Reducing energy consumption of heat pumps by introducing a thermal energy storage system

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Abstract. We proposed a short-term thermal energy storage system by applying Phase Change Materials (PCMs) in air ducts connected to an indoor air conditioning unit within a ceiling space. PCM tubes were arranged in staggered layout in the air duct and cooled by chilled air from the air conditioning unit during “charging” mode, whereas the PCM tubes released cold energy to the room space without operation of the compressor during “discharging” mode. The charging mode allowed for an increase in the partial load ratio of the heat pump to prevent low-efficiency operation at <20% load, which can increase seasonal energy efficiency. We constructed an energy simulation program, where the heat exchange coefficient between the PCM tubes and air was varied with airflow rate and the temperature difference between the PCM and air, in accordance with results from CFD analysis. Switching between the charging and discharging modes was based on the outlet temperature (Ton) and 30% of the rated capacity of the indoor unit (Li). The heat balance between the charging and discharging modes was improved by this change to the system control. A higher load factor and higher COP resulted in a 7.6% reduction in energy consumption.

1 Introduction

The Japanese government has promoted several activities to achieve a carbon-neutral society by 2050 [1]. Improving the energy efficiency of building equipment and producing renewable power are important actions to reduce CO2 produced through energy usage, particularly in the context of air conditioning [2]. Heat pumps have attracted global attention as an energy-efficient air-conditioning technology. Inverter-equipped heat pumps are common in Japan and allow efficient operation during both partial load ratios and larger load ratios that are close to the compressor capacity [3]. However, inverter control is invalid at load ratios <20%. At these small load ratios, the compressor is operated with inefficient on-off control [4]. Air conditioning equipment is generally designed with a surplus to provide sufficient cooling and heating in buildings, even during the warmest/coldest days of the year. Most heat loads that occur in a building are partial loads, with relatively short-lived peaks for several tens of hours [5–7].

To reduce energy consumption and CO2 emissions, it is therefore necessary to improve energy efficiency at low load ratios (i.e., in the range below the lower limit of the inverter control). In this study, we address this challenge with a novel system for heat exchange between the phase change materials (PCMs) which filled in multiple vertical tubes and the air in air ducts of conventional individual air conditioning systems.

The proposed system allows heat pumps to increase the actual load ratios and operate with effective inverter control by simultaneously cooling rooms and PCMs. Once the PCMs complete cooling storage, the stored thermal energy is discharged to the room without operation of the compressor. After heat exchange with PCMs, the outlet air temperature is maintained at approximately phase-change temperatures for 1-2 hours during both charging and discharging [8].

In this study, we constructed an energy simulation program, where we varied the heat exchange coefficient between the PCM tubes and the air according to airflow rate and temperature difference, in accordance with 3D Computational Fluid Dynamic (CFD) analysis. Finally, we compared seasonal energy consumption between scenarios with and without PCM tubes, assuming standard office building cooling loads.

2 Thermal energy storage system

A conceptual schematic of the proposed system is shown in Figure 1. The PCM was installed in multiple vertical tubes within an additional air duct connected to an indoor air conditioning unit in the ceiling space. Inlet air from the room space passes through an evaporator, through the PCM duct, and then flows back into the room space. The PCM was cooled by chilled air from the evaporator in the “charging” mode, whereas the PCM released cold energy to the room space without operation of the compressor in the “discharging” mode. These charging and discharging modes repeated during a cooling period. This operation allowed an increase in the partial load ratio of the heat pump to prevent low-
efficiency operation at <20% load, which increased seasonal energy efficiency.

Figures 2 and 3 show the plan and section views of the PCM duct model, respectively. The PCM duct is 20 cm wide, 20 cm high, and 200 cm long in the air flow direction.

The PCM was assumed to be enclosed in a cylindrical tube with a diameter of 1 cm and height of 20 cm, as shown in Figure 4. A total of 198 PCM tubes were arranged in a staggered layout with six tubes per row and 33 rows in the flow direction.

3 Determining the heat exchange coefficient based on CFD analysis

3.1 CFD analysis condition

We performed CFD analysis (see Table 1) using Ansys Fluent to determine the heat exchange coefficient ($K_{ex}$) by considering the spatial and temporal variation of the phase change status in the PCM duct. We conducted simulations assuming charging mode, in which chilled air from the indoor air conditioning (AC) unit was introduced into the duct and cooled the PCM. To obtain the heat exchange performance during solidification of the PCM, we carried out a non-steady state analysis for a maximum duration of 50 min. The inlet temperature was constant at 12.6 °C. We assumed that the phase change of the PCM ranged from 16.2 to 20.2 °C, with a latent heat of 229 kJ/kg, based on Differential Scanning Calorimetry (DSC) analysis of sample material. A total of 2.58 kg of PCM was installed in the duct.

3.2 Calculation of heat exchange coefficients

We calculated variation in both air and PCM temperature in the duct for air flow volumes of 300, 700, and 1000 m$^3$/h. The heat exchange coefficient, $K_{ex}$, was determined by the following equation:

$$K_{ex} = \frac{Q}{A \Delta T_{air-pcm}}$$

$$Q = C_a p_a V \Delta T_{in-out}$$

$$\Delta T_{air-pcm} = T_{air} - T_{pcm}$$

Where $K_{ex}$ is the heat exchange coefficient between air and PCM (W/m$^2$/K); $Q$ is the amount of heat exchanged between air and PCM (W); $A$ is the total surface area of PCM tubes (m$^2$); $C_a$ is the specific heat of air (J/kg/K); $p_a$ is the density of air (kg/m$^3$); $V$ is the air supply flow volume (m$^3$/h); $\Delta T_{in-out}$ is the temperature difference between the duct inlet and outlet (°C); $T_{air}$ is the average air temperature (°C); and $T_{pcm}$ is the average PCM temperature (°C).

3.3 Comparison of conventional and modified heat exchange coefficients

Figure 5 shows the $K_{ex}$ values calculated from CFD analysis, compared with previous values from theoretical analysis for a staggered tube layout. The previous $K_{ex}$ values were defined using only the air flow volume and a constant inlet air temperature. This
previous calculation of $K_{ex}$ indicated that heat flow exchange is proportional to the air flow volume regardless of the phase change status.

Our new CFD-based $K_{ex}$ estimate differs from the conventional $K_{ex}$ estimate by up to 25% depending on the temperature difference ($\Delta T_{air-pcm}$ in Equation (1)), even for a given air flow volume. The new CFD-based $K_{ex}$ values are approximately 40% lower than conventional $K_{ex}$ values at an air flow volume of 1000 m$^3$/h. This could be explained by the consideration of heat transfer in the radial direction of each PCM tube in the CFD analysis. This additional calculation results in a decrease in the heat exchange performance compared to previous methods that assumed a uniform temperature.

### 3.4 A multiple regression analysis

Using the results in the previous section, we conducted a multiple regression analysis to predict variation in $K_{ex}$. We considered $V$, $\Delta T_{air-pcm}$, $T_{air}$, $T_{pcm}$, and $\Delta T_{in-out}$ as independent variables in the stepwise method. As a result, in Table 2, $V$ and $\Delta T_{air-pcm}$ were chosen as to calculate $K_{ex}$. $K_{ex}$ can thus be expressed by the following equation:

$$K_{ex} = 13.807 + 0.045 V - 2.604 \Delta T_{air-pcm} \quad (2)$$

We used this equation in the simulations described in Section 4.

### 4 Simulation methods

#### 4.1 Heat load calculation and AC equipment

Using the Building Energy Simulation Tool (BEST) Program, we calculated the seasonal variation in heat loads for a standard office space with a floor area of 300 m$^2$. The rated cooling capacity of the AC equipment was defined to be 65 kW to achieve an average load ratio of 30% against typical heat loads in July. Accordingly, we chose an indoor unit with a rated capacity of 7.3 kW and a fixed air flow volume of 1170 m$^3$/h per indoor unit. In total, eight indoor units were connected to an outdoor unit [8].

#### 4.2 Control of charging and discharging modes

We constructed a system simulation program based on the Life Cycle Energy Management (LCEM) tool, where heat exchange between the PCMs and air was calculated using the values of $K_{ex}$ determined in Section 3.4. For the charging mode, the return air from the room was cooled in the evaporator; this met the heat load for cooling both the PCMs and the room space. The cooled air was partially exchanged in the PCM duct and then supplied to the room. For the discharging mode (i.e., air circulation mode), the return air was cooled only in the PCM duct.

Once the air passed through the PCM duct and the air temperature ($T_{out}$) dropped to the lower limit ($T_{low}$, 15 °C) during charging, the mode was switched to discharging. Similarly, as $T_{out}$ increased to its upper limit ($T_{high}$, 21 °C) during discharging, the mode was switched to charging. This simple control based on $T_{out}$ is named as Case 1 as in Table 4.

For Case 2, we set an upper limit on the heat loads in the room space during discharging to prevent unstable control. If the heat load in the room space increased beyond $L_r$ (2.19 kW, at 30% of the rated capacity of the indoor unit) during discharging, the mode was switched to charging. We also applied normal cooling (without PCMs) for startup heat loads at 8:00 h in Case 2.

Cooling was operational from 8:00 to 22:00 on weekdays and did not operate during weekends and national holidays. The calculation period was May–June and September–October, with low to moderate heat loads. The physical properties of the PCM used in the calculations are shown in Table 3. A total weight of 6 kg was contained per duct per indoor unit. The PCM was assumed to charge or discharge 229 kJ/kg of latent heat for a temperature range of 16–20 °C.

### 4.3 Simulation cases

Table 4 shows the simulation cases. Case 1 and Case 2 included PCMs, with the control methods described in Section 4.2. Case 0 considered heat exchange without PCMs for comparison.
5 Results and discussion

5.1 Remaining heat loads

Figure 6 shows remaining heat loads, which were caused by a shortage of heat generated in the evaporator, against the input heat load in Case 1 and Case 2. Daily maximum heat loads were typically observed at the start of operation. Case 1 shows a maximum remaining heat load of approximately 2.6 kW at the start time, during which the cooling capacity of the compressor was insufficient to meet the heat demand in both the room space and PCMs. However, this problem was overcome in Case 2, where cold storage in PCMs was avoided for the first 5 min in the morning.

5.2 Stability of on–off control

Figure 7 shows daily variation in the air temperature ($T_{\text{out}}$) supplied to the room space in each case and the corresponding heat loads in the room space for two representative days (a) May 24th and (b) October 10th.

On May 24th, the heat load-based control was not operational, and the heat load varied below the upper limit $L_r$ (2 kW). The first charging mode started at 8:00 and continued until 10:00 in Case 1. Once $T_{\text{out}}$ reached $T_{\text{low}}$ (15 °C), $T_{\text{out}}$ increased to approximately 20 °C and the discharge mode was initiated. Case 2 showed a rapid decrease in $T_{\text{out}}$ to 12.6 °C at the start time, owing to the normal cooling operation without charging of PCMs. Subsequently, the first charging mode was operational until 11:00, similar to Case 1. Both cases showed that a discharge period followed a 2-hour charging period, during which the room space was sufficiently cooled only by discharging heat from the PCMs and requiring no operation of the compressor. In both Case 1 and Case 2, a second charging period was observed for 2.5 hours in the afternoon. Discharging followed charging again until the end of daily operation (22:00). Such long discharging periods were achieved because small heat loads in the evening required a relatively small air flow volume, leading to preservation of heat by the PCMs. Therefore, we found that for the given heat load, a stable cycle of charging and discharging for approximately 2–3 h was possible in both cases.

However, there were some differences in $T_{\text{out}}$ between Case 1 and Case 2 on October 10th, with heat loads above $L_r$. For Case 1, $T_{\text{out}}$ varied widely between 11:00 and 14:00, during which time the charging and discharging modes frequently switched. This is because large amounts of the heat load above 2.0 kW required relatively large air flow volumes in the room space, leading to rapid heat exchange between PCMs and air, and thus a rapid increase or decrease in $T_{\text{out}}$. For Case 2, $T_{\text{out}}$ remained low in the morning, even dropping below $T_{\text{low}}$ (15 °C), because additional controls that prevented the switch to discharging mode were effective for a heat load >2 kW. This led to increased cold energy storage and extended the discharging time, which allowed for a more stable cycle in Case 2 than in Case 1. We observed discharge for 30 minutes from 12:00 to 12:30, and one hour from 13:30 to 14:30. These periods were shorter for smaller heat loads, i.e., in May. Additional work is required to achieve long-term discharge with a low $T_{\text{out}}$ by improving the control and heat exchange performance of the PCM duct.

6 Comparing energy consumption

Figure 8 shows the relationship between the hourly Coefficient of Performance (COP) and the load ratio of the indoor AC units in Case 0 (without PCMs) and Case 2 (with PCMs). For Case 0, the load road was typically <20% while for Case 2, the load ratio was typically >20%. The increase in load ratio, which was due to the additional charging load of PCMs, generally resulted in improvement of the COP in Case 2. However, frequent switching between charging and discharging led to slight decreases in the COP at load ratios of approximately 30%.

![Figure 6. Sensible heat load and unprocessed load in the mid load periods](image1)

![Figure 7. The supply air temperature and sensible heat load in Case 1 and Case 2](image2)

![Figure 8. Comparison of COP and load ratio](image3)
Figure 9 shows monthly electricity consumption (May–October) of air conditioning for Case 0 and Case 2. Compared with Case 0, Case 2 indicated energy reductions of 22.7%, 7.0%, and 8.2% in May, June, and October, respectively. For Case 2 compared with Case 0, there was a slight increase in energy consumption in September, which had relatively large heat loads compared to October. The seasonal energy reduction between Case 0 and Case 2 was 7.6%, excluding July and August, and 4.5%, including July and August. Although the new system we proposed here achieved energy reduction via cold storage for PCMs at small heat loads, our future work will consider other solutions to save energy for large heat loads.

7 Summary and conclusion

In this study, we propose a short-term thermal energy storage system by adding phase-change materials (PCMs) to air ducts that are connected to an indoor air conditioning unit in the ceiling space. We examined the energy performance of the system by determining the heat exchange coefficient, \( K_{ex} \), through numerical analysis.

We determined a stable cycle of charging and discharging every few hours for small heat loads. We achieved an increase in COP and an energy reduction of 7.6% compared with conventional systems (i.e., without PCMs). However, cold storage for PCMs did not lead to an increase in energy efficiency in September, when there were relatively large heat loads. Therefore, more research is required to improve the control or operation of our proposed system in order to reduce energy for large heat loads, including in July and August. We also experimentally validate the heat exchange coefficient between the PCMs and air and examined the practical feasibility of the system in terms of heat exchange performance and pressure loss.

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