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Research Paper

Techno-economic heat transfer optimization of large scale latent heat energy storage systems in solar thermal power plants

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HIGHLIGHTS

- Methodology for optimization of heat transfer and enhancement structures.
- Complex, longitudinal fin profiles made of aluminum determined as best profile.
- Hexagon determined as optimum cross section geometry of one finned tube.
- Simplified model for coupling of LHTESS' and steam turbine's part load behavior.

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ABSTRACT

Concentrated solar power plants with integrated storage systems are key technologies for sustainable energy supply systems and reduced anthropogenic CO_2-emissions. Developing technologies include direct steam generation in parabolic trough systems, which offer benefits due to higher steam temperatures and, thus, higher electrical efficiencies. However, no large scale energy storage technology is available yet. A promising option is a combined system consisting of a state-of-the art sensible molten salt storage system and a high temperature latent heat thermal energy storage system (LHTESS). This paper discusses the systematic development and optimization of heat transfer structures in LHTESS from a technological and economic point of view. Two evaluation parameters are developed in order to minimize the specific investment costs. First, the specific product costs determine the optimum equipment of the latent heat storage module, i.e. the finned tube. The second parameter reflects the interacting behavior of the LHTESS and the steam turbine during discharge. This behavior is described with a simplified power block model that couples both components.

1. Introduction

1.1. DSG solar power plants

State-of-the-art concentrated solar power (CSP) plants use thermal oil as heat transfer fluid (HTF). This limits the maximum operation temperature to 400 °C and, thus, the overall plant efficiency. Instead, solar thermal power plants based on direct steam generation (DSG) with steam temperatures up to 550 °C offer higher efficiencies [1,2]. For further cost reduction cost efficient thermal energy storage (TES) system is required. However, a latent heat storage system that absorbs and provides the heat of evaporation is not commercially available.

This paper focuses on the design optimization of the latent heat storage system's main equipment, i.e. vertical tubes equipped with fins. Longitudinal fins manufactured by extrusion allow very flexible and adaptable fin design and low manufacturing costs. In addition, longitudinal fins lead to minimum mechanical load on the fins, when the phase change material (PCM) volume changes due to the solid–liquid phase change.

The considered latent heat storage system is a part of a combined latent-sensible TES system as proposed by Seitz et al. [3]. The design basis is a CSP plant with a nominal power output of 50 MW_{el}, a solar field capacity of 250 MW_{th} with a solar multiple of 2 and a direct normal irradiation of 850 W/m^2 (e.g. Andasol plant). The maximum discharge time of the fully charged storage system is set to 8 hours. Sodium nitrate with a phase change temperature of 306 °C is applied as latent heat storage medium, while the solar salt, a

Abbreviations: CSP, concentrated solar power; DNI, direct normal irradiance; DSG, direct steam generation; FEM, finite element method; HTF, heat transfer fluid; LCOE, levelized cost of electricity; LHTESS, latent heat thermal energy storage system; NL, nominal load; O&M, operation and maintenance; PCM, phase-change-material; PL, part load; SF, solar field; SoC, state of charge of LHTESS; TES, thermal energy storage.

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mixture of 60% sodium nitrate and 40% potassium nitrate, is used as the sensible storage medium.

1.2. Heat transfer enhancement in latent heat storage systems

In high temperature latent heat storage systems alkali salts such as sodium nitrate or potassium nitrate are typically used [3]. However, these materials have a very low thermal conductivity in the range of 0.5 W/m/K. However during freezing heat transfer is constraint by thermal conduction, since natural convection at the solid–liquid borderline is severely limited. According to Pointner et al. [4], there are different concepts of how to overcome the limited heat transfer rate. On the one hand stationary systems with either an increased effective thermal conductivity or an extended heat transfer area have been researched. On the other hand the overall heat transfer rate can be improved by moving the PCM and avoiding the constantly growing solid layer.

1.3. Stationary systems

Steinmann and Tamme [5] and Do Couto Aktay et al. [6] have investigated enhancing the thermal conductivity by integrating the PCM into a matrix of highly conductive material such as expanded graphite. Although they applied different manufacturing techniques, experimental results showed different disadvantages such as anisotropic thermal conductivity and dynamic material parameters due to thermal cycling.

In contrast, the concept of applying fins to the heat transfer fluid (HTF) tubes was studied successfully in different small and midscale experiments. Bayón et al. [7] demonstrated the evaporation and condensation of water/steam in a 100 kW_{th} test facility by using graphite foil fins. Another 200 kW_{th} graphite fin based storage module was integrated into a sand-lime bricks plant for waste heat recuperation purposes [8]. In order to avoid reactions between graphite and nitrate salts, which occur at temperatures exceeding 250 °C, aluminum fins have been introduced [9]. Based on the results of a lab module a 700 kWh_{th} demo plant using radial aluminum fins in $NaNO_3$ was developed and built. The system was successfully operated in fixed and sliding pressure mode simulating its combination with a DSG solar power plant [10,11]. A further improvement was achieved due to the application of longitudinal fins that are clipped to the steel tube with a special steel clip. This clip concept enables the assembly to cope with different thermal expansion factors and is suitable for mass production. It was successfully operated for approximately 200 cycles in a lab scale plant consisting of seven tubes [12].

An alternative method is to macro-encapsulate the PCM in tubes or spheres enabling direct heat transfer between the HTF and the encapsulated PCM. These spheres have been investigated widely in low temperature applications [13]. However, in high temperature applications using alkali salts as PCM only steel and nickel with an internal polymer coating have been identified as a noncorrosive and temperature resistive encasing material [14]. Tamme et al. [15] and Buschle [16] reported that in comparison to an external PCM arrangement this technique is less efficient due to a higher amount of required encasing material.

2. Methods and calculations

The optimization of the LHTESS's heat transfer structure is based on a finite-element method (FEM) simulation model being applied to a multitude of heat transfer structures. The set up and boundary conditions of this model are explained in Section 2.1, whereas Section 2.2 presents the three main targets and the general optimization procedure. This includes the development and the parameterization of different heat transfer structures and fin profiles as well as an overview of all compared structures. The main evaluation parameter – the specific product costs – that is calculated from the FEM simulation results is introduced in Section 2.3. A second parameter – the levelized cost of electricity (LCOE) – is then introduced in Section 2.4, in order to prove the validity of the main evaluation parameter. Finally Section 2.5 presents the model for coupling the power block and the LHTESS that is required for calculating the LCOE parameter.

2.1. Numerical simulation model

In order to optimize the LHTESS's heat transfer structures it is assumed that all tubes and fin profiles are identical and that HTF distribution is uniform across all tubes. At discharge the storage system is operated in forced circulation mode with high volume flows. Thus, freezing along the length of the tubes takes place simultaneously as it has been observed in 3-D simulations. As a consequence, only a two-dimensional cross section of one finned tube is considered for simulation. The cross section consists of a central steel tube with 25.4 mm in outer diameter, the extruded fin profile of variable size and the surrounding PCM. The extruded fins cohere to a second central tube of the same material that is mounted on the inner steel tube. Both central tubes have a wall thickness of 2 mm. The whole structure is enclosed by the basic geometry, i.e. a hexagon, a square or a triangle, that holds the phase change material (Fig. 1). Only these three basic geometries allow for a dense packing of multiple tubes inside the LHTESS without any additional void volume between the tubes. Finally, the size of the basic geometry determines the tube pitch.

The freezing process with a growing resistive solid layer on the heat transfer surface is considered to be the limiting process that dictates the LHTESS's design. At discharge the cold HTF enters the tubes from the bottom limiting the occurrence of natural convection to a minimum. Hence, the heat transfer problem is reduced to a transient heat conduction problem that is solved with Comsol Multiphysics®. The software applies a finite element method and the phase change problem is addressed by the apparent heat capacity formulation.

Prior to the FEM-simulation a detailed verification of the simulation model has been performed based on the analytical solution of a simple 1-D semi-infinite case [17]. Adequate results at acceptable computation times were obtained for a free triangular mesh with "finer" sized elements and the intermediate time stepping method with a maximum time step of 2 s. A further reduction of computation time is achieved by taking advantage of symmetries

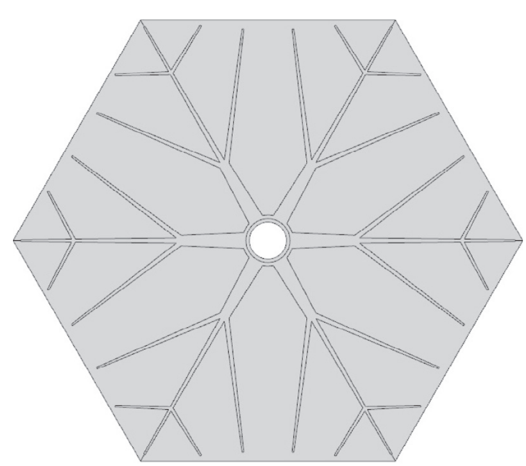

Fig. 1. Fin profile inside the basic hexagonal geometry with 2nd order fin branching level.