INVESTIGATION OF EFFECTS ON HEAT TRANSFER AND FLOW CHARACTERISTICS OF Cr-Ni ALLOY AND ALUMINUM PINS PLACED IN AISI 304 TUBE

by

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In this study, the effects of cylindrical aluminum and Cr-Ni alloy pins placed in different arrangements on the inner wall of the pipe in the turbulent flow, the effects of heat transfer and flow characteristics on different Reynolds numbers have been experimentally investigated. The experiments were carried out under forced flow and constant heat flow conditions. Air is preferred as the fluid and the fluid velocity is adjusted between Reynolds number of 10000 and 50000. It has been observed that the Nusselt values obtained over the number of Reynolds number for the 5 different test tubes are arranged in a line from large to small, sequential row aluminum pin, sequential row Cr-Ni alloy pin, diagonal row aluminum pin, diagonal row Cr-Ni alloy pin, plain tube. There are also CFD analysis for each material, arrangements and pins geometry sets. On the other hand, it was determined that friction coefficient is directly proportional to the increase of heat transfer coefficient. As a result, it is observed that experimental results are compatible with both literature and numerical study.

Key words: *convection heat transfer, heat transfer enhancement, aluminum and Cr-Ni pins, turbulent flow in tube, friction and pressure losses in channels.*

Introduction

Heat exchanger systems with different surface geometries are used, such as energy conversion facilities, houses, chemistry, and food industry. Nowadays, various methods are used to improve the heat transfer and the efficiency of these heat exchangers. The most commonly used method is the passive method in which different types of plugging elements are used in the pipe. For this reason, numerous numerical and experimental studies have been carried out to improve energy efficiency and heat transfer by reducing operating costs. Generally, in these studies, investigators have studied on heat transfer and flow characteristics with the various elements that placed in and into the channel. Darici [1], have experimentally investigated the effect of an occluding element in the inlet, which is placed in the inlet, on the heat transfer to a pipe heated by using the walls as resistance, in a constant surface heat flow boundary condition and in turbulent air-flow. Sara *et al.* [2] investigated heat transfer and pressure differentials by placing rectangular section perforated pipes in a rectangular tube. When this research was

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carried out, they made different hole diameters and plate numbers. Gunes [3] experimentally investigated the effect of heat transfer and flow characteristics of helically wound wires placed in the pipe in operation on the Reynolds number range of 3514-27188. We observed that the results obtained from the plain tube experiments are consistent with the literature results. He carried out experiments by placing helical wires on the pipe. Abou-Ziyan *et al.* [4] studied the effects of concentric ring-shaped particles on heat transfer and pressure drop. They have made their experiments for hollow pipes and three winged pipes with helical spacing. They worked between the range of Re =1428-3008. Eren and Caliskan [5] experimentally investigated the Nusselt number and friction values using cylindrical pins and triangular pins in a rectangular channel. They prefer air as the fluid and the pins are positioned radially in the channel. When the temperature was detected, they used an infrared thermal camera. Yang *et al.* [6] studied total heat absorbing bulk, pin material volume and pressure drop. They tried to determine the optimum number of pins for this study. Kirsch and Thole [7] used four different pin arrays in their research. They performed pressure loss and heat transfer measurements at different Reynolds values. They found that the pressure losses due to friction were high. El-Sayed *et al.* [8] experimentally examined the heat transfer and flow profile for laminate flow by placing blades in different rows, numbers and positions in the pipe. As a result, values of pressure drop along a pipe with nonstop fins are higher than that of the in-line arrangement, and lower than that of the staggered arrangement. The values of the average Nusselt number for with the staggered array fins are lower than that of in-line array fins, at high Reynolds numbers. Huq and Aziz-ul Huq [9] have experimentally examined the effects of eight axial fins placed in the tube on heat transfer for turbulent flow regime. As a result, they observed that the heat transfer increased by 97-112% compared to the plain tube. Hsieh *et al.* [10] experimentally, they investigated the heat transfer in the developing turbulent flow regime by placing strip elements into a horizontal circular channel. The study was studied between Re = 6500-19500, Gr = 0-108. As a result, they found that heat transfer optimization in circular channels with strip elements is 2-3 times more than in plain tube. They stated that the results were comply with similar study finding in the literature. Alam and Ghoshdastidar [11] studied the heat transfer in a circular cross-section pipe with four different fins. It is a flow laminator and has applied constant heat flux to the test pipe. In this study, they used finite differences method. Heat transfer coefficient and viscosity change with temperature took into account. They evaluated energy and momentum equations for laminar flow in the pipe separately for seven different situations. As a result, they observed that the outer fins of the channel provide better heat transfer improvement. In his study, Gurlek [12] examined heat transfer enhancement by placing turbulators in different geometries and sizes inside the pipe. In his experimental and numerical study between Re = 7000-15000 for 130, 150, 200, and 250 mm pitch and three different models, he investigated the effect of flat type turbulators positioned in concentric tubes on heat transfer and pressure losses. During solid modeling of turbulators, Solidworks program, Gambit program for the creation of network structure, and ANSYS FLUENT program for the analysis part. According to numerical analysis, the heat transfer and flow characteristics of the turbulators were visualized by different factors such as pitch, model, fluid velocity, and temperature. The maximum number of Nusselt number and friction values obtained in *V*-type turbulators with 130 mm pitch ratio were obtained in the results obtained. Yakut [13] have done their study using fins placed in a heat exchanger. In this study, they performed the measurements of the fins according to the type of rotation and the distance between the fins. They observed the changes in the heat transfer and pressure difference in the measurements. Hiravennavar *et al.* [14] performed measurements by placing fins in different geometries into a channel. While doing this, they changed the fin thickness and measured it. This study has observed improvement in heat transfer. Sertkaya [15] experimentally investigated the convection and radiation and heat transfer of pins placed at different angles in the pipe. As a result, it observed that the heat transfer increased significantly by increasing the number of pins and the angle of pins compared to the plain pipe. Bilen *et al.* [16] numerically examine the effect of aluminum porous fins placed in a rectangular duct on heat transfer and pressure drop, at 5000 < Re < 35000. It was seen that heat transfer enhancement decreased with increase in the Reynolds number as well as increase in C/H. Zeitoun and Hegazy [17] is submitted for fully developed laminar convective heat transfer in a tube provided with internal longitudinal fins, and with uniform outside wall temperature. The fins are arranged in two groups of different heights. The use of different fin heights in internally finned pipes enables the enhancement of heat transfer at reasonably low friction coefficient. On the other hand, CFD analysis are made at FLOEFD which is a 3-D CFD software delivered by Mentor Graphics®.

In this study, the effects of aluminum cylindrical and Cr-Ni alloy pins placed in different arrangements on the inner wall of the pipe in the turbulent flow, the effects of heat transfer and flow characteristics on different Reynolds numbers have been experimentally investigated.

Material and method

Experimental set-up

The experimental set-up used in the study is schematically shown in fig. 1. The test set-up consists of a three part flow pipeline consisting of inlet, test and outlet and various measuring instruments on it. In all three sections, AISI 304 L grade steel pipes with an internal diameter of 70 mm and a wall thickness of 3 mm were used. The inlet pipe is designed to be 30 diameters long to transfer the air sucked in 220/380 V, 0.75 kW, 2800 rpm and 1.8/3.1 a fencer to the test tube in the advanced flow.

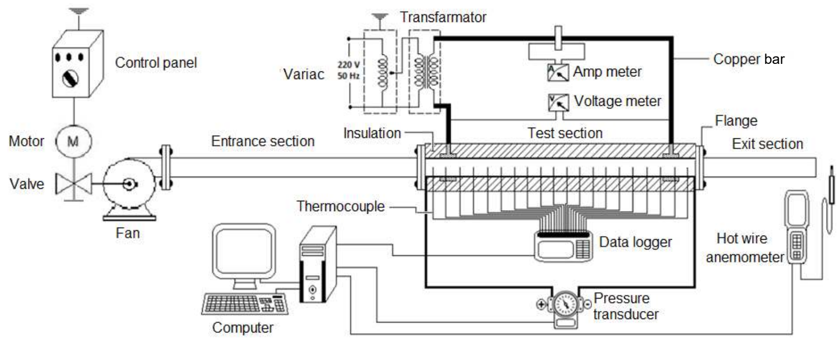


Figure 1. Schematic representation of the test set-up

The gate valve and the electric motor speed controller installed before the plant are adjusting the air-flow through the pipe. In addition, to prevent vibration caused by the fan motor, the fan inlet pipe connection is provided by a hose made of flexible rubber material. Bakelite gaskets are placed between the flanges to reduce heat loss at the inlet and outlet of the test tube and to provide airtightness. The 15 diameters were taken to provide thermal development of the heated air in the test pipe. In addition, the outer surface of the test tube is insulated with heat insulation material (glass wool) to reduce heat loss. The test tube is heated by direct electrical energy to the tube. The heater circuit consists of 2 kW variac, amperemeter and voltmeter with measuring range 0-1000 A and 0-1 V, respectively. The current is supplied by thick Cr-Ni alloy bars attached to the inlet and outlet of the test probe. Thus, a homogeneous heat flow distribution

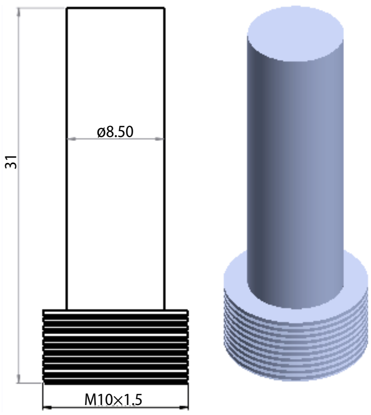


Figure 2. Aluminum and Cr-Ni pin geometry

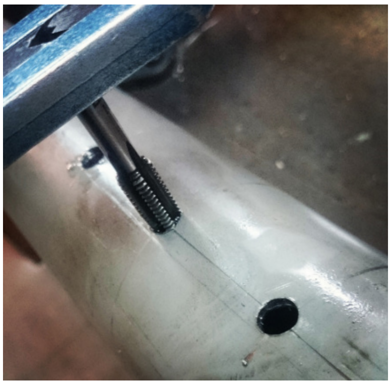


Figure 3. Tap in the test tube

is provided at each end of the test tube surface. In this experimental set-up, the effects of heat transfer and friction losses on the pins placed in the circular channel inner wall were investigated. Experiments were first carried out with the plain tube. Then the same procedure was repeated for the other test tubes. Four holes were drilled in 900 around the pipe, 20 holes in one row for the test pipe. Thus, a total of 80 holes were drilled on the pipe at equal distances and at equal angles. The pins to be installed in these holes are made of aluminum and Cr-Ni alloy material. These pins are located in the test tube, all of which are fitted sequential row and diagonal row. Thus, experiments were carried out for five different test tubes in total. One of them was examined for plain tube, two coupled for diagonal and sequential array of different material pins. Aluminum and Cr-Ni alloy pins inserted into the test tube are shown in fig. 2.

To assemble these pins to the test tube, 3 mm long teeth were drilled in the holes and pins. The holes were first drilled with an 8.5 mm diameter drill and then threaded with an M10 × 1.5 guide. At equal intervals along the test tube 20 stations were threaded as in fig. 3.

If the pins are inserted into the wall continuously at 90°, it is named *sequential row*. When the pins are positioned along the test tube and the radial axes are spaced apart, they are expressed in a diagonal row, fig. 4 and 5.

Analysis of experimental data

Among the measurements taken during the experiments are temperature of the test tube and outer surface of the insulation, input and output temperatures of air feed,

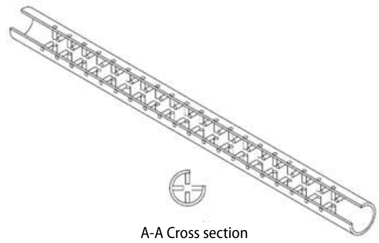


Figure 4. Tube with sequential row pin

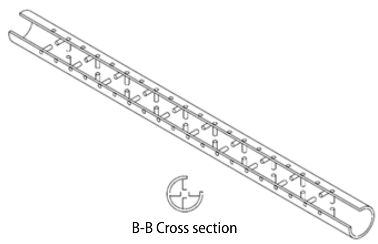


Figure 5. Tube with diagonal row pin

environmental air temperature, fluid velocity, pressure difference between test tube input and output ends, heater circuit current, and voltage. Accordingly, the following can be calculated: the heat transferred to the environment from the outer surface of the insulation:

$$Q' = 1.24\pi D'L\left(\overline{T'} - T_{\infty}\right)^{4/3} \tag{1}$$

Net electrical power due to direct electrical current at the input and output ends of the test tube:

$$P_{\text{net}} = \Delta VI - Q' \tag{2}$$

The heat flux obtained from the electrical current applied to the test tube:

$$q_w = \frac{P}{2\pi L R_{w_i}} \tag{3}$$

Heat generated per unit volume of the tube wall:

$$\dot{q} = \frac{P}{2\pi L\left(r_{w_0}^2 - r_{w_i}^2\right)} \tag{4}$$

Inner surface temperature in association with the outer surface temperature:

$$T_{w_{i_x}} = T_{w_{0_x}} - K\dot{q} \tag{5}$$

The factor, K, used in this equation:

$$K = \frac{(r_{w_0})^2}{2k_w}\left\{\ln\frac{r_{w_0}}{r_{w_i}} - \frac{1}{2}\left[1 - \frac{(r_{w_i})^2}{(r_{w_0})^2}\right]\right\} \tag{6}$$

Equation (7) is obtained considering the assumptions of a hollow cylinder where heat is generated evenly distributed inside the walls, having insulated outer surface, and where a fluid with constant temperature is heated with a constant heat transfer coefficient in the interior part using 1-D heat transfer analysis.

Bulk temperature of the fluid is calculated using:

$$T_{b_x} = T_{b_i} + \frac{P\left(\frac{x}{L}\right)}{\rho C_p \dot{V}} \tag{7}$$

Local heat transfer coefficient throughout the test tube at the x-axial distant is calculated using:

$$h_x = \frac{q_w}{T_{w_{i_x}} - T_{b_x}} \tag{8}$$

The dimensionless Nusselt number, a temperature gradient is calculated:

$$\text{Nu}_x = \frac{2h_x r_{w_i}}{k} \tag{9}$$

The pressure difference between the inlet and outlet points of the test tube and the coefficient of friction in the tubes with the help of the air-flow can be calculated:

$$f = \frac{\Delta P}{\frac{1}{2}\rho U_m^2 \frac{L_p}{D}} \tag{10}$$

Uncertainty analysis

The accuracy of the experimental data may be erroneous due to the nature of the measuring devices and the measurement operator. It is not always possible to prevent faults caused by measuring devices while user faults can be recovered. The errors from the measuring devices, expressed as uncertainty, were calculated by the eqs. (11) and (12) given by Kline and McClintock [18] on the experimental findings.

$$W_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \ldots\ldots\ldots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2} \tag{11}$$

$$R = R(x_1, x_2, x_3, x_4 \ldots x_n) \tag{12}$$

where R is the size to be measured in the system, n – the number of units affecting this magnitude, x_1, x_2, x_3, x_4, ..., x_n are independent variables and w_R – the error rate of R size. The error amounts for each measured parameter are determined according to eq. (11) and the total uncertainty giving the total error amount is found. As a result, the uncertainty amount of ±0.0089 in the heat transfer coefficient, ±0.0942 in the heat transfer coefficient and ±0.05 in the friction coefficient was determined.

The CFD analysis

Mesh independence study is for specifying how much mesh is enough for CFD analysis. It is done with gradually increasing the number of mesh. A result goal is selected and check for all mesh orientation. If goal value does not change as mesh number is increased this mesh orientation is fine for this model. Mesh independence study result is at following.

After this study mesh number fixed as 95474. There are four criteria for analysis which are three type of pins shape, three material of pins, two kinds of pins arrangements, and nine different velocity of air. Hence there are 162 analysis for this paper. Common parts of these analysis are that heating operation is defined as surface heat generation and exit pressure is defined as environment pressure. Ambient temperature is taken as 20 °C and pressure is taken as 1 atm. On the other hand, each analysis run on approximately 1500 seconds. Details of the analysis are given in tab. 1. This table is derived from the automatically generated software report of FloEFD software.

Table 1 . Details of the analysis

Product	FloEFD FE17.0. Build: 4208
Computer name	DESKTOP-KLEPI6S
User name	EEM03
Processors	Intel(R) Core(TM) i7-7700 CPU at 3.60GHz
Memory	8121 MB / 134217727 MB
Operating system	Windows 10 (or higher) (Version 10.0.16299)
CAD version	SOLIDWORKS 2017 SP3.0
CPU speed	3601 MHz

Research findings

In order to investigate the effect of aluminum and Cr-Ni alloy pins with cylindrical geometries and arrangements on the heat transfer and flow characteristics in the in-tube flow, experiments were first carried out with a plain tube between Reynolds number of 10000 and 50000. The results obtained are compared with the existing and widely used equations and evaluations. Figure 6 shows the comparison of the Nusselt numbers obtained from the experiments for the plain pipe with the equations of Petukhov, Colburn, and Ditus-Boelter [19]. Figure 7 shows that Nusselt curves obtained with experimental results are compatible with

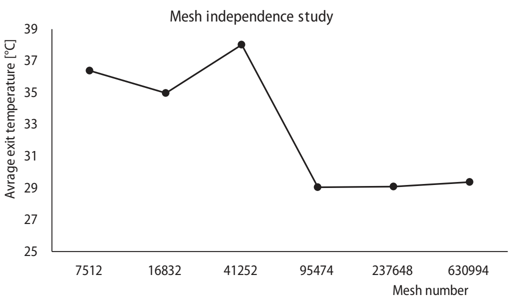


Figure 6. Mesh independence study according to average exit temperature and mesh number

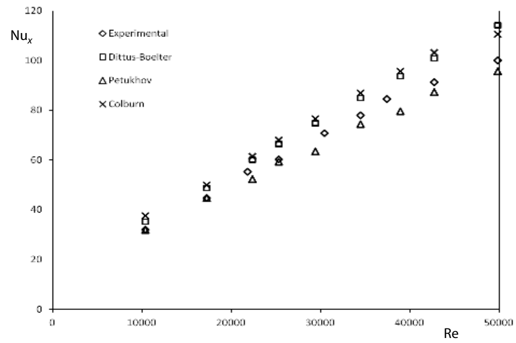


Figure 7. Comparison of experimental Nusselt number with correlations in the literature for plain tube

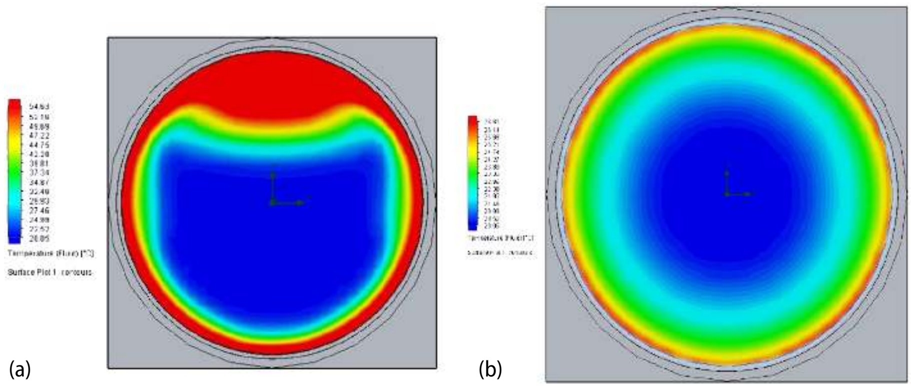


Figure 8. Temperature distribution in Re = 10000 (a) and Re = 50000 (b)
(*for color image see journal web site*)

Re = 25000. After Re = 25000, it is seen that the values have gradually moved away from the literature results with the increase of Reynolds number. Nevertheless, even in the greatest Reynolds number, the experimental Nusselt number seems to have approached about 15% with Petukhov equation.

According to CFD analysis fig. 8 shows that temperature distribution of plain tubes for Re = 10000 and Re = 50000. When the fig. 7 was examined, it was observed that temperature decrease toward cylinder centre due to friction factor and flow velocity. It was seen that temperature values for u = 3.84 m/s are higher than temperatures obtained for u = 18.9 m/s.

Comparison of the friction coefficients obtained from the experimental results for the plain tube with the equation obtained by Petukhov [19] is given in fig. 9. When the figure is examined, it can be seen that the friction coefficient change curve for all Reynolds numbers agrees with Pethukov' s equation.

In order to investigate the effect of cylindrical aluminum and Cr-Ni alloy pins on the heat transfer and flow characteristics, the pins were sequential row in a straight line and diagonal row in the wall of the test tube and the experiments were repeated for values ranging from Reynolds number of 10000 to 50000. The results are given in terms of Nusselt numbers for heat transfer and friction coefficients for flow characteristics. Variation of the Nusselt number along the tube for two different situations in which the aluminum and Cr-Ni alloy pins are diagonal row and sequential row, are given in figs. 10, 11, 14, and 15. When the shapes are examined, the first result is that the Nusselt curves show a similar tendency for all Reynolds numbers. As the number of Reynolds increases along the test tube, Nusselt numbers also increase. The heat transfer coefficient along the x-axial distance of the cylindrical diagonal row pin, cylindrical sequential row pin and tubes increases with a declining decline of up to $6D$ in diameter. Then, it decreases up to $10D$ distance, then increases up to $12D$ again and finally curve decrease up to $15D$ axial distance. Fluctuations in Nusselt values continue until $x = 15D$ diameter provided by the thermal development. Due to the flow inhibiting effect of the pins, a sudden increase in the Nusselt values is observed until the separated flow reaches the point of restitution. The reason for the Nusselt values here being too great is that additional turbulence, turbulence, and vortices significantly increase the heat transfer through the convection with the flow impinging on the pins. Then, the fluid moving along a certain x-axial distance arises a declination in the Nusselt curves due to the fact that there is no element to break up the lamina boundary-layer on the surface before reaching the next pintle. The lowest Nusselt value for Re = 9287 is 46.83, while the highest value for Re = 47843 is 118.71 in diagonal row aluminum pin. The lowest Nusselt value is 32.19 for Re = 9147, while the highest value is 150.87 for Re = 48237 in tube with sequential row aluminum pin. The lowest Nusselt value for Re = 9166 is 37.76, while the highest value for Re = 42952 is 120 in tube with sequential row Cr-Ni alloy pin. In tube experiments with diagonal row Cr-Ni alloy pin, the lowest Nusselt value is 35.75 for Re = 9327, while the highest value is 115 for Re = 47579.

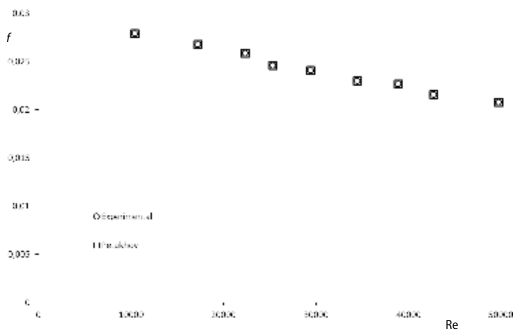


Figure 9. Comparison of experimental friction coefficient with Petukhov correlation for plain tube

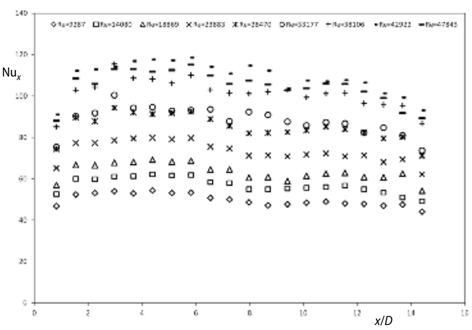


Figure 10. Change of Nusselt numbers along the channel of tube with diagonal row aluminum pin

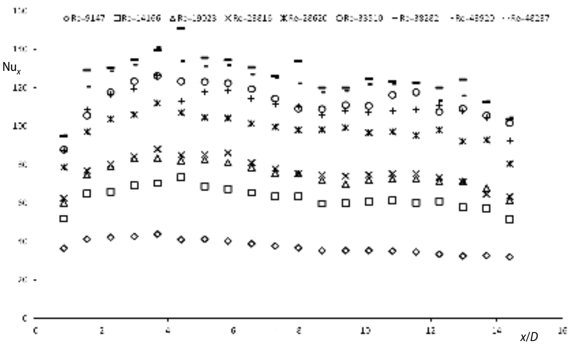


Figure 11. Change of Nusselt numbers along the channel of tube with sequential row aluminum pin

After obtaining experimental Nusselt numbers of tube with aluminum pin and sequential row, numeric analysis were performed using CFD. Temperature distribution, flow lines and thermal development for Re = 10000 and Re = 50000 in test tube are shown between figs. 12(a) and 13(b).

After obtaining experimental Nusselt numbers of tube with Cr-Ni alloy pin and sequential row, numeric analysis were performed using CFD.

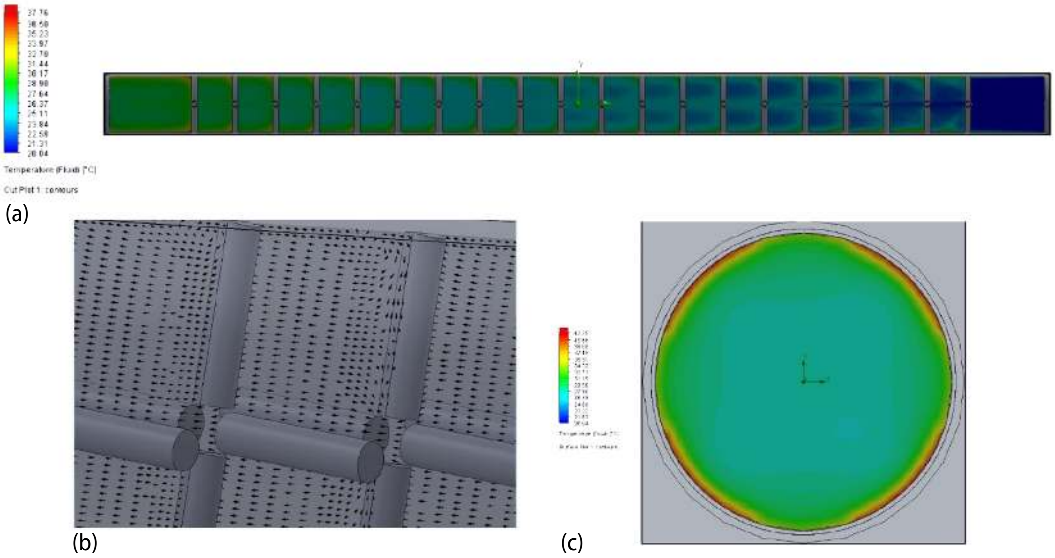


Figure 12. (a) Thermal development for tube with aluminum pins in Re = 10000, (b) flow characteristics for tube with aluminum pins in Re = 10000, and (c) temperature distribution for tube with aluminum pins in Re = 10000 (*for color image see journal web site*)

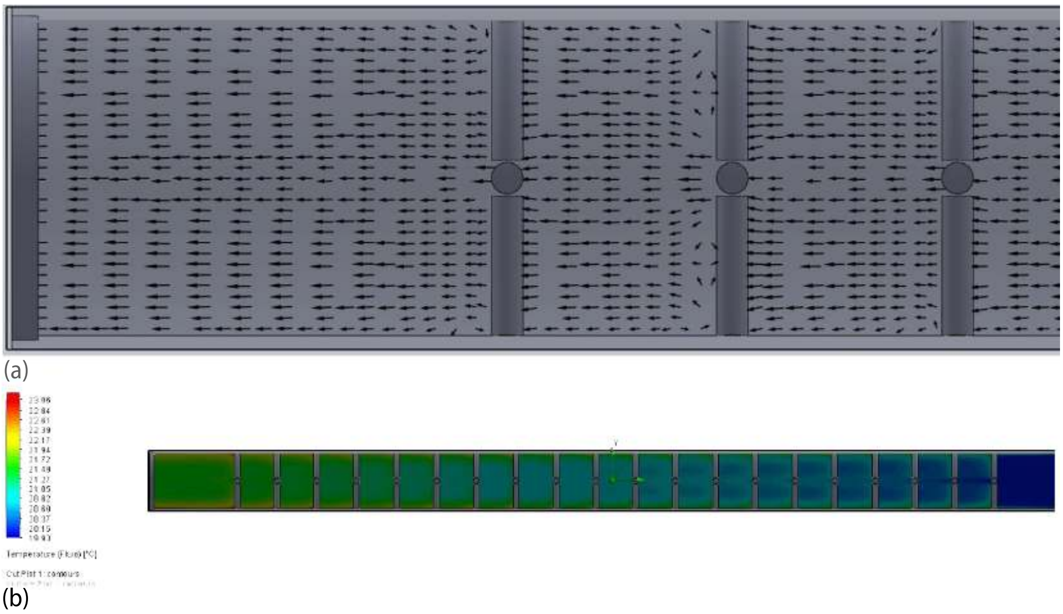


Figure 13. (a) Flow characteristics for tube with aluminum pins in Re = 50000, (b) thermal development for tube with aluminum pins in Re = 50000 (*for color image see journal web site*)

Temperature distribution, flow lines and thermal development for Re = 10000 and Re = 50000 in test tube are shown between figs. 16(a) and 17(b)

Figure 18 shows the average Nusselt number, Nu_{fd}, variation for Reynolds number for tubes with pin and plain tube with different numbers and arrangements. When the curves are

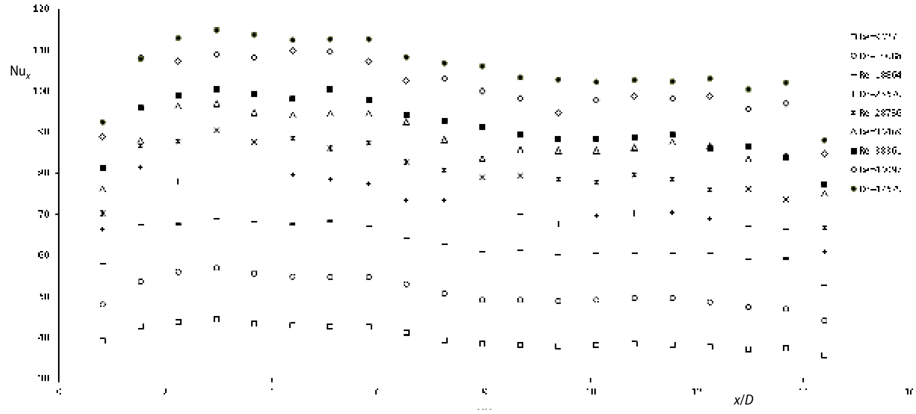


Figure 14. Change of Nusselt numbers along the channel of tube with diagonal row Cr-Ni alloy pin

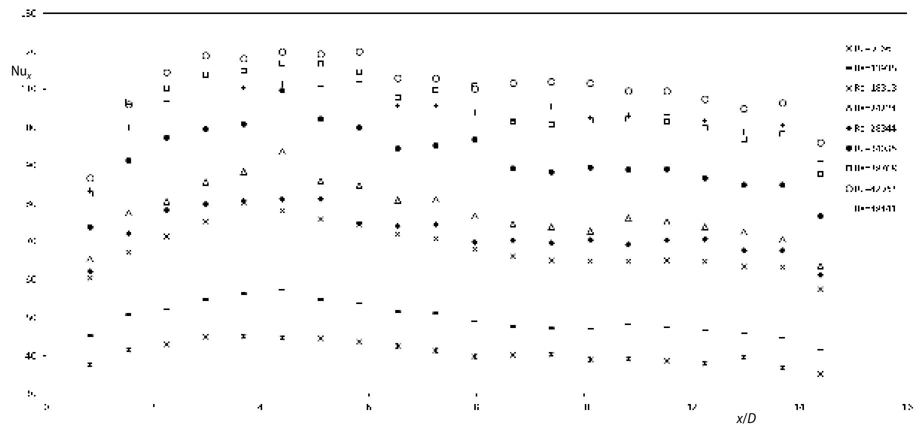
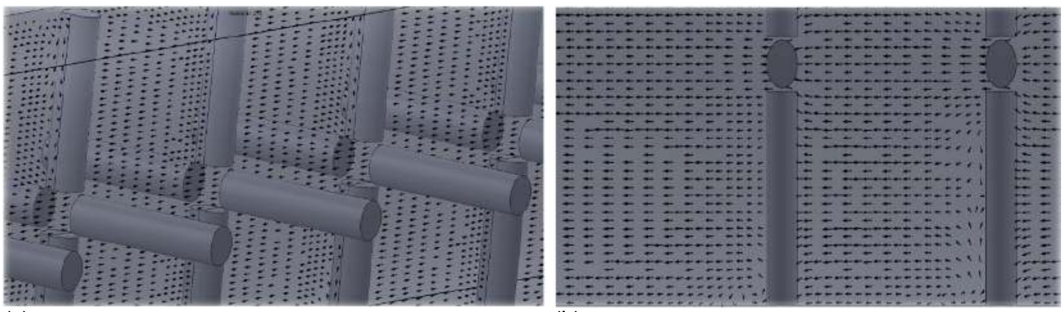


Figure 15. Change of the Nusselt numbers along the channel of tube with sequential row Cr-Ni alloy pin



(a) (b)

Figure 16. (a) Flow characteristics for tube with Cr-Ni pins in Re = 10000, (b) flow characteristics for tube with Cr-Ni pins in Re = 50000

examined, it is seen that the Reynolds numbers increase while the Nu_{ort} values increase with a decreasing slope. Reynolds number is the lowest value of fluid velocity and Nu_{ort} data is lowest.

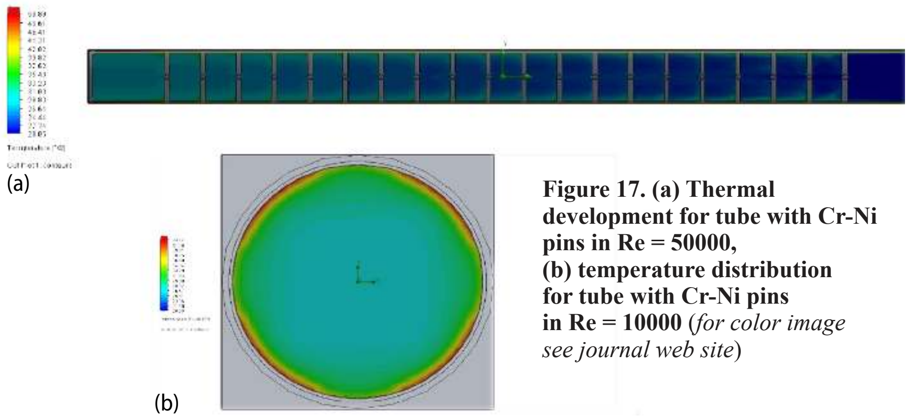


Figure 17. (a) Thermal development for tube with Cr-Ni pins in Re = 50000, (b) temperature distribution for tube with Cr-Ni pins in Re = 10000 (*for color image see journal web site*)

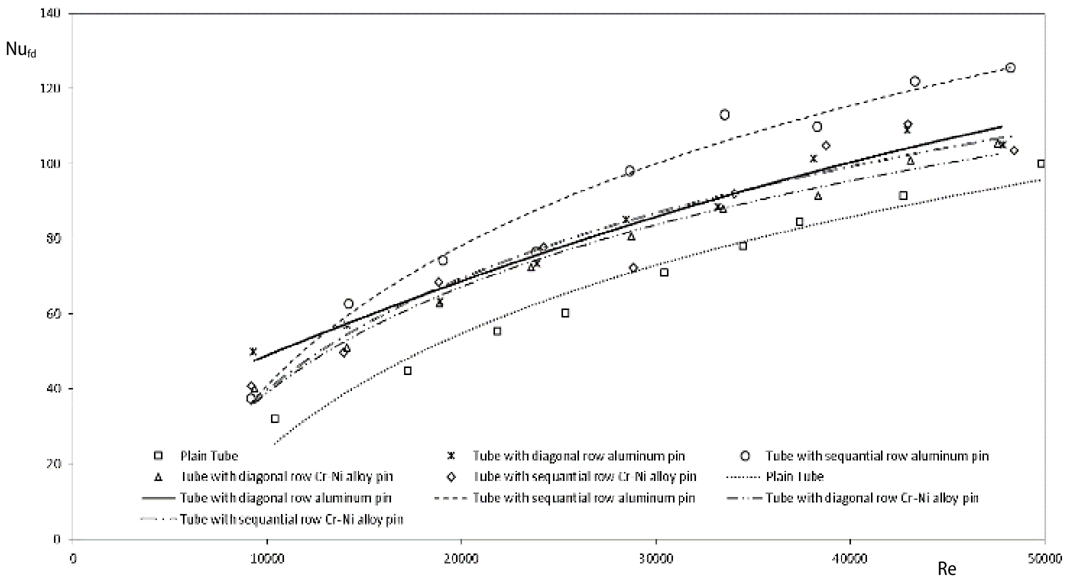


Figure 18. Change along the Reynols number of average Nusselt numbers for all test tubes

It has been observed that the closest values to the calculations made for plain tube and tubes with pin appeared at the Reynolds values where the lowest fluid velocity was again. It is also observed that the heat transfer coefficient increases with increasing pin count and frequency. The lowest heat transfer coefficient obtained for the plain tube was 32.125 at Re = 10384, whilst the highest was found to be 125.73 for Re = 48237 in the tube with sequential row aluminum pin.

Figure 19 shows the variation of friction coefficients according to Reynolds number for five different test tubes. When the figure is examined, it is seen that the friction coefficient is more than that when the pins are arranged in order. Friction factor value in test tube with sequential row aluminum and diagonal row Cr-Ni alloy pin shows a sudden rise, for Re = 10000-15000. Friction coefficients are exposed to fluctuations except for the time it takes for the fluid to hit the surface again with the impact of each pin surface. In experiments for five different test tubes, the lowest friction coefficient is 0.0279 in Re = 10384 for the plain tube, while the highest friction is 0.854 at Re = 33510 for the tube with cylindrical sequential row aluminum pin.

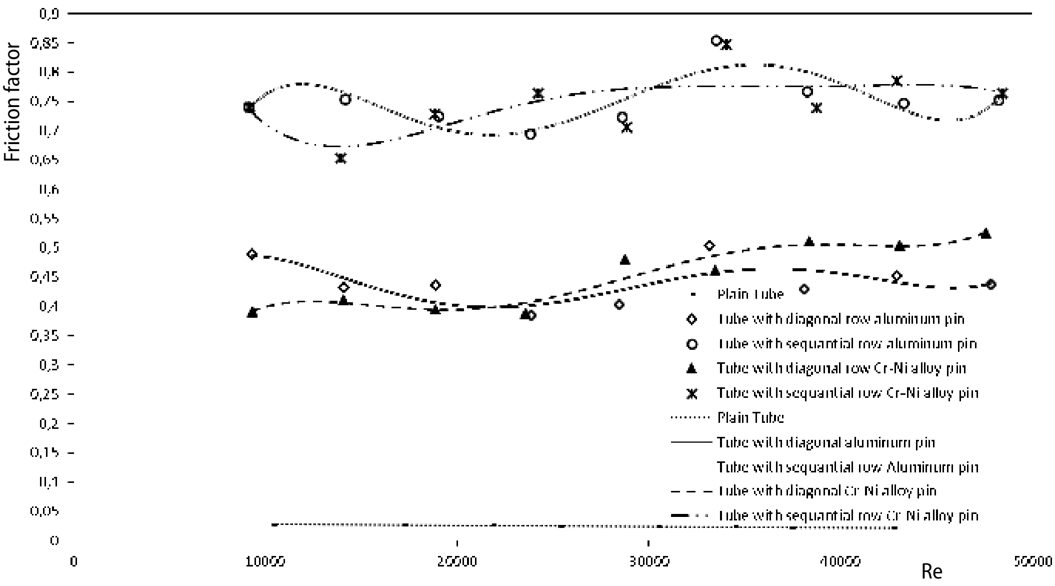


Figure 19. Change along Reynolds number of friction coefficients

Conclusion

The conclusions obtained can be summarized as follows. Nusselt values obtained in the measurements made on the tubes with pin were found to be higher than those of plain tube. The Nusselt values obtained along Reynolds number for five different test tubes are shown in the order of magnitude; are tube with sequential row aluminum pin, sequential row Cr-Ni alloy pin, diagonal row aluminum pin, diagonal row Cr-Ni alloy pin and plain tube. Here, it was concluded that the pins increased in surface area and pin counts as well as in Nusselt values varying along Reynolds number. The rate of improvement in heat transfer obtained by using pins with respect to the plain tube is 38.47-39.21% in tube with sequential row aluminum pin, 36.3-51.6% in tube with diagonal row Cr-Ni alloy pin, 38.48-59.8% in tube with diagonal row aluminum pin, 31.5-42.6% tube with sequential row Cr-Ni alloy pin. On the other hand, it was determined that the coefficient of friction increased with the increase of heat convection coefficient. It is determined that the friction coefficient is between 0.0208-0.0279 in the plain tube, 0.694-0.854 in tube with sequential row aluminum pin, 0.652-0.846 in tube with sequential row Cr-Ni alloy pin, 0.437-0.489 in the tube with diagonal row aluminum pin, 0.390-0.526 in the tube with diagonal row Cr-Ni pin. As a result, it is observed that experimental results are compatible with both literature and numerical study.

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