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Article

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Performance evaluation of latent heat TES system-case study: Dimensions improvements of annular fins exchanger

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Abstract

Thermal Energy Storage (TES) is an important challenge for the development and implementation of Concentrated Solar Power (CSP) plants. The heat can be stored in a sensible, thermochemical or latent form. Recently, the latent heat thermal energy storage (LHTES) kind has gained the most of research interest as it has the advantage of increasing the energy density of the current commercial TES systems through the use of a Phase Change Materials (PCM). However, the main limitation of PCMs technology is the low thermal conductivity, which affects the system performances and lead to significant energy losses. In this sense, a good container design can impact positively the system performances. For that sake, several designs have been proposed to overcome the decreasing heat transfer rate. For this purpose, this study propose many enhancements and improvements of the fins method based on the integration of PCM in an exchanger with annular fins. The main objective of this paper is to study the impact of finned exchangers on the total flow transferred and also on the height of these annular fins. Finally, the physical model describing the thermal behavior of the studied system is developed and simulated after 80 min of charging process.

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Keywords: Annular fins; Exchanger fins; Latent heat thermal energy storage; Phase change material; Thermal conductivity; Length of the fins

1. Introduction

Thermal energy storage (TES) is an essential component for concentrated solar power plants (CSP), as it allows them to improve the performance and energy availability of CSP plants. The use of TES can overcome the gap between energy production and demand [1]. The three thermal storage methods are thermochemical storage (TCS), sensible heat storage (SHS) and latent heat storage (LHS). The latent heat thermal energy storage (LHTES) based on

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Nomenclature

D	Diameter [m]
$D_{\text{-tube,out}}$	Diameter of outside tube [m]
$D_{\text{-tube,in}}$	Diameter of inside tube [m]
H	Height [m]
r	Radius [m]
C_p	Heat capacity [J/(kg K)]
k	Thermal conductivity [W/(m K)]
ρ	Density [kg/m ³]
L	Latent heat [kJ/kg]
f_L	Liquid fraction
T_m	Melting temperature [°C]
P	Pressure [Pa]
Re	Reynold number
v	Velocity [m/s]

phase change material (PCM) is the method that most attracted the attention of researchers in the last years because of its higher storage density, often greater than 100 kJ/kg with relatively small volumes compared to SHS reservoirs, and its small temperature variation between storage and recovery [2,3]. Main disadvantage of LHTES systems is the low thermal conductivity [4]. Regarding this problem, the literature proposes several ideas and techniques for improving the heat transfer, among these techniques there are finned exchangers based on the surface exchange between the heat transfer fluid (HTF) and the PCM [5–7]. In this context, this work consists in studying the thermal conductivity improvement of the PCMs based on an annular fins exchanger. And this through evaluation of fins length effect on the heat transfer rate. This paper evaluates the performance of six different lengths to have the optimal length for this system. This evaluation is done by a numerical study based on the finite element method. Calculations are used to perform several simulations after 80 min of charging process. The system is a two tube exchanger with annular fins fixed to the inner tube. The PCM (NaNO₃–NaOH) in the annular space between the two tubes, and HTF (DelcoTerm) circulates inside the inner tube, with an inlet temperature of 300 °C.

2. Two tubes model with annular fins

2.1. Numerical method

Axisymmetric two-dimensional (2D) numerical modeling of charging processes was performed using a numerical study based on the finite element method. The finite element approach was adopted to discretize the governing equations of the problem. Due to the energy transfer coupling between HTF and PCM, and to solve the problem of heat transfer, the energy conservation equations for HTF and PCM were integrated across the physical domain.

2.2. Physical model

The retained geometry (Fig. 1) consists of the copper tube, around which are 99 annular aluminum fins. The diameter of the inner copper tube is $D_1 = 2.4$ cm, with a thickness of 2 mm. The diameter of the outer tube is $D_2 = 5.8$ cm, with a thickness of 2 mm. The height of the computation domain is $H = 30$ cm. Fig. 1 presents the description of the system.

The HTF inlet in the top of the tube with an inlet diameter $D = 2$ cm. the upper and lower flanges are not represented in the geometry; they are modeled via boundary conditions.

In this work the objective is to have the optimal geometry at the length level of the fins. For this we have chosen six different geometries at the level of length and with the same thickness and the same distance between the fins. The dimensions of the fins are summarized in Table 1.

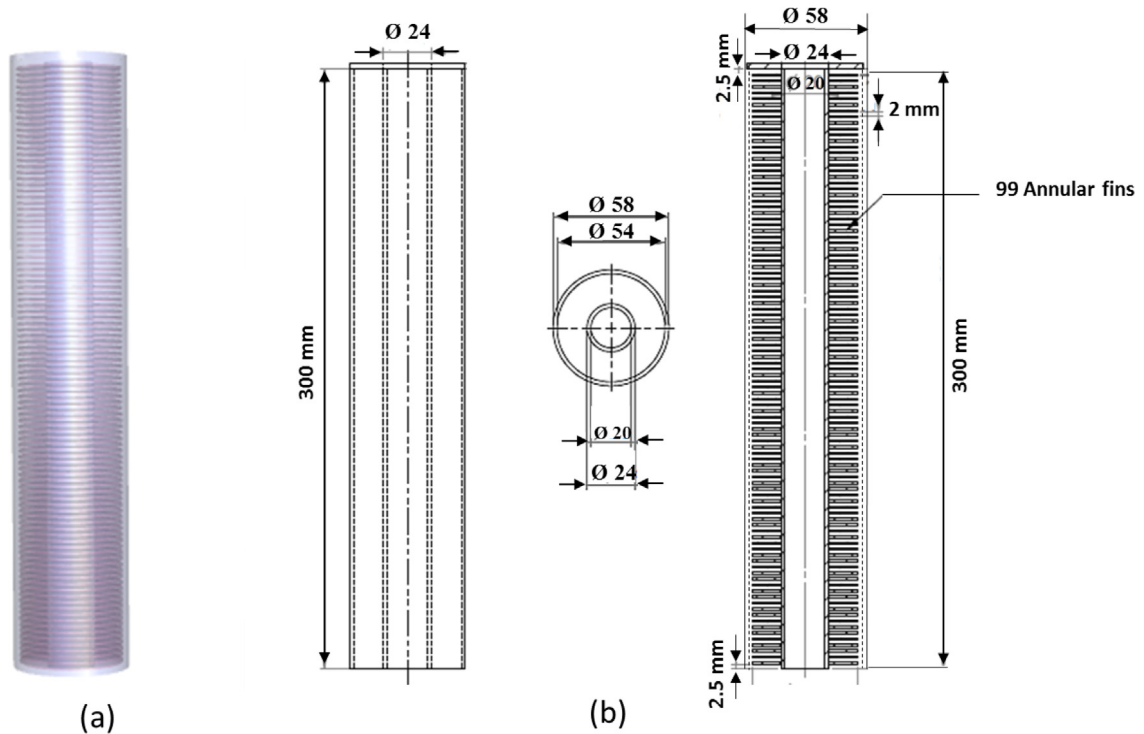


Fig. 1. Two pipes with annular fins attached to the inner tube, (a) 3-D visual description; (b) 2-D description.

Table 1. Geometric comparison of the different fins.

	Geo 1	Geo 2	Geo 3	Geo 4	Geo 5	Geo 6
D-tube, out/D-tube, in [mm]	40/24	44/24	48/24	50/24	52/24	53/24
Thickness [mm]	1	1	1	1	1	1
Length [mm]	08	10	12	13	14	14.5

2.3. Numerical model

For the storage of latent heat, the principle is based on the absorption energy during the transformation of a material from the solid state to the liquid state. During the inverse transformation, the energy is released. In our case study, we model the system using mathematical equations that are based on time, temperature, and material properties.

Heat conduction is the physical phenomenon relating to the change of temperature in a homogeneous material without movement. This phenomenon is modeled by the equation of heat, with physical properties assumed to be constant.

$$\frac{\partial h}{\partial t} + \nabla(-k\nabla T) = 0 \quad \text{avec } h = \rho C_p T \quad (1)$$

For this relation (1), C_p shows the heat capacity of the PCM (J/(kg K)), k the thermal conductivity (W/(m K)) and ρ the density (kg/m³). The function h (J/m³) is the enthalpy function related to the elevation of the temperature by conduction in the material.

If the material absorbs or releases heat (this is the case during melting), it is necessary to add a source term in the thermal equation to respect the energy balance. So relationship (1) becomes:

$$\frac{\partial h}{\partial t} + \rho C_p \vec{u} \cdot \nabla T + \nabla(-k\nabla T) = Q \quad (2)$$

The enthalpy model was introduced with the aim of simplifying the writing of the phase change problem on the basis that the same equations govern the thermal transfer in the liquid phase and in the solid phase. Eq. (3) presents the enthalpy model as a function of the evolution of the total liquid fraction in the PCM volume.

$$Q = -\rho L \times \frac{df_L(T, t)}{dt} \quad (3)$$

where L is the latent heat of fusion of PCM (J/kg) and f_L the fraction of liquid. For an isothermal change of state, the liquid fraction $f_L(T, t)$ is given by the following discontinuous expression:

$$f_L(T, t) = \begin{cases} 0 & \text{solid} & T < T_{\text{melting}} \\]0, 1[& \text{mushy} & T = T_{\text{melting}} \\ 1 & \text{liquid} & T > T_{\text{melting}} \end{cases} \quad (4)$$

The value of the liquid fraction is zero in the solid phase, it is 1 in the liquid phase and it varies between 0 and 1 in the mushy zone. This zone is a source of instabilities, it corresponds to the whole coexistence domain of the solid and liquid phases and where the specific enthalpy is between that of the solid and that of the liquid.

- Boundary conditions

The initial conditions (at $t = 0$) are the same for all six geometries. The initial temperature is uniform; it is less than T_m in the case of melting, where the initial system temperature is 180 °C. The initial and uniform liquid fraction. It is 0 in the case of the merger (PCM is in solid phase). It is 1 in the case of all PCM is liquid. The initial velocity of the HTF is considered zero.

The HTF inlet is at the top, defined by a mass flow rate of 60 kg/h and a temperature of 300 °C. The output condition is a differential pressure set at $P = 0$ Pa. At the top, bottom, and around the PCM domain, the exchange coefficient with the outside is zero. The pressure at PCM is constant and corresponds to atmospheric pressure 101 325 Pa.

- Simplifying hypotheses

- The outer wall surface is supposed to be adiabatic.
- The fluid is supposed to be Newtonian and incompressible.
- HTF and PCM have constant thermophysical properties.
- The inner tube is supposed to be thin and the material that constitutes it is supposed to be a good thermal conductor.

2.4. Physical properties of constituent materials

The properties of the materials listed below (Table 2) are those retained for the tube (copper), the fins (aluminum) and the PCM (NaNO₃–NaOH).

Table 2. Thermophysical property of PCM, copper and aluminum [8,9].

	NaNO ₃ –NaOH (86–14)	Copper	Aluminum
Melting temperature [°C]	250	–	–
Thermal conductivity, solid [W/m K]	0.66	398	220
Thermal conductivity, liquid [W/m K]	0.6	–	–
Latent heat [kJ/kg]	160	–	–
Specific heat, solid [J/kg K]	1190	386	900
Specific heat, liquid [J/kg K]	1860	–	–
Density, solid [kg/m ³]	2241	8930	2700
Density, liquid [kg/m ³]	1975	–	–

Fig. 2 shows the evolution of total energy as a function of temperature. The blue curve represents the evolution of the specific enthalpy, blue and red dashed lines represent tangents on both sides of specific enthalpy. the black dotted line represents the tangent of the phase change curve. This curve indicates that the melting ranges are between 246 °C and 254 °C.

The DelcoTerm (HTF) specifications are shown in Table 3. The HTF circulates in laminar flow ($Re = 1939 < 2300$) inside the inner tube and is subjected to a constant inlet temperature of 300 °C. The mass flow rate is 60 kg/h leading to an average velocity of $v = 0.074$ m/s.

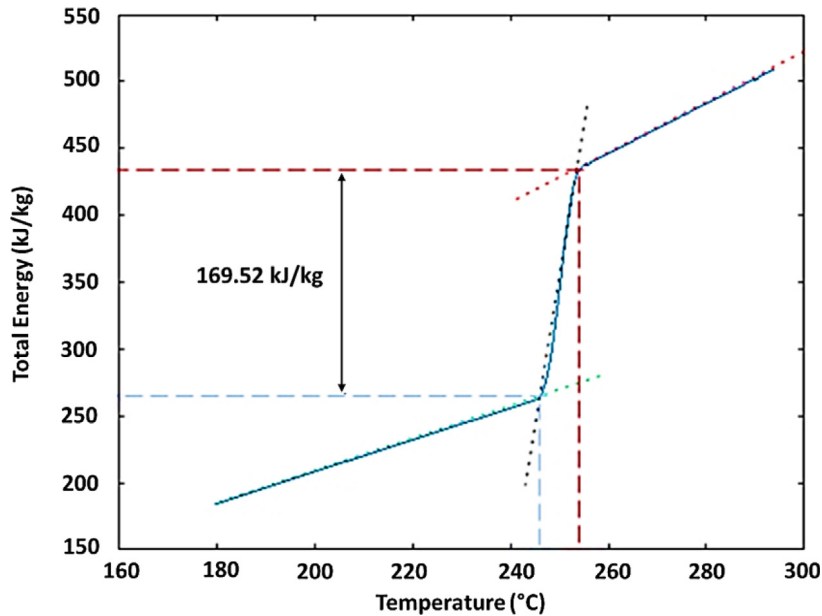


Fig. 2. Evolution of total energy as a function of temperature of the NaNO₃–NaOH [10]. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

Table 3. Specifications of DelcoTerm Solar E 15 [11].

HTF	DelcoTerm Solar E 15
Working temperature range [°C]	180–300
Mass flow rate (charge) [kg/h]	60
Velocity [m/s]	0.074
Specific heat (180 °C) [J/kg K°]	2477
Thermal conductivity (180 °C) [W/m K]	0.1234
Density (180 °C) [kg/m ³]	717.8

3. Results and discussion

In the PCM the points to be studied are determined by dividing of the whole system into four axial sections of similar dimensions. These sections are referenced from 1 to 4, the four sections are 75 mm each. Section 1 being the upper section, and Section 4, the lower section. The study points are located between the sections. That is, between Sections 1 and 2 there is point A, between Sections 2 and 3, point B and between Sections 3 and 4, point C. The three points are located at 1 mm of the outer tube and at 14 mm of the base of the fin (inner tube) with different positions on the length of the system. A: $z = 0.225$ mm. B: $z = 0.15$ mm. C: $z = 0.075$ mm.

Figs. 3–5 illustrates three graphs corresponding to recorded values at the three selected points in this study, and that for the six geometries. All graphs show the temperature evolutions in the PCM during melting processes.

The PCM temperature increase speed is decreasing from the top to the bottom of the system (A, B, and C respectively). This result corresponds to the charging process which takes place from the top of the system to its bottom, thus the energy content of the HTF is decreasing in this direction. Moreover, this finding is clear on the three parts of the curve: (1) the cold sensible zone, (2) the latent zone, and (3) the hot sensible zone. As the most significant zone for the LHTES system is the second one, the emphasis of the following discussions should be restricted to this zone. First of all, and corresponding the above mentioned result, the time length of the latent zone increase from top to bottom (from point A to point C respectively), which means that: the phase change time decreases in this direction due to the same already explained reasons. Secondly, it can be seen clearly that the overall heat transfer mechanisms within the LHTES system are enhanced as the fins length increases. This result is demonstrated in the three graphs corresponding to the three chosen points of measurement (A, B, and C),

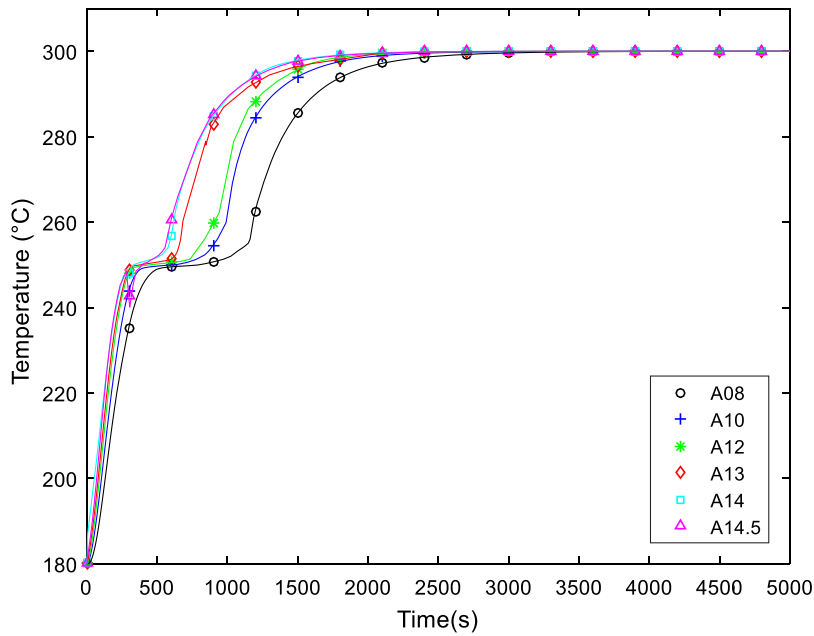


Fig. 3. Temperatures evolution in the PCM during charging processes for the 6 geometries, point A: $z = 0.225$ m, $r = 0.026$ mm.

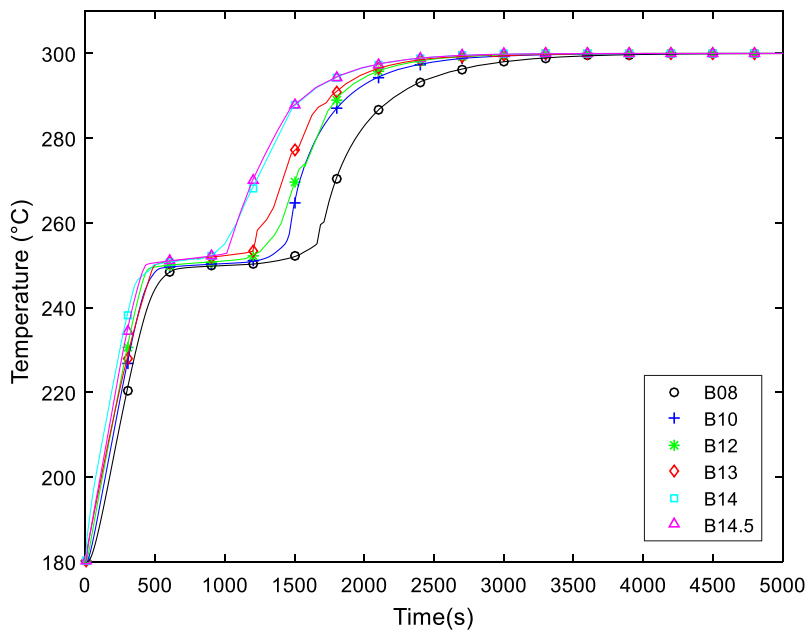


Fig. 4. Temperatures evolution in the PCM during charging processes for the 6 geometries, point B: $z = 0.15$ m, $r = 0.026$ mm.

i.e.: the time length of the latent zone in the three positions decreased with respect to the fines length extension. Consequently, the proposed enhancement improves significantly the charging time of the LHTES system. Finally, it was found that the geometry enhancement process has an optimal point where the overall LHTES performances reach a stagnation point. In the current study, this point corresponds to the fine length value of 14 mm (geometry 5). It is to be noted that the overall heat transfer speed was increased by at least 50% as shown in Table 4.

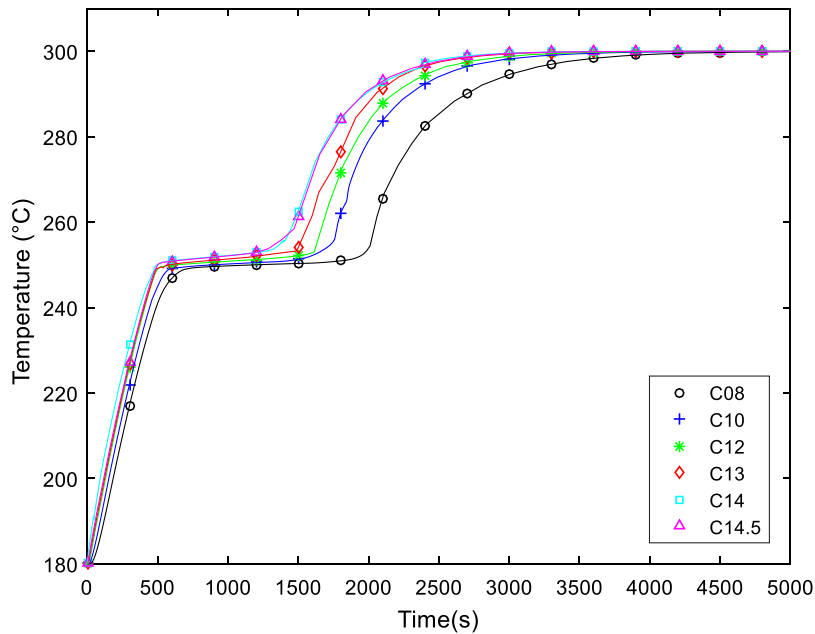


Fig. 5. Temperatures evolution in the PCM during charging processes for the 6 geometries, point C: $z = 0.075$ m, $r = 0.026$ mm.

Table 4. Effect of increasing the height of the fins on the time during the charge process, in comparison with the first geometry (08).

	Geo 2 (10)	Geo 3 (12)	Geo 4 (13)	Geo 5 (14)	Geo 6 (14,5)
Melting	19,31%	26,03%	42,77%	50,64%	51,3%
Charging	15,24%	21,07%	26,6%	29,17%	29,34%

4. Conclusion

In this work a numerical modeling of the melting process was carried out in order to evaluate the performance of an annular fin exchanger at different lengths of the fins. Used DelcoTerm as HTF and NaNO₃–NaOH as PCM. The evolution of the temperature profiles in the PCM domain were compared for the six geometries in the three graphs corresponding to the three chosen points of measurement (A, B, and C). The time length of the charging system in the three positions decreased with respect to the fins length extension. In order to choose the optimal geometry, we conclude that geometry 5 which is 14 mm in length is the optimal geometry for our system. It should be noted that the overall rate of heat transfer was increased by at least 50%.

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