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Effects of Fin Height, Fin Thickness and Reynolds Number on Heat Transfer Enhancement of Flat-Plate Thermal Collector: A Numerical Analysis

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ABSTRACT

A flat-plate thermal collector combined with a PV panel also called a PV/T collector is a device that converts solar irradiation into thermal energy and electrical energy simultaneously. The unused thermal energy of the PV/T collector is absorbed by the flat-plate thermal collector. This may then contribute directly to an enhancement of the electrical efficiency of the PV/T collector. In the present study, the effect of geometry and Reynolds Number on the thermal performance of flat-plate thermal collectors is numerically investigated. Thereafter, CFD simulation is then implemented to characterize the thermal performance in terms of absorber temperature and convection heat transfer coefficient. To disturb the fluid flow pattern in this work the 45° inclined fins are attached underneath the surface of the collector model and they act as an absorber. Monthly average weather data of inlet fluid temperature and ambient temperature as well as solar irradiation level implemented in this study were obtained from the Meteorological, Climatological and Geophysical Agency of Bandarlampung regency. Several different cases have been considered by varying the fin height from 20 to 80 mm, fin thickness from 1 to 4 mm, and Reynolds number from 1500 to 6000. The results show that increasing the fin geometry (the fin thickness and fin height) and Reynolds Number reduce the flat-plate surface temperature due to the more conductive and convective heat transfer process. However, in terms of the convective heat transfer coefficient parameter, the Reynolds Number implemented has a dominant effect compared to the fin geometry. Moreover, by increasing Reynolds Number by four times, there is a decrease in the mean surface temperature by 26% and an increase in the mean convective heat transfer coefficient of 146% compared to the initial conditions.

1. Introduction

Fins are one method to enhance the heat transfer rate by extending the surface area of the solid and the contact with a fluid. The fins have been applied to various applications such as refrigeration systems, electronic devices, combustion engines, and Photovoltaic panel cooling systems [1]. Under the thermal performance of fins cooling, several kinds of research have been conducted to get

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optimal geometry and operating conditions [2, 3]. The fins can operate in both natural and forced heat convection. The more surface area of fins increases the heat transfer rate therefore more fins or bigger fins need to be applied [4-6]. In most cases, fins have size and manufacturing cost limits. Moreover, forced convection is preferred because a better heat transfer rate can be achieved by increasing the fluid flow rate under the same fin size [7, 8].

Another method to increase the heat transfer rate in forced convection is to apply a better fin arrangement to disturb flow patterns into turbulence resulting bigger convective heat transfer coefficient [9-11]. Comparative studies of various fin arrangements have been conducted. Among straight and various inclined arrangements, fins with inclined 45^{o} show the best performance. Thus, more physical and operational parameters need to be evaluated in terms of thermal performance characteristics of 45^{o} inclined rectangular fin [12]. The thermal characteristic of the finned plate-flat collector also attributes to PV/T collector development with operating temperature conditions. Besides, the electrical efficiency of the PV/T collector itself increases due to the operating temperature decrease [13-16].

Concerning the implementation of fins cooling on PV/T collectors, they are attached inside the channel to keep the fluid flow in direction [17, 18]. An experimental study on the effect of the fin ratio and channel height has been conducted, it shows that the bigger the ratio between fin and channel height the better thermal performance [19, 20].

Another physical property that affects the performance of solar thermal collectors is fin thickness. The fins contribute by implementing conduction and convection methods during the heat transfer process. The conduction occurs from the surface of the fin to the tip while the sides exchange heat by convection method. Although the thicker the fins the bigger the surface area on the tip, it is mostly neglected due to the ratio of the tip area and the sides being too small [21]. Nevertheless, temperature distribution due to conduction heat transfer has a significant impact on fin performance. Studies on fin thickness have been conducted and the results show the effect varies depending on fin types, arrangements, and applications [22]. In this context, the heat transfer enhancement for inclined fins of plate absorbers should be beneficial for many thermal device applications including PV/T solar collectors under tropical climate conditions.

Afzanizam *et al.,* [23] investigated several parametric studies including the Reynolds Number effect on the thermal performance of the PV/T collector. It is reported that in most cases, the total energy efficiency in a turbulent regime is higher, while the total exergy in laminar regime efficiency is superior. In terms of the testing under different weather conditions, Mila Ebrahim [24] studied the performance of PV/T collectors by varying the amount of irradiance, fluid inlet temperature and ambient temperature in the Stockholm climate in a Nordic Country. It is reported that various climatic conditions affect the performance of the PV/T collector. The results on the performance of the PVT collector in different weather conditions show that the inlet water temperature can significantly affect operating time and the amount of thermal energy that can be extracted during the year, especially if the collector operates in a colder climate like Sweden.

Based on previous studies, a finned flat-plate collector is advantageous in terms of the geometric and fluid flow phenomenon. However, references available are still limited in giving insight into the performance enhancement of this kind of collector, including in tropical climate applications. The present study aims to investigate the effect of fin geometry and Reynolds Number on the thermal performance of plate flat absorbers using the CFD method. Moreover, fin height and fin thickness are varied, while fin inclination is fixed at about 45^{o}. In this case, several Reynolds Numbers are kept constant, and the hydraulic diameter of the channel is set by adjustment of fin height, consequently, the air mass flow rate is then varied. The tropical climatic data collected are implemented in this work. Thus, average operating temperature and convective heat transfer coefficient are used as a thermal performance indicator under various fin heights and fin thicknesses, as well as different and constant Reynolds Number.

2. Methodology

The procedures of the simulation study are described in the flowchart below as presented in Figure 1. Firstly, the geometry of the model is designed and the meshing process is then generated by using a CFD simulation program which is validated with the reference data. Once the simulation result obtained from the validation process agrees with the previous experimental data, therefore further simulation studies can be created and analyzed deeply under tropical climate conditions. Detail procedures are then described in the following flowchart and sections.

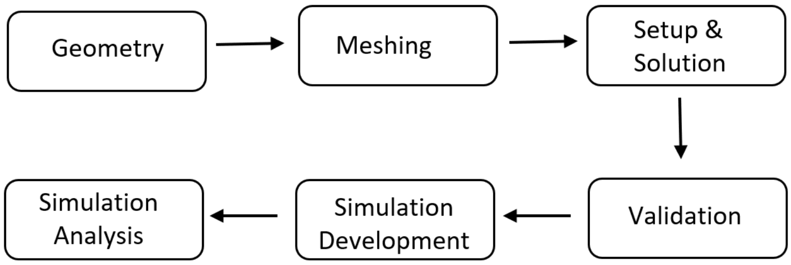

Fig. 1. Flowchart of the simulation

2.1 Model Geometry and Meshing

A model proposed in the present study was drawn using Inventor Program and was constructed by 24 fin elements attached underneath the flat-plate collector. The flat-plate absorber is made of aluminium material. Besides, it is also provided by a rectangular channel to circulate the working fluid as seen in Figure 2. The model is then run in Ansys Discovery AIM [25] to generate a mesh which is created with a tetrahedral shape as illustrated in Figure 3. While the mesh structure of the fluid domain is presented in Figure 4. To increase the stability and robustness of the simulation results, the proper grid structure is important to minimize the possible effects of bad mesh quality on the solid and fluid domains.

Moreover, to fulfil the minimum mesh number requirement, it is evaluated by using mesh independency tests. It is then found to be about 1.2845 x 10^6 as an optimal test value as presented in Table 1. Therefore, to ensure the accuracy for further simulation study the mesh number created should be higher than the value obtained from mesh-independency tests. This means that if the mesh number is less than the optimal one, the results are not acceptable in numerical simulation.

Table 1
Mesh Independency Tests

Grid Number	PV Temperature (°C)
328535	26.63
510665	30.51
718870	35.57
955698	39.10
1284500	40.17
2108692	40.23
2886322	40.64
3535866	40.40

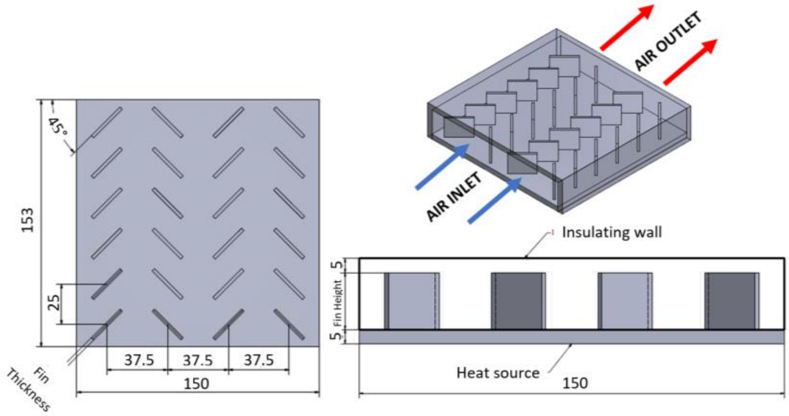

Fig. 2. Model geometry in mm

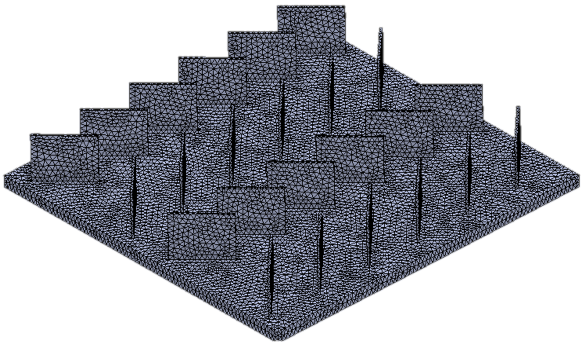

Fig. 3. The mesh structure of the solid domain

Fig. 4. The mesh structure of the fluid domain

2.2 Set Up and Solution

The mathematical model of heat and airflow in the present study is governed by several basic equations such as continuity equation, momentum conservation and energy equation. The equations consist of mathematical forms as the following [26].

Continuity equation

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

x-momentum equation

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x_j}\left[(\mu+\mu_T)\left(\frac{\partial u}{\partial x_j}+\frac{\partial u_j}{\partial x}\right)\right] \tag{2}$$

y-momentum equation

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\frac{\partial}{\partial x_j}\left[(\mu+\mu_T)\left(\frac{\partial v}{\partial x_j}+\frac{\partial u_j}{\partial y}\right)\right] \tag{3}$$

z-momentum equation

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\frac{\partial}{\partial x_j}\left[(\mu+\mu_T)\left(\frac{\partial w}{\partial x_j}+\frac{\partial u_j}{\partial z}\right)\right] \tag{4}$$

energy equation

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{\partial}{\partial x_j}\left[\left(\lambda+\frac{\mu_T c_p}{Pr_T}\right)\frac{\partial T}{\partial x_j}\right] \tag{5}$$

where ρ and cp are the density and heat capacity. While u, v and w are velocities in the x, y, and z directions, respectively. Thus, μT is the turbulent viscosity, and PrT is the turbulent Prandtl number.

Inlet fluid working temperature, ambient temperature and solar irradiation level used in this work are set as boundary conditions. Furthermore, monthly average data were obtained from the Meteorological, Climatological and Geophysical Agency of Bandarlampung regency. Moreover, Reynolds Number (Re) is kept constant by varying the hydraulic diameter (D_h) and inlet fluid velocity (V) which can be achieved by Eq. (1).

$$Re=\frac{\rho\, V\, D_h}{\mu} \tag{6}$$

The thermal result of a numerical calculation is in the form of the surface temperature of the flat plate collector. Besides, the convective heat transfer coefficient also needs to be evaluated as given by Eq. (2).

$$h=\frac{q"}{T_p-\left(\frac{T_p+T_{in}}{2}\right)} \tag{7}$$

where Dh = hydraulic diameter (m), V = velocity (m s^{-1}), ρ = density (kg m^{-3}), q"= heatflux (W m^{-2}), μ = dynamic viscosity (kg $m^{-1}s^{-1}$), T_p = flat plate temperature (C), T_{in} = inlet fluid temperature (C).

2.3 Validation and Simulation Study

To make sure the simulation numerical software generates robust value, the comparison between experimental and simulation results is then applied under the same setup conditions. On the other hand, the validation process is needed to ensure the CFD simulation program used in this work should represent the real conditions. Concerning the validation process, working fluid through the rectangular channel is described by nondimensional Rayleigh Number in the range of $4\text{x}10^7$ to $7\text{x}10^8$ as seen in Figure 5. The numerical result solved by Ansys Algorithm with a fluid-solid heat transfer template was compared with the experimental result provided by Dogan *et al.,* [19]. As seen in Figure 5 the difference in the heat transfer coefficient value obtained from this comparison is less than 10 %. Furthermore, a further simulation study is then feasible to do.

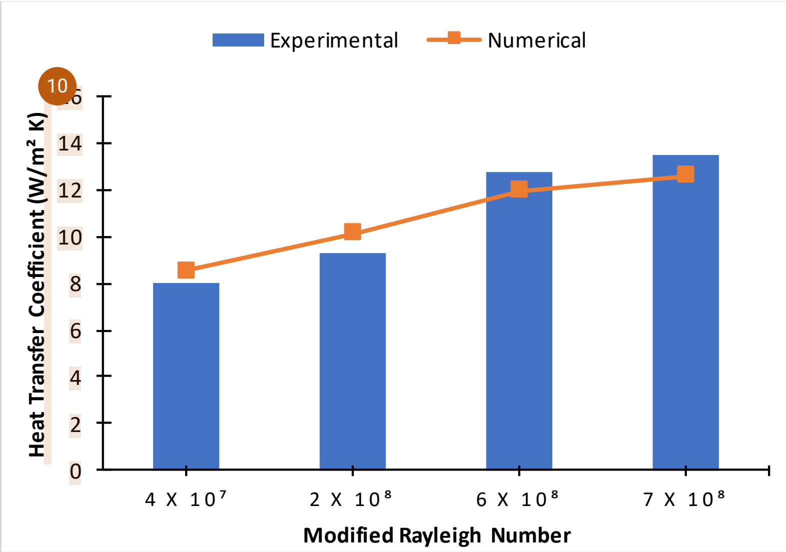

Fig. 5. Comparison of results between experimental and numerical

According to the further simulation study created in this work, each of the four different Reynolds Numbers is kept constant, while fin absorber geometry is then varied to see the effect of increasing heat transfer area. Concerning the number of size elements generated, they are varied to be 1.297×10^6 for 20 mm fin height, 1.345×10^6 for 40 mm fin height, 1.735×10^6 for 60 mm fin height, and $2{,}225 \times 10^6$ for 80 mm fin height. All of the mesh numbers used in this work are generated higher than the optimal mesh value.

3. Results and discussion

To present the detailed effect of fin geometry and Reynolds Number on heat transfer enhancement, the model collector was simulated and analyzed simultaneously. Thus, the Reynolds Number, fin thickness and fin height are varied such as (1500, 3000, 4500, 6000), (1,2,3,4) mm and (20,40,60,80) mm, respectively. Meanwhile, the tropical climates with a monthly average ambient temperature and solar radiation are found to be 30oC and 860 W/m2 respectively. Thermal performances are then parametrically analyzed based on the surface temperature of the model collector and the convective heat transfer coefficient.

3.1 Surface Temperature of Flat Plate Collector

As seen in Figure 6 shows that increasing all parameters such as Reynolds Number, fin thickness and fin height leads to decreasing the surface temperature of the model collector. Furthermore, the surface temperature enhancement is more dominantly affected by the increase in Reynold Numbers than the fin geometry effect as indicated by about 15^{o}C of the mean temperature difference between Re 1500 and Re 6000.

Meanwhile, under the same Reynolds Number implemented in this work, Re 1500 itself has the highest mean increment of temperature difference compared with the others. As shown in Figure 6a and Figure 6d, it is described that the increment of the mean temperature difference within the same Reynolds Number is found to be about 3^{o}C for Re 1500, while the lowest one is given by 1.25 ^{o}C for

Re 6000. This may be because of the low velocity of air as a fluid working to allow more time in contact with the fin absorber compared to the high velocity, therefore the mean increment of the temperature difference for the various Reynolds Numbers used here will be different from each other. Additionally, as can be seen in Figure 6d since air velocity moves quickly through the fin of the collector in the higher Reynold Numbers, therefore the convection heat transfer rate may tend to be rather the same as indicated by the graphic lines close together.

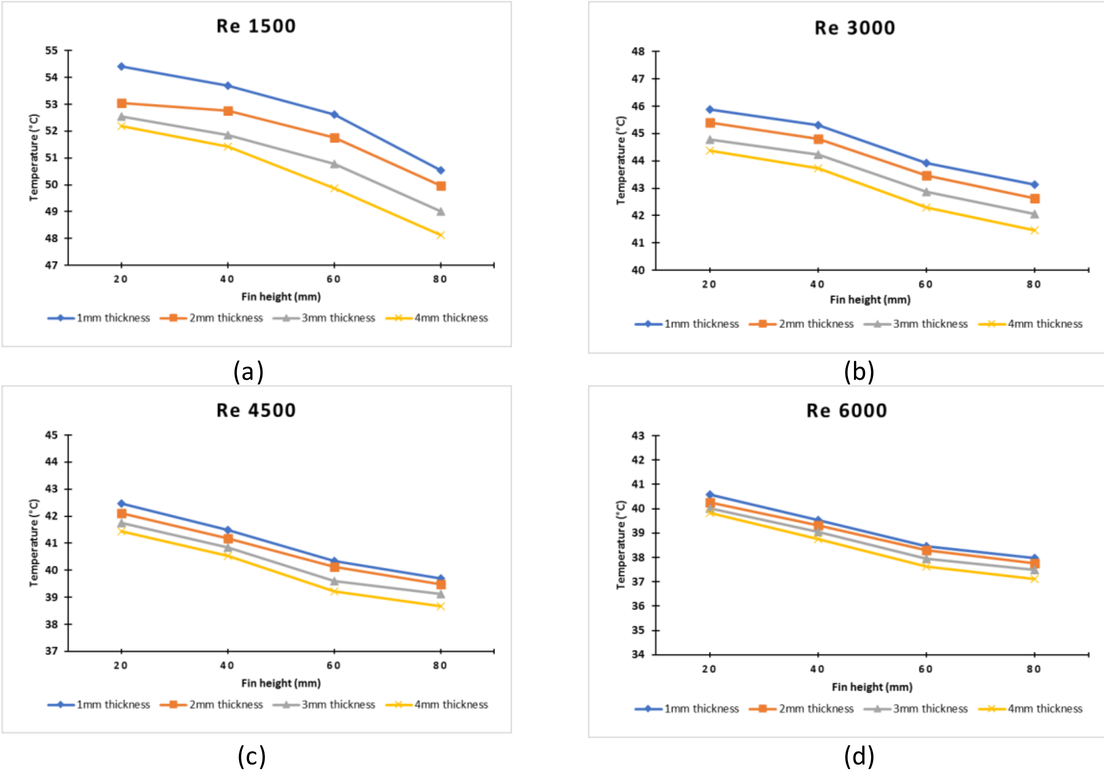

Fig. 6. Effect of fin height and fin thickness on the surface temperature within the same and different Reynolds Number

Furthermore, the surface temperature contours of the model collector with various fin thicknesses and Reynold Numbers are presented in Figure 7. The thicker the fins, the lower temperature achieved within the same Reynold Numbers as illustrated by the colour contour of the simulation result. Thus, in Figure 7d the blue colour of fin thickness of 4 mm over the inlet baseplate dominantly appear to illustrate the temperature distribution. This phenomenon may occur due to the volume of the absorber fins increasing as the fins get thicker. The heat transfer from a higher volume solid into the ambient air is attributed to the lower surface temperature and increase in the heat transfer convection from the solid surface to ambient air. Moreover, since Reynolds Number (Re) is kept constant but the inlet fluid velocity (V) is allowed to vary, for that reason, the increase in fin thickness will also lead to a change in airflow field characteristics that affect the surface temperature as seen in Figure 6-7.

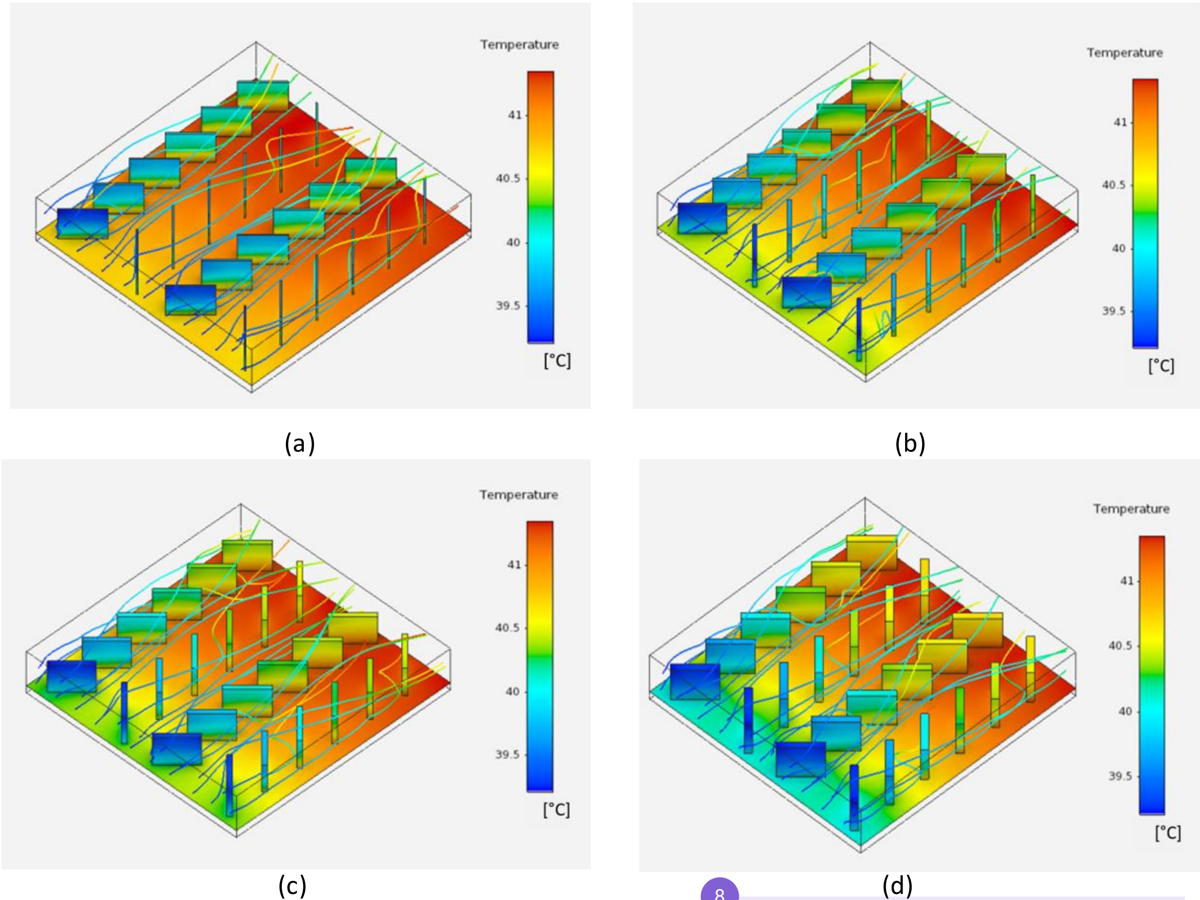

Fig. 7. Temperature distribution on different fin thicknesses (a) 1 mm, (b) 2 mm, (c) 3 mm, (d) 4 mm within Reynolds Number 1500.

3.2. Coefficient of Convective Heat Transfer

Concerning the convection heat transfer investigated in the present study, it depends on contact area, coefficient of convective heat transfer and temperature difference between a solid surface and the moving fluid. As can be seen in the trend graph in Figure 8b-d, the gradient of the convective heat transfer coefficient for Re 3000, 4500 and 6000 tends to be similar, while the convective heat transfer coefficient values are different. If Reynolds Number is increased by four times, there is an increase in the mean convective heat transfer coefficient of 146% compared to the initial condition. Furthermore, there is also an increase in the mean convective heat transfer coefficient of 28% for Reynolds Number from 3000 to 4500 as depicted in Figure 8b-c.

In addition, the higher the Reynolds Number, the gradient of the convective heat transfer coefficient graph has a sharper tendency than that of Reynolds Number 1500. However, the graph tendency between Reynold Numbers 4500 and 6000 gives a little different as seen in Figure 8c-d even though the convective heat transfer coefficients of the two Reynold Numbers are completely different. This may be due to the slight thermal effect, especially under the higher Reynold Numbers conditions that have less time to transfer the heat compared with the lower airflow. In terms of fin geometry and Reynolds Number effects, it shows that increasing both the Reynolds Number and fin thickness parameters leads to an increasing coefficient of convection heat transfer. However, it is not followed by the effect of increasing the fin height in which the coefficient of convection heat transfer tends to be conversely decreased. This may be because the Reynolds Number is kept constant in each of the simulation processes as shown in each Figure 8a-d. Therefore the velocity of the working fluid may be different depending on the fin height used. To achieve the constant value of each Reynolds Numbers, the air velocity is allowed to vary by adjusting the cross-section area of every channel under different fin heights. Consequently, the increasing cross-section area of the channel leads to a specifically decreasing air velocity and therefore it has a low impact on the coefficient of convective heat transfer.

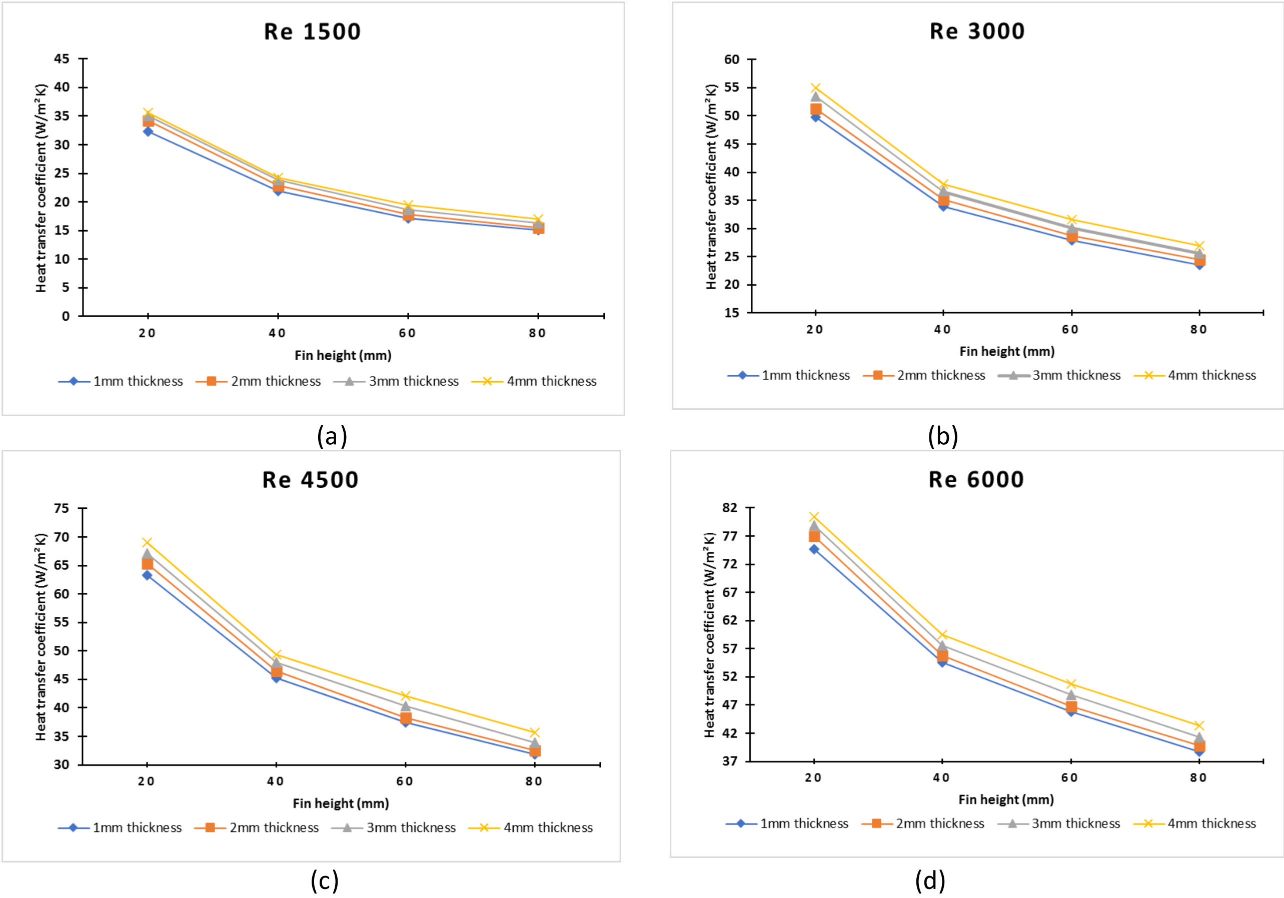

Fig. 8. Effect of fin thickness and fin height on convective heat transfer coefficient within the same and different Reynolds Number

In addition, even though the effect of increasing the fin height on the convective heat transfer coefficient is decreased, however, increasing the fin height conversely enhances the surface temperature as described in section 3.1. This means that the extended surface area by increasing the fin height has a significantly more dominant effect than that of the constant Reynolds Number effect on the surface temperature due to different air velocities.

4. Conclusions

The effects of fin geometry and Reynolds Number on heat transfer enhancement have been numerically studied. Several different fin heights and fin thicknesses with four different constant Reynolds numbers were simulated to investigate the thermal performance of the flat-plate collector such as the surface temperature of the collector absorber and convective heat transfer coefficient.

The results show that increasing the fin geometry such as the fin thickness and fin height as well as Reynolds Number reduce the flat-plate surface temperature due to the more conductive and convective heat transfer process. However, in terms of the convective heat transfer coefficient parameter, the Reynolds Number implemented has a dominant effect compared to the fin geometry. Moreover, by increasing Reynolds Number by four times, there is a decrease in the average surface temperature by 26% and an increase in the average convective heat transfer coefficient of 146% compared to the initial conditions. It is described that the increment of the mean temperature difference within the same Reynolds Number is found to be about 3°C for Re 1500, while the lowest one is given by 1.25 °C for Re 6000.

To assess the created model collector as proposed in the present study, further deep work leads to the experimental testing for a PV/T air collector integrated with cooling fins design, TEG etc. The result should provide a valuable reference regarding the optimal utilization and development of PV/T and thermal collectors in tropical climates.

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