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A COMPUTATIONAL MODEL OF A PHASE CHANGE MATERIAL HEAT EXCHANGER IN A VAPOR COMPRESSION SYSTEM WITH A LARGE PULSED HEAT LOAD

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ABSTRACT

Phase change materials (PCMs) use latent heat to store a large amount of thermal energy over a narrow temperature range. While PCMs are commonly used for thermal storage applications, they may also be used to dampen large pulsed heat loads, which are commonly generated by high-power electronics and direct-energy weapons. During a pulse, the PCM absorbs some of the large heat load, and between pulses the heat is dissipated to a cooling system, which minimizes the instantaneous heat load applied to the cooling system, reducing its physical size and power consumption.

To minimize the size of a PCM heat exchanger, a simple computational model that can capture the transient thermal response of a flat plate PCM heat exchanger in a vapor compression cooling system with a pulsed heat load was developed. Using this model, the effect of PCM thermal conductivity, melt temperature, and latent heat on the size of the PCM heat exchanger was studied. PCM thermal conductivity and melt temperature had the greatest impact on the PCM heat exchanger size. The ideal PCM heat exchanger would contain relatively high thermal conductivity PCM with a melt temperature close to the desired heat source temperature.

Keywords: phase change material, energy storage, thermal damping, pulsed power

INTRODUCTION

Direct-energy weapons generate large, pulsed heat loads during operation. This heat load is typically characterized by a maximum heat load during firing that must be dissipated for a given duty cycle, followed by a fraction of the maximum heat load that must be dissipated for the remainder of the period (Figure 1). Since most direct-energy weapons must be mobile, these large heat loads must be rejected with a compact, lightweight cooling system.

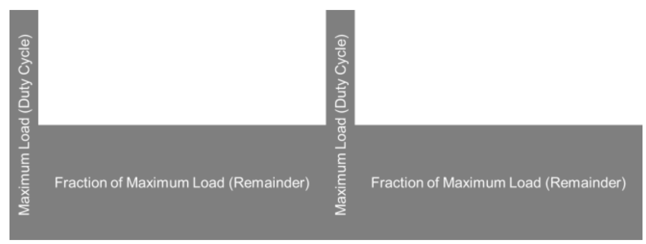

Figure 1. A visual representation of the pulsed heat load generated by a direct-energy weapon.

A vapor compression system is a potential cooling solution for dissipating high power transient heat loads. Typically, the vapor compression system would be sized for the maximum heat load that it will experience. For direct-energy weapons, this system would be relatively simple, but unnecessarily large, especially considering that it will only experience the maximum heat load for a fraction of the pulse duration. An alternative option would be to employ thermal damping and size the vapor compression system for the average heat load that it will experience during a pulse, resulting in a much smaller vapor compression system.

A schematic of a vapor compression system with a thermal damping component is shown in Figure 2. During firing, the large, transient heat load will be dissipated to the thermal damping component, which will store any heat that the vapor compression system cannot immediately absorb. Upon completion of the maximum heat load pulse, the thermal damping component will release the stored heat to the vapor compression system prior to the next cycle.

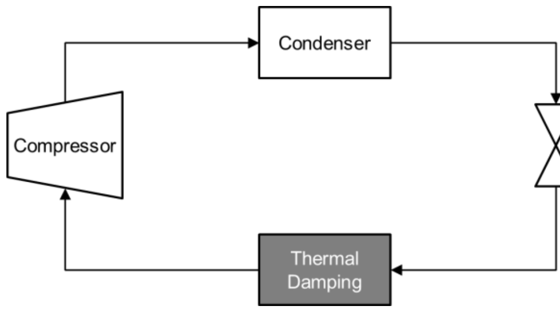

Figure 2. A schematic of a potential vapor compression cooling system for direct-energy weapons. This cooling system contains a thermal damping component to store excess thermal energy during large pulses.

For the particular direct-energy application investigated in this study, the thermal damping component will be a flat plate, 3-circuit heat exchanger filled with a paraffin-based PCM. Paraffin-based PCMs have melt temperatures within the range of -10 to 100˚C, and their latent heats are relatively large, typically between 150 and 250 $kJkg^{-1}$. However, one of their major drawbacks is their low thermal conductivity (~0.2 $Wm^{-1}K^{-1}$ [1]), which makes it difficult to effectively conduct heat through the PCM [1-4]. Graphite nanofibers have been impregnated in paraffin-based PCMs to boost their thermal conductivity, usually by an order of magnitude (2 $Wm^{-1}K^{-1}$ [5]).

In the PCM heat exchanger, the PCM will be sandwiched between coolant and two-phase refrigerant streams (Figure 3). The coolant stream will transport heat from the direct energy weapon to the PCM heat exchanger, where it is either stored or, if possible, rejected to the two-phase refrigerant. However, the weapon must be maintained at 30˚C, and therefore, the coolant stream must stay below this temperature.

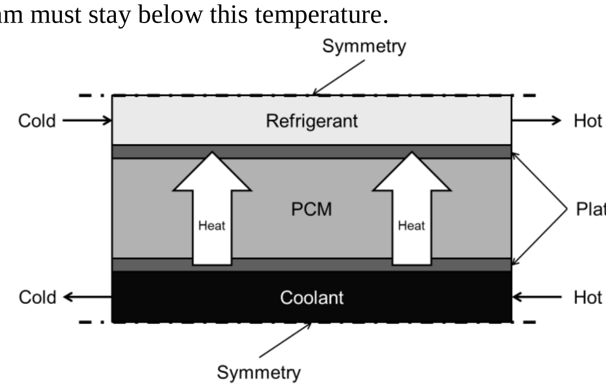

Figure 3. A single layer of PCM in the PCM heat exchanger. Coolant transfers heat from the direct-energy weapon to the PCM heat exchanger where it is either stored or rejected to the two-phase refrigerant.

The objective of this study is to determine the effect of PCM thermal conductivity, melt temperature, and latent heat on the size of the PCM heat exchanger for a representative set of requirements. This work will aid in minimizing the size of the PCM heat exchanger and therefore the vapor compression system. Furthermore, it will also provide a methodology for optimizing cooling systems for similar applications.

NOMENCLATURE

T – Temperature
y – Spatial y-direction
ρ – Density of the PCM
c – Temperature dependent specific heat of the PCM
k – Thermal conductivity of the PCM
Δh – Latent heat of the PCM
T_{melt} – PCM melt temperature
ΔT_{melt} – PCM melt temperature range
N_{layers} – Number of PCM layers
A – Area of a heat exchanger plate
$\bar{h}$ – Average heat transfer coefficient
l – Height of model domain (distance between plates)

METHODS

A one-dimensional model of the PCM heat exchanger was developed to determine the effect of PCM (1) thermal conductivity, (2) melt temperature, and (3) latent heat on its size.

The partial differential equation governing the PCM is:

$$\frac{\partial^2 T}{\partial y^2} = \frac{\rho c}{k}\frac{\partial T}{\partial t}$$

where k is the PCM's thermal conductivity, ρ is its density, and c is its temperature-dependent specific heat, which accounts for latent heat storage. As shown in Figure 4, a piecewise linear function was used to model the variation of specific heat with temperature. To generate this function, it was assumed that the phase change of the PCM occurred over a temperature range (ΔT_{melt}) surrounding the melt temperature (T_{melt}). The latent heat capacity of the PCM is the area underneath the specific heat/temperature curve and is shaded in Figure 4. If the latent heat capacity is known and a melt temperature range is assumed (3 K for the model), the magnitude of the specific heat at the peak of the piecewise linear function (c_{max}) can be determined.

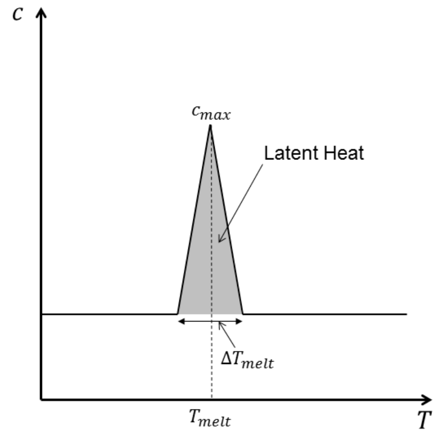

Figure 4. Variation of PCM specific heat with temperature. A piecewise linear function was used to account for the latent heat storage.

The model domain is shown in Figure 5. The height (l) of the domain is 2.4 mm, which is the distance between plates in a readily available 3-circuit flat plate heat exchanger [6]. The boundary conditions imposed on the model are shown in Figure 5. The bottom surface of the domain is exposed to a pulsed heat flux corresponding to the heat applied to the PCM heat exchanger via the coolant stream. For this particular application, the maximum heat load is applied for a 10% duty cycle followed by half of the maximum heat load for the remainder of the period. The heat flux at the bottom surface is calculated by dividing the heat transfer rate by the number of PCM layers (N_{layers}) and the area of a single heat exchanger plate (A), 0.128 m^2 for a readily available 3-circuit flat plate heat exchanger [6]. The top surface of the model domain corresponds to the two-phase refrigerant flowing through the vapor compression system and is modeled as a convection boundary condition. The average heat transfer coefficient, $\bar{h}$, is estimated from manufacturer's data [6]. It is assumed that the initial temperature of the PCM is 5°C, the evaporating temperature (T_{evap}) of the refrigerant.

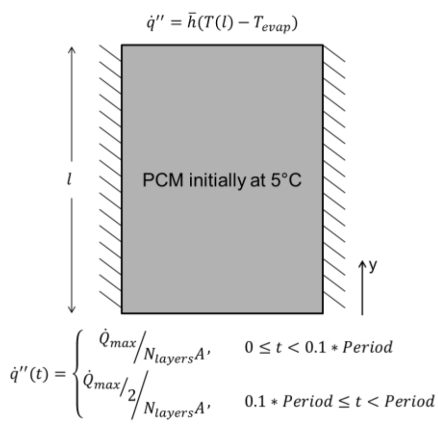

Figure 5. The PCM model domain with boundary conditions. The top surface is exposed to a pulsed heat flux representing the heat applied to the PCM heat exchanger via the coolant stream. The bottom surface represents two-phase refrigerant at constant temperature.

A parametric study was performed by varying the thermal conductivity (k), melt temperature (T_{melt}), and latent heat (Δh) of the PCM, as well as the number of PCM layers (N_{layers}). Since the manufacturer could not vary the separation distance between heat exchanger plates, the thickness of the PCM layer was not varied. Initially, a total of 600 runs were performed, with additional runs performed afterwards to more accurately identify trends. Table 1 shows the values of the parameters used in the model.

Table 1. Parameters used in the one-dimensional PCM heat exchanger model.

Parameter	Units	Value
Varied Parameters		
Thermal conductivity (k)	$Wm^{-1}K^{-1}$	0.2:0.7:3
Latent heat (Δh)	$kJkg^{-1}$	150:50:250
Melt temperature (T_{melt})	K	280:5:295
Number of PCM layers (N_{layers})	-	10:10:100
Constant Parameters		
Density (ρ)	kgm^{-3}	880 [8]
Specific heat of liquid and solid PCM (c)	$Jkg^{-1}K^{-1}$	810 [8]
Melt temperature range (ΔT_{melt})	K	3
PCM layer thickness (l)	mm	2.4 [7]
Area of a heat exchanger plate (A)	m^2	0.128 [7]
Average heat transfer coefficient ($\bar{h}$)	$Wm^{-2}K^{-1}$	1000 [7]
Heat transfer rate to PCM	kW	$\dot{Q}_{max}$ (10% duty) $\dot{Q}_{max}/2$ (remainder)

The model was solved using the method of lines - the spatial derivative was discretized and the resulting time derivatives were solved with Hindmarsh's ODE solver, LSODE, in GNU Octave [8]. After the parametric study runs were completed the results were separated based on three requirements:

1. The temperature at the bottom of the domain ($y = 0$) cannot exceed 30°C (the coolant temperature must be below the temperature specified by the design requirements)
2. The maximum temperature at any location within the domain at any time must exceed the melt temperature (the PCM must melt).
3. The temperature distribution must reach a periodic steady state.

Initially, the results were sorted based on the first two requirements. If those were satisfied, the average heat transfer rate to the refrigerant during a single pulse was calculated to verify that a periodic steady state was reached. To do this, the transient heat flux distribution was calculated by taking the gradient of the transient temperature distribution. This was used to determine the rate of heat transfer at the refrigerant surface ($\dot{q}(y = l)$). The rate of heat transfer at the refrigerant surface was then numerically integrated over a single pulse period to determine the total amount of energy transferred to the refrigerant during a pulse. This value was divided by the period to calculate the average rate of heat transfer to the refrigerant during a pulse. If this value equaled the average applied heat load, the results reached a periodic steady state and were considered viable; otherwise, the results were discarded.

RESULTS AND DISCUSSION

Using the results of the parametric study, the effect of PCM thermal conductivity, melt temperature, and latent heat, on the size (number of PCM layers) of the PCM heat exchanger was investigated.

Results

The impact of PCM thermal conductivity, melt temperature, and latent heat on PCM heat exchanger size is summarized in Table 2, which shows the normalized minimum number of PCM layers required for every combination of these parameters modeled (the scenario requiring the most layers had a thermal conductivity of 0.2 $Wm^{-1}K^{-1}$, melt temperature of 280 K, and latent heat of 150 $kJkg^{-1}$).

Table 2. The normalized minimum number of PCM layers required for every combination of thermal conductivity, melt temperature, and latent heat modeled.

T_{melt} (K)	150 $kJkg^{-1}$	200 $kJkg^{-1}$	250 $kJkg^{-1}$
0.2 $Wm^{-1}K^{-1}$			
280	1	0.99174	0.98347
285	0.88430	0.87603	0.86777
290	0.79339	0.78512	0.77686
295	0.71901	0.70248	0.70248
0.9 $Wm^{-1}K^{-1}$			
280	0.29752	0.29752	0.29752
285	0.27273	0.26446	0.26446
290	0.24793	0.23967	0.23140
295	0.22314	0.21488	0.21488
1.6 $Wm^{-1}K^{-1}$			
280	0.20661	0.20661	0.20661
285	0.20661	0.19835	0.19008
290	0.17355	0.16529	0.16529
295	0.16529	0.15702	0.14876
2.3 $Wm^{-1}K^{-1}$			
280	0.16529	0.16529	0.16529
285	0.16529	0.16529	0.16529
290	0.14050	0.14050	0.13223
295	0.13223	0.12397	0.12397
3 $Wm^{-1}K^{-1}$			
280	0.14876	0.14876	0.14876
285	0.14876	0.14876	0.14876
290	0.12397	0.12397	0.11570
295	0.11570	0.11570	0.10744

The most evident trend shown in Table 2 is that increasing the thermal conductivity of the PCM exponentially reduces the number of PCM layers necessary to meet the three requirements outlined in the Methods section. This is shown graphically in Figure 6 for a latent heat of 250 $kJkg^{-1}$. At thermal conductivities of 0.2, 0.9, 1.6, 2.3, and 3 $Wm^{-1}K^{-1}$, at least 0.70248, 0.21488, 0.14876, 0.12397, and 0.10744 normalized layers are respectively required at the highest melt temperature (295 K) and latent heat (250 $kJkg^{-1}$) (Figure 6). Similarly, at the lowest melt temperature (280K) and latent heat (150 $kJkg^{-1}$), at least 1, 0.29752, 0.20661, 0.16529, and 0.14876 normalized layers are required.

Although the size of the PCM heat exchanger is largely influenced by the thermal conductivity of the PCM, it is also affected by the melt temperature, particularly at low thermal conductivities. At a thermal conductivity of 0.2 $Wm^{-1}K^{-1}$ and latent heat of 250 $kJkg^{-1}$, the minimum number of normalized layers required at melt temperatures of 280, 285, 290, and 295 K are 0.98347, 0.86777, 0.77686, and 0.70248, respectively (Figure 6). At the same melt temperatures, configurations with a thermal conductivity of 0.9 $Wm^{-1}K^{-1}$ and a latent heat of 250 $kJkg^{-1}$ require at least 0.29752, 0.26446, 0.23140, and 0.21488 normalized layers, respectively. This trend continues at higher thermal conductivities, but becomes less pronounced. For instance, configurations with a thermal conductivity of 3 $Wm^{-1}K^{-1}$ and latent heat of 250 $kJkg^{-1}$ required 0.14876 normalized layers at melt temperatures of 280 and 285 K, and only 0.11570 and 0.10744 normalized layers at melt temperatures of 290 and 295 K, respectively.

Of all of the parameters varied, latent heat had the least influence on the PCM heat exchanger size. However, it is evident that increasing the latent heat reduces the required number of PCM layers.

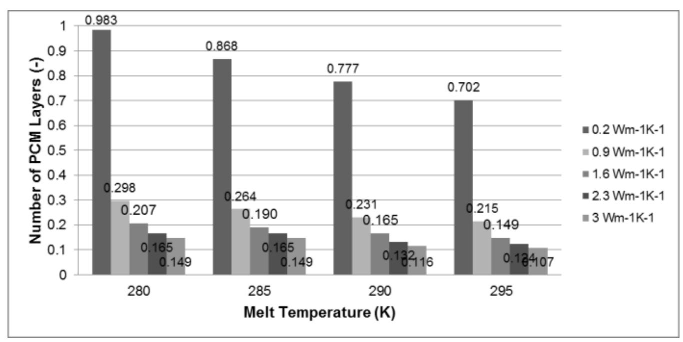

Figure 6. The minimum number of normalized PCM layers required at a latent heat of 250 $kJkg^{-1}$, every melt temperature, and every thermal conductivity.

Discussion

The two major trends observed in the results above are;

1. Increasing the thermal conductivity of the PCM reduces the required number of PCM layers.
2. Increasing the melt temperature of the PCM reduces the required number of PCM layers.

To better understand these trends the heat applied to the PCM during a high heat flux pulse was tracked. The applied heat is either;

1. Stored latently within the PCM.
2. Transferred through the PCM to the refrigerant.

3. Stored sensibly within the PCM.

Since the coolant temperature ultimately dictates the number of PCM layers, intuition says that for this application it is undesirable to sensibly store the heat applied to the PCM because a large temperature gradient will develop within it. To remedy this, a larger surface area (more PCM layers) is required to reduce the applied heat flux, or the majority of the heat must be stored latently within the PCM or transferred through it to the refrigerant. However, transferring the heat to the refrigerant is undesirable, as this would necessitate a large vapor compression system, defeating the purpose of the PCM heat exchanger. Therefore, the optimal configurations store a large quantity of the applied heat latently with the fewest number of PCM layers. The remainder of the section will focus on analyzing the distribution of the applied heat load in two contrasting configurations: PCM with conductivities of 0.2 and 3 $Wm^{-1}K^{-1}$ (latent heat of 250 $kJkg^{-1}$, at all melt temperatures, and the minimum number of required PCM layers).

The amount of latent heat stored within the PCM during a pulse can be examined by tracking the melt front during a pulse. Figure 7 shows the melt front location in a PCM layer at all melt temperatures for PCM thermal conductivities of 0.2 and 3 $Wm^{-1}K^{-1}$ normalized to the thickness of the PCM layer (y/l). The gray area effectively describes the amount of PCM that melts during a pulse. The top bound of this area is the location of the melt front at the end of a high heat flux pulse (10% duty), while the bottom bound is the location of the melt front at the end of a low heat flux pulse (beginning of a pulse). The most evident trend shown in Figure 7A is that a very small portion of the low thermal conductivity PCM layer melts during a pulse. As the melt temperature increases, the mean location of the melt front moves towards $y = 0$ (the coolant stream). Additionally, a slightly larger portion of the PCM layer melts because more PCM remains close to the melt temperature at that location throughout the pulse. A much larger portion of the PCM melts when it has a high thermal conductivity (Figure 7B). Similar to low thermal conductivity PCM, the mean location of the melt front in a PCM layer tends toward $y = 0$, and a slightly larger portion of the PCM layer melts as the melt temperature increases. Unlike low thermal conductivity PCM, the mean location of the melt front, and therefore, the melt temperature, dramatically affects the amount of PCM that melts during a pulse. At a melt temperature of 285 K, the melt region is bounded by the top of the model domain (the heat exchanger plate in contact with the refrigerant stream), which limits the quantity of PCM that can melt during a pulse. As the melt temperature increases, the mean melt front location moves towards $y = 0$, and the melt region is no longer physically bounded, enabling a much greater portion of the PCM to melt.

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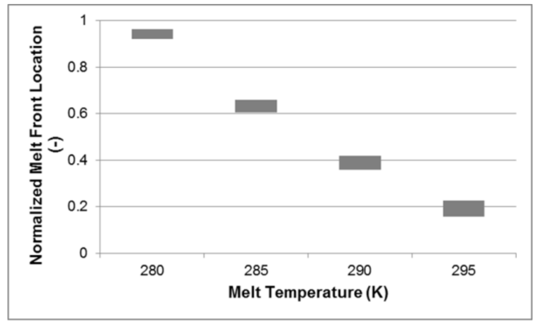

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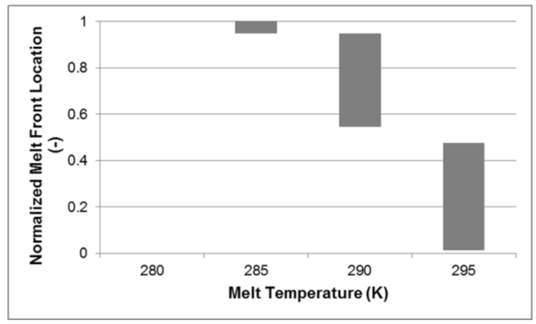

Figure 7. Normalized melt front location (y/l) during a pulse at thermal conductivities of (A) 0.2 $Wm^{-1}K^{-1}$ and (B) 3 $Wm^{-1}K^{-1}$, latent heat of 250 $kJkg^{-1}$, all melt temperatures, and the minimum number of required PCM layers at each melt temperature (see Figure 6 or Table 2). For the scenario at 3 $Wm^{-1}K^{-1}$ and a melt temperature of 280 K, all of the PCM melted during the first pulse and remained in liquid form upon reaching a periodic steady state.

The melt fronts clearly explain why high thermal conductivity PCM requires far fewer layers than low conductivity PCM: the temperature gradient in high thermal conductivity PCM is much lower, meaning a larger quantity of PCM remains near the melt temperature, and therefore melts. It also explains why the minimum number of PCM layers linearly decreases with increasing melt temperature at lower thermal conductivities: As melt temperature increases a larger portion of the PCM layer melts. Similarly, it explains the sudden drop in the minimum number of PCM layers at a melt temperature of 285 K observed at higher thermal conductivities. At that melt temperature the melt front is bounded by the refrigerant surface, which limits the amount of PCM that can melt. The melt temperature is no longer bounded as the melt temperature increases, allowing a much greater portion of the PCM layer to melt. This causes a reduction in the number of required layers.

Figure 8 shows the rate of heat transferred through the PCM to the refrigerant ($\dot{q}(y = l)$) at all melt temperatures and thermal conductivities of 0.2 and 3 $Wm^{-1}K^{-1}$. As expected, the heat is transferred to the refrigerant at a low rate during a pulse for 0.2 $Wm^{-1}K^{-1}$ PCM at all melt temperatures. The maximum rate of heat transfer to the refrigerant normalized to the maximum heat applied at each melt temperature is 0.68710, 0.58813, 0.57697, and 0.57323, respectively. The melt temperature has a larger impact on the heat transferred to the refrigerant for 3 $Wm^{-1}K^{-1}$ PCM. At melt temperatures of 280 and 285 K, the normalized maximum rate of heat transfer to the refrigerant is 1, which is highly undesirable, as the vapor compression system would have to dissipate the maximum heat load. At melt temperatures of 290 and 295 K, the normalized rate of heat transfer to the refrigerant is dramatically reduced, with maximum values of 0.67397 and 0.63477, respectively. This reduction can be attributed to the large quantity of PCM that melts during a pulse.

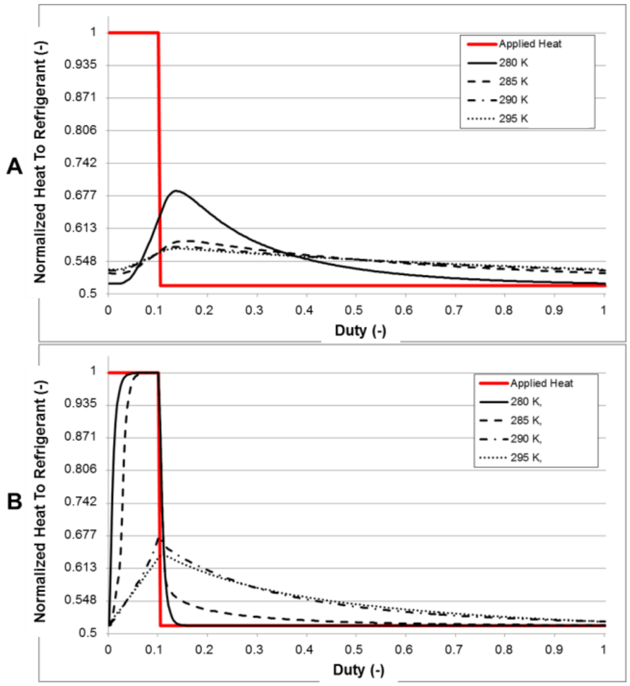

Figure 8. Heat transferred to the refrigerant ($\dot{q}(y = l)$) normalized to the maximum applied heat load at thermal conductivities of (A) 0.2 $Wm^{-1}K^{-1}$ and (B) 3 $Wm^{-1}K^{-1}$, latent heat of 250 $kJkg^{-1}$, all melt temperatures, and the minimum number of required plates at each melt temperature (see Figure 6 or Table 2).

Finally, the amount of heat stored sensibly within the PCM was determined. First, the heat transferred to the refrigerant during a pulse was calculated by numerically integrating the curves in Figure 8 (from 0 to 10% duty). This value was then subtracted from the total amount of heat applied during a pulse ($\dot{Q}_{max}$*10% duty) to determine the amount of heat stored within the PCM during a pulse. This heat is either stored sensibly or latently. The amount of heat stored latently within the PCM during a pulse is directly related to the difference in the melt front location at the start and end of a high heat flux pulse (Figure 7). Therefore, the remainder of the applied heat must be stored sensibly.

Figure 9 shows how the heat applied to 0.2 (A) and 3 $Wm^{-1}K^{-1}$ (B) PCM is distributed during a pulse at each melt temperature. For 0.2 $Wm^{-1}K^{-1}$ PCM (Figure 9A), approximately 54% of the heat applied during a pulse is rejected to the vapor compression system at all melt temperatures, leaving ~46% of the applied heat to be stored within the PCM. The percentage of heat stored latently increases from 28% to 40% at melt temperatures of 280 and 295 K, respectively, while the percentage of heat stored sensibly decreases from 18% to 5.2% at the same melt temperatures. Consequently, a larger temperature gradient develops within the PCM at lower melt temperatures, which necessitates more PCM layers. A different trend is observed for 3 $Wm^{-1}K^{-1}$ PCM (Figure 9B), particularly at melt temperatures of 280 and 285 K. At theses relatively low melt temperatures, the majority of the heat applied during a pulse (~96% and ~86%, respectively) is rejected to the refrigerant. At a melt temperature of 280 K, the remaining 4.4% of the applied heat is stored sensibly within the PCM since a phase change does not occur. At a melt temperature of 285 K, 6.8% of the remaining heat is stored latently, while 6.7% is stored sensibly. It is somewhat perplexing that the amount of heat stored sensibly increases as the melt temperature increases; while the number of required PCM layers does not change (both melt temperatures required at least 0.14876 normalized layers). This trend was also observed in other high thermal conductivity configurations and can be attributed to the appreciable amount of PCM that melts during a pulse, which dramatically affects the temperature distribution within the PCM, in some cases, enabling more sensible heat storage without an appreciable temperature rise at the coolant surface. At melt temperatures of 290 and 295 K, the amount of heat stored sensibly reduces to 3.2% and 0.36%, respectively. At both of these melt temperatures approximately 57% of the applied heat load is rejected to the vapor compression system, respectively leaving 39% and 43% of the heat to be stored latently.

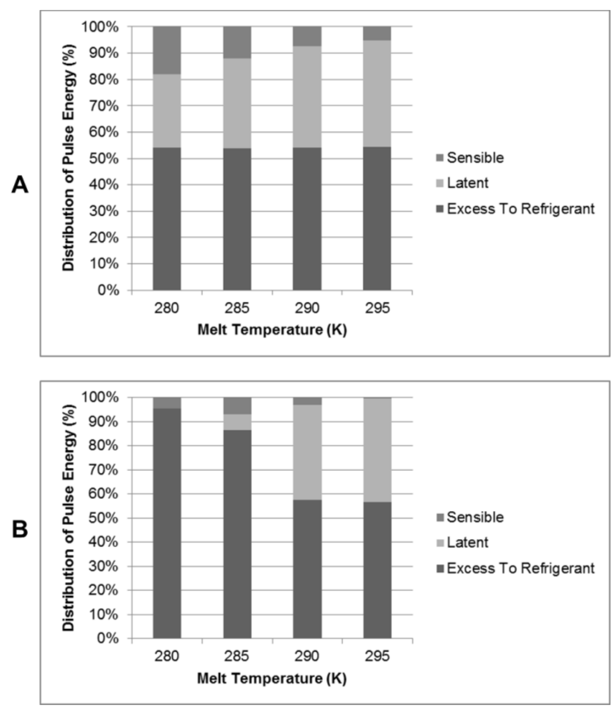

Figure 9. Distribution of applied heat during a high heat flux pulse at thermal conductivities of (A) 0.2 $Wm^{-1}K^{-1}$ and (B) 3 $Wm^{-1}K^{-1}$, latent heat of 250 $kJkg^{-1}$, all melt temperatures, and the minimum number of required plates at each melt temperature (see Figure 6 or Table 2).

FUTURE WORK AND CONCLUSIONS

Future Work

One of the assumptions made in this study was that the refrigerant temperature remained constant regardless of the instantaneous heat load that it experienced. In reality, the capacity of a vapor compression system is highly dependent on the temperature lift across the compressor. The goal of this initial study was to obtain general trends and gain a better understanding of using PCMs for thermal damping applications, and therefore, this was a convenient assumption to make. One of the major lessons learned in this study is that the PCM heat exchanger cannot be fully optimized without considering the entire cooling system. Therefore, integrating models of vapor compression system components with this PCM heat exchanger model will be the focus of future work. One potential method for doing this is to make the refrigerant temperature dependent on the instantaneous heat flux that it experiences. Compressor manufacturers often provide data indicating the temperature lift and refrigerant mass flow rate required to handle a range of heat loads. Using this data, it would be possible to more effectively optimize PCM heat exchanger parameters to minimize the size of the entire cooling system.

Conclusions

The ultimate goals of the PCM heat exchanger are to control the heat source temperature and dampen the transient applied heat load in order to reduce the size of a vapor compression system. Keeping this in mind, it is useful to think about the impact PCM heat exchangers with low and high thermal conductivity PCMs would have on such a system. From the results presented in this study it is clear that a PCM heat exchanger with a low thermal conductivity PCM can substantially dampen the applied heat load, which would reduce the size of a vapor compression system's compressor and condenser. However, the size of the vapor compression system would be dramatically increased by the PCM heat exchanger itself. The most useful conclusion from this study is that increasing the amount of heat stored latently during a pulse not only acts to substantially dampen the amount of heat transferred to the vapor compression system, but also minimizes the amount of PCM required to prevent an unacceptable temperature rise at the coolant surface. For low thermal conductivity PCM, a small portion of a PCM layer melts during a pulse, which means a greater quantity of PCM is required to effectively regulate the coolant temperature. Conversely, a PCM heat exchanger with a high thermal conductivity PCM and relatively low melt temperature minimizes the amount of PCM required, but does little to dampen the applied heat load. Therefore, the vapor compression system would still require a large compressor and condenser. However, at higher thermal conductivities increasing the melt temperature causes a greater portion of the PCM layer to melt, effectively dampening the heat load and minimizing the required number of PCM layers. Therefore, the optimal cooling system would contain a PCM heat exchanger with a high thermal conductivity PCM and a melt temperature close to the desired temperature of the coolant surface.

Another important conclusion of this study is that the size of the PCM heat exchanger exponentially decreases with increasing thermal conductivity. There is clearly a point where increasing the PCM's thermal conductivity does little to reduce the number of layers required (Figure 6). Therefore, depending on the application, only moderate (or no) thermal conductivity enhancement may be necessary to obtain a sufficiently small PCM heat exchanger.

This study aimed to determine the effect of (1) PCM thermal conductivity, (2) PCM melt temperature, (3) PCM latent heat on the size of a PCM heat exchanger. It was found that PCM heat exchangers with high thermal conductivity PCM are smaller than those with low thermal conductivity PCM, but that they cannot effectively dampen the applied heat load unless they have a relatively high melt temperature. At all thermal conductivities, increasing the PCM's melt temperature decreased the size of the PCM heat exchanger. Since the ultimate objective of the PCM heat exchanger is to reduce the size of the vapor compression system cooling the NLVS, future work will focus on optimizing PCM parameters to minimize the size of this system. Ultimately, this will enable us to better design compact cooling systems for large, transient heat loads.

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