

Experimental characterization and analysis of phase change material-based thermal energy storage system for refrigerated display case

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ABSTRACT

Refrigerated display cases that are used to store and exhibit food products in supermarkets and retail spaces consume a significant portion of the buildings' total electricity. Importantly, the refrigeration-related energy cost and demand charges are greatly affected by the time-of-use electricity pricing and demand rates, which are at their maximum during peak hours, typically when refrigeration energy consumption is also high. Using energy storage to shift the refrigeration load from peak to off-peak hours can greatly reduce the operational costs in the supermarket. This study demonstrates a phase change material-based thermal energy storage (TES) system, specifically designed in stackable units, that can be integrated with an open vertical refrigerated display case. We perform numerical and experimental characterization that includes finite-difference modeling for the TES, prototype fabrication, and laboratory evaluation, followed by a preliminary system-level analysis to predict the impact of TES on the refrigerated case performance, energy use, and energy cost. The results show that the dedicated latent TES for refrigerated cases can provide a specific energy of 50.4 Wh/kg and a specific power of 15.5 W/kg. The TES can be charged during 12 h of the off-peak period and discharged at various rates during 4 to 6 h of the peak period, thereby shifting the refrigeration load from the peak to the off-peak period. Consequently, annual cost savings up to 19% can be achieved, depending on the thermal load, the summer/winter peak electricity pricing, and the transition temperature of the phase change material used.

1. Introduction

Refrigeration in supermarkets and food retail stores can account for more than 40% of stores' total electricity consumption [1]. A significant portion of this energy is related to the refrigerated display cases that are used to store and display food products. Notably, energy consumption related to refrigeration is typically high during hot hours, such as in the afternoon, when the electricity demand and time-of-use prices are also high. Reducing the energy use and related costs while continuously operating the refrigerated cases to maintain the required product storage conditions is a challenge. The situation becomes even more challenging during unplanned events, such as extreme weather or prolonged power outages.

There are various ways to improve energy efficiency and the resilience of refrigerated cases. Previous studies in literature investigated many potential solutions to reduce the inefficiencies in the refrigerated display cases. For instance, a simple and effective method to enhance the

performance of refrigerated cases is to reduce infiltration by using glass doors. Studies report that infiltration can contribute up to 70–80% of the cooling load in an open display case [2]. An experimental evaluation by Faramarzi et al. [3] suggested that retrofitting a glass door in an open vertical case could reduce the cooling load by 68%, the compressor power by 87%, and the average product temperature by 6°F (3.3°C). Another study by Fricke and Becker [4] reported that although the compressor energy consumption in doored display cases was significantly lower than in open display cases, the anti-sweat heaters that are needed in doored display cases to prevent condensation on the glass doors required a significant amount of energy, resulting in the overall energy consumption of open display cases to be only 1.3 times higher than similar doored display cases. Although adding a door is not always possible, air curtains are often recommended to minimize air infiltration. Demirpolat [5] reported that optimally designed multi-flow air curtains could reduce up to 40% of the air entrained from the outside to the indoor environment of an open vertical display case. Several studies

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in the literature examined the effect of air curtains on the energy efficiency of open display case-type commercial refrigerators using experimental and computational fluid dynamics methods [6–8] and noted an improvement in the product storage condition and energy efficiency of the display case. In addition to air infiltration, heat conduction through the case panels and walls is another source of inefficiencies in refrigerated display cases. Tao et al. [9] noted that increasing insulation in the display cases by using vacuum insulation panels and vacuum insulating glass could improve the energy efficiency by up to 30%. A technical report by Walker [10] provides a detailed investigation of various sources of inefficiencies in refrigerated display cases, including thermal radiation from the ambient to the product and display case interior, as well as other internal thermal loads, such as lights, evaporator fans, periodic defrosts, and anti-sweat heaters.

Most prior studies have focused on reducing the undesired heat and mass flow into the display cases, and there has been less focus on solutions that can provide load-shifting capability and resilience in the display cases. As the contribution of intermittent and unpredictable renewable energy sources increases in the energy generation mix, the time mismatch between the energy demand and supply is becoming critical. Various energy storage techniques have been increasingly investigated in buildings to offset the peak energy demand and shift the energy use from on-peak hours to off-peak hours. Although electrical energy storage systems in the form of electrochemical batteries can be potentially used for refrigerated display cases, the need for large capital investments and a long payback period can make the investment unjustifiable. Considering that the purpose of refrigerated display cases is to store and cool the products, thermal energy storage (TES) appears to be the most logical solution among other energy storage alternatives.

Thermal energy storage for display cases can be achieved in various ways. Fricke et al. [6] suggested using the internal thermal mass of the products in a display case by precooling them during the off-peak period and turning off the refrigeration system during the peak period. More specifically, it was noted that the average product temperature in a medium-temperature refrigerated display increased by 5.5°C during 2.5 h of no refrigeration, i.e., at a rate of approximately 1°C per 30 min. Based on this study, it was suggested that the thermal mass of the stored product in refrigerated display cases could provide a significant load reduction if the product temperature is allowed to safely drift while reducing refrigeration system operation. Although it is a technically feasible solution to achieve load shifting, the nature of the products in the display cases greatly determines the duration and degrees of precooling temperature to avoid compromising the products' shelf life, quality and safety.

Achieving load-shifting potential in refrigerated display cases without significantly altering the internal operating condition of the stored products is possible by using a dedicated TES system, which can be either a sensible or a latent-based system. Typically, sensible TES requires a much larger volume than a latent TES; therefore, a latent TES based on phase change materials (PCMs) is often preferred for buildings and other applications where space is a constraint (e.g., a display case) [11]. TES for building components—such as envelopes [12–14]; heating, ventilating, and air-conditioning systems [15–17]; and refrigeration systems [18,19]—has recently been an active research topic. Considering the broad range of subjects, it is difficult to cover the entire literature here; therefore, only a brief review is provided.

Marques et al. [20] investigated the design and operation of a household refrigerator integrated with PCM-based thermal storage. The study included numerical modeling and experimental validation to estimate the TES charge and discharge rate and the refrigerator operation during on/off cycles under various ambient conditions. It was noted that a 5-mm PCM slab integrated into the refrigerator provided 3–5 h of continuous operation, depending on the thermal load, without a power supply. In another similar study, Azzouz et al. [21] conducted experiments on a household refrigerator attached with a PCM layer located at the back side of the evaporator. The study suggested that PCM-based

TES could provide up to 5–9 h of refrigerator operation without electrical supply and up to 10–30% improvement in the coefficient of performance. Hirmiz et al. [22] presented a numerical and analytical evaluation of a hybrid PCM-water thermal storage tank and proposed analytical methods to size thermal storage systems for heat pumps for demand management. The study noted that TES tanks containing 75% PCM with 2-cm thick PCM modules reduced the storage volume by more than a factor of three compared to water-only storage. The proposed TES, depending on the storage capacity, was able to cater up to 2–6 h of thermal loads in a residential household during peak periods. Huang et al. [23] evaluated the potential benefits of PCM-based TES in a cascade heat pump. The study used a clustering method to develop seasonally representative thermal loads in residential buildings and simulated the peak shaving performance of the cascade heat pumps using three different TES systems, including ice, tetradecane, and a commercially available PCM alternative. It was reported that higher TES capacity resulted in greater power reductions until achieving a threshold set by the building load characteristics. Further, the commercial salt hydrate PCM with 10.5°C transition temperature was found to provide the highest energy shavings along with the lowest leveled cost of storage, whereas ice was found to be the least effective.

Many other studies in the literature describe a broad range of thermal storage systems and their impacts and benefits on building components and equipment [24–26] and refrigeration systems such as display cases, household refrigerator, and refrigerated vehicles. Note that in literature, terms display 'case' and 'cabinet' are used interchangeably; however, in this paper, we have used the term 'case' for consistency with the product name used by the most US manufacturers. Table 1 provides a summary of some relevant studies related to PCM based TES systems in refrigeration related applications.

Based on the literature survey, we noted that TES systems dedicated to refrigeration applications can provide significant energy savings by enhancing the COP of the vapor compression cycle. Therefore, the most prior studies related to refrigerated display cases were focused on the energy saving aspects of the TES systems. However, a major benefit of using thermal storage technology is its capability to shift the energy demand from the peak period to off-peak period. This is critical because the utility grids are increasingly implementing time-of-use pricing (electricity rate based on time of use not just energy use) and demand rates (electricity rate based on electricity demand not just energy use). To our knowledge, none of the prior studies have investigated the TES systems that are targeted for load shifting aspects of refrigerated display cases. Additionally, while most of previous studies were focused on technical evaluation of the TES systems, the energy cost savings and system-level deployment costs are not well reported. To bridge this research gap and to evaluate the technoeconomic of TES for load shifting applications, in this study, we investigate the PCM-based TES integrated with a vertical open refrigerated display case. Note that in this study we select a vertical open refrigerated display case because it is still used extensively in the food retail stores due to the visual and tactile customer experience of picking the product directly from the shelves. More specifically, we perform a detailed component level evaluation using numerical as well as experimental techniques and a preliminary system-level modeling and cost-benefit analysis using a representative electricity pricing and peak demand rate in the U.S.

The key objectives of the study are to: 1) evaluate the design of a PCM-based TES for a medium-temperature refrigerated display case using a numerical model; 2) fabricate and characterize the TES system and measure its energy storage capacity as well as its charging and discharging power to achieve the desired load-shifting capability during demand response events; 3) perform a system-level analysis by integrating the TES with the refrigerated display case and predict the TES impact on the refrigeration-related compression power, the daily energy use, and the peak load reduction; and 4) conduct a preliminary cost-benefit analysis to examine the impact of TES on the refrigeration-related energy cost of commercial display cases.

Table 1

A brief literature review and summary of some relevant studies related to PCM-based TES systems in refrigeration related applications.

TES application	Method	PCM types	Key outcome/savings	Reference
Open refrigerated display case/cabinet	Experimental	Water gel as PCM radiators	Up to 5% of energy savings and lower cabinet temperatures	Alzuwaid et al. [27]
Glass-doored refrigerated case/cabinet	Experimental	Distilled water with a nucleating agent and gelling and thickening agent to prevent phase segregation	4.4% energy savings by using PCM on the shelves. During power failure situation, PCM on shelves increases food shelf life	Yilmaz et al. [28]
Open refrigerated display case/cabinet	Numerical and experimental	Water gel as PCM heat exchanger installed after evaporator coils	Energy savings up to 6.4%. Defrost and compressor off intervals increased by about 98% and 50%	Alzuwaid et al. [29]
Open refrigerated display case/cabinet	Experimental	Water filled with a finned tube heat exchanger	With compressor off for 2 h, product temperature increased by $\sim 1^\circ\text{C}$ with PCM compared to $\sim 2^\circ\text{C}$ without PCM	Ben-Abdallah et al. [30]
Open refrigerated display case/cabinet	Experimental	Rubitherm organic PCM: RT3, RT4 and RT5 filled with composite shelves and connected with heat pipes	Reduction in product average temperature by 13.7–32% and reduction in the product temperature fluctuation by 53.3–83.3% during defrosting	Wu et al. [31]
Household refrigerator	Experimental	A4 PluseICE organic PCM as heat exchanger attached to evaporator coils	12% reduction in power and 8% increase in coefficient of performance (COP)	Elarem et al. [32]
Household refrigerator	Experimental	Shape-stabilized heat conduction enhanced PCM, prepared with paraffin, high density polyethylene and expanded graphite, integrated with condenser.	Energy savings of 12% due to lower and stable condenser temperature	Cheng et al. [33]
Household refrigerator	Numerical	n-octadecane organic PCM filled in an annular tube around refrigerant line near condenser	10% increase in COP due decrease in the outlet refrigerant temperature of the condenser	Bakhshpour et al. [34]
Household refrigerator	Experimental	Water as PCM in a finned box attached to bottom surface of evaporator	Up to 13.7% decrease in energy consumption	Rahimi et al. [35]
Household refrigerator	Numerical and experimental	Water as PCM in a finned box attached to bottom surface of evaporator	Up to 17.4% decrease in energy consumption	Zarajabad et al. [36]
Household refrigerator	Experimental	PCMs were placed on the backside of the wire-and-tube condenser in a decreasing order of their melting temperatures	Up to 13% reduction in energy consumption	Pirvaram et al. [37]
Household refrigerator	Experimental	Water and three other binary aqueous mixtures in flexible packaging for evaporator tubes	Up to 9.4% energy savings	Yusufoglu et al. [38]
Refrigerated transport	Experimental	Eutectic PCM plates added to refrigerated vehicle compartment and charged during off-peak period	3°C lower minimum interior temperature and up to 91.4% reduction in energy cost	Liu et al. [39]
Refrigerated transport	Experimental	Two options: PCM layer added to external side of refrigerated enclosure walls, and PCM air heat exchanger placed close to the evaporator	PCM layer resulted in 5.5% reduction and phase displacement in daily heat load. With PCM air heat exchanger, fewer compressor's on-off cycles, resulting in 16% lower electric power consumption	Principi et al. [40]

Some of the key novelties of the paper include the development and validation of a finite-difference model, which can be used to design a PCM-based TES for refrigerated cases and predict the performance under various charging and discharging conditions. The model is validated using experiments that show the fabrication of a scalable TES that comprises of several stackable TES units that can be arranged in various patterns and quantities to achieve the required footprint and energy storage capacity and power. Additionally, the paper also presents a system-level modeling and cost-benefit analysis that guides the integration of the TES with the refrigerated case in a supermarket and quantify the impact of TES technology on refrigeration related energy use and cost savings. The system-level model specifically highlights the potential impact of TES on peak load shifting and estimates a simple pay-back period for the return on investment.

2. Methodology

2.1. PCM heat exchanger-based TES components and characteristics

The thermal storage system described here comprises three main components: a graphite-PCM composite (henceforth referred to as a “composite phase change material” [cPCM]), the thermal fluid for the heat exchange, and the aluminum microchannels at the interface between the thermal fluid and the cPCM (see Fig. 1). Thermal energy is stored and released in and out of the cPCM slabs that comprise Rubitherm SP-11 PCM [41] encapsulated in the graphite matrix. SP-11 is an inorganic PCM, with a transition temperature of approximately -11°C and a heat storage capacity of approximately 290 kJ/kg. Thermal conductivity is relatively small (0.6 W/m·K); therefore, a graphite matrix is

needed to enhance the overall thermal conductance of the cPCM and to minimize the phase change-related volume change.

Considering the low-temperature operation, we used silicon oil (Julabo Thermal C5[42]) as the thermal fluid, which has a lower viscosity (5 mm²/s at 20°C) and better thermal properties and working temperature range than the conventional glycol–water mixture that is typically used as a heat exchange liquid. The microchannels and the header manifolds for the fluid flow in the TES were custom-designed and fabricated to maintain the appropriate fluid distribution among the different microchannels.

As shown in Fig. 1(a), multiple cPCM slabs and microchannels can be stacked together to achieve the required energy storage capacity. In this study, only one TES unit comprising a microchannel with two cPCM slabs was used for experimental validation and characterization. A set of springs was used to apply compression on the top of the TES, which reduces the thermal contact resistance between the slabs and the microchannels [43]. Figs. 1(b)–(d) depict some images of the experimental prototype and components.

The main rationale behind using PCM-graphite TES is based on the cost scaling analysis for thermal batteries reported in our prior study [44], which shows that improving the thermal conductivity of TES by adding graphite fillers in PCM reduces the TES cost by more than 2 times. TES comprising only PCMs requires large heat exchangers due to its poor thermal conductivity. Although the addition of graphite fillers slightly reduces the PCM energy density, it significantly enhances the overall thermal conductivity, enabling a reduction in heat exchanger size and, thus, the overall TES cost. Additionally, a key advantage of impregnating PCM into the expanded graphite matrix is that its thermophysical properties remain nearly constant as the PCM changes its

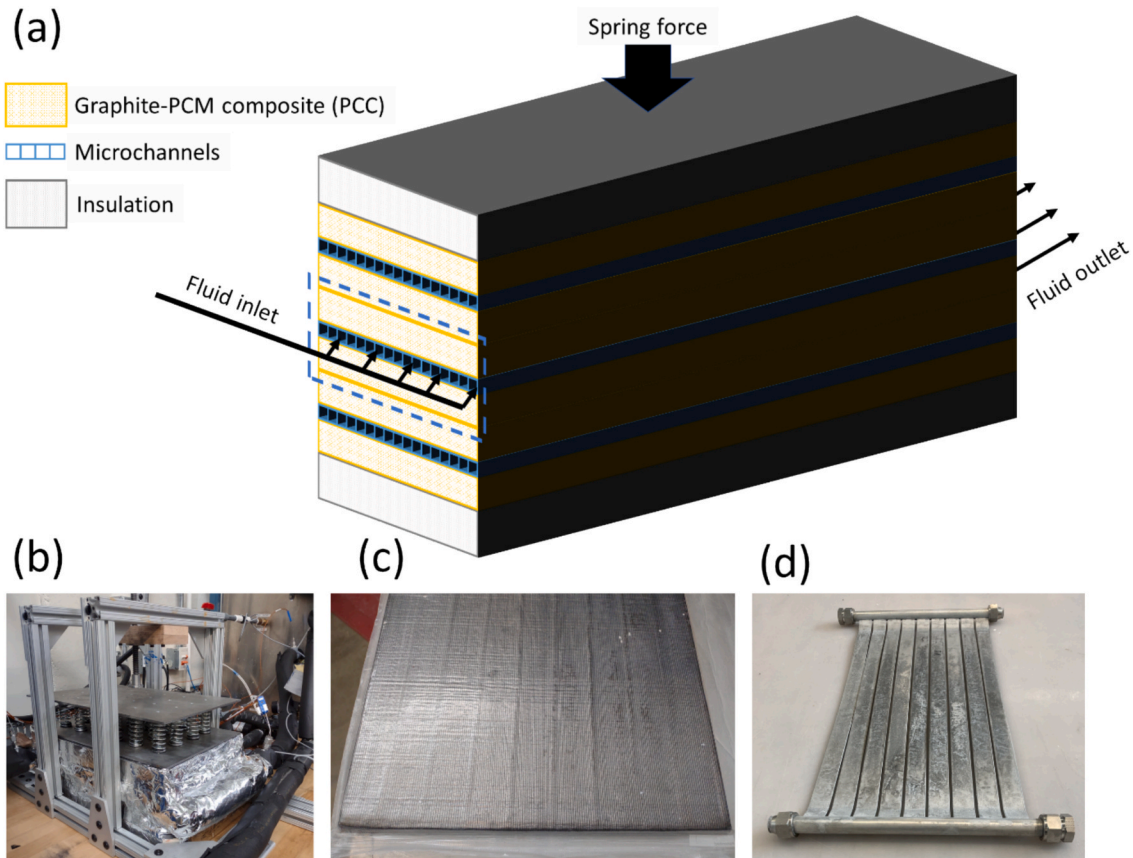


Fig. 1. PCM heat exchanger-based TES system comprising various stackable units to achieve the desired energy storage capacity. (a) CAD rendering illustrating various components of the TES that consists of cPCM, microchannels, fluid flow, and insulation; (b) experimental setup to characterize TES systems; (c) graphite-PCM composites (cPCM); (d) microchannels.

phase. Volume change could be an issue, but the use of springs ensures proper contact between cPCM slabs and microchannels.

2.2. Material synthesis and characterization

Expanded graphite is commonly used as a matrix to enhance the heat transfer rate of a PCM-based thermal storage system and to provide structural stability during phase transition. To produce expanded graphite, natural graphite flakes are treated with intercalating agents and then subjected to high-temperature treatment. The intercalating substances vaporize, forcing the graphite layers apart. The resulting material has a worm-like structure with high porosity.

To form compressed expanded graphite or graphite blocks, the expanded graphite worms were compacted to a prescribed density (Fig. 2(a)). The graphite worms initially have a porosity of about 99% before compression, which decreased to approximately 90% after compression. The manufacturing process of the compressed expanded graphite is described in detail in our previous study [45]. The resulting compressed blocks were then immersed in containers with liquid SP-11 until they reached a fully saturated state to form composite PCM (cPCM) blocks. Due to hygroscopicity, SP-11 must be encapsulated to mitigate moisture uptake and possible changes in physical and chemical properties. Therefore, the blocks were encapsulated under vacuum using poly-nylon bags (Fig. 2(b)).

Preliminary experiments on smaller-scale graphite blocks were conducted to choose the optimal graphite density, determine the cPCM saturation point, and show cycling stability over time. Impregnation studies were carried out at room temperature using graphite blocks of densities 100, 150, and 180 kg/m³. The blocks were submerged in liquid SP-11 and removed at certain time intervals to measure cPCM uptake. As

a result of the soaking study, 180 kg/m³ graphite density was selected for the full-scale unit, as it showed the highest volumetric content of SP-11 after 150 h of soaking (Fig. 2(c)).

Volume change during phase transitions – often up to 10–20% – is a common disadvantage of PCMs. Therefore, thermal cycling was performed in the Thermotron SE-400–6–6 Environmental Test Chamber to estimate both volumetric changes due to density changes of SP-11 and thermal cycling stability of the composite. The cPCM samples based on graphite blocks of different densities (100, 150, and 180 kg/m³) were cycled between –30 and 0°C, ensuring complete phase change of SP-11 each cycle. During cycling, the samples were removed from the chamber daily for a quick inspection, and their mass and height were recorded. Removals were performed alternately during the solid and liquid phases of the PCM. As can be seen from Fig. 2(d), all cPCMs exhibit expansion upon crystallization, which is associated with the presence of water in the composition of SP-11. Thus, even though the expansion stabilizes within 5% after an initial increase, this needs to be considered when designing the thermal energy storage systems. No mass loss was recorded for any of the cPCM blocks, irrespective of the graphite density, confirming the structural integrity and containment efficiency during operation. The cycling was carried out in stages. We began with very small samples (25 mm × 25 mm × 12 mm), then proceeded to larger samples (150 mm × 75 mm × 12 mm). All these tests demonstrated no noticeable cPCM mass loss over 200 cycles. We then tested two full-sized cPCM slabs and performed 150 cycles on each, again observing no mass loss.

It should be noted that the applied uniaxial compression induces preferential alignment of the expanded graphite worms along the pressing plane, which results in anisotropy of the formed blocks (Fig. 2(e)). Therefore, thermal conductivity was measured both along the

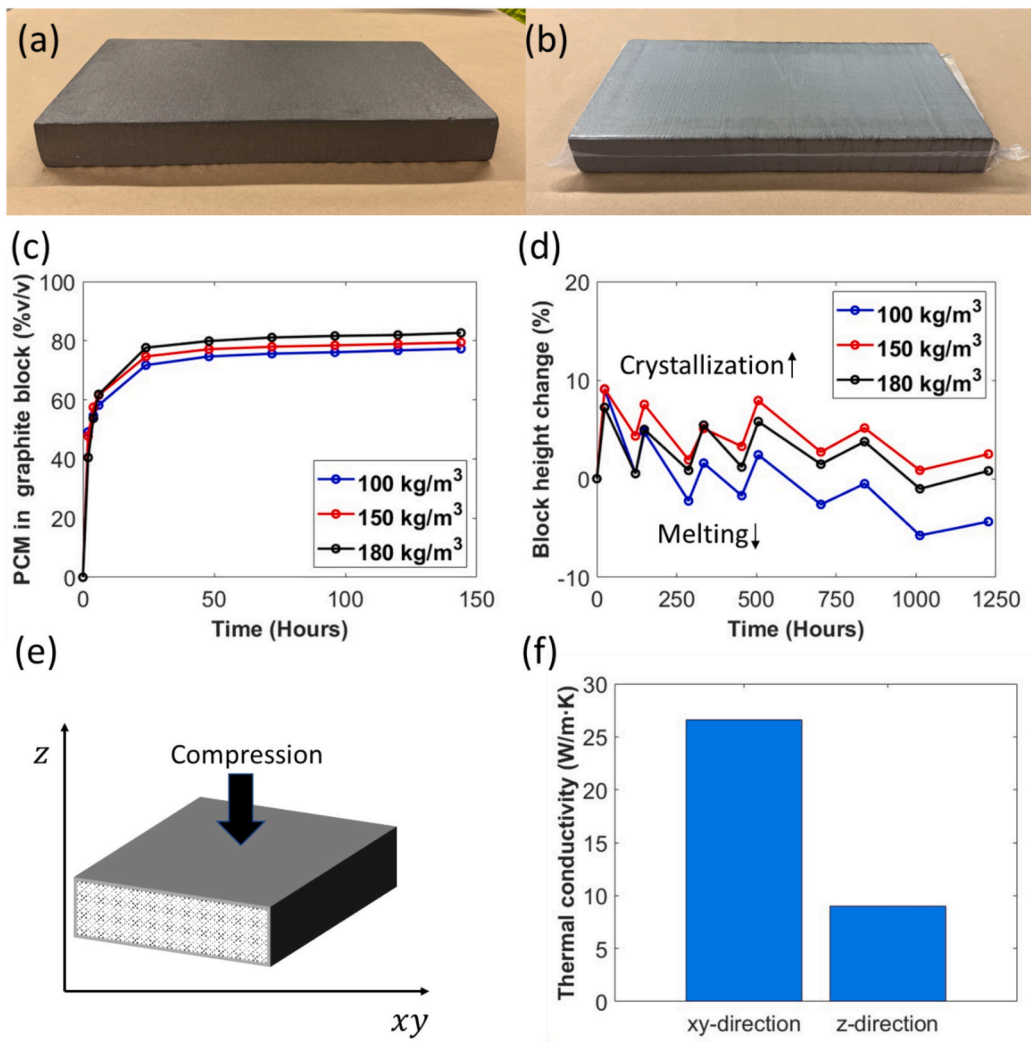


Fig. 2. Material synthesis and characterization of graphite-PCM composites (cPCM). (a) Full-scale cPCM blocks before encapsulation; (b) cPCM blocks encapsulated under vacuum using poly-nylon bags; (c) impregnation and (d) cycling study for small-scale c-PCM samples; (e) compressed expanded graphite block layout; (f) thermal conductivity measurements.

compression axis (xy -direction) and within the plane perpendicular to compression (z -direction) using the guarded hot plate method. The results demonstrate thermal conductivities of 9.0 W/m·K in the z -direction and 26.6 W/m·K in the xy -direction (Fig. 2(f)).

For future work, we plan to conduct comprehensive studies to further optimize cPCM density and soaking time to minimize the impact on long-time cycling and to enable scale-up and lower manufacturing costs.

2.3. TES optimization using the cPCM heat exchanger model

The numerical model for the cPCM heat exchanger was formulated using a finite-difference method. Similar models have been reported and validated in our prior research [17,46,47]. The key advantage of using the cPCM heat exchanger model over experiments is that the model predicts the performance of TES systems with different geometric parameters and under various operating conditions, thereby minimizing the time consumption and expensive experiments.

In this study, we envisioned TES as a combination of various stackable units, with the TES unit having a footprint of 18 in $\times$ 10 in (45.7 cm $\times$ 25.4 cm), which was decided based on the dimension of the microchannels available off the shelf and the in-house cPCM fabrication capability. Given that the cross-sectional area is fixed, the thickness of

the cPCM slabs becomes a crucial parameter because it determines the overall thermal resistance of the TES unit, which eventually governs the maximum charging or discharging power (heat transfer rate at which the TES can be charged or discharged).

The two-dimensional finite-difference model is depicted in Fig. 3(a). As in the experimental prototype, the fluid flows through the microchannel having a length of 45.7 cm and a width of 25.4 cm. The fluid stream was discretized into 30 subdomains along the length of the microchannel. Likewise, the cPCM slabs were discretized into 30 nodes along the length and 15 nodes along the height. The energy balances were imposed at each fluid domain, including the advection of the fluid, the heat transfer to the surrounding cPCM elements, and the energy storage in the fluid domain. The heat transfer rate between the fluid nodes and the cPCM nodes ($\dot{q}_{f \rightarrow PCM}$) was approximated using the convective fluid resistance and the total conductive resistance, including the conduction resistance of the cPCM, the conduction resistance of the microchannel material, and the contact resistance.

The contact resistance was measured using a guarded hot plate and three thicknesses of graphite, encapsulated with the mylar film. By using the different thicknesses, we can obtain thermal conductivity as well as contact resistance. These measurements confirmed the z -direction thermal conductivity of cPCM as 9.0 W/m·K and provided a value of 2×10^{-3} m²·K/W for the contact resistance. This contact resistance includes

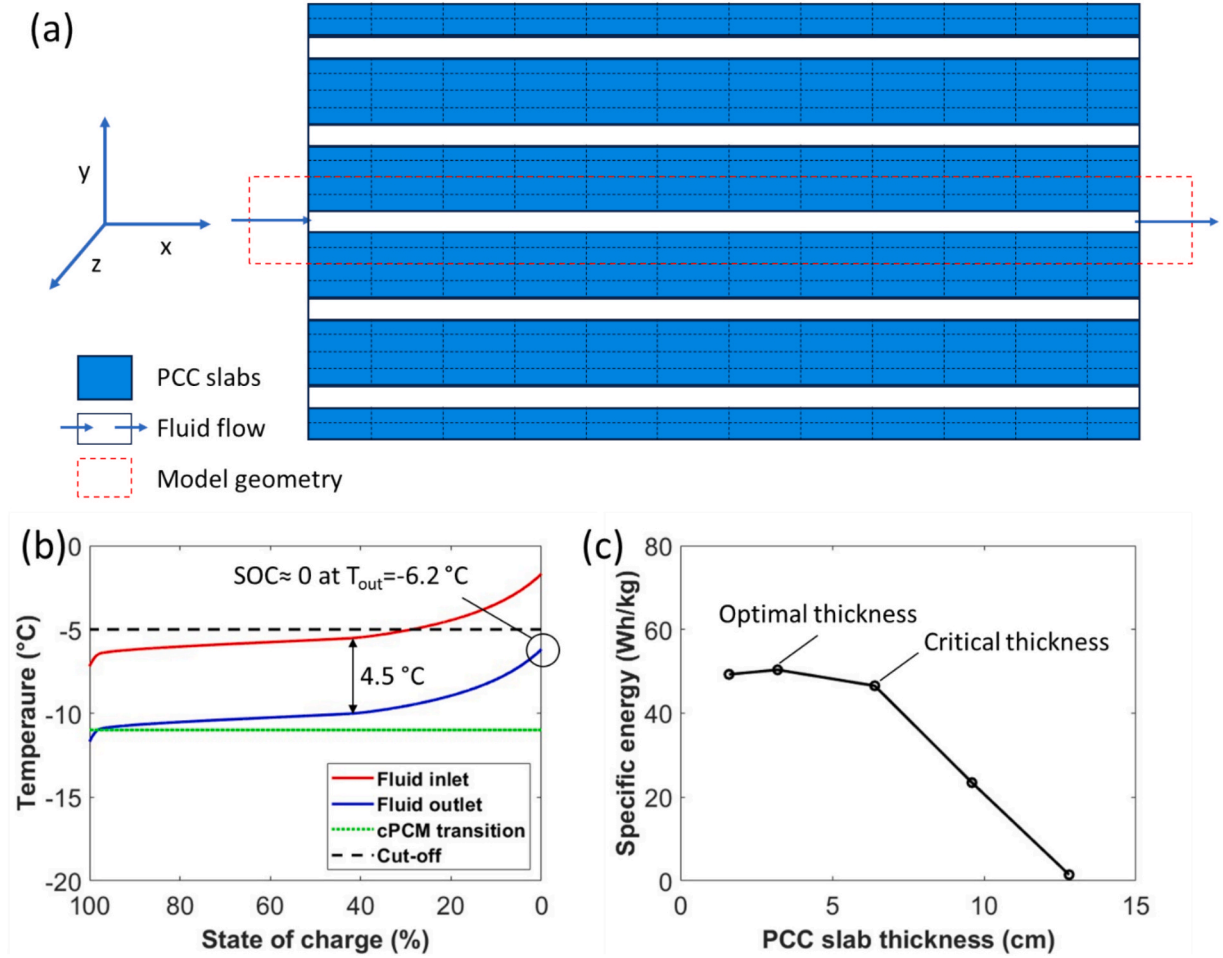


Fig. 3. The two-dimensional finite-difference model and simulation results of the TES. (a) The model geometry and fluid flow, comprising only a portion of the TES with two cPCM slabs and a microchannel; (b) rate capability plot of a TES design with a cPCM slab thickness of 3.2 cm and a discharge rate of C/3, illustrating the cutoff temperature before the TES reaches the fully discharged state (SOC: state of charge); (c) specific energy density during discharging at C/3 versus cPCM thickness, showing the critical cPCM thickness and cPCM thickness selected for fabrication.

the mylar film, and the contact between the cPCM and mylar and the mylar and the metal surface of the guarded hot plate instrument. We assumed this contact resistance is the same as in the prototype using the mylar-encapsulated cPCM and the aluminum microchannels.

The detailed discussion on the contact resistance measurement for PCM-graphite based TES is provided in our previous study [43], where it was reported the contact resistance varies with the applied pressure. In the current study, a spring force of around 700 lbf (3114 N) was used during experiments, which resulted in a contact pressure of 2.7 kPa (4 PSI). Under similar applied pressure conditions, the contact resistance reported in the prior study [43] is around $2.4 \times 10^{-3} \text{ m}^2 \cdot \text{K/W}$, which is close to the measurement done in the current study using the guarded hot plate.

Mathematically, the heat transfer rate between the fluid node, i , and the cPCM interface node ($i, 1$) is given as:

$$\dot{q}_{f \rightarrow \text{PCM}, i} = \frac{T_{f,i} - T_{i,1}}{R_{i,1}} \quad (1)$$

where $T_{f,i}$ is the temperature of the fluid at node i ; $T_{i,1}$ is the temperature of the cPCM node ($i, 1$) at the microchannel interface; and $R_{i,1}$ is the total thermal resistance between the fluid node and the cPCM node, which is given as:

$$R_{i,1} = \frac{1}{dA} \left(\frac{1}{k_{\perp}} \frac{dy}{2} + \frac{1}{h_{f,i}} + \frac{l_{\text{wall}}}{k_{\text{wall}}} + R_{\text{contact}} \right) \quad (2)$$

where $k_{\perp}$ is the cPCM thermal conductivity in the perpendicular direction, $h_{f,i}$ is the fluid convection coefficient, l_{wall} and k_{wall} are the thickness and thermal conductivity of the microchannel walls, and R_{contact} is the contact resistance. Last, $dA = dxW_{\text{HX}}$ is the heat transfer area, where W_{HX} is the width of the heat exchanger, and dx is the discretized length along the direction of the fluid flow.

The fluid temperature at node i along the length of the microchannel is described as:

$$\frac{dT_f}{dt} \Big|_i = \frac{1}{c_{p,f} \dot{m}_{f,i}} \left[\dot{m}_f c_{p,f} (T_{f,i-1} - T_{f,i}) - \dot{q}_{f \rightarrow \text{PCM}, i} \right] \quad (3)$$

where $\dot{m}_f$ is the mass flow rate of the fluid; and $c_{p,f}$, $\dot{m}_{f,i}$, and T_f are the specific heat, mass, and temperature of the fluid in each node. All thermodynamic properties of the fluid required for the simulations were taken from the technical data sheet of Julabo Thermal C5 thermal fluid [42].

Given that PCM is impregnated into the graphite matrix, the cPCM is assumed to be static, providing conduction as the only heat transfer mechanism in the composite. The heat transfer rate between adjacent cPCM nodes is calculated using a discretized version of Fourier's law and accounting for the enthalpy change of each cPCM node with time and directional differences in the thermal conductivity of the cPCM materials:

$$\frac{dh}{dt}\bigg|_{ij} = \frac{1}{M_{ij}} \left[\frac{k_{\perp} dy W_{HX}}{dx} (T_{ij} - T_{i-1,j}) + \frac{k_{\parallel} dx W_{HX}}{dy} (T_{ij} - T_{i+1,j}) + \frac{k_{\perp} dx W_{HX}}{dy} (T_{ij} - T_{i,j-1}) + \frac{k_{\parallel} dy W_{HX}}{dx} (T_{ij} - T_{i,j+1}) \right] \quad (4)$$

where M and T are the cPCM nodal mass and nodal temperature at the given node (i,j) ; $k_{\perp}$ and $k_{\parallel}$ are the thermal conductivities in the perpendicular and parallel directions, respectively; dx and dy are the length of the nodes in the x- and y-direction; and W_{HX} is the width of the heat exchanger.

Solid, liquid, and mushy phase of the cPCM node (i,j) in the model can then be determined using its enthalpy h_{ij} [48,49].

$$\theta_{ij} = 0 \text{ if } h_{ij} \leq h_s; \text{ solid node} \quad (5)$$

$$\theta_{ij} = 1 \text{ if } h_{ij} \geq h_l; \text{ liquid node} \quad (6)$$

$$\theta_{ij} = \frac{h_{ij} - h_s}{h_l - h_s} \text{ if } h_s \leq h_{ij} \leq h_l; \text{ mushy node} \quad (7)$$

where θ_{ij} denotes state (0: solid, 1: liquid, and a value between 0 and 1 shows mushy) of the cPCM node (i,j) . h_s and h_l are the enthalpy of the

cPCM in frozen and melted states, respectively.

At every time step, the cPCM enthalpy and fluid temperature were numerically integrated to calculate the temperature of the cPCM and fluid nodes for the subsequent time step. For all the simulations in this paper, the thermophysical properties of the cPCM were measured as follows: latent heat of 179 kJ/kg, thermal conductivity of 26.6 W/m-K in the parallel direction and 9.0 W/m-K in the perpendicular direction, an average density of 1,083 kg/m³, and an average specific heat of 2,000 J/kg-K. The microchannels were assumed to be aluminum with a thermal conductivity of 200 W/m-K and a density of 2662 kg/m³.

Fig. 3 depicts the fluid inlet and outlet temperatures for a TES comprising 3.2-cm thick cPCM slabs that are being discharged at the C-rate (inverse of the total time of discharge in hours) of 1/3. The cPCM thickness here is defined as the height of the cPCM material between the two microchannels shown in Fig. 3; therefore, the top and bottom cPCM slabs in the TES are only half thick. The cPCM transition temperature (-11°C) and the fluid cutoff temperature (-5°C) are also shown. The fluid temperature at the inlet and outlet maintains a constant temperature difference of 4.5°C , providing a constant discharge power of 15.5 W/kg of cPCM materials. The specific energy density (total discharge energy per unit mass of cPCM materials) is noted to be 50.4 Wh/kg. The

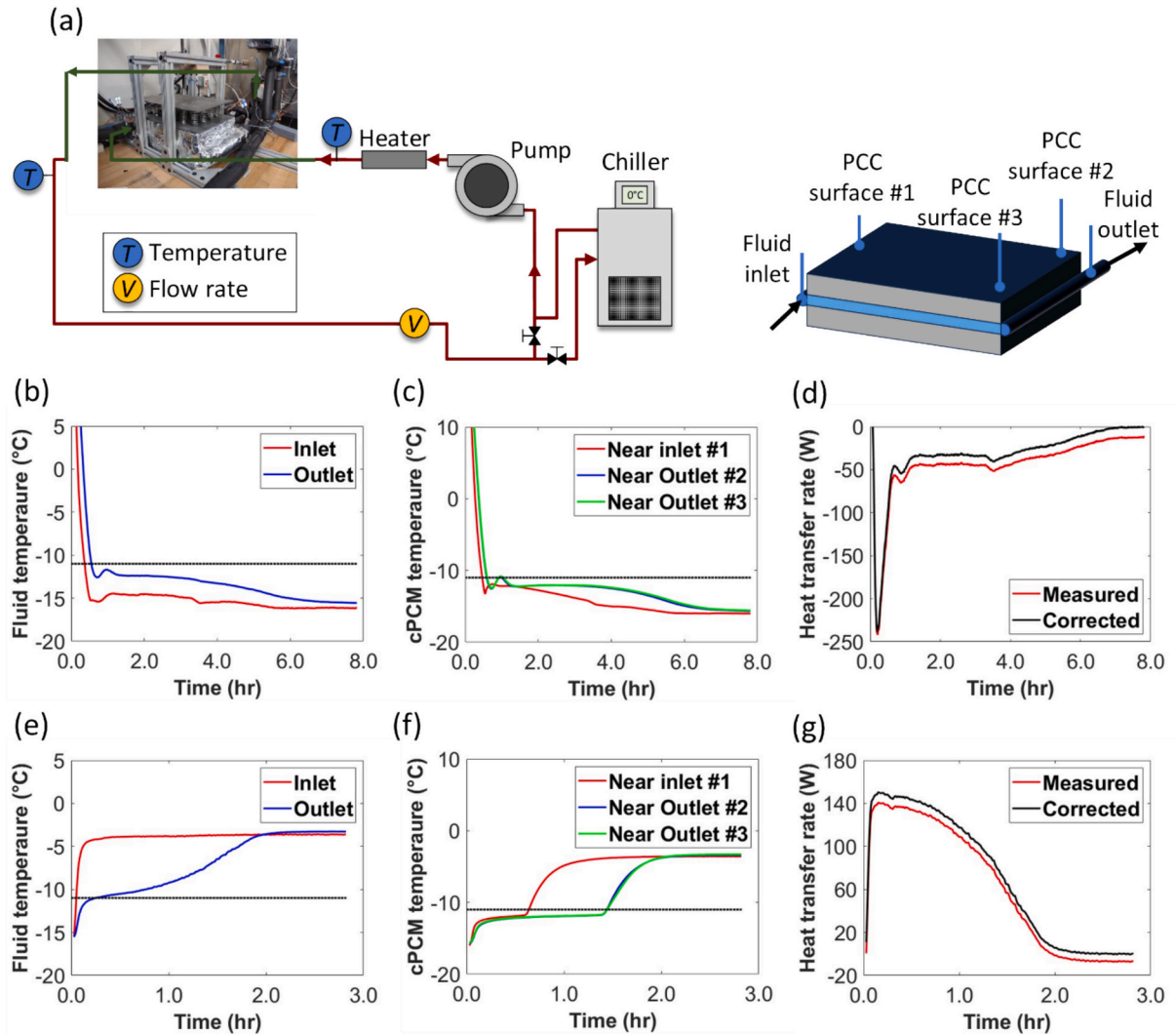


Fig. 4. Experimental evaluation of charging and discharging performance of the TES. (a) Schematic illustrating the experimental setup and locations of the thermocouples; (b–d) TES during charging when the chiller set point temperature is fixed at -19°C : fluid temperature, cPCM temperature, and heat transfer rate versus time; (e–g) TES during discharging when the chiller set point temperature is fixed at -7°C : fluid temperature, cPCM temperature, and heat transfer rate versus time. The charging and discharging power (heat transfer rate) has been corrected to account for heat gains to the cPCM from the ambient. The cPCM transition temperature is shown using the black dotted line.

fluid temperature at the outlet remains below the cutoff value as the TES fully discharges (state of charge or SOC reaches zero). As shown in Fig. 3, the specific energy density decreases with an increase in cPCM thickness. Notably, the specific energy significantly decreases if the cPCM thickness is greater than 6.4 cm, it gradually increases as the cPCM thickness is reduced from 6.4 cm to 3.2 cm, but it reduces again with a further decrease in the cPCM thickness beyond 3.2 cm. The decrease in specific energy with an increase in the slab thickness beyond this critical value of 6.4 cm occurs due to an excessive increase in the cPCM thermal resistance, causing incomplete melting before the fluid cutoff temperature is reached. The reduction in the specific energy with a decrease in the cPCM thickness below 3.2 cm occurs due to a higher contribution of the microchannel mass in the total mass of the TES unit when the cPCM slabs are thin. Based on the modeling results, a cPCM slab thickness of 3.2 cm was found to be optimal, and thus it was chosen for the fabrication.

2.4. Experimental

The schematic and the experimental setup to test the TES, along with the position of the thermocouples, are shown in Fig. 4(a). The experimental characterization procedure of the TES system is similar to what has been reported in our previous studies [11,43]. The test setup includes a chiller (Julabo CORIO CD-1000F), a variable-speed diaphragm pump (Pentair SHURflo 8000–543-236), a heater (Watlow FLUENT in-line 1 kW), and a feedback loop to manage the set point temperature. Three thermocouples each at the inlet and outlet of the fluid flow were used for the temperature measurement. A flow meter with a feedback loop was employed to manage the preset flowrate.

In the charging process, the cPCM freezes with the chiller operating in cooling mode. Fig. 4(b–d) depict the fluid inlet and outlet temperatures, the cPCM surface temperatures, and the heat transfer rate during the charging process. In the discharging process, the cPCM melts with the chiller operating in heating mode or when the in-line heater is switched on. Fig. 4(e–g) show the fluid inlet and outlet temperatures, the cPCM surface temperatures, and the heat transfer rate during the discharging process with the chiller running in heating mode. In the subsequent sections, the in-line heater is used to discharge the cPCM while maintaining a constant discharge rate. The fluid temperature at the inlet varies between -15°C and -16°C during the charging cycle and between -3°C and -4°C during the discharging cycle. The inlet and outlet temperatures asymptote when the TES is fully charged or discharged. The difference in the inlet and outlet temperature near the saturation region suggests some heat gains from the ambient.

Fig. 4(c) and (f) depict the cPCM upper-surface temperatures measured at the mid edge near the fluid inlet and at the two corners near the fluid outlet. The cPCM portion near the fluid outlet reaches the transition temperature later than the cPCM portion near the inlet. This difference in the charging or discharging time is visible in the cPCM surface temperature plots. Notably, the cPCM surface temperatures measured at two corners near the fluid outlet are nearly the same, which confirms the proper flow distribution across different microchannels.

Accounting for the heat gains from the ambient, the TES energy storage capacity (E_{TES}), the charging and discharging power (Q_{TES}), the specific energy density (SE), and the specific power density (SP) can be calculated by accounting for the heat transfer rate and integrating it over the total time of the discharge:

$$E_{TES} = \int_{t=0}^{t=\tau} \dot{m}_f C_{p,f} (T_{f,out} - T_{f,in}) dt \pm \int_{t=0}^{t=\tau} \dot{Q}_{gain} dt \quad (5)$$

$$Q_{TES} = \dot{m}_f C_{p,f} (T_{f,out} - T_{f,in}) \pm \dot{Q}_{gain} \quad (6)$$

$$SE = \frac{E_{TES}}{M_{TES}} \quad (7)$$

$$SP = \frac{Q_{TES}}{M_{TES}} \quad (8)$$

where $T_{f,in}$ and $T_{f,out}$ are the fluid inlet and outlet temperatures, $\dot{m}_f$ and $C_{p,f}$ are the fluid mass flow rate and specific heat, τ is the total charging/discharging time, and M_{TES} is the TES total mass that includes the mass of the cPCM slabs and the microchannels. The heat gain from the ambient ($\dot{Q}_{gain}$) can be estimated using the experimental heat transfer rate data when the TES is fully charged/discharged.

The method to estimate the ambient heat gain $\dot{Q}_{gain}$ using steady state discharge power and fluid and TES temperatures is detailed in our prior research [11]. In addition, we have reported uncertainty propagation analysis using sensor accuracy and data acquisition frequency (5 Hz) in our previous study [43]; therefore these discussions are not included in this paper to avoid repetition.

Table 2 shows the TES charge and discharge capacities as measured using the raw experimental data, the corrected value after accounting for the ambient heat gains, and the theoretically calculated value as the reference. Note that the measured TES charge and discharge capacities are different (~ 250 kWh for the charge and ~ 200 kWh for the discharge). This is due to the different TES initial and final temperature conditions before and after the charging and discharging cycles. During charging, the TES is nearly at room temperature when the cooling starts, and it reaches approximately -15.7°C when the charging completes. On the other hand, during the discharging, the TES is at -15.7°C when the heating starts, and it reaches approximately -3.4°C when the discharging ends. This disparity results in an approximate 45 Wh difference in sensible energy storage of the TES.

2.5. Vapor-compression systems model

Fig. 5 shows a conceptual design of the TES-integrated refrigerated display case. The refrigeration system is modified to integrate the TES in a manner that it can be charged and discharged in different time durations. The modified cooling system comprises of a brazed plate heat exchanger (BPHX) that acts as an evaporator for the vapor compression cycle. BPHX is connected to a glycol coil and TES using a glycol loop and two 3-way valves (V1 and V2). Fig. 5 (b–d) depicts various operational modes. During cooling only mode (Fig. 5(b)), TES is fully bypassed by closing V1 outlet to TES, so that heat exchange occurs directly from the evaporator coil to the glycol coil. During charging and cooling mode (Fig. 5(c)), evaporator operation remains unchanged, but TES and glycol coil are connected in series via the glycol loop, so that colder glycol from BPHX first reaches TES, allowing charging, and then to glycol coil for cooling of the display case. During charging only mode (Fig. 5(d)), there is no flow in the glycol coil, and TES is directly connected to evaporator via glycol loop. Lastly, during discharge and cooling mode (Fig. 5(e)), there is no refrigerant flow, and stored energy in TES is discharged to cool the display case via glycol loop.

We quantified the potential for this cPCM heat exchanger-based TES to provide load shifting for an 8-ft (4.72 m) vertical open refrigerated display case. The TES is assumed to scale to this larger size using stacked slabs and channels, as shown in Fig. 1. The refrigerated case is assumed to have a constant load of 1.6 kW_{th}. While the load will actually vary due to product loading, this was not assessed in this study. Therefore, this 1.6

Table 2
TES charging and discharging capacity at constant fluid inlet temperature conditions.

	Charge	Discharge
Raw measured energy storage capacity (Wh)	338.5 $\pm$ 22.4	173.8 $\pm$ 8.2
Estimated ambient heat gains (Wh)	84.5 $\pm$ 6.1	22.7 $\pm$ 1.4
Net measured energy storage capacity (Wh)	254.0 $\pm$ 23.2	196.6 $\pm$ 8.3
Theoretical calculated storage capacity (Wh)	272.6	227.7

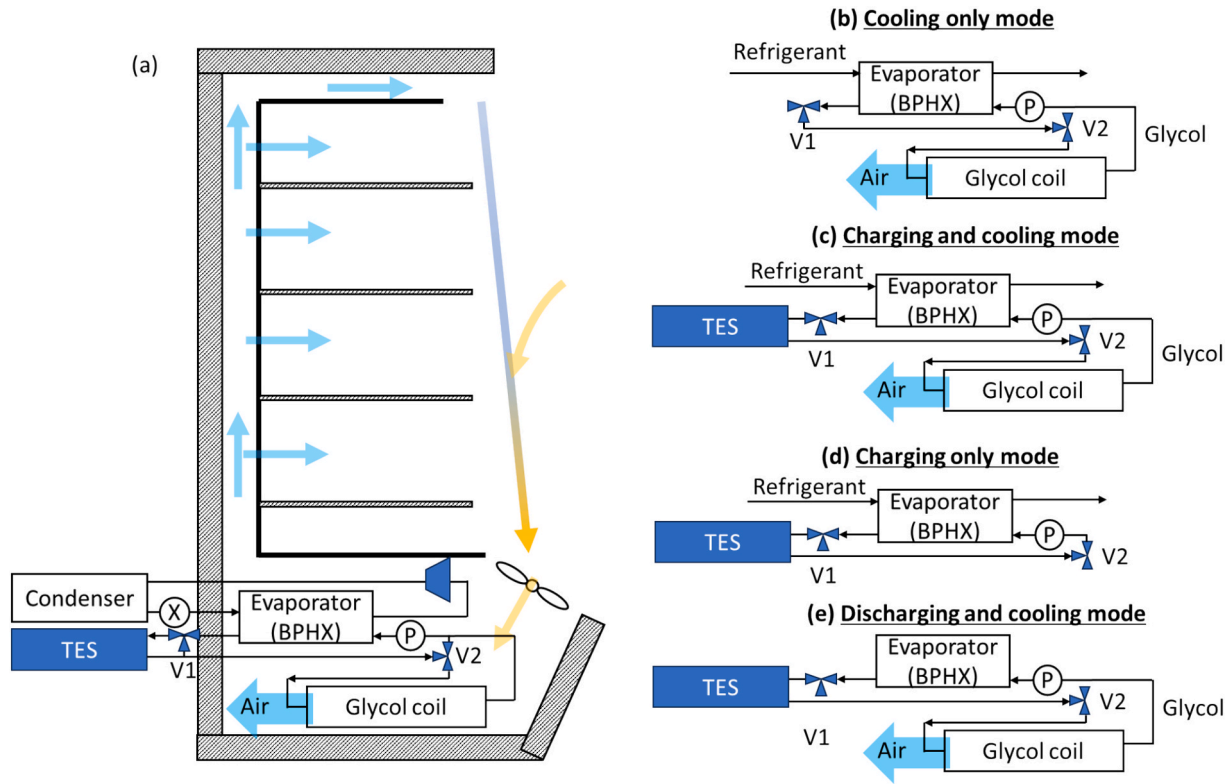


Fig. 5. A conceptual design of the TES-integrated open refrigerated display case and various operational modes (BPHX: Brazed plate heat exchanger). (a) The refrigeration system modified to integrate the TES and a glycol coil; (b) Cooling only mode; (c) Charging and cooling mode; (d) Charging only mode; (e) Discharge and cooling mode.

kW_{th} should be considered as an effective average load during the discharge period. Higher loads due to loading of warm products would result in the need for a slightly larger TES size, to ensure it is available even during a period with above-average thermal load.

We analyzed the performance using a typical daily weather profile, with a high temperature of 33°C and a low temperature of 23°C . We looked at scenarios where the cPCM is discharged in either 4 or 6 h, based on typical peak demand periods from electric utilities. The cPCM was charged from 20:00 to 8:00 (12 h) and discharged from either 13:00–19:00 or 15:00–19:00. The TES size is 6.4 kWh_{th} and 9.6 kWh_{th} for the 4- and 6-hour peak periods, respectively. This is 32 and 48 times larger than the prototype, which would likely use microchannel heat exchangers that are approximately twice as long and three times as wide ($\sim 100\text{ cm} \times \sim 75\text{ cm}$) and require five microchannel heat exchangers for the 6.4 kWh_{th} design and eight microchannel heat exchangers for the 9.6 kWh_{th} design. The total TES mass, excluding auxiliary components like manifolds and pumps, is estimated to be around 125–188 kg.

We compare the performance of two refrigerated cases: one without TES (baseline) and one with TES. The case uses a self-contained R290 refrigeration circuit with a Copeland YHV0211 scroll compressor [50]. We modeled the compressor with a 20-coefficient map that outputs the mass flow rate and the electric power using the inputs of the compressor rotating speed, the evaporating dew-point temperature, and the condensing dew-point temperature [51]. The refrigerant flow rate is corrected for the difference between the refrigerant density at the compressor inlet and the density at the nominal 10 K superheat for the map. The case has a water-cooled condenser, which is assumed to operate at a condensing temperature of 3°C above the water temperature. We assume that the water enters the condenser 5°C above ambient, after rejecting heat to the ambient air in a dry cooler.

The evaporator for the baseline display case and the display case with TES are different. The baseline uses a refrigerant-to-air coil, with an evaporating temperature 2°C below the supply air temperature. The

case with TES uses a brazed-plate water-to-refrigerant heat exchanger and is assumed to operate with an evaporating temperature 1°C below the glycol/water temperature, which is 2°C below the supply air temperature.

We estimate the pump power based on a 50% efficient pump and estimated pressure drops. These are assumed to be the following during each mode: Cooling: 18 W (8 L/min and 70 kPa pressure drop); Charging: 5 W (3 L/min and 50 kPa); Charge-cool: 27 W (8 L/min and 100 kPa); Discharge: 18 W (8 L/min and 70 kPa).

The display case with TES charges the TES with a thermal fluid that is 2°C below the cPCM transition temperature. During the discharge, fluid is provided 2°C above the cPCM transition temperature. These are both simplifications because the glycol temperature for charging and discharging will depend on the TES state of charge, as shown in the following results, but they are adequate for understanding the potential impact of the TES and the cPCM transition temperature.

We evaluate the financial benefits of the display case with TES by considering a typical commercial building utility rate for the United States, which includes $\$/kW_e$ demand charges and a time-varying $\$/kW_e$ energy charge. We assume a $\$25/kW_e$ demand charge during the peak period, which is assessed on the peak for each month, and energy charges of $\$0.10/kW_e$ and $\$0.16/kW_e$ for the off-peak and on-peak periods, respectively.

3. Results and discussion

3.1. TES discharge time

Fig. 6 depicts the TES discharge performance in terms of the fluid inlet and outlet temperatures versus time under various discharge rate conditions. The discharge C-rate is defined as the inverse of the time in hours needed to fully discharge a TES system from an initial fully charged state to a fully discharged state at a constant heat transfer rate

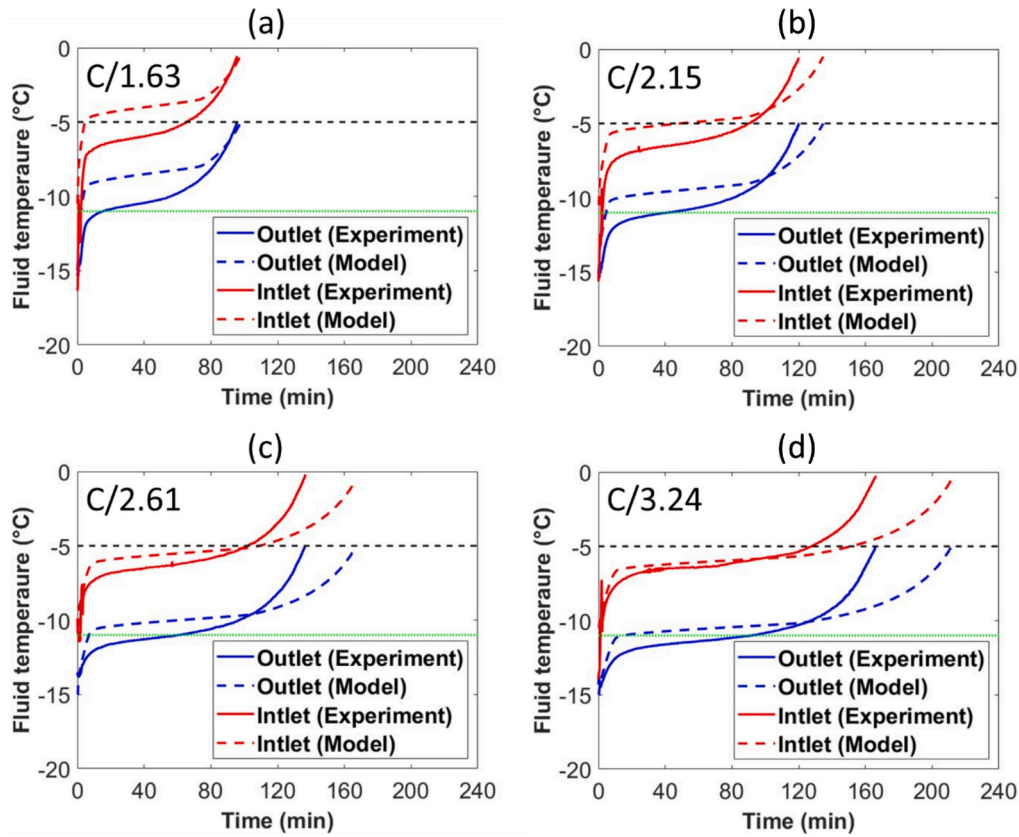


Fig. 6. TES discharge characteristics with the fluid inlet and outlet temperatures at various discharge rates: (a) C/1.63, (b) C/2.15, (c) C/2.61, and (d) C/3.24. All plots terminate at the fluid outlet cutoff temperature of -5°C . The green dotted line represents the cPCM transition temperature, and the black dashed line represents the fluid outlet cutoff temperature.

(or discharge power). In other words, a high discharge C-rate of a TES implies a high discharge power and a shorter discharge time. Likewise, a TES charge C-rate implies the inverse of the time in hours needed to fully charge a TES system at constant charging power. In these experiments, we always charge the TES using a nearly constant chiller set point temperature (inlet fluid temperature varies between -16.2°C and -14.5°C); therefore, the charge power or charge C-rate is variable. But during the discharge, we achieve a constant C-rate by fixing the temperature difference to 4.5°C between the inlet and outlet fluid flow, continuously adjusting the heating rate in the in-line heater, and maintaining a predefined fluid mass flow rate.

As expected, the discharge time increases with a decrease in the discharge C-rate. All lines in Fig. 6 (a)–(d) terminate when the fluid outlet temperature reaches a predefined cutoff of -5°C , indicating the end of the discharge process even though the TES might still have some remaining energy storage. A predefined cutoff temperature is crucial for the refrigerated display case, as it needs to maintain the temperature of the stored products under the required conditions. Note that, in this study, we do not evaluate temperature of the products stored in the refrigerated display case. A complex coupled model comprising heat and mass transport in display case and TES is needed, which is beyond the scope of current study. However, the glycol cut-off temperature of -5°C is colder than the refrigerant (typically -2°C to -3°C) used during normal display case operation, which would ensure the product temperature within the desired limits.

Table 3 compares the discharge time of the TES under various discharge C-rate conditions. Note that the discharge time here is calculated based on the time the fluid outlet temperature reaches the cutoff temperature of -5°C . It is not the time needed to fully discharge the latent thermal storage capacity of the TES. In other words, we use the additional sensible energy stored in the TES while the fluid outlet

Table 3

TES discharge time when the fluid outlet temperature reaches the cutoff value of -5°C .

Discharge C-rate (1/hour)	Experiment (Hours)	Model (Hours)	Difference (%)
C/1.63	1.60	1.63	1.9%
C/2.15	2.00	2.25	12.5%
C/2.61	2.28	2.79	22.4%
C/3.24	2.77	3.53	27.4%
Mean absolute percentage error			16.0%

temperature is still below the cutoff threshold. In nearly all cases, the discharge time for the fluid to reach the cutoff temperature is predicted to be higher in the model than the experiments. This is due to the ambient heat gain from the surroundings, which is not accounted for in the model, but it occurs in practice, causing a faster discharge in the experiment than the idealistic modeling conditions. As the discharge C-rate decreases, the discharge time increases, and thus the total ambient heat increases. For a high C-rate (C/1.63), the difference between the model and the experimental discharge time is small; however, as the discharge time increases, the discrepancy between the model and the experimental results becomes larger.

Note that the model and experimental temperature output shown in Fig. 6 is primarily for visual comparison and it highlights the overall trend of the discharge process at various constant power discharge rates. For comparison between model and experimental results, the absolute percentage difference and mean absolute percentage error are calculated in Table 3, which shows that the percentage difference between model and experiments increases with decrease in C-rate and the average discharge time is about 16% higher in the model than

experiments. This is expected since the TES during experiments undergo heat gain from ambient, which is not accounted in the model.

3.2. TES discharge rate

Fig. 7 shows the TES discharge power under various C-rate conditions. As expected, the discharge heat transfer rate increases with a decrease in C-rates. Although the average discharge power of the model is only slightly higher than that of the experiment, the discharge time is significantly different in the two cases. As shown in Table 4, the average experimental discharge power is approximately 119 W at C/1.63, 91 W at C/2.15, 80 W at C/2.61, and 67 W at C/3.24. On the other hand, the discharge power predicted by the model is approximately 123 W at C/1.63, 93 W at C/2.15, 77 W at C/2.61, and 62 W at C/3.24. Overall, the TES model exhibits good agreement with the experimental results, validating the benefits of the model in predicting the TES performance under various operating conditions.

Again, the model and experimental power output shown in Fig. 7 is mainly for visual comparison and illustrates the overall trend of the discharge power at various constant discharge rates. For comparison between model and experimental results, the absolute percentage difference and the mean absolute percentage error are calculated in Table 4, which shows that, on the average, the model underpredicts the discharge power by about 4.3% than the experiments. This is related to the ambient heat gain in experiments, which is not accounted in the model, leading to higher discharge power particularly at smaller C-rates.

Fig. 8(a) shows the TES discharge capacity, which is the time integral of the discharge power from the start of the discharge to when the fluid outlet temperature reaches the cutoff. The total discharge energy in this duration is nearly constant under the various C-rate conditions used for the experiments. This confirms that the thickness of the cPCM slabs is properly optimized, resulting in a small thermal resistance under the

Table 4

TES discharge power averaged over the time when the discharge starts and when the fluid outlet temperature reaches the cutoff value of -5°C .

Discharge C-rate (1/hour)	Experiment (W)	Model (W)	Difference (%)
C/1.63	119.44	122.77	2.8%
C/2.15	91.06	93.07	2.2%
C/2.61	79.82	76.67	3.9%
C/3.24	67.46	61.76	8.4%
Mean absolute percentage error			4.3%

predefined experimental conditions. This is more evident in the Ragone plot shown in Fig. 8(b). The Ragone plot represents the specific power density of a TES versus its specific energy density under various C-rate conditions. All the data points for experimental C-rates occur in the near-vertical region of the Ragone plot, indicating nearly constant specific energy density despite the change in the discharge C-rate. As shown using the modeling results, if the discharge C-rate, and thus the specific power, significantly increases, the specific energy density starts to reduce, resulting in a smaller discharge energy capacity of the TES.

Additionally, to understand the impact of contact resistance on the modeling results and the impact of ambient heat gain on the experimental results, a sensitive analysis was performed. The contact resistance was increased and decreased by 50% against the reference value of $2 \times 10^{-3} \text{ m}^2 \cdot \text{K/W}$. The discharge capacity obtained using the model changed by around $\pm 2\%$. Likewise, the ambient heat gain used for the correction in the experimental measurements was increased and decreased by 50% against the reference value, and the maximum change in discharge capacity was noted to be within $\pm 6\%$.

3.3. Cyclability test

Fig. 9 shows the TES discharge capacity under the cyclability test

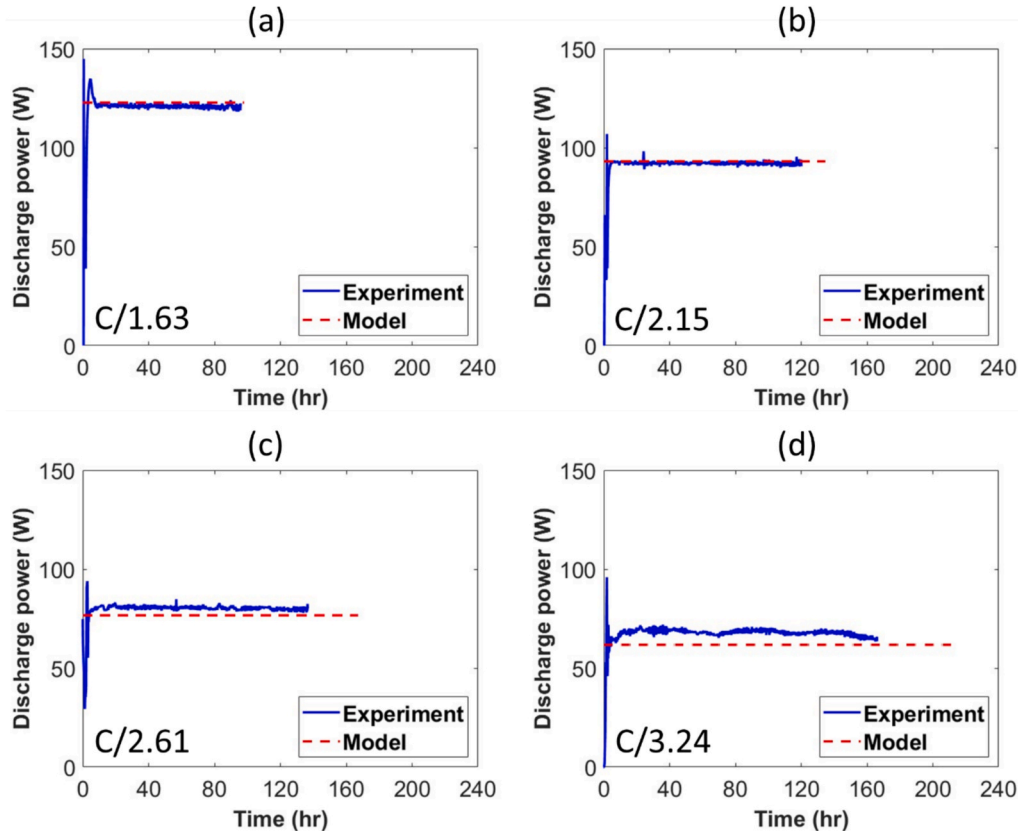


Fig. 7. TES discharge power under various C-rate conditions: (a) C/1.63, (b) C/2.15, (c) C/2.61, and (d) C/3.24. All plots terminate at the fluid outlet cutoff temperature of -5°C .

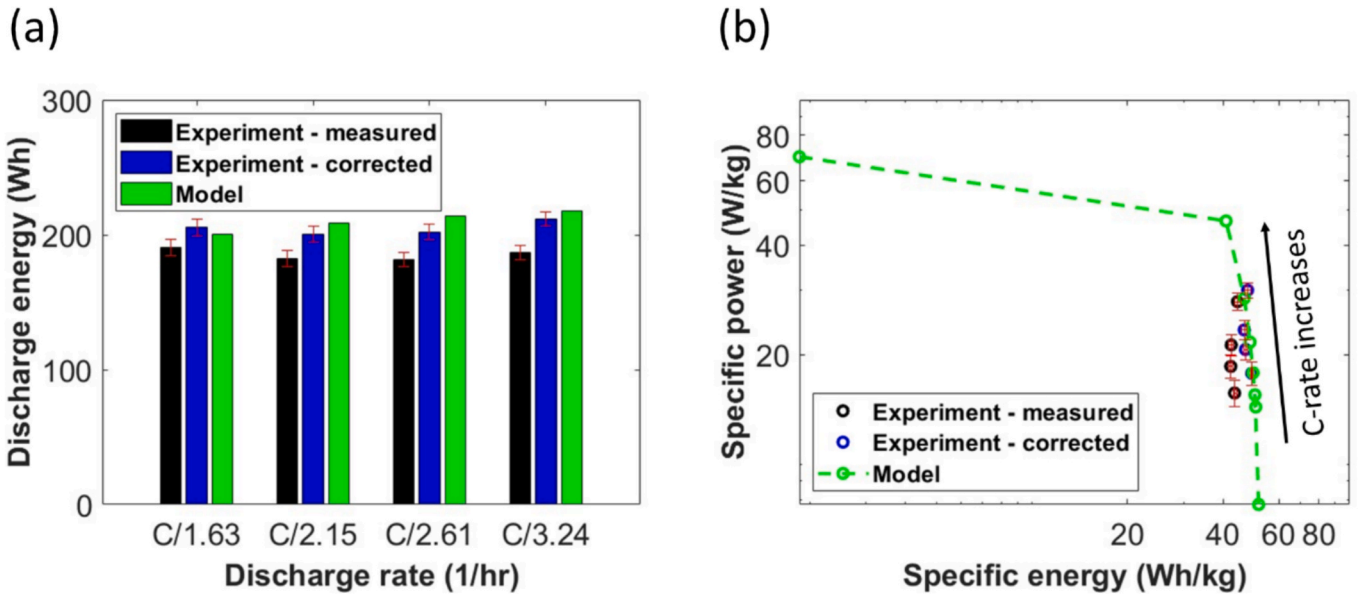


Fig. 8. TES discharge capacity and Ragone plot. (a) Total energy discharge in the time interval between the start of the discharge process and when the fluid outlet temperature reaches the cutoff temperature under the various C-rate conditions. (b) TES-specific power density versus TES-specific energy density under the various C-rate conditions. The error bars represent the uncertainty calculated using the root-sum-of-squares method and the measurement uncertainty of the sensors used in the experiments.

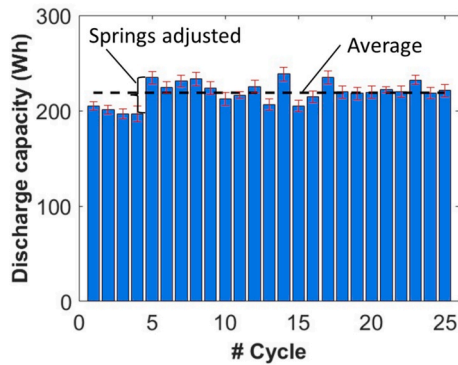


Fig. 9. TES discharge capacity under the cyclability test over 25 charging-discharging cycles. The error bars represent the uncertainty calculated using the root-sum-of-squares method and the measurement uncertainty of the sensors used in the experiments.

over 25 charging-discharging cycles. For brevity, only the discharge capacity is shown here. The TES discharge capacity remains nearly constant over the first four cycles; however, the actual value was noted to be lower than expected using the modeling results. The test setup was adjusted, and the spring force was increased to further reduce the contact resistance. The TES discharge capacity shows some variations during the initial cycles between 5 and 17, but the performance appears nearly consistent for the remaining cycles.

Note that during the first few cycles of the cyclability tests, the discharge capacity is ~ 200 Wh, which is close to the value reported in Table 2. This also nearly matches which discharge capacity reported in Fig. 8. After the test set-up was altered to increase spring force that reduced the contact resistance between cPCM slabs and microchannels, TES thermal performance increased resulting in an average discharge capacity of ~ 220 Wh, which is close to theoretical value reported in Table 2. There are some uncontrolled experimental variabilities, which are difficult to estimate. For instance, ambient heat gain (Q_{gain}) term used for correction in the discharge data is only an estimate. In practice, it varies based on the temperature difference between the TES and

ambient conditions. The exact charging temperature was also noted to be slightly different for different cycles. Although we maintained the same chiller set-point temperature, the actual fluid recirculation temperature varied slightly based on the chiller performance. Lastly, all the cyclability tests were not performed at the same discharge rate; therefore, discharge duration and conditions are different and thus some variations are expected.

Overall, there is no degradation trend observed in the cyclability test over 25 cycles. Note that such short-term cyclability tests only highlight the immediate mechanical and thermal stability of the TES system. A long-term cyclability test comprising numerous cycles is necessary in the future study to evaluate the TES performance over its life cycle.

3.4. System-level impact

Based on these results, we analyzed the potential of the TES system to provide load shifting in a refrigerated case. This is not a comprehensive analysis but rather an illustrative example. As described, we considered a 24-hour period with a constant refrigeration load. Fig. 10 shows the compressor electric power for both the baseline and the TES scenarios, with a 4-hour peak demand period (15:00 to 19:00). The baseline power is lower at night, when it is cooler outside, and it increases with the ambient temperature during the day. Two cPCMs are considered for the TES scenario: one each with a transition temperature of -7°C and -11°C . The longer charge and discharge periods (12 h and 4 or 6 h) allow us to use the higher transition temperature and still fully discharge the cPCM.

There are three distinct operating modes for the display case with TES. From hour 0 (midnight) to 8:00, the system is in *charging mode* and freezing the cPCM. The compressor operates at a higher speed, providing thermal fluid at 2°C below the transition temperature to freeze the cPCM. The fluid exiting the TES is used to meet the load of the case. The higher evaporator load, as well as the lower evaporator temperature, means the compressor uses much more electricity than the baseline. The higher power for the lower cPCM transition temperature (-11°C) is due to the lower efficiency at the lower evaporator temperature. The cPCM is fully charged at 8:00. Note that in some cases, the system may run in charge mode, when there is no cooling. This could include defrost cycles, which may occur one to two times per day. Warm air from the case is

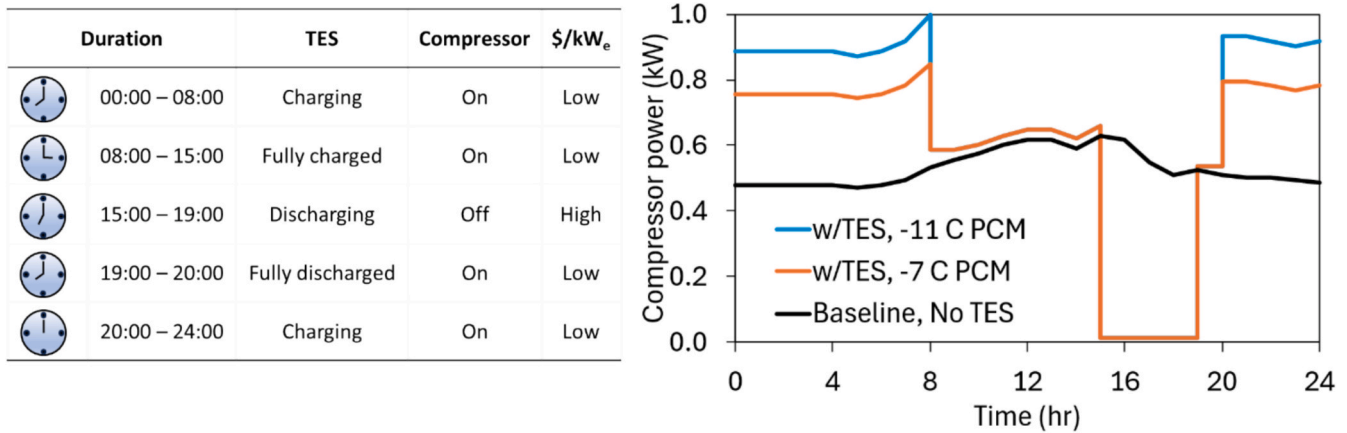


Fig. 10. Illustrative twenty-four-hour load profiles for a typical summer day for both the baseline display case and a display case with cPCM-based TES with a transition temperature of either -11°C or -7°C .

used to defrost the coil, and the compressor, which is normally off during defrost, can be used to charge the TES.

From 8:00 to 15:00, the display case with TES is in *cooling mode*. It delivers the same cooling to the display case as the baseline, but it has a slightly lower evaporating temperature due to the secondary loop. This increases the electric power slightly above the baseline.

From 15:00 to 19:00, the TES system is in *discharge mode*, and the compressor is off, while the TES delivers cold glycol the heat exchanger coil. This period has the highest $\$/\text{kWh}_e$ energy rates and the highest $\$/\text{kW}_e$ peak demand rates.

From 19:00 to 20:00, the system returns to *cooling mode*, and then to *charging mode* from 20:00 to 24:00 (midnight). This final *charging mode* brings the cPCM back to the same state of charge as it was from the prior day (hour 0).

As noted, the most expensive time to operate the display case compressor is the 4- or 6-hour peak window due to both the energy and demand charges. Based on the assumed electric rates described in the methods section, we estimate the electricity costs to operate the refrigerated case for the day shown in Fig. 10. We then assume the same weather for each day throughout the 4-month summer period. We also simulated a representative non-summer day (average temperature $\sim 8^{\circ}\text{C}$) and then summed the costs for the summer and non-summer days. The summer includes weekdays, which have time-of-use rates, and weekends/holidays, which do not. The winter is assumed not to have demand charges or time-of-use rates.

Although the display case with TES increases the demand above the baseline for the *charging mode*, this would typically not add to the

demand charges for the supermarket. This is because the peak demand falls in the late afternoon due to other loads in the store, including air conditioning. This was confirmed by looking at simulated load profiles for supermarkets, including submetered refrigeration electric power, from our prior research [52]. The typical whole-building electric demand during the charging period was 300 kW lower than the discharge period. If the entire refrigeration load of approximately 250 kW was shifted from a 6-hour discharge period to a 12-hour charge period, the peak would increase by approximately 150 kW, only half the difference between the discharge and charge windows; therefore, it would not create a new peak demand during the charging period. Examples of where this may not be the case is for the winter months in a building with electrified heating that maintains the store temperature at night, since this is the coldest time of the day.

We can compare both the savings for only the summer and for the full year (Fig. 11). Considering only the summer shows the high end of the savings and is indicative of a situation where winter also has peak demand and time-of-use rates. Considering the full year shows the low end of the savings and is indicative of a situation where there are no time-of-use or demand charges in the winter.

The annual savings are 12–13% for the -11°C cPCM and 16–19% for the -7°C cPCM, as shown in Fig. 11. Situations with winter peak pricing would likely approach the summer-only percentage savings of 20–32%. If TES is included in all display cases in a supermarket, the savings could be $\$2,000$ – $\$5,000/\text{year}$, depending on the peak period duration, the cPCM transition temperature, and whether the TES is included in only medium-temperature display cases (as tested here) or includes both

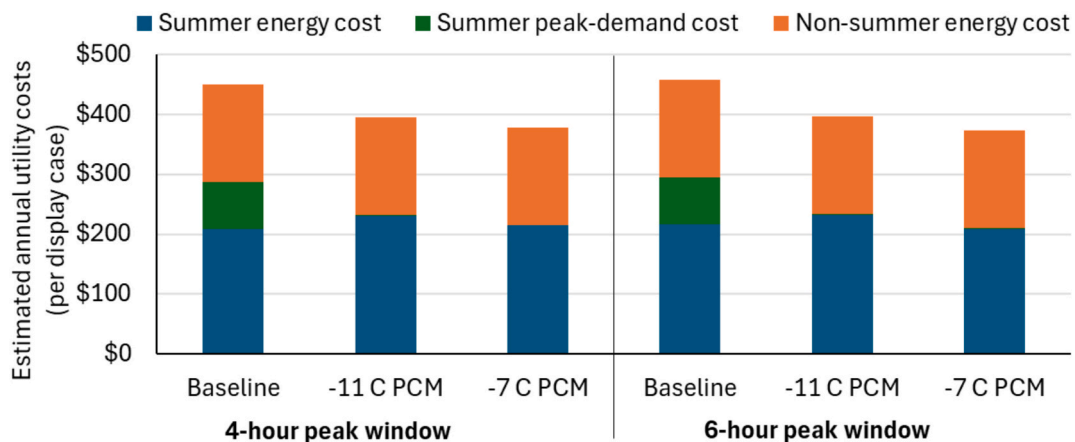


Fig. 11. Estimated annual utility costs per 8-ft (4.72 m) display case for one illustrative scenario. Costs are split into summer energy costs, summer peak demand costs, and non-summer energy costs.

medium-temperature and low-temperature display cases.

The savings per display case, shown in Table 5, are important to consider when comparing the savings to the cost of adding a thermal storage system to a display case. A previous study on cost scaling analysis of thermal batteries predicted TES cost in the range of \$69–\$155 per kWh [44]. The cost of graphite-PCM composite TES was specifically low (\$69/kWh) due to high thermal conductivity. For the open refrigerated display case considered in this study, we calculated the TES capacity of about 6.4 kWh–9.6 kWh, resulting in TES cost of \$440–\$660. Table 5 shows the annual cost savings of \$54–\$85, resulting in a simple payback period of around 6–8 years. Therefore, TES for refrigerated display case seems techno-economically feasible; however, careful consideration needs to be paid to further minimize the integration costs.

4. Conclusions

In this paper, we introduced a dedicated PCM-based TES system for a refrigerated display case and performed a comprehensive analysis using a numerical model as well as experimental characterization. An illustrative system-level analysis, integrating the TES with the refrigerated display case, was also performed to estimate the impact of the TES system on the required refrigeration compressor power, the daily energy use, and the peak load reduction. A preliminary cost-benefit analysis was also conducted to examine the annual energy cost savings due to the TES and its potential payback period for capital investment.

Some key findings and observations of the study are summarized as follows:

- A two-dimensional finite-difference model was formulated to design a PCM-based thermal storage system and was later validated using experimental results. The TES system was designed in the form of stackable units of cPCM slabs so that the required energy storage capacity can be achieved using multiple TES units.
- We determined an optimal thickness (3.2 cm) of the TES units, which provided a maximum specific energy of 50.4 Wh/kg and a specific power of 15.5 W/kg during the discharging process in 3 h. The specific energy significantly reduced if the TES layer thickness was greater than a critical value of 6.4 cm or lower than 3.2 cm.
- Experiments performed under various charge and discharge rate conditions showed good agreement with the modeling results, validating the benefits of the model in predicting the TES performance under various operating conditions. The TES discharge energy was also found to be nearly constant between 180 Wh and 190 Wh when the TES discharge rate was experimentally varied between C/1.63 to C/3.24, confirming the optimal thickness and design of the TES units. The TES discharge energy increased to about 220 Wh with increase in spring force due to lower contact resistance.
- Cyclability tests were also performed to evaluate the durability and consistency of the TES performance. Although we noted some variations in the TES discharge energy with the number of cycles, the variability was found to be small and within the experimental uncertainty. Notably, such short-term cyclability tests only highlight the immediate mechanical and thermal stability of the TES system. A long-term cyclability test comprising numerous cycles is necessary in the future study to evaluate the TES performance over its life cycle.
- The system-level analysis when integrating the TES with the refrigerated case showed the possibility of fully charging the TES for 12 h during the off-peak period and discharging the TES for 4 to 6 h during the peak period, thereby shifting the refrigeration load of the display case from the peak to the off-peak period.
- The cost-benefit analysis showed an annual cost savings of 12–13% for the -11°C cPCM and 16–19% for the -7°C cPCM, which could further increase to 20–32% if there was winter peak pricing. This reduction in energy cost provides cost savings of \$2,000–\$5,000/year if the TES is integrated in all display cases in a supermarket.

Table 5

Illustrative utility cost savings (absolute and percent) for the display case with TES considering only summer utility costs and annual utility costs.

	Savings per display case		Percentage savings	
	-7°C PCM	-11°C PCM	-7°C PCM	-11°C PCM
Annual savings – 4-hour peak period	\$71.43	\$53.98	15.9%	12.0%
Annual savings – 6-hour peak period	\$84.90	\$60.77	18.5%	13.3%
Summer-only savings – 4-hour peak period	\$71.84	\$54.39	31.7%	24.0%
Summer-only savings – 6-hour peak period	\$85.30	\$61.18	28.9%	20.7%

- Considering the savings per display case and assuming TES cost at \$69/kWh, a simple payback period was noted to be around 6–8 years, demonstrating promising application of TES technology for refrigerated display cases. However, further reduction in the integration costs is desired to make TES more techno-economically viable for short-term return on investment.

It is important to note that, in this study, the system-level modeling and cost-benefit analysis was performed using a representative electricity pricing and peak demand rate in the U.S. The scope of the present study was limited to providing an initial TES design for the refrigerated display cases and a preliminary techno-economic evaluation to assess its technical feasibility and present a cost-benefit scenario that could confirm a potential use-case of the technology. Overall, for the load shifting applications, higher differences in time-of-use electricity pricing and peak demand rates would favor economic feasibility of the TES systems for the refrigerated display cases. However, the projected cost savings and payback period would vary based on the local electricity pricing structure.

Next steps could include evaluating different rate structures and their impact on cost savings. It could also include optimization of the PCM transition temperature, which would benefit from an exergy-based analysis and would help select PCMs for different case types, climates, and the risk tolerance for higher product temperatures.

CRediT authorship contribution statement

Ravi Anant Kishore: Writing – review & editing, Writing – original draft, Validation, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Jason Woods:** Writing – review & editing, Writing – original draft, Validation, Supervision, Investigation, Formal analysis. **Yana Galazutdinova:** Writing – review & editing, Methodology, Investigation, Formal analysis. **Monica Cook:** Writing – review & editing, Supervision, Investigation. **Said Al-Hallaj:** Writing – review & editing, Supervision, Funding acquisition. **Kyle Foster:** Writing – review & editing, Methodology, Formal analysis. **Ramin Faramarzi:** Writing – review & editing, Supervision, Funding acquisition, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

Data will be made available on request.

References

- [1] B. Fricke, T. Kuruganti, J. Nutaro, D. Fugate, J. Sanyal, Utilizing thermal mass in refrigerated display cases to reduce peak demand, (2016).
- [2] Faramarzi R. Efficient display case refrigeration. *ASHRAE J* 1999;41(11):46.
- [3] Faramarzi RT, Coburn BA, Sarhadian R. Performance and energy impact of installing glass doors on an open vertical deli/dairy display case. *ASHRAE Trans* 2002;108:673.
- [4] B. Fricke, B. Becker, Energy use of doored and open vertical refrigerated display cases, (2010).
- [5] Demirpolat H. The energy efficiency and environmental analysis of open-type commercial display cabinet with a multi-flow air curtain design. *Sustainable Energy Technol Assess* 2023;60:103555.
- [6] Cui J, Wang S. Application of CFD in evaluation and energy-efficient design of air curtains for horizontal refrigerated display cases. *Int J Therm Sci* 2004;43(10):993–1002.
- [7] Navaz HK, Faramarzi R, Gharib M, Dabiri D, Modarress D. The application of advanced methods in analyzing the performance of the air curtain in a refrigerated display case. *J Fluids Eng* 2002;124(3):756–64.
- [8] Cao Z, Gu B, Han H, Mills G. Application of an effective strategy for optimizing the design of air curtains for open vertical refrigerated display cases. *Int J Therm Sci* 2010;49(6):976–83.
- [9] Tao W-H, Huang C-M, Hsu C-L, Lin J-Y. Performance study of an energy-efficient display case refrigerator. *Chem Eng Commun* 2004;191(4):550–65.
- [10] D.H. Walker, Investigation of energy-efficient supermarket display cases, in, Oak Ridge National Lab.(ORNL), Oak Ridge, TN (United States), 2005.
- [11] Kishore RA, Mahvi A, Singh A, Woods J. Finned-tube-integrated modular thermal storage systems for HVAC load modulation in buildings. *Cell Rep Phys Sci* 2023;4(12).
- [12] Kishore RA, Bianchi MVA, Booten C, Vidal J, Jackson R. Modulating thermal load through lightweight residential building walls using thermal energy storage and controlled precooling strategy. *Appl Therm Eng* 2020;180:115870.
- [13] Kishore RA, Bianchi MVA, Booten C, Vidal J, Jackson R. Enhancing building energy performance by effectively using phase change material and dynamic insulation in walls. *Appl Energy* 2021;283:116306.
- [14] Wijesuriya S, Booten C, Bianchi MVA, Kishore RA. Building energy efficiency and load flexibility optimization using phase change materials under futuristic grid scenario. *J Clean Prod* 2022;339:130561.
- [15] Zeng Y, Clark R-J, Galazutdinova Y, Odukamaiya A, Al-Hallaj S, Farid M, et al. Open-cycle thermochemical energy storage for building space heating: Practical system configurations and effective energy density. *Appl Energy* 2024;376:124218.
- [16] Bulk A, Odukamaiya A, Simmons E, Woods J. Processing Compressed Expanded Natural Graphite for Phase Change Material Composites. *J Therm Sci* 2023;32(3):1213–26.
- [17] James N, Mahvi A, Woods J. Optimizing phase change composite thermal energy storage using the thermal Ragone framework. *J Storage Mater* 2022;56:105875.
- [18] Ullah KR, Saidur R, Ping HW, Akikur RK, Shuvo NH. A review of solar thermal refrigeration and cooling methods. *Renew Sustain Energy Rev* 2013;24:499–513.
- [19] Batlles FJ, Puertas AM, Romero-Cano MS, Rosiek S, Gil B, Kasperski J, et al. Application of thermal storage in over-night refrigeration of an institutional building. *Sol Energy* 2021;220:450–61.
- [20] Marques AC, Davies GF, Maidment GG, Evans JA, Wood ID. Novel design and performance enhancement of domestic refrigerators with thermal storage. *Appl Therm Eng* 2014;63(2):511–9.
- [21] Azzouz K, Leducq D, Gobin D. Enhancing the performance of household refrigerators with latent heat storage: An experimental investigation. *Int J Refrig* 2009;32(7):1634–44.
- [22] R. Hirmiz, H.M. Teamah, M.F. Lightstone, J.S. Cotton. Performance of heat pump integrated phase change material thermal storage for electric load shifting in building demand side management. *Energy Buildings* 2019;190:103–18.
- [23] Huang R, Mahvi A, James N, Kozubal E, Woods J. Evaluation of phase change thermal storage in a cascade heat pump. *Appl Energy* 2024;359:122654.
- [24] Kishore RA, Bianchi MVA, Booten C, Vidal J, Jackson R. Optimizing PCM-integrated walls for potential energy savings in US Buildings. *Energy Buildings* 2020;226:110355.
- [25] Kishore RA, Bianchi MVA, Booten C, Vidal J, Jackson R. Parametric and sensitivity analysis of a PCM-integrated wall for optimal thermal load modulation in lightweight buildings. *Appl Therm Eng* 2021;187:116568.
- [26] Parameshwaran R, Kalaiselvam S, Hari Krishnan S, Elayaperumal A. Sustainable thermal energy storage technologies for buildings: A review. *Renew Sustain Energy Rev* 2012;16(5):2394–433.
- [27] Alzuwaid F, Ge YT, Tassou SA, Raeisi A, Gowreesunker L. The novel use of phase change materials in a refrigerated display cabinet: An experimental investigation. *Appl Therm Eng* 2015;75:770–8.
- [28] Yilmaz D, Mancuhan E, Yilmaz B. Experimental investigation of PCM location in a commercial display cabinet cooled by a transcritical CO₂ system. *Int J Refrig* 2020;120:396–405.
- [29] Alzuwaid FA, Ge YT, Tassou SA, Sun J. The novel use of phase change materials in an open type refrigerated display cabinet: A theoretical investigation. *Appl Energy* 2016;180:76–85.
- [30] Ben-Abdallah R, Leducq D, Hoang HM, Fournaison L, Pateau O, Ballot-Miguet B, et al. Experimental investigation of the use of PCM in an open display cabinet for energy management purposes. *Energy Convers Manage* 2019;198:111909.
- [31] Wu X, Li W, Wang Y, Chang Z, Wang C, Ding C. Experimental investigation of the performance of cool storage shelf for vertical open refrigerated display cabinet. *Int J Heat Mass Transf* 2017;110:789–95.
- [32] Elarem R, Mellouli S, Abhilash E, Jenni A. Performance analysis of a household refrigerator integrating a PCM heat exchanger. *Appl Therm Eng* 2017;125:1320–33.
- [33] Cheng W-L, Mei B-J, Liu Y-N, Huang Y-H, Yuan X-D. A novel household refrigerator with shape-stabilized PCM (Phase Change Material) heat storage condensers: An experimental investigation. *Energy* 2011;36(10):5797–804.
- [34] Bakhsipour S, Valipour MS, Pahamli Y. Parametric analysis of domestic refrigerators using PCM heat exchanger. *Int J Refrig* 2017;83:1–13.
- [35] Rahimi M, Ranjbar AA, Hosseini MJ. Experimental investigation on PCM/fin slab incorporation in an evaporator side of a household refrigerator. *Energy Rep* 2023;10:407–18.
- [36] Zarababab OG, Ahmadi R. Employment of finned PCM container in a household refrigerator as a cold thermal energy storage system. *Therm Sci Eng Prog* 2018;7:115–24.
- [37] Pirvaram A, Sadrameli SM, Abdolmaleki L. Energy management of a household refrigerator using eutectic environmental friendly PCMs in a cascaded condition. *Energy* 2019;181:321–30.
- [38] Yusufoglu Y, Apaydin T, Yilmaz S, Paksoy HO. Improving performance of household refrigerators by incorporating phase change materials. *Int J Refrig* 2015;57:173–85.
- [39] Liu G, Li Q, Wu J, Xie R, Zou Y, Marson A, et al. Improving system performance of the refrigeration unit using phase change material (PCM) for transport refrigerated vehicles: An experimental investigation in South China. *J Storage Mater* 2022;51:104435.
- [40] P. Principi, R. Fioretti, B. Copertaro, Energy saving opportunities in the refrigerated transport sector through Phase Change Materials (PCMs) application, in, IOP Publishing, pp. 012043.
- [41] rubitherm.eu, Data sheet | SP-11: www.rubitherm.eu/media/products/datasheets/Techdata-SP-11_EN_12072022.PDF Accessed on June 25, 2025.
- [42] julabo.com, Data sheet | Thermal C5: <https://www.julabo.com/en-us/products/accessories/bath-fluids-water-bath-protective-media/thermal-c5-8891403> Accessed on June 25, 2025.
- [43] Mahvi A, Shete KP, Odukamaiya A, Woods J. Measuring the maximum capacity and thermal resistances in phase-change thermal storage devices. *J Storage Mater* 2022;55:105514.
- [44] Kocher JD, Woods J, Odukamaiya A, Mahvi A, Yee SK. Thermal battery cost scaling analysis: minimizing the cost per kW h. *Energy Environ Sci* 2024;17(6):2206–18.
- [45] Mills A, Farid M, Selman JR, Al-Hallaj S. Thermal conductivity enhancement of phase change materials using a graphite matrix. *Appl Therm Eng* 2006;26(14–15):1652–61.
- [46] Woods J, Mahvi A, Goyal A, Kozubal E, Odukamaiya A, Jackson R. Rate capability and Ragone plots for phase change thermal energy storage. *Nat Energy* 2021;6(3):295–302.
- [47] Kommandur S, Mahvi A, Bulk A, Odukamaiya A, Aday A, Woods J. The impact of non-ideal phase change properties on phase change thermal energy storage device performance. *J Storage Mater* 2023;72:108470.
- [48] Biswas K, Abhari R. Low-cost phase change material as an energy storage medium in building envelopes: Experimental and numerical analyses. *Energy Convers Manage* 2014;88:1020–31.
- [49] Biswas K, Lu J, Sorousian P, Shrestha S. Combined experimental and numerical evaluation of a prototype nano-PCM enhanced wallboard. *Appl Energy* 2014;131:517–29.
- [50] media.copeland.com, Copeland scroll for R290 applications: <https://media.copeland.com/c35cd3b3-ac35-40ec-9fc6-b2ab00acba9/DSC167-Comfort-Propane-Scroll-Compressors-EN.pdf> Accessed on June 25, 2025.
- [51] webapps.copeland.com, Copeland, Copeland Online Product Information, 2025. Available from: <https://webapps.copeland.com/online-product-information/>.
- [52] Gilleran M, Bonnema E, Woods J, Mishra P, Doeber I, Hunter C, et al. Impact of electric vehicle charging on the power demand of retail buildings. *Adv Appl Energy* 2021;4:100062.