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Effects on Heat Transfer and Flow Characteristics of Aluminum Pins Placed in Circular Channel

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ABSTRACT

In this study, the effects of aluminum cylindrical and shaved 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 Re 10000 and 50000. It has been observed that the Nusselt values obtained over the number of Re for the 5 different test tubes are arranged in a line from large to small, cylindrical sequential row, shaved sequential rows, cylindrical diagonal row, shaved diagonal row, plain tube. On the other hand, it was determined that friction coefficient is directly proportional to the increase of heat transfer coefficient.

Keywords: Convection heat transfer, heat transfer enhancement, aluminum pins, turbulent flow in pipe, friction and pressure losses in channels.

1. 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. Darıcı [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 carried out, they made different hole diameters and plate numbers. Güneş S. [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. 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 Reynolds=1428-3008. Eren and Çalışkan [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.

In this study, the effects of aluminum cylindrical and shaved 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.

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2. MATERIAL AND METHOD

A. Experimental Setup

The experimental setup used in the study is schematically shown in Figure 1. The test setup 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.

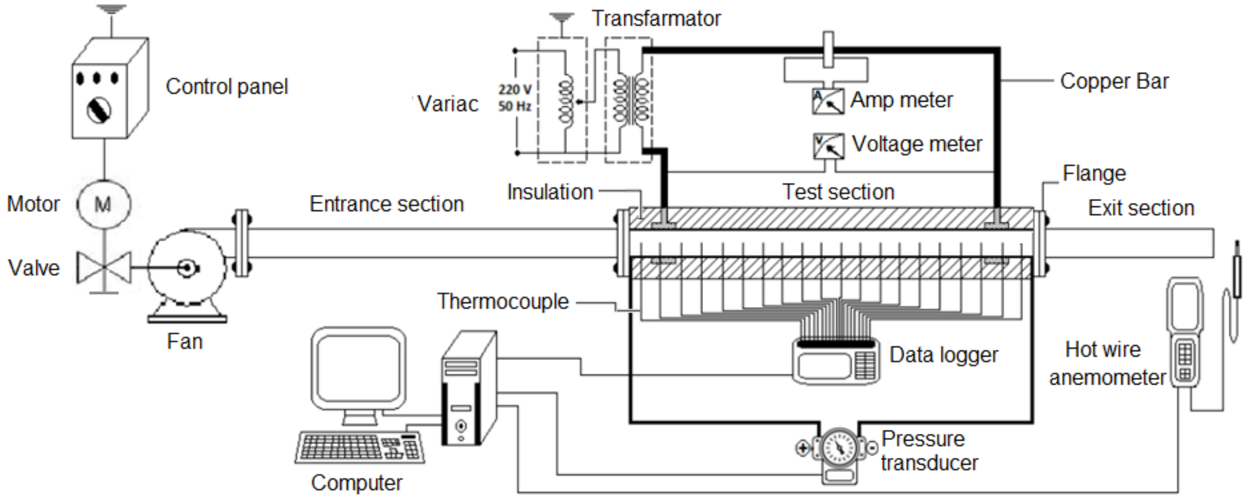

Figure 1: Schematic representation of the test setup

The chimney 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. 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 and 0-1000A and 0-1V ampermeter and voltmeter. The current is supplied by thick copper-nickel alloy bars attached to the inlet and outlet of the test probe.

Thus, a homogeneous heat flow distribution is provided at each end of the test tube surface. In this experimental setup, 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 hole were drilled in 90^0 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 material and are manufactured in 2 types as cylindrical and shaved. These pins are located in the test tube, all of which are fitted sequential row and diagonal row. Thus, experiments were carried out for 5 different test tubes in total. One of them was examined for plain tube, 2 for shaved and cylindrical pins and 2 for diagonal or sequential array of pins. Aluminum pins inserted into the test tube are shown in Figure 2.

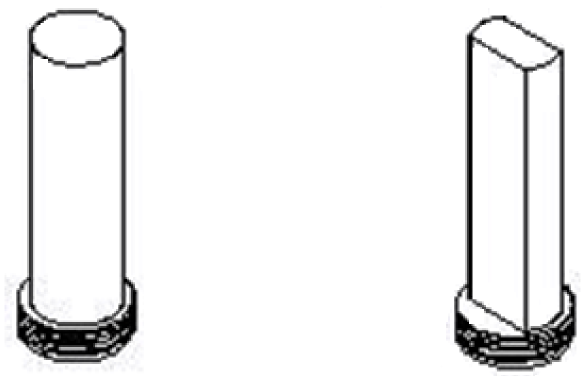

Figure 2: Cylindrical and shaved aluminum pins

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 a 8.5 mm diameter drill and then threaded with an M10x1.5 guide. At equal intervals along the test tube 20 stations were threaded as in Figure 3.

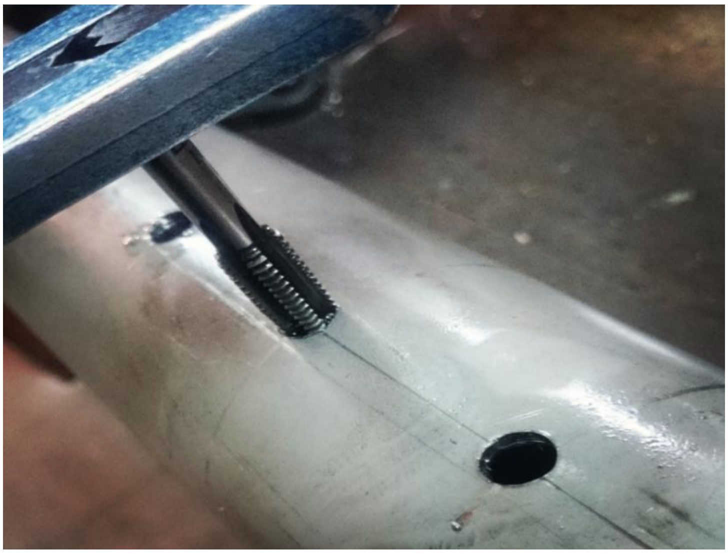

Figure 3: Tap in the test tube

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.

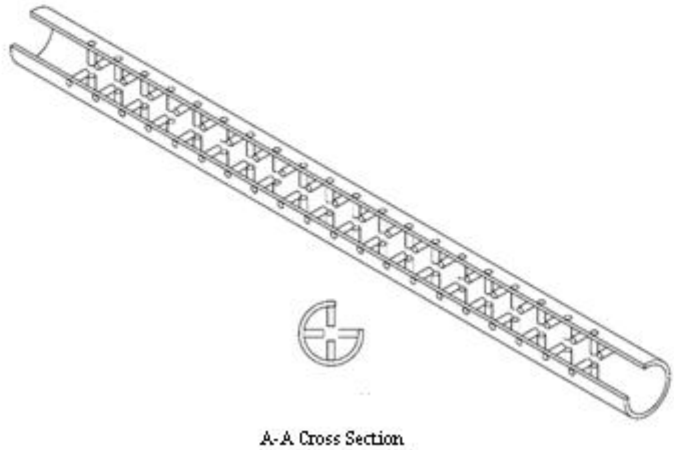

Figure 4: Tube with sequential row pin

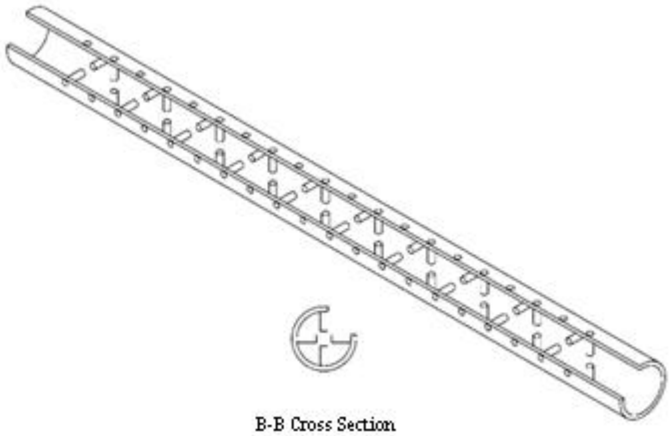

Figure 5: Tube with diagonal row pin

B. 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, 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,

$$Q' = 1{,}24\,\pi\,D'\,L\,(\,\overline{T}' - T_\infty\,)^{4/3} \tag{1}$$

net electrical power due to direct electrical current at the input and output ends of the test tube;

$$P_{net} = \Delta V\,I - Q' \tag{2}$$

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,

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

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 one-dimensional heat transfer analysis.

Bulk temperature of the fluid is calculated using;

$$T_{b_x} = T_{b_i} + \frac{P\,(\,x/L\,)}{\rho\,C_P\,\dot{V}} \tag{7}$$

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 using;

$$Nu_x = \frac{2\,h_x\,r_{w_i}}{k} \tag{9}$$

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 by Equation 10.

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

C. 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 with the equation given by Kline and McClintock [8] on the experimental findings. This equation,

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

$$R = R(x_1, x_2, x_3, x_4, \ldots\ldots x_n) \quad (12)$$

is shown with equation 11 and 12. Where R is the size to be measured in the system, n is the number of units affecting this magnitude, x_1, x_2, x_3, x_4, ..., x_n are independent variables and wR is the error rate of R size. The error amounts for each measured parameter are determined according to equation 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.

3. RESEARCH FINDINGS

In order to investigate the effect of aluminum pins with different geometries and arrangements on the heat transfer and flow characteristics in the in-pipe flow, experiments were first carried out with a plain pipe between 10000 and 50000 Reynolds number. 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 [9]. Figure 6 shows that Nusselt curves obtained with experimental results are compatible with Reynolds 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.

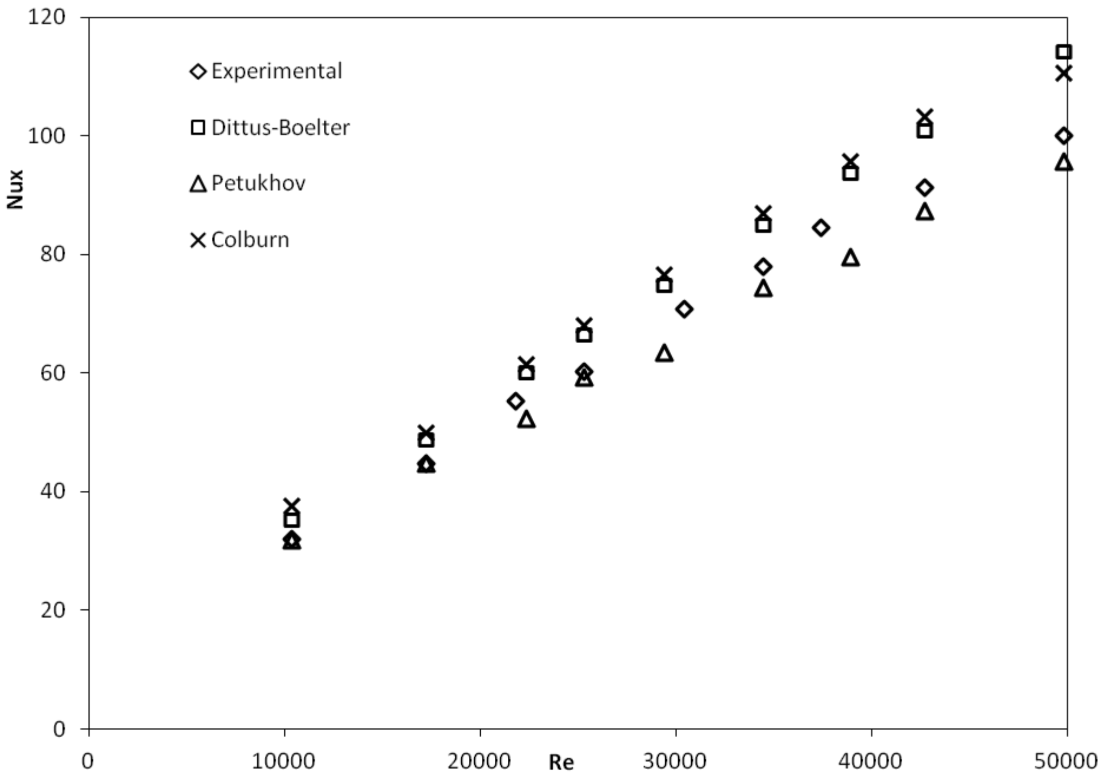

Figure 6: Comparison of experimental Nusselt number with correlations in the literature

Comparison of the friction coefficients obtained from the experimental results for the plain tube with the equation obtained by Petukhov [7] is given in Figure 7. When the figure is examined, it can be seen that the friction coefficient change curve for all Reynolds numbers agrees with Petkhov's equation.

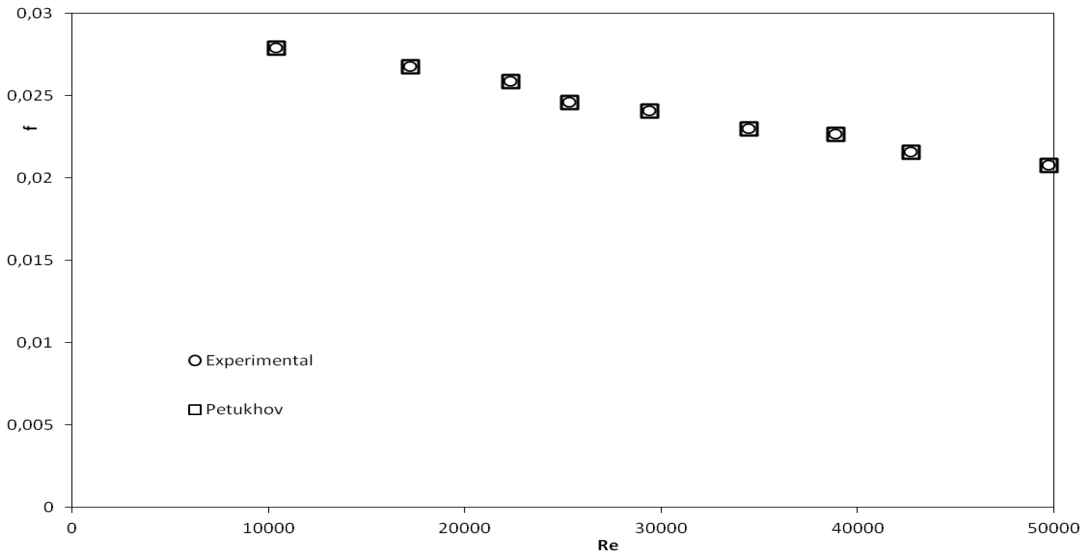

Figure 7: Comparison of experimental friction coefficient with Petukhov correlation

In order to investigate the effect of cylindrical and shaved aluminum 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 10000 to 50000 Reynolds number. 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 pipe for four different situations in which the aluminum pins are cylindrical diagonal row, cylindrical sequential row, shaven diagonal row and shaved sequential row are given in Figure 8, Figure 9, Figure 10 and Figure 11. 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 shaved diagonal pin tubes increases with a declining decline of up to 6D in diameter. Then, It decreases to a certain distance and then increases again. 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 Nu 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 Nu value for Re = 9287 is 46.83, while the highest value for Re = 47843 is 118.71. The lowest Nu value is 32.19 for Re = 9147, while the highest value is 150.87 for Re = 48237 in tube experiments with cylindrical sequential row pin. The lowest Nu value for Re = 9285 is 31.68, while the highest value for Re = 47807 is 117.50. In the case of shaved sequential row aluminum pipe experiments, the lowest Nu value is 30.89 for Re = 9064, while the highest value is 137.11 for Re = 47830.

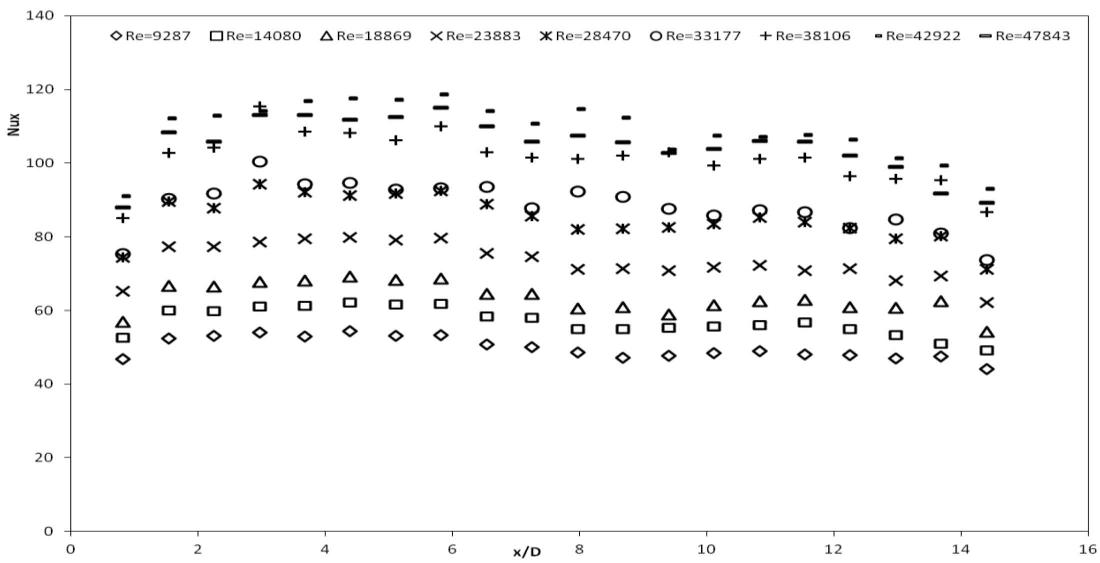

Figure 8: Change of Nusselt numbers along the channel of tube with cylindrical diagonal row pin

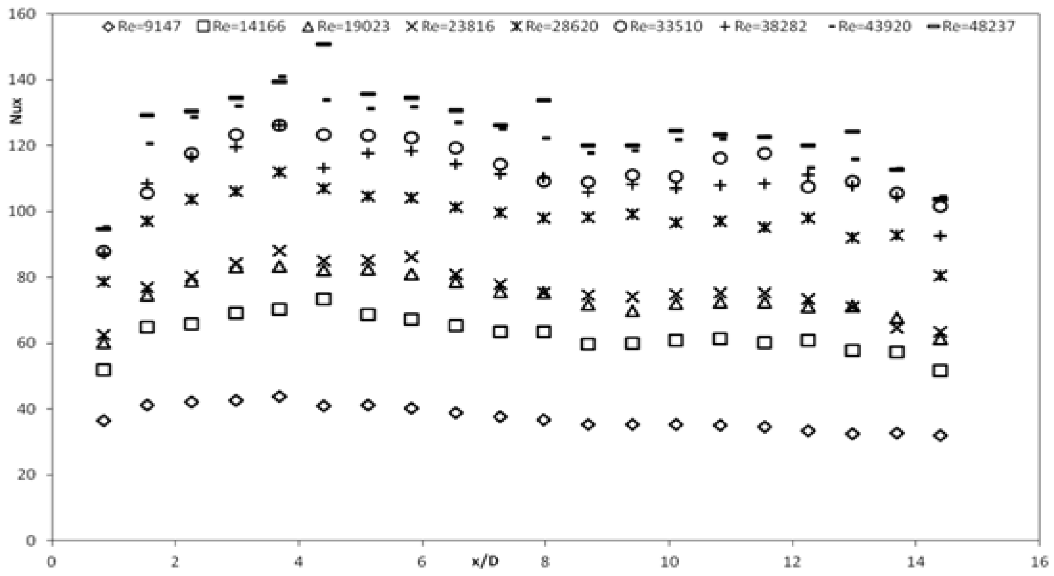

Figure 9: Change of Nusselt numbers along the channel of tube with cylindrical sequential row pin

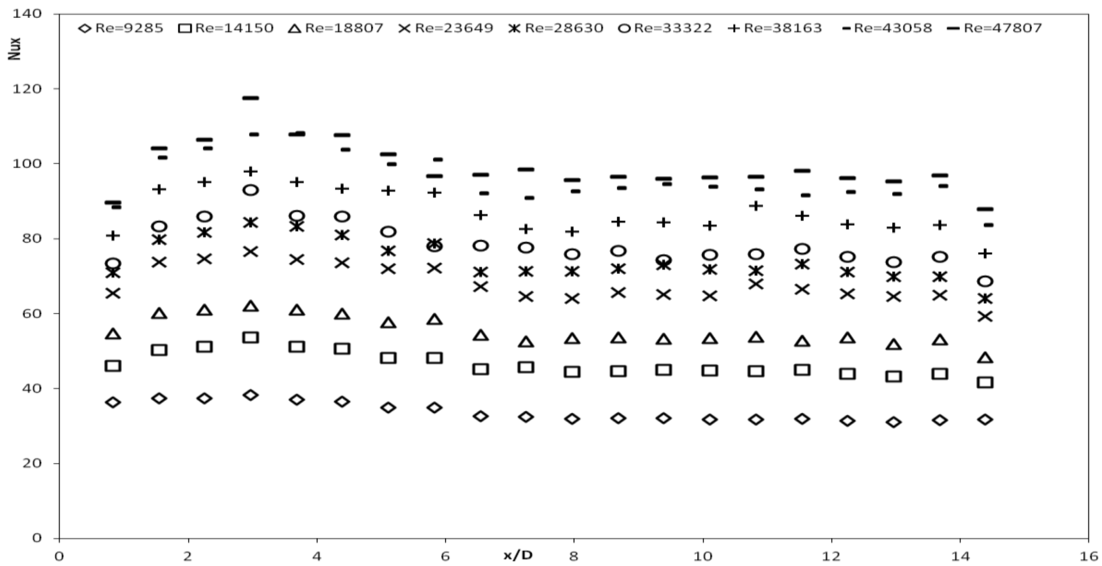

Figure 10: Change of Nusselt numbers along the channel of tube with shaved diagonal row pin

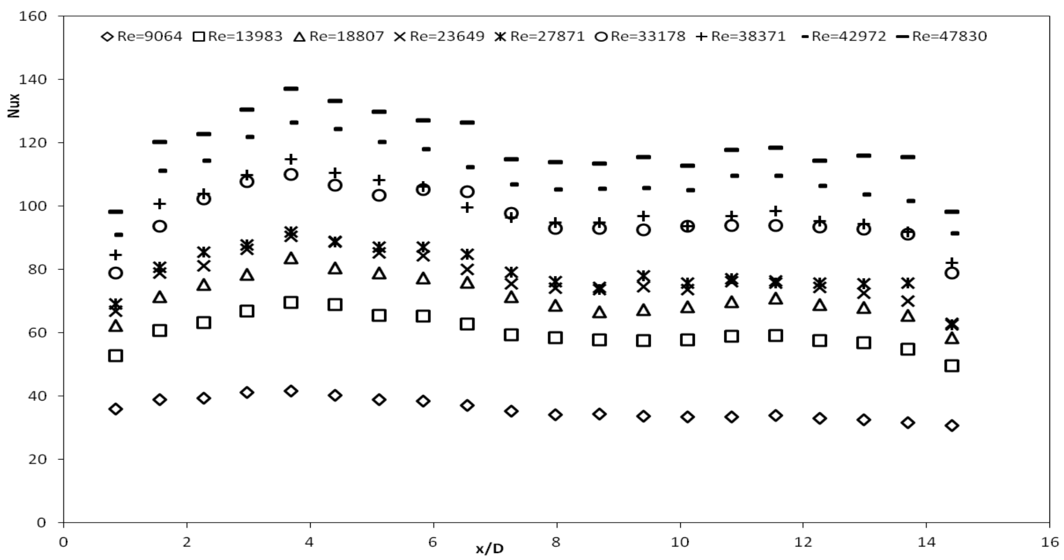

Figure 11: Change of the Nusselt numbers along the channel of tube with shaved sequential row pin

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Figure 12 shows the average Nusselt number (Nu_{ort}) variation for Reynolds number for pipes with pin and plain tube with different numbers and arrangements. When the curves are examined, it is seen that the Re numbers increase while the Nu_{ort} values increase with a decreasing slope. Re is the lowest value of fluid velocity and Nu_{ort} data is lowest. It has been observed that the closest values to the calculations made for plain tube and tubes with pin appeared at the Re 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 straight pipe was 32.125 at Re = 10384, whilst the highest was found to be 125.73 for Re = 48237 in the tube with cylindrical sequential row pin.

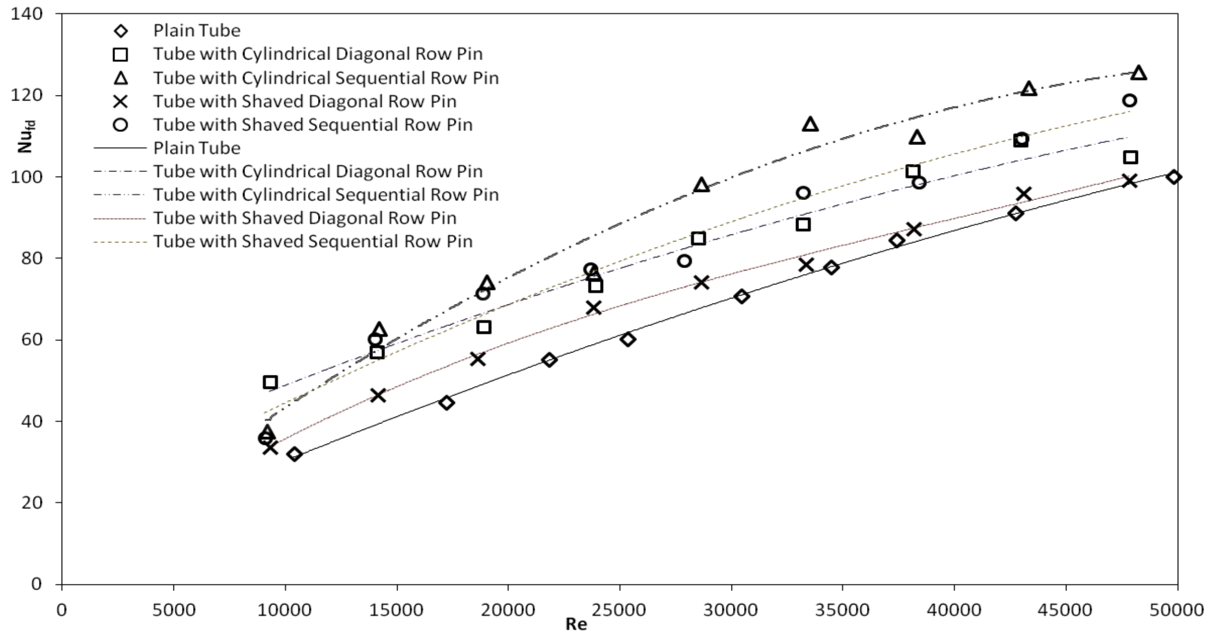

Figure 12: Change along the Re number of average Nusselt numbers for all test tubes

Figure 13 shows the variation of friction coefficients according to Reynolds number for 5 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. In the case of perfectly straight road and shaved crossed pipe test, the coefficient of friction at Re = 10000-15000 shows a sudden rise, while the entire shaved surface shows a decrease in straight row and crossed row pipe test. 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 5 different test tubes, the lowest friction coefficient is 0.0208 at Re = 10384 for the plain tube, while the highest friction is 0.854 at Re = 33510 for the tube with cylindrical sequential row pin.

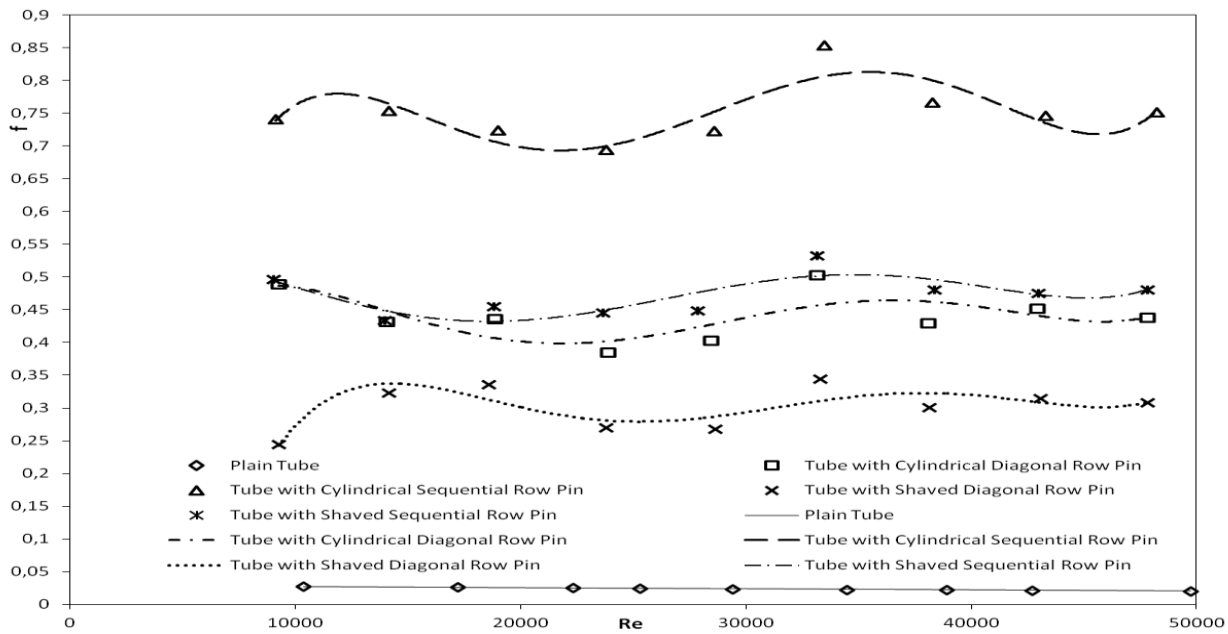

Figure 13: Change along Re number of friction coefficients

CONCLUSIONS

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 5 different test tubes are shown in the order of magnitude; are tube with cylindrical sequential row pin, shaved sequential row pin, cylindrical diagonal row pin, shaved diagonal row 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 Re number. The rate of improvement in heat transfer obtained by using pins with respect to the plain tube is 17%-33% in tube with cylindrical sequential row pin, 12%-20% in tube with shaved sequential row pin, 14%-19% in tube with cylindrical diagonal row pin, 9.9%-12.9%, tube with shaved diagonal row 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 cylindrical sequential row pin, 0.434-0.533 in tube with shaved sequential row pin, 0.385-0.503 in the tube with cylindrical diagonal row pin, 0.244-0.344 in the tube with shaved diagonal row pin.

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