THEORETICAL ANALYSIS OF LATENT THERMAL ENERGY STORAGE SYSTEM WITH ADSORPTION CHILLER

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ABSTRACT

The paper deals with a theoretical analysis of using thermal energy storage filled with Phase Change Material (PCM) to optimise adsorption chiller application. The lumped model equations were developed to analyse heat transfer dynamics inside cooling installation. Model equations result from energy balances of the chiller, PCM thermal storage and heat load. Influence of heat transfer fluid flowrate control, heat capacity of the system components as well as heat losses to the ambient were taken into account. This study shows the great potential of PCM storage unit to optimize cooling systems.

INTRODUCTION

The thermal driven absorption, adsorption or ejection refrigeration systems are known and used for many years. Nowadays, this type of cycle became more and more popular especially in trigeneration and in applications utilizing waste heat. This study is focused on real-industry situation of a mismatch between heating system and adsorption chiller in an office building (Fig. 1). Namely, available hot water mass flow rate is too small to properly supply the adsorption chiller in on-peak hours. Therefore thermal energy storage system with PCM was suggested to reduce peak cooling load demand.

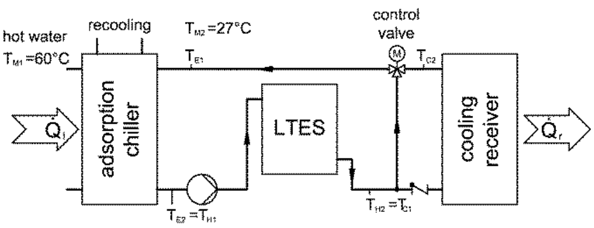

Fig. 1. Schematic of the process

Nowadays phase change materials (PCM) are used in low and high temperature applications (Karwacki et al. 2016, Mizera et al. 2015, Mizera et al. 2016). PCM heat storage materials utilize the process of phase change between solid and liquid to store thermal energy. The heat transfer process during melting and solidification proceeds at nearly constant temperature. In a narrow range around phase change temperature, the PCM stores and releases large quantities of thermal energy. In the cooling applications, water is the most popular PCM. Ice banks with a chiller are used in cooling and air conditioning systems to reduce on-peak hours thermal energy load. They work during the night, when the energy costs are low and use accumulated cooling energy during the day. Ice bank systems allow to use cooling aggregate with much smaller cooling power than the peak load of cooling energy appearing during the process.

Phase change material

The thermal driven absorption chiller works within different range of HTF temperature than the ice bank cooling system. Because chilled water temperature is 8–21 °C, the PCM nominal temperature range of phase transition has to be the same. A building air conditioning system requires chilled water temperature below 16 °C, therefore commercial PCM RUBITHERM RT15 was chosen. The partial enthalpy distribution for this material is presented on Fig. 2.

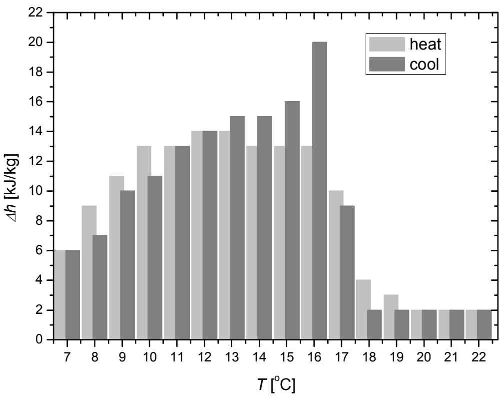

Fig. 2. Phase change material RUBITHERM RT15 partial enthalpy distribution (manufacturer data)

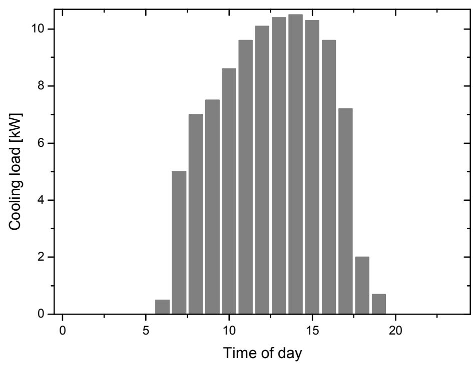

Fig. 3. Hourly profile of office building thermal energy consumption (data from personal correspondence)

Adsorption chiller and office cooling load

Total system load is presented in Fig. 3. Adsorption chiller chosen to use in the building air conditioning system has refrigeration power up to 16 kW for hot water temperature 95 °C. However available temperature of supply water is up to 62 °C. In addition, mass flow rate of water is limited to allow the application of more efficient adsorption chiller. In this way, the chilling system refrigeration power is reduced, Fig. 4. Preliminary analysis shows that on-peak hours building energy consumption will be higher than the chilling power. More accurate calculation will be done basing on solution of lumped model equations.

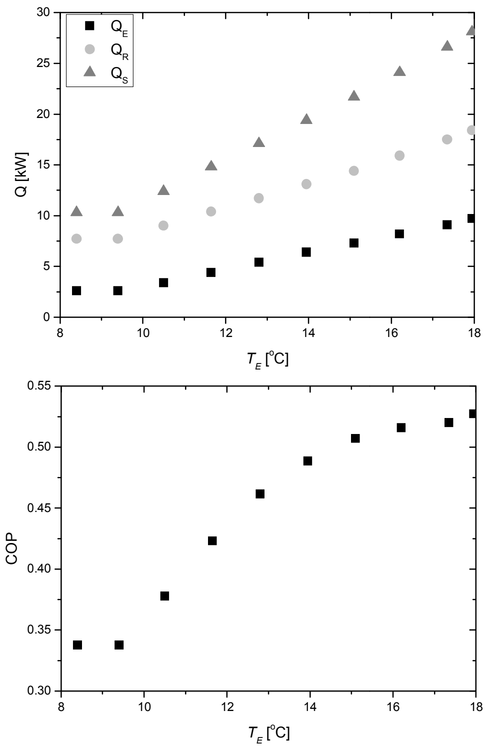

Fig. 4. Adsorption chiller load $\dot{Q}$ and COP for hot water temperature 62 °C and recooling water temperature 27 °C (manufacturer data)

COMPUTATIONAL MODEL

The generic view of configuration that was studied is shown in Fig. 1. It was intended to cool down an office building by an adsorption chiller through energy storage devices. The storage geometry used in calculations is similar to a shell and tube heat exchanger with PCM outside the tubes. The total volume of PCM is calculated from a substitute thickness of an adjacent layer assigned to each tube. The schematic view of a selected storage geometry is presented in Fig. 5. In a real application commercial capillary tube mats will be applied. The main parameters of the capillary mats are summarised in Table 1.

Thermal dynamics of the cooling system consisting of the adsorption chiller, cold receiver (office building), thermal energy storage system and control valve is analysed with the heat balance equations of 0D mathematical model. In the model liquid loop, the heat capacity of all these units was taken into account.

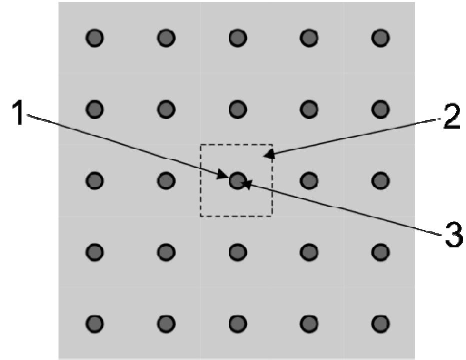

Fig. 5. Schematic view of a cross section through tube bundle and PCM: 1 – tube, 2 – PCM, 3 – HTF (dashed line indicates substitute thickness of PCM assigned to each tube)

Table 1. Capillary mat characteristics

Material	Polipropylene
Capillary tube outer diameter	3.35 mm
Capillary tube wall thickness	0.5 mm
Capillary tubes spacing	10 mm
Capillary tube length	6000 mm
Spacing between tubes rows	10 mm
Thermal conductivity	0.24 W/m·K
Specific heat	2000 J/kg·K
Substitute PCM layer thickness	3.325 mm

The proposed numerical model of the heat storage describes heat transfer and storage between the HTF, and the PCM and the construction material of the storage tubes as well as the heat transfer to or from the surroundings. The equations of the numerical model are based on a balance of the heat accumulated in the material of a mass M and specific heat c, and the difference between the supplied heat flux $\dot{Q}_d$ and the dissipated heat flux $\dot{Q}_t$. This balance can be written in the following general form:

$$\dot{Q} = Mc\frac{dT}{d\tau} = \sum \dot{Q}_d(\tau) - \sum \dot{Q}_t(\tau) \qquad (1)$$

where τ denotes time. Basing on (1), balance equations for each material accumulating heat in the storage can be written, namely:

HTF:

$$M_H c_H \frac{dT_H}{d\tau} = \dot{m}_H c_H \left[T_{H1}(\tau) - T_{H2}(\tau)\right] + \frac{T_W(\tau) - T_H(\tau)}{R_{HW}} \qquad (2)$$

PCM:

$$M_P c_P(T_F)\frac{dT_P}{d\tau} = \frac{T_W(\tau) - T_P(\tau)}{R_{PW}} + \dot{Q}_a \qquad (3)$$

capillary wall (heat transfer area):

$$M_W c_W \frac{dT_W}{d\tau} = \frac{T_P(\tau) - T_W(\tau)}{R_{PW}} + \frac{T_H(\tau) - T_W(\tau)}{R_{HW}} \qquad (4)$$

where: $T_H = (T_{H1} + T_{H2})/2$, $\dot{Q}_a$ – transfer of thermal energy to/from the ambient.

In the above equations, heat transfer from the liquid zones and surroundings was treated as purely convective. In the remaining zones only conductive heat transfer was assumed. Thermal resistance between the PCM and tube wall and between the tube wall and HTF can be written as, respectively:

$$R_{PW} = \frac{\delta_P}{\lambda_P A_P} + \frac{\delta_W}{2\lambda_W A_{FW}}, \qquad (5)$$

$$R_{HW} = \frac{1}{\alpha_H A_H} + \frac{\delta_W}{2\lambda_W A_{HW}}. \qquad (6)$$

For the analysed geometry, the length of tubes is much more larger than their inner diameter. Thus, it was assumed after [Baehr & Stephan 2006] that Nusselt number value in thermal, fully developed laminar flow can be taken as

$$Nu_{lm} = 4.364 \qquad (7)$$

The convective heat transfer coefficient for HTF is given by the relation

$$\alpha_H = \frac{Nu\lambda_H}{d} \qquad (8)$$

Time evolution of the liquid temperature T_C in the receiver is modelled with heat balance equation in the form

$$M_C c_H \frac{dT_C}{d\tau} = \dot{m}_C c_H \left[T_{C1}(\tau) - T_{C2}(\tau)\right] + \dot{Q}_l(\tau) \qquad (9)$$

where: $T_C = (T_{C1} + T_{C2})/2$, $\dot{Q}_l$ – receiver thermal energy load (Fig. 3).

Time evolution of the liquid temperature T_E in the adsorption chiller evaporator is modelled with heat balance equation in the form

$$M_E c_H \frac{dT_E}{d\tau} = \dot{m}_H c_H \left[T_{E1}(\tau) - T_{E2}(\tau)\right] - \dot{Q}_E(T_{M1}, T_{M2}, T_E) \qquad (10)$$

where: $T_E = (T_{C2} + T_{H1})/2$, $\dot{Q}_E(T_{M1},T_{M2},T_E)$ – adsorption chiller load (Fig. 4).

Adsorption chiller load depends on hot water inlet temperature T_{M1}, recooling water inlet temperature T_{M2} and HTF temperature T_E.

Leaving the PCM storage, the HTF stream is divided and part of it is directed to the receiver at a flow rate $\dot{m}_C$ while the rest is by-passed. The two streams mix in a three-way valve where their temperatures meet the energy balance

$$\dot{m}_H c_H (T_{H2} - T_{E1}) = \dot{m}_C c_H (T_{H2} - T_{C2}) \qquad (11)$$

The flow rate division ratio $\dot{m}_C/\dot{m}_H$ is governed by proportional controller expressed in the model by equation

$$\frac{\dot{m}_C}{\dot{m}_H} = K(T_C - T_{set}) \qquad (12)$$

where K is the controller gain and T_{set} is a set-point temperature.

The differential equations (2), (3), (4), (9) and (10) of the considered model can be easily transformed into the normal form (i.e. solved with respect to the differentials). They were solved in engineering math software Mathcad using adaptive Runge-Kutta method.

CALCULATION RESULTS AND DISCUSSION

The aim of the theoretical study is to determine adequate size of the thermal energy storage, its arrangement and operating conditions. Total cooling load required by an office building is 99 kWh (Fig. 3) with temperature available from storage less or equal to 16 °C. Estimates of a storage capacity are calculated based on total system load, number of hours in charging and discharging mode and chiller performance. The results for the PCM storage unit were compared with water-filled buffer tank used as a reference.

The calculations were done using the presented computational model for the following parameters and assumptions:

- constant hot water inlet temperature – T_{M1} = 60 °C,
- constant recooling inlet water temp. – T_{M2} = 27 °C,
- heat transfer fluid (HTF) – water,
- constant HTF mass flow rate – 2900 kg/h,
- phase change material – RUBITERM RT15,
- volume of evaporator and cold receiver – 5 dm^3,
- required average receiver temperature T_C = 17 °C.

Due to the fact that adsorption chiller cooling power depends on temperature T_E, it is difficult to estimate suitable storage capacity. Therefore, the following analysis comprises four different cases. In the first three, water was considered as reference medium to estimate storage volume. In the case 4, the RUBITHERM RT15 was analysed as PCM operating under conditions the same as in case 3. Calculation was conducted up to achieve steady day-to-day state conditions. Summary of the results, pertaining to the last 24 h of calculation, is presented in the Table 2.

Table 2. Operating conditions and simulation results

case no.	storage material	tube no.	M [kg]	SI [kWh]	Q_S [kWh]	Q_R [kWh]	T_{C_max} [°C]
1	water	600	328	10.7	203.9	305.2	33.0
2	water	2000	1093	19.1	216.2	319.2	21.5
3	water	3000	1640	24.1	224.7	328.8	19.4
4	RT15	3000	1264	36.6	235.9	339.5	16.4

Q_S – total input thermal power,
Q_R – total recooling capacity,
M – mass of material used in reservoir to cold storage,
SI – maximum net storage inventory,
T_{C_max} – maximum water temperature in cold receiver.

In general, the main aim of a cooling system is to achieve suitable cooling power and inlet water temperature T_{C1}. In analysed situation adsorption chiller cooling power increases when heat load is rising. In this situation, the best way to check the cooling system working correctness is to analyse the receiver temperature. The temperature T_C range changes with the volume of cold storage. It was assumed that to correctly power the air conditioning system of the building, the average temperature of the HTF in the receiver cannot be higher than 17 °C. Fig. 6 shows calculated cooling water temperature T_C for all cases. From all examined cases, only in the last one T_{C_max} is below the assumed limit.

It is assumed that the adsorption chiller is running for a whole day (around the clock). When the load is less then chiller output, from 0 a.m. to 6 a.m. and from 5 p.m. to 12 p.m. of a design day, excess cold is stored. When the load exceeds the chiller capacity the additional cooling load is discharged from the storage.
In the first case, the dynamic behaviour of the chilling system with small water buffer was examined. Fig. 7 shows that adsorption chiller works almost all the time with a maximum power. In this period the receiver temperature increases significantly above the required limit (Fig. 6.). Time period of storage charge is 5 hours. From 9 p.m. to 6 a.m. next day, the chiller works with a minimum working temperature and unfavourable efficiency.

In calculation case No. 2, the storage reservoir with 2000 tubes was used, which corresponds to 1.09 m^3 volume of thermal storage material. Fig. 8 shows that available cooling capacity of the reservoir is still not sufficient. Period of storage charge is 9 hours from 4 p.m. to 1 a.m. next day. Colling system is not capable to store enough energy to compensate for a system load during the whole day. Lack of capacity reflects on temperature of water T_C which exceeds set up temperature from 1 p.m. to 6 p.m. (Fig. 6).

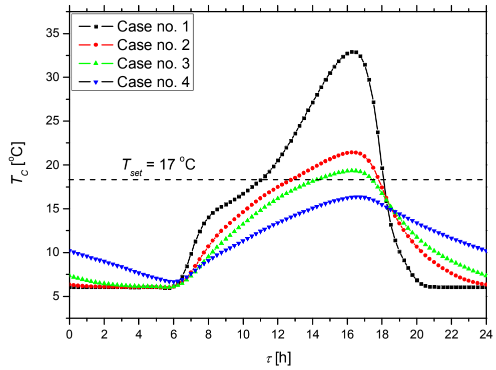

Fig. 6. Daily changes of liquid average temperature in cold receiver T_C

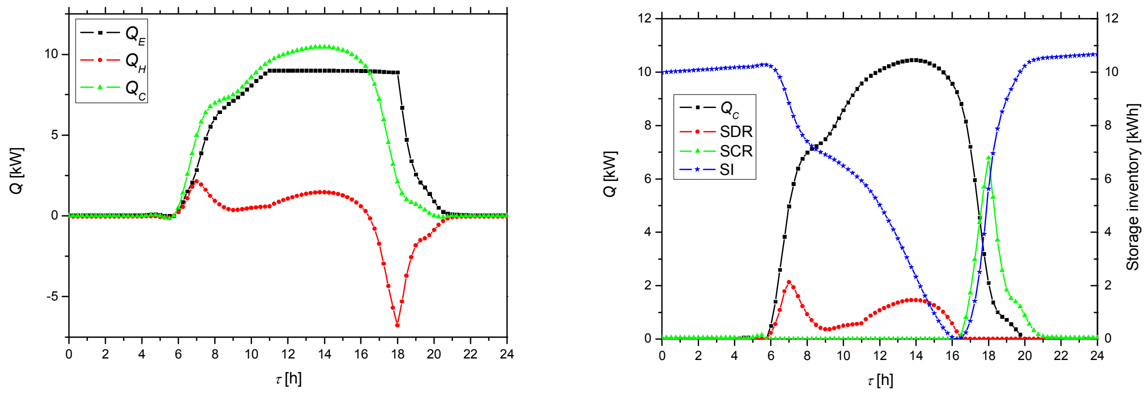

Fig. 7. Dynamic behaviour of heat flow rates, storage loads (SI), storage charge and discharge rates (SCR, SDR) for case 1

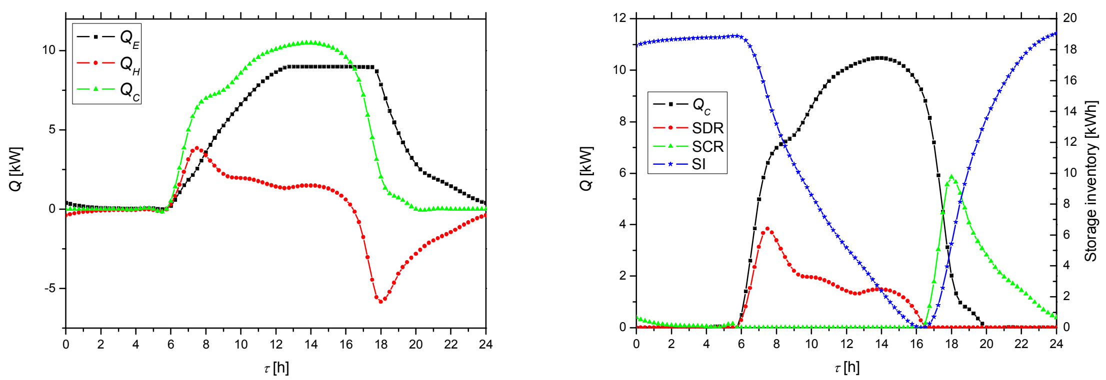

Fig. 8. Dynamic behaviour of heat flow rates, storage loads (SI), storage charge and discharge rates (SCR, SDR) for case 2

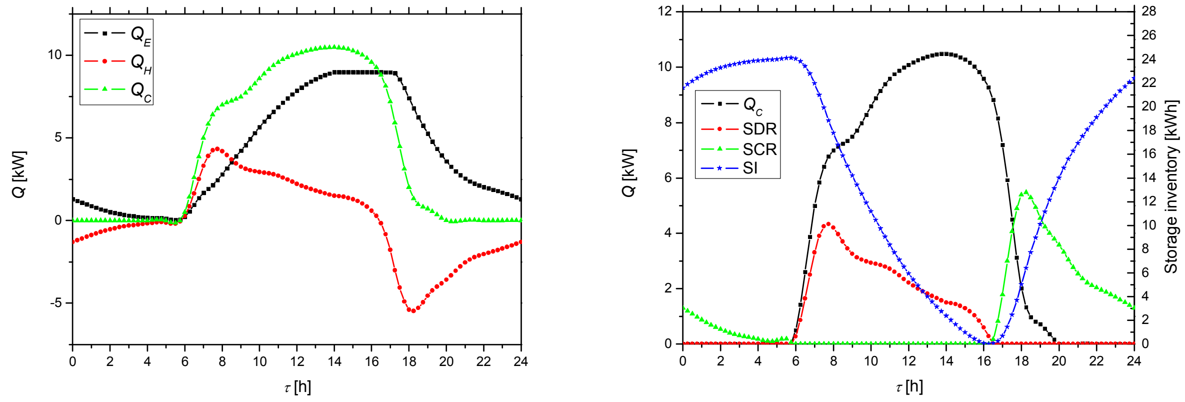

Fig. 9. Dynamic behaviour of heat flow rates, storage loads (SI), storage charge and discharge rates (SCR, SDR) for case 3

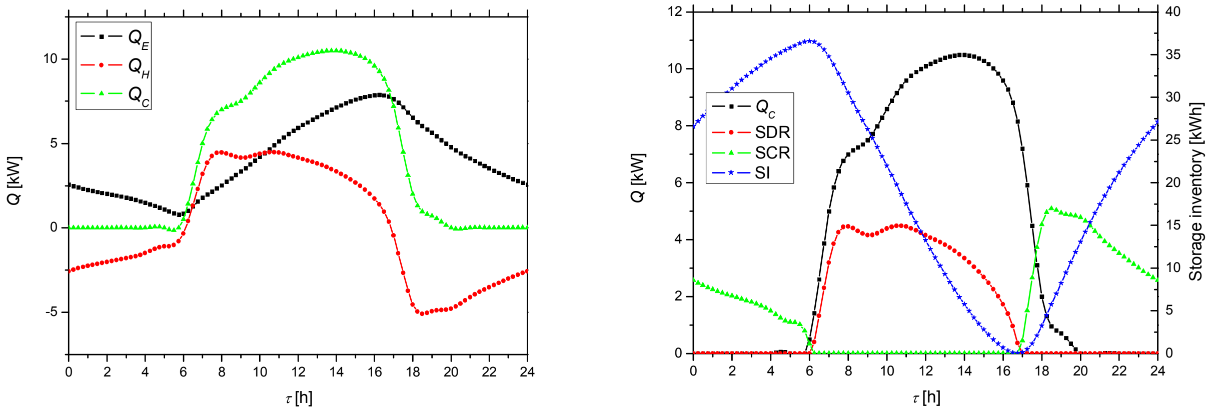

Fig. 10. Dynamic behaviour of heat flow rates, storage loads (SI), storage charge and discharge rates (SCR, SDR) for case 4

In the next case, the number of tubes was increased to 3000 what corresponds to 1.64 m^3 volume of thermal storage material. In this case the period of storage charge is about 13 hours from 4 p.m. to 5 a.m. next day (Fig. 9). Charging process covers almost all time when adsorption chiller power exceeds the office building thermal energy consumption. Among all the examined cases, only the case 3 yielded satisfying results and therefore was taken as a reference.

In the last case RUBITHERM RT15 was taken as thermal storage material with reservoir geometry identical to case No. 3. Despite the lower mass of PCM, accumulated amount of thermal energy was higher than for water. Fig. 10 shows that charge rate at the end of the charge process (at 6 a.m.) is far from zero. It seems that optimal size of thermal energy storage reservoir should be lower. It should also be noted that in more realistic simulation, the influence of temperature control system should be included in the model calculations.

CONCLUSIONS

In this study, theoretical analysis of a cooling cycle with adsorption chiller and thermal storage reservoir was done. Lumped parameter model equations of the system were formulated and solved to determine its thermal dynamics. Presented iterative method allows to estimate the optimal volume of thermal storage material. Generally, using PCM instead of water allows to reduce the size of cold thermal buffer especially when the range of working temperatures is narrow. For the selected here thermal energy consumption of the office building and the adsorption chiller power characteristic, the buffer with 3000 tubes should be chosen. Required mass of phase change material RUBITHERM RT15 is about 1300 kg.

Due to a high process complexity it is impossible to find simple and accurate method to describe the performance of such system over time. In the presented model control system and receiver with real thermal inertia should be taken into account. In the future, experimental validation in limited scale of the model predictions is planned. This is particularly necessary to verify the heat transfer coefficients used in the model.

NOMENCLATURE

A	surface area	m2
c	specific heat	J/kg·K
d	diameter	m
□h_{SL}	latent heat	J/kg
HTF	heat transfer fluid	
k	overall heat transfer coefficient	W/m^2K
l	length	m
$\dot{m}$	average mass flow rate	kg/s
M	mass	kg
Nu	Nusselt number	
PCM	phase change material	
$\dot{q}$	heat flux	W/m^2
R	thermal resistance	K/W
SCR	storage charge rate	W
SDR	storage discharge rate	W
SI	storage inventory	Wh
Q	heat	J
$\dot{Q}$	heat flow rate	W
T	temperature	K (°C)

Greek symbols

α	convective heat transfer coefficient	W/m^2K
□	thickness	m
ρ	mass density	kg/m^3
λ	thermal conductivity	W/m·K
□	time	s

Subscripts

1, 2	fluid inlet and outlet, respectively
P	phase change material (PCM)
H	heat transfer fluid (HTF)
W	tube wall
C	cold receiver

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RUBITHERM RT15 data sheet, www.rubitherm.com