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# Annual operation of PCM storage system for offices

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## Abstract

*This article presents operation of latent heat storage unit with phase change materials (PCMs). Such kind of system reduces energy consumption in buildings since it stores cold during summer nights and delivers it during the day and thus reduces cooling load. In winter system is connected with solar air collector and heat is stored for heating during morning and evening hours. Proposed is a stand-alone unit suitable for offices, which consists of plates filled with paraffin RT22HC. The objective of the paper is to examine the functioning of the suggested TES system on an annual basis, to explore the feasibility of using it for both, cooling and heating and to carry out parametrical analysis in order to find the best set of parameters for given conditions. Heat transfer from and into PCM is simulated with Ansys Fluent code for 2D geometry, temperature of warm air at the outlet from solar air collector is calculated in Trnsys, TRY for Ljubljana (Slovenia) is taken as a boundary condition. Results are compared to energy demands of a typical office (calculated in Trnsys) for Slovenian climate. Calculations reveal that in winter the largest savings are in March, as larger quantities of heat are available at that time. In summer, the maximum amount of accumulated cold is in July and August, because of larger temperature fluctuations between day and night.*

**Keywords:** thermal energy storage, component, modelling

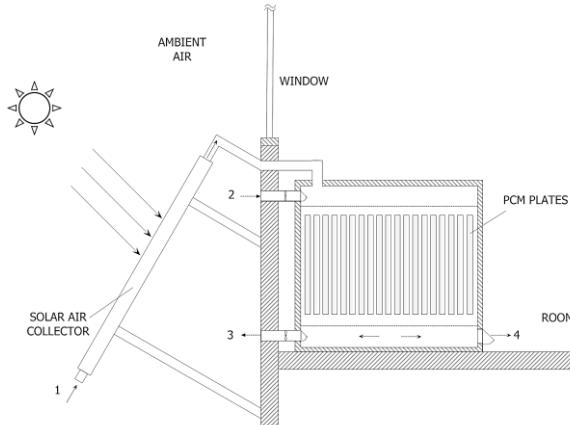
## 1. Introduction

In developed countries buildings account for a 20–40% of the total final energy consumption, in the European Union this number is at 40% [1]. Therefore one of the priorities of the EU is to minimize the energy consumed by buildings. The EPB Directive states that it is necessary to choose alternative solutions for heating and cooling and one option is thermal energy storage (TES) using phase change materials (PCMs).

Latent heat storage has been gaining importance in recent years and literature review shows a variety of approaches, which are summarized in the papers written by Farid et al. [2], Soares et al. [3], Zhang et al. [4] and Osterman et al. [5]. These storage systems can be integrated into the

building's envelope (i.e., walls, roofs, and floors), into HVAC or operate as stand-alone units.

The idea is to utilize natural sources within stand-alone active system. Outdoor cold during summer nights is stored and supplied to the indoor environment during the day when the cooling load increases. For the heating requirements, energy from the sun can be exploited in conjunction with solar air collectors as in [6]. It happens often that heating during the day is not needed, because of high energy gains. Heating load increases after sunset, so in such cases day-cycle TES can be advantageously used. When the available energy exceeds the energy demand, energy is stored and later released to completely or partially replace conventional systems.



*Figure 1: Conceptual design of a storage unit*

Even though previous studies have dealt with this topic, a key limitation was that they mainly addressed operation of TES for either heating or cooling, but none of the studies proposed a system that would unite these two options. Nonetheless, with our concept (Figure 1) it would be possible to further improve output of TES unit and utilize low exergy sources throughout the year. Moreover, research carried out so far was mainly focused on single thermal response of TES and parametrical analyses were carried out only in conjunction with it.

The main objective of this paper is to examine closely the functioning of the suggested system on an annual basis, to explore the feasibility of using it for building cooling as well as for heating and to carry out parametrical analysis which will reveal best set of parameters (such as geometry, air flow rate, melting point). The focus of recent research has been on thermal response of

TES, however, to the authors' best knowledge, very few publications [7–9] are available that discuss operation over a longer period of time. None of them addresses parametrical study on an annual basis. Deeper investigation and finding best parameters is very important because it is the only clear-cut way to realistically assess performance of TES in buildings and to make the most out of it.

## 2. Methods

### 2.1. Numerical model

In order to define thermal response of a storage unit, two important processes need to be considered, namely the heat transfer from air to paraffin, and the heat transfer within the paraffin. The problem is tackled with finite volume method, where both, flow and temperature field, are being solved.

Numerical model of TES can be reduced to a 2D problem due to negligible changes in z-direction. Moreover, if we assume that the distribution of flow between plates is uniform, only one plate and one air channel can be considered. Going further, plate and air channel are symmetric, so only half of each of them needs to be included in the domain. Dimensions and the schematic diagram of the 2D computational domain used in this investigation are depicted in Figure 2.

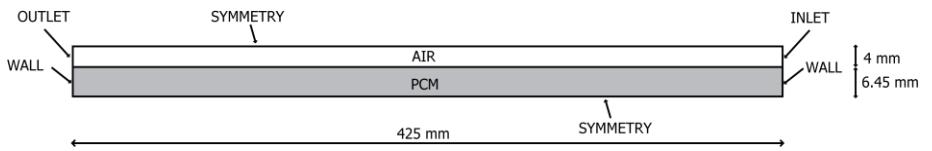


Figure 2: Schematics of a numerical domain

In order to mathematically describe the process inside the storage unit, additional assumptions have to be met for the air as well as for the PCM and they can be found in Osterman et al. [6].

The steady equations governing the flow are solved only for the air and two unsteady energy equations are applied to the entire computational domain (PCM and air). The two-dimensional equations for conservation of mass, momentum and energy for air are expressed in Eqs. 1-3.

$$\frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} = 0 \quad (1)$$

$$\begin{aligned} \rho \left( v_x \frac{\partial v_x}{\partial x} + v_y \frac{\partial v_x}{\partial y} \right) &= -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 v_x}{\partial x^2} + \frac{\partial^2 v_x}{\partial y^2} \right) \\ \rho \left( v_x \frac{\partial v_y}{\partial x} + v_y \frac{\partial v_y}{\partial y} \right) &= -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v_y}{\partial x^2} + \frac{\partial^2 v_y}{\partial y^2} \right) \end{aligned} \quad (2)$$

$$\rho c_p \left( \frac{\partial T_a}{\partial t} + \bar{v} \cdot \nabla T_a \right) = k_a \nabla^2 T_a + S \quad (3)$$

As there is no convection in PCM, only unsteady energy equation is being solved (Eq. 4). Phase change is taken into account via apparent specific heat (presented in Figure 3 with solid curve).

$$\rho_{PCM} c_{p,PCM,app} \left( \frac{\partial T_{PCM}}{\partial t} + \bar{v} \cdot \nabla T_{PCM} \right) = k_{PCM} \nabla^2 T_{PCM} \quad (4)$$

With these equations air temperature at the outlet can be determined and thus thermal response of TES. Of the main interest is thermal power and amount of stored heat/cold, respectively. Former is calculated by Eq. 5. More on numerical modelling can be found in [6].

$$P = \frac{dQ}{dt} = \dot{m}_a c_{p,a} (T_{a,out} - T_{a,in}) \quad (5)$$

## 2.2. Experimental work

In order to verify the validity of the numerical model, an experimental rig for testing storage unit's thermal response was set up. Casing was made of 0.8 cm thick PMMA and the external dimensions were 77 cm x 67 cm x 42 cm. Storage unit was insulated with 8 cm thick EPS. The unit contained 15 or 30 CSM plates (depending on the case) filled with paraffin RT22HC [10]. Air gap between plates was 0.8 cm.

Outer dimensions of plates were 30 cm x 45 cm x 1.5 cm and they were vertically positioned in the storage tank. The actual volume of PCM is smaller, because plates have edges and the surface is not flat. So the dimensions that are also used in the numerical model are: 27.0 cm x 42.5 cm x 1.29 cm.

Analysis of: PCM's specific heat dependence on the temperature (DSC step and T-history measurements, Figure 3), thermal conductivity and density was performed in order to gain data needed for the numerical model. Detailed description and results of experimental work can be found in Osterman et al. [11].

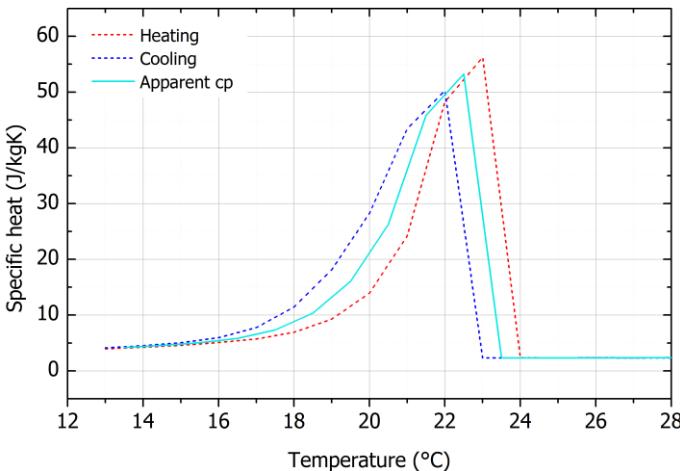


Figure 3: Measured and apparent specific heat of RT22HC

### 2.3. Principle of annual operation

As already mentioned, TES is intended for heating and cooling, therefore two modes of operation need to be distinguished, ie. in summer and winter. Ljubljana, Slovenia is taken as a reference city (but it could be done for any other location as well) for which it is assumed that the heating season lasts from October to April and in the rest of the year, office needs to be cooled. Principle of operation is presented in Osterman et al. [6].

We are interested in operation throughout the year, but from the perspective of simulations this would be time consuming. Therefore the following will be assumed: the first week of each month is representative for the whole month. Data for ambient temperature are taken from the test reference year

for Ljubljana and combined into temperature profile, which consists of 12 weeks.

Procedures were implemented into Fluent through script written in Scheme, which actually commanded Fluent. First, velocity field for minimum air flow was calculated and written to a file then velocity field for maximum air flow was calculated and written to another file. Both of these were steady state calculations and they were succeeded by transient simulation, which embodied only energy equation. Data regarding flow field were read from the previously created two files according to the conditions presented previously. Used time step in yearly calculations was 300 s.

#### 2.4. Parameters for annual operation

One of the objectives set in the introduction was to find such set of parameters in which the energy savings are the greatest. Influential parameters are air flow rate, melting temperature of PCM ( $T_{m,shift}$  means shift of  $c_p(T)$  curve left or right in Figure 3) and geometrical properties. Air flow rate refers to the maximal, i.e. the one that is set during charging, minimum airflow is in all cases equal to 40 m<sup>3</sup>/h. Length and width of plates were fixed (0.425 m and 0.275 m) and thus length of TES. Third dimension varied between 5.0 mm and 15.0 mm, whereas air gap varied between 2.5 mm and 10.0 mm. Overall width of TES was constrained to 0.6 m, therefore we could calculate maximal number of plates that would fit inside TES. In Table 1 are given dimensions of selected geometries, number of plates and overall width of TES.

Chosen parameters:

- Shift of melting temperature:  $T_{m,shift} = -4$  K, -2 K, 0 K and 2 K,
- Volume flow rate:  $q_{v,max} = 75$  m<sup>3</sup>/h, 100 m<sup>3</sup>/h, 125 m<sup>3</sup>/h and 150 m<sup>3</sup>/h,
- Geometry.

*Table 1: Properties of selected geometries*

| Geometry | Thickness– plate<br>(mm) | Air gap<br>(mm) | Number of plates | Width of TES<br>(cm) |
|----------|--------------------------|-----------------|------------------|----------------------|
| 50_25    | 5                        | 2.5             | 80               | 60                   |
| 75_25    | 7.5                      | 2.5             | 60               | 60                   |
| 100_25   | 10                       | 2.5             | 48               | 60                   |
| 125_25   | 12.5                     | 2.5             | 40               | 60                   |
| 150_25   | 15                       | 2.5             | 34               | 59.5                 |
| 50_50    | 5                        | 5               | 60               | 60                   |
| 75_50    | 7.5                      | 5               | 48               | 60                   |

|         |      |     |    |      |
|---------|------|-----|----|------|
| 100_50  | 10   | 5   | 40 | 60   |
| 125_50  | 12.5 | 5   | 34 | 59.5 |
| 75_75   | 7.5  | 7.5 | 40 | 60   |
| 100_75  | 10   | 7.5 | 34 | 59.5 |
| 125_75  | 12.5 | 7.5 | 30 | 60   |
| 75_100  | 7.5  | 10  | 34 | 59.5 |
| 100_100 | 10   | 10  | 30 | 60   |
| 125_100 | 12.5 | 10  | 27 | 60.8 |

Criterion for the best set of parameters was the total quantity of stored heat in a given period of time. We assumed that ambient air needs to be heated up or cooled down to the desired room temperature. If this air is at least partly heated or cooled in the TES, then this part, presumably, represents energy savings  $E_{accu}$  due to TES as defined in Eq. 6. Energy use for fan operation also needs to be considered, which is described by Eq. 7. If the actual energy savings  $E_{saving}$  have to be determined, then energy consumption needs to be subtracted from accumulated energy as in Eq. 8. It is necessary to point out, that these two (heat, electricity) are, from exergy point of view, two different quantities despite the same units. But for the purposes of estimation of yearly savings, this premise will be good enough.

$$E_{accu} = \int q_m c_{p,a} (T_{a,out} - T_{a,in}) dt \quad (6)$$

$$E_{consum} = \int \frac{q_v \Delta p}{\eta} dt \quad (7)$$

$$E_{saving} = E_{accu} - E_{consum} \quad (8)$$

### 3. Results

#### 3.1. Annual parametrical analysis

This subsection presents the results of parametrical analysis on an annual basis. Examples of temperatures in summer and winter as well as details on validation part can be found in Osterman et al. [6].

##### *Influence of melting temperature*

Figure 4 shows temperature in August with melting point lowered for 4 K (left) and increased for 2 K (right), respectively. It can be seen that in the case of lower melting point PCM doesn't reach latent heat region because of high ambient temperatures. With higher melting point it can be clearly seen

that air outlet temperature remains below 22 °C and TES successfully damps temperature fluctuations.

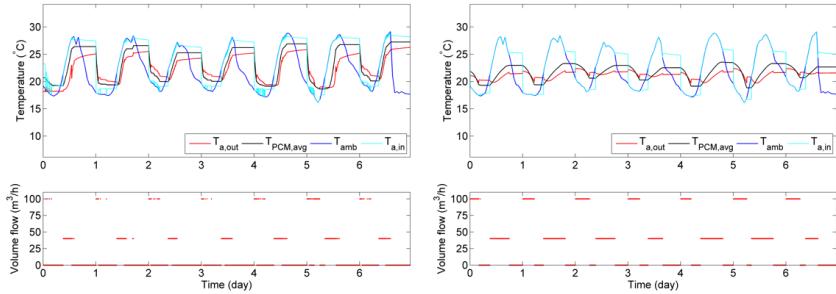


Figure 3: Temperatures in August,  $T_{m,shift}=-4 K$  (left),  $T_{m,shift}=2 K$  (right)

### Influence of volume flow

Comparison of different air flow rates ( $75 m^3/h$  and  $150 m^3/h$ ,  $T_{m,shift} = 0 K$ , Figure 4) reveals that differences in the amount of accumulated cold in May, June and September are negligible, because of low ambient temperature. In that case increased air flow does not affect the heat transfer. Major differences occur in July and August, because of higher temperatures and only increased flow ensures that TES gets charged sufficiently.

Comparison of electricity consumption for both flows indicates that the use of electricity to power the fan increases significantly with higher air flow. Use of electricity represents more than 30% in the case of  $75 m^3/h$ , while in the case of  $150 m^3/h$ , it represents more than 60%. This indicates inefficiency of larger flow rates, therefore it is expected that the most suitable flow rate will be in the range between  $75 m^3/h$  and  $100 m^3/h$ .

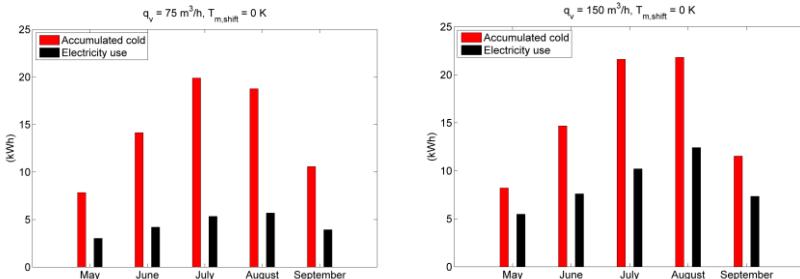


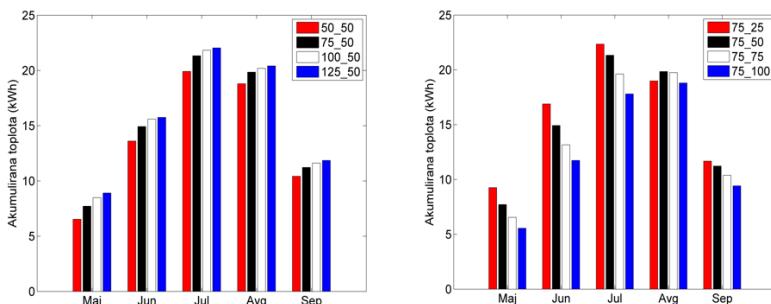
Figure 4: Monthly accumulated cold and electricity use for  $75 m^3/h$  (left) and  $150 m^3/h$  (right)

### **Influence of plate's thickness**

Influence of plate's thickness (5 mm, 7.5 mm, 10 mm, 12.5 mm) was carried out for both summer and winter and for different flow rates and melting temperatures, but here we present summer case with 100 m<sup>3</sup>/h, no shift in melting temperature and for air gap 5 mm. From the Figure 5 it can be seen that the most accumulated heat is in the case of the thickest plate (12.5 mm), because of the largest amount of PCM in the selected TES volume.

### **Influence of air gap**

A similar comparison can be carried out for different air gaps (2.5 mm, 5 mm, 7.5 mm and 10 mm), assuming the same thickness of the plates (7.5 mm). The largest amount of heat is accumulated in the case of the smallest air gap (2.5 mm), because of the largest amount of PCM in the selected TES volume (Figure 5). From these findings it can be predicted that the best geometry will be the one with a greater thickness of plate and a smaller air gap.



*Figure 5: Accumulated heat for different thicknesses of plates (left) and for different air gaps (right).*

### **Best set of parameters**

To confirm predictions, simulations for all geometries, air flows and melting temperatures were carried out. It turns out that the best set of parameters for Ljubljana's weather conditions is: geometry 125\_25 (plate's thickness 12.5 mm, air gap 2.5 mm), volume flow 100 m<sup>3</sup>/h, and shift in melting temperature -2 K. Annual savings with such a configuration equal to 189 kWh. Similarly holds true for other geometries with the same air gap, flow rate and melting temperature. In real application one would probably choose geometry 150\_25, as it means fewer plates and consequently lower price.

## 4. Conclusions

Numerical investigations of a latent heat storage unit for space heating and cooling were carried out. It consisted of 30 plates filled with paraffin RT22HC. The developed model was used for evaluation of TES on a yearly basis. From the parametrical analysis it was found out, that it is more appropriate to choose lower melting temperature in colder summer months, whereas in warmer months higher melting temperature is better. Similar observations hold true for winter. Calculations reveal that larger flow rates are inefficient therefore the most suitable flow rates are between 75 and 100 m<sup>3</sup>/h. Analysis of influence of plate's thickness and air gap shows that the best option is thick plate (in our case 12.5 mm) and small air gap (2.5 mm), because of the largest amount of PCM in the given TES volume. From the calculations on an annual basis it turns out that the best set of parameters for Ljubljana's weather conditions is geometry 125\_25 (plate's thickness 12.5 mm, air gap 2.5 mm), volume flow 100 m<sup>3</sup>/h, and shift in melting temperature -2 K. Annual savings with such a configuration equal to 189 kWh.

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