In France the main solution is to reduce heat losses by addition of thermal insulation either on external or on internal facade according to architectural problems. The last solution often leads to thermal discomfort during summer period. Indeed the thermal inertia is reduced and thus sensible energy of the building structure (i.e. concrete) is not available to store the energy due to solar gains and internal heat gains.

In the litterature several methods are proposed to improve the thermal comfort of a building. Among those solutions, Fraisse et al. (2006) analysed the influence on the thermal comfort of a ventilated double wall. The idea is to increase the house's thermal inertia using a concrete wall through which fresh air circulates during the night to evacuate the accumulated heat. Other solutions are to use Phase Change Materials (PCMs) introduced in building walls in which the heat is stored and released according to internal temperature (Kuznik et al., 2011; Baetens et al., 2010; Zhu et al., 2009; Zhou et al., 2007; Zalba et al., 2004). However the numerical results show that it can be a good solution, in practice this solution is limited by the low heat transfers at the interface wall/air (David et al., 2011). That is why we propose a solution using phase change materials in an active system like the ventilation. Some papers are available in the litterature on this topic. For instance, Arkar et al. (2007), Arkar and Medved (2007) present a study of free cooling applied to a low energy building. The cylindrical heat storage is filled with organized structured spheres containing PCM. A numerical study with TRNSYS (TRaNsient System Simulation program) was performed. The authors conclude in a better thermal comfort even if they have just analysed the air temperature of the building. No results are given about energy savings in this study. Nagano et al. (2006) present a new floor supply air conditioning system, using phase change material to enhance building mass thermal storage. The results from measurements simulating an air conditioning schedule in office buildings indicate that 89% of daily cooling load could be stored each night in a system that used a 30 mm thick packed bed of the granular PCM. No building simulation was performed in this study.

This work deals with the evaluation both on thermal comfort and on energy savings during summer period of a PCM air heat/exchanger. The first section deals with the simulation arrangment. Details are given about the investigated building and the software used to perform the numerical simulation. Then, the main assumptions about the numerical model of the PCM air/heat exchanger are indicated. The second section gives the main results of the study which are finally discussed.

SIMULATION ARRANGEMENT

The work is carried out by numerical simulations in the TRNSYS software. The Type 56 is used to model the building whereas a specific Type was developed under MATLAB to model the PCM ventilation system.

Investigated Building

To achieve the desired objectives the TRNSYS Type 56 was used to model a single French family house with a floor area of approximatively 100 m^2 (figure 1). The house was assumed to be 2.5 m high, on crawl space and with a false roof. It consists of three bedrooms, one kitchen, one bathroom, one corridor, a

living-room and a garage on the west side. The modelling of the house is defined by two different thermal zones. The day zone is the living-room whereas the night zone is represented by the other rooms of the house.
To provide an accurate representation, schedules were used for lighting, occupancy (4 persons), heating, cooling and electrical equipments leading to internal heat gains shown on figure 2.

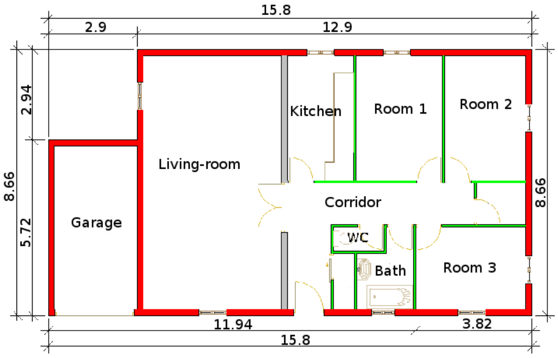

Figure 1: Schematic plan view of the house

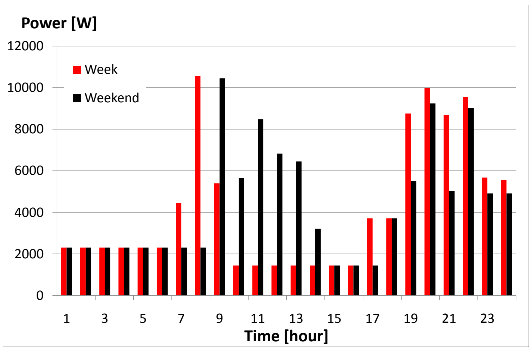

Figure 2: Internal heat gains

Table 1 provide the thermal characteristics of the materials used in TRNSYS. The composition (inside to outside) of the walls, the roof and the floor are respectively the following:

- gypsum(1cm)-polystyrene(25cm)-brick(11cm)-siding(1cm)
- plaster board(1.3cm)- mineral wool(40cm)
- concrete(20cm)- polystyrene(10cm)

The other parameters needed in TRNSYS are listed in table 2

Numerical modelling of the PCM/air heat exchanger

The design of the PCM/air heat exchanger is presented in figure 3. The heat exchanger is composed by several PCM plates which are introduced in an insulated box. The air is forced inside the heat exchanger by a

Table 1: Thermophysical characteristics of the walls

Material	λ $[W/(m.K)]$	ρ $[kg.m^{-3}]$	Cp $[J/(kg.K)]$
Gypsum	0.32	1200	837
Polystyrene	0.04	25	1380
Brick	0.5	720	794
Siding	1.15	1700	1000
Plaster board	0.32	1200	837
Mineral wool	0.03	35	1180
Concrete	1.75	2300	920

fan in order to store/release the heat through the PCM. During the day, supply air in the building release heat to the PCM in order to cool the room. During the night, fresh external air is used to release the heat inside the PCM. The PCM thermal storage system is 1.2m long, 1.2m width and 0.462 m high. There is 10 plates of PCM of 0.03 m thick and the airgap between two plates is 0.018 m. The thermal conductivity of the PCM is 0.2 $W.m^{-1}.K^{-1}$, the density is 1000 $kg.m^{-3}$ and the specific heat at constant pressure is given by figure 4. The phase change material is a paraffin with a latent heat of 170 $J.g^{-1}.K^{-1}$.
Concerning the numerical modelling, see (Kuznik

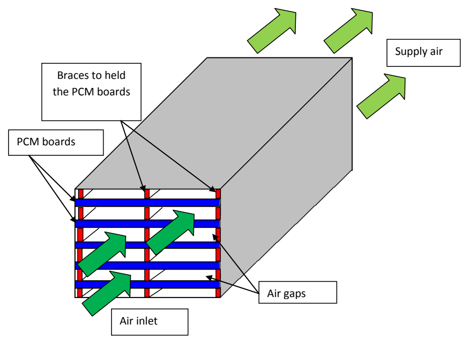

Figure 3: PCM air/heat exchanger design

et al., 2008; Borderon et al., 2010) for detailed information, the energy conservation equation (1) is solved thanks to an implicite finite differences scheme in two dimensions. The two dimensions are the thickness and the length of the PCM layer.

$$\frac{\partial h_{PCM}}{\partial t} = C_{PCM}(T)\frac{\partial T}{\partial t} \quad (1)$$

Due to convective exchange with the air gap, the convective heat transfer coefficient is evaluated according to the flow regime (transition at $Re \geq 2500$). For laminar flow, the Nusselt number is calculated with the Gratz-Nusselt relation adapted to this geometry:

$$Nu = 7.541 + \frac{0.0235.Re.Pr.X}{L} \quad (2)$$

Where X is the hydraulic diameter, L the length of the PCM layer.

Table 2: operating parameters in TRNSYS

Parameter	Value
Weather data file	Trappes or Nice
Mechanical ventilation	0.6 volume air change per hour
Vertical/horizontal convective heat transfer coefficient(inside)	9/11
Vertical/horizontal convective heat transfer coefficient(outside)	16.5/11
Solar absorption of wall	0.6

For turbulent flow, the Nusselt number is calculated with the Colburn correlation:

$$Nu = 0.023.Re^{0.8}Pr^{1/3} \qquad (3)$$

The whole system to solve is therefore the following:

$$\{T\}^{n+1} = [M(T^n)]\left(\{T\}^n + \{B\}^{n+1}\right) \qquad (4)$$

This numerical model has been devloped in MATLAB and is linked to TRNSYS with the Type 155. The inputs of the numerical model are the outside air temperature and the air rate change. The outputs of this type are the supply air temperature as well as the cooling power.

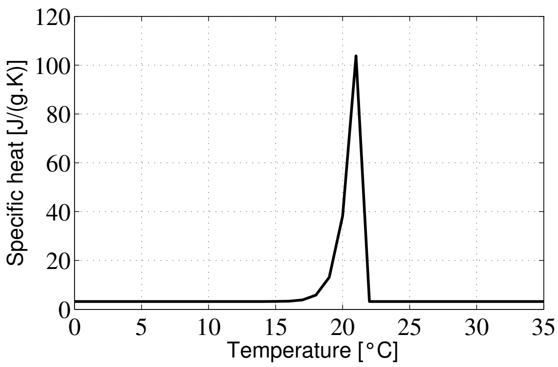

Figure 4: Evolution of the specific heat according to the temperature

RESULTS AND ANALYSIS

On the basis of the building described above the operative temperature T_{op}, the cooling load Q and the peak cooling demand P_{max} have been analysed in several configuration. The period analyzed is from the beginning of May to the end of September. In order to know the impact of the PCM/air heat exchanger both on thermal comfort and on energy savings, the influence of four parameters are analyzed, i.e the ventilation air rate, the use of a cooling system, the temperature of fusion T_f and the climate (Nice or Trappes). Note that Nice (south of France) is a very hot climate with small gap between the day and the night temperature wheras Trappes (close to Paris) is colder with high gap between temperatures. The different studied configurations are shown in table 3.

Table 3: Simulated configuration for both climate

Case	PCM	Air rate	cooling	T_f
1	No	0.6	No	-
2	No	0.6	Yes	-
3	No	0.6 day 3 night	No	-
4	No	0.6 day 6 night	Yes	-
5	No	6	No	-
6	Yes	3	No	21
7	Yes	6	No	21
8	Yes	6	Yes	21
9	Yes	6	Yes	23
10	Yes	6	Yes	26

No cooling system

Results obtained in case1 have to be considered as reference results since it represents the results obtained for a refurbished single family house. Figure 5 shows the evolution of the operative temperature for the climate in Nice and in Trappes. It shows that in Nice the operative temperature is 23% of the time over 26°C. This phenomenon is less important in Trappes because the external temperature and the sun radiation are lower than in Nice. However, the operative temperature is often over 26°C in both climates leading to thermal discomfort. If the percentage of time where the operative temperature over 28°C in Trappes is not significant, it represents more than 15% of the time in Nice. The figure also shows the good influence of the

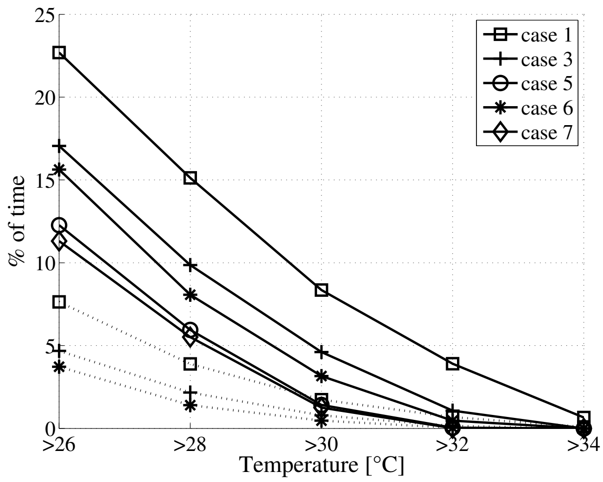

Figure 5: Evolution of Top according to the test case - dotted line for Trappes, solid line for Nice

night ventilation. Indeed the night ventilation reduces the operative temperature over 26°C in Nice to 17% while it represents 10% of the time over 28°C for an air change rate of 3 volume per hour. If the air change rate is equal to 6 volume per hour therefore the results are better since the operative temperature is only 12.5 % of the time over 26°C and 5% of the time over 28°C. Case 6 and case 7 use the PCM/air heat exchanger with

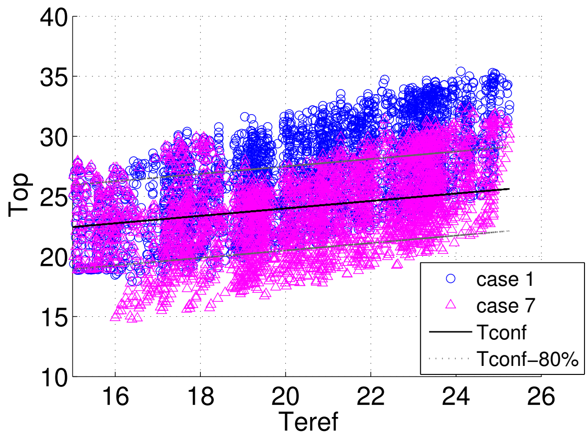

Figure 6: Comparison of comfort in Nice (adaptative comfort)

respectivly an air change rate of 3 and 6 volume per hour in the summer period. The gain in thermal comfort due to PCM is very low with the air change rate of 3 and negligeable with 6 volume per hour compared to case 3 and case 5.

Figure 6 presents the gain in comfort in Nice based on the adaptative comfort method (Fraisse et al., 2010; van Hoof and Hensen, 2007). It presents a comparison of the operative temperature between case 1 and 7 according to the reference external temperature T_{eref} defined in the adaptative comfort method. The comfort temperature T_{conf} as well as the 80% acceptability limits are also indicated. This figure shows that case 7 leads to a better thermal comfort since the temperatures are inside the limits of the 80% acceptability. It has to be underlined that the previous results does not take into account the extra electrcity consumption of the night ventilation.

With cooling system

Although the French regulation forbids the use of cooling system in single family house, more and more families use it to improve their thermal comfort. That is why the market of such systems has significantly increased the last years in France. The use of a cooling system avoid therefore thermal comfort problems but leads to a more important cooling energy demand!

Figure 7 shows the evolution of the cooling energy demand in Nice for each month according to the test case. Case 2 is the reference case 1 using a cooling system. It can be underlined that according to the month the effects of the night ventilation (case 4) and the PCM/air heat exchanger (case 5-10) are not the same. In May and September the effects on the cooling demand are the same for the night ventilation and for the PCM system. It can be noted that nor the night ventilation nor the PCM system reduce the cooling energy demand in May. In June, the effect due to the

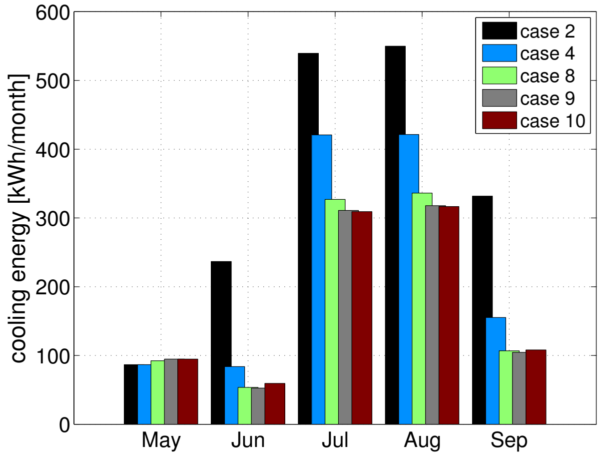

Figure 7: Monthly cooling energy demand in Nice

night ventilation is better than the one due to the PCM system. On the other hand, in July and August effects of both systems are the same. The cooling energy savings, except May, can vary from 70% to 45%. Using night ventilation of 6 volume per hour leads to a yearly energy demand of $5.1\ kWh.m^{-2}.year^{-1}$ whereas the use of the PCM/air heat exchanger leads to a yearly energy demand of $4\ kWh.m^{-2}.year^{-1}$ for the best case. Compared to the reference system the yearly cooling demand can be reduced by 46%.

Figure 8 shows the evolution of the cooling energy demand in Trappes for each month according to the test case. Even if the cooling energy demand is lower in Trappes than in Nice, the same conclusion can be done concerning the effect of the night ventilation or the PCM system.

Whatever the climate the figures 7 and 8 show that the temperature of fusion of the PCM has no significant effect on the cooling demand.

Figure 9 presents the peak cooling demand obtained

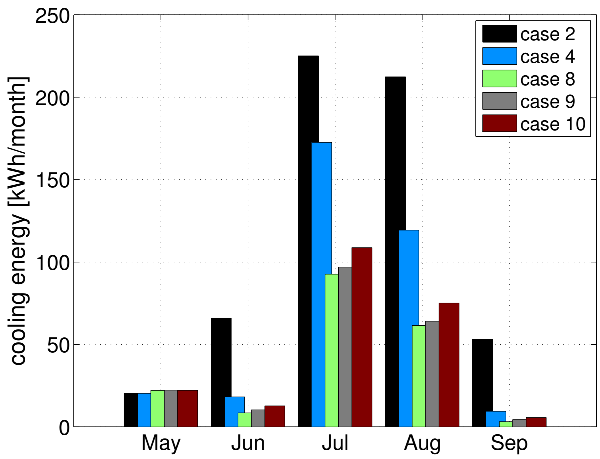

Figure 8: Monthly cooling energy demand in Trappes

in each test case and for both climate. For Trappes the PCM system, whatever the temperature of fusion, considerably reduces the needed power of the system. The needed power is therefore about two times less in case 9 than in case 2. It is the same in Nice but due to important cooling needs the power reduction is less significant and is equal to $P_{max} = 6.8\ kW$ in the best case compare to the $P_{max} = 9.5\ kW$ in the reference case. As it was underlined in the previous section the results does not take into account the extra electrcity consumption of the night ventilation.

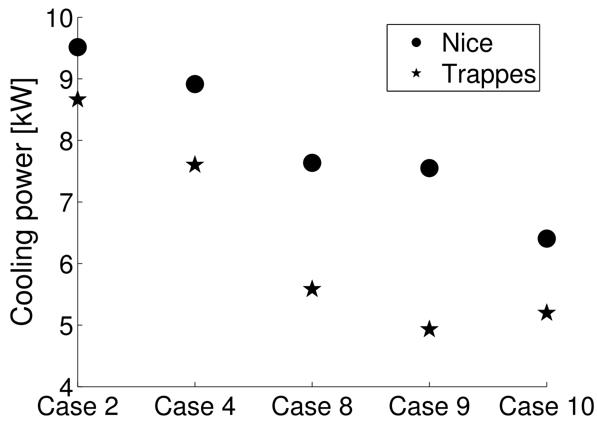

Figure 9: Peak cooling demand

Sensitivity to the convective heat transfer coefficient

Looking the previous results, the efficiciency of the PCM system is not sufficient due to low heat transfers between the air and the PCM. As presented in the numerical modelling section, the convective heat transfer depends on the flow regime. The analysis of the experimental configuration gives for an air rate change of 0.6 volume per hour a convective heat transfer of 5.5 $W.m^{-2}.K^{-1}$. The night ventilation at 3 volume per hour leads to a convective heat transfer of 6.16 $W.m^{-2}.K^{-1}$ while for a night ventilation at 6 volume per hour, it leads to 12.4 $W.m^{-2}.K^{-1}$. The velocities inside the PCM/air heat exchanger are respectively 0.193 $m.s^{-1}$, 0.96 $m.s^{-1}$ and 1.93 $m.s^{-1}$. This analysis also underlines that the flow regime is turbulent only for 6 volume per hour with a Reynolds number equal to 4551. Therefore it is interesting to know the influence of the convective heat transfer coefficient. The following figures show the impact of this coefficient for the climate of Nice. It is respectively fixed to 20 $W.m^{-2}.K^{-1}$ and 40 $W.m^{-2}.K^{-1}$ for an air change rate of 6 volume per hour, a temperature of fusion fixed at 21°C and with/without cooling system. The figure 10 shows that whatever the convective heat transfer coefficient used, there is no significant impact on the operative temperature since it is the same as case 7. With cooling system (figure 11) the results are the same. There is no significant impact on the cooling demand.

The figure 12 shows the evolution of the outside temperature T_{ext}, the inside air temperatures $T_{int-h=20}$ and $T_{int-h=40}$ and the supply air temperatures of the PCM/air heat exchanger $T_{shx-h=20}$ and $T_{shx-h=40}$.

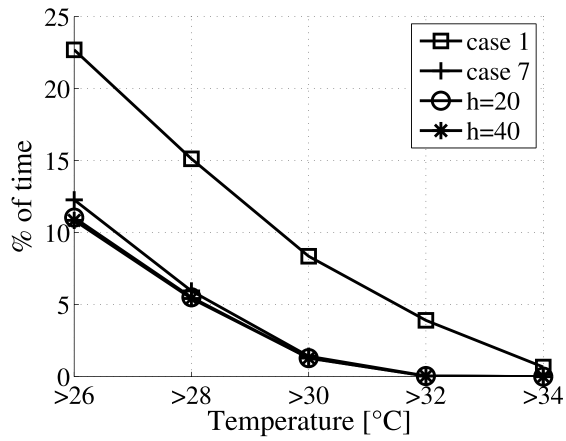

Figure 10: Operative temperature as a function of the convective heat transfer coefficient

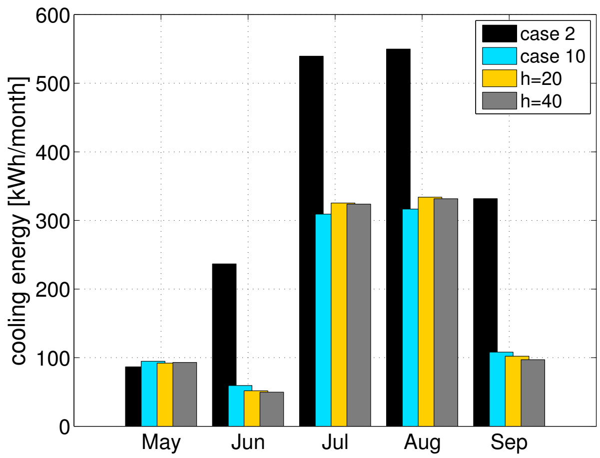

Figure 11: Cooling energy demand as a function of the convective heat transfer coefficient

The indices $h = 20$ and $h = 40$ give the convective heat transfer coefficient. The analysis of these temperatures underlines that the difference between the two supply air temperatures T_{shx} are too small to influence the air temperature. Therefore very small differences can be noted between $T_{int-h=20}$ and $T_{int-h=40}$. Note that the bigger day/night external temperature difference the bigger the T_{shx} difference. It can be also noted that at time 4733 and 4757, the supply temperatures during this period are nearly equal to 20/21°C due to a high outside temperature T_{ext}. This result indicates that the available latent energy is thus null. At time 4742, the temperature $T_{shx-h=20}$ is lower than $T_{shx-h=40}$. This shows the importance of the history of the phase change. Indeed, before this time the high convective heat transfer coefficient has permitted to melt all the PCM and thus all the enthalpy has been used. With the lower convective heat transfer coefficient $h = 20$ the PCM was not completely melt at time 4742. Therefore the temperature $T_{shx-h=20}$ becomes lower than $T_{shx-h=40}$. Integrating all these phenomena, the results detailed with figures 10 and 11 can be well understood, especially in July and August where the results are better with the lower convective heat transfert coefficient. During July and August, the PCM is always melt and therefore there is no latent effect. It seems that the effect of the PCM/air heat exchanger will be greater with another climate. Indeed if the climate presents greater day/nigth fluctuations of the outside temperature, the PCM would melt/cristallize easily. Moreover the regulation of the air change rate has to be optimized.

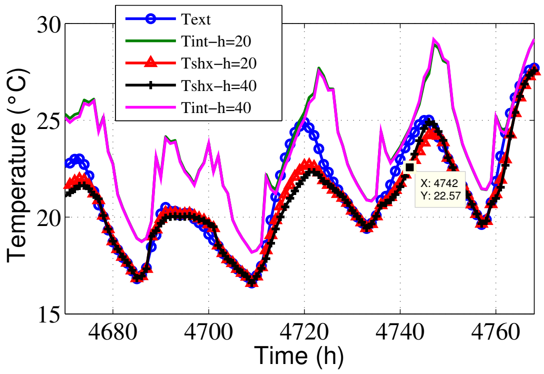

Figure 12: Temperature evolutions during some days in June

CONCLUSIONS

Simulations studies have been performed for a typical single family house which has been refurbished to have a low consumption building. Due to thermal comfort problems inside this building, a PCM/air heat exchanger has been used to find out how it affects the operative temperature T_{op}, the cooling load Q and the peak cooling demand P_{max} under the climatic conditions of Nice and Trappes, two french cities.

First, the results show that night ventilation, without the PCM/air heat exchanger, is effective in reducing the percentage of time where the operative temperature is over 26°C. It varies from 23% to 12.5% with a constant air change rate of 6 volume per hour during the summer period, for the climatic conditions of Nice. Due to lower thermal comfort problems, the results for Trappes are less significant! The effect of the PCM system in reducing the percentage of time over 26°C or 28°C is however less significant. The results also show the gain in comfort based on the adaptative comfort method. If the reference system leads to high thermal discomfort, the PCM/air heat exchanger coupled with the night ventilation allows to improve significantly the thermal comfort during the summer period. Indeed, with such a system the operative temperature is regularly within the acceptability limit (80% : $T_{conf} + 3.5°C$).

Then, the results show that the PCM/air heat exchanger system is effective to reduce the cooling energy demand. If the PCM system effect was not really significant on the operative temperature, the use of a cooling system increases substantially the efficiency of the PCM system. Thus, the yearly cooling demand can be reduce by up 46% under the climatic conditions of Nice. Moreover the results underline the efficiency of the PCM system to reduce the peak cooling demand leading to cheaper cooling system. So, in Nice the peak cooling demand P_{max} is reduced to $6.8\ kW$ against $9.5\ kW$ in the reference case. The results are better in Trappes due to lower cooling demand.

Finally, the results show that the convective heat transfer coefficient has no sighnificant influence. This phenomenon has been explained with the temperature evolutions. Indeed, it was underlined fusion/cristallisation problems. The history of the melting/critalization has a great affect on the phase change process thus to have a great convective heat transfer coefficient is not necessary a good way. Therefore more analysis is needed especially on the control command of the system. This will avoid to reduce the efficiency of the PCM system with high convective heat transfer coefficient.

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