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ASSESSMENT OF NATURAL CONVECTION HEAT TRANSFER IN AN ISOCHORIC THERMAL ENERGY STORAGE SYSTEM

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ABSTRACT

Most of the renewable energy sources, including solar and wind suffer from significant intermittency due to day/night cycles and unpredictable weather patterns. Energy Storage systems are required to enable the renewable energy sources to continuously generate energy for the power grid. Thermal Energy Storage (TES) is one of the most promising forms of energy storage due to simplicity and economic reasons. However, heat transfer is a well-known problem of most TES systems that utilize solid state or phase change. Insufficient heat transfer impairs the functionality of the system by imposing an upper limit on the power generation. Isochoric thermal energy storage system is suggested as a low-cost alternative for salt-based thermal energy storage systems. The isochoric thermal energy storage systems utilize a liquid storage medium and benefit from enhanced heat transfer due to the presence of buoyancy-driven flows. In this study, the effect of buoyancy-driven flows on the heat transfer characteristics of an Isochoric Thermal Energy Storage system is studied computationally. The storage fluid is molten elemental sulfur which has promising cost benefits. For this study, the storage fluid is stored in horizontal storage tubes. A computational model was developed to study the effect of buoyancy-driven flow and natural convection heat transfer on the charge/discharge times. The computational model is developed using an unsteady Finite Volume Method to model the transient heat transfer from the constant-temperature tube wall to the storage fluid. The results of this study show that the heat transfer process in Isochoric thermal energy storage system is dominated by natural convection and the buoyancy-driven flow reduces the charge time of the storage tube by 72-93%.

INTRODUCTION

Solar energy is one of the most promising renewable energy sources for the future grid. The governments and industries are providing extensive resources to advance the use of solar energy and reduce the Levelized Cost of Energy provided by solar power plants. Thermal Energy Storage (TES) is an important part of solar power plants and enables them to generate electric power during off-sun hours. Over-generation of solar energy during noon time and unchanging peak of power consumption later in the day leads to formation of a large rate of demand (Duck Curve). Using TES system can help the stability of power grid by storing part of the over-harvested solar energy during off-peak hours and return back the power to the grid during peak hours [1].

Thermal energy can be stored in the form of internal energy of a storage medium. The state-of-the-art of TES is the two-tank molten salt system in which the storage medium is pumped from the "low temperature tank" to the "high temperature tank" through a heat exchanger. In the heat exchanger, the thermal energy is absorbed from the Heat Transfer Fluid (HTF). When the stored thermal energy is required, the storage medium is pumped back to the "low temperature tank" through a heat exchanger to release the heat to the HTF.

Molten salt mixtures are attractive candidates for TES due to their low vapor pressure; however, the high cost of molten salt imposes an elevated cost for solar power plants that utilize conventional two-tank TES systems. Therefore, a challenging goal has been set by the U.S. Department of Energy to reduce the cost of thermal storage to \$15/ kWh_t. Many research studies have been performed on thermal energy storage to develop high-

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density and cost-efficient storage fluids using sensible and latent heat methods [2-3].

Elemental sulfur has interesting characteristics as a TES system which is scalable and robust as recently proposed by Wirz, et al. [4]. Sulfur is the 13th most abundant element on earth. 21.7% of global sulfur supply is produced in the United States and Canada [5]. The typical price of sulfur is $0.075 - $0.16/kg which is about one order of magnitude lower than the cost of nitrate salt mixtures as the state-of-the-art of thermal storage medium. Chemical stability in high temperature condition is a fundamental requisite of any thermal storage medium over a 30-year plant lifetime. As a basic element, sulfur exhibits negligible thermal degradation and is an ideal candidate for high-temperature operation.

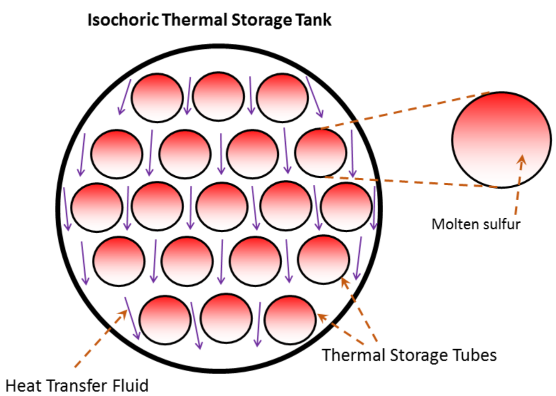


Figure 1 – Schematic of the proposed isochoric Thermal Energy Storage tank.

Figure 1 shows a schematic of the proposed isochoric TES system using molten sulfur as the storage medium. The storage tank is very similar to shell-and-tube heat exchangers with a difference that the molten sulfur is stationary inside the sealed storage tubes. The vapor pressure of sulfur is relatively low in the temperature range of this study (250 – 400 °C); therefore the thickness of the wall is not impacted by the internal pressure of the tube. The shell of the tank is essentially a low-pressure environment through which the HTF flows and exchanges heat with the storage tubes. The thermal energy is transferred from the solar field to the storage tank by the HTF and is stored in the molten sulfur. In the discharge cycle, the HTF extracts the heat from the storage tubes. The typical charge time of the storage tank is about 4 to 6 hours corresponding to the daily peak of solar radiation.

Heat transfer plays an important role in any TES system. The thermal energy is transferred from the HTF to the wall of the storage tubes and consequently to the storage medium. Since forced convection heat transfer is absent inside the storage tubes, the heat is only transferred through natural convection and conduction. Molten sulfur does not have a high thermal conductivity ($k \cong 0.17\ \mathrm{W/m{\cdot}K}$); therefore, availability of desired charge and discharge speeds needs to be verified. On the other hand, the viscosity of molten sulfur goes through significant changes due to polymerization and formation of different species of elemental sulfur (i.e., S_8, S_6, and S_4) in liquid phase [6]. Figure 2 shows the variations of dynamic viscosity of elemental sulfur as a function of temperature. On the same plot the viscosity of some common liquids at elevated temperatures has been shown for comparison. Molten sulfur is highly viscous (sour cream-like) at 250 °C and shows two orders of magnitude of viscosity reduction at 400 °C. The significant change in molten sulfur viscosity imposes an additional complexity on the natural convection inside the proposed isochoric TES storage tubes.

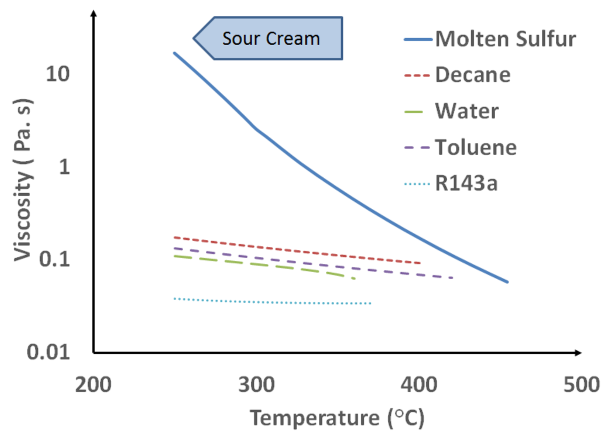


Figure 2 – Variation of dynamic viscosity of molten sulfur, pressurized liquid decane, water, toluene, and R143a. The viscosity of molten sulfur shows two order of magnitude change in the range of application.

In this study, the heat transfer mechanism inside the storage tubes of the proposed isochoric TES is investigated considering the dramatic variations of molten sulfur viscosity. A computational model is developed and validated to investigate the buoyancy-driven flow and natural convection inside the storage tubes.

GEOMETRY OF THE PROBLEM AND MATHEMATICAL FORMULATION

The geometry of the problem is shown in Fig. 3. The tubes are assumed to be long and end effects are neglected; hence, a two-dimensional solution domain is utilized. The tubes are initially filled with molten sulfur. The temperature-dependent properties of molten sulfur is obtained from DIADEM data base [7]. It is assumed that the molten sulfur and the tube wall are initially at 250 °C. The outer surface of the storage tube is in contact with the HTF which flows continuously around the storage tubes. It is assumed that the outer surface of the tube is at a constant temperature of 400 °C during the charge cycle. The heat is then transferred to the molten sulfur through the tube wall which has a high thermal conductivity.

The initial Rayleigh number of the problem is defined as

$$Ra = \frac{g\beta D_o^3(T_w - T_o)\rho^2 c_p}{k\mu} \quad (1)$$

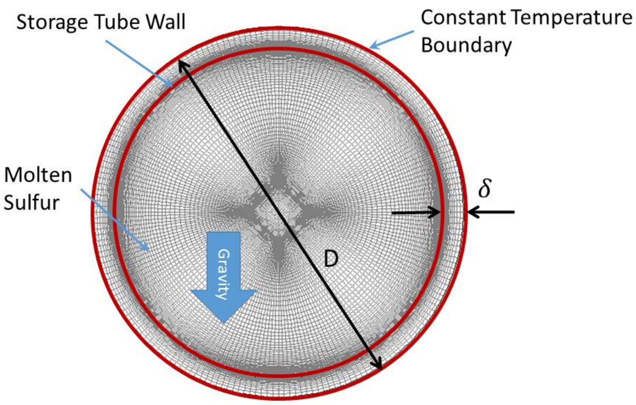


Figure 3 - Geometry of the problem and the schematic of the computational grid.

The computations were performed for small- to moderate-sized storage tubes with radii of D_o = 60 mm and 200 mm leading to the initial Rayleigh number of 1.34×10^5 and 4.96×10^6. Experimental studies show that the transition to turbulence in horizontal tubes occurs at about Ra ~ 10^7 to 10^9 and a fully turbulent flow is established beyond Ra = 10^9 [7-8]; therefore the computations in this study were performed with laminar formulation.

The governing equations of the problem are continuity, momentum, and energy equations which are given below in indicial notation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_j)}{\partial x_j} = 0 \quad (2)$$

$$\frac{\partial(\rho v_i)}{\partial t} + \frac{\partial(\rho v_i v_j)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial v_i}{\partial x_j} + \frac{\partial v_j}{\partial x_i}\right)\right] + \rho g_i \quad (3)$$

$$\frac{\partial(\rho h)}{\partial t} + \frac{\partial(\rho v_j h)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[k\frac{\partial T}{\partial x_j}\right] \quad (4)$$

The boundary conditions at the tube wall are no-slip ($v_i = 0$). The temperature of the outer surface of the tube wall is considered to be non-changing and equal to 400 °C. The molten sulfur is assumed to be at rest at the beginning of the computations. It should be noted that this study is performed for pure molten sulfur in liquid phase; therefore the two-phase region is avoided.

Equations (2)-(4) were discretized using a finite-volume method and an implicit and second-order upwind scheme was used to interpolate the convective terms in momentum, energy. The cells are clustered near the tube wall to resolve the high gradients of velocity and buoyancy force. The commercial software ANSYS FLUENT 15 was utilized for the computations. The initial time steps were as small as 10^{-4} seconds. The convergence criteria were to reduce the residuals of all equations to 10^{-6} at each time step. The computations were continued until the dimensionless mean temperature of the fluid, T^*, was more than 0.9 and the charge time, $\tilde{t}$, was found for 90% of the possible energy storage. The model is also used to solve the energy equation in the absence of gravity so that the charge time corresponding to pure conduction is obtained.

The adopted computational model was previously verified by an extensive grid refinement study and validated by reproducing experimental results available in the literature [9-10]. A grid refinement study was performed to obtain the optimal number of grid points for accurate computational results. The results of the grid refinement study is summarized in Fig. 4. The mean temperature of the storage fluid has been plotted as a function of time in the first 20 seconds of the flow time using three significantly different grids, i.e., with 10,000, 23,000, and 200,000 grid points. The asymptotic behavior of the results show that increasing the number of grids beyond 23,000 increases the computational time without providing significantly more accurate results. Therefore, the computations of this study were performed with 23,000 cells.

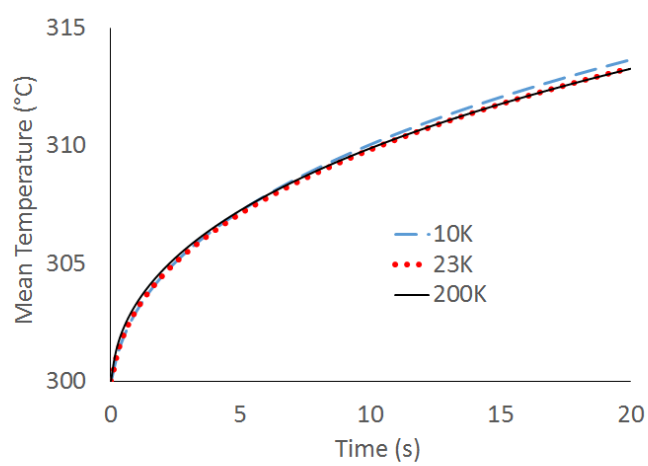


Figure 4 – Grid refinement study results using grids with 10,000, 23,000, and 200,000 grid points

RESULTS AND DISCUSSION

The variations in buoyancy-driven flow, temperature field, and viscosity distribution in the 60-mm storage tube are shown in Fig. 5. The heat is initially conducted to the sulfur layer attached to the tube wall and increases the temperature of this layer. Consequently, the density and viscosity of this layer decreases rapidly and a buoyancy force is formed in either side of the tube which causes a rising flow. Conservation of mass requires formation of a returning flow at the center of the tube. Once the flow is established, two counter-rotating vortices are formed inside the storage tubes. The center of the vortices are initially along the centerline of the cross section (as observed in Fig.3); however, in the later times of the charge cycle the center of the vortices fall below the center line and the recirculating regions lean to the storage tube walls. The temperature field is highly affected by the buoyancy-driven flow field. As the charge cycle progress, the hotter fluid is pushed toward the top by the buoyant flow causing the maximum temperature of the fluid to occur at the top of the tube. The temperature field in the core region of the fluid is impacted by the recirculating regions. The center of the vortices remain at a lower temperature and higher viscosity due to less flow activities.

The variation of the average temperature of storage fluid, T_m, and average heat flux delivered to the storage tube during the charge cycle are plotted in Fig. 6. On the same figure the results corresponding to the pure conduction condition are superimposed. As expected, the results of the natural convection and pure conduction are identical in the beginning of the charge cycle. The natural convection results deviate from pure conduction at about 100 s after the start of the charge cycle. The peak of heat flux at 0.1 s is due to the transient conduction through the tube wall [11]. The increase of heat transfer to the storage fluid by buoyancy-driven flow is observed beyond t = 100 s. The presence of the buoyancy-driven flow decreases the charge time of the 60-mm storage tube from 46 minutes to about 12 minutes. The reduction of the charge time is very critical for the performance of a TES system that ideally charges in 4-6 hours.

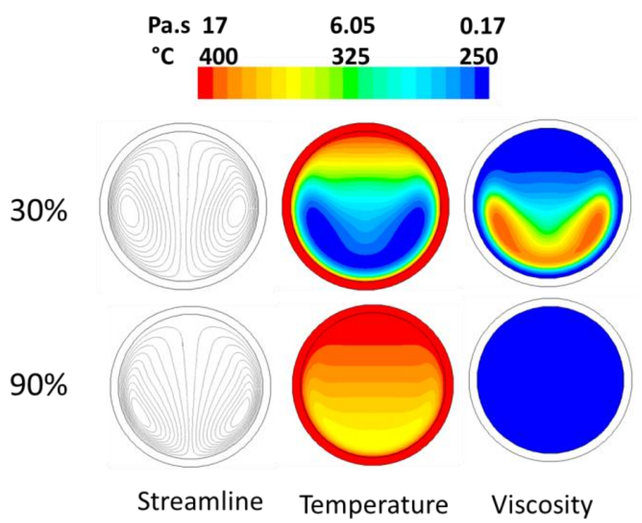


Figure 5 – Buoyancy-driven flow, temperature, and viscosity distribution at 30% and 90% of the charge cycle for the 60-mm storage tube

In the next step, the effect of increasing the size of the storage tube on the charge time is investigated. Figure 7 illustrates the evolution of the buoyancy-driven flow in a 200-mm storage tube that has the same wall thickness. The buoyancy-driven flow is initially similar to the developed flow in the smaller storage tube. The recirculating regions are formed on either side of the storage tube and cause a stratified temperature distribution. The hottest region is at the top of the cross section and the high temperature front progresses to the bottom during charge cycle. The flow field is slightly different from the 60-mm tube at the later times of the charge cycle. After about 27.5 minutes of the charge cycle (~ 40% of energy storage), the flow field becomes more complicated. A second pair of counter-rotating vortices is formed at the bottom of the tube cross section, which makes the flow and temperature field slightly asymmetric at the bottom while the rest of the solution is symmetric. As the solution progresses, the symmetric flow and temperature fields are regained. The formation of the secondary vortices and the partly-asymmetric flow in horizontal tubes has previously been reported by Xia et al [12].

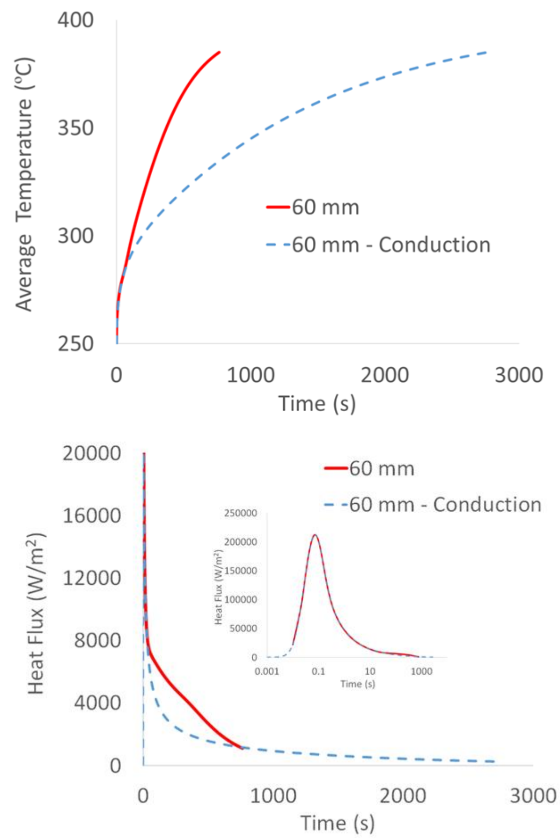


Figure 6 – Average temperature of the storage tube (top) and the average heat flux (bottom) for a 60mm storage tube during charge cycle.

Figure 8 shows the variation of the average temperature of the storage fluid and the average heat flux for the 200-mm storage tube with and without the effect of buoyancy-driven flow. The general trend of the average temperature and heat flux is similar to the 60-mm storage tube; however, the appearance of the secondary counter-rotating tubes in the 200-mm causes a slight increase in the heat flux at t = 27.5 min. Natural convection heat transfer reduces the charge time in the 200-mm storage tube from 36 hours (pure conduction) to 2.5 hours.

Although the charge time is significantly impacted by the buoyancy-driven flow, it takes about 2.5 hours for the 200-mm storage tube to charge. This fact imposes an evident lower limit on the required time for the TES tank charge time during on-sun hours. In other words, increasing the diameter of the storage tubes in the proposed isochoric TES system from 60 mm to 200 mm increases the minimum required charge time of the TES system from 12 minutes to 2.5 hours. The significant increase of charge time in larger tubes is due to low thermal conductivity of molten sulfur and its high viscosity at the beginning of charge cycle.

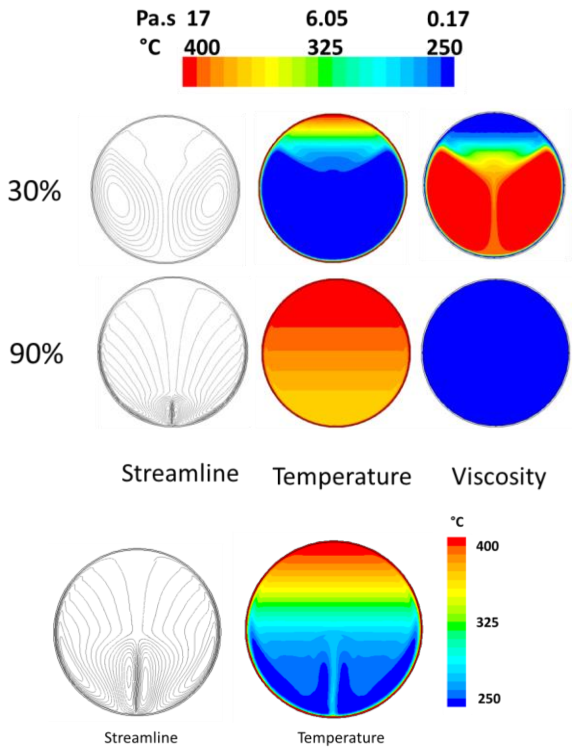
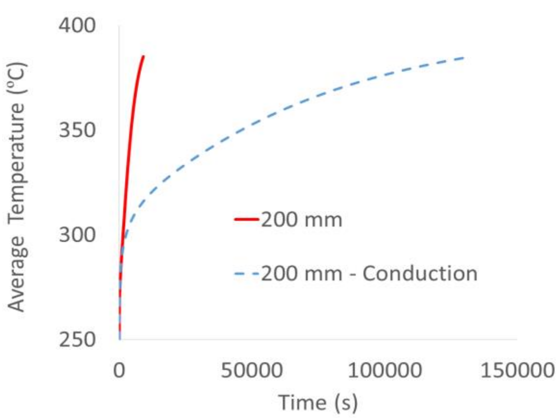


Figure 7 – Buoyancy-driven flow, temperature, and viscosity distribution at 30% and 90% of the charge cycle for the 200-mm storage tube (top) – formation of the secondary counter-rotating vortices at t = 27.5 min (bottom)

The design of the isochoric TES system may require adopting larger tubes due to cost and system-level reasons; however, increasing the tube size has a negative impact on heat transfer and additional modifications might be required to reduce the overall charge time of the proposed isochoric TES system.



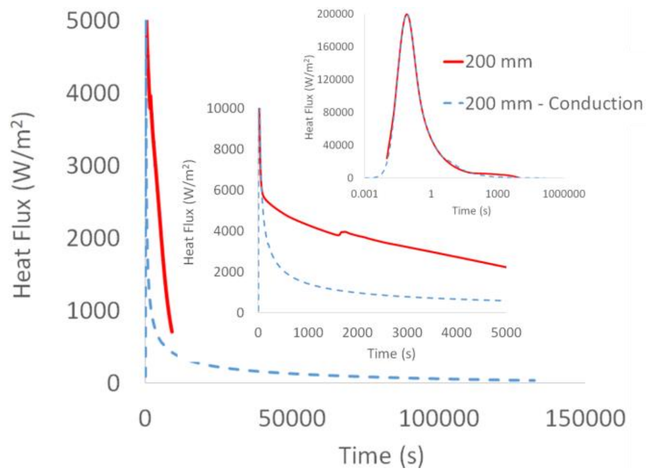


Figure 8 –Average temperature of the storage tube (top) and the average heat flux (bottom) for a 200-mm storage tube during charge cycle.

CONCLUDING REMARKS

In this study, the heat transfer effectiveness for the charging of an isochoric thermal energy storage system was computationally assessed. The storage fluid is molten sulfur in the range of 250-400 °C. A computational model was developed to estimate the charge time of the storage tubes with two different tube diameters.

The results of this study show that the buoyancy-driven flow on the cross section of the horizontal storage tubes reduce the charge time of the thermal energy storage tank compared to pure conduction. The reduction of charge time compared to pure conduction is more significant in the tubes with larger diameters. Natural convection reduces the charge time of 60-mm-diameter storage tube from 46 minutes to about 12 minutes, making this a viable approach for solar thermal energy storage with desired charge times on the order of 4-6 hours.

Increasing the size of the storage tubes from 60 mm to 200 mm has a significant impact on the charge time. The results show that the charge time of the 200-mm-diameter tube is about 2.5 hours. If larger tube diameters are to be used for sulfur-based TES system, additional active or passive heat transfer methods should be considered to reduce the charge time. Further studies are underway to similarly examine the discharge time.

NOMENCLATURE

c_p	Specific heat (J/kg.K)
D_i	Inner diameter of the storage tube (m)
D_o	Outer diameter of the storage tube (m)
$g = 9.81$	Gravity (m/s^2)
$h = \int_{T_{ref}}^{T} c_p dT$	Sensible enthalpy (J/kg)
k	Thermal conductivity (W/m.K)
P	Static Pressure (Pa)
q_w	Wall heat flux (W/m^2)

Ra	Rayleigh Number
t	Time (sec)
$\tilde{t}_{conv}$	Charge time with natural convection (sec)
$\tilde{t}_{cond}$	Charge time with pure conduction (sec)
T	Temperature (K)
T_m	Spatially-averaged sulfur temperature (K)
$T^* = (T_m - T_i)/(T_w - T_i)$	Dimensionless Mean Temperature
v_j	Velocity components (m/s)
x_j	Spatial coordinates (m)

Greek symbols

β	Thermal expansion coefficient (1/K)
$\delta =$ 0.06 mm	Tube wall thickness (m)
μ	Dynamic Viscosity (Pa-s)
ρ	Density (kg/m^3)

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