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ABSTRACT

The growing usage of heat pumps increases the global electricity demand. During winter, the external temperature peak is at noon whereas electricity peak is at the end of the afternoon. Latent heat storage can be used to benefit from the time lag between these two periods. In this application phase change material (PCM) is integrated in the condenser of an air-to-air heat pump. The condensing temperature is used to melt down the PCM at noon. Then, during electricity peak load, the compressor is turned off and the indoor air of the house can be heated by exchanging heat with the melted PCM through the condenser. The condenser design is important to fulfill heating requirements since PCMs have in general low thermal conductivity. A dynamic model using Dymola is applied to size and simulate a prototype. Results show that a well sized heat exchanger can fulfill any heating requirements, however it is voluminous compared to an equal capacity regular condenser.

1. INTRODUCTION

Electricity supply and demand are subject to uncertainties (unavailability of production means, climate), since electricity cannot be stored easily, it is difficult to ensure their adequacy in all circumstances. Today in France, the main variation imposed on electricity consumption depends on the outdoor temperature with sensitivity up to 2100 MW.°C^{-1} in winter whereas it was less than 1500 MW.°C^{-1} a few years ago. Primarily responsible for this thermal sensitivity is the Joule heating, it is then urgent to find alternative controllable solutions. Possible solutions may lead to seek new sources of cutting off through intelligent management systems.

This study aims at the development of a solution coupling an air-to-air heat pump and a thermal storage capacity of phase change materials (PCMs) to meet the peak cut-off between 6:00 pm and 8:00 pm. In addition to the peak cut-off, this study seeks to improve the heat pump energy performance. In fact, this is possible, and consistent with the cut-off, by operating the heat pump during the outdoor highest temperature of the day (just after noon). Besides, the storage capacity of the thermal energy improves the seasonal coefficient of performance (COP) by a more continuous operation of the heat pump for low heat loads.

The solution consists of a condenser design that allows heat transfer between PCM, air, and refrigerant with low temperature difference.

2. STATE OF ART

Several studies have been done on residential heating systems with thermal storage. Many systems coupling heat pumps to latent heat storage systems are studied, like Long et al. (2008) for an air-to-water heat pump producing hot water. Agyenim et al. 2010 use latent heat storage for residential heating by means of horizontal cylinder shape PCM tank (paraffin) absorbing and releasing through internal water tubes.

Stirih (2003; 2004) designed two paraffin containers to store latent heat useful in residential applications. The first storage system is to incorporate paraffin between the layers of building external walls (Stirih 2003). The second tank has a rectangular shape, and exchanges heat with water circulating on one of its facets (Stirih 2004).

The PCM integration in air ducts is also used for latent heat storage. Dolado et al. (2011) and Zukowski (2007) studied the heat transfer between air and PCM for different structures. These studies led to the conclusion that the air must be blown with higher speeds than conventional to ensure desired thermal capacities. This leads to greater fan consumption, higher noise levels, and poorer exergy and energy efficiencies.

Literature allows concluding that the main problem of PCMs is their low thermal conductivity, which limits the absorbing and releasing of the thermal capacity.

3. Numerical modeling (Finite volumes)

In order to choose a suitable design of the condenser, the heat transfer inside the PCM is modeled. A widely used method to model the phase-change phenomenon (Dumas, 2002) considers the whole volume as a single phase with highly depending thermo-physical properties on temperature. It consists of solving the energy equation for each elementary volume. This method is particularly well suited to problems involving the use of PCM mixtures where the phase change occurs over a temperature range. The PCM physical properties, such as density, thermal conductivity, and specific heat, depend on its physical phase. Thus, a variable describing the physical state of each elementary volume must be introduced: the liquid quality, which is the ratio of the liquid mass on the total mass.

$$v_i = \frac{m_{L\,i}}{m_{L\,i} + m_{S\,i}} = \frac{m_{L\,i}}{m_i} \tag{1}$$

$$v_i = \frac{H_i - H_S}{H_L - H_S} = \frac{H_i - H_S}{h_{LS}} \tag{2}$$

The physical properties are equal to

$$\varphi_i = v_i\, \varphi_L + (1 - v_i)\varphi_S \tag{3}$$

The internal heat transfer flux is proportional to the temperature gradient in the material, while the phase change takes place at constant temperature. This complexity could be simplified by using the "enthalpy form" of the energy equation (Equation 1.4), because the variable enthalpy remains monotonic (strictly increasing or decreasing) during the phase change. The energy equation in the solid and liquid phases is expressed as follows:

$$\rho \frac{\partial H}{\partial t} = \frac{1}{x^m} \frac{\partial}{\partial x}\left(\lambda_i x^m \frac{\partial T}{\partial x}\right) \tag{4}$$

Where i refers to liquid or solid phase and m = 0, 1 or 2 depending on the geometry (plane, cylindrical or spherical).

Temperature (T) is expressed as function of enthalpy (H).

$$T = \begin{cases} T_f + \dfrac{H}{Cp_S} & pour\ H < 0 \\ T_f & pour\ 0 < H < h_{LS} \\ T_f + \dfrac{H - h_{LS}}{Cp_L} & pour\ H > h_{LS} \end{cases} \tag{5}$$

Figures 1 and 2 show rectangular and cylindrical meshed volumes of thickness e. Examples of internal heat transfer and boundary conditions are also illustrated.

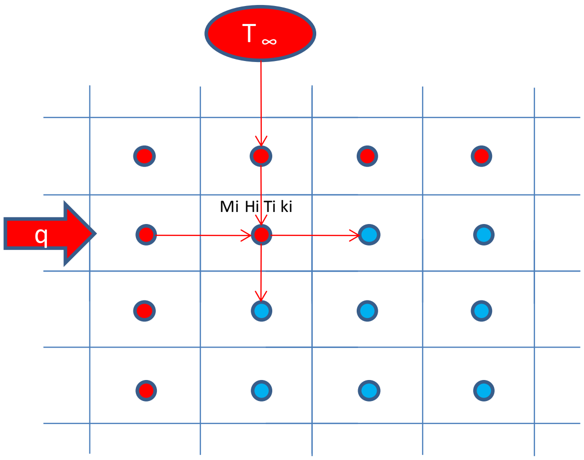

Figure 1: Rectangular volume mesh

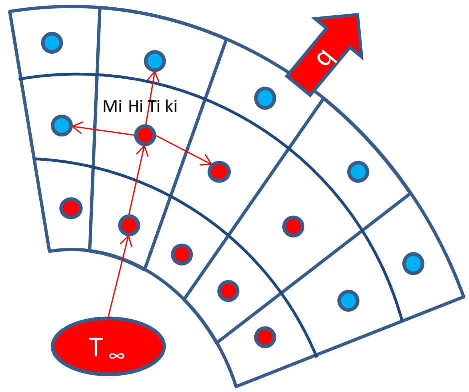

Figure 2:Cylindrical volume mesh

The thermal resistance between two adjacent cells having same dimensions is:

$$R_{ij} = \frac{2}{\lambda_i + \lambda_j} * \frac{d_{ij}}{S_{ij}} \tag{6}$$

The boundary conditions can be external heat flux and / or an imposed wall temperature.

4. Heat-exchanger design

4.1 Energy and capacity needs

For a 100-m^2 low energy house, a stock of 4000 Wh heat energy is sufficient to ensure thermal comfort between 6:00 pm and 8:00 pm during the coldest day. The rest of the days, the cut-off period will necessarily be greater than two hours. The system must be capable to deliver 2000 W heat capacity at maximum.
Conduction inside the PCM is the limiting factor for energy transfer in the system. To solve this problem, it is necessary to design a heat exchanger in which the PCM will be well distributed on small thicknesses. This allows generating the requested thermal capacity. However, a very small thickness distribution requires voluminous structure. For this reason, the PCM layer should have the maximum thickness that can deliver thermal requirements during the release. In the case of cylindrical or finned structures, the thickness considered is "the equivalent thickness".
The multi-split heating system has six indoor units. Therefore each of these units must be able to store around 670 Wh heat energy and to deliver a maximum thermal power of about 335 W during all the releasing period.

4.2 Sizing the heat exchanger

A heat exchanger (HX) that meets the requirements listed previously is designed. Its structure allows heat exchange between each two of the three active components: PCM, refrigerant, and air. Exchange surfaces are doped with fins to increase the capacity of absorbing and releasing heat: internal fins for PCM and external ones for air. It can contain 12.25 liters of PCM. The used PCM is paraffin with characteristics presented in Table 1.

Table 1: Paraffin characteristics.

T_f (°C)	T_S (°C)	h_{LS} (kJ.kg^{-1})	Cp (kJ.kg^{-1}.K^{-1})	ρ_S (kg.l^{-1})	ρ_L (kg.l^{-1})	λ (W.m^{-1}.K^{-1})
35.0	36.0	160	2.00	0.88	0.76	0.20

The heat energy amount that can be stored is the sum of metal sensible heat, PCM latent and sensible heat. This sum is about 670 Wh for HX temperature interval [20°C; 36°C]. The heat exchange surface between air and casing tubes is about 30 m².

4.3 Heat-exchanger behavior

The heat exchanger is simulated in releasing phase where the heat capacity is lower than at the absorbing phase.
The inlet air is blown at room temperature, 19°C. Two blowing speeds are investigated. The first regime consists in thermal release with maximum capacity where the air speed (V_{air}) at the fins is limited to 1 m.s^{-1} to avoid high noise level. The air volume flow rate is 750 m^3.h^{-1} and the external heat transfer coefficient is around 15 W.m^{-2}.K^{-1} (Hewitt). The second regime is studied for an air speed on the fins of 0.2 m.s^{-1}. The air volume flow rate is then 150 m^3.h^{-1} and the external heat transfer coefficient is around 5 W.m^{-2}.K^{-1}.

Paraffin particles solidify between 35°C and 36°C, considered 35 °C in the model. The simulation results with thermal release regime for V_{air} = 1 m.s^{-1} are shown in Figure 4. The blue curve represents the PCM critical point temperature (the last volume to solidify), and the purple one the air outlet temperature. The temperature difference is always associated with dominant paraffin due to its low thermal conductivity. The outlet air temperature slides from a maximum of 30°C at the beginning of the release to 20.3°C at the end. Thus the generated capacity decreases from 2700 W to 335 W at the end of the release (Figure 5).

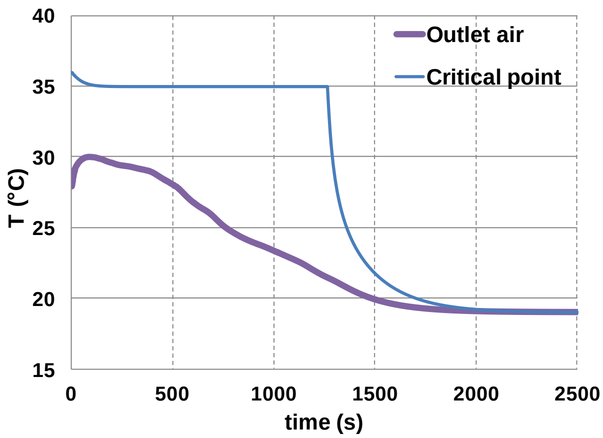

Figure 4: Paraffin critical element and outlet air temperatures.

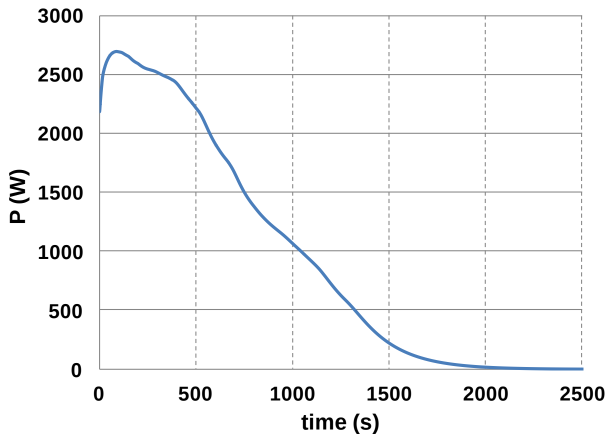

Figure 5: Generated capacity during the release for V_{air} = 1 m.s^{-1}.

In the second regime (0.2 m.s^{-1}) the temperature of the heated air reaches 34°C at the beginning of the release phase (Figure 6). The red curve represents the wall temperature. During the second half of the releasing period the

temperature inside the PCM is higher than the one between the casing tube and outlet air. This fact refers to the main problem of low PCM conductivity.
The size of the PCM mesh used for these two simulations is about 7 mm². Figure 7 shows the heat capacity curves in the second regime for a regular meshing (7 mm²) and a refined one (1.8 mm²). The absolute deviation is about 20 W, which validates the regular mesh.

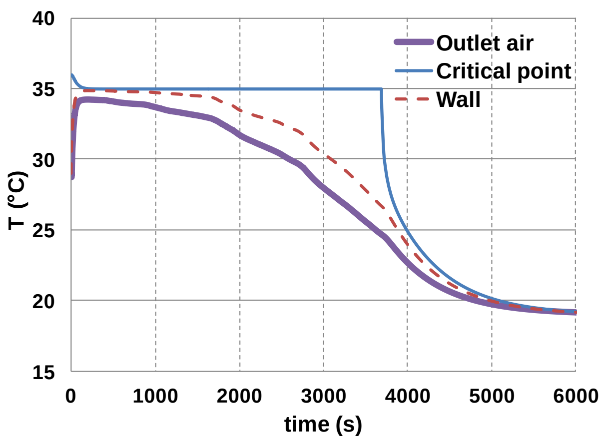

Figure 6: Paraffin critical element; wall and outlet air temperatures for $V_{air} = 0.2$ m.s^{-1}.

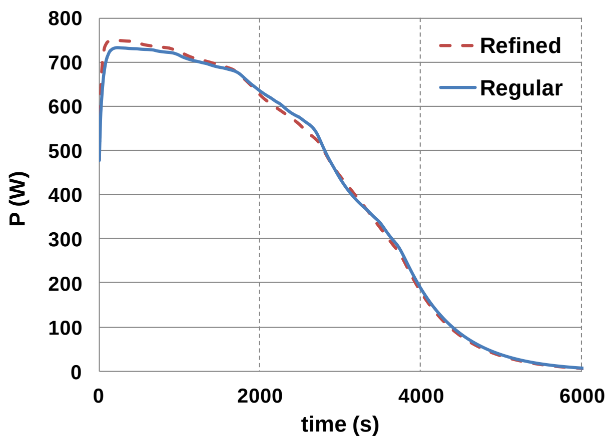

Figure 7: Generated capacity during the release for $V_{air} = 0.2$ m.s^{-1}.

4.4 Heat capacity control

As shown in the previous paragraph, the generated heat capacity decreases from a maximum, at the beginning of discharge period, to zero at its end. However, the dwelling heat load at the discharge period is quasi constant. Thus variable ventilation velocity should be used to get a constant discharge heat capacity. Figure 8 shows the volumetric air flow rate needed to ensure the 335 W at each time of the two hours discharging period. For the first 100 minutes, blowing air flow rate below 150 m^3.h^{-1} is sufficient to heat the dwelling. During the last 20 minutes the air flow rate must be increased from 150 m^3.h^{-1} to about 800 m^3.h^{-1}.

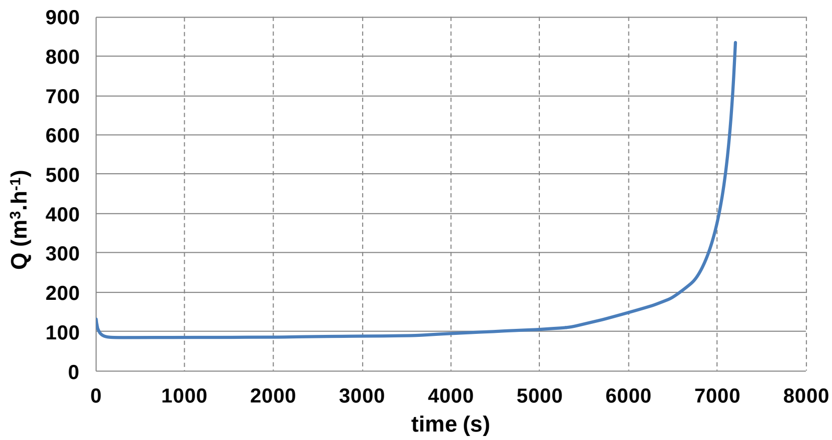

Figure 8: Volumetric air flow rate needed at the heat exchanger to ensure the 335 W.

6. CONCLUSIONS

- Heat pump coupled to a heat storage system allows a cut off during the maximum demand period and will improves automatically its performance.A dynamic model developed using Dymola is used to simulate heat transfer in regular and phase change materials.

- A condenser is designed to be able to store and deliver the required amount of heat energy.
- The limiting factor in latent heat storage systems is the low thermal conductivity of PCMs.
- A variable ventilation velocity should be used to ensure the constant heat capacity load.

NOMENCLATURE

Cp	Thermal heat capacity	$(kJ.kg^{-1}.K^{-1})$	**Subscripts**	
d	Distance between mesh	(m)	i	Volume line index
H	Enthalpy	$(kJ.kg^{-1})$	j	Volume column index
h_{LS}	Latent heat	$(kJ.kg^{-1})$	f	Fusion
m	Mass	(kg)	S	Solid phase, Solidification
S	Contact surface	(m^2)	L	Liquid phase
T	Temperature	(° C)	HX	Heat exchanger
v	Liquid quality	(-)		
λ	Thermal conductivity	$(W.m^{-1}.K^{-1})$	ϕ	Physical property

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