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Full Paper

Effects of sinusoidal turbulator in cylindrical channel on heat transfer and flow characteristics

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Abstract: Effects of heat transfer and flow characteristics on Reynolds numbers of different sinusoidal turbulators placed separately from a pipe in the turbulent flow were investigated experimentally. The experiments were carried out under forced flow and constant heat flow conditions. The elements, which drew a sine curve used as a turbulator, were positioned axially to the channel centre. The experiments were repeated for 3 width values of sinusoidal turbulators that correspond to 3D/4, D/2 and D/4 (D = inner diameter of test pipe). The experiments showed that although there were increases in the rate of heat transfer with the turbulator, they caused considerable pressure drop. As a result, as the width of the turbulator increased, the heat transfer ratio (the Nusselt number) and coefficient of friction increased. Increase in Nusselt number was about 39-86% at D/4 width, 70-119% at D/2 width and 72-170% at 3D/4 width. It was observed that the coefficient of friction was significantly affected by the width of the sinusoidal turbulator, being about 0.02 in the case of straight pipe, 0.184 - 0.242 at D/4 width, 0.345 - 0.415 at D/2 width, and 0.416 - 0.451 at 3D/4 width. In addition, increasing coefficient of friction increased pressure drop and power required for pumping.

Keywords: turbulent flow in pipe, heat transfer enhancement, sinusoidal turbulator, pressure drop and friction in pipe

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ABBREVIATIONS

A'	Surface on heat transfer	**R**	Real measured value
a	Width	**Re**	Reynolds number
c_p	Specific heat	r_{wi}	Inner radius of pipe
D	Inner diameter of test pipe	r_{wo}	Outer radius of pipe
D_o	Outer diameter of test pipe	r_m'	Isolation radius
f	Friction coefficient	T_{bx}	Bulk temperature
f_s	Friction coefficient in experimental with plain tube	T_m'	Isolation temperature,
h_m'	Isolation heat convection coefficient	T_{wo}	Wall outer temperature in pipe
h	Heat convection coefficient	T_{wi}	Wall inner temperature in pipe
I	Current	T_∞	Ambient temperature
K	Thermal conductivity factor	U_m	Average fluid velocity
k	Heat conduction coefficient	$\dot{V}$	Specific volume
L	Pipe length	W_R	Error rate
Nu_x	Local Nusselt number	ΔP	Pressure difference
Nu_{fd}	Average Nusselt number	ΔV	Potential difference
n	Amplitude of sinusoidal turbulator	Δx	Axial distance
Q	Heat transfer	η	Thermal performance factor
$\dot{q}$	Heat in unit volume	λ	Wavelength
q	Heat flux	ρ	Density
Q'	Heat loss	ϑ	Kinematic viscosity
P_{net}	Net electrical power		

INTRODUCTION

Pipe heat exchangers are widely used in industrial areas, housing, recycling systems, and chemical and food industries. Nowadays various methods are used to increase heat transfer and improve the effectiveness of these heat exchangers. The most frequently used technique is the passive method in which various internal elements are placed in pipe. Therefore, there are numerous experimental and numerical studies conducted on this method in order to reduce operating costs and improve effectiveness and heat transfer. In these studies some researchers examined the effects of various elements placed in the pipe inlet on heat transfer and flow characteristics. Sparrow and Chaboki [1] investigated the effect of non-symmetric blockage elements placed in the pipe inlet on heat transfer and turbulent flow characteristics. As a result, experiments made for different Reynolds numbers (Re) and blockage rates have shown that when flow separation and reattachment are observed, the highest value of heat transfer is obtained. In a similar study Chen at al. [2] also examined the effect of a decaying axisymmetric swirl element on heat transfer. They observed that the swirl element placed into the inlet increased significantly the heat transfer in the pipe. Many researchers have investigated the effects of various elements placed along pipe on heat transfer and flow characteristics. A comprehensive literature review on these studies was given by Gurdal [3].

Among these studies, Sethumadhavan and Rao [4] investigated the effect of variable pitch, helical angle and thickness on heat transfer of wire tightly positioned in the pipe. They compared results of the studies with those in the literature and developed a general correlation for different types of rough surfaces. Hsiesh et al. [5] numerically investigated the effects of different dimensions of flat strip elements placed in the pipe on heat transfer and turbulent flow. According to the results obtained, it was determined that a circular pipe with strip element increased heat transfer 2-3 times compared to a smooth pipe. Eiamsa-ard and Promvonge [6] conducted an experimental study on friction characteristics and heat transfer in a tube having a uniform heat flux and being fitted with V-nozzle turbulators. The authors reported an increase in the average heat transfer for different pitch ratios (270%, 236% and 216%). Moreover, they suggested that the highest enhancement in heat transfer was obtained at low pitch ratio and the enhancement was inversely proportional to the increase in Re. Tandıroglu [7] conducted experimental studies on the effects of semi-circular baffles placed along a tube on the heat transfer and turbulent flow. The author explored the Nusselt number (Nu) and pressure drop by using the results obtained from tests conducted with different Re, aspect ratio, baffle spacing to diameter ratio, fitting angle, baffle orientation angle, baffle to tube length ratio, tube length to baffle spacing ratio, baffle area ratio and cross-sectional area ratio. Eiamsa-ard and Promvonge [8] investigated heat transfer and friction characteristics of conical-nozzle turbulators fitted in a test tube with different pitch ratios (2.0, 4.0 and 7.0). They determined the effects of two types of turbulators (C-nozzle turbulator and D-nozzle turbulator) in terms of the Re ranging between 8000 and 18000. The experimental results showed that heat transfer increased from 236% to 344% depending on the Re and turbulator arrangement compared to a straight tube. Anvari et al. [9] examined the effects of conical turbulators fitted in a tube on heat transfer and pressure drop for forced convection. They compared the results obtained from the experiments conducted on C-nozzle and D-nozzle turbulator arrangements with Nu correlations in the literature. Although it was possible to achieve enhancement in Nu by 521% for D-nozzle arrangement and 355% for C-nozzle arrangement, they suggested that turbulators cause a significant pressure drop.

Fan et al. [10, 11] numerically examined effects of conical turbulators fitted in a tube on heat transfer ratio, friction coefficient and thermal-hydraulic performance of turbulent flow. The authors found that Nu increased by~ 5 fold (Nu=98.35- 400.41) as coefficient of friction increased by~10-fold (f=0.062-0.36), compared to the smooth tube. Therefore performance assessment criterion ranged between 1.67 and 2.06. When the modified tube was compared with the smooth tube, 3.70 - 5.51fold increase in Nu and 0.59-1.51fold increase in the friction coefficient was observed. They also noted that the performance assessment criterion ranged between 1.17 - 2.97. Besides, they numerically repeated the previously mentioned study for louvered tape under similar conditions. The authors found that Nu increased 2.75 - 4.05fold (Nu=108.71 – 423.87) compared to a straight tube and the performance assessment criterion ranged between 1.60 - 2.05.

Muthusamy et al. [12] studied the effects of conical cut-out turbulators fitted in a tube with turbulent flow on heat transfer ratio, friction coefficient and thermal performance for several Re's. The authors also investigated the effects of turbulators fitted with internal fins of varying arrangements and pitch angles. Results showed that maximum heat transfer was 315% greater in a D-nozzle arrangement with a pitch angle of 3 compared to that in a straight tube, and its friction coefficient increased 3.2fold as its thermal performance factor increased 2.4fold, compared to those of the straight tube. Zhang et al. [13] explored numerically effects of helical baffles fitted in a tube on heat transfer and flow characteristics of 4 widths. It was reported that performance criterion for varying widths ranged between 1.58 - 2.35. The authors also analysed synergy and entropy production. Wenbin et al. [14] studied the effects of small cylinder inserts on heat transfer and pressure drop of a tube by means of an experimental setting. Experiments were repeated with small cylinders fitted in different arrangements and with radial pitch variation in Re ranging from 4000 to 18000. Results showed that Nu and f (friction coefficient) increased with decreasing baffle-pitch. Anvari et al. [15] investigated numerically and experimentally effects of conical cut-off turbulators of different sizes fitted in a tube on heat transfer and pressure drop. It was reported that the Nu calculated numerically were in agreement with those obtained from experiments.

Turgut and Kizilirmak [16] numerically determined heat transfer characteristics of turbulent flow in a tube with semi-circular baffles. The authors analysed using different pitch arrangements, pitch angles and a range of Re from 3000 to 50000. Results showed that pitch arrangement, pitch angle and Re had significant impact on heat transfer and flow. Tu et al. [17] investigated effects of S-shaped channels fitted in a tube with turbulent flow in specific arrangements and numbers on heat transfer and fluid flow characteristics for several Re's. Using water as the fluid, the authors stated that S-shaped channels offered very high heat transfer performance. Xu et al. [18] investigated numerically the effects of delta wing type vortex generators (VGs) fitted in the tube on heat transfer and friction. The results of their analysis showed that VGs had a significant effect on Nu and f, which also indicated thermal performance of the VGs. Many researchers demonstrated heat transfer enhancement of twisted baffles fitted in tubes. Hasanpour et al. [19] published a literature review on twisted tapes of different specifications. This study addressed experimental studies focusing on the above-mentioned technique and underlined the importance of the need to select the most efficient type of twisted tape.

Effects of Re for different sinusoidal decoupled strip elements placed separately from the pipe in turbulent flow on heat transfer and flow characteristics were experimentally investigated by Altun et al. [20]. Corrugated strip elements were positioned axial to the pipe centre, which caused a sine wave used as turbulator. Experiments were repeated for three different amplitudes of sinusoidal strip elements with 3D/4, D/8, 3D/16 and D/4 widths (D = inner diameter of test pipe). It was seen that Nu increased as the amplitude value increased. Berber et al. [21] experimentally investigated effects of cylindrical aluminum and Cr-Ni alloy pins placed in pipe for turbulent flow on heat transfer and flow characteristics at different Re's.

In this study effects of sinusoidal wavy strip elements on heat transfer and pressure drop are investigated experimentally. Experiments were repeated at different Re's for turbulent flow. These types of flow and flow arrangements are common in food and chemical industries.

MATERIALS AND METHODS

Experimental Set-up

Figure 1 shows schematically the experimental set-up used in the study. The test set-up consists of three parts: inlet, test and outlet. The set-up includes various measuring instruments. In all three sections, AISI 304 L-grade steel tube with an internal diameter of 70 mm and a wall thickness of 3 mm was used. The inlet channel was designed to be 30 pipe diameters long to transfer the sucked air. The air flow rate was achieved by a fan (220/380 V, 0.75 kW, 2800 max RPM) with 1.8 - 3.1 m^3/min. flow rate. The air flow rate was adjusted by a chimney valve and an electric motor speed controller. In addition, in order to prevent vibration caused by the fan motor, the fan inlet channel connection was provided by a hose made of flexible rubber material. Bakelite gaskets were placed between the flanges to reduce heat loss and to provide airtightness at the inlet and outlet of the test channel. In order to achieve thermal development, the test section was selected as 15 pipe diameters long. The outer surface of the test channel was insulated with glass wool to reduce heat loss. The test tube generated heat because of direct application of electricity to the tube. The electrical circuit had a 2-kW variac, 0-1000A ampermeter and 0-1V voltmeter. The current was supplied through thick copper-nickel alloy bars attached to the inlet and outlet of the test probe. Thus, a homogeneous heat generation was achieved.

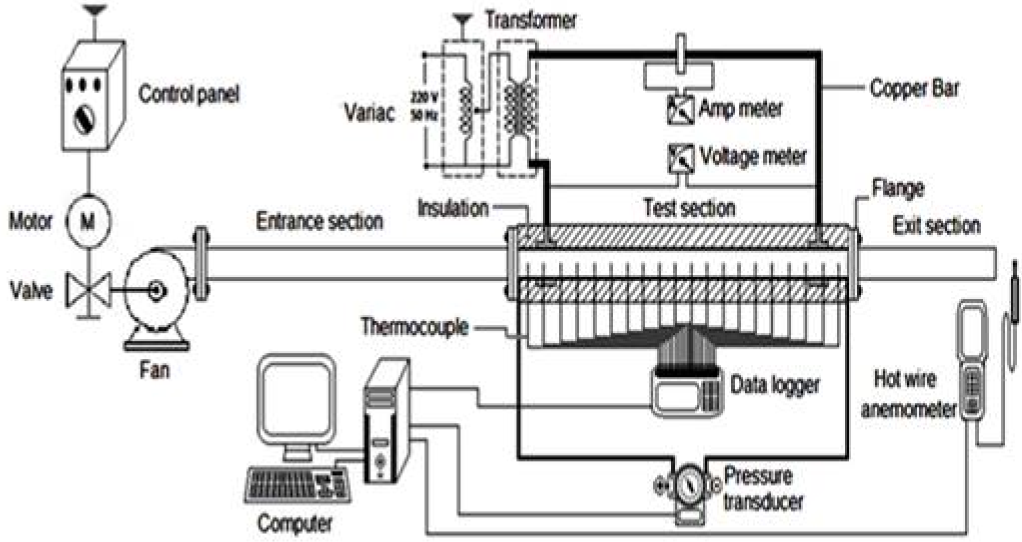

Figure 1. Schematic diagram of experimental set-up

In order to measure the surface temperature of the heated test tube, 20 stations were equally spaced along the tube. To prevent the asymmetric effect of the radial air flow due to the turbulators, two thermocouples were placed at each station such that the circumferential angle was 90°. In this way a total of 40 K-type thermocouples were placed on the outer surface of the test tube. The thermocouples were also installed to measure the inlet and outlet temperatures of the test pipe at 2 separate points. The outer surface temperature of the insulation was also measured. The pressure difference between the inlet and outlet of the test section was measured by means of a digital pressure gauge. The velocity of the air passing through the test tube was measured by an anemometer positioned at 8-pipe-diameter length from the inlet of outlet pipe. Figures 2 and 3 show the sinusoidal turbulators located in the test tube in the experiments and Table 1 shows the geometric dimensions of the turbulator. Dimensions of the strip elements were determined according to Altun et al. [20]. The amplitude value was chosen as it gave the best thermal performance. In order to determine the effects of width of this amplitude value on heat transfer and flow characteristics, three different widths were tested. The turbulator made of 1-mm-thick stainless AISI 304 steel was placed in the test tube as in Figure 4.

Table 1. Geometrical dimensions of sinusoidal turbulator

n (amplitude)	a (width)	λ (wave length)
D/4	3D/4	D/2
	D/2	
	D/4	

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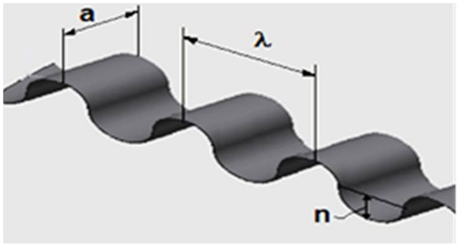

Figure 2. Sinusoidal turbulator

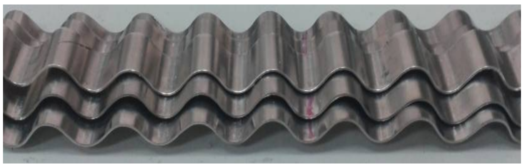

Figure 3. Turbulators used in experiment

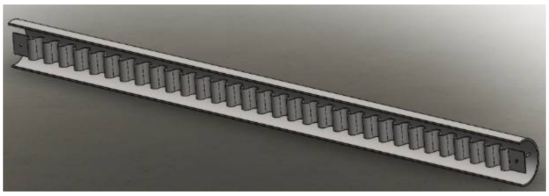

Figure 4. Sinusoidal turbulator in test channel

Data Reduction

The experiments were repeated at different Re's with constant surface heat flux and constant flow rate. Temperature data obtained from the experiments were recorded instantly at 2 div./Hz via a data logger. Before starting the experiments, the air flow was changed to the value read from the flow meter by means of the gate valve to reach the desired Re value. During the experiments, the temperature was allowed to fix along the test tube so that the flow regime became thermally stable. It was observed that the experimental set-up reached the thermal regime approximately 7200 seconds after the initiation. Then this period was observed to be approximately 5400 seconds in each trial. The data obtained through the data collector were plotted over time and experimental measurements were taken when the thermal equilibrium was achieved. The measurements taken during the experiments were the temperature of test tube, temperature of outer surface of the insulation, input and output temperatures of air feed, environmental air temperature, fluid velocity, pressure difference between test tube inlet and outlet, heater circuit current, and voltage. The cross-sectional view of the test pipe is given in Figure 5.

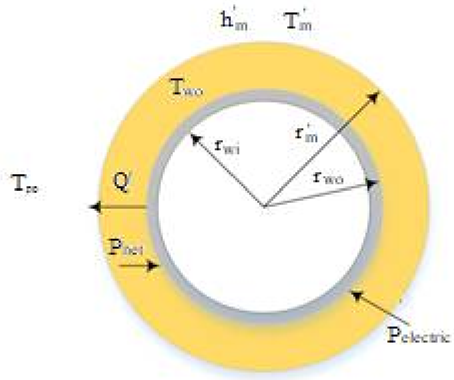

Figure 5. Cross section of test pipe (Q'=heat loss, P_{net}=net heating power, $P_{electric}$=electrical power, h_m'= isolation heat convection coefficient, T_{wo}=outer surface temperature in pipe, T_∞=ambient temperature, r_{wi}= inner radius of pipe, r_{wo} = outer radius of pipe, r_m'= isolation radius, T_m'= isolation temperature)

Accordingly, the heat loss from the outer surface of the insulation to the environment can be calculated [22]:

$$Q' = h_m' \cdot A' \cdot \left(T_m' - T_\infty\right) \quad ; \tag{1}$$

the heat transfer coefficient of the outer surface of the insulation can be calculated as [22] :

$$h_m' = 1.24 \cdot \left(T_m' - T_\infty\right)^{1/3} \quad ; \tag{2}$$

and the net electrical power due to direct electrical current at input and output ends of the test tube is [22]

$$P_{net} = \Delta V \text{ I} - Q' \ , \tag{3}$$

where Q' in equation 3 is the amount of heat lost from the outside of the test tube. According to the heat balance at the steady-state flow condition, Q' was found between 2-5% of the input electrical power ($P_{elec} = \Delta VI$). Other data were obtained as follows [22, 24]:

- Heat flux (obtained from the electrical current applied to the test tube):

$$q = \frac{P_{net}}{\pi D_o L} \tag{4}$$

- Heat generated per unit volume of the tube wall :

$$\dot{q} = \frac{P}{2\pi({r_{wo}}^2 - {r_{wi}}^2)L} \tag{5}$$

- Inner surface temperature in association with outer surface temperature:

$$T_{wi_x} = T_{wo_x} - K\dot{q} \tag{6}$$

- Factor K used in above equation:

$$K = \frac{r_{wo}^2}{2k_w}\left[\ln\left(\frac{r_{wo}}{r_{wi}}\right) - \frac{1}{2}\left(1 - \frac{r_{wi}^2}{r_{wo}^2}\right)\right] \tag{7}$$

- Bulk temperature of the fluid:

$$T_b(x) = T_b(x - \Delta x) + \frac{P_{net}\Delta x}{\rho \dot{V} c_p L} \tag{8}$$

- Local heat transfer coefficient throughout the test tube at the x-axial distance:

$$h(x) = \frac{q}{T_{wi} - T_b(x)} \tag{9}$$

- Nu, temperature gradient:

$$Nu(x) = \frac{h(x)D}{k_h} \tag{10}$$

- Pressure difference between the inlet and outlet of the test tube and coefficient of friction in the tubes with the help of the air flow:

$$f = \frac{\Delta P}{\frac{1}{2}\rho U_m^{\,2} \frac{L_p}{D}} \tag{11}$$

- Performance factor:

$$\eta = \left(\frac{Nu_t}{Nu_s}\right)\left(\frac{f_s}{f_t}\right)^{1/3} \tag{12}$$

Uncertainty Analysis

Errors from measuring devices on the experimental findings expressed as uncertainty were calculated with the equation given by Kline and McClintock [23]:

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \dots\dots\dots\dots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2\right]^{1/2} \tag{13}$$

$$R = R\ (x_1, x_2, x_3, x_4, \dots\dots x_n) \tag{14}$$

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 w_R is the error rate of R size. As a result, the uncertainty was determined as ± 0.0230 for heat transfer coefficient, ± 0.0271 for Nu and ± 0.0145 for friction coefficient.

RESULTS AND DISCUSSION

To investigate the effect of the sinusoidal turbulator on heat transfer and flow characteristics, experiments were first carried out with a straight tube having an Re ranging between 17000-42000. The results obtained were compared with the common existing data.

Figure 6 shows a comparison of Nu's obtained from our experiments performed for the plain tube with those from the equations of Petukhov, Colburn and Dittus-Boelter [24]. It is seen that the Nu obtained those from the present experiments are in good agreement with the literature values for all tried Re's.

Figure 7 shows a comparison of the friction coefficients obtained from our experiments for the plain tube with those from the equations of Petukhov [24]. Our friction coefficient curve for all tried Re's agrees with that from Petukhov's equation. With increase in Re, the coefficients of friction obtained from both the present work and the Petukhov equation decrease with a straight slope.

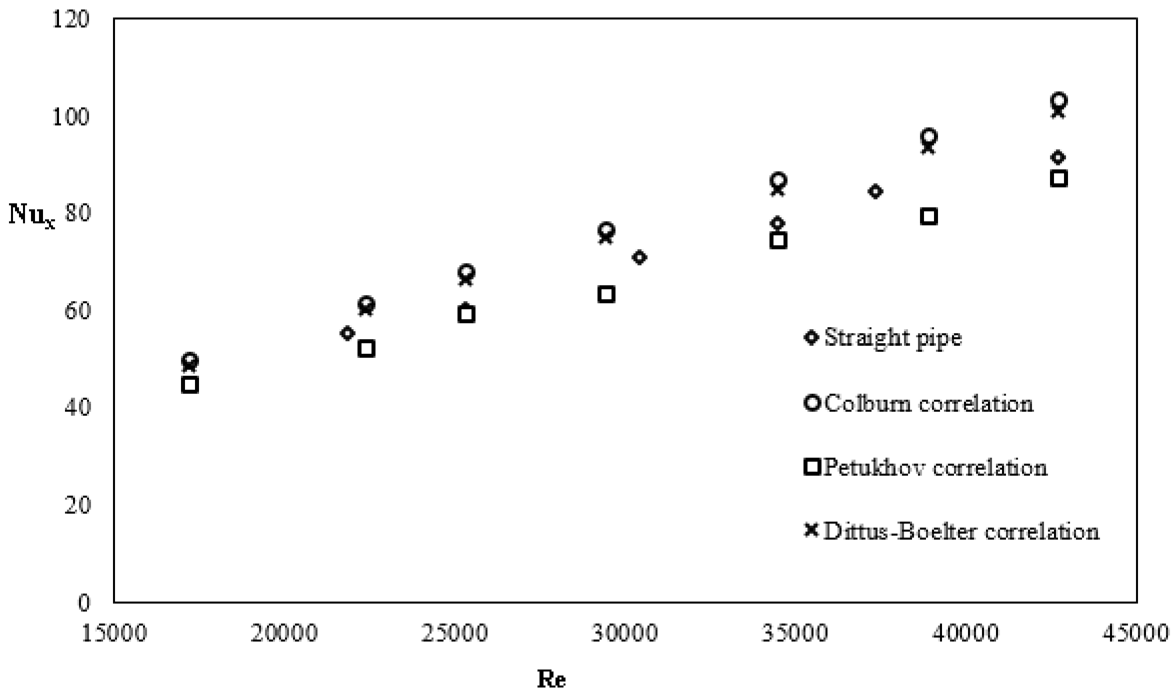

Figure 6. Comparison of experimental Nu's with literature correlations

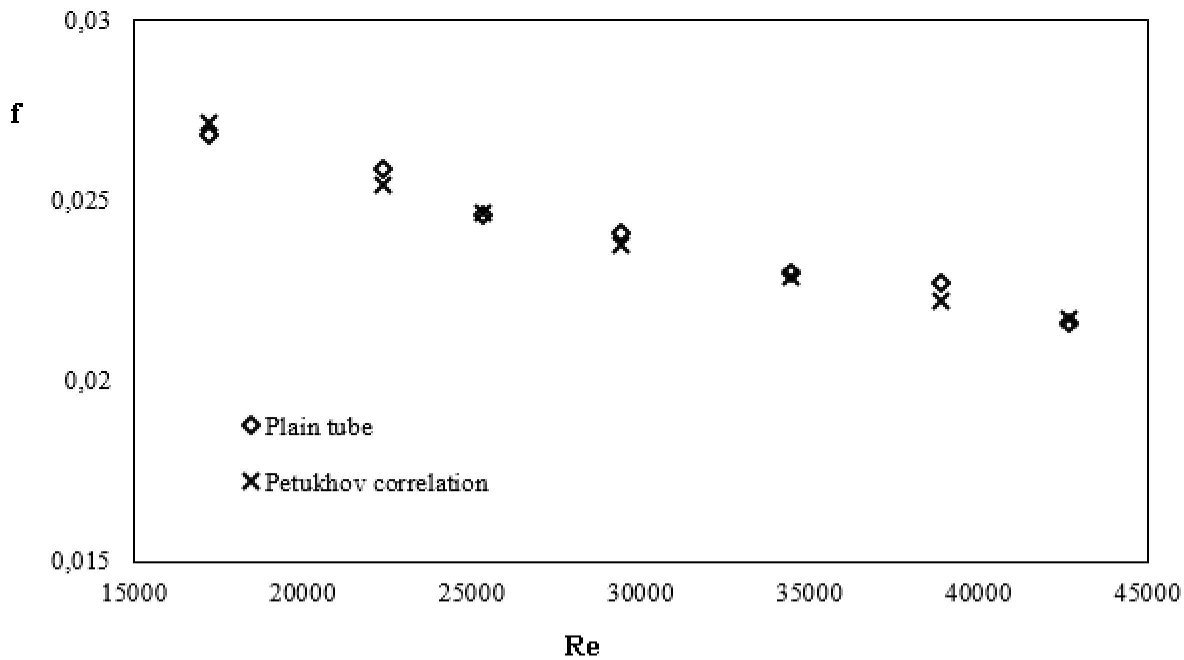

Figure 7. Comparison of experimental friction coefficients (f) with Petukhov correlation

Three widths (a = D/4, D/2 and 3D/4) and Re ranging between 17000-42000 were used for the experiments to resolve the effects of sinusoidal turbulator of D/4 amplitude on heat transfer and flow characteristics. The results are given in terms of Nu for heat transfer and in terms of friction coefficients for flow characteristics. Figures 8.1-8.3 show the variation of sinusoidal turbulator with three widths along the channel. When the shapes are examined, the first result is that the Nu curves show a similar tendency for all Re's. As Re increases, Nu also increases. However, when the curves in Figures 8.1-8.3 are examined, it is also observed that the increased rate of Nu is almost the same compared to Re, whereas Re ranging from 17000 to 22000 is larger than the others. It is also found that the Nu is higher at the inlet parts of the test tube as seen in Figures 8.1-8.3. This is due to the fact that the fluid flow is in contact with the turbulators at inlet of the test pipe, which leads to a further increase in heat transfer through the wall by creating additional turbulence and vortices. After the inlet, it is seen that Nu decreases along the channel and it is almost unchanged after about x/D = 13 (x/D = axial distance of test pipe / inner diameter of test pipe). When the variation of Nu according to x/D is examined, it is observed that Nu increases with increasing width value. However, it is understood that the increase rate at high Re is less and the Nu's for D/2 and 3D/4 are close to each other.

In Figure 9 results are given for 7 flow rates, i.e. corresponding to 15000 <Re <45000 and 3 width values. Nu values in the thermally developed region are determined for the turbulator tube with D/4, D/2, 3D/4. In addition, Nu values in the thermally developed region are determined in the straight pipe. When 4 different Nu curves are examined, slope lines increase proportionally with the Re. When there are no obstructions blocking the flow in the pipe, the Nu values range between 40-80. As the width of the turbulator increases, convective heat transfer also increases. At the maximum flow rate of 15.42 m/s, the highest Nu is obtained. Nu is also a parameter of the thermal performance factor. Since thermal performance factory (η) and Nu parameters are directly proportional, the maximum thermal recovery is achieved when the turbulator with a = 3D/4 is used. As a result, the four curves in Figure 9 indicate that the Nu value varies as a function of Re, fluid velocity and turbulator width.

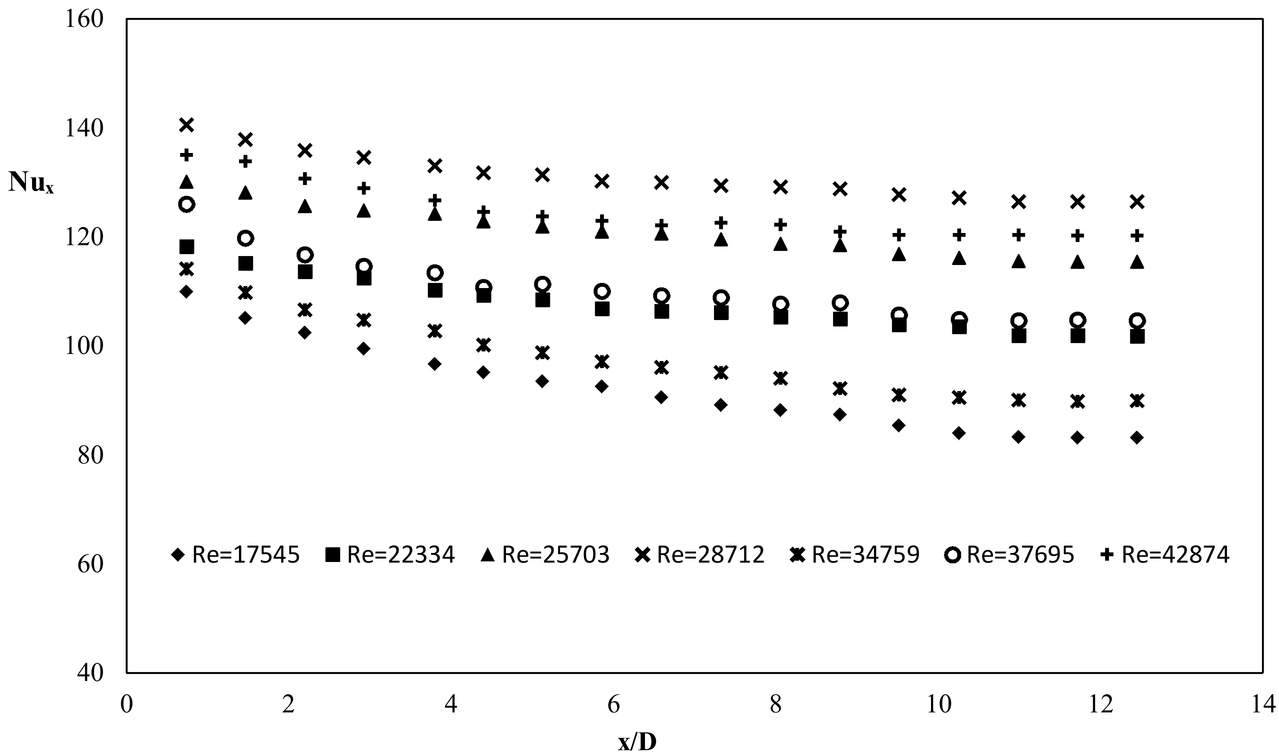

Figure 8.1. Axial distributions of Nu for turbulator with a=D/4

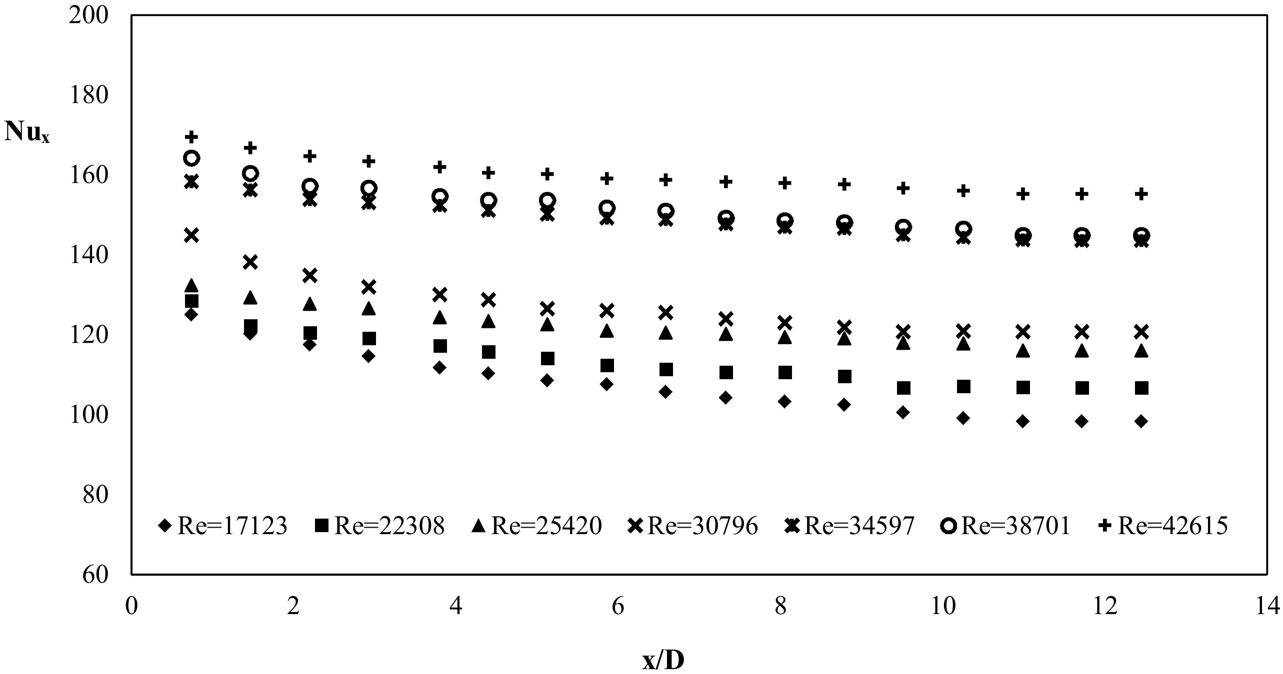

Figure 8.2. Axial distributions of Nu for turbulator with a=D/2

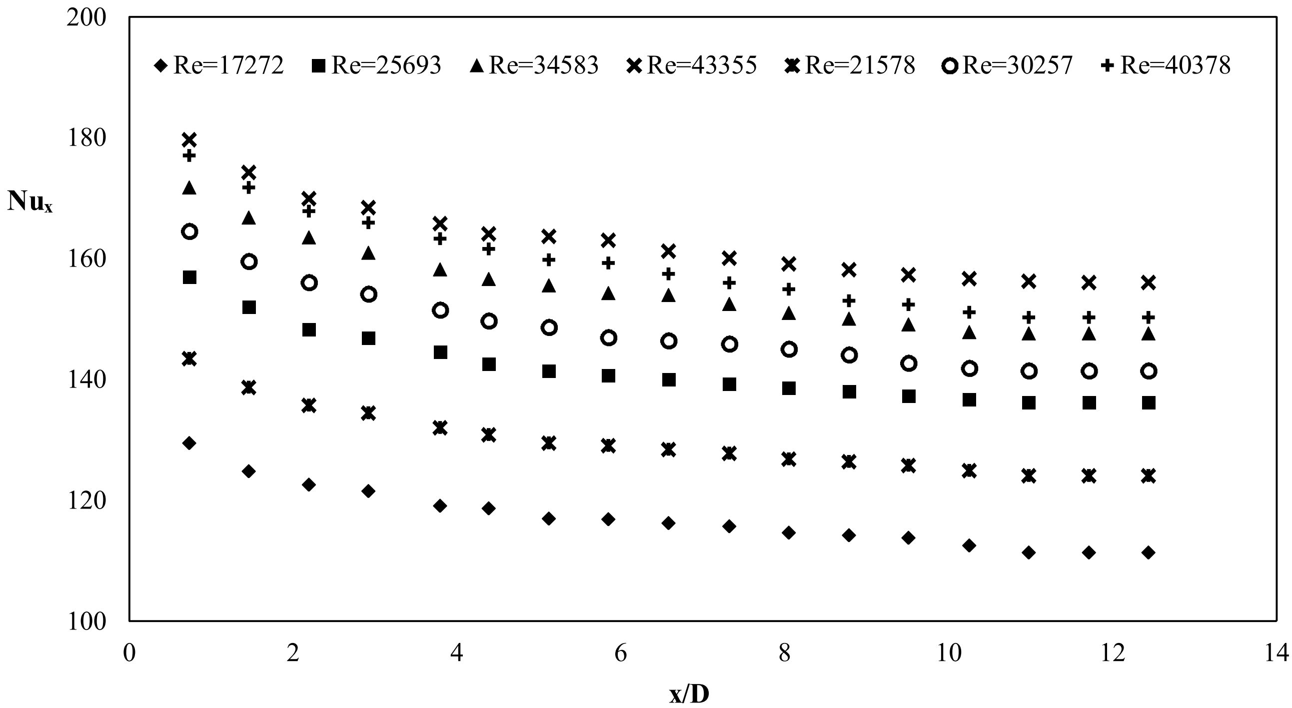

Figure 8.3. Axial distributions of Nu for turbulator with a=3D/4

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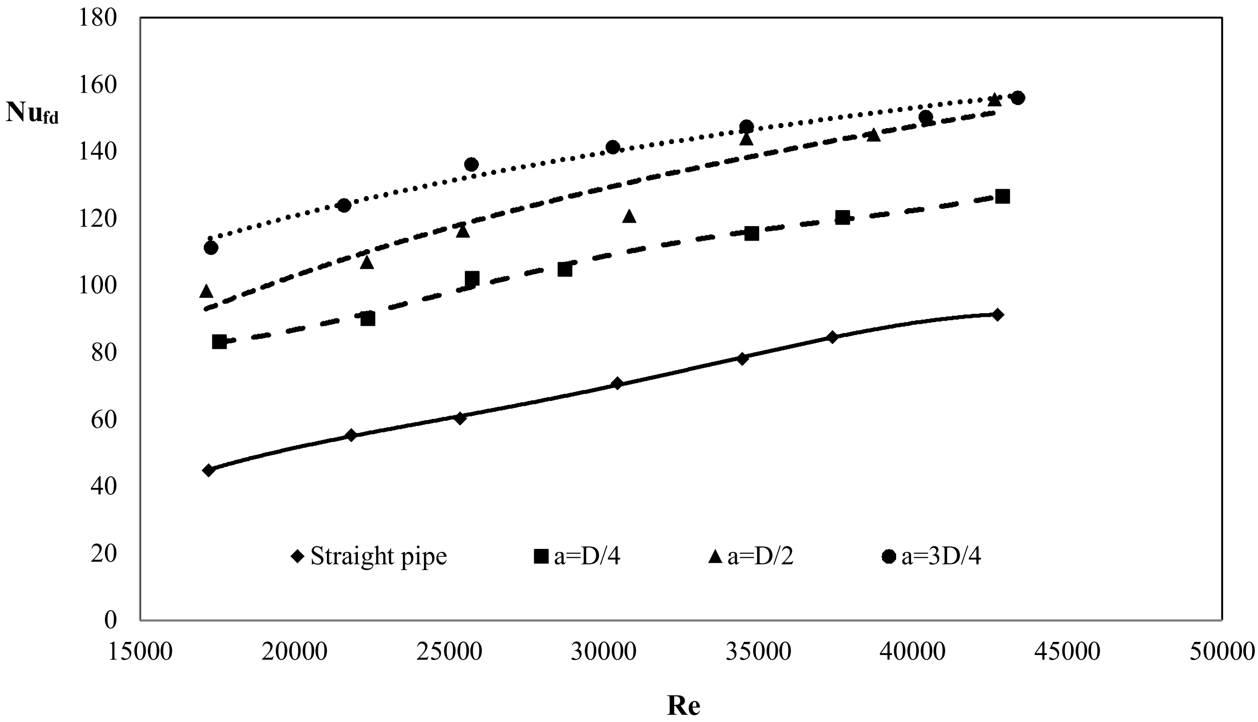

Figure 9. Change in Nu for fully developed turbulent pipe flow (Nu_{fd}) as a function of Re of sinusoidal turbulator

Figure 10 shows the variation of the friction coefficient of the turbulators at different Re's. The friction coefficient values increase with increasing width values due to the blockage rate. The most important result is that the coefficient of friction is much more influenced by the width value. As the coefficient of friction increases, pressure drop causes obstructions and the time required for the flow to leave the test tube increases. Therefore, pumping demand also increases. Friction is an undesirable physical phenomenon as it increases pumping costs. Increased friction value in thermal energy storage systems has a negative effect on thermal performance factor in the system. Therefore, the friction value in such systems can be controlled only by considering the thermal recovery-cost balance.

Figure 11 shows thermal performance factor values for all experiments. The thermal performance factor is an important parameter for determining the potential of a sinusoidal turbulator for most of industrial applications. The thermal performance factor which depends on constant pumping power criteria is calculated by Equation (12). Figure 11 shows a correlation between the thermal performance factor and Re for different widths of sinusoidal turbulators. The use of turbulator with 3D/4 width gains benefit in the enhancement of heat transfer compared to twisted sinusoidal turbulator with D/4 and D/2 widths. However, the decrease in thermal performance factor is due to friction loss with increasing Re. The thermal performance factor is less than unity for turbulator with width ratios of 3D/4, D/4 and D/2 for constant amplitude of D/4 and wavelength of D/2. That is to say, the friction factor penalty is more dominant than the heat transfer enhancement ratio for all tried cases. Thus, the use of turbulator width of 3D/4 at higher Re's is thermodynamically disadvantageous due to the higher pressure loss and friction factor compared to turbulator widths of D/4 and D/2. The maximum value of thermal performance factor of 0.969 is

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found in the case of sinusoidal turbulator width ratio 3D/4, amplitude ratio D/4, and wavelength ratio D/2.

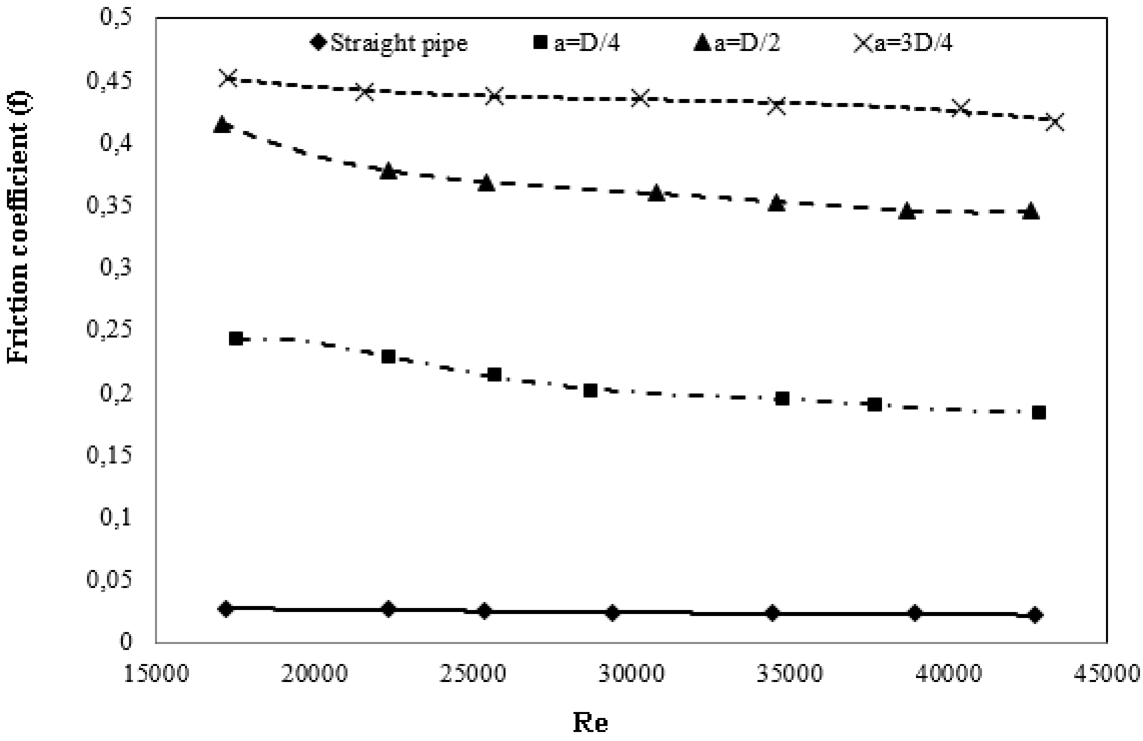

Figure 10. Friction coefficient as a function of Re's of all experiments

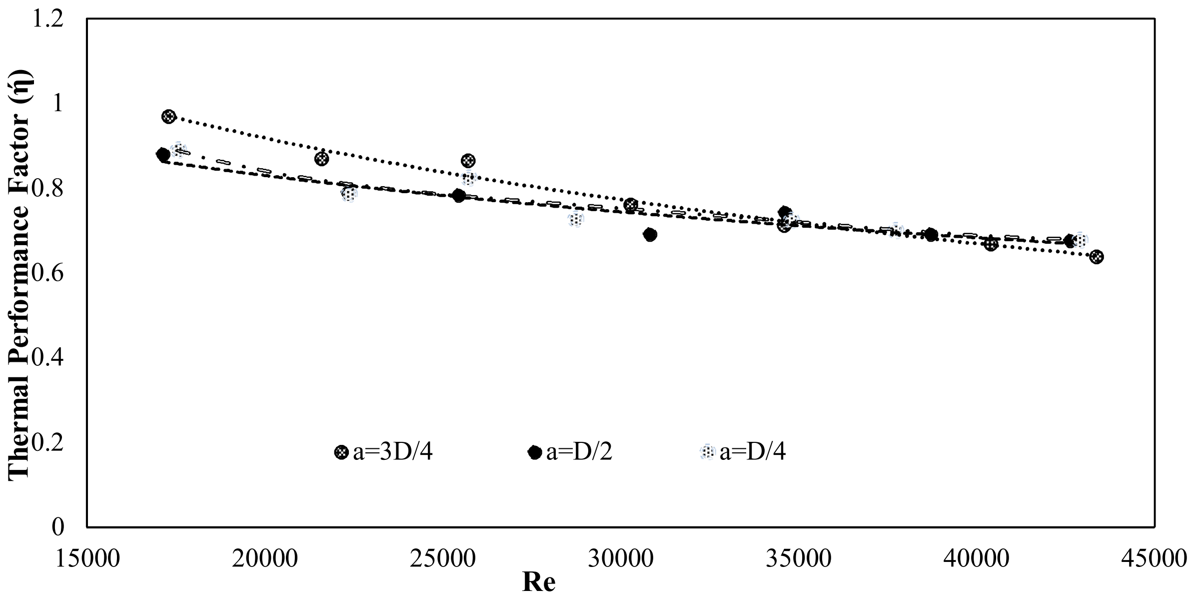

Figure 11. Change in thermal performance factor with respect to Re's for all experiments

In order to validate the accuracy of the study, results were compared with literature. For example; S.Eiamsa-ard and P.Promvonge [25] experimentally investigated the effects of V-nozzle inserts placed under the effect of a uniform heat flux in a channel on the heat transfer and friction properties. In their studies, they arranged the V-nozzle elements in the test tube for the three different step ratios. They worked in the range of 4000<Re<20000. The increase rate of Nu value ranged from 60% to 200%. They found that the thermal recovery rate for the smallest step ratio was between 0.82-1.19. In another study, S. Eiamsa-ard et al. [26] placed twisted strip elements positioned on different axes in a circular channel. Forced-convection heat transfer was investigated in a working range of 5200 <Re <22000. According to the straight tube, Nu in the tube with strip element increased by 24-62%. The thermal performance factor varied between 0.85-1.03 depending on Re. They also observed that thermal performance factor had very close values according to Manglik and Bergles correlations [27]. In another similar study Ibrahim et al. [28] studied experimentally and numerically the thermal and hydrodynamic behaviour of the flow due to conical turbulators. They worked in the range of 6000<Re <25000. The conical turbulators were fixed in different arrangements within the channel. In addition, different turbulator inlet and outlet diameter ratios (d/D) were used. They observed that thermal performance factor varied in the range of 0.544-0.893 based on plain tube for d/D = 0.3-0.4-0.5. Nu varied between $Nu_{0.3}$ = 102-160, $Nu_{0.4}$ = 138-230, and $Nu_{0.5}$ = 164-280. Dittus Boelter, Petukhov, Colburn correlations [24] and the results obtained from the above-mentioned studies are compatible with the experimental results.

CONCLUSIONS

In this study heat transfer and friction loss were investigated experimentally for sinusoidal turbulators placed in tubes having turbulent air flow. The experiments were performed for different width values and Reynolds numbers, and axial distributions of heat transfer and flow characteristics were observed. The results obtained can be summarised as follows.

The reason for high Nu values at the inlet of the test tube is that the flow significantly increases the heat transfer by convection of the turbulence and entrainment of the vortex. As the width increases, Nu increases. Nu values are 39-86% in the test tube with turbulator D/4 wide, 70-119% with turbulator D/2 wide and 72-148% with turbulator 3D/4 wide, compared to the straight pipe. It has been determined that the coefficient of friction is much more affected by the sinusoidal decay than by the width of the turbulator. It is also determined that the coefficient of friction is about 0.02 in the straight pipe, 0.184-0.242 in n = D/4, 0.345-0.415 in n = D/2, and 0.416-0.451 in n = 3D/4. In addition, an increase in the friction coefficient increases the pressure losses and the power required for pumping. Therefore, as the turbulator width increases, the thermal performance factor decreases based on increasing pressure losses.

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