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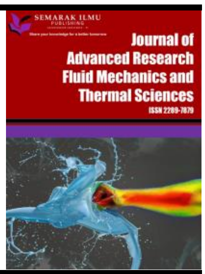

The Phenomenon of Flow and Heat Transfer in Annular Heat Exchanger on Plain Tube Condition

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ABSTRACT

Advanced industrial development in technology necessitates using equipment that can effectively and efficiently transfer heat energy. A heat exchanger is the most common device used in industry to transfer heat energy. The Annular Heat Exchanger is one of the heat exchangers used in industry that is simple to manufacture and inexpensive to finance. The annular heat exchanger is commonly found in the food, beverage, chemical process, pharmaceutical, and refrigeration technology industries. This type of heat exchanger is used due to its simple design, easy to manufacture, and low cost. The presence of a heater in a concentric pipe with a straight construction is the general form. Experimentation was carried out in this study. A heater is used to transfer the heat, and there is a cold fluid flow through the annulus. The heater is linked to an AC Voltage Regulator and has five heat settings ranging from 400 W to 600 W. Simultaneously, cold fluid flows in the annulus via a closed system with six flow variations ranging from 2.5 GPM to 5 GPM. A cooling system is used to keep cold fluids at a constant temperature. The purpose of this research is to demonstrate the flow and heat transfer phenomena in an annular heat exchanger. This study obtained the average heat transfer convection coefficient, *h*, and *Nusselt* number, *Nu*. The steady-state conditions for the heat variations of 400 W, 500 W, and 600 W are 2400 s, 3600 s, and 5400 s, respectively. The flow calibration results in this apparatus experiment have an error of 0.05%. At a heat variation of 500 W and Reynolds number 10669, the highest average heat transfer convection coefficient (*h*) reaches 6703 W/m^{2}K, and the average *Nusselt* number (*Nu*) of 314 is obtained. The *Reynolds* number of critical transition points for heat variations of 400 W, 450 W, 500 W, 550 W, and 600 W are 6986, 7125, 8444, 8616, and 10009, respectively.

1. Introduction

The performance of a heat exchanger is essential for the industrial world. Both those related to the chemical process industry, food and beverage industry, oil and gas, pharmaceuticals, and refrigeration that use heat exchangers require performance [1]. The current use of heat exchangers

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considers the heat transfer coefficient, pressure drop, and the effectiveness. Enhancing the heat transfer through a heat exchanger is one technique to improve performance.

Heat transfer enhancement can be achieved by increasing the heat transfer convection coefficient (h) and the surface area coefficient [2]. The heat transfer convection coefficient can be increased by producing eddies through inserts, such as twist tape inserts, double-sided delta-wing tape inserts, and louvered strips insert [3-7]. The insert contains a thin strip as a flow barrier to increase turbulent flow. The results of the insert experiment revealed that heat transfer increased as the *Reynolds* number increased. At the same *Reynolds* number, the heat transfer rate generated by the tube with the insert increases.

The flow can also be manipulated in the heat exchanger’s annulus section. The presence of a perforated baffle installed in the heat exchanger annulus increases the average Nusselt number and overall heat transfer coefficient [8]. The same goes for research on the effect of spacing baffles [9]. Research using a perforated turbulator and a discontinuous helical turbulator was also carried out on a water-to-air heat exchanger [10,11]. This study’s findings also show a higher heat transfer coefficient. Using helical fins and vortex generators can also increase the heat transfer coefficient and convection surface area [12]. All of these indicate the occurrence of enhanced heat transfer in the heat exchanger.

Other methods for improving heat transfer include changing the working fluid. The most used working fluids are water, air, and ethylene glycol. Using a solution of Carboxyl methyl cellulose (CMC) or pseudoplastic fluid as a working fluid was carried out experimentally by Moradi *et al.,* [13]. An increase in CMC causes a decrease in viscosity, increasing the heat transfer rate.

Another researcher investigated heat transfer in the inner tube of a heat exchanger utilizing supercritical carbon dioxide (SCO_2) [14]. According to the study, increasing SCO_2 pressure lowers the total heat transfer coefficient, which lowers the overall heat transfer rate. Wu *et al.,* [15] and Zamzamian *et al.,* [16] investigated the use of alumina as a working fluid. Where there is a 0.37 percent to 3.43 percent increase in heat transfer, and 2 percent to 50 percent increase in the heat transfer convection coefficient [15,16]. Hosseini *et al.,* [17] developed the Clove-Treated Multi-Walled Carbon Nanotube (C-MWCNT), which increases thermal conductivity. Hosseini conducted research using an annular heat exchanger, which previous researchers rarely did.

The existence of research using inserts on the inner pipe opens opportunities for further research. As with inserts made of thin strips, they also have the same conditions as inserts. It's just that the term “outsert” is given because its position is placed in the annulus. To the best of author's knowledge, the use of outsert in annular heat exchangers has not been thoroughly investigated, nor have the flow and heat transfer characteristics that occur. So, it is indispensable to show the flow and heat transfer phenomena in the annular heat exchanger.

2. Methodology

2.1 Experimental Setup

The experiment was carried out using the test equipment depicted in Figure 1. This test equipment supplies a tank filled with water to the test section using a pump with a closed system. Water flows through a rotameter, which is used to measure the flow of water. A bypass valve is used to control flowing water flow back to the reservoir tank. Water through the inlet enters the annulus test section and exits through the outlet. After taking off the outlet, the water returns to the reservoir tank after going through a cooling system, such as radiator and fan.

There is a heater along with the test rig in the test section, which is used to heat running water. The heater is connected to the AC Voltage Regulator to get the variation of heat. The wattage used by the heater is shown on a display. In the test section, 7 thermocouples are installed, where 2 thermocouples are installed at the inlet and outlet. At the same time, the other 5 thermocouples are installed along the test section. Thermocouples are installed to determine the temperature inlet and outlet, also surface temperature at 5 points along the test section. In the test section, a pressure transducer is also implemented to measure the pressure drop during the experiment. A data logger is used to record the experiment data connected to a computer to store the data results.

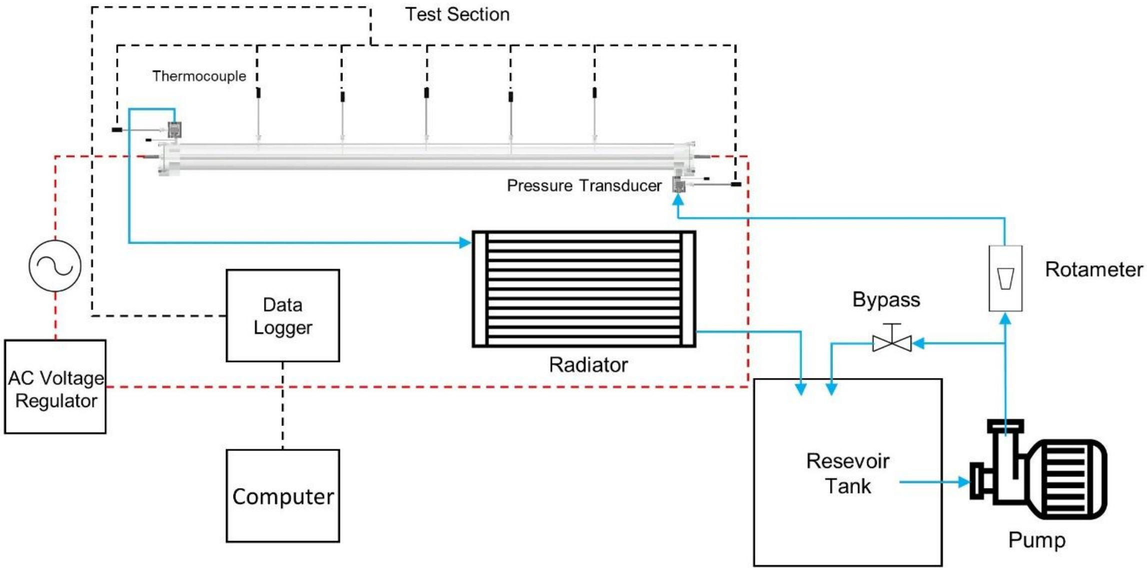

Fig. 1. Schematic diagram of the test rig

A shallow well water pump with a maximum total head specification of 33 meters and a maximum capacity of 24 liters/min is used to drain water from the reservoir tank to the test section and back to the reservoir tank through the radiator. The flow rate is measured using an H_2O Omega rotameter with a capacity of 0.5 to 5 GPM.

A cooling system, consisting of a radiator and a fan, is used to cool the hot water passing the heater. A tubular heater is used to heat the flowing water, which is placed in the center of the test section. The tubular heater is turned on using an AC Voltage Regulator connected to both ends of the tubular heater to make the desired heat variation. A display shows the value of the voltage, current, power, and power-factor used during the experiment.

The thermocouple used to measure the temperature in this experimental apparatus is a type K thermocouple of 7 units. The arrangement of the thermocouple placement is at the inlet and outlet points and 5 points along with the heater with sizes of 180 mm, 340 mm, 500 mm, 660 mm, and 820 mm from a distance to the inlet point. This thermocouple measures the surface temperature of the heater's outer wall. While the pressure sensor used to measure pressure is Autonics PSAN-C01CV. The goal is to get the pressure drop value in the annular heat exchanger. A Graphtec midi logger GL820 data logger was used to record data throughout the experiment, which was connected to a computer to store all experimental data.

Detailed dimensions of the test section can be seen in Table 1. In the test section, an annular heat exchanger with a length 1000 mm is used. The material used in the test section uses a see-through pipe made of Plexiglas with a 40 mm inner diameter and a 50 mm outer diameter. In the centreline of the pipe is placed a heater with a diameter of 11 mm made of cooper. This heater has a power of up to 700 W at 220 V voltage.

Table 1
Dimensions of the test section

No.	Items	Dimension (mm)
1	Outer pipe length	1000
2	Outer pipe inner diameter	40
3	Outer pipe outer diameter	50
4	Heater length	1000
5	Heater diameter	11

To obtain the phenomenon that occurs on the plain tube condition in this annular heat exchanger, flow variations of 2.5 GPM, 3 GPM, 3.5 GPM, 4 GPM, 4.5 GPM, and 5 GPM are used on the Omega rotameter. This variation is obtained by rotating the bypass valve so that these 6 flow variations can be carried out. Meanwhile, to get the effect of the heat given in this experiment, 5 heat variations were carried out by turning the AC Voltage Regulator at 400 W, 450 W, 500 W, 550 W, and 600 W. A wattmeter display shows the magnitude of values such as voltage, current, power, and power-factor.

2.2 Experiment Procedure

In this experiment, the test is focused on where the heater is in plain tube conditions. This condition is where the heater is really in a plain condition without using an outsert as a flow repellent so that it becomes turbulent. All data is stored in steady-state conditions.

The experiment was started by starting the pump. The pump is turned on to see the condition of leakage in the experimental apparatus and water fluid that runs in a closed cycle. After ensuring no leaks in the entire experimental apparatus, the instrument is then calibrated.

The following are the equations for single-phase forced convection in this experiment. The properties of the working fluid in this experiment can be seen from the mean temperature obtained from the equation.

$$T_{m,f} = \frac{T_{f,i}+T_{f,o}}{2} \tag{1}$$

The surface temperature of the heater is obtained in Eq. (2).

$$\bar{T}_s = \frac{\sum T_{s,n}}{n} \tag{2}$$

The mass flow rate can be calculated using Eq. (1) as:

$$\dot{m} = \dot{V} \times \rho \tag{3}$$

The *Reynolds* number value is obtained through the equation.

$$Re = \frac{\rho v D_h}{\mu} \tag{4}$$

The heat transfer rate can be calculated through the equation.

$$q = \dot{m} c_p (T_{in} - T_{out}) \tag{5}$$

Then, the heat transfer convection coefficient (h) can be calculated with Eq. (6) as:

$$h = \frac{q}{A(T_s - T_f)} \tag{6}$$

Furthermore, the *Nusselt* number (Nu) value is obtained through the equation.

$$Nu = \frac{hD_h}{k} \tag{7}$$

3. Results

Steady-state conditions in this experiment can be seen in Figure 2. With different heat variations, the surface temperature distribution on the heater reaches steady-state conditions, resulting in different steady-state times. Conditions with the heat of 400 W reach steady-state conditions in 2400 s, faster than the others. The heat variation of 500 W to reach steady-state conditions takes 3600 s, longer than the 400 W of heat variation but faster than the 600 W. Meanwhile, at 600 W of heat variation, it takes longer to reach steady-state conditions in 5400 s.

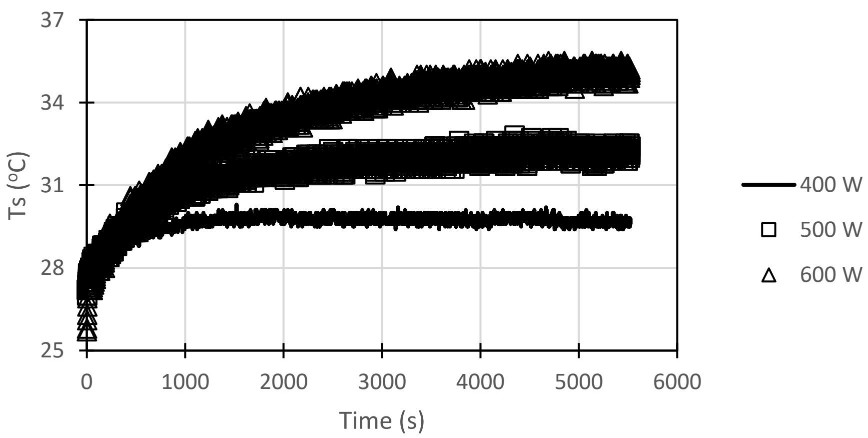

Fig. 2. Surface temperature (T_s) heater distribution

The results of the flow calibration conducted by this apparatus experiment can be seen in Figure 3. The test was carried out on 4 flow variations, namely at 3 GPM, 3.5 GPM, 4 GPM, and 4.5 GPM. From the figure, it can be described that there is a flow calibration. The Q_{actual} value has a value that is not significantly different from Q_{read}. It explains that experiments can be conducted well on this apparatus. From this figure, the flow calibration results in this apparatus experiment have an error of 0.05%, where this value shows exceptionally satisfactory results to be continued in the following study.

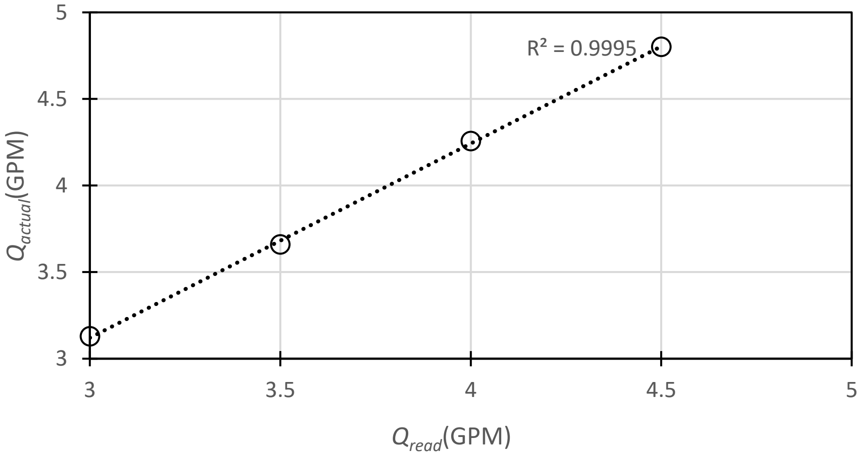

Fig. 3. Flow calibration

Figure 4 depicts the experimental results for the average heat transfer coefficient (h), taken at one heat variation of 550 W. The result of the tests revealed that as the *Reynolds* number increased, so did the average convection heat transfer coefficient (h). The findings of this experiment were also compared to those of Hosseini *et al.,* [17]. The average convection heat transfer coefficient (h) from the experiment shows the same thing in Hosseini. The value increases as the *Reynolds* number increases.

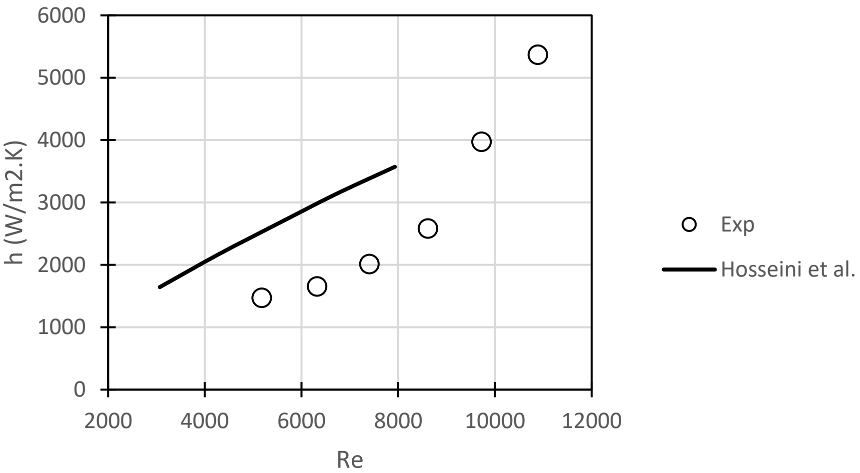

Fig. 4. Comparison of the average heat transfer convection coefficient (h) with other studies

This increase can be caused by the influence of the velocity distribution in the boundary layer. As shown in the image description, the shape of the boundary layer increases the average convection heat transfer coefficient (h). However, the difference lies in the trends generated in these two experiments. Hosseini's experimental results show a trend of increasing convection heat transfer coefficient (h), which is more linear on average. Meanwhile, the result of this experiment shows a significant increase in the average heat transfer convection coefficient (h) beginning at *Reynolds* number 8616. The result also can be seen in the average *Nusselt* number (Nu), which experienced similar results.

The experimental results showing the average heat transfer coefficient (*h*) with the overall heat rate variation can be seen in Figure 5. This figure shows heat variations of 400 W, 450 W, 500 W, 550 W, and 600 W. The experimental results show that the heat variation of 500 W has greater effect on the average heat transfer convection coefficient (*h*) than the other variations. The heat variation of 450 W, on other hand, causes a smaller increase in the average heat transfer convection coefficient (*h*) than the other variations. At a heat variation of 500 W, the highest heat transfer convection coefficient (*h*) is obtained, which can reach 6703 W/m^{2}K at *Reynolds* number 10669. Meanwhile, the lowest is obtained at a heat variation of 450 W, at 761 W/m^{2}K for *Reynolds* number 4956.

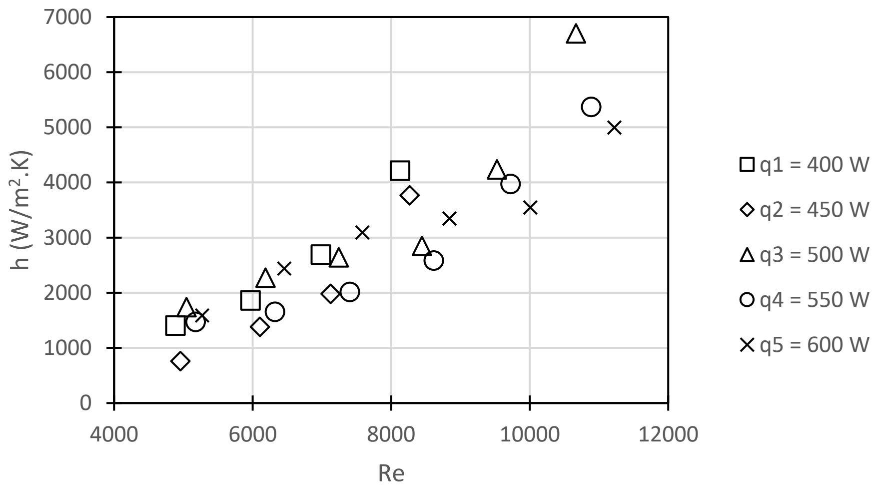

Fig. 5. The result of average heat transfer coefficient (*h*) with heat variations

All experimental results are validated with a reliable empirical correlation. One of them is the *Petukhov* and *Gnielinski* correlation.

$$Nu = \frac{(f/8)R_e P_r}{1.07+12.7(f/8)^{1/2}\left(P_r{}^{2/3}-1\right)} \tag{8}$$

Which $\mathrm{f} = (0.790 \ln \mathrm{R_e} - 1.64)^{-2}$
With $3000 \leq R_e \leq 5 \times 10^6$

$$Nu = \frac{(f/8)(R_e-1000)P_r}{1+12.7(f/8)^{1/2}\left(P_r{}^{2/3}-1\right)} \tag{9}$$

The experimental results that show the average *Nusselt* number (*Nu*) value obtained at a heat rate variation of 550 W can be seen in Figure 6. The result of the test revealed that the average *Nusselt* number (*Nu*) value increased in tandem with the *Reynolds* number. The experiment compared empirical correlations that other researchers commonly use, such as the *Petukhov* and *Gnielinski* correlation. The average *Nusselt* number (*Nu*) value resulting from the *Petukhov* and *Gnielinski* correlation shows the same things. The value increases as the *Reynolds* number increases.

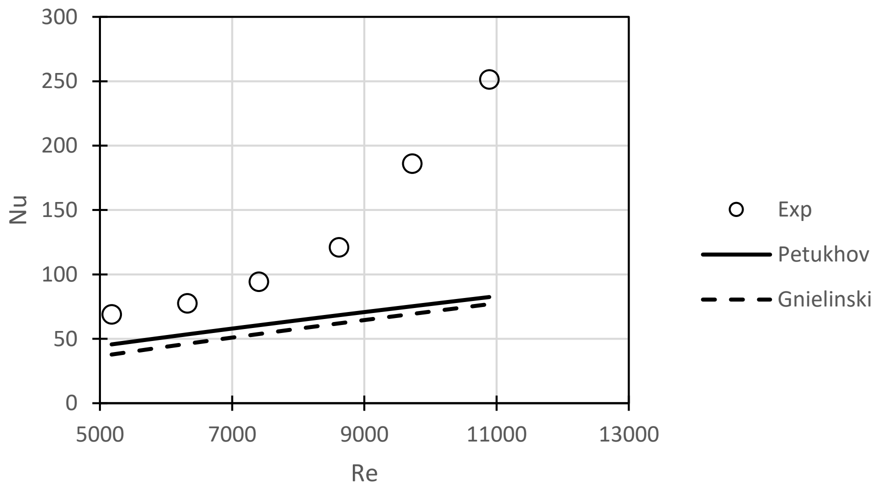

Fig. 6. Comparison of the average *Nusselt* number (*Nu*) to the empirical correlation

The increase in the average *Nusselt* number (*Nu*), like the increase in the average heat transfer convection coefficient (*h*), could be attributed to the influence of velocity distribution on the boundary layer. As shown in the figure description, the shape of the boundary layer increases average heat transfer. However, the difference in the experimental results on the average *Nusselt* number (*Nu*) is the increase generated by the experiment and the empirical correlation, which has a different trend. The increase in the empirical correlation increases linearly but not significantly. The increase in the experimental results on the average *Nusselt* number (*Nu*) has a different trend.

The experimental results show a deflection, with the difference in the average *Nusselt* number (*Nu*) beginning at *Reynolds* number 8616, which has increased significantly more than the *Reynolds* number below 8616. According to Everts and Meyer [18], these results indicate flow conditions in laminar and transitional states. The critical point of the transition starts at *Reynolds* number 8616. Figure 7 depicts the critical point more clearly. According to the average *Nusselt* number (*Nu*) and *Colburn j*-factor, the transitional flow regime began at *Reynolds* number 8616.

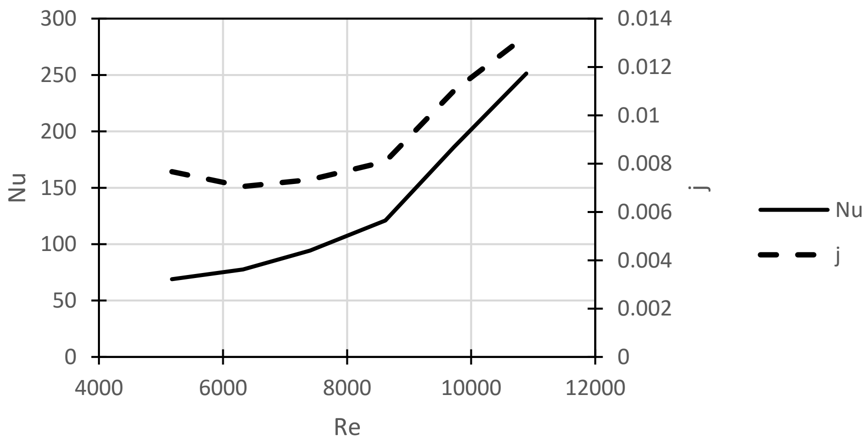

Fig. 7. The average *Nusselt* number (*Nu*) with the *Colburn j*-factor

The overall result of the average *Nusselt* number (*Nu*) with various heat variations can be shown in Figure 8. The increase in all the average *Nusselt* number (*Nu*) of heat rate variation follows the same pattern. At the heat variation of 500 W and *Reynolds* number 10669, the highest average *Nusselt* number (*Nu*) reaches 314. Furthermore, the lowest of the *Nusselt* number (*Nu*) is 35 at the heat variation of 450 W and *Reynolds* number 4956. There are deflection and the difference increase in the average *Nusselt* number (*Nu*) in all heat variations. These results indicate transformation from the laminar flow to the transition with different critical points. At a heat variation of 400 W, the critical point of the transition starts at *Reynolds* number 6986.

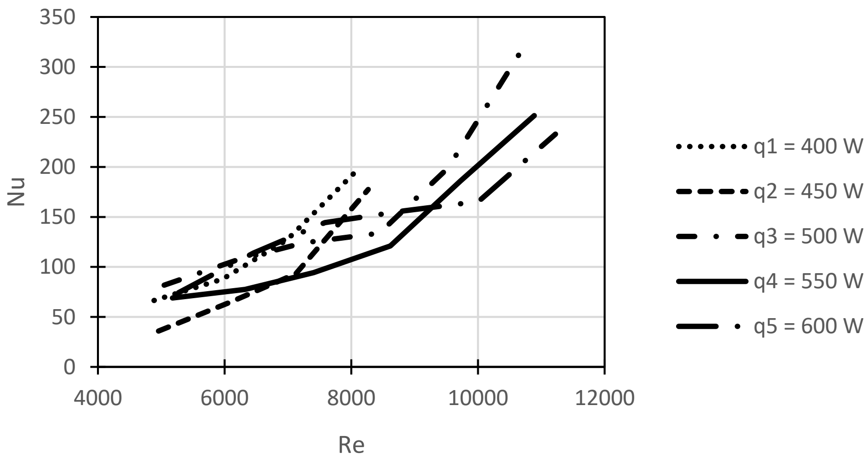

Fig. 8. The result of the average *Nusselt* number (*Nu*) with heat rate variations

Meanwhile, in a heat variation of 450 W, the critical point of transition begins at *Reynolds* number 7125. In a heat variation of 500 W, the critical transition point starts at a *Reynolds* number greater than 400 W and 450 W, at the *Reynolds* number 8444. Likewise, the more considerable heat variations such as 550 W and 600 W have a critical transition point starting at the more significant *Reynolds* number, namely 8616 and 10009. These results show that the greater the variation of heat, the *Reynolds* number value of the critical transition point is getting bigger.

4. Conclusions

The experiment described the phenomenon of heat and heat transfer in an annular heat exchanger under plain tube conditions in this paper. Heat variations are more noticeable because it takes longer to reach a steady-state condition. The steady-state conditions for the heat variations of 400 W, 500 W, and 600 W are 2400 s, 3600 s, and 5400 s, respectively. The flow calibration results in this apparatus experiment have an error of 0.05%. The experiment result shows that as the *Reynolds* number increases, so does the average heat transfer convection coefficient (*h*). Similarly, as the *Reynolds* number rises, so does the average *Nusselt* number (*Nu*). At a heat variation of 500 W and *Reynolds* number 10669, the highest average heat transfer convection coefficient (*h*) reaches 6703 W/m^2K, and the average *Nusselt* number (*Nu*) of 314 is obtained. Heat variations, on the other hand, did not significantly increase the average heat transfer convection coefficient (*h*) or the average *Nusselt* number (*Nu*). Heat variations have the potential to transform the *Reynolds* number at a critical point of transition from laminar to transition flow. The higher the heat variation, the higher the critical transition point's the *Reynolds* number value. The *Reynolds* number of critical transition points for heat variations of 400 W, 450 W, 500 W, 550 W, and 600 W are 6986, 7125, 8444, 8616, and 10009, respectively. Future research on outsert in annular heat exchanger will be linked to the entire flow and heat transfer phenomenon.

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