Improving Thermal Performance of Solar PTC Using Internal Finned Structure

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Abstract

In the present work, simple and finned absorber tubes of a parabolic trough solar collector (PTC) are considered and analyzed numerically. The three-dimensional numerical simulations have been done by using a COMSOL Multiphysics 6.1 software. This work has also been carried out to investigate temperature distribution, useful energy gains and thermal efficiency of absorber tubes. The variable input data of a solar heat flux condition is applied to receiver tubes for time dependent solver configuration. It can be seen from the analysis that finned receiver tube showed better performance comparing to common one. In the discussion section of this article, temperature variations, useful energy gain, thermal efficiency of the receiver of Solar PTC are provided.

Keywords: Solar PTC, Absorber tube, Finned structure, Numerical model.

1. Introduction

There are many types of solar collectors, and the parabolic trough concentrator (PTC) has several advantages over other solar collectors. The possibility of using these concentrators in low, medium and high temperature systems, the higher efficiency compared to flat, evacuated tube collectors, and the fact that they have a single-axis solar tracking system are simpler than other collectors with two and three-axis tracking systems. In addition, the receiver tube of the concentrator can be installed along the common focal line [1].

In addition to the reflector, support and tracking system components of the concentrator, the absorber tube is also the main working part. The main reason for this is that the performance and efficiency of the concentrator depend on the design and efficiency of this absorber tube, and its cost is 30% of the cost of a parabolic trough concentrator. Many materials have been studied as a material for the heat sink tube. For example, copper, aluminum, stainless steel, iron and other materials can be cited as examples. Among these, the material with the highest thermal conductivity is copper [2]. In addition, the application of selective coatings to this tube and the use of a glass-coated vacuum also increase heat absorption and reduce heat loss. Many researchers have also worked on increasing the useful work coefficient of the parabolic trough concentrator. The lower surface of the heat transfer tube has a higher temperature than the upper surface, which leads to rapid failure. To solve these problems, it has been proposed to rotate the absorber tube at a certain frequency to reduce the high surface temperature and maximize the absorption of solar energy [3], [4].

An example of the work of Antar M.M. and Peng Liu on improving the efficiency of the absorber tube by modifying its internal design and using internal spiral fin structures is [5], [6]. A design with a baffle with a conical fin inside was numerically studied by Peng Liu, Nianben Zheng, etc. [7]. The results of the study show that using this design leads to an increase in the thermodynamic efficiency of the collector and a decrease in the rate of entropy generation.

2. System description

A design of receiver tube of Solar PTC in Figure 1 is built in 3D geometry in COMSOL Multiphysics 6.1 software. Water is used heat transfer fluid [8], [9], stainless steel is used receiver tube material [10], [11]. The length of the receiver is 2 m, inner and outer diameters of tubes are 0.03 and 0.06 m.

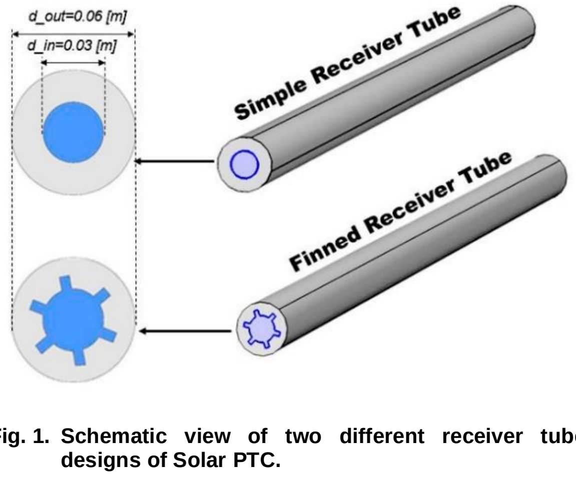

Fig. 1. Schematic view of two different receiver tubes designs of Solar PTC.

2.1. Boundary conditions and Method

Heat transfer in Solid and Fluid, Laminar flow interfaces of COMSOL Multiphysics 6.1 software are utilized for modelling. Boundary conditions and material properties in the Table 1 and Table 2 are inserted to the parameters to the application. Concentration ratio is equal to 4 which calculated with aperture area of the concentrator and receiver area.

Table 1. Variation of solar radiation.

	Time	q_inlet_func(t)
0	8:00 am	400$[W/m^2]$
1	9:00 am	550$[W/m^2]$
2	10:00 am	700$[W/m^2]$
3	11:00 am	850$[W/m^2]$
4	12:00 am	900$[W/m^2]$
5	1:00 pm	850$[W/m^2]$
6	2:00 pm	700$[W/m^2]$
7	3:00 pm	500$[W/m^2]$

The mass flow rate is 0.005 [kg/s]. Inlet temperature (T_{in}) of HTF is 298.15 [K]. External natural convection is also conspired across all boundaries of receiver tubes.

2.1.1. Finite element analysis

The COMSOL Multiphysics finite element method was used to solve the two-dimensional Navier-Stokes equations with the energy equation. Finite element analysis (FEM) is a numerical computational method used in engineering and scientific research to model and analyze complex physical processes. FEM is widely used to solve differential equations and analyze environments with complex geometries. Boundary conditions are applied to the all materials which is provided in the Table 1. Normal Mesh size is generated using Physics-Induced Sequence.

Table 2. Finite element solver

Elements type	Domain elements	Boundary elements	Edge elements
Simple	23705	30330	1420
Elements type	Tetrahedra	Mesh vertices	Edge elements
Finned	27562	7809	1730

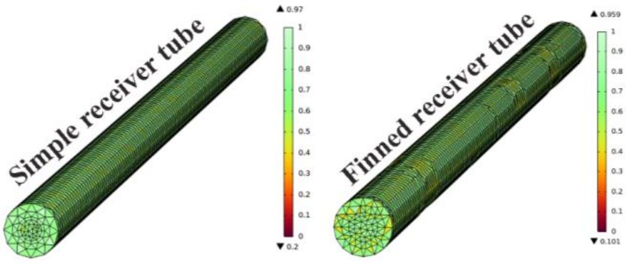

Fig. 2. Generated mesh for two different receiver tubes

Table 3. Thermophysical properties of materials [12], [13], [14]

№	**Materials**	Density $(\boldsymbol{kg/m^3})$	Specific heat $(\boldsymbol{J/kg\cdot K})$	Thermal conductivity $(\boldsymbol{W/m\cdot K})$
1	Water	998.2	4280	0.6
2	Stainless steel	7850	475	44.5

2.2. Equations, Formulas, Symbols, Units

Following Eq.1 is utilized for solving the Heat Transfer in Solids Interface [15], [16], [17].

$$\rho C_p\left(\frac{\partial T}{\partial t} + u_{trans}\cdot\nabla T\right) + \nabla\cdot(q + q_r) = -\alpha T:\frac{dS}{dt} + Q \qquad (1)$$

For a steady-state problem the temperature does not change with time and the terms with time derivatives disappear.

The first term on the right-hand side of Eq.1 is the thermoelastic damping and accounts for thermoelastic effects in solids:

$$Q_{ted} = -\alpha T:\frac{dS}{dt} \qquad (2)$$

It should be noted that the *d ⁄ dt* operator is the material derivative, as described in the Time Derivative subsection of Material and Spatial Frames.

For Heat Transfer in Fluids

The Heat Transfer in Fluids Interface solves for the following Eq.3:

$$\rho Cp\left(\frac{\partial T}{\partial t} u\cdot\nabla T\right) + \nabla\cdot(q + q_r) = \alpha_p T\left(\frac{dp}{dt} + u\cdot\nabla_p\right) + \tau:\nabla u + Q \qquad (3)$$

the Cauchy stress tensor, σ, is split into static and deviatoric parts as in:

$$\sigma = -pI + \tau \qquad (4)$$

for ideal gases, the thermal expansion coefficient takes the simpler form $\alpha_p = 1/T$

$$\alpha_P = \frac{1}{\rho}\frac{\partial\rho}{\partial T} \qquad (5)$$

For a steady-state problem, the temperature does not change with time and the terms with time derivatives disappear. The first term of the right-hand side of Eq.1 is the work done by pressure changes and is the result of heating under adiabatic compression as well as some thermoacoustic effects. It is generally small for low Mach number flows.

$$Q_P = \alpha_P T\left(\frac{dp}{dt} + u\cdot\nabla_p\right) \qquad (6)$$

The second term represents viscous dissipation in the fluid:

$$Q_{vd} = \tau:\nabla u \qquad (7)$$

where, ρ is the density (kg/m^3), C_p is the specific heat capacity at constant stress (J/(kg·K)), T is the absolute temperature (K), u_{trans} is the velocity vector of translational motion (m/s), q is the heat flux by conduction (W/m^2), q_r is the heat flux by radiation (W/m^2), α is the coefficient of thermal expansion (1/K), S is the second Piola-Kirchhoff stress tensor (Pa), Q contains additional heat sources (W/m^3).

Useful energy gain is defined as:

$$Q_u = \dot{m}\cdot C_p(T_{out} - T_{in}) \qquad (8)$$

Where $\dot{m}$ is mass flow rate of fluid (kg/sec), T_{out} is outlet temperature (K), T_{in} is inlet temperature (K). C_p is specific heat capacity $(J/kg\cdot K)$.

$$\eta = Q_u/(I_0\cdot A_a) \qquad (9)$$

2.3. Results and Discussion

Simulations are conducted under time dependent solver configuration. Variable solar radiation during the day is provided in the Table 1. The variable solar radiation is applied with q_inlet_func(t), t-is time.

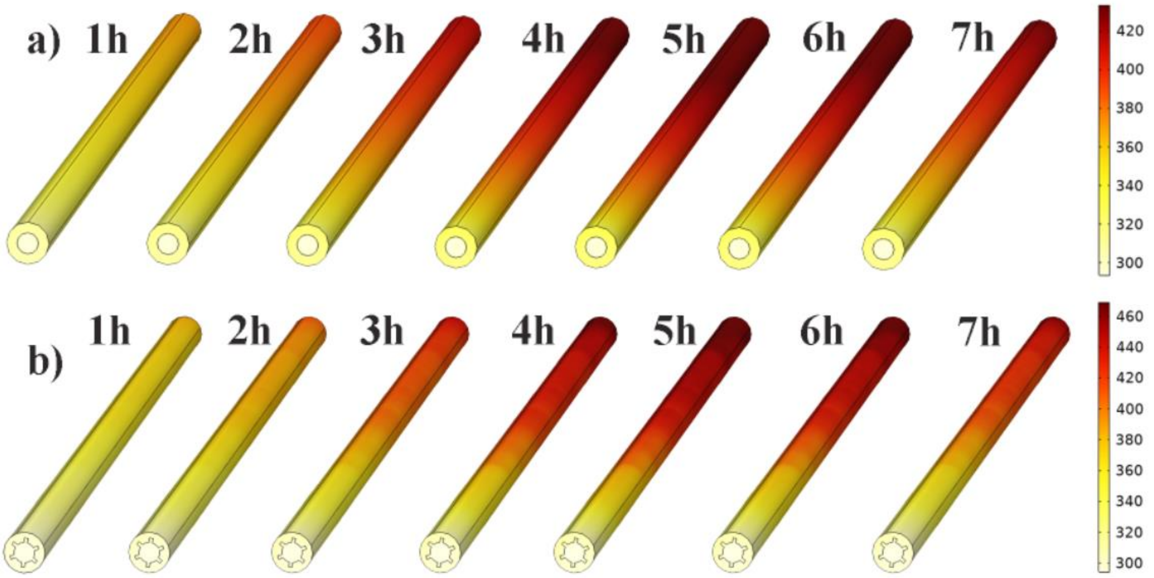

Fig. 3. Variation of the surface temperature of receiver tube, a) Simple receiver tube, b) Finned receiver tube.

Surface temperature variation of receiver tubes at 0.005 [kg/s] mass flow rate is given in the Figure 3. These processes demonstrated steadily increasing temperature distributions during 7 hours. Maximum temperature ranges are observed at 4 to 6-hours period during charging process.

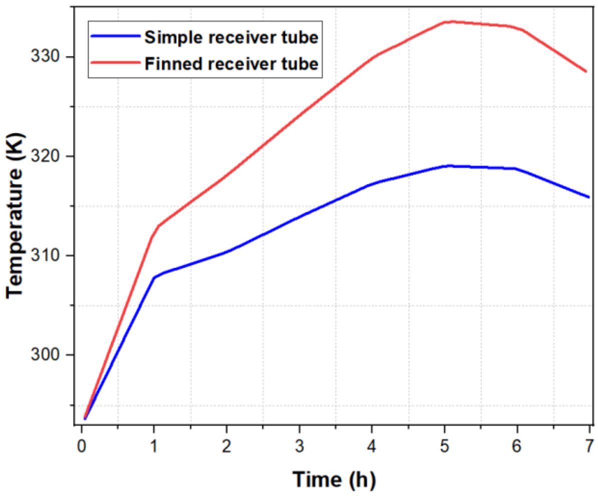

Fig. 4. Outlet temperature variation of the receiver tubes at 0.01[kg/s] mass flow rate

It can be seen from Figure 4 that temperature at outlets points were increased continuously at the given time interval. Maximum outlet temperature reached to 335.15 [K] with finned receiver tube at around 5 and 6-hours. In contrast, simple receiver tube gained 320 [K] at the outlet. This temperature differences between the outlet temperatures shows the better performance of finned receiver tubes. Because, finned structure increase the heat transfer surface area to the working fluid.

Useful energy gain is calculated by applying Eq.8 to the variable part of COMSOL Multiphysics. Figure 5 illustrates the change of useful energy gain at the given time interval. Useful energy reached to its maximum value about 1400 (W) with finned receiver tubes at 5 and 6-hours see the Figure 5, while simple receiver tube presented 1000 (W). This means that finned receiver tube gained more useful energy comparing to simple structure.

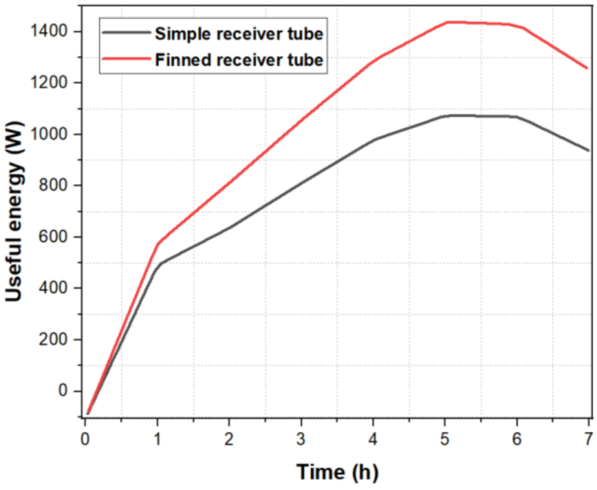

Fig. 5. Useful energy gain through the receiver tubes

Thermal efficiency is one of the most important unit of a receiver tube in Solar parabolic through collector

(PTC) system. It is defined as the ratio of the useful thermal energy gained through the heat transfer fluid (HTF) to the total solar energy incident on the collector's aperture area which is provided in Eq.9. This equation also applied to the variable part of application.

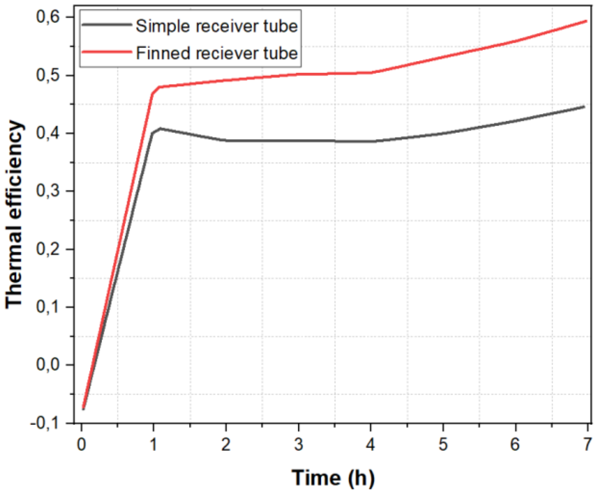

Fig. 6. Thermal efficiency through the receiver tube

Figure 6 illustrates the thermal efficiency of the receiver tube at the given time interval. It can be seen that thermal efficiency of finned receiver tube is higher than the simple receiver.

3. Conclusion

A numerical model of two different receiver tubes are modelled in COMSOL Multiphysics 6.1. Surface and outlet temperature variation of the receiver tubes are investigated. Maximum outlet temperature of the heat transfer fluid finned and simple receiver tubes are 335.15 [K] and 320.15 [K] respectively. Furthermore, useful energy and thermal efficiency of the receiver tubes are determined. Useful energy gain through the finned absorber tube was about 1400 (W) while it was 1000 (W) in simple absorber tubes. Thermal efficiencies are 0.6 and 0.4 for finned and simple receiver tubes. It can be concluded that finned receiver tubes showed better performance comparing to the simple receiver one. This is because, finned structure significantly increases heat transfer surface areas to heat transfer fluid (HTF).

Acknowledgments

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