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Energy Procedia 00 (2014) 000–000

Energy Procedia

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The 6th International Conference on Applied Energy – ICAE2014

Modelling of heat exchangers based on thermochemical material for solar heat storage systems

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Abstract

In solar thermal energy storage systems the operation modes involve charging and discharging. This paper focuses only on the charging leading to an endothermic reaction and therefore an efficient heat exchanger is required to transfer the heat for fast and complete charging. Two different heat exchangers are studied in this paper. A plate fin and helical coil heat exchangers embedded in a magnesium chloride bed is modelled and solved using the software Comsol 4.3a based on finite element method. Meshing analysis is performed for parameters sensibility and the results show a temperature variation of 13 °C (helical coil) and 19 °C (plate fin) in the material bed during the charging mode of the thermochemical heat storage system. The pressure distribution in the heat transfer fluid and the temperature distribution in the material bed are presented and the calculated overall heat transfer coefficient of 173 $W/m^2\cdot K$ (helical coil) and 236 $W/m^2\cdot K$ (plate-fin) are obtained on the base of the total heat transferred (Q) to through the system. The fluid flow is in turbulence regime (Re = 13200) in the fin-plate, but in laminar mode (can be kept up to Re = 20000) [1] in the coil because the flow is affected by secondary flow cause by centrifugal forces. This study allows the choice of the heat exchanger wherein with first experiment has been made and compared.

Keywords: Heat exchanger; Fin plate; Helical coiled tube; Thermochemical material; Heat transfer coefficient.

Nomenclature

- c_p specific heat capacity (kJ/kg)
- μ dynamic viscosity, kg/(m·s)
- ρ density (Kg/m^3)
- ΔT_1 Inlet temperature difference between fluid-solid (K)
- ΔT_2 Oulet temperature difference between fluid-solid (K)
- q Heat Transfer rate or heat flux (W)

1. Introduction

A decade ago, it has been shown that helical coil and plate fin heat exchanger increases the heat transfer coefficient and the temperature rise of fluid for helically coiled depends on the tube geometry and the flow rate [2]. So, to develop a long or short term solar energy storage system using solid-gas reaction system, just like the development of thermal energy storage systems, heat exchanger is an absolute requirement in close process. In that field of thermal energy storage, various research projects [3]–[5] where different heat exchangers were used, recommend that heat exchanger

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should be as compact as possible. Compact, because of large area density which leads to a high heat transfer surface per volume. For compact heat exchangers (CHEs) the ratio of heat transfer surface to heat exchange volume, is determined to be over 700 m^2/m^3. This large area density indicates small hydraulic diameter for fluid flow [4] and lead to high heat transfer coefficient.

A good heat exchanger performance includes small temperature differences across the heat exchanger to maximize the heat transfer coefficient. The heat exchanger therefore has to be designed in a compact way (heat exchange surface-to-mass ration, small volume, allowing good vapour transport), but also has to work with a high heat transfer coefficient [4]. That is why we focus on these heat exchangers. Coil tubes and plate fins are feasible passive enhancement methods of compact heat exchanger. The authors have not identified an experimental work or detailed modelling using a helical coil tube as heat exchanger in gas-solid sorption storage process, except in condenser/evaporator and hydrogen storage applications where interaction is involved [6]. However plate fins are quite broadly used due to their large heat transfer surface area [4], [7], [8]. This work focuses on charging phase of a low thermal energy storage systen with a thermochemical material ($MgCl_2 \cdot 6H_2O$) in order to evaluate the performance of parameters such as temperature, pressure and heat transfer coefficient. A three-dimensional model in Comsol Multiphysics 4.3a has been developed to simulate the charging of the storage bed using the two different heat exchangers and comparison is made in term of thermal performance in order to choose the appropriate heat exchanger. The magnesium chloride properties used in this work is based on the values reported by Rammelberg et al [9].

2. Heat Exchangers Design and Comsol Model

In this type of heat exchanger design, the shell consists of a cylinder and a helical coiled tube inserted inside (Fig. 1). The selection criteria for a proper combination of components are dependent upon the operating pressures, temperatures, and thermal stresses, corrosion characteristics of fluids, fouling, cleanability, and cost. Since the desired heat transfer in the exchanger takes place across the tube/fin surface, the selection of tube/fin geometrical variables (coil pitch, coil diameter, fin dimensions, tube flow rate, etc.) is important from the performance point of view [6], [10]. Therefore, according to our low temperature (90°c – 120°C) application and the field of thermochemical energy storage, we define the design parameters mentioned in nomenclature.

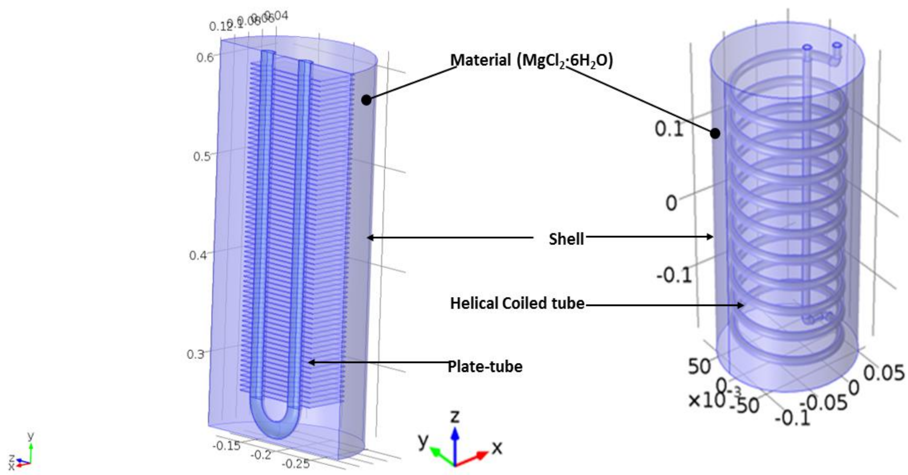

Fig. 1. 3D-schematic view of (a) fin plate heat exchangers and (b) shell-helical coiled tube for solar heat storage.

3. Energy balance and heat transfer coefficient

The heat from the heat transfer fluid (HTF) is supplied to the material bed via conduction (convection is neglected since the conducted wall is a metal) and leaves the latter via advection (water vapor) after the phase change of water (inside the material) from liquid to gas. The stationary energy balance in the bed can be written by the following equation:

$$C_p \cdot \rho \boldsymbol{u_f} \cdot \nabla T = \nabla \cdot \left(\lambda_{meff} \nabla T\right) + q_a \quad (1)$$

where $C_p \cdot \rho$ is the specific volumetric heat capacity of the solid material, λ_{meff} (W/m·K) the mean effective thermal conductivity of the bed, q_a the source term for the energy generated in the bed and $\boldsymbol{u_f}$ the fluid velocity. In this study, the bed is considered as being immobile, bed pressure constant. The radiative heat transfer and the work done by pressure changes are not taken into account here because of the minor effect under vacuum and relatively low temperature. Also thermal resistance of walls is neglected due to the fact that model idealize the contact heat exchanger/material bed. Based on finite element method, Comsol Multiphysics was applied to calculate the overall transfer coefficient, U_0 (W/m^2·K). It is calculated from the temperature data and the total heat transferred during the charging (Equation 2).

$$U_0 = q/(A_0 \Delta T_{LM}) \tag{2}$$

where A_0 (m^2)is the outside surface area of coiled tube or plate fin, q is the heat transfer rate and ΔT_{LM} is the log mean temperature difference, based on inlet temperature difference, ΔT_1, and the outlet temperature difference, ΔT_2, using the following equation:

$$\Delta T_{LM} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) \tag{3}$$

In this case of fluid-solid heat transfer, which differs from co-flow and counter flow, the used temperature in the solid is an average at different positions of the solid.

4. Results and discussion

The material bed initially at 25 °C is heated up to 120 °C with uniform temperature distribution in the both case in the simulation, see Fig. 4b. This variation of about 95 °C is in agreement with the temperature need to charge the heat storage system and which can be supplied by a solar collector. The system with a plate fin heat exchanger (right picture in the Fig. 3) exhibits a higher temperature variation (19 °C) than one with the helical coil heat exchanger (13 °C). The plate fin with 50 plates has a bigger heat transfer surface than of the helical coil. The reason of temperature difference can be that the flow is affected by secondary flow cause by centrifugal forces in the helical coil. The overall heat transfer coefficient of 236 W/m^2·K for plate fin heat exchanger and of 173 W/m^2·K for helical coil tube heat exchanger is obtained, showing why we should use the plate fin instead of helical coil heat exchanger in the reactor.
A comparison of pressure drop for the two exchangers calculated with Comsol Multiphysics based on fluid velocity is presented in the Fig. 3 (below). The pressure drop in fin-plate heat exchanger is as low as hundredth the result in coil heat exchanger and this favors our low temperature application. Although the fin-plate heat exchangers are mostly subjected to fouling issue, it can be seen that compared to coil heat exchanger in our application fin-plate is the most appropriate, even if with more thermal mass, the higher heat transfer surface dominates.

5. Validation attempt with first experiment

Using the heat transfer coefficient, fluid pressure drop and temperature distribution in the bed as thermal performance parameters, we have compared two different heat exchangers: the fin-plate and helical coil. It result that for our application in gas-solid solar heat storage, the fin-plate will be used and then compared to experimental results.

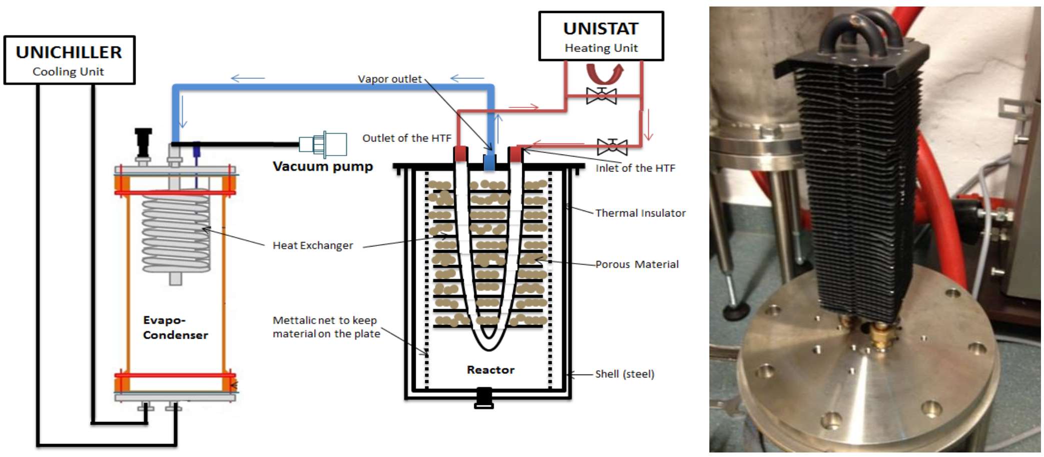

Fig. 2. Lab-scale design for first experimentation and the used fin-plate heat exchanger in the solar heat storage system

After the choice was made based on numerical result, the fin plate heat exchanger was ordered and introduced in the cylindrical reactor (Fig. 2). The experimental principle consists of charging and discharging the thermochemical material bed via fin-plate heat exchanger, but we focus here only on charging mode. In the charging, desorption heat comes from the Unistat, representing here a solar panel. The idea is to use heat from the solar panel to heat the bed via heat exchanger. A by-pass is used in order to obtain the required charging temperature of the fluid for decomposition, before opening the valve at the inlet of the tube-plate heat exchanger and the sufficient heat provided by the heating fluid (here the ethylene glycol) is then transported in the bed through heat exchanger. The Unichiller cooled the outlet vapour from the reactor into liquid water, later use for discharging phase.

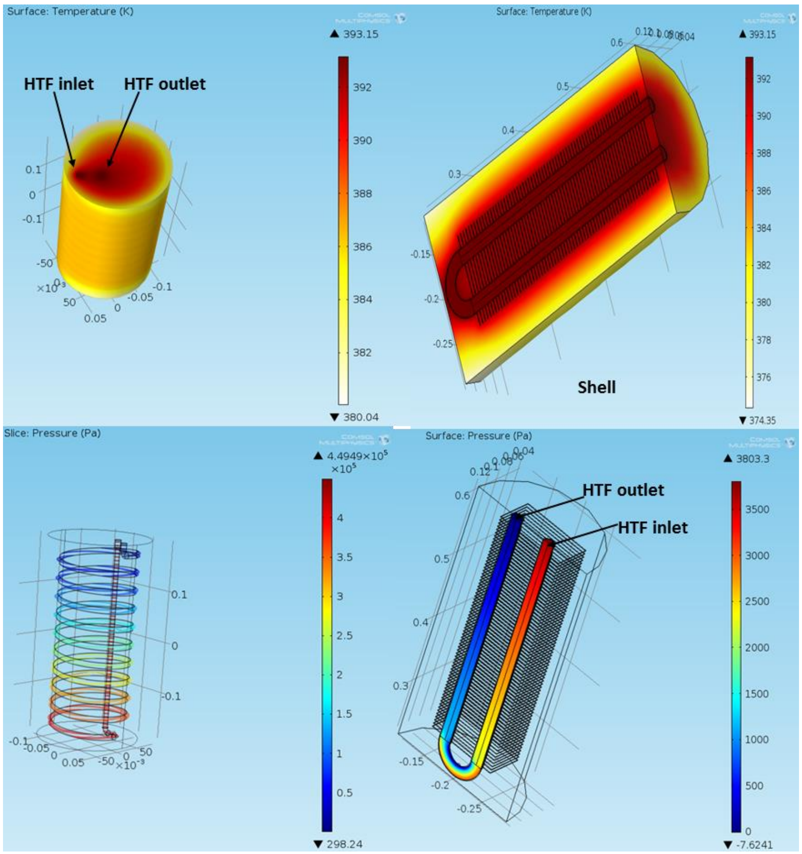

Fig. 3. Temperature distribution over the material bed, and pressure drop with helical coil heat exchanger (left) and plate fin heat exchanger (right).

Let remember that in experiment, power are picked up manually since there is not yet an adapted data logger. This can consequently leads to a systematic error which has not been taking into account. The experimental power is read out on the Unistat, which correspond when calculating with fluid heat capacity, mass flow and fluid temperature difference. The peak power in the experiment and simulation are 655 W and 2500 W respectively. The latter value lies on the given heat source for simulation (equation 1), depending on the convective heat transfer, as it was difficult to evaluate it before experiment. Using the log mean temperature difference and the same heat transfer surface (simulation design reproduce the exactly 50 plates with same dimension as in experiment), the experimental overall heat transfer coefficient is 99 W/m^2.K. It obvious, that the major difference in Fig. 4 concern temperature and the power supply for the charging.

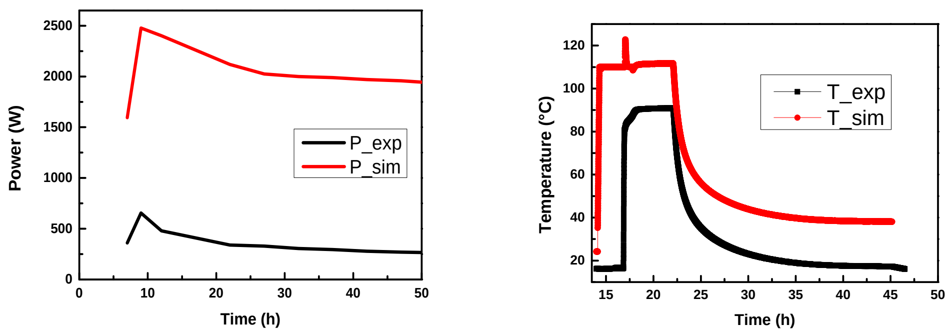

Fig. 4. (a) Power supply for charging phase (b) Temperature evolution during the charging phase of the heat storage system

6. Conclusion

This work mainly on heat exchanger modelling shows some validation attempts with first lab-scale experiment. Although the discrepancy caused by the overestimation of the numerical value, we can clearly see that the mechanisms are kept. The Figure 4b reveals that even a normal solar panel can afford the charge with magnesium chloride as thermochemical material, though the simulation value of about 120 °C is also possible. This can be explained as more power supplied, more temperature increased. Experimentally the 90 °C obtained do not charge completely the material according to the decomposition [11], that leads to first agglomeration problem during first discharging, making the heat exchanger inefficient for vapor transport into the bed. Further works are ongoing to solve this issue. The deviation between real and simulated power open our mind on the fact that, contact layer between the heat exchanger and the material bed is not perfect as in the numerical model. We can see that numerical solution shows that experimental optimization can be made in order to have full charging and therefore good discharging.

Acknowledgements

This work was performed under research project innovation-inkubator / competence tandem “thermische batterie”. The authors would like to acknowledge the support of the EU-foerdert niedersachsen (EFRE), the Leuphana University of Lueneburg to have financed this project.

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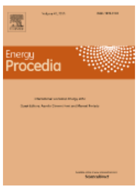

Biography

Armand Fopah Lele is graduated from the University of Yaoundé 1 in Cameroon (Physics-Materials Science) and the International Institute for Water, Energy and Environmental Engineering in Burkina Faso (Environment). He is actually research associate and doctorate student in the project Inkubator-thermal battery at the Leuphana University of Lüneburg in Germany. He is focusing on heat and mass transfer coupled to chemical reaction, material characterization, lab-scale experiments and modelling.