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EXPERIMENTAL AND NUMERICAL INVESTIGATION OF AN AIR-PCM HEAT-STORAGE UNIT

EKSPERIMENTALNA IN NUMERIČNA PREISKAVA ENOTE ZRAK – PCM ZA SHRANJEVANJE TOPLOTE

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The phase-change materials (PCMs) are quite promising heat-storage media for applications that require a high heat-storage capacity in a relatively narrow temperature interval. One of the areas of the latent-heat-storage application is thermal-energy storage in solar air systems, where a lower heat-storage temperature leads to an increase in the overall efficiency of the system. The paper deals with numerical and experimental investigations of the thermal performance of an air-PCM heat-storage unit. The studied unit contained 100 aluminum containers filled with a paraffin-based PCM. Both experimental and numerical investigations were done for a constant air temperature at the inlet of the heat-storage unit.

Keywords: heat exchanger, PCM, simulation

Materiali s fazno premeno (PCM) so obetavni mediji za shranjevanje toplote za uporabo, ki zahteva veliko kapaciteto shranjevanja toplote v relativno ozkem temperaturnem intervalu. Eno od področij uporabe shranjevanja latentne toplote je shranjevanje toplotne energije v solarnih sistemih, kjer nizka temperatura shranjevanja toplote povečuje splošno učinkovitost sistema. Članek obravnava numerične in eksperimentalne preiskave toplotne zmogljivosti enote zrak – PCM za shranjevanje toplote. Preiskovana enota je sestavljena iz 100 aluminijevih vsebnikov, napolnjenih s parafinskim PCM. Eksperimentalne in numerične preiskave so bile izvršene za konstantno temperaturo zraka, pri vhodu v napravo za shranjevanje toplote.

Ključne besede: izmenjevalnik toplote, PCM, simulacija

1 INTRODUCTION

Most of the solar thermal systems cannot effectively operate without thermal storage.[1] Water is generally used as a heat-storage medium in water-based solar systems, but it is less practical for air-based systems. The rock beds, where solid materials (usually pebbles) are used for heat storage,[2] can be used in air-based solar thermal systems, but they have certain disadvantages. The rock beds use a lot of space, they are difficult to clean, the air-flow distribution in the beds is usually non-uniform causing highly non-uniform temperature distribution in the heat-storage medium and thus decreasing the energy efficiency of the system. In order to achieve a higher heat-storage density, the rock beds need to be heated up to high temperatures leading to a decrease in the overall energy efficiency. Promising media for thermal storage in these applications are phase-change materials.[3]

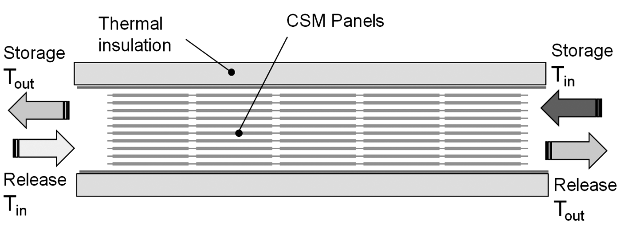

Figure 1: Schematic view of the heat-storage unit
Slika 1: Shematski prikaz enote za shranjevanje toplote

The phase change of a material provides a rather high thermal-storage capacity (and also energy-storage density) in a narrow temperature interval around the melting point of the material.[4] Latent heat is absorbed or released when the material changes phase from solid to liquid or from liquid to solid. The most commonly used PCMs are paraffins, fatty acids and esters, and various salt hydrates.[5]

2 HEAT-STORAGE UNIT

PCM-based heat-storage units can be of various designs. The studied heat-storage unit had a rather simple design (**Figure 1**). It was a thermally insulated box that contained 100 aluminum panels filled with the PCM (arranged in 5 rows). The Rubitherm CSM panels were used in the unit. The panels have the dimensions of 450 mm × 300 mm × 10 mm and each of them can accommodate approximately 700 ml of PCM. The panels were filled with the Rubitherm RT42 paraffin-based PCM. The RT42 has a melting range from 38 °C to 43 °C, heat-storage capacity of 174 kJ/kg (in the temperature range between 35 °C and 50 °C) and a thermal conductivity of 0.2 W/(m K). The overall heat-storage capacity of the unit was 12.3 kJ (3.4 kW h) in the temperature interval between 25 °C and 55 °C.

In the situations when the unit is not fully charged it can make sense to reverse the air-flow direction in the discharging mode (in comparison to the charging mode – as indicated in **Figure 1**) in order to increase the outlet air temperature and, thus, to increase the efficiency. Another issue is the position of the PCM panels in the unit. When the panels are positioned horizontally the PCM in the fully melted state collects at the lower part of the panel and there will be an air gap between the PCM and the upper surface of the container. That gap can significantly influence the heat transfer between the PCM and the air passing through the heat-storage unit. The volume change between the solid and liquid state is rather significant for many PCMs (it can be larger than 15 %) and thus some empty space needs to be kept in the containers to allow for that volume change. An experimental set-up was prepared in order to investigate the thermal performance of the unit. The experimental set-up consisted of the unit, a fan, an electric air heater, and the temperature and air-velocity probes connected to a data logger (**Figure 2**).

The experiments were carried out at a constant air-flow rate throughout the unit and a constant inlet air temperature.

3 NUMERICAL MODEL OF THE UNIT

The simulation tool TRNSYS 17 was used for numerical investigations. TRNSYS 17 is a 1D simulation tool that can be used for energy-performance simulations of systems and buildings. A schematic of the numerical model of the heat-storage unit is shown in **Figure 3**. Simulations were done for 5 rows of the CSM panels with 20 panels in each row as was the case in the experimental investigations. Since there are 19 geometrically equal air channels between the CSM panels, only one channel was modeled. Actually, assuming the planar symmetry, only a half of a channel with a half of the CSM panel thickness was modeled. The numerical model of the heat transfer including a phase change in the PCM unit was implemented in MATLAB and connected to TRNSYS.[6]

The numerical model for the heat transfer in the PCM created in MATLAB is based on the implementation of the 1D heat-transfer equation that includes the source of the latent heat of the phase change:[6]

$$\frac{\partial}{\partial t}(\rho c T) = k\frac{\partial^2 T}{\partial x^2} + \dot{Q} \tag{1}$$

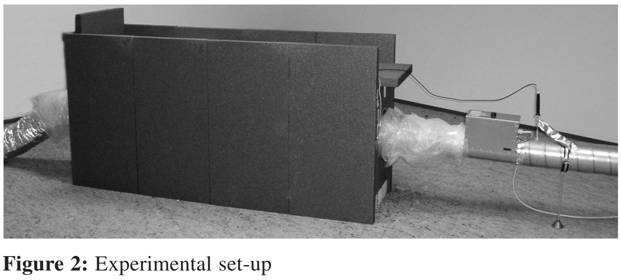

Figure 2: Experimental set-up
Slika 2: Eksperimentalni sestav

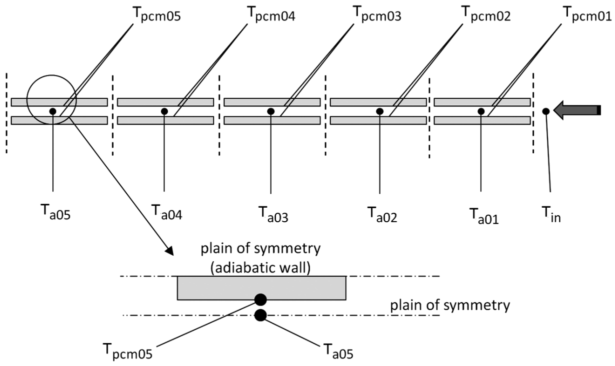

Figure 3: Simplification of the storage unit for the numerical model
Slika 3: Poenostavitev naprave za shranjevanje za numerični model

where ρ represents the density, c denotes the heat capacity, k stands for the thermal conductivity, t is the time, T represents the temperature and x is the spatial coordinate. The term $\dot{Q}$ in equation (1) can be expressed as follows:

$$\dot{Q} = \rho \Delta H \frac{\partial f_s}{\partial t} \tag{2}$$

where ΔH denotes the latent heat and f_s is the solid fraction that represents the ratio between the solid and liquid phases. If $f_s = 0$, the material is in the liquid state and, therefore, only thermo-physical properties related to the liquid state are considered. Conversely, if $1 f_s = 0$, the material is in the solid state. The theoretical analysis of solidification is based on the equilibrium with the assumption that a complete diffusion occurs between the solid and liquid phases. A simple premise is to assume that the latent heat increases linearly with the temperature:

$$f_s = \frac{T_L - T}{T_L - T_S} \tag{3}$$

In equation (3) T_S and T_L represent the solidus and liquidus temperatures, respectively. The solution of equation (1) strictly depends on the initial and boundary conditions. The initial condition describes the temperature distribution for $t = 0$:

$$T(x, t=0) = T_0(x) \tag{4}$$

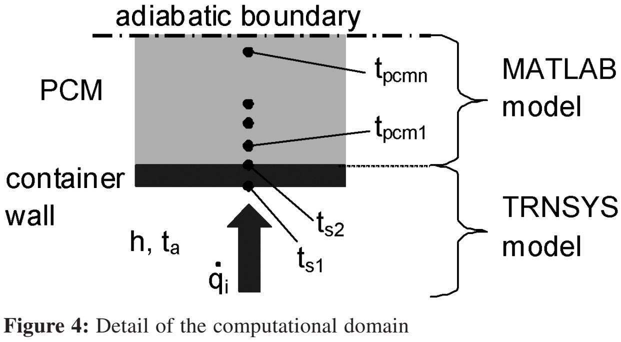

Figure 4: Detail of the computational domain
Slika 4: Detajl računske domene

The numerical model takes into account the Neumann type of boundary conditions determined by the heat fluxes $\dot{q}$ on the surface:

$$-\lambda \frac{\partial T}{\partial x}(x=0, f)=\dot{q} \tag{5}$$

The adiabatic boundary condition ($\dot{q}=0$) is used at the plain of symmetry in the middle of the container.

A detail of the computational domain is shown in **Figure 4**. The communication between the TRNSYS model and the MATLAB model is described below.

The computations started from a given initial temperature profile in the PCM layer (temperatures t_{pcm1} to t_{pcmn}). A constant temperature across the layer was assumed in the simulations.

The heat flux $\dot{q}_i$ at the surface of the aluminum wall of the container was obtained from the TRNSYS model. No thermal resistance between the PCM and the container wall was assumed and the wall temperature t_{s2} was considered to be equal to the temperature t_{pcm1}. The heat flux obtained from the TRNSYS model was then used as an input for the MATLAB model for the PCM layer. The MATLAB model provided a new value of the t_{pcm1} that was used for calculating the heat flux in the next time step in the TRNSYS model. A time step of 60 s was used in the TRNSYS model while the MATLAB model used a much shorter time step of 1 s in line with the stability condition. The stability condition for explicit formula was used according to[7].

4 RESULTS AND DISCUSSION

Figure 5 shows the comparison of experimental and numerical results for the heat-storage rate in the unit. The heat-storage rate was obtained from the air-mass-flow rate and the air temperatures at the inlet and the outlet of the storage unit. The air temperature at the inlet of the unit in the heat-storage period was 58 °C and the air-flow rate was 230 m^3/h. Though a constant air temperature at the inlet of the unit is of rare occurrence under real operating conditions, it is very illustrative for the theoretical analyses. There are certain discrepancies between the predicted and measured heat-storage rates. The electric air heater used in the experiments had an output of 2 kW, but since the heater needed some time to reach that output the measured heat-storage peak is not as sharp as in the case of numerical simulations. The current version of the simulation model neglects the heat loss to the surroundings. This is one of the reasons for the increasing discrepancies over longer periods of time. At a certain point the air-temperature difference between the inlet and the outlet of the unit is not due to the storage rate but due to the thermal loss of the unit. Another reason for discrepancies is associated with the air flow inside the unit.

Though the heat transfer in the case of the air flow between two parallel planes is well described in the literature, the uncertainty remains about the air-flow rates in particular air channels. The melting of the PCM is yet another source of uncertainties. A proper simulation of the melting and solidification requires a complex 3D numerical model that takes into account the convection in the liquid PCM as well as the PCM volume change (to address the void formation during solidification).

The heat-release rates can be seen in **Figure 6**. The air temperature at the inlet of the unit was 25 °C during the heat-discharge period. The air-flow rate was the same as in the heat-storage period. Again, there is a relatively good match between the numerical and experimental results at the beginning of the heat-release period, but the discrepancies increase with the elapsed time. The heat-release rate peaks at more than 2 kW at the beginning of the heat-release period, but it very quickly drops to around 1 kW when the sensible heat above the melting range is released. The differences between the numerical and experimental results can be explained with the non-uniform air-flow rates in the air channels between the panels. The numerical model assumes that the air-flow rates in all the air channels are the same and, therefore, the heat fluxes stored to, or released from, all the panels in one row are the same.

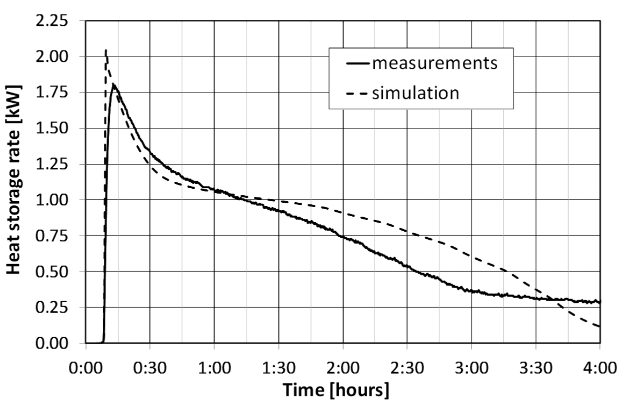

Figure 5: Heat-storage rates
Slika 5: Hitrost shranjevanja toplote

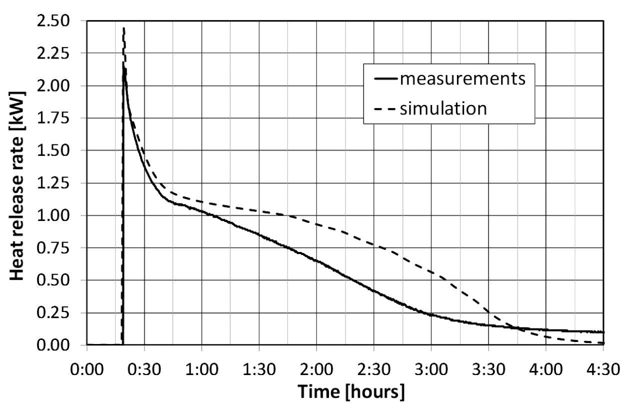

Figure 6: Heat-release rates
Slika 6: Hitrost sproščanja toplote

5 CONCLUSION

A 1D simulation model of a heat-storage unit with a PCM for thermal-energy systems using air as a heat carrier was developed. The model of the unit was created with the use of coupling between the TRNSYS 17 simulation tool and the in-house MATLAB model for PCMs. An experimental setup with a test heat-storage unit was put together in order to validate the developed numerical model experimentally. The comparison of experimental and numerical results revealed certain discrepancies in the predicted and measured temperatures and energies. The simulation model neglected the heat loss to the ambient environment, the uncertainties associated with the air-flow inside the unit and it used a rather simple approach to the model phase change of the PCM. The necessary communication between TRNSYS and MATLAB also significantly slowed down the simulations. To address all these issues a more advanced model of a heat-storage unit implemented as a TRNSYS type (programmed in C++) is under development.

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