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Modelling Of an Encapsulated Molten Salt Thermal Storage System

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Abstract
Thermal storage is both a strong advantage for solar thermal power plants and a component with high improvement potential. The research institute solar energy and new energies (IRESEN) is currently developing a pilot solar power plant using an Organic Rankine cycle in the framework of the recently launched “GREEN ENERGY PARK” research platform, this plant shall operate at relatively low temperatures in comparison with large scale commercial plants, and then shall require a well suited storage system in order to improve economic efficiency and enable night time production. A study based on the cost and storage capacity has led to the choice of two systems: sensible heat storage in a rock (quartzite) packed bed and latent heat storage on encapsulated molten salt. A model has subsequently been developed in order to enable the assessment of both systems’ performances and in particular for the encapsulated salt solution for which a hexagonal close packing with spherical capsules was selected taking inspiration from hexagonal crystal structures. This model calculates the heat exchange between the heat transfer fluid HTF and the storage medium by computing the dimensionless numbers and HTF flow on each horizontal plane following flow direction and for a whole discharge duration. The results enable to tell whether the output of the encapsulated salt system is high enough to justify its cost. It also allows choosing the best design, sizing and the best configuration to optimize the systems’ output to the maximum.
Keywords: thermal storage, latent heat, molten salt, packed bed, Organic Rankine cycle, hexagonal close packing

I. INTRODUCTION

Morocco is engaged since 2008 in a highly ambitious renewable energy policy, this led to the establishment of the Moroccan solar plan which aims the achievement of a 2000 MW capacity by 2020. To support this solar program bytightening the link between academics and industrials and reinforcing their presence and participation on research and innovation in this field, IRESEN was created in 2011 in the frame of the ministry of energy and mines as a research institute and a funding agency for universities and industries.The main role of IRESEN as a research center is to procure research facilities for Moroccan researchers, this task is now being achieved through the construction of the GREEN ENERGY PARK platform in the New City Mohammed VI – Benguerir.This new research facility shall include many laboratories and demonstration projects for both photovoltaic and thermal solar technologies.

The main and most innovative demonstration project to be installed on the new platform is a pilot CSP plant using an Organic Rankine Cycle. This project shall combine the concept of ORC with solar energy as a heat source. The main advantages of such a new combination are:

- Modularity of the power block that makes transport, installation and operation much easier,
- The use of relatively low pressures and condensers at atmospheric pressure reduces maintenance efforts,
- The low condensation temperature of the organic fluids and their other properties which allow the use of relatively low working temperatures ($T_{turbine\text{-}inlet}$<300C°),
- The use of low working temperatures on the power block reduces the needed temperatures at the heat source (solar field) and then gives more opportunity to profit from the solar collectors and optimize them.

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II. SYSTEM CONFIGURATION

The proposed system was dimensioned to cover the required thermal energy of 1MW plant for three hours of continuous operation. Assuming that the power block has a rated efficiency of 20%, this means a thermal input of 5MW and a total storage capacity of 15MWh.Among the existing storage technologies, (two molten salt tanks, HTF tanks, chemical storage…), the choice was made on two different technologies:

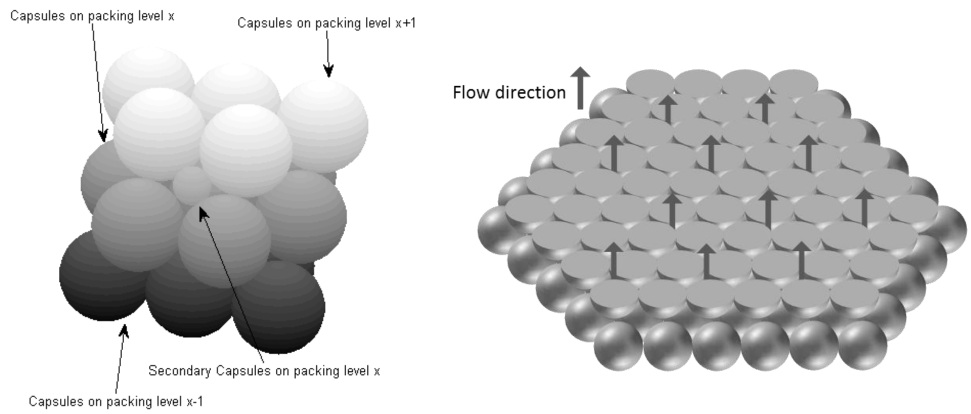
- Thermocline with rocks as filler materials: this approach has a high potential for application in Morocco and may use locally available resourcesand materials.
- Latent heat storage: this is a more ambitious and promising approach which has a high improvement potential and enables high storage densities and relatively stable power and temperature delivery. The simulation model focuses on this particular solution but can be also used (with slight adaptations) for the thermocline system.

The working temperatures of the organic cycle are between 260°C and 290°C which is relatively low when compared with standard commercial plants using steam cycles. These temperatures make impossible the use of conventional two tanks molten salt solution due to salt freezing risks inside the system. Special salt mixtures as Hitec and Hitec XLhave freezing temperatures below 150°C [1], however, the use of such mixtures can hugely raise the total system's cost due to high sodium nitrite and calcium nitrate share in those respective mixtures. The only way to use standard solar salt in our case is as phase change material with a passive storage approach; in that case salt has to be separated from the heat transfer fluid and until now many configurations were studiedespecially the cylindricaloption and its diverse enhancements [2].

We decided here to opt for an encapsulation approach where salt (phase change material) is restrained inside sealed steel spherical capsules [3];[Figure 1 (a)] shows the proposed system's packing disposition. The choice for spherical capsules results from the fact that most phase change materials "PCM" have relatively low thermal conductivity, leading to slowcharging and discharging rates [2]. A sphere capsule will enable a larger contact surface and a better heat transfer.Then the decisive parameter is the diameter of the capsules. As the diameter increases, the total contact surface between the PCM and the HTF decreases and also system's cost. On the other hand, a lower capsules diameter will lead to a larger contact surface and a higher heat transfer rate but will cause a higher system cost due to the higher number of capsules.

The spherical shape of the capsules allows the use of close packing. In our case capsules disposition in the system was inspired by the hexagonal close packing in some minerals, this distribution allows an efficient HTF circulation and enhanced heat transfer between the capsules and the fluid. This choice led to a system (tank) witha hexagonal prism shape with a diameter of 5 m an 6 m height[Figure 1 (b)]. In addition this distribution leaves a room for additional "smaller" capsules or (secondary capsules) to be located at the octahedral voids as shown in [Figure 1 (a)]. These secondary capsules main role is to raise local HTF flow velocity and heat exchange rates.

The system was simulated using three capsules diameters: 5cm, 10cm and 20cm. In those respective cases, secondary capsules had diameters of 2.05 cm, 4.1 cm and 8.2 cm. The close packing distribution allows the capsules to fill up to 74% of storage volume (36% being occupied by the HTF), the use of secondary capsules may achieve 79% of volume occupation. The simulated capsules are made of steel and the outer shell thickness varies with capsule's diameter. However for this simulation the only modelled part is the heat transfer between the capsules and the HTF, the capsules are considered as homogeneous volumes with a specific heat quantity stored on them.



(a) Capsules packing (b) Horizontal cut on a packing layer

Figure 1: Capsules packing and overall disposition

III. ASSUMPTIONS AND BOUNDARY CONDITIONS

Regarding the existing CSP plants and especially the planed CSP-ORC pilot operation mode, systems charging will not constitute a big issue, despite the low thermal conductivity limitation; thermal storage systems have generally sufficient time for charging as solar resource is basically available all day time and peak solar irradiation can span up to eight hours in summer days. A greater attention was granted to discharging mode as the system will have to deliver a specific power to the organic cycle to enable an operation load that guaranties acceptable cycle's efficiency.

The following assumptions were made for the results presented in this work:

- Convective transfer was the only heat exchange mean considered for calculations;
- Flow velocities were supposed homogeneous at all point on horizontal planes, it is onlyfunction of the height;
- Capsules were considered as solid volumes and heat transfer inside the PCM was not taken into account.

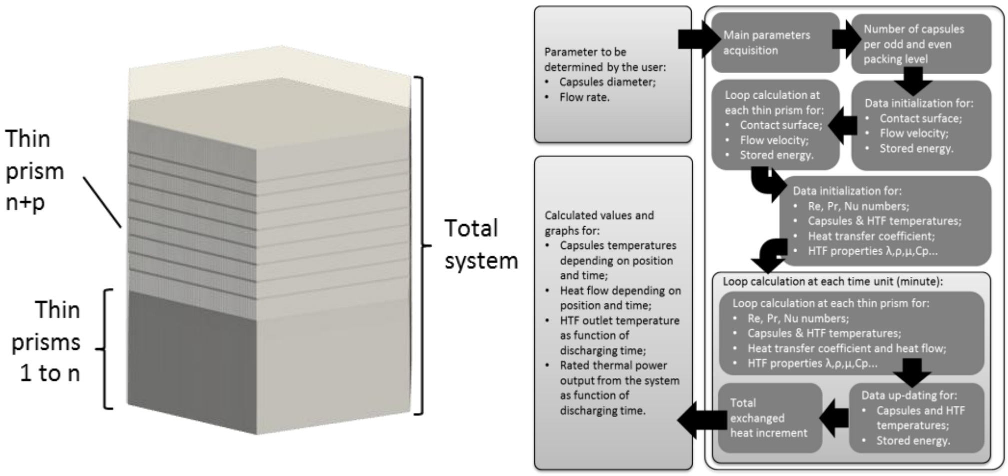
Initial and boundary conditions were defined as follows:

- Flow rate is 0.03 m^3/s;
- Initial HTF and PCM temperatures respectively 170°C and 270°C;
- Initial stored energy density is 185kWh/m^3;
- PCH freezing temperature is 260°C.

IV. CALCULATION PROCESS

The system is divided into thin horizontal hexagonal prisms; each prism shares the same base as the whole system and has a height of 2.8 mm.

The model is built in such a way that enables to calculate the average heat transfer rate at each horizontal plane and the total heat exchanged at each volume element (thin prism) between two successive planes[Figure 2 (a)].The diagram below[Figure 2 (b)] shows model operation and the different calculation phases.



(a) Systems layout and thin prisms disposition (b) model in/output & operation diagram

Figure 2: Storage systems layout and operation diagram

Calculations start at the bottom of the system, a first phase calculates steady values for the whole system, including:

- Contact surfaceSc_i;
- Initially stored energy at time (t=0)$Ec_{i,0}$;
- Flow velocityV_i,…

The second phase concentrates on dimensionless numbers and heat transfer.

Before running those specific parts, the model starts by calculating the number of capsules needed for the system at each packing level depending on capsule diameter, this number shall determine the contact surface and flow section but will have no effect on the stored energy quantity. Flow velocity (V_i) is determined from flow section(S_i)and the volumetric flow rate ($\frac{dv}{dt}$) as follows:

$$V_i = S_i \frac{dv}{dt} \qquad (1)$$

where $i$is the plane coefficient.

HTF properties are calculated at each loop as function of the temperature [1]; these properties in addition to the parameters calculated earlier (flow velocity, stored energy and contact surface) are necessary to determine the

dimensionless numbers (Re, Pr, Nu). Those are calculated for all volume elements and for each minute from discharge process trigger to complete systems discharging (3 hours later).
Convective heat transfer coefficient (H)and heat flow (φ) are calculated through equations (2) and (3). The amount of energy that was transferred is then deducted from the stored heat on the capsules which leads HTF and capsules temperatures to be updated at the end of each calculation loop.

$$H_{i,t} = \frac{k_{i,t} Nu_{i,t}}{L} \quad (2)$$

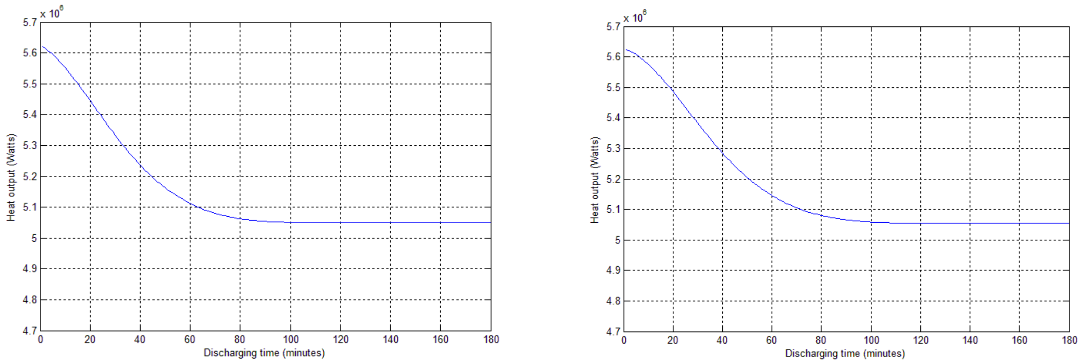
$$\varphi_{i,t} = H_{i,t} Sc_i \left(Tc_{i,t} - Tf_{i,t} \right) \quad (3)$$

where L is the characteristic length, k the thermal conductivity of the HTF,Tc and Tf are PCM and HTF temperatures.
The total heat output (P) of the system is measured by summing-up heat flow at a specific time and for the whole system.

$$P_t = \sum_i \varphi_{i,t} \quad (4)$$

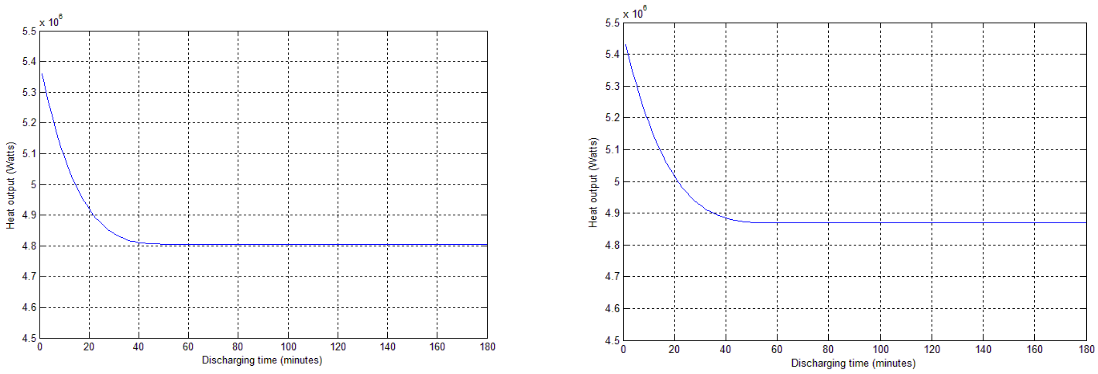
V. RESULTS AND CONCLUSIONS

In the present work, the presented results are those for system using 20 cm, and 10 cm capsulesdiameter with and without additional secondary capsules [Figure 3] and[Figure 4].



(a) System without secondary capsules (b) System with secondary capsules

Figure 3: Heat output for a system with 20 cm diameter capsules



(a) System without secondary capsules (b) System with secondary capsules

Figure 4: Heat output for a system with 10 cm diameter capsules

The curves show a first slope due to PCM's liquid phase temperature that drops until phase change starts, at this moment, output heat and HTF outlet temperature starts to stabilise until the end of the cycle. In all these cases heat output and temperatures are constant until the end of the third hour (180 minutes) this is due to an over-dimensioning of the system which was intended to simplify systems cutting and overall calculations.

The results showed that even with 20 cm capsules, the heat output was still at an acceptable range (4.8 MW) compared to the 5 MW needed for the ORC, the use of 10 cm capsules enabled a slightly higher output but may raise system's costs significantly.

Another issue that must be pointed is that the use of secondary capsules raises the total heat output by less than 2% while adding 6.7% to PCM volume. Adding secondary capsule is also economically ineffective as systems cost can increase or even double.

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