The Use of Phase Change Materials in Heating Buildings: A Solar-adaptive Passive System

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Abstract

This research presents a novel solar-adaptive passive heating system called Sol-PH. Sol-PH integrates Phase Change Materials (PCM) to offer passive heating while meeting ventilation requirements in winter. The thermal behaviour of PCM panels with different PCM-encapsulated cell shapes, orientations, cell sizes, and inlet air velocity was comprehensively studied. Simulation results reveal that PCM encapsulated within hexagonal cells with a horizontal orientation and a radius of 16.5mm exhibit superior thermal performance, as evidenced by the highest average outlet air temperature and a substantial enthalpy change over time. Implementing the proposed system on the south-facing façade of a 20m² office space in Boston, with a balanced mixture of indoor and outdoor air, results in 19.7% of energy reduction.

Highlights

- PCM-integrated solar-adaptive passive heating system.
- Parametric study on PCM-encapsulated cell shape, orientation and size.
- Hexagonal cells with horizontal orientation and smaller cell size offer the best thermal performance.
- The proposed system can provide passive heating while meeting ventilation requirements, contributing to about 19.7% heating energy savings.

Introduction

Thermal energy loss through building envelope significantly affects building energy efficiency, particularly in glazing systems that are commonly used for outdoor visual connections and daylighting. Integrating phase change materials (PCM) into building elements presents substantial opportunities for improving overall building performance and energy efficiency. PCM possesses the ability to store and release heat within a narrow temperature range. Researches exploring applications of PCM integration in building elements such as concrete, glass, and gypsum board have demonstrated notable benefits in reducing peak loads, energy consumption, and enhancing thermal comfort (Erlbeck et al., 2018; Karapetký et al., 2016).

However, the low thermal conductivity of PCM, whether organic or inorganic, poses challenges as it slows down the exchange of thermal energy between the PCM and its environment, resulting in system inefficiencies. Many thermal conductivity enhancement techniques have been proposed to address this issue, including using porous media such as metal foams (Wang et al., 2019), incorporating nanoparticles (Ebadi et al., 2018), employing extrinsic methods like embedded fins (Kamkari & Shokouhmand, 2014), and adopting PCM encapsulation structures (Hasse et al., 2011; Liu et al., 2018) to increase the heat transfer area. The latter two methods have demonstrated more positive outcomes and hold higher potential with comparatively lower cost and convenience. Notably, the utilization of external fins and honeycomb structures has demonstrated significant enhancements in the thermal performance of PCM macro-encapsulation within the building envelope (Liu et al., 2018). Studies have also shown that incorporating aluminum honeycomb as a structural support can greatly enhance the thermal conductivity of building wallboards, facilitating rapid heat transfer into the PCM (Lai & Hokoi, 2014).

Moreover, it is worth noting that different geometrical shapes of PCM encapsulated shells have varying effects on the melting rate of PCM (Erlbeck et al., 2018; Liu et al., 2018). Recent studies have provided evidence of distinct melting rates of PCM in metallic cylindrical and rectangular encapsulates (Raj et al., 2019). Furthermore, the shape of PCM inclusions significantly influences the dynamic thermal behavior during the melting process (Dhaidan & Khoddadi, 2015). However, there is a lack of comprehensive research on evaluating the impact of various design parameters on the thermal performance of PCM-integrated elements. Specifically, there is a need to investigate how the thermal performance of PCM panel can be enhanced through passive means by altering the design of the encapsulated shell, without relying on the addition of any chemicals or additives.

This study, therefore, proposes a novel solar-adaptive PCM-integrated passive heating system called Sol-PH. This decentralized system is mounted on the glass façade and is specifically designed to provide passive heating by harnessing solar radiation while meeting the ventilation requirements. Sol-PH incorporates an aluminum honeycomb structure to encapsulate PCM and utilizes an optimized shell size to achieve optimal thermal performance (Figure 1). To identify the most efficient Sol-PH system design, comprehensive parametric studies were conducted to analyze the effects of various factors on the thermal performance of the
PCM-integrated panel. This includes the geometrical form of the PCM encapsulated shell, the shell orientation, the size of the shell, the stack structure of individual shells, and the inlet air velocity. Furthermore, the study assesses the energy-saving potential associated with the implementation of the proposed Sol-PH system. Sol-PH offers an innovative and comprehensive solution by integrating passive heating and ventilation to improve building energy efficiency while ensuring the occupants' thermal and visual comfort.

**Conceptual Design of Sol-PH System**

Figure 1 shows the conceptual design of Sol-PH system which integrates PCM and offers climate-responsive capabilities to store and utilize solar radiation for heating purposes. The system is mainly designed for colder climates where heating demands exceed cooling requirements annually. With dimensions of 700mm x 900mm, the system consists of two key components: the air mixing box and the PCM-integrated energy storing panels. The air mixing box, positioned at the bottom, extracts and mixes the indoor cool air near the glass facade (assumed to be 15°C) and outdoor cold air (assumed to be 0°C) to meet the ventilation requirements. PCM panels are formed by a stack of individual hexagon PCM-contained cells.

![Figure 1. Solar-adaptive PCM Integrated Passive Heating System (Sol-PH) Conceptual Design.](image)

Sol-PH functions as an advanced passive thermal control system designed to absorb latent heat from the sun during the morning to early afternoon period. Once fully charged, the air mixing box extracts air from both indoor and outdoor environments, then directs the mixed air into the charged PCM panels. The PCM panels release the stored energy, effectively heating the mixed cool air, and thus remain the space warm without rely on the mechanical heating system. Figure 2 illustrates the energy storing and releasing processes of the Sol-PH system. Blue and red of PCM panels represent the solid and liquid states of the material respectively. By absorbing solar radiation, solar energy is stored in PCM panels. As the temperature gradually rises beyond the phase change temperature of the PCM, the PCM encapsulated within the hexagon shell begins to melt (Figure 2(a)). Figure 2(b) illustrates the heat transfer effect to warm the cold air. At this stage, the PCM undergoes the solidification process.

![Figure 2. (a) Energy storing. PCM melting Process; (b) Energy releasing. PCM solidification process.](image)

The entire system is seamlessly integrated into the glass curtain wall on the glass façade as shown in Figure 3. To preserve external aesthetics, ensure even daylight distribution, and maintain visual continuity with the outdoors, PCM panels are covered with the single-pane glass which is designed with a solar heat gain coefficient of 0.86, allowing for optimal heat absorption during the charging process.

![Figure 3. Dimension and Installation of the Sol-PH system in relation to the glazing façade](image)

The study aims to achieve the following objectives:

- Conduct a comprehensive parametric study to evaluate the impact of various design parameters on the thermal performance of PCM-integrated panel, including the geometrical form of the PCM encapsulated shell, the shell orientation, the size of the shell, and the inlet air velocity.
- Develop an innovative solar-adaptive PCM-integrated passive heating system to provide passive heating and fulfil ventilation requirements.
- Evaluate the energy-saving potential of the proposed Sol-PH system under Boston climate context.

**Methodology**

The study utilized ANSYS Fluent 16.1 software to conduct computational simulations and analyze the thermal performance of the Sol-PH system. The thermal behavior of the PCM was simulated using melting and solidification models in Fluent, the model function is based on enthalpy porosity techniques. The Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm was chosen to solve the numerical equations, considering both velocity and pressure fields. For the convective terms of momentum and energy equations, the spatial discretization method employed
the second-order upwind scheme (Wesseling, 2009). Under-relaxation factors of 0.3, 1.0, 0.7, and 0.9 were used for pressure, density, momentum, and energy, respectively. Solution convergence was assessed at each time step, setting scaled residuals to $10^{-6}$ for continuity and momentum equations, and $10^{-10}$ for the energy equation. To simplify the problem, all simulations were conducted in two dimensions. The validation and calibration of the Fluent model were performed by comparing the simulation results of the previous study that analyzes the melting rate of PCM with different shape of enclosure (Duan et al., 2019).

The modelling process made the following assumptions:  
• The melting and solidification process of PCM is considered to be transient.  
• The direction of the gravitational acceleration is consistently opposite to the y-axis and has a constant value of 9.81 m/s².  
• Volume expansion of the PCM is neglected.  
• No internal heat generation occurs within the PCM.

Model Description
An inorganic PCM consisting of salt hydrates with a melting temperature of 35°C was chosen for modelling and simulations in this study. Detailed properties of the selected inorganic PCM are presented in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>C_p,s</td>
<td>2800</td>
<td>J/(kg K)</td>
</tr>
<tr>
<td>C_p,l</td>
<td>3200</td>
<td>J/(kg K)</td>
</tr>
<tr>
<td>L</td>
<td>220000</td>
<td>J/kg</td>
</tr>
<tr>
<td>p_pcm</td>
<td>800</td>
<td>kg/m³</td>
</tr>
<tr>
<td>T_m</td>
<td>35</td>
<td>°C</td>
</tr>
<tr>
<td>T_f</td>
<td>37</td>
<td>°C</td>
</tr>
</tbody>
</table>

Table 1. Properties of inorganic PCM

Influence of the Inlet Air Velocity & Geometrical Shape
The first parametric study was conducted to examine the influence of inlet air velocity, geometrical shape of PCM-encapsulated shell and its orientation on the thermal behavior of PCM panels and the resulting outlet air temperature. Inlet air velocity varied within the range from 0.1 m/s to 1.0 m/s, while three different shapes and orientations of PCM containers were simulated: cylinder, hexagon with vertical orientation (hexagon-vertical), and hexagon with horizontal orientation (hexagon-horizontal) (Figure 4). The total cross-sectional areas of these cells and the total number of cells in each panel were kept the same, thereby maintaining a fixed PCM volume and same thermal storage capacity for all three models.

Figure 4. Geometrical shapes and orientations of PCM cells selected for parametric study.

Figure 5 illustrates simulation model set-up with the flow of cold air into the PCM panel with a specified velocity. As the cold air interacts with the charged PCM cells, it absorbs thermal energy and undergoes a temperature increase. The heated air then exits the PCM panel through the air outlet located at the top. The boundary conditions for the left and right side walls of the panel were assumed to be adiabatic. To account for the high thermal conductivity of aluminum, which was used for the PCM encapsulation shell frame, all boundaries of the numerically simulated cells were set to have an isothermal condition.

Figure 5. PCM Panel Model set-up for simulations

Influence of the PCM-encapsulated Cell Size
The second parametric study aimed to evaluate the influence of individual PCM cell size on the thermal performance of the system. For this purpose, three distinct sizes of PCM shells were simulated and analyzed. Model A, B, C represent PCM cells with a radius of 33mm, 22mm, 16.5mm respectively, as depicted in Figure 6. The selection of these specific radii was based on the requirement of maintaining a consistent total volume of PCM within each panel, this ensured a uniform thermal capacity across all three models.

Figure 6. Model set-up for simulating the PCM cell size

Feasibility Study for Energy Saving Potential
Furthermore, to assess the energy-saving potential of the proposed Sol-PH system and ensure compliance with ventilation requirements, the optimized design configuration of the system was evaluated through simulating three different scenarios, as outlined in Table 2. The mixed air temperature is determined using Equation 1, which considers the outdoor ($T_{\text{out}}$) and indoor ($T_{\text{in}}$) air temperatures, the percentage of outdoor air ($P_{\text{OA}}$) and indoor air ($P_{\text{IA}}$) in volume, and the
efficiency (eff) of the mixing process. Assuming an 80% effectiveness during the mixing process, scenario 1 mixes an equal amount of indoor air (15°C), and outdoor cold air (0°C), resulting in the mixed air temperature of 6°C. Scenario 2 involves combining two portions of indoor air (15°C), with one portion of outdoor air (0°C), resulting in mixed air temperature of 8°C. In scenario 3, the amount of indoor air at 15°C is three times greater than the amount of outdoor air at 0°C, leading to the mixed air temperature of 9°C. ASHRAE Standard 62.1 2013 (Standard, 2013) recommends a basic combined office fresh air supply rate of 8.5 L/s per person. In our case, considering a 40 m² office accommodating 16 people, the mass flow rate ($\dot{m}$) of fresh air is calculated using Equation 2, taking into account the density of air ($\rho$) and the volume flow rate of air ($V$) that is derived from multiplying the total number of people by the required fresh air supply rate per person. Since each proposed system unit consists of four PCM panels, the mass flow rate of fresh air directed into each PCM panel is calculated to be 0.041 kg/s for all three scenarios, ensuring the fulfilment of the ventilation requirement. The air velocity ($v$) under different scenarios is calculated using Equation 3, where $A$ is air inlet area of PCM panel.

\[
T_{\text{mix}} = \frac{T_{\text{out}} \times \rho_{\text{OA}} + T_{\text{in}} \times \rho_{\text{IA}}}{\rho_{\text{OA}}} \times \text{eff} \quad \text{Equation 1}
\]

\[
\dot{m} = \rho \times V \quad \text{Equation 2}
\]

\[
v = \frac{\dot{m}}{\rho \times A} \quad \text{Equation 3}
\]

Table 2. Summary of three scenarios simulated for assessing the energy saving potential.

<table>
<thead>
<tr>
<th>Scenario 1</th>
<th>Scenario 2</th>
<th>Scenario 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate of 0°C outdoor air ($\dot{m}_{\text{OA}}$)</td>
<td>0.041 kg/s</td>
<td>0.041 kg/s</td>
</tr>
<tr>
<td>Mass flow rate of 15°C indoor air ($\dot{m}_{\text{IA}}$)</td>
<td>0.041 kg/s</td>
<td>0.082 kg/s</td>
</tr>
<tr>
<td>Total Mass flow rate in one PCM panel ($\dot{m}_{\text{total}}$)</td>
<td>0.082 kg/s</td>
<td>0.123 kg/s</td>
</tr>
<tr>
<td>Inlet Air Velocity</td>
<td>0.45 m/s</td>
<td>0.66 m/s</td>
</tr>
<tr>
<td>Mixed air Temperature (80% effectiveness)</td>
<td>6°C</td>
<td>8°C</td>
</tr>
</tbody>
</table>

**Initial and Boundary Conditions**

The well-defined initial and boundary conditions were set to ensure a good processing in simulations. At the start of the simulations ($t = 0s$), the initial temperature of PCM was set to its final liquidus temperature ($T_f$) (i.e. $T_f = 37°C$). This initial condition assumes that the PCM is in a fully charged liquid state at the beginning of the simulations. The specific initial and boundary conditions for the different models are summarized in Table 3.

Table 3. Initial and boundary conditions for different simulation models

<table>
<thead>
<tr>
<th>Scenario 1:</th>
<th>Scenario 2:</th>
<th>Scenario 3:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial conditions</td>
<td>$t = 0s$</td>
<td>$t = 0s$</td>
</tr>
<tr>
<td>$T_{\text{inlet}} = 15°C$</td>
<td>$T_{\text{inlet}} = 15°C$</td>
<td>$T_{\text{inlet}} = 8°C$</td>
</tr>
<tr>
<td>$V_i(t) = 0$</td>
<td>$V_i(t) = 0$</td>
<td>$V_i(t) = 0$</td>
</tr>
</tbody>
</table>

**Simulation Results and Analysis**

**Influence of Inlet Air Velocity & Geometrical Form**

Figure 7 presents the correlation between the outlet air temperature and the inlet air velocity for different geometrical shapes of the PCM container. Generally, a higher inlet air velocity, which corresponds to a higher mass flow rate, leads to a lower outlet air temperature. Among the various geometrical shapes, the hexagon-horizontal configuration exhibits the best thermal performance, followed by hexagon-vertical and cylinder.

To gain further insights into the influence on the outlet air temperature of the proposed system, the study conducted simulations of the inlet air flow (with an initial temperature of 15°C) through PCM panels with different geometrical shapes for a duration of 1000s. Figure 8, Figure 9, and Figure 10 present the simulation results for panels with cylindrical, hexagonal-vertical, and hexagonal-horizontal shaped PCM cells, respectively. Consistent with the findings in Figure 7, the thermal behaviour of PCM in composite cells with different shapes and orientations varies. In general, a lower air velocity allows for a longer duration of heat transfer between the cold air and the heated PCM cells, leading to higher temperatures of the discharged outlet air. For instance, as depicted in Figure 8, when the inlet air velocity is 0.1 m/s, the outlet air temperature reaches to 32°C. However, with an increase in the inlet air velocity to 0.5 m/s and 1.0 m/s, the outlet air temperatures decrease to 23°C and 21°C, respectively.
The hexagon-vertical PCM encapsulated shell delivers higher outlet air temperatures, demonstrating its superior thermal performance compared to cylindrical counterparts. As shown in Figure 9, over a 1000s duration, the hexagon-vertical PCM consistently achieves outlet air temperatures that are 2-3°C higher than those of cylindrical PCM panel at various inlet air velocities. This observation aligns with the notion that hexagons offer a larger heat transfer surface area relative to cylinders, resulting in improved thermal performance. However, it is noted that some heat is trapped at the intersection between two hexagon-vertical cells, leading to ineffective heat exchange.

The hexagon-horizontal PCM cells exhibit the most favourable thermal performance comparatively. Figure 10 demonstrates that, with an inlet air velocity of 0.5 m/s, the hexagon-horizontal PCM panel achieves an outlet air temperature of 26°C, surpassing the temperatures delivered by cylindrical and hexagon-vertical cells by 3°C and 2°C, respectively. Furthermore, when the inlet air velocity increases to 1 m/s, the hexagon-horizontal PCM panel attains the highest outlet air temperature among all the simulated shapes, reaching 25°C. Moreover, this model does not exhibit the phenomenon of heat being trapped at the cell intersection, indicating more effective heat transfer between the PCM-encapsulated cells and the cold air.

The relationship between inlet air velocity and enthalpy change for different geometrical shapes of the PCM container is investigated in this study. The enthalpy change is calculated using Equation 4, which quantifies the energy transfer during the heat exchange process. Figure 11 reveals that hexagon-horizontal cells consistently exhibit the highest enthalpy change across all values of inlet air velocities compared to hexagon-vertical and cylindrical PCM containers. This finding indicates that, with the same inlet air velocity, hexagon-horizontal cells have the capability to transfer more energy and effectively heat up the cold air compared to other shapes. Therefore, the geometrical characteristics of the PCM container play a significant role in its thermal behaviour, with hexagonal cells featuring a horizontal orientation demonstrating superior thermal performance in this study.

Even though lower inlet air velocities generate higher outlet air temperatures, it also leads to lower enthalpy change (Figure 11), which will ultimately affect the overall effectiveness of the system. To further illustrate this point, we investigated the liquid fraction remaining in the PCM cells after 10,000s of airflow duration for different inlet air velocities. As shown in Figure 12, at an inlet air velocity of 0.1 m/s, the liquid fraction remaining in the PCM cells is 80%, indicating that only 20% of the stored energy has been released. This is an indication that the system does not function optimally in a desired manner. Thus, it is crucial to establish a balance between the inlet air velocity and its corresponding enthalpy change in order to ensure optimal system operation and achieve the desired outcomes.
Figure 11. Relationship between the inlet air velocity and the enthalpy change with different geometrical shapes of the PCM container

Figure 12. Liquid fraction of PCMs after 10,000s operation with different inlet air velocity

Influence of the PCM-encapsulated Cell Size

To further optimize the design of PCM-integrated panels from a geometrical perspective, the impact of individual PCM cell sizes on the system's thermal behavior was studied. Building upon the previous finding that the hexagon-horizontal cell yielded the best thermal performance, the subsequent analysis focused on this specific configuration. Simulations were carried out with an inlet air temperature of 15°C for a duration of 5000s. As depicted in Figure 13, at a relatively low inlet air velocity of 0.3 m/s, all three models achieved the desired outlet air temperatures of 32°C, 33.7°C, and 35.5°C for cell radii of 33mm, 22mm, and 16.5mm, respectively. However, when the inlet air velocity was increased to 1.0m/s, the average outlet air temperatures delivered by the 33mm and 22mm hexagonal cells fell below 30°C. Conversely, the hexagonal cell with a radius of 16.5mm maintained an average outlet air temperature of 31.1°C. Across different levels of inlet air velocities, the cells with a radius of 16.5mm consistently delivered the highest average outlet air temperatures compared to the other two models.

Furthermore, the outlet air temperature and the corresponding enthalpy change for containers with different cell sizes were analyzed. The simulations were conducted under three scenarios with air velocities of 0.3m/s, 0.5m/s, and 1.0m/s. It was observed that, in all three cases, smaller cell sizes resulted in higher outlet air temperatures and larger enthalpy changes. Therefore, smaller cell sizes exhibited superior thermal performance in PCM panels, as they achieved optimal outcomes in terms of delivering the highest average outlet air temperature and achieving larger enthalpy changes over time.

Feasibility Study for Energy Saving Potential

In order to evaluate the energy-saving potential of the proposed system and fulfill ventilation requirements, three specific scenarios outlined in Table 2 were investigated. Simulations were carried out until the point at which the liquid fractions of the PCM decreased to 10%, as a liquid fraction of 10% signifies a substantial utilization of the stored thermal energy.

Figure 14 depicts the temporal evolution of the average outlet air temperature for the three studied scenarios. It is evident that as the duration of the airflow increases, the average outlet air temperature decreases across all scenarios. After 1000s of airflow, all three scenarios achieve the desired outlet air temperature of above 30°C (34.1°C for scenario 1, 33.0°C for scenario 2, and 31.7°C for scenario 3). However, with a prolonged airflow duration of 6000s, both scenario 2 and scenario 3 result in average outlet air temperatures below 30°C. Specifically, scenario 3 exhibits an average outlet air temperature of 27°C, while scenario 2 yields 29.6°C. In contrast, even after 6000s of airflow, scenario 1 maintains an average outlet air temperature of 32.1°C, fulfilling the desired objective. Although, scenario 3 has the highest inlet air temperature (i.e., 9°C) compared to scenario 1 (i.e., 6°C) and scenario 2 (i.e., 8°C), it requires the largest mass flow rate which is controlled directly by the inlet air velocity, resulting in lower corresponding outlet air temperature. This observation suggests that the impact of the mass flow rate surpasses the influence of the inlet air temperature in determining the overall system performance in this case study.

The liquid fraction of the PCM after certain durations was examined. The solidification process of PCM involves the release of stored thermal energy, and thus the liquid fraction of PCM serves as an indicator of the remaining thermal energy content. Figure 15 illustrates the liquid fraction remaining in the PCM cells for different time durations of cold air flow (1000s, 3000s, and 6000s) for the three simulated scenarios. It is observed that across all airflow durations, scenario 1 consistently exhibits the highest liquid fraction remaining in the PCM cells, with the slowest rate of decrease. After an operation time of 6000s, the liquid fraction of PCM in scenario 3 reaches approximately 10%, indicating that the stored energy within the PCM panel has been extensively utilized. In contrast, for scenario 1 and scenario 2, the liquid fractions in the PCM panel remain at approximately 30% and 17%, respectively. This suggests that a substantial amount of thermal energy remains in the PCM panel, and the system can continue to operate beyond 6000s for both scenario 1 and scenario 2.
By heating exhibit a air, an heat Sol-PH provide 20m consistently respectively. Heating energy the different a a 13.1% and the system from hours) Sol-PH the energy 20m the shown enthalpy a rate (with liquid in system for space office (d) system; + energy of the a implementing ∆ air the a the an Figure (c) increase is energy Where liquid the liquid the inlet assessed a purposes shorter 40m the and the scenario office the system Changes system is evaluate heat (60m the a be which larger resulting the temperature of the fraction effective PCM-integrated heating PCM-integrated Sol-PH the scenarios. system for ultimately liquid operation generated of the Figure scenarios the outlet of hours, loss − indoor 1 40m proposed in leading G in × efficiency by in scenarios total 3 16 3. of Evolution the inlet can was and 17 heating bars the the was designed temperature of the improvement of building envelope, Pfan represents the fan power in the air mixing box.

\[ Q_{\text{enthalpy}} = \dot{m} \times c_p \times |T_{\text{outlet}} - T_{\text{inlet}}| \] \hspace{1cm} Equation 4

\[ Q_{\text{heat}} = \dot{m} \times c_p \times \Delta T - G + L - P_{\text{fan}} \] \hspace{1cm} Equation 5

By implementing scenario 1 conditions on a winter day in Boston with strong solar radiation, the outdoor air temperature was assumed to range from -6°C to 1°C, between 8am and 6pm. By installing the proposed system on a south-facing façade, Figure 17 illustrates the heating loads variation with and without the proposed system. The blue bars represent the heating loads in the space during the daily operation hours, while the orange bars are the energy saving potential delivered by the Sol-PH system (Figure 17(a) & (c)). By installing the system in a 40m² (120m³) office space and offsetting the energy delivered by Sol-PH system, a 13.1% of total energy savings was achieved. If the proposed system was installed in a 20m² (60m³) room, the overall energy saving potential for heating purposes can increase to 19.7%. Figure 17(b) and (d) exhibit the reduced heating loads resulting from the installation of the Sol-PH system. Since the system was designed to meet the minimum ventilation requirements for a 40m² office space, allocating the system in a 20m² room can provide an even better ventilation.

The energy saving potential provided by the proposed Sol-PH system was assessed by calculating the enthalpy change of the system over 3 hours of discharging process, as shown in Equation 4. Meanwhile, the heating loads was calculated using Equation 5. Where \( Q_{\text{enthalpy}} \) and \( Q_{\text{heat}} \) are the enthalpy change of the system and heating loads respectively. \( \dot{m} \) represents the mass flow rate of air, and \( c_p \) is the specific heat capacity of air. \( T_{\text{inlet}} \) and \( T_{\text{outlet}} \) stand for the inlet and outlet air temperatures from the system respectively. \( \Delta T \) is the temperature difference between outdoor air and system supply air. \( G \) is the internal heat gain, \( L \) refers to the heat loss through building envelope, \( P_{\text{fan}} \) represents the fan power in the air mixing box.

Figure 14. Changes in average outlet air temperature over time under three different scenarios

Figure 15. Evolution of liquid fraction over time for different scenarios

Figure 16 provides a comprehensive view of the evolution of outlet air temperature and liquid fraction for the three simulated scenarios. Figure 16(a) demonstrates that scenario 1 consistently achieves the highest outlet air temperature throughout the longest operation time when compared to scenarios 2 and 3. Figure 16(b) reveals that scenario 1 sustains an effective operation duration of 9000s (2.5 hours) before the liquid fraction inside the PCM panel decreases to 10%. In contrast, scenario 2 and scenario 3 exhibit shorter effective operation durations of 7000s and 6000s, respectively. All scenarios are designed to satisfy the ventilation requirements. Despite scenario 1 incorporating a smaller fraction of indoor air, it leads to a lower mass flow rate and inlet air velocity, ultimately resulting in a desirable outcome over an extended operation period. Conversely, although scenarios 2 and 3 generate larger amounts of heat at the initial stage, it appears to be excessive as the heat generated surpasses the required loads, leading to shorter operation duration. In conclusion, scenario 1, which involves an equal mixture of indoor and outdoor air, exhibits the highest energy-saving potential among the studied scenarios.

Figure 16. (a) Evolution of outlet air temperature over time; (b) Evolution of liquid fraction over time

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Conclusion

This study introduces a novel solar-adaptive passive heating system, Sol-PH, which integrates PCM to facilitate passive heating while simultaneously fulfilling ventilation requirements. Extensive investigations were conducted to analyse the thermal characteristics of PCM with different PCM-encapsulated cell shapes, orientation, and cell sizes. Additionally, energy saving potential of Sol-PH was assessed under three different scenarios. The results indicate that PCM encapsulated within hexagon cells with a horizontal orientation and a cell size of 16.5mm exhibits superior thermal performance. This can be attributed to the larger surface area available for heat transfer and the unobstructed airflow pathway within the panel. This design configuration yields the highest average outlet air temperature and a larger enthalpy change over time. Furthermore, the study reveals that by employing an equal mixture of indoor and outdoor air and installing the Sol-PH system in a 20m² office space located in Boston, 19.7% of energy reduction can be achieved. It should be noted, however, that the actual energy-saving potential is influenced by various factors such as the specific location, facade orientation, solar radiation intensity, choice of glazing materials, and room size, etc. The comprehensive sensitivity analysis and uncertainty analysis will be included in the future works. However, challenges related to timed control of thermal energy release still exist. Promising advancements in photo-switch technology (Han et al., 2017) may address these concerns in the near future. Additionally, the design of interchangeable PCM panels allows for the use of different PCM materials with varying melting points, enabling adaptation to changing seasonal requirements. In conclusion, this study, along with others, highlights the potential of PCM as a natural resource-driven approach for reducing building energy consumption and presents avenues for further exploration.

References