Numerical investigation on sensible thermal energy storage with porous media for high temperature solar systems

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Abstract. In this paper different high temperature TES components are numerically analyzed. The difference is defined by the different type of porous medium employed in the storage. Two different porous media are considered: spheres or foams. In all cases a ceramic material is considered. In the formulation of the model it is assumed that the system geometry is cylindrical, the fluid and the solid thermophysical properties are temperature independents, the radiation heat transfer mechanism is taken into account. The commercial CFD Fluent code is used to solve the governing equations in transient regime and in local thermal non-equilibrium (LTNE). Numerical simulations are carried out at different mass flow rates of the heat-carrying fluid. The results show the effects of the porosity and of the working fluid mass flow rate on the stored thermal energy and on the storage time.

1. Introduction

Energy conservation and management are needed in several industrial and commercial applications in order to supply thermal energy. Various devices are employed to satisfy the energy demand in commercial, industrial and utility sectors which can vary on daily, weekly and seasonal bases. The main devices to realize the energy conservation are the thermal energy storage (TES) systems and they allow to align energy production with consumer demand. The use of TES for thermal applications, such as space and water heating, cooling, air-conditioning, has recently received much attention [1-7]. The use of TES in concentrated solar power (CSP) technology is very important to deliver high-temperature heat in the form of sensible heat storage in a packed bed of rocks or other ceramic materials and it is especially suitable when air is used as the heat transfer fluid (HTF) in the solar receiver [8-12].

Packed beds represent the most suitable storage units for air-based solar system. A packed bed storage system consists of loosely packed solid material through which the heat transport fluid is circulated. Heated fluid (usually air) flows from solar collectors into a bed of graded particles from top to bottom in which thermal energy is transferred during the charging phase. Coutier and Farber [13] mentioned...
that packed bed generally represents the most suitable energy storage unit for air based solar systems. During the charging mode, solar heated air is forced into the top of the container, i.e. upper plenum and then passes evenly down through the bed heating the storage and passes out through the lower plenum. Air is drawn off at the bottom and returned to the collectors. When energy is needed from storage, the airflow is reversed. However, some other type of porous media such as ceramic foams or honeycomb could be employed as material in the storage unit to realize a different storage system with lower thermal capacity and pressure drop. Several studies describe numerical models for sensible heat storage in porous media [12].

The heat transfer to and from a flowing fluid to a packed bed has been the subject of many theoretical and experimental investigations since Schumann’s original work [14]. One-dimensional two-phase model for packed bed system was assumed by ignoring the thermal capacity of the fluid, axial conduction in the fluid and axial conduction in the bed material. Some elaborations were performed in [15-17] to facilitate the extraction of numerical information from the solutions by providing nomograms, extensive graphs and tabulations. Numerical simulations were carried out to solve the governing equations for the packed bed by finite difference methods [18-20]. A mathematical model to evaluate the dynamic response of a packed column to an arbitrary time dependent inlet air temperature was accomplished Saez and McCoy [21]. Different aspects of sensible heat storage systems were analyzed by Dincer et al. [22]. They reported that the selection of sensible heat storage system depends upon the storage period, economical viability and operating conditions. A study on different energy storage techniques and materials used in sensible heat storage systems was presented in [23]. A comparative numerical investigation on packed bed thermal models suitable for sensible and latent heat thermal storage systems was reported by Ismail and Stuginsky [24]. A method of preserving the stratification by segmenting the storage bed was numerically studied in [25]. An experimental investigation on heat transfer and pressure drop characteristics of packed bed solar energy storage system with large sized elements of storage material was presented in [26]. Correlations were developed for Nusselt number and friction factor as function of Reynolds number, sphericity and void fraction. An extensive literature review of research work on packed bed systems was presented [27]. The effect of multiple charge and discharge cycles was studied in detail in [28]. An high temperature TES was numerically parametrically analyzed by using CFD code to solve the governing equation in porous media in transient regime [29]. High temperature TES in a packed bed of rocks was studied by Hanchen et al. [13] for air-based concentrated solar power plants. A 1-D porous medium model in local thermal non equilibrium was assumed and two-phase energy conservation equations for combined convection and conduction heat transfer were solved numerically for charging/discharging cycles. A comparison between numerical results and their experimental data was accomplished for a packed bed of crushed steatite (magnesium silicate rock) at 800 K.

In this paper different high temperature TES components are numerically analyzed. The difference is defined by the different type of porous medium employed in the storage. Two different porous media are considered: spheres or foams. In all cases a ceramic material is considered. The commercial CFD Fluent code is used to solve the governing equations in transient regime and in local thermal non-equilibrium (LTNE). Numerical simulations are carried out at different mass flow rates of the heat-carrying fluid. The results show the effects of the porosity and of the working fluid mass flow rate on the stored thermal energy and on the storage time.

2. Mathematical Description
The physical system and geometry under investigation are shown in Fig. 1. It consists of a cylinder whose diameter is equal to 0.60 m and height is 1 m. The difference is defined by the different type of porous medium employed in the storage. Two different porous media are considered: spheres or foams. In all cases a ceramic material is considered. Material storage is made of alumina for ceramic foam, while for the package bed configuration with spheres is employed cordierite. The heat-carrying
fluid is air. The radiation heat transfer mechanism is taken into account for some configuration. Heat losses with the external environment, T\textsubscript{amb} equal to 300 K, are considered by setting the surface heat transfer coefficient equal to 5 W/m\(^2\)K. A charging-discharging cycle is studied. In the charging phase air enters at 1473 K whereas in discharging phase air enters counter-current at 1073 K. The study is focused on the first charge of the thermal storage. The charge, in the cycle, starts with the heat storage system temperature set at 300 K.

![Figure 1. Geometric configuration and physical domain in charging phase](Image)

In the porous medium region, the generalized flow model, known as the Brinkman-Forchheimer-extended Darcy model, is used in the governing equations and the local thermal non equilibrium is assumed. The equations for mass, momentum and energy are, in the storage region:

\[
\frac{1}{r} \frac{\partial}{\partial r} (rv_p) + \frac{\partial u_p}{\partial z} = 0
\]

(1)

\[
\frac{\partial \rho_f v_p}{\partial t} + \rho_f \frac{\partial v_p}{\partial r} + \rho_f u_p \frac{\partial v_p}{\partial z} = -\frac{\partial p}{\partial z} + \nu \left[ \frac{\partial^2 v_p}{\partial r^2} + \frac{1}{r} \frac{\partial v_p}{\partial r} - \frac{v_p}{r^2} + \frac{\partial^2 v_p}{\partial z^2} \right] - \frac{\nu}{K} v_p + \frac{C_s^2 \rho_f}{2} \left( u_p^2 + v_p^2 \right)
\]

(2)

\[
\frac{\partial \rho_f u_p}{\partial t} + \rho_f \frac{\partial u_p}{\partial r} + \rho_f u_p \frac{\partial u_p}{\partial z} = -\frac{\partial p}{\partial r} + \nu \left[ \frac{\partial^2 u_p}{\partial r^2} + \frac{1}{r} \frac{\partial u_p}{\partial r} + \frac{\partial^2 u_p}{\partial z^2} \right] - \frac{\nu}{K} u_p + \frac{C_s^2 \rho_f}{2} \left( u_p^2 + v_p^2 \right) - \varepsilon \frac{C_s^2 \rho_f}{2} \frac{u_p^2 + v_p^2}{u_p} \frac{v_p}{u_p}
\]

(3)

\[
\varepsilon (\rho c_p)_f \frac{\partial T_f}{\partial t} + \varepsilon (\rho c_p)_f \left( v_p \frac{\partial T_f}{\partial r} + u_p \frac{\partial T_f}{\partial z} \right) = \varepsilon k_f \left[ \frac{\partial^2 T_f}{\partial r^2} + \frac{1}{r} \frac{\partial T_f}{\partial r} + \frac{\partial^2 T_f}{\partial z^2} \right] - h_f a_f (T_s - T_f)
\]

(4)

\[
(1-\varepsilon)(\rho c_s) \frac{\partial T_s}{\partial t} = (1-\varepsilon) k_s \left[ \frac{\partial^2 T_s}{\partial r^2} + \frac{1}{r} \frac{\partial T_s}{\partial r} + \frac{\partial^2 T_s}{\partial z^2} \right] + h_f a_f (T_s - T_f)
\]

(5)

Where C\(_s^2\) = 2CK\(^{0.5}\).

The permeability K and inertia coefficient C of porous medium, in packed bed configuration are based on two relations given in [30]. The term h\(_{sf}\)a\(_f\) is present due to the local thermal non-equilibrium assumption and it is related with the local convective heat transfer inside the porous medium between the fluid and the solid porous matrix surfaces. Convective heat transfer coefficient and interface area
per volume of packed bed are evaluated by means of the relations given in [31,32]. For ceramic foams permeability, inertia coefficient, convective heat transfer coefficient and interface area per volume of foams are evaluated by relations given in [33,34]. The radiative heat transfer was taken into account in the case of ceramic foams considering an effective thermal conductivity for the porous matrix which takes into account both heat conduction and thermal radiation as given in [35].

3. Numerical model
For packed bed scheme the boundary condition are the following: at the inlet, fluid phase temperature is equal to 1473K in the charging phase or 1073 K in the discharging phase, and three several mass flow, G, are studied, equal respectively, 0.1, 0.2, and 0.3 kg/m$^2$s. Five different porosity values are analyzed in ranging from 0.2 to 0.60, the radiation heat transfer mechanism and thermal energy losses toward external ambient are neglected. For configuration with Alumina ceramic foam, at inlet section the thermal boundary condition are the same that the configuration depicted previously, moreover three several mass flow rate are analyzed equal to 0.05, 0.1 and 0.2 kg/s., at external wall of the cylinder a surface heat transfer coefficient is assumed to 5 W/m$^2$K. the porosity of alumina is equal to 0.858, corresponding to 20 PPI. At channel exit an outflow condition is imposed on the fluid phase, while an adiabatic condition is imposed on the solid phase. A 2D-axialsymmetric option is also enabled to simulate the phenomenon under investigation.

The commercial CFD code Fluent is employed to solve the governing equations. The solid and fluid thermophysical properties are temperature independents. The SIMPLE scheme is chosen to couple pressure and velocity. The porous medium model is active in the porous region and the nonlocal thermal equilibrium model is employed by using an UDF code. In this UDF code the term $h_{sf,a}$ is evaluated taking into account the relations for packed bed, given in [31,32], and for ceramic foams, given in [33,34]. The convergence criteria of $10^{-6}$ for the residual of continuity equation and velocity components and $10^{-8}$ for the residuals of energy are assumed.

A grid dependence test is accomplished to realize the most convenient grid size by monitoring the percentage error among three different grids tested, 40x120, 80x240 and 160x480. Volumetric fluid average temperature is monitored, in the changing phase in the first cycle with m=0.05 kg/s. The maximum variations of the average temperature values when the number of the nodes are 80x240 with respect to the value, obtained with the finest grid, 160x80, was 0.3 %. The mesh 80x240 is employed in this investigation because it ensures a good compromise between computational time and accuracy requirements.

4. Results and Discussion
For packed bed configuration, the results are presented in terms of stored energy profiles for cordierite spheres. Five porosity values are analyzed in ranging from 0.2 and 0.6, mass flow per unit cross section, G, inlet has values equal 0.1, 0.2, and 0.3 kg/m$^2$s for all cases.

In Figure 2, energy stored is reported as a function of time, for $\varepsilon$ values in the range from 0.2 to 0.6 and G equal to 0.1 kg/ m$^2$s, for charge and discharge cycles. Decreasing the porosity the charging and discharging time increases due to the thermal capacity increase. Increasing the porosity value steady state conditions are reached at lower time both in charging and discharging phases.

In Figures 3 and 4, stored thermal energy as a function of time is depicted for mass flow per unit cross section equal to 0.2 and 0.3. It is noted that the steady state is attained at lower time value due to a more efficient convective heat transfer between the fluid and the solid matrix. However, increasing the mass flow rate the energy level, in the case with adiabatic external surface of the storage, any variation in the energy stored values is detected. In this case, the main result is that charging and discharging times decrease as the mass flow rate increases.

Also for ceramic foam configuration, the results are presented in terms of energy stored profiles.
Figure 2. Energy stored for several porosity values and $G$ equal to 0.10 kg/m$^2$s a) charge cycle, b) discharge cycle

Figure 3. Energy stored for several porosity values and $G$ equal to 0.20 kg/m$^2$s a) charge cycle, b) discharge cycle

Figure 4. Energy stored for several porosity values and $G$ equal to 0.30 kg/m$^2$s a) charge cycle, b) discharge cycle
Alumina foam with porosity equal to 0.858 and 20 PPI. For charge phase the initial temperature of solid is equal to 300 K while at inlet the temperature is equal to 1473 K. In the discharge phase the air has the inlet section opposite at the one in the charge phase and the inlet temperature is 1073. In this configuration also the effect of heat transfer losses is analyzed. For all studied configurations, the heat transfer coefficient on the external surface of cylinder is assumed equal to 5 W/m$^2$K. Three different mass flow rate are analyzed and equal to 0.050, 0.10 and 0.20 kg/s. For mass flow rate equal to 0.20 kg/s also the case with radiative heat transfer is analyzed.

Figure 5 shows stored energy profiles as a function of time for adiabatic and with heat losses cases. At mass flow per unit cross section increases charging and discharging time decrease. Steady state conditions are reached in any case. It is noticed that for the cases with heat transfer losses with a external heat transfer coefficient equal to 5.0 W/m$^2$K, the energy stored value is less than the adiabatic ones.

In Figure 6 is showed the effect of radiation heat transfer mechanism. The energy values stored are the same for both configuration studied, but for configuration with radiation heat transfer charging and discharging time decreases.

Figure 6. Energy stored for several mass flow rate values and $\varepsilon$ equal to 0.858 with and without radiative heat transfer.
5. Conclusions
The stored energy and the charging and discharging times related to the first cycle of a high
temperature thermal energy storage are numerically evaluated for different storage materials. The
difference is defined by the different type of porous medium employed in the storage. Results showed
that both for packed bed and ceramic foams the charging and discharging times decrease increasing
the mass flow rate. The heat losses and the presence of radiative heat transfer in porous medium
determined an increase in charge and discharging times.

6. Nomenclature

\( a_{sf} \) specific surface area \( m^{-1} \)

\( C \) inertia coefficient

\( c \) solid specific heat \( Jkg^{-1}K^{-1} \)

\( D \) cylinder diameter \( m \)

\( h \) surface heat transfer coefficient \( Wm^{-2}K^{-1} \)

\( h_{sf} \) interfacial heat transfer coefficient between \( Wm^{-1}K^{-1} \)

solid matrix and fluid

\( K \) permeability \( m^{2} \)

\( k \) thermal conductivity \( Wm^{-1}K^{-1} \)

\( L \) cylinder height \( m \)

\( m \) mass flow rate \( kg/s \)

\( p \) pressure \( Pa \)

\( q \) heat flux \( Wm^{-2} \)

\( r, z \) cylindrical coordinates \( m \)

\( T \) Temperature \( K \)

\( u, v \) horizontal and vertical velocity components \( ms^{-1} \)

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