

Numerical simulation of heat transfer of an absorber with fin of a flat-plate type solar collector

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Abstract. In this study a flat plate type solar collector with fin is studied numerically. The fins are installed to the plate absorber to increase the thermal efficiency of the collector. The objective of this paper is to explore the temperature and velocity distribution in the solar collector. In addition, the rate of the heat transfer from the plate absorber to the tube is examined. The governing equations are developed based on mass conservation, momentum equation and energy conservation. The developed governing equations are solved numerically using commercial computational fluid dynamic code. Pressure distributions, temperature distributions, and velocity distributions are plotted and discussed. The heat transfer rate from the plat absorber to the tube is estimated. Effect of the fins to the heat transfer rate is examined. The results showed that there is no significant effect of installing fins to the performance of the solar collector. It is suggested to use the present result in designing an optimum solar collector.

1. Introduction

The limit of fossil-based energy is a driving force for many countries to improve the renewable energy utilization. One of the most potential renewable energy resources is solar energy due to availability and can reach many counties [1]. Solar energy is useful for in many life activities [2]. Solar energy has many advantages such as unlimited, sustainability, etc. Thus, solar energy is predicted becomes the main energy resources in the future. In the present day, solar energy utilization is not a new technology. Even though, solar energy can be harvested in many countries, however the density is not uniform. In particular for Indonesia, due to its archipelagos located in the equator line, the potency of solar energy is considerable and has long daily radiation time. This is a typical characteristic of solar energy of Indonesia. This potency can be explored in order to support small scale and high scale solar industry. Harvesting solar energy can be divided into solar thermal and solar photovoltaic. This work focuses on the solar thermal application.

In the solar thermal solar collector, the solar energy is absorbed using absorber plate and it converted into heat energy. The heat is transferred into working fluid and then it can be categorized as useful energy or can be used directly for a certain function [6]. Many applications of solar thermal technologies are in service such as solar water heater. This application is very promising that can be used to replace conventional water heater. As a note,



the conventional water heater typically powered by fossil fuel or electricity. Thus, the conventional water heater uses a significant amount of fossil energy and expensive operational cost [5]. On the other hand, Indonesia has a big potency of solar energy with averaged solar radiation of 4 to 4.5 hours per day. This potency can be explored for solar water heater application. However, there are many challenges still exists.

There are many types of solar collector that can be used in solar water heater application such as flat-plate type, evacuated tube, parabolic through, etc. These types of solar collectors have been topics of research around the world. The objective is to improve the performance of the solar collector [4]. The flat-plate solar collector is a collector that is known for its simplicity, easy to make and the price is relatively lower. However, the thermal efficiency is relatively lower [6]. The main component of the flat-plate type solar collector is the plate absorber. As mentioned above that the plate absorber absorb the solar radiation and converted it into heat [4,7]. In order to improve the efficiency of a flat-plate type solar collector fin might be useful. Installing fin to the plate absorber will increase the heat transfer area of the solar collector. The objective of this work is to explore the effect of installing fins to the solar collector. A commercial CFD can be used to simulate fluid flow and heat transfer in a flat-plate type solar collector [8 – 11]. In this stud, CFD will be employed to explore the effect of fins to the performance of the solar collector. The solar collector with fins will be compared with solar collector without any fin.

2. Methods

In the present work, the numerical simulation using CFD will be referred to the previous experimental work [11, 12]. The resulted data will be used as an input for the CFD simulation. Here, the solar collector that analysed is a double-glasses flat plate type solar collector. It consists of double-glasses cover, plate absorber, fins and insulation material. Taking advantage of the symmetry, the solar collector is analysed only for two-dimensional case. The computational domain of the analysed solar collector is presented in Figure 1.

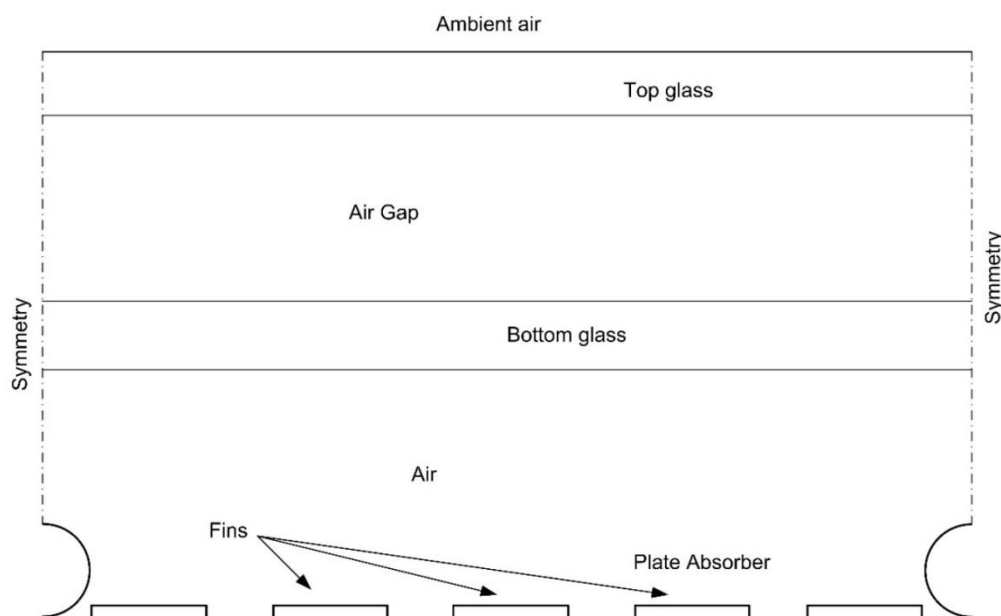


Figure 1. Model of the flat plate collector and computational domain

2.1. Design and boundary condition

It is shown in Figure 1 that the plate absorber of the solar collector is not a flat plate. Several fins are installed. In order to save the computational cost, it is assumed that every pipe is symmetry. The top glass affected by ambient air with temperature fixed at 35°C and convection heat transfer coefficient is 15 W/m²K. Temperature of the plate absorber and temperature of the pipe is fixed at 100°C and 40°C, respectively.

2.2. Mathematical model

Since the computational domain is a two-dimensional case, the governing equations are developed for two-dimensional domain. The assumptions was made, it is a laminar incompressible flow and steady state condition. The continuity, momentum equations and energy equation are formulated as follows [3].

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) + g\beta(T - T_\infty) \sin \phi \quad (2)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + g\beta(T - T_\infty) \cos \phi \quad (3)$$

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

In the analysis, the non-dimensional parameters are formulated below.

$$Ra = \frac{g \beta \Delta T H^3}{\nu^2} Pr \quad (5)$$

Where Pr is Prandtl number and calculated by

$$Pr = \frac{\nu}{\alpha} \quad (6)$$

Local convective heat transfer coefficient is formulated in equation (7).

$$h = \frac{Nu_x k}{H} \quad (7)$$

where k [W/(m.K)] is the conductive heat transfer coefficient of the air and H [m] is the distance of the double glasses cover. The local Nusselt number on the lower surface and upper surface are formulated by equation (8) and equation (9), respectively.

$$Nu_x = \frac{H}{(T_1 - T_2)} \frac{\partial T}{\partial y} \Big|_{y=0} \quad (8)$$

$$Nu_x = \frac{H}{(T_1 - T_2)} \frac{\partial T}{\partial y} \Big|_{y=H} \quad (9)$$

The average Nusselt number is calculated by

$$\bar{Nu} = \frac{1}{L} \int_{x=0}^L Nu_x dx \quad (10)$$

In the literature, several empirical correlations to calculated Nusselt number on the solar collector are available. In this study, the below correlations are used [12,13].

$$\bar{Nu} = 0.195 Ra_H^{0.25} \text{ for } 10^4 < Ra_H < 4 \times 10^5 \quad (11)$$

$$\bar{Nu} = 0.068 Ra_H^{1/3} \text{ for } 10^5 < Ra_H < 4 \times 10^7 \quad (12)$$

The below equation proposed by Holland et al [14] can also be used.

$$Nu = 1 + 1.44 \left[1 - \frac{1708(\sin 1.8 \phi)^{1.6}}{Ra \times \cos \phi} \right] \left[1 - \frac{1708}{Ra \times \cos \phi} \right]^+ + \left[\left(\frac{Ra \times \cos \phi}{5830} \right)^{\frac{1}{3}} - 1 \right]^+ \quad (13)$$

3. Results and Discussions

3.1. Temperature distributions

Numerical simulations have been carried out. Figure 2 shows temperature distributions on the computational domain. The maximum temperature is 373 K shown by red colour and the minimum temperature is 313 K shown by blue colour. In the computational domain, temperature varies between these values. The lowest temperature close to upper glass which is close to ambient temperature. On the other hand, the highest temperature close to the absorber plate. Temperature of the lower glass of the cover is 341.5 K. This can be categorised low.

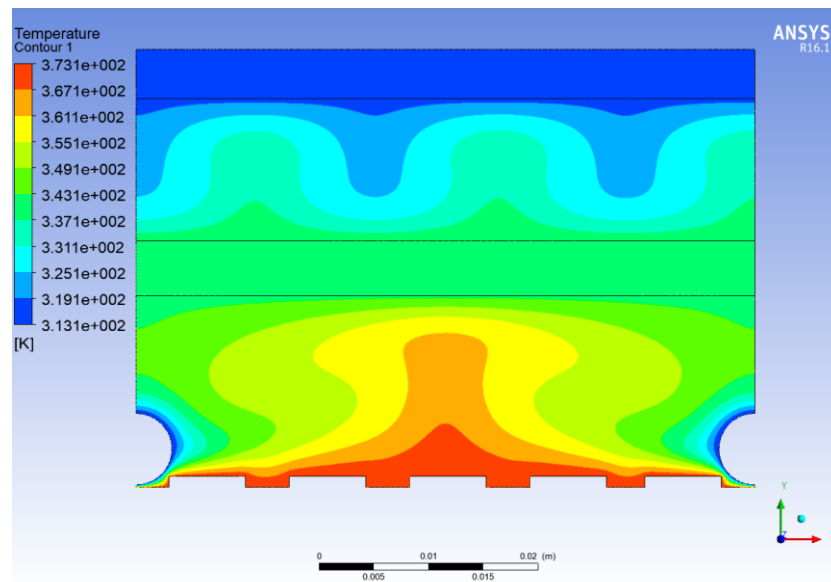


Figure 2. Temperature contour in the computational domain

3.2. Velocity vector

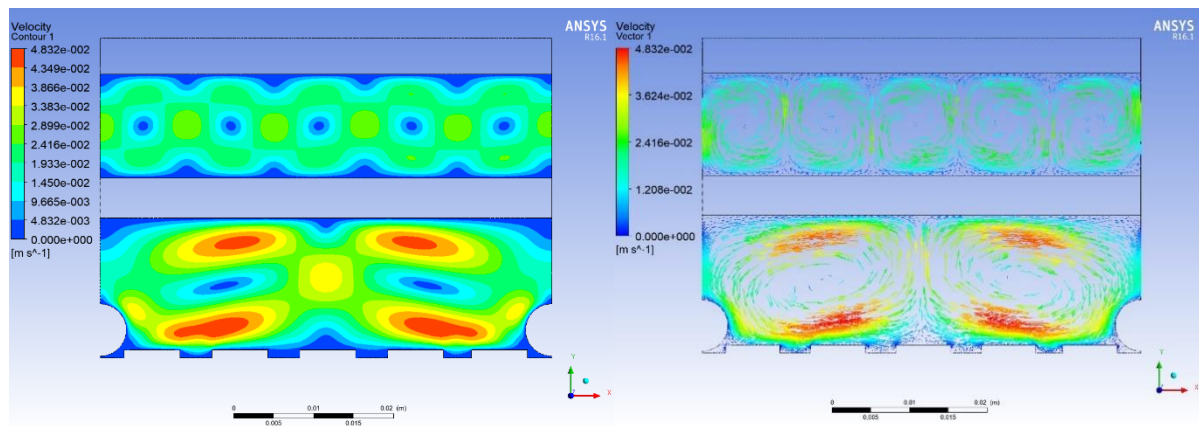


Figure 3. Velocity contour and velocity vector in the computational domain

Figure 3 shows contour velocity and vector velocity in the computational domain. The red colour and blue colour in the figure indicate high velocity and low velocity, respectively. It can be seen that in the air gap between double glass cover, five circulations are captured. The circulation was known as Bern Cell. On the other hand, in the air space between lower glass and absorber plate two circulations are captured. The flow characteristics of the air circulation will affect the heat transfer coefficient.

3.3. Pressure distribution

Figure 4 shows the pressure distribution in the computational domain. It can be seen that the highest pressure in the domain is captured in the upper part of the air between the lower glass cover and absorber plate. In the figure, this area is shown by red colour. This is because in this area the velocity tends to be zero after accelerated from the absorber plate. In contrast, the lowest pressure is captured in the lower part below the red area. This is because the fluid is accelerated from this area. In addition, pressure distribution in the air gap is relatively lower than air in the absorber plate.

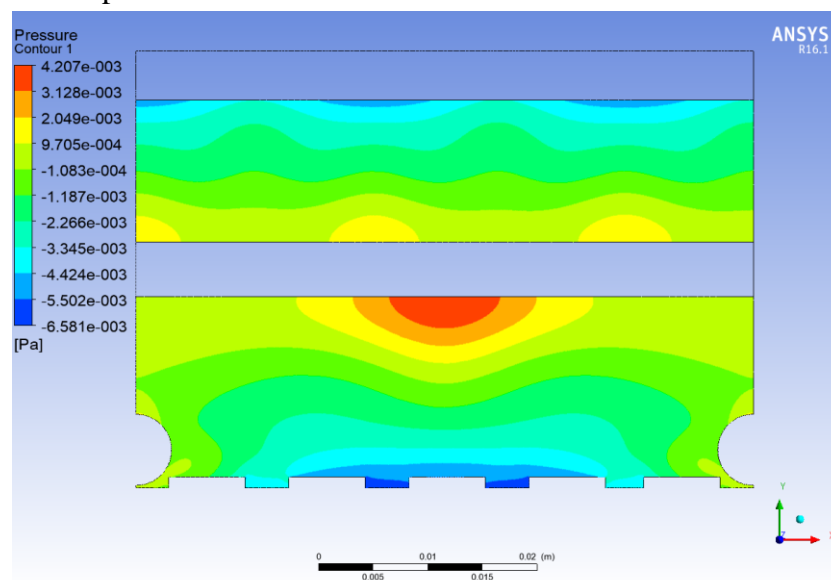


Figure 4. Pressure dstribution in the computational domain

3.4. Heat transfer rate

As a note, the heat transfer mechanism in a flat-plate solar collector can be explained as follows. The heat leaving the plate absorber and goes to working fluid in the pipe and to the ambient air thorough the upper glass cover. By using the temperature distribution, heat transfer rate from the absorber plate to the working fluid inside the pipe is calculated. The heat loss to the ambient air is also calculated. The results show that the heat to the pipe and to the ambient air is 26.82 W and 5.12 W, respectively. Furthermore, the heat transfer rate from the plate absorber is 31.94 W. It can be said that the heat transfer rate obeys the energy conservation law. These values reveal that the heat from the absorber plate is mainly goes to working fluid in the pipe. It is 83.97%.

3.5. Comparison with plain absorber plate

Comparison between absorber plate without and with fins has been made. Figure 5 shows the temperature distribution in the computational domain with and without fins. It can be seen that there is no significant different between the two cases. Only a slightly different on the temperature distribution in the middle area of the plate absorber. By using the temperature distribution, heat transfer rate for both cases are calculated. The heat transfer rate from the absorber plate without fin and absorber with fins is 28.95 W and 31.94 W, respectively. This fact suggests that there is a slightly enhancement. It is about 10.3%.

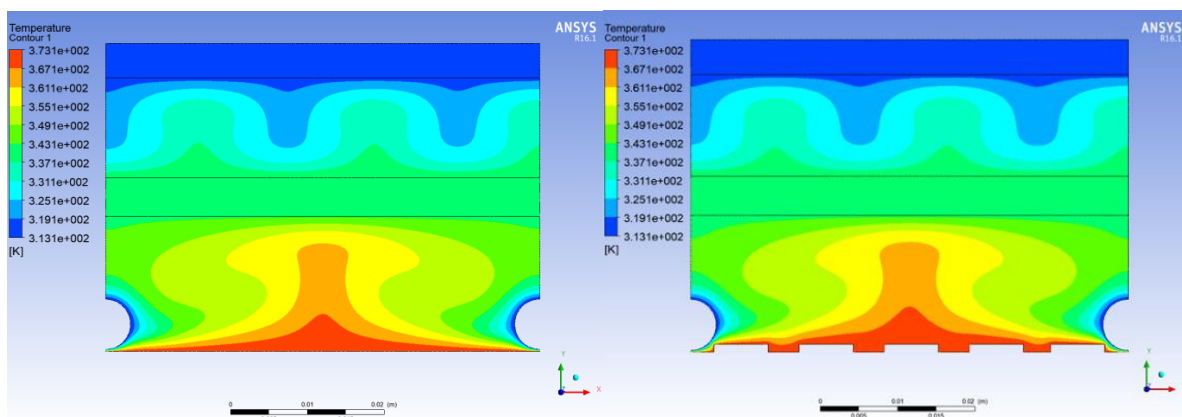


Figure 5. Comparison temperature distribution between absorber plate without and with fins

4. Conclusions

Effect of fins on the absorber plate has been investigated numerically. A commercial CFD code has been used to explore the fluid flow and heat transfer characteristics of the solar collector. Pressure distribution, temperature distribution, vector velocity have been plotted. The heat transfer rate from the absorber plate has been calculated. The results shows that the heat transfer rate from the absorber plate with fins to the working fluid can be up to 83.97%. Installing fins to the absorber plate increases the heat transfer rate to the working fluid about 10.3% in comparison with absorber plate without fins.

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