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Thermo-physical effect of solid filler on the performance of a packed-bed thermal storage

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ABSTRACT

This paper attempts to investigate the effect of the diameter of the solid filler and the porosity of the heat transfer fluid on the selection process of suitable material given best performance in terms of storage capacity of thermal energy storages coupled with parabolic trough concentrating solar power plants. In addition, an optimization model is proposed. The model is formulated as a constrained optimization problem to optimally design a rock-bed storage system. The main objective is to determine optimal values of the porosity and the solid filler diameter of appropriate materials maximizing the storage capacity. A comparative study is considered, where the operation of the thermocline storage is simulated taking into account two different materials, the concrete HT and the granite. The results show that the selection of suitable material depends strongly on the porosity and the solid filler diameter.

1. Introduction

Currently, concentrating solar power is one of the promising technologies, having low impacts on the environment, economically viable, and can meet the various energy challenges. This technology, which uses intermittent and fluctuating source, cannot cover the energy needs of consumers over time, which highlights the need to develop new storage solutions. Sensible heat storage technology in the solid is considered to be the most mature, exploited, and economical technology. In 2003, the German aerospace center successfully tested the first heat storage unit in concrete at the Almeria Solar Platform. This unit mainly consists of heat transfer tubes embedded in parallel in the concrete allowing the fluid flow for temperatures between 340 °C and 380 °C. Although, substantial amount of stored energy requires a large storage system. The best solution remains the case of materials with a high heat capacity, which can store the maximum of energy at low cost in a relatively small space.

In general, solar power plants have shown significant efficiency in terms of energy conversion at low costs. There is great interest in building solar thermal power plants as a clean and renewable energy source. The fluids used in solar power plants may reach typical operational temperatures of few hundred degree Celsius, which may allow to an energy saving of 40%. Electricity produced from solar power plants is still more expensive than the one coming from traditional thermal power plants. In addition, solar energy production depends strongly on sunlight, which is available less than 12 h a day. Commonly, the electrical power demand is highest in the afternoon during the summers, and early in the evening in the winters. Hence, the solar power plants need to extend their operation until nighttime. One of the key solutions to solve this issue is the integration of thermal energy storage systems with solar power plants. These thermal energy storages can serve to store excess of thermal energy during the day and can be considered as a source of thermal energy at nighttime. Furthermore, they may play a key role in ensuring the power balance, stabilizing the power supply and coping with fluctuation of the solar irradiation. Currently, several energy storage technologies exist to meet the night peak loads [1–6]. Studies have shown that thermocline heat storage systems offer the lowest cost of storing thermal energy. These thermocline storages, like storage with two tanks, can operate directly or indirectly. Rock-bed thermocline storage systems have gained a great interest, and many examples of implementing such storage systems for concentrated solar power plants exist, namely, Coolidge solar powered irrigation pumping project, USA, IEA-SSPS project, Almeria, Spain, solar one project, Barstow, USA, and Sandia project, USA.

In general, storage material should have certain essential characteristics and properties, namely: high thermal storage capacity of the material per unit of volume and mass; High thermal conductivity within the operating temperature limits; High density; Excellent charging/discharging capacity with a large number of operating cycles; Long-term thermal reliability and stability, even after several thousand

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Nomenclature

ε	porosity of dual-media region
C	specific heat capacity, J kg^{-1} K^{-1}
d_p	diameter of the solid filler, m
H	heat transfer coefficient, W m^{-2} K^{-1}
H_{eff}	effective lumped capacitance heat transfer coefficient, W m^{-2K-1}
H	tank height, m
H_1	overall heat transfer coefficient between fluid and solid for the fluid energy equation, s^{-1}
H_s	overall heat transfer coefficient between fluid and solid for the solid energy equation, s^{-1}
K	thermal conductivity, W m^{-1} K^{-1}
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
S_s	total surface area of solid filler per unit length of a cylindrical storage tank, m
T	average temperature over a cross section perpendicular to the tank axial, K
t	time, s
v	intrinsic velocity of the fluid, m s^{-1}

Greeks

ε	porosity of dual-media region
μ	viscosity, kg m^{-1} s^{-1}
ρ	density, kg m^{-3}

Subscripts

f	fluid
in	inner size
out	outer size
s	solid

operating cycles; Chemical stability for long periods without decomposition; Non-toxic, non-explosive, low corrosion potential or non-reactive in the heat transport medium; mechanically stable; Low cost and environmental impact.

In the literature, numerous studies have been performed. Although, considerable efforts have been concentrated on numerical and experimental studies of thermal performance of the thermal energy storage. Authors in [7] compared two type of rocks with molten salts. The thermophysical and mechanical properties were evaluated, the results validate adoption of rocks given their energy potential, thermal stability, low cost and environmental impacts. Ref. [8] presented a chronology of numerical simulation methods to evaluate the performance of the rock bed storage system, while displaying the main empirical correlations in order to consider the natural convection inside the spheres or the effective thermal conductivity of the fluid. In [9] authors studied the influence of the concrete on the performance of the storage system of four different structures, namely: the channel-embedded structure, the parallel-plate structure, the rod buddle structure and the packed-bed structure using the so-called lumped capacitance method. The rock bed shows best discharging results even considering variation of the flow speed and size of the storage systems. A comparison between an experimental rock bed storage model and a numerical simulation for temperatures ranged between 20 and 650 °C is reported in [10]. The model is applied to the design of two thermal storage units on industrial scale, each with a capacity of 7.2 GWhth. Authors in [11] developed a heterogeneous model of storage in a rock bed using the Modelica. The model has been validated using data from the literature. A storage system composed of two rock beds has been studies in [12] using 3D simulation on the Fluent software. The results show that the temperature is constant, which proves that the thermocline can be integrated in the concentrating solar power. Authors in [13] demonstrated that the efficiency of a rock bed storage system is linked to the thermocline zone. In addition, they validated that the height of the reservoir and the thermo-physical properties of the filling materials strongly affect the efficiency of the tank. Ref. [14] studied the influence of external conditions on the charge and discharge cycles of molten salts through a reservoir filled with rock. The results show that when working in a non-adiabatic environment, the heat losses strongly affect the temperature distribution, thus causing a rapid drop in the temperature at the outlet. While, when the Reynold number is greater than 250, the discharge efficiency is almost the same for both adiabatic and non-adiabatic environmental conditions. The authors in Refs. [15] and [16] have proposed a comprehensive decision model to optimally identify the solid medium and heat transfer tubes material composing the thermal energy storage and formulated a constrained optimization problem to optimally design the geometry variables that maximize the net present value associated to the thermal storage investment. To simulate the behavior of the storage system in the rock bed reservoir, several methods [17–19] were applied in order to solve the equations of Schuman [20] who was the first author who presented a set of equations governing the conversation of the energy of the fluid through a porous medium. Ref. [21] investigated the thermal performances of a PCM and designed a PCM- metal foam composite thermal storage device. Authors in [22] performed a theoretical study of the forced convection heat transfer in a microchannel partially filled with a porous medium. An experimental validation of a modified collocated unit cell framework estimating the effective thermal conductivity of packed beds is presented and described in [23]. Authors in [24] compared the performance of multi-tank modular system with that of conventional single tank system. Authors in [25] studied the ability of latent heat thermal energy storage system with phase change material to store/release a large quantity of energy in a small volume. A numerical investigation on the influence of the location of the phase change material in a concentric double pipe latent heat thermal energy storage system is reported in [26]. An experimental study of a thermal energy storage based phase change materials is investigated in [27]. Ref. [28] compared the efficiency of multi-tank modular system with that of conventional single tank unit. The flat heat pipe efficiency is investigated in [29] considering various structures. Ref. [30] reviewed divers approaches of thermal energy storage comprising sensible, latent and thermochemical energy storages, concentrating on phase change materials.

The previous literature discussion shows that prior works present complex methods and models that are challenging to implement, and necessitate computation time requirement. Furthermore, some of the previous methods are computationally intensive, or may generate suboptimal solutions. Compared to previous works, the strengthen of this paper might be summarized in proposing a reduced and efficient decision support framework formulated as a constrained optimization problem to optimally design a rock-bed storage system.

The originality of this paper can be summarized in studying the effect of the diameter of the solid filler and the porosity of the heat transfer fluid on the selection process of suitable material given best performance in terms of storage capacity. In addition, we developed a decision model formulated as a constrained optimization problem to optimally design a rock-bed storage system. The main objective is to determine optimal values of the porosity and the filler diameter of appropriate materials maximizing the storage capacity.

2. Operation of thermocline thermal storage systems

Rock-bed thermocline storage uses a single tank filled with solid material such as rocks as reported in Fig. 1. It was originally designed to use air as a heat transfer fluid, but it has also become possible to use liquid fluids. During the storage process, a thermal gradient separates the hot zone from the cold zone. During charging and discharging modes, the thermocline zone moves axially and can represent up to a third of the height of the tank. The tank is fully insulated with supply ports on the top and bottom. Diffusers at both ends keep the existing stratification in the tank. During the charging period, the hot fluid is introduced into the upper part of the tank, flows down through the rock-bed and the cold fluid is evacuated through the diffuser from the lower part. During this stage, the hot fluid raises the temperature of the storage materials. During the discharging period, the direction of the fluid flow is reversed and the cold fluid enters from the lower part. This fluid is heated as it circulates through the solid filler. During these charging and discharging phases, the thermocline zone will move until it leaves the tank.

The temperature trends of the thermocline storage vary accordingly with the height of the rock-bed. These curves cover a range of charging percentage varying between 0% (fully discharged) and 100% (fully charged). In the fully discharged stage the ambient temperature occupy up to 1/3 of the height of the storage system, then, it begins to rise along the height of the bed. At the top, the temperatures are close, but still below the maximum temperature of the material and do not reach the maximum air temperature. While in the charging process, the hot air passes through the rock-bed from top to bottom and the time required for full charge depends on the design of the storage system. At the start of the charging process, only the rocks near to the top are heated and reach the maximum temperature, while the lower levels are less heated. When the material of the upper layer reaches the maximum temperature, the hot air transfers the heat to a lower layer. Over time, the lower layers will also be heated to the maximum temperature. The storage system is said to be fully charged when the hot air passing through it transfers so little heat to the material at the outlet. In general, the storage system is fully charged when about 2/3 of the height reach the maximum temperature.

3. Modelling the rock-bed thermocline storage tank

3.1. Description of the rock-bed model

As an example, the granite densely fill the vertically placed cylindrical reservoir and the fluid flows pass through the spaces between the solid filler. The simplified rock-bed unit is illustrated in Fig. 2; the storage unit consists of a tank of height *H* filled with rocks considered uniform with diameter *dp*. For simplification purposes, we assumed that the heat conduction in the axial direction of the fluid and rock tank is neglected; the thermal properties of solid and fluid are invariable; heat loss neglected.

3.2. Mathematical modelling

In the charging and discharging process, the filler material and the heat transfer fluid will have a temperature difference in each position. At the end of the charging or discharging process, a temperature equilibrium will held between the material and fluid. In this case, the energy balance can be expressed by [31]:

$$\varepsilon\rho_f C_f T_{f-initial} + (1-\varepsilon)\rho_s C_s T_{s-initial} = \varepsilon\rho_f C_f T_{f-final} + (1-\varepsilon)\rho_s C_s T_{s-final}$$

The thermal energy balance of the fluid in the volume dz is given by:

$$\varepsilon\rho_f \pi R^2 U(h_z - h_{z+dz}) + hS_s(T_s - T_f)dz = \varepsilon\rho_f C_f \pi R^2 dz\frac{\partial T_f}{dt}$$

where the mean fluid speed in the rock bed is:

$$U = \frac{\dot{m}}{\rho_f a_f}$$

The energy balance of the filler material in a control volume dz can be formulated as:

$$hS_s(T_s - T_f)dz = -\rho_s C_s(1-\varepsilon)\pi R^2 dz\frac{\partial T_s}{dt}$$

Considering the assumptions listed in the previous section, the energy equations for both fluid and solid that can be formulated in Schumann equations are given by:

$$\frac{\partial T_f}{\partial t} + v\frac{\partial T_f}{\partial z} = H_1(T_s + T_f)$$

$$\frac{\partial T_s}{\partial t} = -H_s(T_s + T_f)$$

where *Ts* is the average rock temperature, *v* is the velocity of the fluid, H_l and H_s express the overall heat transfer coefficients, given by.

$$H_s = \frac{6h_{eff}}{\rho_s * C_s * d_p}$$

$$H_1 = \frac{6(1-\varepsilon)h_{eff}}{\rho_f C_f \varepsilon d_p}$$

where h_{eff} is the corrected heat transfer coefficient between solid and fluid for the lumped capacitance method derived from Ref. [9].

3.3. The corrected heat transfer coefficient

Ref. [19] proposed a corrected heat transfer coefficient, which considers the temperature gradients within solid filler. The correction factor is expressed in terms of the Biot number, and is described as follow:

$$h_{corrected} = \frac{h}{1 + B_i/5}$$

Ref. [32] developed a one-dimensional model expressing the heat exchange coefficient for very large Biot numbers using the “global thermal capacity” method. Furthermore, authors checked the reliability of this model with the analytical solution of Ref. [33]. In case of the thermocline thermal storage, the corrected exchange coefficient is written as follow:

$$h_{eff} = \frac{1}{\frac{1}{h} + \frac{d_p/2}{5k_s}}$$

where *h* is the heat transfer coefficient for laminar flow, which is

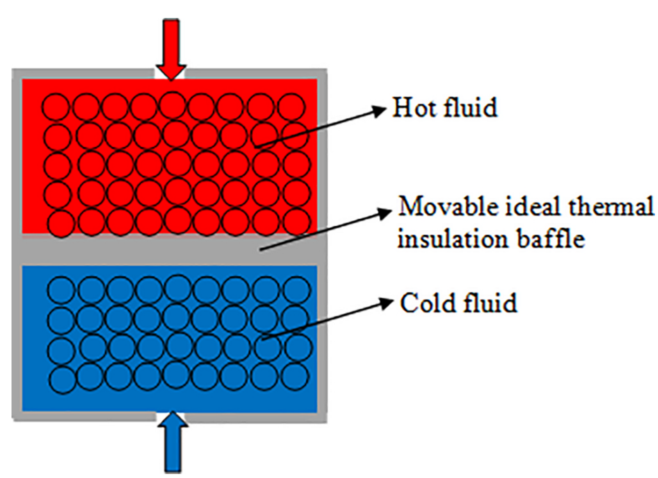


Fig. 1. Operation of thermocline thermal storage.

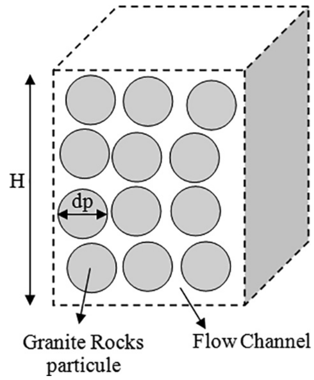


Fig. 2. Rock-bed reservoir model.

defined by:

$$h = \frac{k.\ Nu}{d_p}$$

$$Nu = 2 + 1.1Re^{0.6}\ Pr^{1/3}$$

where μ_f is the dynamic viscosity, kg.m^{-1} m^{-3}, and v_f is the intrinsic velocity of the fluid, m.s^{-1}

3.4. Optimization problem formulation

The objective of the optimization problem is to maximize the amount of energy released or stored considering the operational constraints of the system. The energy released from the storage unit is formulated in the following the form:

$$Q_{dis} = \rho_s c_{ps}.\ V\,(T_{s,int} - T_{e,end}).\ \varphi$$

$$\varphi = \frac{Q_{released}}{Q_{stored}} = \frac{\rho_s c_s.\ (T_{s,int} - T_{e,mean})}{\rho_s c_s.\ (T_{s,int} - T_{e,end})}$$

$$V = (\pi \times R^2 \times H)(1 - \varepsilon)$$

Subject to the following constraints:

$$0.10 \le \varepsilon \le 0.50$$

$$Q_{dis} \le 1000\ \text{ kW h}$$

4. Numerical results and discussions

In this case study, we considered the therminol VP-1 as a heat transfer fluid, which is widely used in concentrating solar power systems. It is assumed that in the charging process, both rock and heat transfer fluid have an initial temperature equal to 340 °C. The fluid is conducted from the top of the tank to a high temperature of 380 °C, then crosses the solid down through the channel with an input velocity v = 0.04 m/s. We mention that we assumed a laminar flow with low Reynold number, and temperature gradient will be carried out by conduction. On the other hand, a low velocity is necessary to optimize the exchanges between the rock-bed and the fluid. Furthermore, it is supposed that the thermal properties of the fluid are constant with the temperature and the height *H*, the radius *R*, the porosity and the solid filler diameter of the equivalent tank are set to be equal respectively to 7 m, 4 m, 0.25 and 0.04 m. The diameter of rocks is assumed to have low value, this is will increase the exchange surface between the fluid and the solid. In addition, with the thermal properties of the storage system reported in Table 1, there will be a low Biot number (Bi < 0.1) and therefore the temperature will be more homogeneous within the material. The porosity in the storage system represents the existing volume of fluid relative to the total volume of the reservoir. For example, a porosity equals to one means that the total volume of thermocline is completely filled with the fluid.

The proposed model has been compared and validated considering small charging and discharging cycles (reservoir height = 0.767 m, reservoir diameter = 0.241 m, porosity = 0.326, rock diameter 0.02 m) as reported in [34], where authors used method of characteristics. The comparison shows approximately the same results taking into account difference among parameters used in both methods. Furthermore, the lumped capacitance model adopted in this paper has been validated in [35].

4.1. Scenario1: Rock diameter variation

The effect of the spherical diameter of the solid filler on the performance of the system is investigated in this section through testing different diameters of the rock ranging from 0.01 to 0.04 m. The results show that the heat transfer between the solid and the fluid is significantly influenced by the diameter of the solid filler. The increase in solid filler diameter reduces the rate of heat transfer between the solid filler and the heat transfer fluid. Consequently, the charging time will increase as reported in Fig. 3, where variation of the stored energy during the charging process is displayed. While, when the solid filler size is small enough, the tank charges promptly.

The time varying temperature of the solid filler in both charging and discharging modes are respectively shown in Figs. 4 and 5. As the solid filler diameter increases, the temperature of the solid begins to decrease faster and the discharge time increases. In addition, the time required for the temperature to drop from 380 to 340 °C rises significantly. The system can be evacuated faster when reducing the solid filler diameter. Considering a diameter of 0.01 m the discharging time reduced from 34 min to 1h15min for a diameter of 0.002 m, 2 h 05 min for a diameter of 0.03 m and about 3 h for a diameter 0.04 m.

4.2. Scenario 2: Variation of the porosity

The porosity is defined as the percentage of the quantity of existing fluid compared to the total volume of the reservoir. Fig. 6 shows the effect of porosity on the performance of the system, where different porosity values ranging from 0.25 to 0.50 m are tested. The results show that the heat transfer between the solid filler and the fluid is significantly influenced by the diameter of the solid filler. High values of the porosity improve the rate of heat transfer between the filler material and the heat transfer fluid, and this consequently leads to a reduction in the discharging time. I addition, this effect can be seen in Fig. 6, where the effect of the porosity on the stored thermal energy is reported. When the porosity of the storage unit is sufficiently small, the tank is slowly loaded.

Figs. 7 and 8 show the time varying of the temperature of the solid filler respectively in both charging and discharging periods. As the porosity of the tank increases, the temperature of the solid rises gradually and the charging and discharging time decreases. In addition, the time required for the temperature to drop from 380 °C to 340 °C

Table 1
Thermo-physical parameter of the thermocline tank.

Dimensions and thermo-physical parameter of the thermocline tank	
$H_{tank} = 7$ m	$R_{tank} = 4$ m
Fluid (Therminol® VP-1)	
$C_f = 2474.5$ J/kg K	$k_f = 0.086$ W/mK
$\mu_f = 1.810 - 4$ Pa s	$\rho_f = 753.75$ kg/m^3
$v_f = 0.004$ m/s	
Filler material (granite rock)	
$d_p = 0.04$ m	$K_s = 2.8$ W/mK
$\rho_s = 2630$ kg/m^3	$C_s = 775$ J/kgK

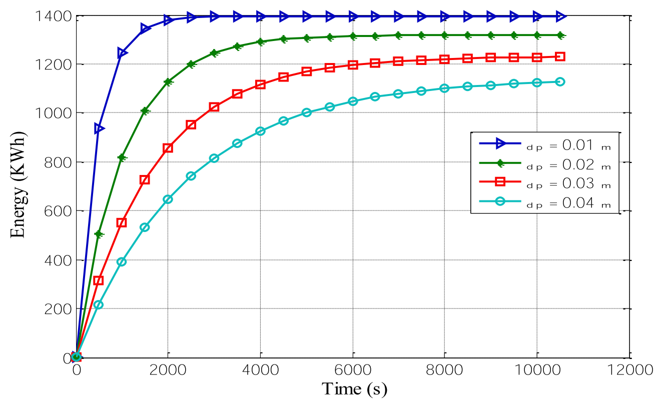


Fig. 3. Variation of the stored energy during the charging process.

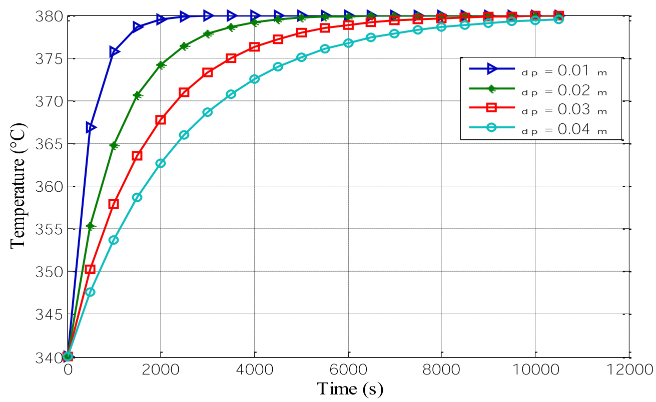


Fig. 4. Temperature variation during the charging process.

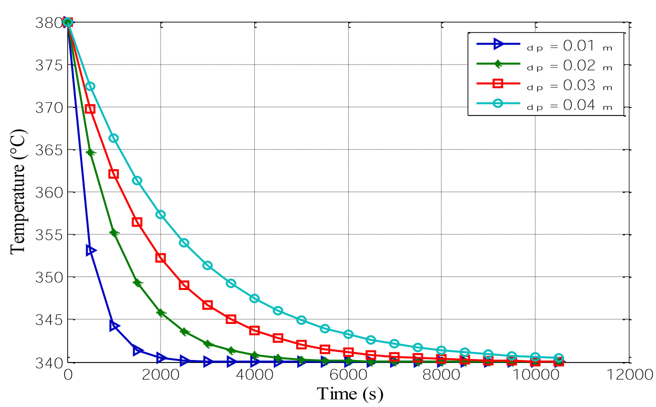


Fig. 5. Temperature variation during the discharging process.

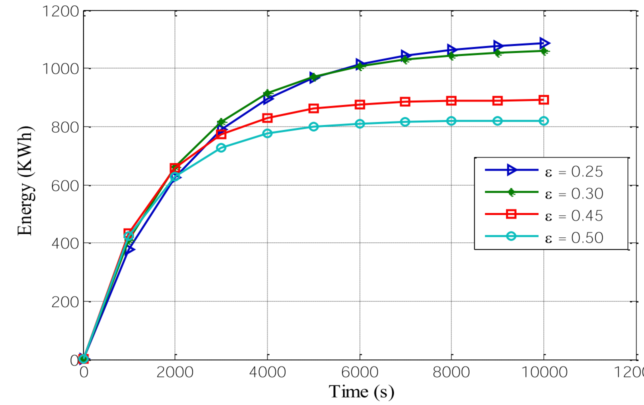


Fig. 6. Variation of stored energy during the charging process vs. porosity.

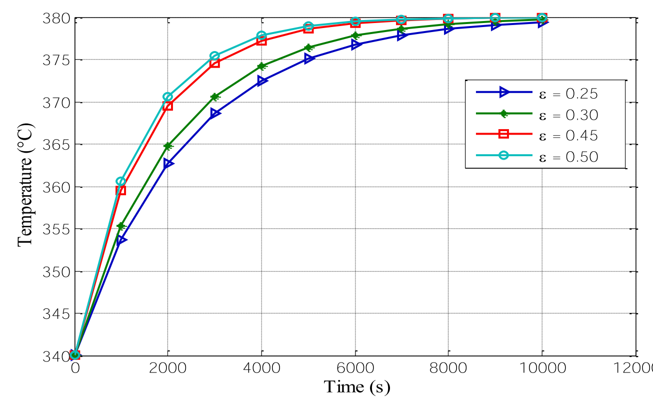


Fig. 7. Variation of temperature during the charging process vs. porosity.

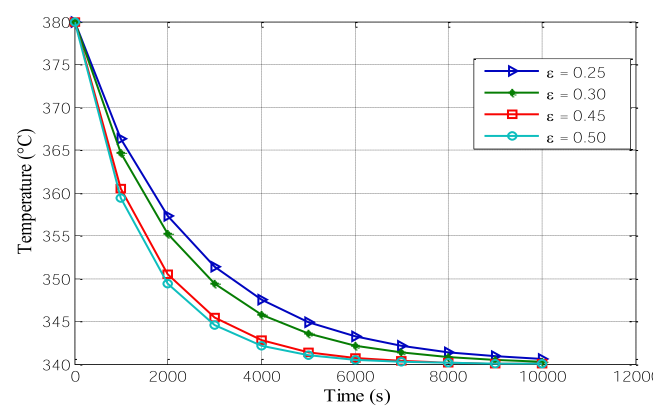


Fig. 8. Variation of temperature during the discharging process vs. porosity.

increases very slowly. With a diminution in porosity, the system can be evacuated slowly, and the discharge time decreases respectively by about 3 h for a porosity of 0.25, to 2 h 31 min for a porosity of 0.30, 1 h 50 min for a porosity of 0.45 and 1 h 41 min for a porosity from 0.50.

4.3. Scenario 3: Optimal porosity determination

The model proposed in this scenario is formulated as a constrained optimization problem to optimally design a rock-bed storage system. The main objective is to determine the optimal porosity, which allows obtaining the maximum energy. Considering the parameters chosen previously and mentioned in Table 1, considering a temperature ranged between 340 °C and 380 °C, a height of 7 m, a radius R of 4 m and a diameter of the solid filler equals to 0.04 m. The optimization results show an optimal value of the porosity equals to $\varepsilon = 0.22$, which maximizes the discharging energy. In addition, the proposed model can be used to select other variables affecting the optimal design of the thermocline thermal storage system.

4.4. Scenario 4: Comparative study

The first objective of this scenario is to investigate the effect of diameter of the solid on the selection process of suitable material given best performance in terms of stored energy. The operation of the thermocline storage is simulated considering two different materials, the first one is the concrete HT that is defined by its thermophysical characteristic ($\rho = 2750[\text{kg/m}^3]$, $C_p = 619[\text{J/kg. K}]$), while the second one is the granite rock defined by ($\rho = 2630[\text{kg/m}^3]$, $C_p = 775[\text{J/kg. K}]$). Furthermore, we assumed temperatures ranged between 340 and 380 °C, as well as a fixed value for the porosity equals to 0.25. It can be seen from Fig. 9, that by varying the diameter the energy stored changes considerably. For small diameter, the two materials show the same behavior, while for high diameter, concrete HT demonstrates higher storage capacity than granite.

The second objective aims to study the influence of the porosity variation on the storage capacity of the two materials considering the same conditions. The results demonstrate that the concrete shows higher storage capacity than the granite for a porosity of 0.25 as reported in Fig. 10 (a). While, the granite shows a bit higher storage capacity than the concrete HT considering a porosity equals to 0.15 as illustrated in Fig. 10 (b). This proves that deep investigation should be considered to determine optimal values of the porosity and the solid filler diameters in selecting appropriate materials maximizing the storage capacity.

The stored energy as well as the temperature variation are reported in Fig. 11. It can be observed from the figure an increased trend of temperature and stored energy for different average velocities. During one hour of charging, for instance, an increase is observed from 355 °C to 370 °C and from 576 kW h to 1150 kW h for velocities ranging from 0.001 to 0.004 m/s.

The energy utilization may be described as a ratio among the actual stored energy and the maximum stored energy defined as $Q_{max} = \rho_s c_{ps}.\ V(T_{s,int} - T_{e,end})$ in the rock bed storage system. Fig. 12 reports the evolution of the energy utilization according to various average velocities, with a porosity of 0.25 and a rock size equals to 0.04 m. Results show that during one hour of charging, the efficiency of the storage system increases from 38.5% to 68% when the fluid average velocity varies from 0.001 to 0.003 m/s. While, it increases from 68% to 77% for average velocities ranging from 0.003 to 0.004 m/s. This is mainly due to the dominance of convective thermal resistance for low velocities and conductive thermal resistance for high average velocities.

5. Conclusion

The integration of thermal energy storage systems in concentrating solar powers will enhance their performance and minimize the effect of solar irradiation intermittencies and availability. These thermal energy storages can serve to store excess of thermal energy during the day and can be considered as a source of thermal energy at nighttime. Furthermore, they may play a key role in ensuring the power balance, stabilizing the power supply and coping with fluctuation of the solar irradiation. This paper aims to study the effect of the diameter of the solid filler and the porosity of the heat transfer fluid on the selection process of suitable material given best performance in terms of storage capacity. The main objective is to determine optimal values of the porosity and the solid filler diameter of appropriate materials maximizing the storage capacity. A comparative study is considered, where the operation of the thermocline storage is simulated taking into account two different materials. The optimization results show an optimal value of the porosity equals to $\varepsilon = 0.22$, which maximizes the discharging energy. The results demonstrate that the concrete shows higher storage capacity that the granite for a porosity of 0.25, while, the granite shows a bit higher storage capacity than the concrete HT considering a porosity equals to 0.15.

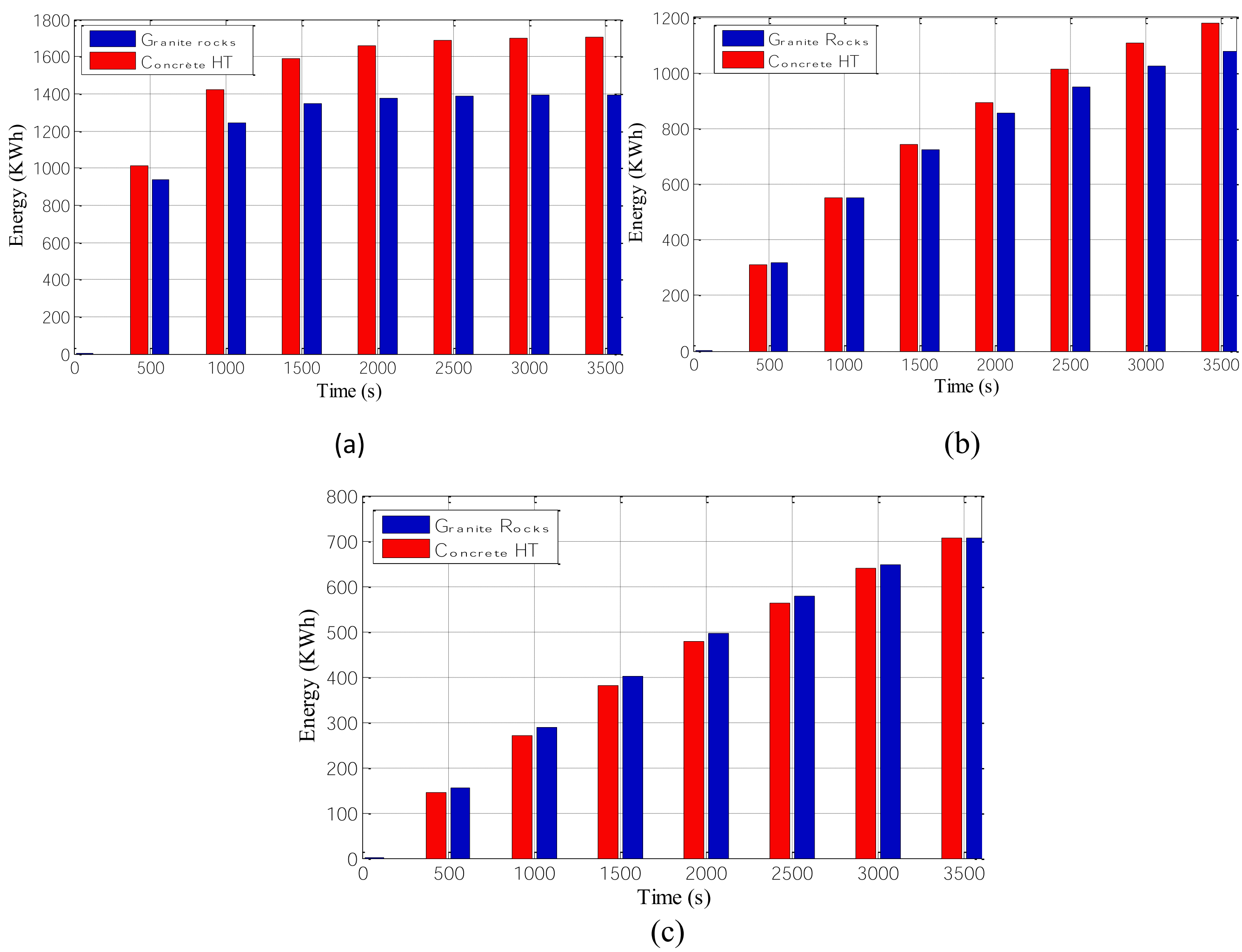
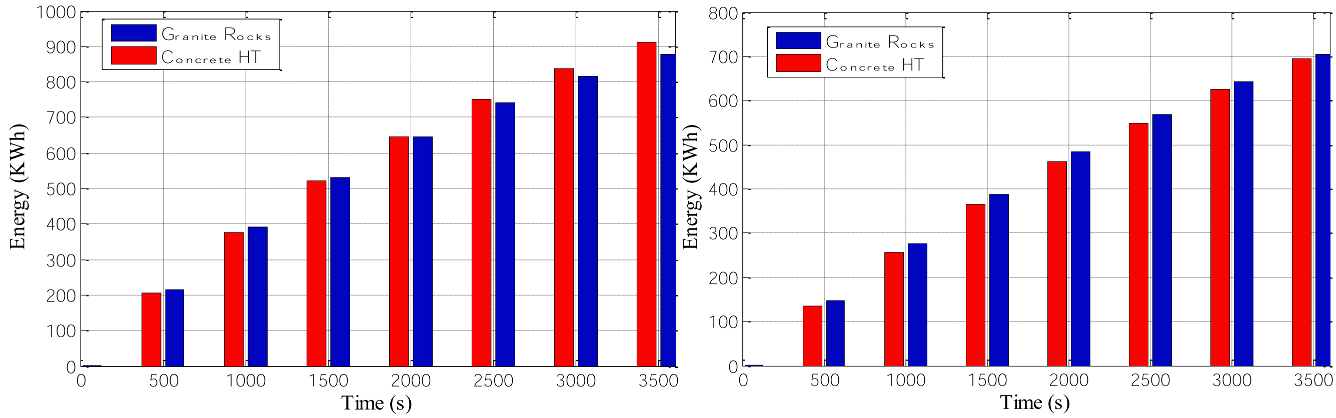


Fig. 9. Storage capacity considering $\varepsilon = 0.25$, (a) $dp = 0.01$ m, (b) $dp = 0.03$ m and (c) $dp = 0.05$ m.



(a) (b)

Fig. 10. Storage capacity considering: (a) $\varepsilon = 0.25$, (b)$\varepsilon = 0.15$ and$dp = 0.04$m.

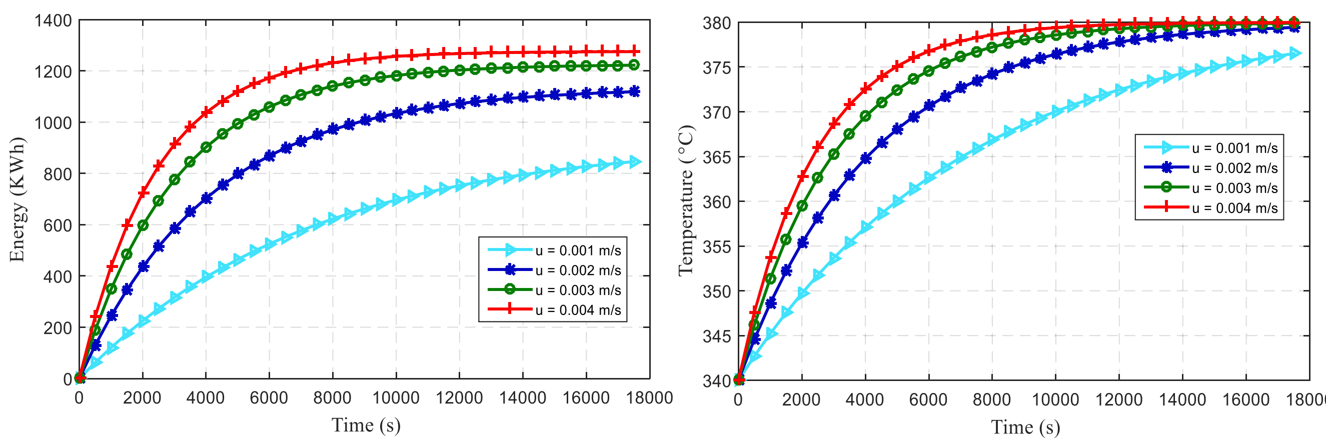


Fig. 11. Stored energy and temperation evolution for various velocities.

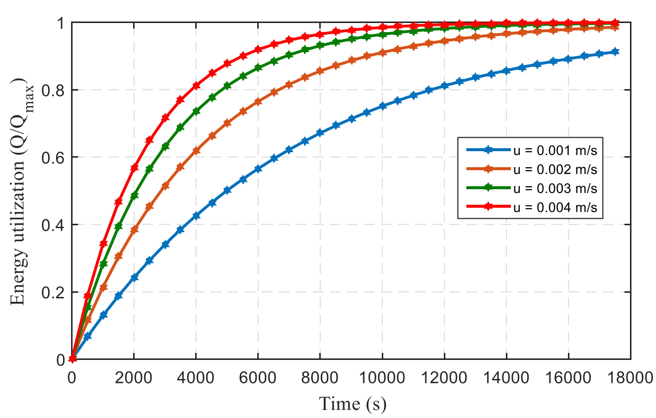


Fig. 12. Energy stored for various velocities.

CRediT authorship contribution statement

Samir Berrhazi: Software, Methodology, Visualization, Writing - original draft, Writing - review & editing. **Ahmed Ouammi:** Software, Methodology, Supervision, Visualization, Writing - original draft, Writing - review & editing. **Rachid Benchrifa:** Supervision, Writing - review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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