A THEORETICAL ANALYSIS OF THE USE OF PHASE CHANGE MATERIALS IN HOUSEHOLD REFRIGERATORS

Carla Soares Rametta, carla.rametta@polo.ufsc.br
Joel Boeng, joel@polo.ufsc.br
Cláudio Melo, melo@polo.ufsc.br
POLO – Research Laboratories for Emerging Technologies in Cooling and Thermophysics
Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, SC, Brazil

Joaquim Manoel Gonçalves, joaquim@polo.ufsc.br
IFSC - Federal Institute of Santa Catarina
Rua José Lino Kretzer, 608 – Praia Comprida, São José, SC, Brazil

Abstract. *Conventional household refrigeration systems have reached a point where considerable effort is required to achieve minimal energy savings. Thus, innovative solutions are welcome and the introduction of phase change materials (PCMs) - substances with a high heat fusion which are capable of storing and releasing large amounts of energy - are a good option for performance improvements. This paper introduces a theoretical analysis of the effect of incorporating PCMs into the refrigerated compartments and the condenser of a household refrigerator. From the literature it is clear that the use of PCMs contributes to the system performance, leading to energy savings. Moreover, the incorporation of PCMs into refrigerator compartments brings temperature stabilization, enhances the food preservation capability and keeps the internal temperature at an acceptable level for longer periods in cases of energy shortage. In this paper we describe how a PCM attenuates the difference between condensing and evaporating temperatures and the behavior of this attenuation as a function of the compressor run time ratio and the global heat transfer coefficients between the system components and PCMs. The coefficient of performance (COP) is evaluated and the values in an ideal cycle with and without PCMs are compared. It can be concluded that the addition of PCMs promotes a positive effect in the system, which can be quantified by the design characteristics of its components. The closer the compressor run time ratio is to unity, the smaller the effect of the PCM, this becoming ineffective in the case of variable capacity compressors (VCCs). It was also observed that the wrapping of the heat exchanger is the most efficient way to apply such materials, minimizing the heat transfer resistance.*

Keywords*: Phase change material, household refrigerator, theoretical analysis*

1. NOMENCLATURE

Variables

a	attenuation, -
COP	coefficient of performance, -
f	compressor run time ratio, -
Q	heat transfer rate, W
s	arbitrary parameter, -
t	time, s
T	temperature, K
u	arbitrary parameter, -
UA	global coefficient of heat transfer, $W.K^{-1}$
v	arbitrary parameter, -
η	reversible efficiency, -

Subscripts

A	ambient
C	condensation
cic	cyclic
cont	continuous
E	evaporation
H	heat exchanger
P	PCM
,P	with PCM
rev	reversible
w	thermal load

2. INTRODUCTION

A phase change material (PCM) is a substance with high heat fusion, which is capable of storing and releasing large amounts of energy while changing phase. This concept has been applied for millenniums, using ice/water as a natural PCM. This technology has more recently been developed to store the maximum amount of thermal energy in order to improve the performance and reliability of energetic systems. The appropriate selection of a PCM for a certain application is mandatory, since there will be no phase change if the PCM does not reach its melting point, and only sensible heat will be stored, jeopardizing the improvement expected in the system.

Over the past decade, the application of PCMs in heat exchangers or in the refrigerated compartments of household refrigerators to improve the energetic efficiency has grown significantly. Recently, Yusufoglu et al. (2015) evaluated the performance of two refrigerator models with different PCMs attached to the evaporator. The study indicated a significant reduction in the compressor run time with consequent energy savings of up to 9.4%. In a similar study, Azzouz et al. (2009) evaluated the effect of using a thick slab of PCM behind the evaporator. The authors reported an enhancement of the system performance of 10 to 30%, depending on the thermal load. In addition, a reduction in the temperature fluctuations and a longer period of acceptable temperature levels without electrical supply were observed. In contrast to Yusufoglu et al. (2015) and Azzouz et al. (2009), Sonnenrein et al. (2014) evaluated the impact of using PCM in the condenser of a household refrigerator. The results indicated a reduction in the average condensation temperature during the entire run time of the compressor. A large part of the heat rejection was taking place during the compressor off time since the energy absorbed by the PCM, during the compressor on time, was slowly released during the off periods. Moreover, the authors reported an energy saving of around 10%.

As mentioned by many authors, the use of PCMs in refrigerated compartments or attached to heat exchangers contributes to (i) enhancing the system performance, (ii) reducing the energy consumption, (iii) stabilizing the internal temperatures, since it leads to a high thermal inertia of the system, (iv) decreasing the temperature stratification, and (v) keeping the internal temperature at acceptable levels for a longer period in the case of energy shortage. Based on these previous findings, the application of PCMs in household refrigerator was the object of this study and this paper introduces a theoretical analysis of the thermal effect of these materials in refrigerator compartments and heat exchangers.

3. PHENOMENOLOGICAL ANALYSIS

In a household refrigerator, the compressor is designed to provide a cooling capacity that overcomes the thermal load imposed on the system. This strategy aims at a faster achievement of the steady state regime after the compressor starts, and also allows proper operation under more severe conditions of thermal load, such as high ambient temperature and product loading. Therefore, to compensate the excessive refrigerating capacity, the compressor works in a cyclic regime, i.e. on/off regime. Thus, the heat transfer between the heat exchanger and the ambient (Q_{cic}) takes place only during the compressor on time. However, during the off periods, the heat transfer is neglected and the heat exchangers become inactive.

In a cyclic regime, the application of a PCM in heat exchangers allows continuous heat transfer through it, as if the heat exchangers were kept constantly active. This can be explained by the capacity of the PCM to absorb or release heat at a constant temperature, working as a regenerator between the heat exchanger and the ambient. Therefore, heat transfer rate between the PCM and the air (Q_{cont}) is continuously distributed in the cycle. However, the heat transfer between the heat exchanger and PCM occurs only during the compressor on time, as shown in Fig 1.

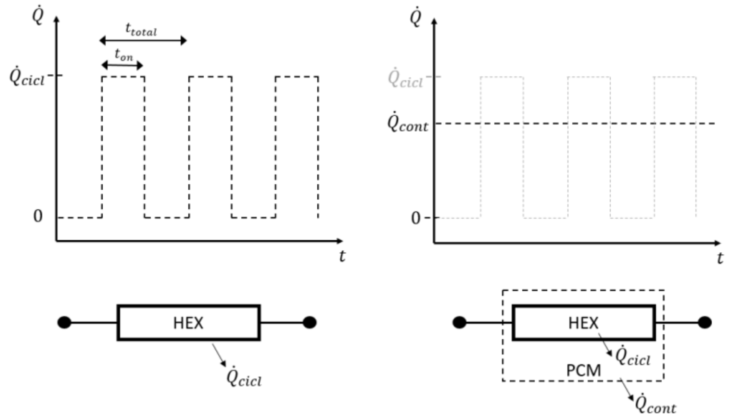


Figure 1 – Heat transfer rate (a) without PCM and (b) with PCM.

The application of a PCM provides continuous heat transfer during the entire cycle, in contrast to a cyclic regime. This means that the absolute heat transfer rate is reduced ($Q_{cont} < Q_{cic}$), decreasing the difference between the heat exchanger and ambient temperatures ($\Delta T_{AH,P} < \Delta T_{AH}$). In summary, the use of a PCM in a condenser or evaporator results in a lower condensation temperature or higher evaporation temperature, respectively. In this context, the attenuation (a) can be defined by considering the values for the difference between the temperatures of the heat exchanger and the ambient in systems with and without a PCM. As shown later, this attenuation positively affects the coefficient of performance of the system. Specifically, the attenuation (a) can be defined as:

$$a = 1 - \frac{\Delta T_{AH,P}}{\Delta T_{AH}} \tag{1}$$

4. THEORETICAL ANALYSIS

Figure 2(a) represents a typical household refrigerator without a PCM where the heat transfer rate between the ambient and the heat exchanger (Q_{AH}) is correlated with the thermal load imposed on the system by the compressor run time ratio. Equation (2) summarizes the energy balance, where f is defined as the ratio between the compressor on time and the entire time of the cycle.

$$\dot{Q}_W = f\,\dot{Q}_{AH} = fUA_{AH}(T_A - T_H) \tag{2}$$

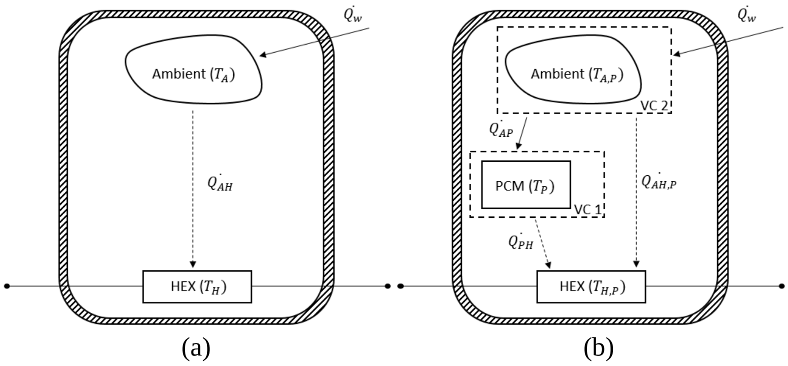


Figure 2 – Energy balances (a) without a PCM and (b) with a PCM in the refrigerated compartment.

Similarly, Fig. 2(b) shows the heat transfer rates when a PCM is inserted into the refrigerated compartment. The heat transfer between the heat exchanger and the PCM (Q_{PH}) and that between the heat exchanger and the ambient with a PCM ($Q_{AH,P}$) occurs only when the compressor is on. However, the heat transfer between a PCM and the ambient (Q_{AP}), as well as the thermal load (Q_W), are continuously distributed through the cycle. Taking this into account, Eq. (3) represents the energy balance for control volume 1, which characterizes the PCM inserted in the refrigerated compartment. The heat transfer rates Q_{AP} and Q_{PH} were defined considering the global coefficients of heat transfer, thus:

$$UA_{AP}(T_{A,P} - T_P) = fUA_{PH}(T_P - T_{H,P}) \tag{3}$$

On rearranging Eq. (3) for the PCM temperature, we obtain:

$$T_P = \frac{UA_{AP}T_{A,P} + fUA_{PH}T_{H,P}}{fUA_{PH} + UA_{AP}} \tag{4}$$

On the other hand, the energy balance for volume control 2 provides the relationship between the thermal load and the heat transfer rates:

$$\dot{Q}_W = UA_{AP}(T_{A,P} - T_P) + fUA_{AH,P}(T_{A,P} - T_{H,P}) \tag{5}$$

On rearranging the Eq. (5) and dividing by UA_{AP}, we obtain:

$$\frac{fUA_{AH,P}T_{H,P}}{UA_{AP}} = \frac{fUA_{AH,P}T_{A,P}}{UA_{AP}} + T_{A,P} - T_P - \frac{\dot{Q}_W}{UA_{AP}} \tag{6}$$

Substituting Eq. (4) in Eq. (6) gives:

$$\frac{fUA_{AH,P}T_{H,P}}{UA_{AP}} = \frac{fUA_{AH,P}T_{A,P}}{UA_{AP}} + T_{A,P} - \frac{UA_{AP}T_{A,P} + fUA_{PH}T_{H,P}}{fUA_{PH} + UA_{AP}} - \frac{\dot{Q}_W}{UA_{AP}} \tag{7}$$

In this context, the dimensionless ratios for the global coefficients of heat transfer are defined by Eqs. (8), (9) and (10):

$$s = \frac{UA_{AH}}{UA_{AP}} \tag{8}$$

$$u = \frac{UA_{PH}}{UA_{AP}} \quad (9)$$

$$v = \frac{UA_{AH,P}}{UA_{AP}} \quad (10)$$

In addition, replacing equations (9) and (10) in Eq. (14):

$$(T_{A,P} - T_{H,P}) = \frac{\dot{Q}_W}{UA_{AP} f (v + \frac{u}{1+fu})} \quad (11)$$

Finally, according to the definition of attenuation given in Eq. (1) and replacing Eqs. (2) and (8) in Eq. (11), the attenuation can be rewritten as:

$$a = 1 - \frac{s}{v + \frac{u}{1+fu}} \quad (12)$$

Equation (12) represents the ordinary case that defines the temperature attenuation as a function of the global coefficients of heat transfer of heat exchangers and PCM and the compressor run time ratio.

The dimensionless parameter u defines the ratio between the global coefficients of heat transfer between the heat exchanger and the PCM and that between the refrigerated compartment and the PCM. In the specific case where a PCM is inserted into the refrigerated compartment, it can be easily concluded that the parameter u is unity. Thus, the attenuation is expressed by Eq. (13).

$$a_u = 1 - \frac{s}{v + \frac{1}{1+f}} \quad (13)$$

On the other hand, the dimensionless parameter v defines the ratio between the global coefficients of heat transfer between the refrigerated compartment with a PCM and the heat exchanger and that between the refrigerated compartment and the PCM. If the PCM is wrapped around the heat exchanger, there is no heat transfer between the ambient and the heat exchanger, i.e. $UA_{AH,P}=0$, and the parameter v is null. Moreover, the dimensionless parameter s defines the ratio between the global coefficients of heat transfer between the refrigerated compartment without a PCM and the heat exchanger and that between the refrigerated compartment and the PCM. When the PCM is wrapped around the heat exchanger, a valid hypothesis is to assume that $UA_{AH} \approx UA_{AP}$, since the PCM acts as a heat exchanger, the heat exchange area and airflow behavior varying minimally. Therefore, assuming $s=1$:

$$a_{v,s} = 1 - (f + \frac{1}{u}) \quad (14)$$

Equation (14) shows that the attenuation is higher when the compressor run time decreases and parameter u increases. This parameter is directly proportional to the global coefficient of heat transfer between the PCM and the heat exchanger (UA_{PH}). Therefore, it can be concluded that the positive effects of a PCM are potentialized when the compressor run time is low and the thermal resistance between the PCM and the heat exchanger is at a minimum.

When the PCM is wrapped around the heat exchanger, a reasonable hypothesis is to assume that the coupling between the PCM and the heat exchanger is ideal, i.e. the thermal resistance between them is close to zero ($u \to \infty$). In this case, the attenuation is dependent on the compressor run time ratio only, as follows:

$$a_{v,s,\infty} = 1 - f \quad (15)$$

In a system operating in a cyclic regime, where the average compressor run time is always lower than 100%, the PCM provides the attenuation of the difference between the heat exchanger and the ambient temperatures. Therefore, the lower the compressor run time ratio the higher the attenuation will be. However, if the compressor run time ratio is close to or equal to 100%, a classic case when a variable capacity compressor is used, the attenuation is minimal or null.

In summary, the attenuation provided by the PCM can be estimated by considering the global coefficients of heat transfer and the compressor run time ratio. Table 1 summarizes the attenuation for each possible configuration.

The dimensionless parameters *u*, *v* and *s* can be estimated for cases other than that shown in Table 1, covering the whole application range. The parameters *v* and *s*, for example, can be calculated from the ratio between the global coefficient of heat transfer of the heat exchanger (typical values are in the range of 10 to 25 W/K for the evaporators of household refrigerators) and the global coefficient of heat transfer of the PCM. When the PCM is inside the refrigerated compartment, it is subjected to a convective coefficient in the range of 10 to 15 W/m².K. Therefore, a rough estimate provides values of *v* and *s* in the range of 1 to 10. It is worth emphasizing that both dimensionless parameters are inversely proportional to the convection imposed on the PCM. Table 2 shows the typical values of dimensionless parameters for different applications.

Table 1 – Attenuation provided by the PCM application

1	General case		$a = 1 - \dfrac{s}{v + \dfrac{u}{1+fu}}$
2	PCM inside refrigerated compartment (uncoupled)	$u = 1$	$a_u = 1 - \dfrac{s}{v + \dfrac{1}{1+f}}$
3	PCM coupled to the heat exchanger	$v = 0$ $s = 1$	$a_{v,s} = 1 - (f + \dfrac{1}{u})$
4	Negligible thermal resistance between PCM and heat exchanger	$u \to \infty$	$a_{v,s,\infty} = 1 - f$

Table 2 – Typical values for dimensionless parameters

Dimensionless parameter	**PCM uncoupled from the heat exchanger**	**PCM coupled to the heat exchanger**
s	$1 < s < 10$	1
u	1	$u >> 1$
v	$1 < v < 10$	0

5. REVERSIBLE EFFICIENCY

The efficiency of a cooling system is evaluated by the coefficient of performance, which is defined as the ratio of the cooling capacity to the work provided by the compressor. The coefficient of performance of a reversible cycle can be defined based on the absolute temperatures of the heat exchangers, as shown in Eq. (16).

$$COP_{rev} = \frac{1}{\dfrac{T_C}{T_E} - 1} \tag{16}$$

Similarly to Eq. (16), the coefficient of performance for a reversible cycle using a PCM can be expressed by Eq. (17).

$$COP_{rev} = \frac{1}{\dfrac{T_{C,P}}{T_{E,P}} - 1} \tag{17}$$

Defining reversible efficiency (η_{PCM}) as the ratio between the reversible coefficients of performance for systems with and without a PCM, and combining Eqs. (16) and (17), gives:

$$\eta_{PCM} = \frac{COP_{rev,P}}{COP_{rev}} = \frac{\dfrac{1}{\dfrac{T_{C,P}}{T_{E,P}} - 1}}{\dfrac{1}{\dfrac{T_C}{T_E} - 1}} = \frac{\dfrac{T_C}{T_E} - 1}{\dfrac{T_{C,P}}{T_{E,P}} - 1} \tag{18}$$

As shown previously, the temperature difference between the heat exchanger and the ambient is attenuated when a PCM is used. Thus:

$$\eta_{PCM} = \frac{\frac{T_C}{T_E} - 1}{\frac{T_{C,P}}{T_{E,P}} - 1} > 1 \quad \therefore \quad \text{COP}_{\text{rev,P}} > \text{COP}_{\text{rev}} \tag{19}$$

Therefore, the addition of a PCM to the heat exchanger has a positive effect on the efficiency of the system, and this is directly proportional to the attenuation of the temperatures provided by the PCM.

The reversible efficiency of the PCM can be also defined by the compressor run time ratio and the dimensionless parameters *s*, *u* and *v*. In this case, Eq. (1) is rewritten to the condensation and evaporation temperatures obtained with the use of a PCM:

$$T_{C,P} = (1 - a_C) \cdot (T_C - T_{A,C}) + T_{A,C} \tag{20}$$

$$T_{E,P} = -(1 - a_E) \cdot (T_{A,E} - T_E) + T_{A,E} \tag{21}$$

Thus, replacing the Eq. (12) in Eqs. (20) and (21) gives:

$$T_{C,P} = \frac{s_C \cdot (T_C - T_{A,C})}{v_C + \frac{u_C}{1 + f \cdot u_C}} + T_{A,C} = \varphi_C + T_{A,C} \tag{22}$$

$$T_{E,P} = -\frac{s_E \cdot (T_{A,E} - T_E)}{v_E + \frac{u_E}{1 + f \cdot u_E}} + T_{A,E} = \varphi_E + T_{A,E} \tag{23}$$

The reversible efficiency of a PCM can be rewritten by replacing Eqs. (22) and/or (23) in Eq. (18). Thus, the values for the reversible efficiency when a PCM is used only in the evaporator, only in the condenser or in both heat exchangers, are given by Eqs. (24), (25) and (26), respectively:

$$\eta_{PCM,E} = \frac{\frac{T_C - T_E}{T_E} \cdot (T_{A,E} - \varphi_E)}{\varphi_E + T_C - T_{A,E}} \tag{24}$$

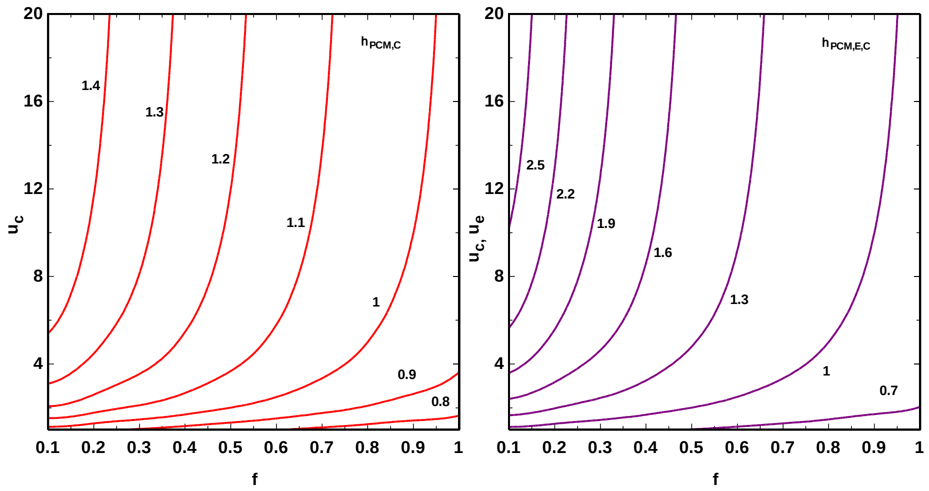
$$\eta_{PCM,C} = \frac{T_C - T_E}{\varphi_C + T_{A,C} - T_E} \tag{25}$$

$$\eta_{PCM,E,C} = \frac{\frac{T_C - T_E}{T_E} \cdot (T_{A,E} - \varphi_E)}{\varphi_C + \varphi_E + T_{A,C} - T_{A,E}} \tag{26}$$

The following figures illustrate the estimation of the reversible efficiency as a function of the compressor run time and the dimensionless parameters, which were estimated in agreement with the typical values for each case presented in Table 2. The temperatures were arbitrated as follows: T_C=*350K*, $T_{A,C}$=*310K*, T_E=*250K* and $T_{A,E}$=*280K*.

Figure 3a shows the reversible efficiency when a PCM is attached to only one heat exchanger, in this case the condenser. As expected, the reversible efficiency increases with a decrease in the compressor run time ratio. This occurs because the PCM has the potential to exchange heat with the ambient during the compressor off time. If the run time ratio is 100%, the PCM becomes ineffective. Refrigeration systems equipped with a variable capacity compressor (VCC) usually have a run time ratio close to 100%, associated with a low potential for improvement when a PCM is used. Moreover, when the parameter *u* is very low, i.e., there is inefficient heat transfer from the PCM to the heat exchanger, the reversible efficiency can be less than 1, indicating that the PCM works as a thermal insulator, reducing the heat transfer and resulting in a loss of efficiency compared with the system without a PCM. In summary, the higher the thermal resistance between the PCM and the heat exchanger the lower the reversible efficiency will be.

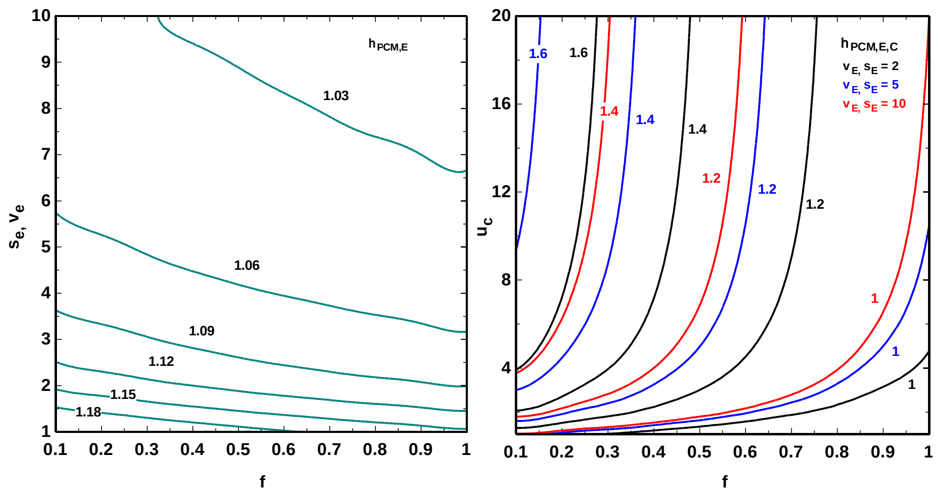
The reversible efficiency obtained when PCMs are coupled to both heat exchangers is shown in Fig. 3b. It can be seen that the behavior is similar to that of the configuration of the PCM attached to a single heat exchanger, however, in this case, the benefits are enhanced.



(a) (b)

Figure 3 –Reversible efficiency when PCMs are attached to (a) the condenser and (b) to both heat exchangers.

Figure 4a shows the reversible efficiency when a PCM is inserted into the refrigerated compartment. Once again, the higher the compressor run time ratio the lower the reversible efficiency will be. However, the dimensionless parameters *s* and *v* are inversely proportional to the convection imposed on the PCM uncoupled from the heat exchanger. Therefore, as expected, the reversible efficiency increases when the convection is higher. Furthermore, it is worth noting that the average value for the reversible efficiency when a PCM is inserted into the refrigerated compartment is much lower than that when the PCM is coupled to the heat exchangers. In other words, wrapping the heat exchanger in the PCM is the most effective way to use this material.



(a) (b)

Figure 4 – (a) Reversible efficiency when a PCM is inserted into the refrigerated compartment and (b) when PCMs are attached to the condenser and inserted into the refrigerated compartment.

Figure 4b shows the reversible efficiency when PCMs are coupled to a heat exchanger and inserted into the refrigerated compartment, as a function of the compressor run time ratio and the parameter u_C. The graph illustrates a range of v_E and s_E. For a fixed compressor run time ratio, the reversible efficiency decreases as the convection imposed on the PCM decreases.

Table 3 summarizes the effect on the reversible coefficient of performance of a refrigeration system operating with a run time ratio of 50%. Coupling PCMs to both heat exchangers can provide a theoretical benefit of up to 59.3%, in view of low thermal resistance between the PCM and the heat exchanger. In contrast, if the coupling is not ideal, the decrease in the COP can reach 30.4%, since the PCM acts as a thermal insulator, reducing the heat transfer of the heat exchanger. The table also shows that, if the PCM is inserted into the refrigerated compartment, the internal convection imposed through it has a significant impact on the system performance.

Table 3 - Theoretical effect on the reversible COP for different configurations.

	Low coupling thermal resistance	High coupling thermal resistance	
PCM coupled to the condenser	23.8%	-16.7%	
PCM coupled to the evaporator	23.6%	-18.3%	
PCMs coupled to both heat exchangers	59.3%	-30.4%	
PCMs coupled to the condenser and uncoupled from the evaporator	27.7%	-14.7%	Low internal air convection
	52.3%	-3.0%	High internal air convection
PCM uncoupled from the evaporator	19.1%	2.7%	
	High internal air convection	Low internal air convection	

6. CONCLUDING REMARKS

The theoretical analysis showed that the attenuation of the condenser and evaporator temperatures provided by the use of PCMs has a positive effect on the coefficient of performance, which occurs as a function of the compressor run time ratio and global coefficients of heat transfer. In this context, it can be concluded that:

- The attachment of PCMs to the heat exchangers has a positive impact on the efficiency of the system, which is directly proportional to the temperature attenuation;
- The reversible efficiency increases as the compressor run time ratio decreases. This occurs due to the capacity of the PCM to exchange heat with the system during the compressor off time. When the run time ratio is equal to 100% there is no gain in the coefficient of performance. Therefore, the insertion of PCMs into systems equipped with a variable capacity compressor provides a low level of improvement;
- If the heat transfer between the PCM and the heat exchanger is inefficient, the reversible efficiency can be less than 1, indicating that the PCM acts as a thermal insulator, decreasing the heat transfer and resulting in an efficiency loss compared with the system without a PCM. Thus, the higher the thermal resistance between the PCM and the heat exchanger, the lower the reversible efficiency will be;
- The insertion of a PCM into the refrigerated compartment brings benefits if the heat transfer is sized appropriately. The higher the convection imposed on the PCM the higher the reversible efficiency will be;
- It was observed that the reversible efficiency of the PCM inserted into the refrigerated compartment is much lower than the case where the PCM is coupled to the heat exchanger. Therefore, the most effective approach is to use the PCM coupled to the heat exchangers;
- When PCMs are coupled to both heat exchangers, the reversible efficiency is maximum;
- Assuming an arbitrary system operating under a run time ratio of 50%, PCMs coupled to both heat exchangers can bring a theoretical improvement of up to 59.3% in the reversible COP, when the thermal resistance between the PCM and the heat exchangers is low. However, if the coupling is not ideal, the decrease in the reversible COP can reach 30.4%.

7. REFERENCES

Azzouz K, Leducq D, Gobin D. Enhancing the performance of household refrigerator with latent heat storage: An experimental investigation, International Journal of Refrigeration, Vol. 32, Pages 1634-1644, 2009.

Sonnenrein G, Elsner A, Baunhogger E, Morbach A, Fieback K, Vrabek J. Reducing the power consumption of household refrigerators through the integration of latent heat storage elements in wire-on-tube condensers, International Journal of Refrigeration, Vol. 51, Pages 154-160, 2014.

Yusufoglu Y, Apaydin T, Yilmaz S, Paksoy HO. Improving performance of household refrigerators by incorporating phase change materials, International Journal of Refrigeration, Vol. 57, Pages 173-185, 2015.

8. RESPONSIBILITY NOTICE

The authors are solely responsible for the printed material included in this paper.