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Thermal characterization of turbulent tube flows over diamond-shaped elements in tandem

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ABSTRACT

Experiments have been conducted to investigate the heat transfer and friction factor characteristics of the fully developed turbulent airflow through a uniform heat flux tube fitted with diamond-shaped turbulators in tandem arrangements. In the experiments, strong turbulence and recirculation flow is expected by using tandem diamond-shaped turbulators (D-shape turbulator) connected to each other by a small rod and placed inside the test tube. The parameters for this study are consisted of Reynolds number (Re) from 3500 to 16,500, the included cone angle ($\theta = 15^\circ$, 30° and 45°), and the tail length ratio ($TR = l_t/l_h = 1.0$, 1.5 and 2.0) defined as the ratio of the tail length (l_t) to the head length of turbulator (l_h). The variation of Nusselt number and friction factor with Reynolds number under the effect of those parameters are determined and presented. The experimental result reveals that the heat transfer rate increases with increasing Reynolds number and the included cone angle (θ) but decreases with the rise of the tail length ratio (TR). This is because of the mixing of the fluid in the boundary layer thereby enhancing the convective heat transfer and increasing pressure loss. For the tube with the turbulator of $\theta = 45^\circ$, the heat transfer enhancement is found to be 67%, 57% and 46% for tail length ratio, $TR = 1.0$, 1.5 and 2.0, respectively. Correlations of the Nusselt number (Nu) and friction factor (f) are developed for the evaluation of interactive effects of using the turbulators on the heat transfer and pressure loss. The good agreement between the experimental and the correlated results is obtained within 5–7% deviation. In addition, the heat transfer enhancement efficiency determined under constant pumping power is also provided.

1. Introduction

Heat transfer augmentation techniques are of interest in engineering designs for increasing the thermal performance of heat exchangers over the past decades. In general, enhancing the heat transfer by using the passive method is popular and applied in many engineering applications for example; heat exchanger system, refrigeration system, electronic cooling, boiling system, etc. This is because this method does not require the extra external power sources, for example the turbulence promoter device (*surface coatings, rough surfaces, extended surfaces, wire coil, conical-ring, and V-nozzle*), the swirl flow devices (*insertion of twisted tape, helical tape, snail entry, guide vane, twisted tube*) and additives for liquid and gases. It is well established that the turbulence intensity created in the medium facilitates efficient transportation of heat from the tube wall. For the high turbulent flow, it can improve convection heat transfer of the tube wall by increasing the effective axial Reynolds number, decreasing the cross-section flow area, and increasing the mean velocity and temperature gradient.

Heat transfer enhancement using several types of the turbulence promoters mounted in a tube has been extensively studied for the past decades [1]. Ayhan et al. [2] carried out experimental and numerical works to study the heat transfer in a tube fitted with truncated hollow cone turbulators. Influences of the cutting out conical turbulators, placed in a heat exchanger, on the heat transfer rates with four different types of the cutting out conical turbulators and four conical angles (5°, 10°, 15° and 20°) were described and reported by Durmus [3]. In his work, the heat transfer, pressure loss and exergy analyses were determined for the conditions with and without turbulators. Yakut et al. [4] studied the effect of conical-ring turbulators on the heat transfer, pressure drop and flow-induced vibrations. Their experimental data were analyzed and presented in terms of the thermal performances of the heat transfer promoters with respect to the heat transfer enhancement efficiencies at constant pumping power. They also found that the maximum entropy generation, at the same Reynolds number,

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Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of air, J kg^{-1} K^{-1}
d	diameter of D-shape turbulator, m
D	inside diameter of test tube, m
f	friction factor
h	heat transfer coefficient, W m^{-2} K^{-1}
I	current, A
k	thermal conductivity of air, W m^{-1} K^{-1}
L	length of the test section, m
l_h	head length of D-shape turbulator, m
l_t	tail length of D-shape turbulator, m
M	mass flow rate, kg s^{-1}
Nu	Nusselt number
ΔP	pressure drop, Pa
Pr	Prandtl number
Q	heat transfer rate, W
Re	Reynolds number
t	thickness of test tube, m
T	temperature, K
$\tilde{T}$	mean temperature, K
TR	tail length ratio, l_t/l_h
U	mean axial velocity inside the test section, m s^{-1}
V	voltage, V
$\dot{V}$	volume flow rate, m^3 s^{-1}

Greek symbols

ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}
μ	dynamic viscosity, N s m^{-2}
θ	included cone angle
η	enhancement efficiency

Subscripts

a	air
b	bulk
con	convection
i	inlet
o	outlet
p	plain tube
pp	pumping power
t	turbulator
w	wall

occurs in sequence by the conical rings with 10, 20 and 30 mm pitches, respectively, while the conical rings are thermodynamically advantageous ($N_{s,a} < 1$) up to 8000 Reynolds number with respect to entropy generation. Yakut and Sahin [5] conducted the flow-induced vibration behaviors of conical-ring turbulators used for heat transfer enhancement in heat exchangers. Vortex-shedding frequencies and amplitude were determined and flow-acoustic coupling was also analyzed experimentally. They observed that as the pitch increases, vortex-shedding frequencies also increase and the maximum amplitudes of the vortices produced by conical-ring turbulators occur with small pitches. Promvonge and Eiamsa-ard [6] carried out an experiment to investigate the heat transfer and friction factor in a uniform heat flux tube with conical nozzle turbulators. In their work, the turbulators were placed in the test tube with two arrangements: diverging nozzle and converging nozzle. Promvonge and Eiamsa-ard [7] again examined the influences of the conical turbulator combined with the snail entrance on the heat transfer rate and flow friction in a heat exchanger tube. They observed that each application of the conical nozzle and the snail can help to considerably increase the heat transfer rate over the plain tube by about 278% and 206%, respectively. Subsequently, Eiamsa-ard and Promvonge [8] studied the heat transfer enhancement in a tube by using V-nozzle turbulators at several pitch ratios. They showed that using the V-nozzle can help to maximise the heat transfer rate at about 270% over the plain tube with a maximum gain of 1.19 on enhancement efficiency, obtained at the smallest pitch ratio. Promvonge [9] investigated the effects of the conical-ring turbulator inserts with three different diameter ratios of the ring to tube diameter ($d/D = 0.5, 0.6, 0.7$) and three different arrangements (converging conical ring, diverging conical ring, and converging–diverging conical ring) on the heat transfer rate and friction factor in a tube. Akansu [10] presented a numerical heat transfer and pressure drop analysis for porous rings inserted in a tube under constant heat flux conditions by using the SST k-ω model. He reported that the highest heat transfer is found at $H/D = 0.2$ and $L/D = 1.0$. Recently, Charun [11] examined the heat transfer and pressure drop characteristics of the water and air through a vertical tube fitted with a nodular turbulizer. The turbulizer was made in the form of a cascade of spherical nodes (nodules) placed in the axis of the vertical tube. He found that the tube with turbulizer leads to the increase in convective heat transfer around 2–4 times.

In the present work, the major interest is to study the heat transfer and friction factor characteristics in a uniform heat flux tube equipped with diamond-shaped turbulators. In the experiments, the air was the tested fluid and flowed through the test tube while the outer surface of test tube was heated with heating wires. The test was conducted for the airflow rates in the range of $3500 < \mathrm{Re} < 16{,}500$. The high level of the turbulence intensity was generated by the separation of a thermal boundary layer occurring between pairs of the turbulators which depend on the included cone angle ($\theta = 15°$, $30°$ and $45°$) and the tail length ratio (TR = 1.0, 1.5 and 2.0) used. Heat transfer and friction factor in the test tube were made for the similar conditions with and without diamond-shaped turbulence promoters. Some empirical correlations expressing the results are also reported.

2. Experimental discription

2.1. Experimental set-up

A schematic diagram of the experimental apparatus with the basic components and fluid flow systems is shown in Fig. 1. The test facility is consisted of (1) test tube with a set of turbulence promoter, (2) a set of thermocouple for check the bulk air temperature and the mean tube wall temperatures, (3) orifice meter to measure the volumetric airflow rates at downstream of the test section, (4) a 7.5 kW blower for the suction of airflow through the test tube, (5) a three-Phase inverter to control the speed of the blower, (6) variac transformer to control the heat flux at the tube wall, (7) a data logger for receiving the data signal from the thermocouple type K or resistance temperature detectors (RTDs), and sending to the main PC (8) for calculation, and (9) U-tube/inclined manometer to check the head loss in the test tube with specific gravity ($SG.$) of 0.826.

2.2. Detail of D-shape turbulators

Fig. 2a–c shows the details of test tube and the arrangement of D-shape turbulence promoters. The inside and outside diameters of

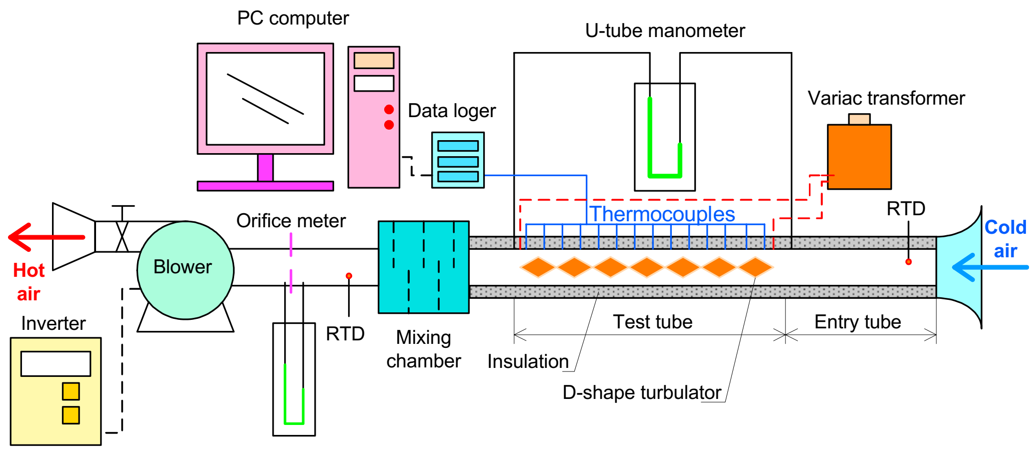

Fig. 1. Schematic diagram of the experiment set-up: (1) inverter, (2) blower, (3) rotameter, (4) insulated test tube, (5) electrical heating coil, (6) variac transformer, (7) U-tube manometer, (8) a set of thermocouple, and (9) data logger.

the copper test tube having a length of $L = 1250$ mm and 1.5 mm thickness (t), were 47.5 mm (D) and 50.5 mm (D_o), respectively. The D-shape turbulator was made of stainless steel with 23 mm (d) in diameter. The turbulators in the present study were consisted of (1) three different tail length ratios (TR $= l_t/l_h = 1.0$ (27 elements), 1.5 (21 elements) and 2.0 (18 elements)) defined as the ratio of the tail length of turbulator (l_t) to the head length (l_h) and (2) three included cone angle; $\theta = 15°, 30°$ and $45°$. In each test run, the head length (l_h) was kept constant at 23 mm. The turbulator was mounted in the test tube by connecting with the small rod along the test tube. The rod was made of stainless steel with 3 mm in diameter. The turbulator placed in the test tube is used as the turbulence promoter by generating the re-circulation/reverse flow along the test tube. Details of the experimental apparatus and procedure are similar as reported in earlier paper [8].

2.3. Procedure

In the experiments, the tube was heated by continuously wound flexible electrical wire, which provided a uniform heat flux

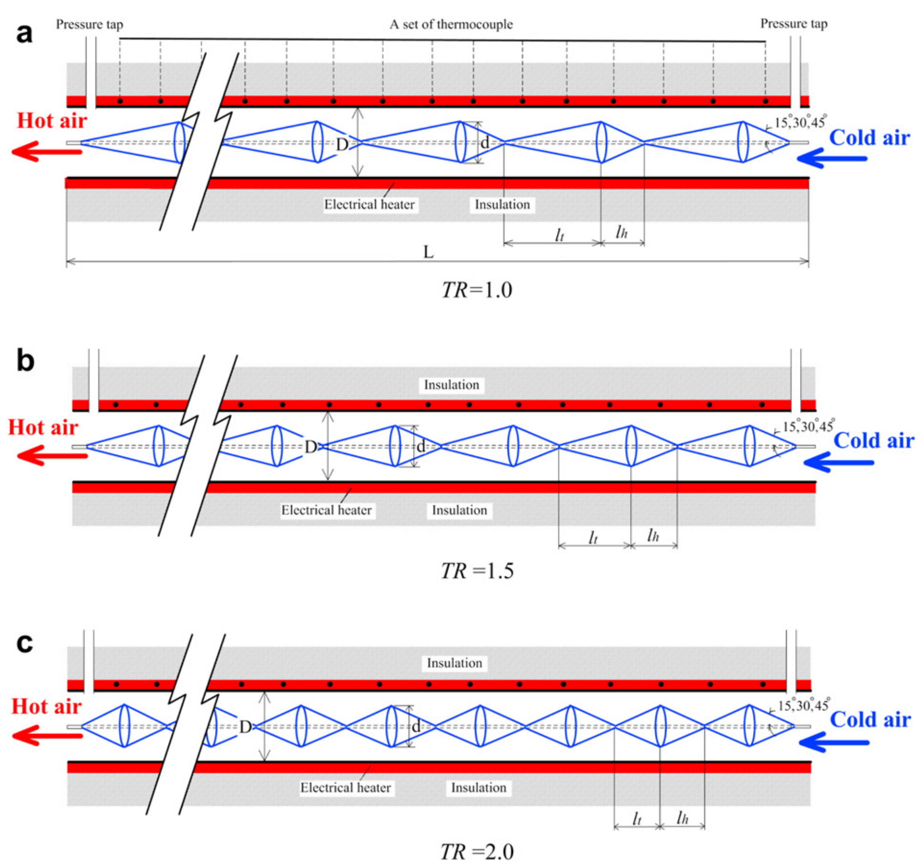

Fig. 2. Details of test tube with D-shape turbulators: (a) $TR = 1.0$, (b) $TR = 1.5$ and (c) $TR = 2.0$.

boundary condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 3 A. The outer surface of the test tube was well insulated to minimize convective heat loss to the surroundings, and necessary precautions were taken to prevent leakages from the system. The inner and outer temperatures of the bulk air were measured at certain points with a multi-channel temperature measurement unit in conjunction with RTD. Ten thermocouples were taped on the local wall of the tube, and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean wall temperature was determined by means of calculations based on the reading of the Chromel-constantan thermocouples. In the apparatus setting above, the inlet bulk air at 27 °C from a high speed blower was directed through the heat transfer test section and passed to an orifice meter. The airflow rate was measured by the orifice meter, built according to an ASME standard. Manometric fluid was used in U-tube manometers with specific gravity (SG) of 0.826 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. The volumetric airflow rates from the blower, situated before the inlet of the test tube, were adjusted by varying the motor speed through the inverter. During the experiments, the bulk air was heated by an adjustable electrical heater wrapped along the test section. Both the inlet and outlet temperatures of the bulk air from the tube were measured by RTD, calibrated within ±0.2 °C deviation by thermostat before being used. It was necessary to measure the temperature at 10 stations altogether on the outer surface of the heat transfer test pipe to determine the average Nusselt number. In each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk air at steady state conditions. A steady state condition was usually obtained after 2–3 h in which all of thermocouples were recorded and noted. The various characteristics of the flow, the Nusselt number, and the Reynolds number were based on the average of tube wall temperature and outlet air temperature. The local wall temperature, inlet and outlet air temperature, the pressure drop across the test section and the airflow velocity were measured for heat transfer of the heated tube. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature. The maximum uncertainties of the non-dimensional parameters are ±5% for Reynolds number, ±10% for Nusselt number and ±15% friction, whereas the uncertainty in temperature measurement at the tube wall was about ±0.5%. The experimental results were reproducible within these uncertainty ranges.

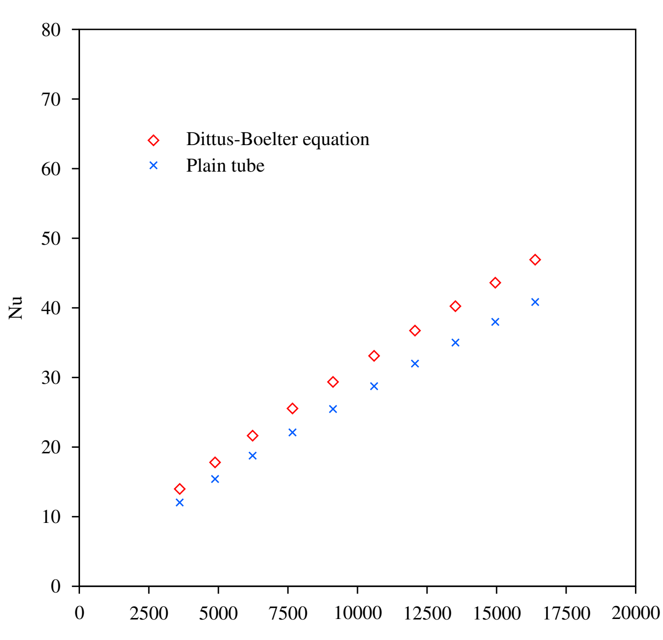

Fig. 3. Validation test of Nusselt number.

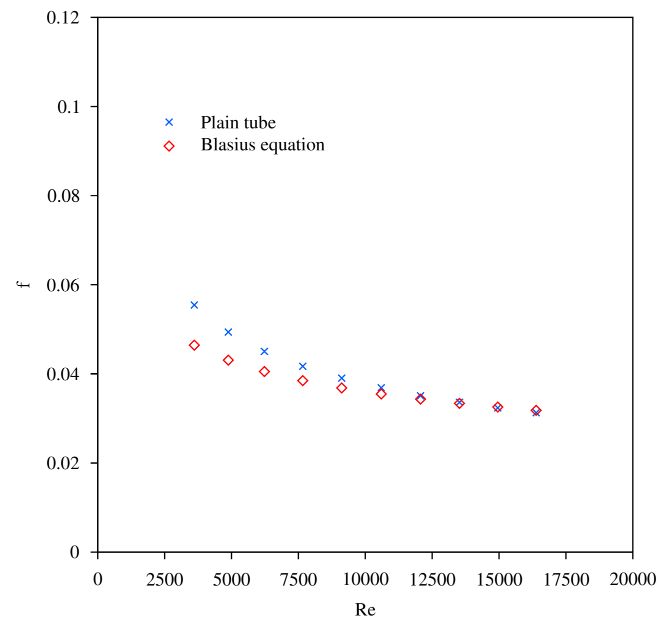

Fig. 4. Validation test of friction factor.

3. Processing data

In the present study, air is used as the test fluid and flowed through a uniform heat flux tube with insulation outside. At steady state condition, the heat transfer increase of the airflow (Q_a) and

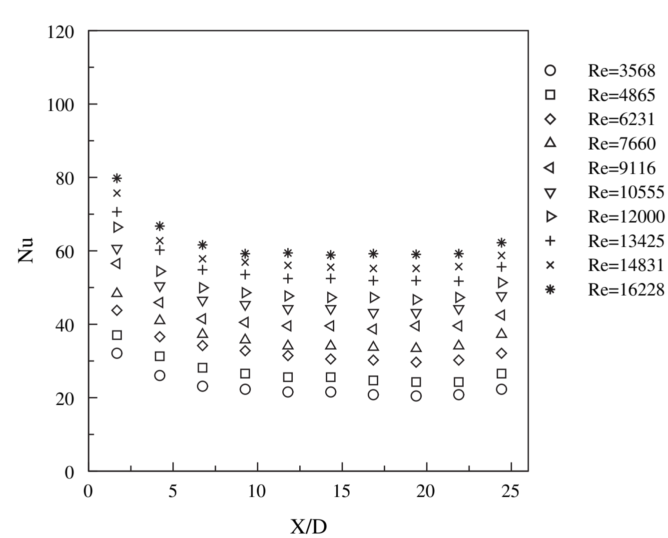

Fig. 5. Axial Nusselt number distribution for $TR = 1.0$ and $\theta = 45°$.

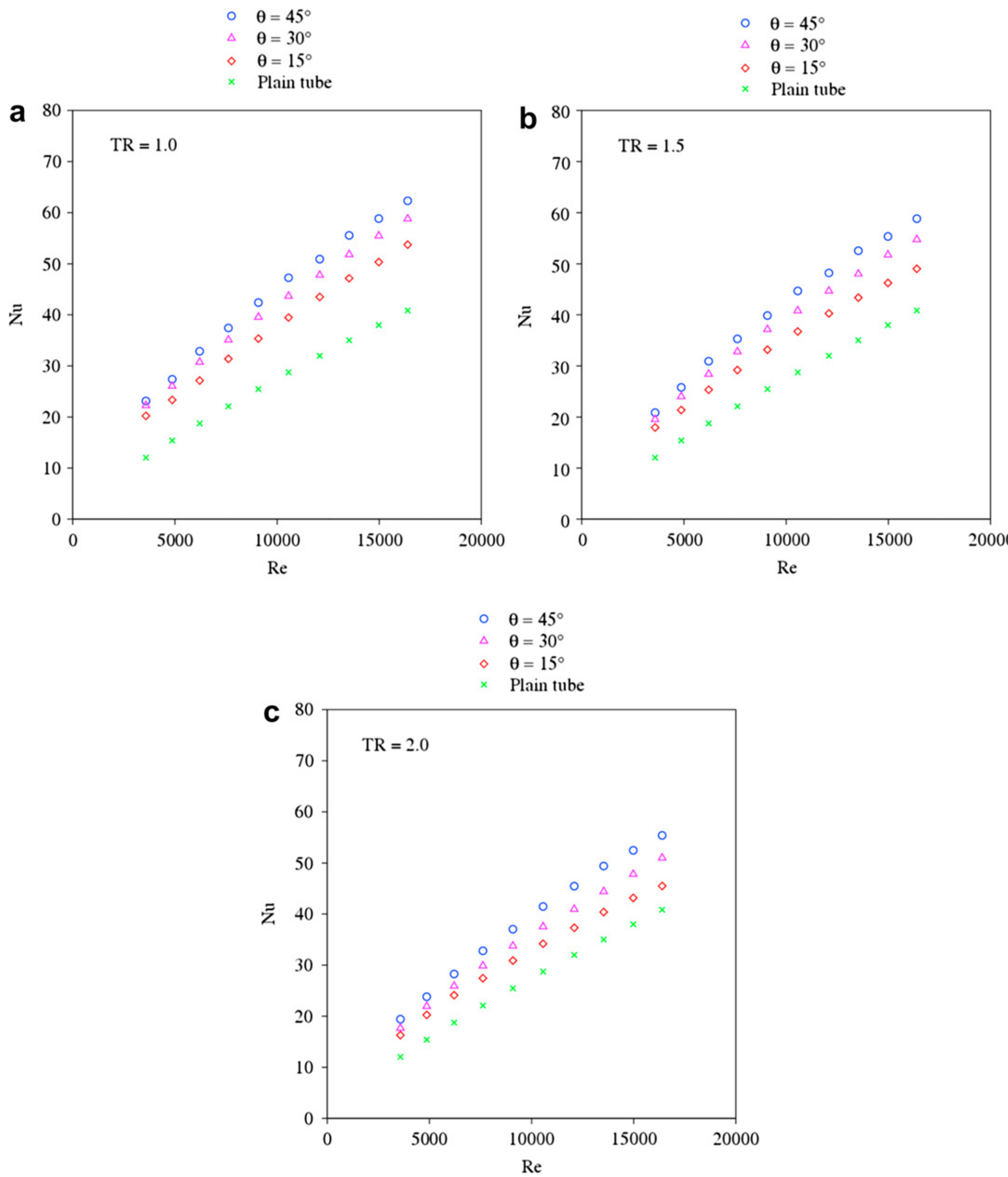

Fig. 6. Effect of included cone angle on Nusselt number for (a) $TR = 1.0$, (b) $TR = 1.5$ and (c) $TR = 2.0$.

the convection heat transfer of airflow in the tube (Q_{conv}) are assumed to be the same which can be expressed as:

$$Q_a = Q_{conv} \tag{1}$$

where

$$Q_a = MC_{p,a}(T_o - T_i) = VI \tag{2}$$

The heat supplied by the electrical winding in the test tube is found to be 0.5–3% higher than the heat absorbed by the fluid for the thermal equilibrium test due to convection and radiation heat losses from the test section to the surroundings. Thus, only the heat transfer rate absorbed by the fluid is taken for the internal convective heat transfer coefficient calculation.

The convection heat transfer from the test section can be written by

$$Q_{conv} = hA(\tilde{T}_w - T_b) \tag{3}$$

The mean bulk temperature of the fluid in the test tube is given by

$$T_b = (T_o + T_i)/2 \tag{4}$$

The mean temperature on the outer wall of the test tube is written as

$$\tilde{T}_w = \sum T_w/10 \tag{5}$$

where T_w is the local wall temperature and evaluated at the outer wall surface of the test tube. The average wall temperature is calculated from 10 points of local wall temperatures lined between the inlet and the exit of the test tube. The average heat transfer coefficient, h and the average Nusselt number, Nu are estimated as follows:

$$h = MC_{p,a}(T_o - T_i)/A(\tilde{T}_w - T_b) \tag{6}$$

$$\mathrm{Nu} = hD/k \tag{7}$$

The Reynolds number is defined as:

$$\mathrm{Re} = UD/\nu \tag{8}$$

Friction factor, f can be written as:

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho\frac{U^2}{2}\right)} \tag{9}$$

in which

$$U = \frac{M}{\rho A} \tag{10}$$

where ΔP is the pressure drop across the test section, L is the length between the pressure taps, D is inside tube diameter, ρ is fluid density, M is air mass flow rate and finally U is the mean axial velocity. All of thermo-physical properties of the air were determined at the bulk air temperature.

4. Results and discussion

The experimental results of the round tube with D-shape turbulator inserts are identified and discussed in the present work. The effects of the included cone angle (θ) and the tail length ratio (TR) on the heat transfer, the flow friction and the enhancement efficiency characteristics in a uniform heat flux tube have been investigated. Experimental data are obtained for the tube with the turbulator inserts of three different tail length ratios, $TR = 1.0$, 1.5 and 2.0 and included cone angles, $\theta = 15°$, 30° and 45°. The data cover a wide range of Reynolds number (Re) between 3500 and 16,500 for heating condition. Figs. 3 and 4 show the data verification of the heat transfer rate and friction factor of the plain tube. The results of the present plain tube are also compared with the published correlations found in the open literature [12]. In the

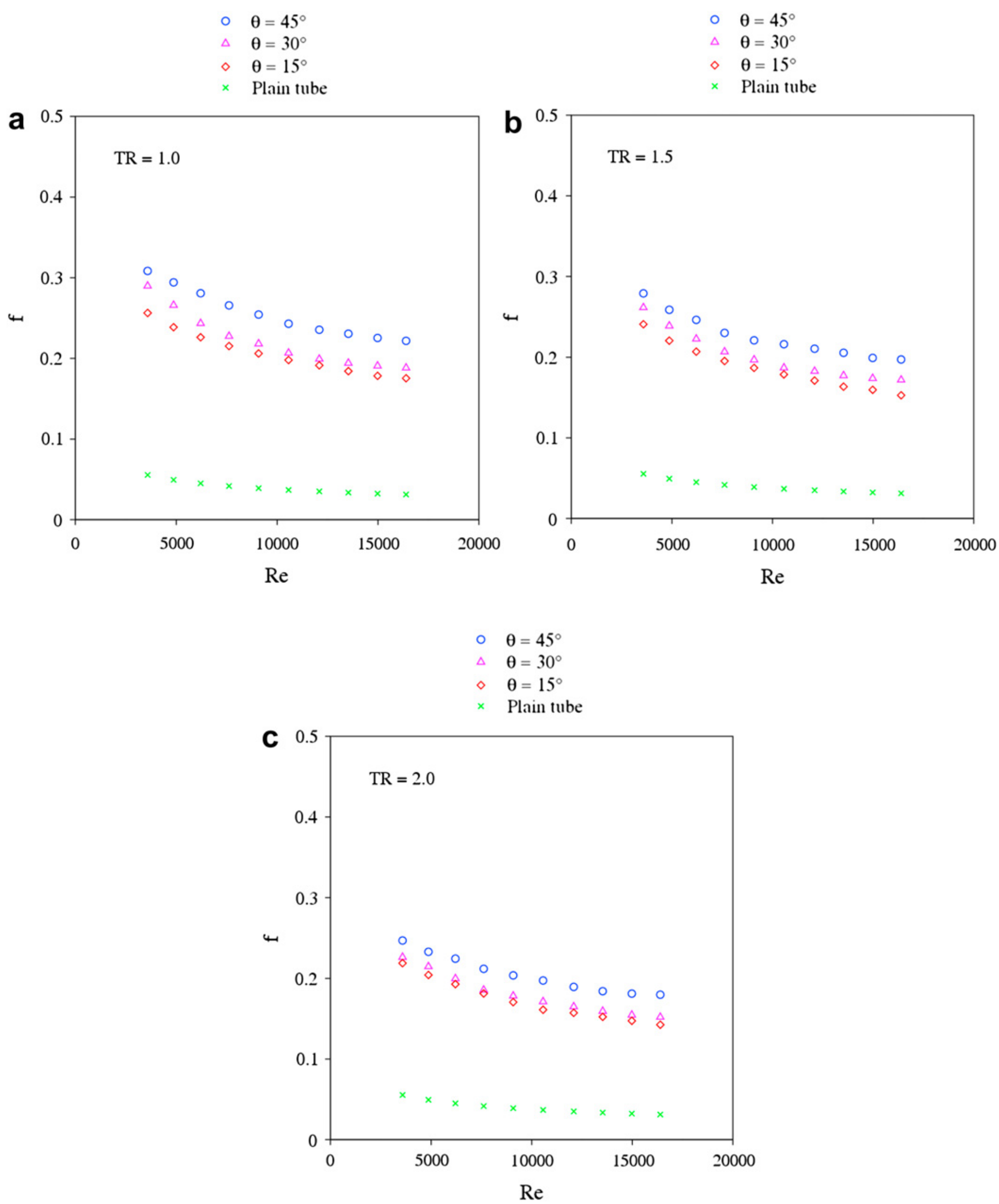

Fig. 7. Effect of included cone angle on friction factor for (a) $TR = 1.0$, (b) $TR = 1.5$ and (c) $TR = 2.0$.

figures, the present results reasonably agree well with the correlations within 5–12% deviation for Nusselt number (Nu) and 10–15% for friction factor (f).

4.1. Axial heat transfer distribution

Fig. 5 presents the axial distributions of Nusselt numbers along the tube fitted with the D-shape turbulator for cone angles $\theta = 45°$ and tail length ratio $TR = 1.0$, for Reynolds numbers ranging from 3568 to 16,228. Obviously, Nusselt number is high at the entry region and gradually decreases to nearly uniform value in region of $7 \le X/D \le 22$. The Nusselt number data in the exit region of $X/D > 22$ slightly increases above the uniform one. It can be also observed that the local Nusselt number increases with increasing Reynolds number due to the increase of turbulent intensity imparted to the flow along the test section.

4.2. Effect of included cone angle (θ)

Fig. 6a–c present the heat transfer variation in terms of Nusselt number of the inserted tube for different the included cone angle ($\theta = 15°$, 30° and 45°) with Reynolds numbers at $TR = 1.0$, 1.5 and 2.0, respectively. In the figures, it is visible that the heat transfer increases with increasing Reynolds number and the cone angle. The larger angle (θ) yields a higher heat transfer rate than the smaller angle. It is expected that the turbulator can give rise to the reverse/turbulent flow and boundary layer eruption causing high convective heat transfer and momentum processes. In addition, the slight difference of the heat transfer rate between $\theta = 30°$ and 45° can be observed. This can be attributed to a strong turbulent flow remain for the large cone angle. The use of smaller the cone angle of $\theta = 15°$ and 30° leads to the decrease in Nusselt number at 16.3% and 7.4%, respectively, in comparison with $\theta = 45°$ of all range Reynolds numbers studied.

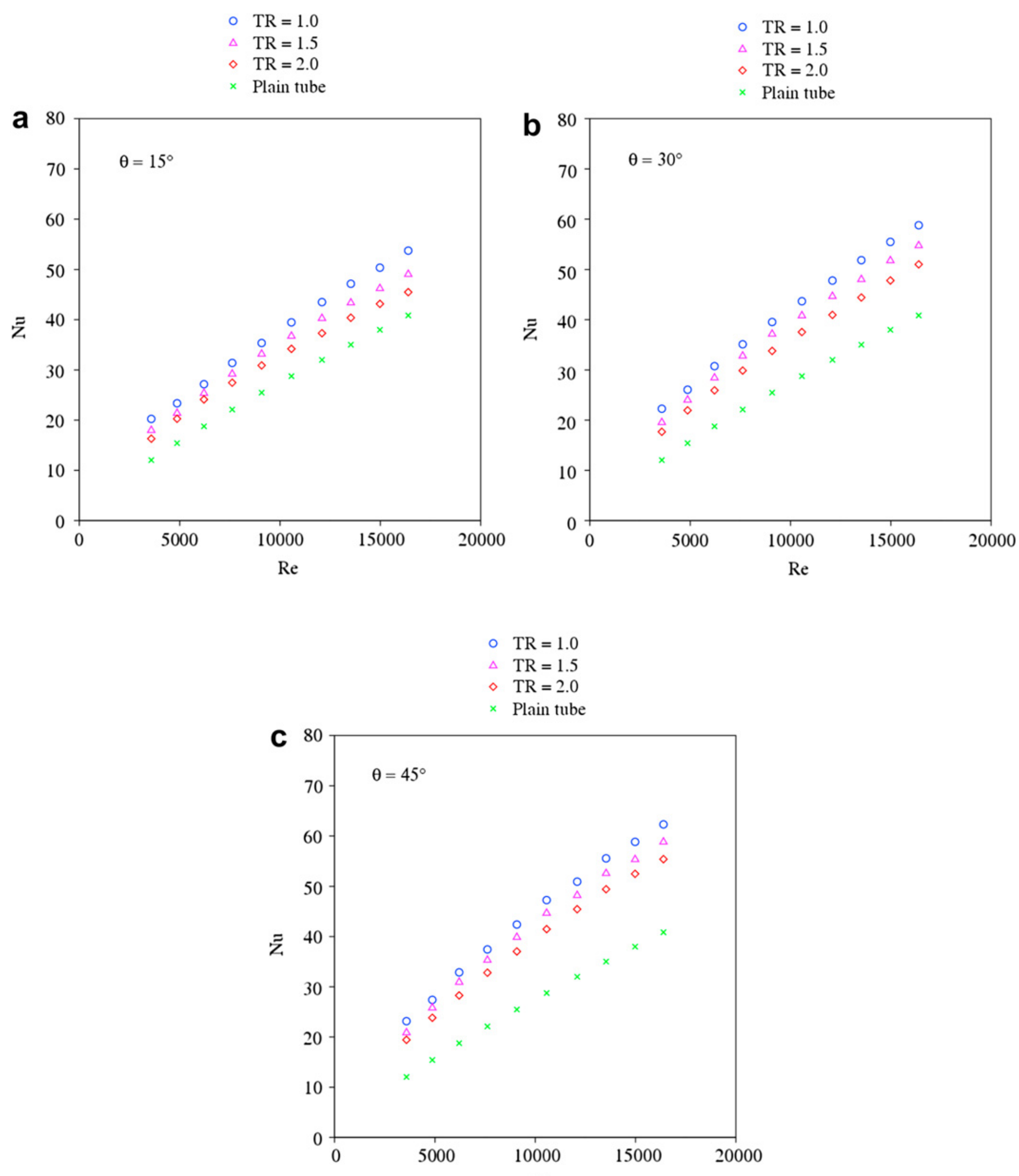

Fig. 8. Effect of tail length ratio on Nusselt number for (a) $\theta = 15°$, (b) $\theta = 30°$ and (c) $\theta = 45°$.

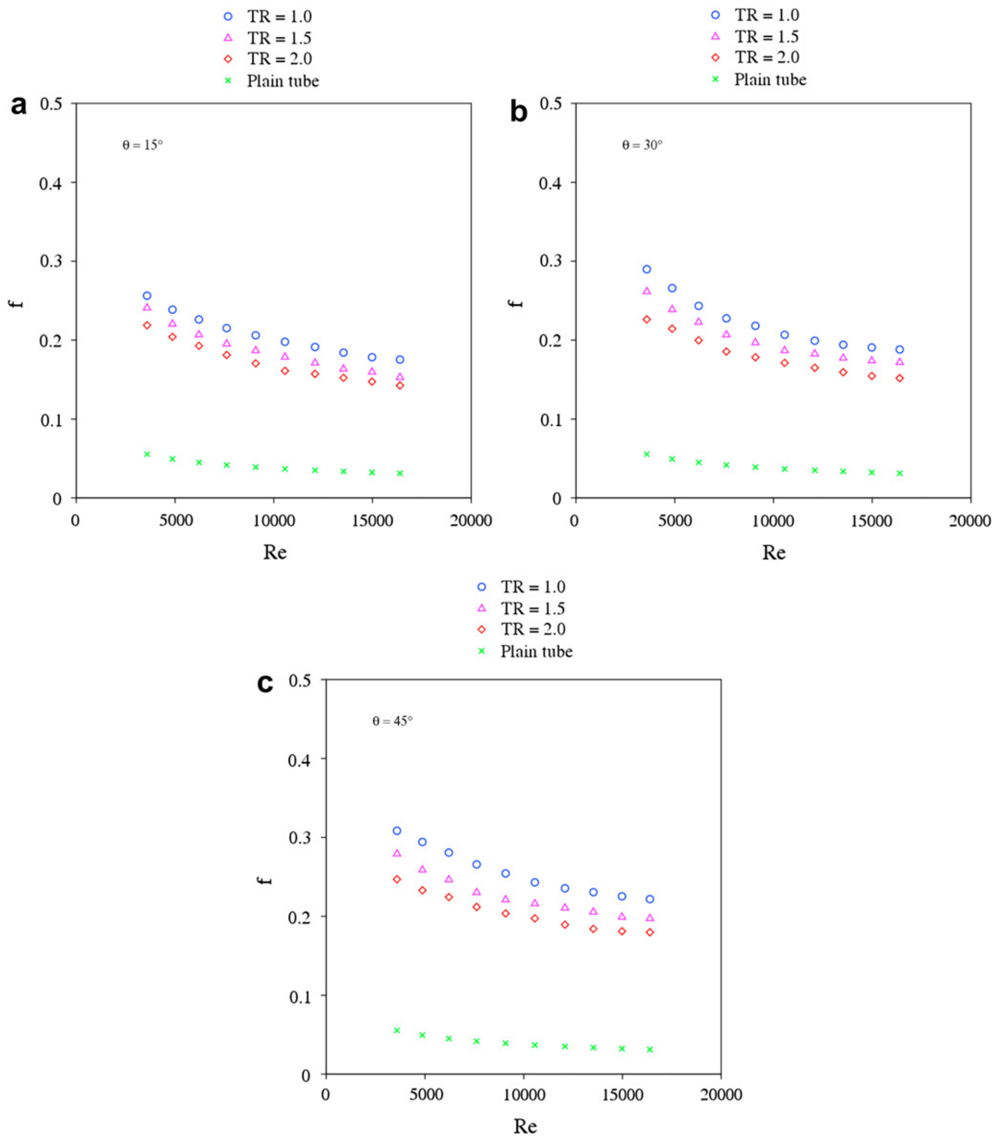

Fig. 9. Effect of tail length ratio on friction factor for (a) $\theta = 15^{\circ}$, (b) $\theta = 30^{\circ}$ and (c) $\theta = 45^{\circ}$.

The friction factor variations of using the three included cone angles ($\theta = 15^{\circ}$, 30° and 45°) with Reynolds number between 3500 and 16,500, are illustrated in Fig. 7a–c for $TR = 1.0$, 1.5 and 2.0, respectively. In this arrangement of the D-shape turbulators, the high turbulence intensity or recirculation can be obtained continuously in the test tube but at different levels depending on the included cone angle. The smaller cone angle would provide lower flow blockage and surface areas that affect considerably the pressure loss of fluid flow. Therefore, the large friction loss comes from the largest cone angle, $\theta = 45^{\circ}$. The use of the small cone angle, $\theta = 15^{\circ}$ results in the reduction of pressure loss around 6.5% and 21.4% lower than that of the larger cone angles, $\theta = 30^{\circ}$ and 45°, respectively.

In addition, it can be found that for using D-shape turbulator with cone angle, $\theta = 15^{\circ}$, 30° and 45° at $TR = 1.0$, the improvement of average heat transfer rate is, respectively, 32.1%, 46.5% and 57.8% higher than those the plain tube while the friction factor is 4.7, 5.25 and 5.67 times of the plain tube.

4.3. *Effect of tail length ratio (TR)*

Effects of the tail length ratios (TR = 1.0, 1.5 and 2.0) on the heat transfer rate in the tube fitted with D-shape turbulators are

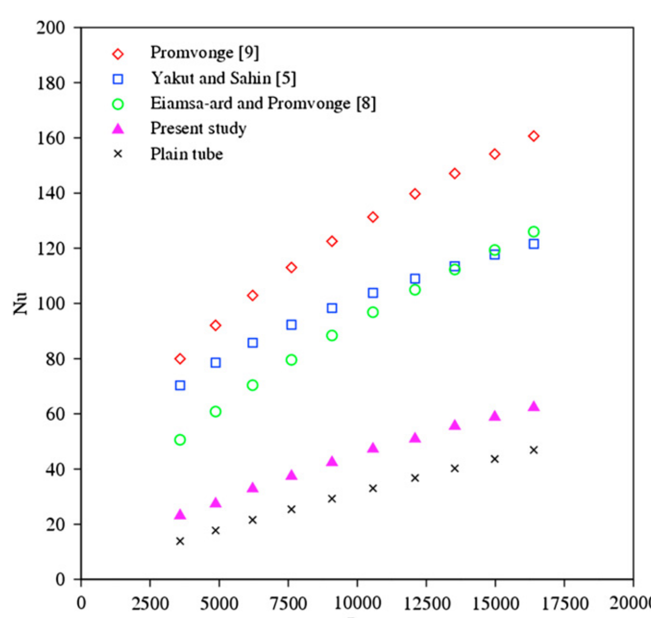

Fig. 10. Comparison between the present work and previous studies.

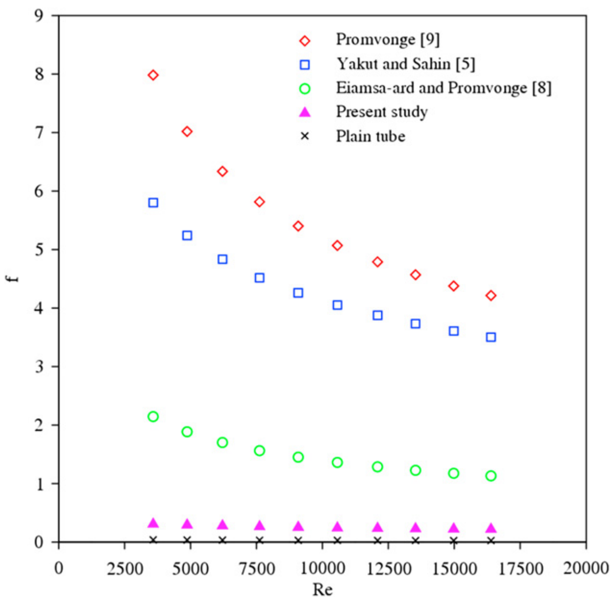

Fig. 11. Comparison of friction factor with previous work.

Table 1
Heat transfer correlations.

Type of turbulator	Investigator	Correlation
1) Plain tube	Present study	$\mathrm{Nu} = 0.02\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}$ for $3,500 \le \mathrm{Re} \le 16,500$
2) D-shape turbulator	Present study	$\mathrm{Nu} = 0.105\mathrm{Re}^{0.676}(\tan\theta)^{0.135}$ $TR^{-0.214}\mathrm{Pr}^{0.4}$ for $3,500 \le \mathrm{Re} \le 16,500$
3) Conical-ring	Yakut and Sahin [5]	$\mathrm{Nu} = 4.497\mathrm{Re}^{0.36}\mathrm{Pr}^{0.4}(t/D)^{-0.0716}$ for $5,000 \le \mathrm{Re} \le 17,000$
4) V-nozzle	Eiamsa-ard and Promvonge [8]	$\mathrm{Nu} = 0.524\mathrm{Re}^{0.6}\mathrm{Pr}^{0.4}(l/D)^{-0.285}$ for $8,000 \le \mathrm{Re} \le 18,000$
5) Conical-ring	Promvonge [9]	$\mathrm{Nu} = 0.863\mathrm{Re}^{0.459}\mathrm{Pr}^{0.4}(d/D)^{-1.32}$ for $6,000 \le \mathrm{Re} \le 26,000$

depicted in Fig. 8a–c for $\theta = 15°$, $30°$ and $45°$, respectively. For all cases, the tube with turbulator inserts provides higher values of heat transfer than the plain tube without turbulator insert. It is seen that the heat transfer increases with increasing Reynolds number

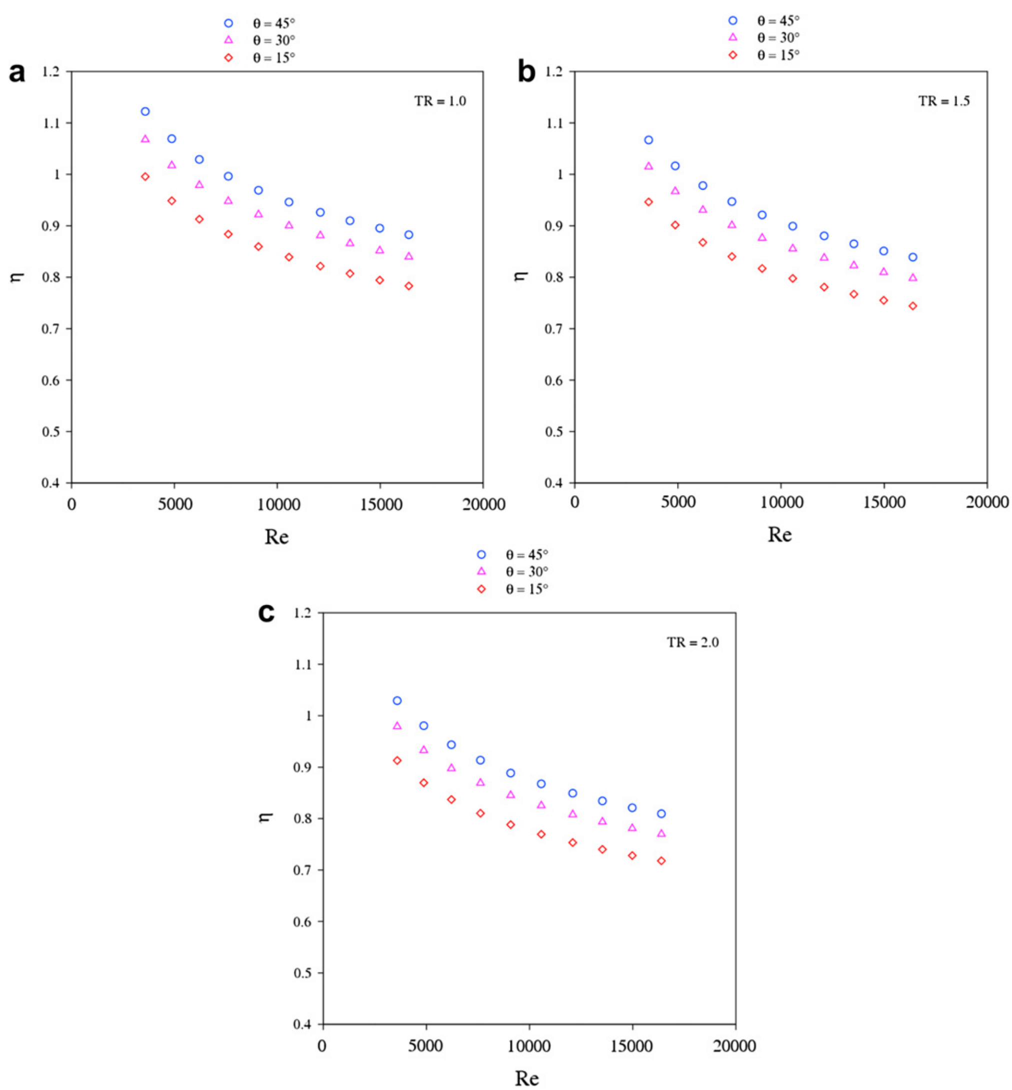

Fig. 12. Effect of included cone angle on enhancement efficiency for (a) $TR = 1.0$, (b) $TR = 1.5$ and (c) $TR = 2.0$.

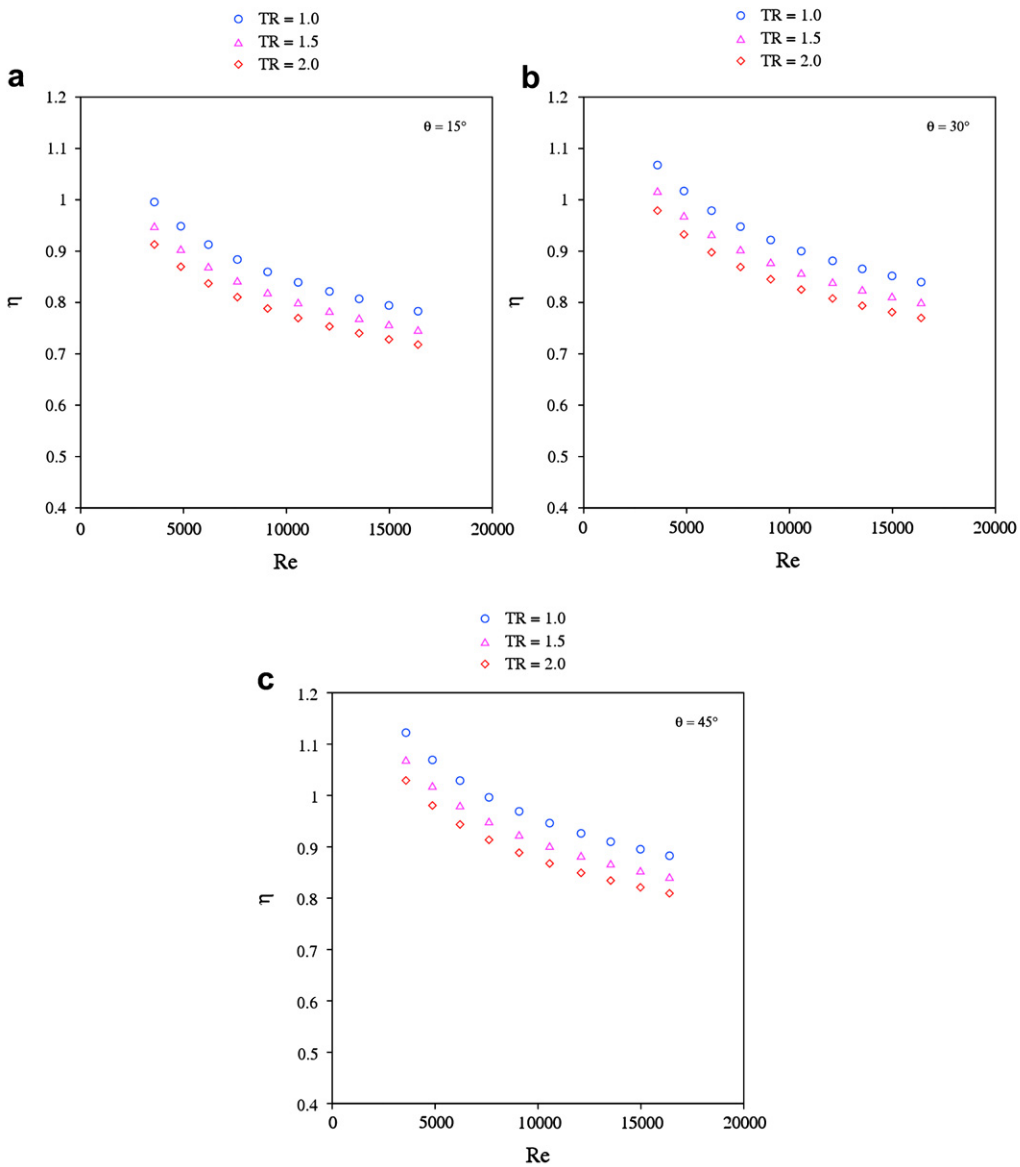

Fig. 13. Effect of tail length ratio on enhancement efficiency for (a) $\theta = 15°$, (b) $\theta = 30°$ and (c) $\theta = 45°$.

but with decreasing the tail length ratio (TR) of turbulators. This indicates that the turbulence intensities in the tube with a short turbulator must be higher than the longer one. The lower tail length ratio means a higher number of the D-shape elements in the test tube. Therefore, the highest heat transfer is achieved with the D-shape turbulator operated in a tube for the lowest tail length ratio (TR = 1.0) in the range of TR investigated.

The variation of the friction loss with the Reynolds number for different tail length ratios (TR) in the test tube is shown in Fig. 9a–c for $\theta = 15°$, 30° and 45°, respectively. It is interesting to note that the friction factor tends to decrease with increasing the Reynolds number and the tail length ratio (TR). As expected, a greater increase in friction factor is found for the turbulator with lower tail length ratio (TR). The maximum increase in friction factor is seen at $TR = 1.0$ which is higher than $TR = 1.5$ and 2.0 around 11.3% and 22.6%, respectively.

Table 2
Friction factor correlations.

Type of turbulator	Investigator	Correlation
1) Plain tube	Present study	$f = 1.19\mathrm{Re}^{-0.375}$ for $3,500 \leq \mathrm{Re} \leq 16,500$
2) D-shape turbulator	Present study	$f = 2.7\mathrm{Re}^{-0.263}(\tan\theta)^{0.143}TR^{-0.291}$
3) Conical-ring	Yakut and Sahin [5]	$f = 97.41\mathrm{Re}^{-0.332}(t/D)^{-0.149}$
4) V-nozzle	Eiamsa-ard and Promvonge [8]	$f = 107\mathrm{Re}^{-0.42}(l/D)^{-0.68}$
5) Conical-ring	Promvonge [9]	$f = 12.52\mathrm{Re}^{-0.42}(d/D)^{-4.31}$

Table 3
Enhancement efficiency correlations.

D-shape turbulator	$\mathrm{Re}_p = 1.36\mathrm{Re}_t^{1.043}(\tan\theta)^{0.055}TR^{-0.11}$ $\eta = 4.096\mathrm{Re}_t^{-0.158}(\tan\theta)^{0.091}TR^{-0.125}$

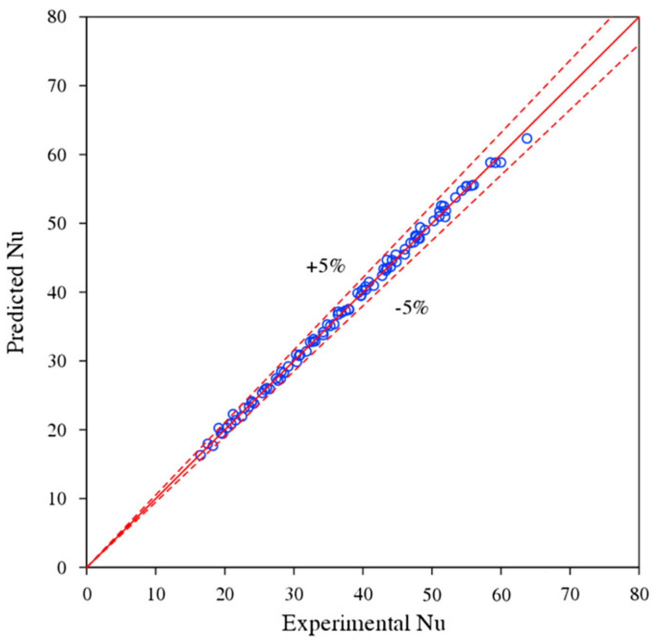

Fig. 14. Correlation predictions of Nusselt number with experimental results.

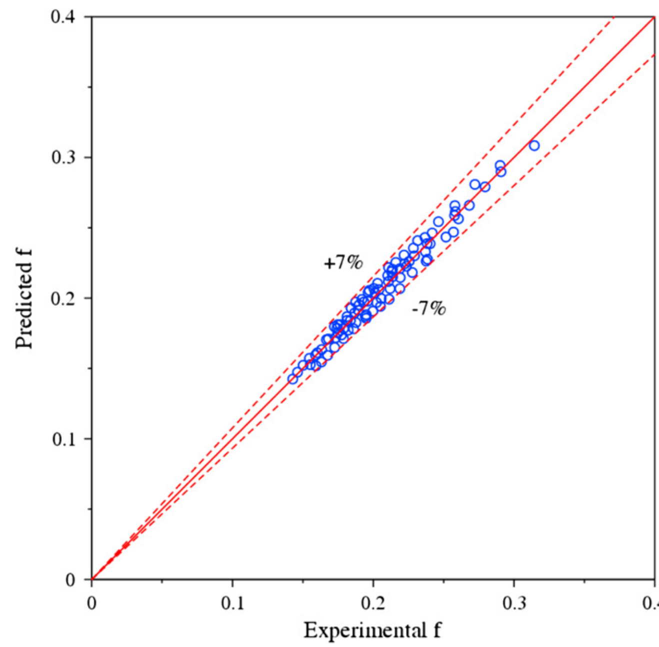

Fig. 15. Correlation predictions of friction factor with experimental results.

From experimental results above, the Nusselt number and friction factor for plain tube are correlated as follows:

$$\mathrm{Nu} = 0.02\ \mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{11}$$

$$f = 1.19\ \mathrm{Re}^{-0.375} \tag{12}$$

The results of the Nusselt number and friction factor for D-shape turbulators with three different cone angles ($\theta = 15°$, $30°$ and $45°$) and tail length ratio (TR = 1.0, 1.5 and 2.0) are correlated as follows:

$$\mathrm{Nu} = 0.105\ \mathrm{Re}^{0.676}(\tan\theta)^{0.135}TR^{-0.214}\mathrm{Pr}^{0.4} \tag{13}$$

$$f = 2.7\mathrm{Re}^{-0.263}(\tan\theta)^{0.143}TR^{-0.291} \tag{14}$$

4.4. Comparison with previous work

The present results are assessed by comparison with the results obtained from similar turbulent flows of the previous studies [5,8,9] for heat transfer and friction factor and plotted in Figs. 10 and 11, respectively. In this investigation, the D-shape turbulators with cone angle, $\theta = 45°$ and tail length ratio, $TR = 1.0$, were chosen to compare with the previous studies under the same Reynolds number and Prandtl number. All of the empirical correlations of the present work and the past studies are described in Table 1. The turbulence airflows of Yakut et al. [5] and Promvonge [9] created by using conical-ring turbulators while those of Eiamsa-ard and Promvonge [8] produced by using V-nozzle turbulators. The correlation data of Yakut et al. [5] and Promvonge [9] given in Fig. 9 were obtained for the conical-ring with $t/d = 2.0$ and $d/D = 0.5$ in a range of $5000 \le \mathrm{Re} \le 17{,}000$. Also, the work of Eiamsa-ard and Promvonge [8] was for the V-nozzles at pitch ratio (l/d) of 2.0. The result from the present work for using D-shape turbulators yields lower Nusselt number than those from the past investigations. The Nusselt numbers of the present study are lower than those of Promvonge [9], Yakut et al. [5], Eiamsa-ard and Promvonge [8] around 2.94, 2.39, and 2.11 times. However, a similar trend is found for both the present work and the previous studies. The considerable reductions in the friction loss of the present study are around 16.9, 5.8, and 21.5 times below those of Yakut et al. [5], Eiamsa-ard and Promvonge [8], and Promvonge [9], respectively.

4.5. Performance evaluation

The comparison of the heat transfer enhancement efficiency (η) of the plain tube with D-shape turbulator inserts for equal pumping power would be more practical and the enhancement efficiency (η) could be derived as below [4,5,8]:

$$(\dot{V}\Delta P)_{\mathrm{p}} = (\dot{V}\Delta P)_{\mathrm{t}} \tag{15}$$

and the relationship between friction and Reynolds number can be expressed as:

$$(f\mathrm{Re}^3)_{\mathrm{p}} = (f\mathrm{Re}^3)_{\mathrm{t}} \tag{16}$$

Therefore, the enhancement efficiency (η) can be written as follows:

$$\eta = \left.\frac{h_{\mathrm{t}}}{h_{\mathrm{p}}}\right|_{\mathrm{pp}} \tag{17}$$

Using Eqs. (12), (14) and (16), the Reynolds number for the plain tube ($\mathrm{Re_p}$) can be written as a function of the Reynolds number for the D-shape turbulator ($\mathrm{Re_t}$):

$$\mathrm{Re}_p = 1.36\ \mathrm{Re}_{\mathrm{t}}^{1.043}(\tan\theta)^{0.055}TR^{-0.11} \tag{18}$$

Employing Eqs. (11), (13), (17) and (18), the enhancement efficiency for D-shape turbulators can be written as:

$$\eta = 4.096\ \mathrm{Re}_{\mathrm{t}}^{-0.158}(\tan\theta)^{0.091}TR^{-0.125} \tag{19}$$

In order to asses thermal performance of using the D-shape element, the enhancement efficiency (η) for a constant pumping power comparison from Eq. (19) is introduced and presented in Figs. 12 and 13 for various θ and TR values, respectively. In the figures, it is worth noting that the enhancement efficiency tends to decrease with increasing Reynolds number for all cases. The highest cone angle ($\theta = 45°$) and the lowest tail length ratio (TR = 1.0) provide the highest enhancement efficiency especially at the lowest Reynolds number value. Enhancement efficiencies are found to be around 0.78–1.0, 0.84–1.07, and 0.88–1.12 for $\theta = 15°$, 30°, and 45° at $TR = 1.0$, depending on Reynolds number. Similar trend is observed for the different tail length ratios with the lowest Reynolds number. The use of $TR = 1.0$, 1.5 and 2.0 at $\theta = 45°$ gives the maximum enhancement efficiency around 1.12, 1.07 and 1.03, respectively.

In the present study, the experimental results of the Nusselt number, friction factor and enhancement efficiency for the tube fitted with D-shape turbulators are correlated as in equations shown in Tables 1–3. Figs. 14 and 15 present the comparison between the present experimental data and the predicted data obtained from the correlations. It is found that the present experimental data agree very well with the correlation data within $\pm7\%$ and $\pm5\%$, for the friction factor and Nusselt number, respectively The empirical correlations are valid for the experimental conditions of $3500 \leq Re \leq 16{,}500$, $Pr \cong 0.7$, $15° \leq \theta \leq 45°$, and $1.0 \leq TR \leq 2.0$. Thus, the correlation could predict the values of Nusselt number and friction factor satisfactorily in the range of parameters studied.

5. Summary

The turbulent forced convective flows through a uniform heat flux tube fitted with D-shape turbulence promoters have been experimentally investigated. The result of the heat transfer demonstrates that the Nusselt number is a function of the cone angles ($\theta = 15°$, 30° and 45°), the tail length ratio (TR = 1.0, 1.5 and 2.0), and Reynolds number (Re). It is found that both the heat transfer rate and friction factor increase with the cone angles (θ), maximum at cone angles, $\theta = 45°$, and with further reduction in the tail length ratio (TR). The increase in the heat transfer rate with increasing the cone angle and decreasing the tail length ratio is due to the higher turbulence intensity imparted to the flow between the turbulators and the heating wall. Since the turbulators are placed directly into the flow core, they cause high friction losses because of high flow blockage. Furthermore, empirical correlations for the Nusselt number and friction factor based on the present experimental study are introduced for practical use with average absolute percentage deviation of $\pm5\%$ and $\pm7\%$, respectively.

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