Fluid flow and heat transfer characteristics of an enclosure with fin as a top cover of a solar collector

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Abstract. To reduce heat losses in a flat plate solar collector, double glasses cover is employed. Several studies show that the heat loss from the glass cover is still very significant in comparison with other losses. Here, double glasses cover with attached fins is proposed. In the present work, the fluid flow and heat transfer characteristics of the enclosure between the double glass cover are investigated numerically. The objective is to examine the effect of the fin to the heat transfer rate of the cover. Two-dimensional governing equations are developed. The governing equations and the boundary conditions are solved using commercial Computational Fluid Dynamics code. The fluid flow and heat transfer characteristics are plotted, and numerical results are compared with empirical correlation. The results show that the presence of the fin strongly affects the fluid flow and heat transfer characteristics. The fin can reduce the heat transfer rate up to 22.42% in comparison with double glasses cover without fins.

1. Introduction

Nowadays the world is facing energy problems includes global warming, lack of energy supplies and environmental concerns as air pollution. Boltzmann [1] concluded that the struggle for life is not a struggle for basic elements or energy, but a struggle for the availability of negative entropy transfer from the hot sun to the cold earth. World populations are expected to double by the middle of the 21st century [2]. To mitigate these problems, fossil fuel replacement with renewable energy is proposed. These problems require energy management as a method of energy use in an efficient and environmentally friendly way. Kabir et al. concluded [3], only three renewable energy sources (i.e., biomass, geothermal, and solar) can be utilized to yield sufficient heat energy for power generation. Of these three, solar energy exhibits the highest global potential since geothermal sources are limited to a few locations, and the supply of biomass is not ubiquitous [4].

There is a need to develop an ingenious method of solar energy conversion systems and then to substitute it where applications of fossil fuels are most vulnerable. One of the most widely developed today is solar energy. Solar energy has been widely investigated for water heating process [5,6], solar cooker [7,8], drying process [9], solar refrigeration [10,11], solar desalination [12,13,14] and so forth. To absorb the heat of the sun and convert it into a source of energy, a solar collector is needed. The main source of a solar collector is the collector itself. Fudholi et al. [15] had concluded that the solar collector
is the vital component of solar thermal systems. In the solar thermal system, the sun heats the collector, and the heat is then transferred to a working fluid. Numerous studies on solar collectors have been reported, and the focuses varied widely. Pinto et al. [16] had concluded, the natural convection study in cavities with inner bodies has been of great interest nowadays due to several engineering applications includes solar collector application. Maliska [17] concluded the advanced in knowledge computational in fluid dynamics (CFD) capability has contributed to more sophisticated equipment development with highest performance levels. Karim and Hawlader [18] studied the mathematical model based on the thermal performance of the finned and V-corrugated absorber plates. The study determined the factors that affect collector performance, such as the optimum angle of the triangular collector and the effect of changes in the shape of absorber plates. The analysis was based on Rayleigh number, inclination angle, AR(aspect ratio) and wavelength laminar. Numerical studies related to solar collector were reported by Beghein et al. [19] who analyzed the influence of buoyancy force on heat or mass transfer and Lewis number in a square cavity.

Utilization of Computational Fluid Dynamic (CFD) commercial code to optimize the design of a flat-plate type solar collector has come under scrutiny. Martinopoulos et al. [20] used CFD to investigate the behavior of a polymer solar collector. Selmi et al. [21] employed CFD commercial code to simulate heat transfer process in a flat-plate type solar collector. Recently, Ambarita [22] explore the characteristics of the enclosure of solar collector double glass cover. The above-reviewed studies show that heat transfer analysis in the enclosure of double glass cover plays an important role in designing highly efficient solar collector. One of the potential innovation to increase the performance of the solar collector is the finned enclosure. This paper focuses on numerical analysis in the finned enclosure between double glass cover of a solar collector. The objective is to explore the heat transfer and flow characteristics in the enclosure with fin. The results are expected to support the necessary information in developing high-performance flat-plate type solar collector.

2. Method

Our research group Sustainable Energy and Biomaterial Centre of Excellent, Faculty of Engineering, Universitas Sumatera Utara is developing high-performance flat-plate solar collector for several applications. Based on the previous studies, it was concluded that the main heat loss from the collector is the top cover. To decrease the loss double glass cover with fin is proposed. Typical of the flat-plate type solar collector is shown in figure 1. The focus of the present study is the comparison between double glass top cover non-finned and finned solar collector.

2.1. Numerical Method

In this study, only the enclosure between the upper and bottom glasses is taken account into domain. The computational domain is shown in figure 1b (enclosure without fin) and figure 1c (enclosure with fin).

Figure 1. Photograph of the system and computational domain
In the present analysis, the domain is assumed to be the two-dimensional case. The flow in the domain is incompressible laminar flow and steady state condition. The additional assumptions are: there is no viscous dissipation, the gravity acts in the vertical direction, and fluid properties are constant except in density of the fluid. The Boussinesq approximation is employed to model the buoyancy force. By using those assumptions, the governing equations are presented below.

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]

\[
u \frac{\partial u}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) + g \beta (T - T_0) \sin \phi
\]

\[
u \frac{\partial u}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + g \beta (T - T_0) \cos \phi
\]

\[
u \frac{\partial T}{\partial x} + \mu \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)
\]

Where \( u \) and \( v \) are the velocity vector in \( x \)- and \( y \)-directions, respectively. In the analysis, non-dimensional parameters are stated in the followings.

\[
Ra = \frac{g \beta \Delta T H^3}{\nu^2}
\]

Where \( Pr \) is Prandtl number and calculated by

\[
Pr = \frac{\nu}{\alpha}
\]

Convective heat transfer coefficient (\( h \)) is calculated using the non-dimensional Nusselt number. It is given by

\[
h = \frac{N_u \times k}{H}
\]

Where \( k [W/mK] \) is the conductive heat transfer coefficient of the air and \( H [m] \) is the distance between the double glasses. The Nusselt number is presented using local and average Nusselt number. The local Nusselt number in the bottom and top surfaces are given by

\[
N_u = \frac{H}{(T_1 - T_2)} \left[ \frac{\partial T}{\partial y} \right]_{y=0}
\]

and

\[
N_u = \frac{H}{(T_1 - T_2)} \left[ \frac{\partial T}{\partial y} \right]_{y=H}
\]

, respectively. The average Nusselt number is calculated by

\[
\bar{N}_u = \frac{1}{L} \int \frac{N_u}{dx}
\]

All of those governing equations are converted into linear equations by employing finite volume method. The system of linear equations for all fields is coupled using SIMPLE algorithm. The commercial code of ANSYS FLUENT is used to carry out the simulation.

2.2. Empirical correlation

The results of the present simulation will be compared with the analytical results. However, there is no correlation of finned surface is found in the literature. Thus only the smooth surface correlation will be used. There several empirical correlations are found in the literature. The first one is given by Jacob [23].
\[ \tilde{Nu} = 0.195 Ra_H^{0.25} \text{ for } 10^4 < Ra_H < 4 \times 10^5 \]  
\[ \tilde{Nu} = 0.068 Ra_H^{1/3} \text{ for } 4 \times 10^5 \leq Ra_H < 10^7 \]

In the more recent experimental study using air inside the enclosure, Holland et al. [24] proposed an empirical correlation between the average Nusselt number and Rayleigh number. The equation is given below.

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

The above equation is claimed to be valid for tilt angle from 0° to 75°. In the equation the meaning of the “+” exponent is that only positive values of the terms in the square bracket are it will be zero if the value is negative. The additional condition for using equation (12) are the aspect ration (AR) of the enclosure and the Rayleigh number must be \( AR \geq 12 \) and \( Ra_H < 10^5 \).

3. Results and Discussion

3.1. Numerical Validation

The present numerical method has been validated in the previous work [22]. The selected numerical validation case is a laminar natural convection heat transfer from a square cavity heated left wall and cooled from right wall. While, the top and bottom walls are insulated. The results are compared for Rayleigh number \( 10^5 \) and \( 10^6 \). The results of the present method show a very good agreement with the previous results. Thus, the present method can be used to explore the problem.

3.2. Flow characteristic and heat transfer of smooth and finned collector

Figure 2 shows flow characteristics of the fluid inside the enclosure for smooth and finned flat-plate collector. The simulation was made for AR of 10. The dimension of the present enclosure is also 1 m × 10 cm with the dimension of fins are 5 cm × 2.5 cm. It is shown clearly that inside the enclosure without fin several circulation flows are generated. The mechanism can be explained in the following. The heated fluid flows upward and before reaching the top surface it will be divided into two flows. Each flow generates a circulation flow and it repeats. The velocity of the fluid follows the Benard cells [22]. The fluid flow characteristics in the enclosure with fin, the Bernard cells clearer and filled the areas between fins. Figure 2 shows this fact very clear.

![Figure 2. Fluid flow characteristics](image-url)
The Benard cells affect local heat transfer characteristics. It is divided the bottom surface into several dead zones, where the velocity almost zero. Between two dead zones the velocity increases and makes temperature gradient higher. Heat transfer coefficient in the dead zone decreases due to lower temperature gradient. On the other hand, the area between dead zones the local heat transfer coefficient increases and reaching maximum value. This is because of higher temperature gradient. This fact is clearly shown in Figure 3. For the enclosure with fin, the local heat transfer coefficient shows a quite different trend. It still shows the sinusoidal pattern. However, on the dead zone areas there is a small wavy heat transfer.

![Figure 3. Locally heat transfer coefficient](image)

### 3.3. Heat transfer coefficient

The heat transfer coefficient from the present result and the estimation using equation (23) and equation (24) are shown in Table 1. The analyses are made at Rayleigh number $Ra = 1.90 \times 10^6$. The table shows that for the smooth enclosure, there is a small difference of average heat transfer coefficient among those methods. However, it is still in the range of less than 10%. For the surface with fin, the heat transfer coefficient decreases significantly. It is reduced 42.27% in comparison with smooth enclosure. This is because installing fin on the surface of glass cover will break some peak heat transfer coefficient as shown in the local heat transfer coefficient of figure 3. Since the peak heat transfer coefficients are broke down by the fins, the average heat transfer coefficient of the finned surface decrease significantly in comparison with the surface without a fin. This fact suggests that glass cover with fin will effective to reduce the heat loss from a solar collector to the ambient.

| Aspect Ratio | Parameter | Present Results | Equation (11) | Equation (12) |
|--------------|-----------|-----------------|---------------|---------------|
|              | $N_u$     | Finned          | Smooth        | Ref. [23]     | Ref. [24]     |
| $AR = 10$    |           | 4.63            | 8.02          | 7.241         | 8.321         |
| $L = 1 m$    | $h$ [W/m²K] | 1.19            | 1.94          | 1.752         | 2.014         |
| $H = 10.0 cm$| $Q'$ [W/m²] | 22.42           | 38.84         | 35.046        | 40.275        |
4. Conclusions

In this work, the heat transfer and fluid flow characteristics in the enclosure between double glasses cover of a smooth and finned flat-plate type solar collector have been numerically studied using commercial code CFD code. The numerical simulation shows that in the enclosure several Benard cells for two types of the solar collector are captured. The presence of the Benard cells strongly affects the local heat transfer coefficient. For the surface with fin, the heat transfer coefficient decreases significantly. It is reduced 42.27% in comparison with smooth enclosure. Since the peak heat transfer coefficients are broke down by the fins, the average heat transfer coefficient of the finned surface decrease significantly in comparison with the surface without the fin. This fact suggests that glass cover with fin will effective to reduce the heat loss from a solar collector to the ambient.

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