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Numerical study on natural convection between double glasses top cover of a flat plate solar collector

H Ambarita[1]

[1]Sustainable Energy and Biomaterial Centre of Excellent, Universitas Sumatera Utara, Jl. Almamater Kampus USU, Medan 20155, Indonesia

*Email: himsar@usu.ac.id

Abstract. Double glasses cover typically used to decrease top heat loss of a flat plate solar collector. However, reported studies show that the top heat loss of a flat plate solar collector is very significant in comparison with other losses. The present work deals with a numerical study on natural convection between double glasses cover of flat plate solar collector. The main objectives are to investigate the heat transfer characteristics of the covers and to propose heat transfer coefficient correlation. Two-dimensional governing equations and boundary conditions are developed. The governing equations are discretized into linear equation system based on finite volume method. The effects of the aspect ratio (AR) and Rayleigh number are explored and fluid flow in the enclosure are plotted. The results show that AR affects fluid flow and heat transfer characteristics significantly. A correlation proposed by the previous researcher is recommended to calculate top heat loss.

1. Introduction

The world population has increased significantly. It is approximated that the present world population is 7.5 billion people. In the last two decades, the population increased more than 1.5 billion. To support the people activities, the world's energy demand also increasing significantly. Preventing an energy crisis is one of the most casual issues of the 21st century. Solar energy, among other renewable energy sources, is a promising and freely available energy source for managing long-term issues in the energy crisis. The solar industry is developing steadily all over the world because of high demand for energy, on the other hand, major energy source, fossil fuel, is limited and other sources are expensive [1]. Solar energy thermal can be used to power solar desalination [2], adsorption refrigeration [3], solar water heater [4], drying process [5], etc. One of the key components of solar collector system is the collector it'self. Many types of solar collector can be found in literature, and flat-plate type solar collector is the simplest solar collector, and it is very popular [6].

In the flat-plate type solar collector, to decrease the heat loss from the top, a double glass cover is typically employed. Studies on the natural heat transfer mechanism in the enclosure between a cover of flat-plate solar collector have been found in the literature. Varol and Oztop [7] reported a comparative numerical study on natural convection in inclined wavy and flat-plate solar collectors. It was shown that flow and thermal fields are affected by the shape of the enclosure and heat transfer rate increases in the case of wavy enclosure than that of a flat enclosure. The effect of inclination to the wavy and flat-plate solar collectors has also investigated [8]. Kumar [9] investigated natural convective heat transfer in a trapezoidal enclosure of box-type solar cooker. It was shown that the values of convective heat transfer coefficient and top loss coefficient for rectangular enclosure are lower by 31-35% and 7 %, respectively.

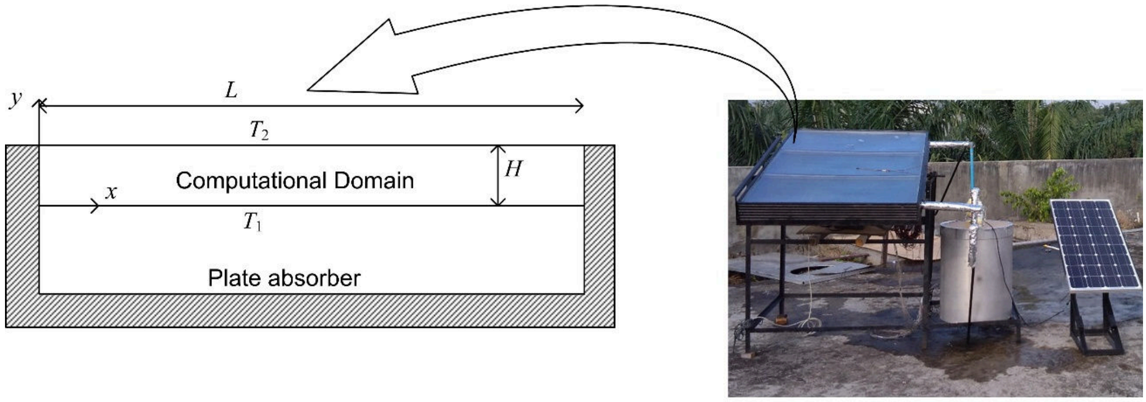
Recently, Computational Fluid Dynamic (CFD) commercial code has been employed to optimize the design of a flat-plate type solar collector. Martinopoulos et al. [10] investigated the behavior of a polymer solar collector experimentally and numerically. Solar irradiation, as well as convection and heat transfer in the circulating fluid and between the parts of the collectors, is considered in the model. Selmi et al. [11] employed CFD commercial code to simulate heat transfer process in a flat-plate type solar collector. The above-reviewed studies show that heat transfer analysis in the enclosure of double glass cover plays an important role in designing and developing a high-performance flat-plate type solar collector. In fact, only a few of empirical correlations are available in the literature. This paper focuses on numerical analysis in the enclosure between double glass cover of a solar collector. The objective is to explore the heat transfer and flow characteristics in the enclosure. The numerical results will be compared with empirical correlation. The results are expected to support the necessary information in developing high-performance flat-plate type solar collector.

2. Method

Our research group in Sustainable Energy and Biomaterial Centre of Excellent, Faculty of Engineering Universitas Sumatera Utara is developing high-performance flat-plate solar collector for several applications such as drying, adsorption cycle, solar desalination, and solar water heater. These applications use flat-plate type solar collector. Typical of the flat-plate type solar collector is shown in Figure 1. In the figure, photograph of a solar water heater that is also shown. The focus of the present study is the enclosure in the top cover of the solar collector.

2.1 Numerical Method

In this study, only the enclosure between the upper and bottom glasses is taken account into consideration. The computational domain is shown in Figure 1(a).



(a) Model for computation (b) Solar Collector System

Figure 1. Computational domain and photograph of a system

In this analysis, the domain is assumed to be the two-dimensional case. The flow in the domain is incompressible laminar flow and steady state. 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 used to model the buoyancy force. By using the above assumptions, the governing equations are as follows.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

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

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

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

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.

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

Where Pr is Prandtl number and calculated by

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

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

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

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

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

and

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

, respectively. The average Nusselt number is calculated by

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

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 from simulation will be compared with the analytical results. In the analytical one, the below empirical correlation will be used. There several empirical correlations are found in the literature. The first one is given by Jacob [11].

$$\bar{N}u=0.195Ra_H^{0.25}\ \text{for}\ 10^4<Ra_H<4\times10^5 \quad (11a)$$

$$\bar{N}u=0.068Ra_H^{1/3}\ \text{for}\ 4\times10^5\le Ra_H<10^7 \quad (11b)$$

In the more recent experimental study using air inside the enclosure, Holland et al. [12] proposed an empirical correlation between the average Nusselt number and Rayleigh number. The equation is given by the following equation.

$$Nu=1+1.44\left[1-\frac{1708\left(\sin1.8\phi\right)^{1.6}}{Ra\times\cos\phi}\right]\left[1-\frac{1708}{Ra\times\cos\phi}\right]^{+}+\left[\left(\frac{Ra\times\cos\phi}{5830}\right)^{1/3}-1\right]^{+} \quad (12)$$

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This equation is claimed to be valid for tilt angle from 0º to 75º. In the equation the meaning of the "+" exponent is that only positive values of the terms in the square bracket are to be used and if the value is negative, the value will be converted into zero. The additional condition for using equation (12) is the aspect ration (AR) and the Rayleigh number must be $AR \geq 12$ and $Ra_H < 10^5$.

3. Results and Discussions

3.1. Numerical Validation

The present numerical method is validated by comparing the result with previously reported studies. The selected numerical validation case is a laminar natural convection heat transfer from a square cavity heated and cooled from side walls, respectively and insulated from the top and bottom walls. The results are compared for Rayleigh number 105 and 106. Table 1 shows the numerical validation of the present result. Data of the table reveals that the present results show a very good agreement with the previous results. Thus, the present method can be used to explore the problem.

Table 1. Comparison of averaged Nusselt number

Reference	Parameter	Rayleigh Number	
		$Ra_H = 10^5$	$Ra_H = 10^6$
Present work	$\bar{Nu}$	4.518	8.879
De Vahl Davis [13]	$\bar{Nu}$	4.519	8.799
	Difference [%]	0.02%	-0.9%
Fusegi et al. [14]	$\bar{Nu}$	4.646	9.012
	Difference [%]	2.8%	1.49%
Tiwari et al. [15]	$\bar{Nu}$	4.450	8.806
	Difference [%]	-1.5%	-0.8%

3.2. Flow characteristics and heat transfer

Figure 2 shows flow characteristics of the fluid inside the enclosure. The AR of the presented enclosure is 20. In the figure temperature distribution, streamlines, and velocity vector are shown in Figure 2a, 2b, and 2c, respectively. It is shown clearly that inside the enclosure several circulation flows are generated. This circulation flow is known as "Benard cell". The mechanism can be explained as follows. The heated fluid flows upward, and before reaching the top surface, it is divided into two flows. Each flow will generate a circulation flow. This circulation will be repeated. The streamlines (Figure 2b) clearly show the number of the Bernard cell. In this case, there are 14 Benard cells generated in the enclosure. The same pattern also is shown by velocity vector. The velocity of the fluid will follow the Benard cells.

Locally heat transfer coefficient on the bottom surface is calculated and plotted in Figure 3. The horizontal axis is the distance from the left edge. As a note, the dimension of the present enclosure is 1 m $\times$ 5 cm. The figure shows that, as expected, locally heat transfer coefficient is strongly affected by Benard cells. The Benard cells divided the bottom surface into several dead zones, where the velocity almost zero or stagnant. Between two dead zones, the velocity increases and make temperature gradient higher. Heat transfer coefficient in the dead zone will decrease because temperature gradient is very low. On the other hand, the area between dead zones the local heat transfer coefficient increases and reaching a maximum value. This is because temperature gradient is higher. This fact is shown in Figure 3.

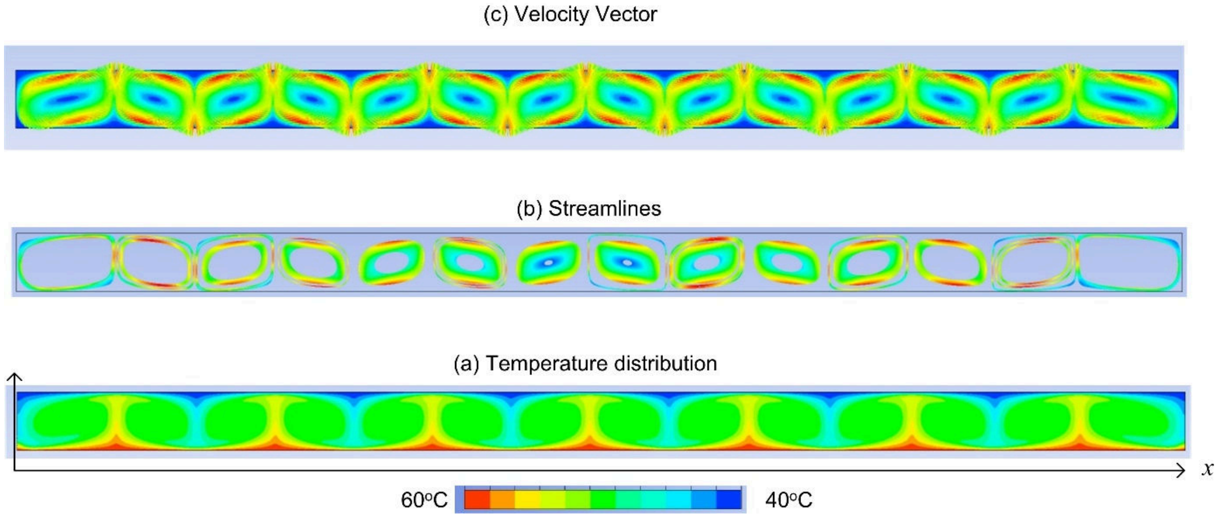

Figure 2. Flow characteristics

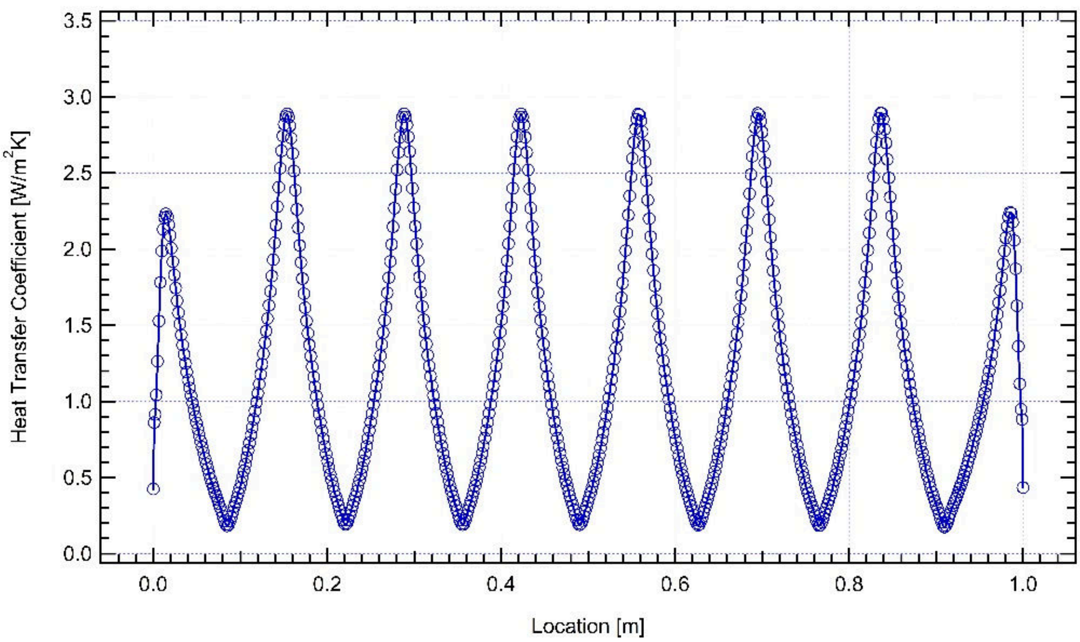

Figure 3. Locally heat transfer Coefficient

3.3. Effect of Rayleigh number

By using equation (7) to (10), the average Nusselt number is calculated. The Nusselt number resulted by empirical correlation is also estimated. The results from the present work and from the empirical estimation are presented in Table 2. The table shows the configuration, Rayleigh number, and averaged heat transfer coefficients, as well as Nusselt number and heat flux. The table shows that there is a clear discrepancy between Nusselt number resulted from the above empirical correlation and with the present results. The difference of the results from Reference [11] and the present results varies from 10.8 – 38.5%, which is lower for higher Rayleigh number. The difference with Reference

[12] varies from 3.5 – 15.2%, this also lower for higher Rayleigh number. This fact suggests that empirical correlation proposed by Jacob [11] shows significant deviation. On the other hand, Holland [12] reveals a better estimation. Thus, it is suggested to use Holland correlation to predict heat transfer rate for a double glass cover of the flat-plate solar collector.

The effect of the Rayleigh number to the performance of a double glass cover can be examined using data shown in Table 2. The Nusselt number increases with increasing Rayleigh number. However, the total heat transfer rate is decreasing.

Table 2. Results comparison

Aspect Ratio	Rayleigh Number	Parameter	Present Results	Equation (11) Ref. [11]	Equation (12) Ref. [12]
$AR = 40$	$Ra_H = 2.97\times10^4$	$\bar{Nu}$	3.545	2.560	3.078
$L = 1m$		$\bar{h}$ [W/m^{2}K]	3.432	2.478	2.979
$H = 2.5cm$		Q'' [W/m^2]	68.641	49.563	59.587
$AR = 20$	$Ra_H = 2.38\times10^5$	$\bar{Nu}$	5.589	4.305	4.871
$L = 1m$		$\bar{h}$ [W/m^{2}K]	2.71	2.084	2.358
$H = 5.0cm$		Q'' [W/m^2]	54.10	41.67	47.150
$AR = 10$	$Ra_H = 1.90\times10^6$	$\bar{Nu}$	8.02	7.241	8.321
$L = 1m$		$\bar{h}$ [W/m^{2}K]	1.94	1.752	2.014
$H = 10.0cm$		Q'' [W/m^2]	38.84	35.046	40.275

3.4. Effect of AR

The data of Table 2 shows the effect of Aspect Ratio (AR) to the performance of a double glass cover. It was shown that increasing AR would increase heat transfer rate. This is because increasing AR will reduce the distance between two glasses and make heat transfer resistance becoming smaller. In contrast, the increasing AR will decrease the Nusselt number. As a note, the Nusselt number is defined by using the distance of the double glasses. Increasing AR will decrease the distance and will decrease the Nusselt number. Since the side wall of the present enclosure is insulated or inactive in transferring heat, it is suggested to define the Nusselt number using the active wall. In this work, as well in the proposed empirical equations [11, 12], the Nusselt number is defined as the distance between double glass cover.

4. Conclusions

A numerical method using commercial code CFD to explore the heat transfer and fluid flow characteristics in the enclosure between double glasses cover of a flat-plate type solar collector has been developed. The method has been validated for a natural convection is a square enclosure heated and cooled from side walls, respectively. The validated method is then used to explore the problem. The results of the developed method are compared with empirical correlations that typically used to analyze the heat transfer in the double glass cover. The conclusions are as follows. In numerical simulation shows that in the enclosure several Benard cells are captured. The Benard cell is a circulation flow and constructs a cell-like flow. The present of the Benard cells strongly affects the local heat transfer coefficient. The average Nusselt number resulted from the present work is higher than resulted from empirical correlations. The discrepancy varies from 10.8 – 38.5% and from 3.5 – 15.2% for empirical correlation proposed by Jacobs [11] and Holland et al. [12], respectively. Based on this fact it is suggested to use the empirical correlation proposed by Holland et al. [12].

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