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Heat transfer enhancement in tubular heat exchanger with double V-ribbed twisted-tapes

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ABSTRACT

An experimental work on heat transfer enhancement in a round tube by insertion of double twisted tapes in common with 30° V-shaped ribs has been conducted. Air as the test fluid flowed through the test tube having a constant wall heat-flux with Reynolds number (Re) from 5300 to 24,000. The combined vortex generators (called "V-ribbed twisted tape") were obtained by incorporating V-shaped ribs into the edges of double co-twisted tapes having a similar twist ratio of 4. The effect of pertinent V-rib parameters such as four relative rib heights, (called "blockage ratio", $B_R=b/D=$ 0.07, 0.09, 0.14 and 0.19) and a relative rib pitch, ($P_R=P/D=1.9$) at an attack angle of rib, $\alpha=30°$ on thermal characteristics was investigated. The experimental results reveal that the heat transfer and pressure drop in terms of the respective Nusselt number and friction factor for the V-ribbed twisted tapes show the increasing trend with the rise of Re and B_R. The V-ribbed twisted tape with $B_R=0.19$ yields the highest heat transfer and friction factor. However, the maximum thermal enhancement factor is about 1.4 for the V-ribbed twisted tape at $B_R=0.09$ but is around 1.09 for the twisted tape with no rib.

1. Introduction

Thermal performance improvement of many heat exchanger systems utilized in engineering and industrial work is needed for energy saving and reduction of operating cost. Heat transfer augmentation methods are often used in the heat exchanger systems in order to enhance the heat transfer rate and increase the thermal performance. In general, a turbulent promoter (called "turbulator") which is one of the passive method is widely employed in heat transfer enhancement in the form of swirl/vortex flow devices such as rib/fin/baffle/winglet/propeller/groove-roughened surfaces [1–8]. Several types of turbulators inserted into the duct flow are to provide an interruption of thermal boundary layer development, to increase the heat transfer surface area and to cause enhancement of heat transfer by increasing turbulence intensity or fast fluid mixing. Therefore, more compact and economic heat exchanger systems with lower operation cost can be obtained. Many

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Nomenclature

A	surface area of test tube, m^2
B_R	relative rib height or blockage ratio, $(=b/D)$
C_p	specific heat of fluid, $J\,kg^{-1}\,K^{-1}$
D	inner diameter of test tube, m
b	rib height, m
f	friction factor
h	average heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
k	thermal conductivity of fluid, $W\,m^{-1}\,K^{-1}$
L	length of test section, m
$\dot{m}$	mass flow rate, $kg\,s^{-1}$
Nu	Nusselt number
P	rib pitch spacing $(=y)$, m
ΔP	pressure drop, Pa
P_R	relative rib pitch or pitch ratio, $(=P/D)$
Pr	Prandtl number
Q	heat transfer rate, W
R	electrical resistance, Ω
Re	Reynolds number
$\tilde{T}$	mean temperature, K
T	temperature, K
t	thickness of tape, m
TT	typical double twisted tape
U	mean air velocity, $m\,s^{-1}$
w	tape width, m
y	pitch length of twisted tape (180° rotation), m

Greek letters

η	thermal enhancement factor
ρ	fluid density, $kg\,m^{-3}$
ν	kinematic viscosity, $m^2\,s^{-1}$

Subscripts

a	air
b	bulk
0	plain tube
conv	convection
i	inlet
o	outlet
pp	pumping power
p	plain tube
v	vortex flow generator
w	wall

attempts have been made to examine the application of various turbulators with different configurations to heat transfer improvement in the heated tube of heat exchanger, for example wire-coils [9,10], twisted-tapes [11,12], dimpled/corrugated/grooved tubes [13,14], combined/compound turbulators [15,16].

In duct/channel heat exchangers, the performance of thermal systems can be enhanced by using rib/baffle/winglet turbulators. Harsha et al. [17] studied the effect of using 90° continuous and 60° V-broken ribs on heat transfer behaviors in a square channel and the V-broken rib performs better than the continuous one was reported. Tamna et al. [18] examined the thermal performance of multiple V-baffles mounted on the two opposite walls of a solar air heater channel. Zhou and Ye [19] experimentally investigated the thermal and flow characteristics in a duct fitted with curved trapezoidal, rectangular, trapezoidal and delta winglets.

In tube inserts, Promvonge [20] conducted measurements using wire coil in conjunction with twisted tape for heat transfer augmentation and reported that the combined wire-coil and twisted-tape yielded the higher heat transfer, friction factor and thermal enhancement factor than the wire-coil/twisted-tape acting alone. Gunes et al. [21] investigated the heat transfer and pressure drop in a tube inserted with coiled wire placed separately from the tube wall with three different pitch ratios ($P/D=1$, $P/D=2$ and $P/D=3$) and two gap distances ($s=1$ mm and $s=2$ mm). They found that the highest overall thermal performance of 50% was achieved for the coiled wire with $P/D=1$ and $s=1$ mm. Promvonge et al. [22] reported the use of the inclined horseshoe-baffles to augment the heat transfer in a circular tube and found that the maximum thermal enhancement factor of about 1.92 was obtained for the horseshoe-baffle at $B_R=0.1$ and $P_R=0.5$. Promvonge et al. [23] also investigated thermal characteristics in a tube fitted with inclined vortex ring (VR) and found that the VR at $B_R=0.1$ and $P_R=0.5$ yielded the best thermal performance. Promvonge et al. [24] again examined the heat transfer behaviors in a square channel inserted with twisted tape together with winglets.

Several modified twisted tapes have been extensively studied by focusing on the rise in heat transfer rather than the reduction of pressure drop. Because of lower pressure loss, the heat transfer enhancement by the modified twisted-tape insert has been extensively investigated. Eiamsa-ard et al. [25] reported the heat transfer behaviors in a double pipe heat exchanger fitted with regularly-spaced twisted-tape elements. Krishna et al. [26] investigated thermal characteristics in a round tube fitted with straight full twist insert with different spacer distances. The measurement of twisted tape consisting wire-nails and plain twisted-tapes inserted in a double pipe heat exchanger on thermal characteristics was carried out by Murugesan et al. [27]. Wongcharee and Eiamsa–ard [28] studied the influence of insertion of twisted tapes with alternate–axes and triangular, rectangular and trapezoidal wings on heat transfer characteristics in a round tube. Eiamsa–ard [29] examined experimentally the application of multiple twisted-tapes for enhancing heat transfer in a channel. Bharadwaj et al. [30] examined experimentally the effect of twisted-tape inserts on thermal behaviors in a grooved tube. Ray and Date [31] proposed a numerical work on laminar and turbulent convection characteristics in a square channel fitted with twisted tapes. Chiu and Jang [32] presented the experimental and numerical analyses on thermal–hydraulic characteristics of air flow inside a circular tube with 5 different tube inserts; longitudinal strip inserts both with/without holes and twisted-tape

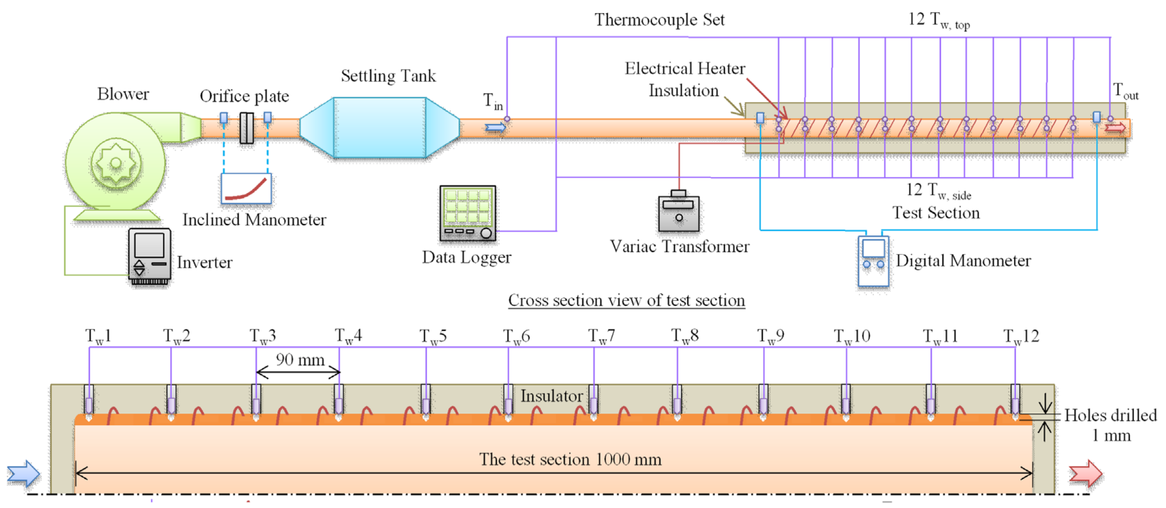


Fig. 1. Schematic diagram of experimental apparatus.

inserts with three different twist angles for inlet velocities ranging from 3 to 18 m/s. Eiamsa-ard et al. [33] numerically studied the convective heat transfer in a circular tube fitted with loose-fit twisted tapes. Liu et al. [34] conducted a numerical investigation on the effect of short-width twisted-tapes inserted in a tube on thermal behaviors in laminar and turbulent flows. Guo et al. [35] examined numerically the thermo-hydraulic performance of laminar flow through a circular tube fitted with center-cleared twisted tape. A simulation of multi-longitudinal vortices in a tube induced by multiple twisted-tapes inserts for the Re from 300 to 1800 was investigated by Zhang et al. [36]. Hong et al. [37] performed a numerical simulation of turbulent flow and heat transfer in converging-diverging tubes and converging-diverging tubes equipped with twin counter-swirling twisted tapes. In their work, the effect of Re, pitch length, rib height, gap distance between twin twisted-tapes and tape number on Nu, f and η was reported.

In the literature review above, the typical or modified twisted-tapes are frequently introduced in round tubes to enhance the degree of turbulence and the fast fluid mixing whereas the rib/fin/baffle/winglets are often offered in ducts/channels to promote the turbulence intensity. For a circular tube, the application of combined double twisted-tapes and V-ribs attached on the tape edge has never been come across in the literature. In general, the use of twisted-tapes alone gives rise to low pressure drop penalty but lower vortex strength while the V-rib which is popular in channels provides higher heat transfer rate apart from lower pressure drop. Considering the merits of both devices, a new enhancement device is offered by incorporating the rib into the twisted-tape edge. The insertion of this compound device is expected to yield stronger turbulence intensity in the near-wall regime by the V-rib and fast fluid mixing by the double twisted tapes, leading to higher heat transfer augmentation in the tube.

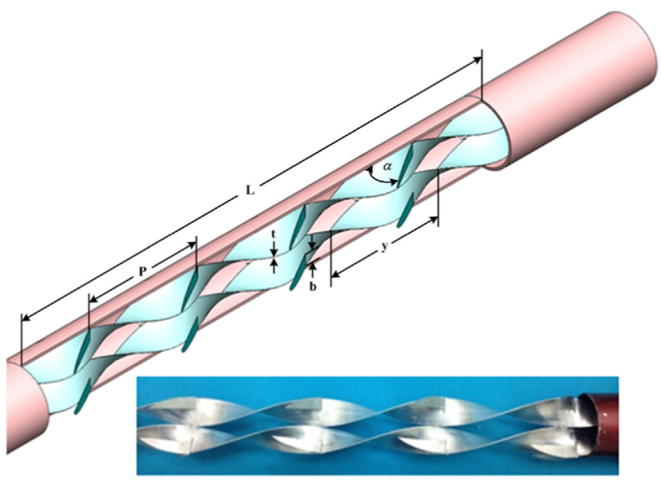


Fig. 2. Test section with double twisted-tapes.

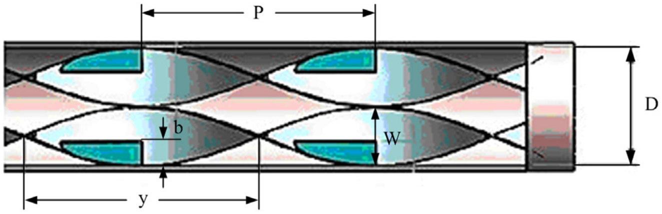


Fig. 3. Dimensions of V-ribbed twisted-tapes.

2. Experimental setup

A detail of the experimental apparatus is displayed schematically in Fig. 1. In the apparatus system, inlet air from a 1.5 kW high pressure blower was directed through an orifice flow-meter to the test section. Using the orifice meter calibrated by hot-wire and vane-type anemometer, the airflow rate was measured. The volumetric airflow rate was adjusted by varying the motor speed of the blower through an inverter. The 3000 mm-long copper tube with inner diameter (D) of 50.8 mm was divided into two sections: calm section and test section (L) as seen in Fig. 1. The test section having a 1000 mm length was heated by continually winding flexible electrical wire on the outer tube wall using a variac transformer to obtain a uniform heat-flux along the entire length of the test section. To minimize heat loss to the surrounding, insulation was wrapped on the most outer tube. The air temperatures at the tube inlet and outlet were measured by resistance temperature detectors thermocouples (RTD-pt100). Two sets of 12 K-type thermocouples were used to measure the surface temperatures (T_w) located equally on the top and the side walls along the test section. All the 24 thermocouples were embedded under the outer surface and centered of the tube walls with axial separation of 90 mm apart. All the temperatures were recorded using a data logger after reaching a steady state condition. To find the pressure drop across the test section, a digital differential manometer was employed in the measurement at an isothermal flow condition.

In Figs. 2 and 3, each of double twisted-tapes made of aluminum sheet was 1000 mm long and 0.8 mm thick (t). The twisted-tape having twist ratio, $y/w=4$, was 24 mm wide (w) with 96 mm twist-length (y). Both twisted-tapes were attached together by gluing before insertion with slightly loose fit. The V-shaped rib made of 0.3 mm aluminum strip with a half V-tip angle of 30° was attached on the edges of twisted-tapes close to the tube wall by fixing the V-tip of the strip on the partially cut edge of the tape before gluing. The four V-rib sizes were 3.6, 4.8, 7.2 and 9.6 mm high (b), in terms of rib-height to tape-width ratios, $b/w=0.038$, 0.05, 0.075 and 0.1, respectively (or equivalent to $B_R=b/D=0.07$, 0.09, 0.14 and 0.19). The ribs were mounted on the edges of tapes with a single rib-pitch to tube-diameter ratios, $P_R=P/D=$ 1.9 and a single rib attack angle (α) of 30°.

3. Data processing

The purpose of the current work is to determine the heat transfer rate in a circular tube fitted with double V-ribbed twisted-tapes. The parameters of interest are Reynolds number (Re) and rib blockage ratio (B_R). The Re is given by

$$Re = UD/\nu \tag{1}$$

The friction factor (f) calculated from pressure drop is written as

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

in which U is mean air velocity in the test tube.

In the experiment, air flowed through the test tube under a uniform heat-flux condition. The steady state of the heat transfer rate is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_a = Q_{conv} \tag{3}$$

where

$$Q_a = \dot{m}C_{p,a}\left(T_o - T_i\right) \tag{4}$$

The convection heat transfer from test section can be written by

$$Q_{conv} = hA\left(\tilde{T}_w - T_b\right) \tag{5}$$

in which

$$T_b = (T_o + T_i)/2 \tag{6}$$

and

$$\tilde{T}_w = \sum T_w/24 \tag{7}$$

where T_w is local wall temperature located equally along the outer surface of the test tube. The average wall temperature, $\tilde{T}_w$ was computed by using 24 points of local wall temperatures. The average heat transfer coefficient (h) and Nusselt number (Nu) are estimated as follows:

$$h = \dot{m}C_{p,\,a}\left(T_o - T_i\right)/A\left(\tilde{T}_w - T_b\right) \tag{8}$$

The heat transfer is calculated from the average Nu which can be obtained by

$$\mathrm{Nu} = \frac{hD}{k} \tag{9}$$

All of thermo-physical properties of air are determined at the overall bulk air temperature (T_b) from Eq. (6).

A fruitful comparison between heat transfer coefficients of vortex flows at equal pumping power can be made, since this is relevant to the operation expense. For constant pumping power,

$$(\dot{v}\Delta P)_p = (\dot{v}\Delta P) \tag{10}$$

and the relationship between f and Re can be expressed as:

$$(f\ \mathrm{Re}^3)_p = (f\ \mathrm{Re}^3) \tag{11}$$

To assess the practical use, thermal performance of the enhanced tube is evaluated relatively to the smooth tube at an identical pumping power in the form of thermal enhancement factor (η) which can be expressed by

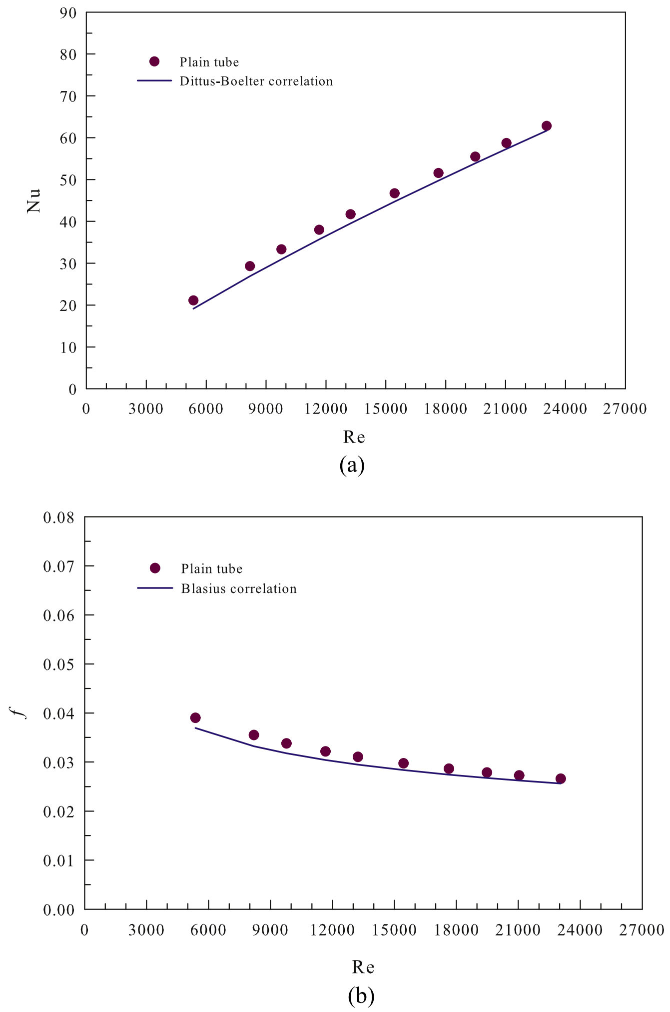
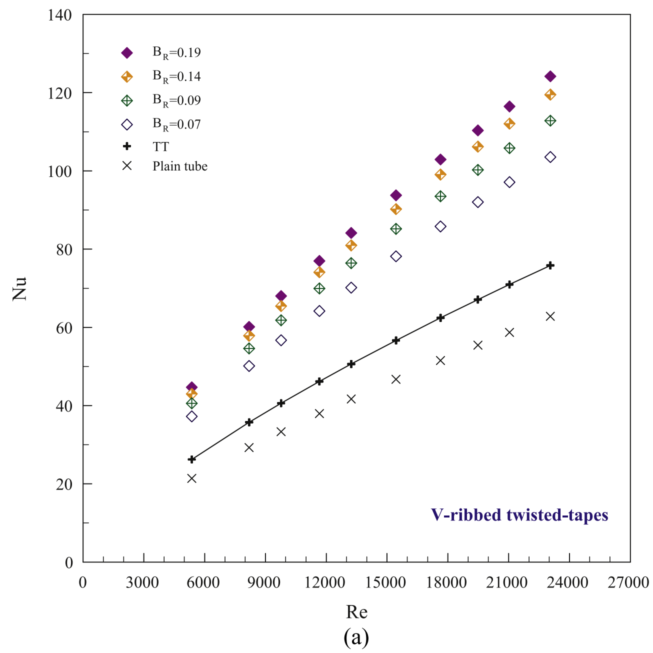


Fig. 4. Verification of (a) Nu and (b) f for plain tube.

$$\eta = \frac{h}{h_p}\bigg|_{pp} = \frac{\mathrm{Nu}}{\mathrm{Nu_p}}\bigg|_{pp} = \left(\frac{\mathrm{Nu}}{\mathrm{Nu_p}}\right)\left(\frac{f}{f_p}\right)^{-1/3} \tag{12}$$

where h_p and h stand for heat transfer coefficients of plain tube and inserted tube, respectively.

The uncertainty calculation was based on Ref. [38]. The maximum uncertainties of non-dimensional parameters were $\pm 5\%$ for Reynolds number, $\pm 7.6\%$ for Nusselt number and $\pm 9.5\%$ for friction factor.



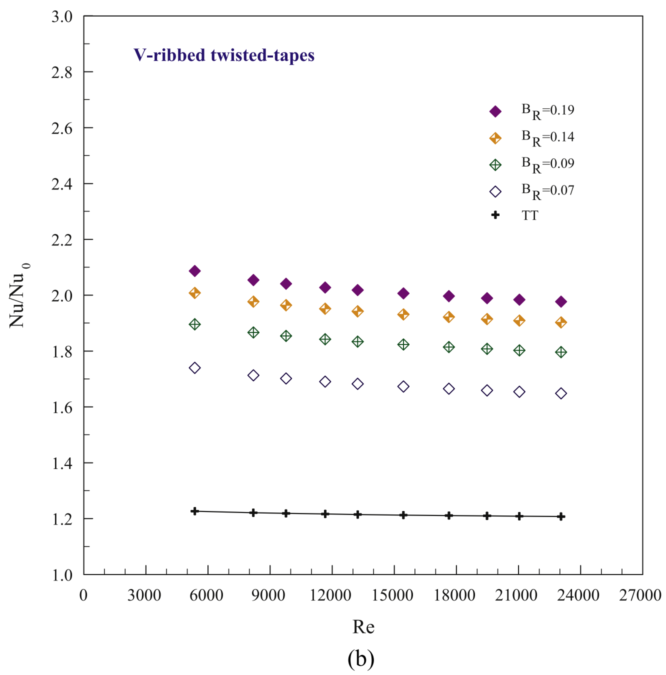


Fig. 5. Variations of (a) Nu and (b) Nu/Nu_0 with Re for V-ribbed twisted tapes.

4. Results and discussion

4.1. VerificatioN OF PLAIN tube

The Nu and f of the plain tube are, respectively, verified first with those from correlations of Dittus–Boelter and Blasius [39], as given in Eqs. (13) and (14), depicted in Fig. 4a and b. In the figure, measured data are in good agreement with correlation's data. The average deviation of the measured and the correlation's Nu and f is about 5% each.

Dittus–Boelter correlation:

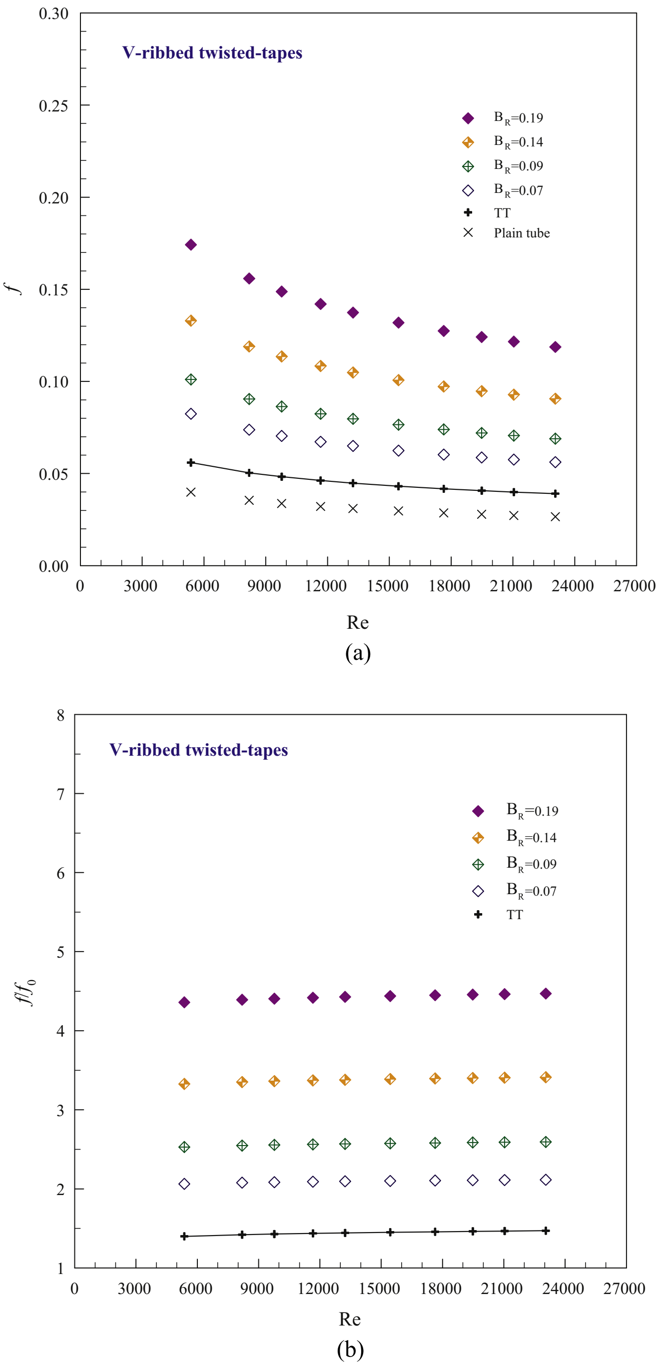


Fig. 6. Variations of (a) f and (b) f/f_0 with Re for V-ribbed twisted tapes.

$$Nu = 0.023Re^{0.8}Pr^{0.4} \text{ for heating} \tag{13}$$

Blasius correlation:

$$f = 0.316 \ Re^{-0.25} \tag{14}$$

4.2. Heat transfer

The variations of Nu and Nusselt number ratio, Nu/Nu_0 with Re for the tube inserted with the V-ribbed twisted-tapes are displayed in Fig. 5a and b, respectively. It is visible that the inserted tube yields considerable heat transfer compared with the plain tube. The Nu shows the uptrend with rising the Re. The Nu of the tube insert is much higher than that of the plain tube. This is due to stronger vortex strength helping to increase turbulence intensity and thinner boundary layer resulting in higher convection. In scrutiny of Fig. 5a, the Nu obtained from the double ribbed twisted tapes is seem to be higher than that from the typical double twisted tape (TT) alone and the plain tube. The double ribbed twisted-tapes with B_R=0.19 yield the highest Nu and the B_R=0.14 provides higher Nu than the B_R=0.09 and 0.07.

The Nu/Nu_0 shows a slightly decreasing trend with the increase in Re as seen in Fig. 5b. For the present data, the Nu/Nu_0 values for the V-ribbed twisted tape are about 1.98–2.09, 1.9–2.01, 1.8–1.89 and 1.65–1.74 times for B_R=0.19, 0.14, 0.09 and 0.07, respectively. The V-ribbed co-twisted tape yields the Nu/Nu_0 around 27–41% higher than the twisted tape (TT) alone. This means that the ribbed twisted tape is advantageous over the twisted tape alone.

4.3. Friction factor

The influence of using the V-ribbed twisted tape on f and f/f_0 against Re is displayed in Fig. 6a and b, respectively. In the figure, it can be observed that the application of the combined devices leads to a substantial increase in f above the plain tube and the f shows the downtrend with the increment of Re. The higher friction loss mainly comes from the increased surface area and higher swirl intensity by the inserts. The f of the V-ribbed twisted tape increases around 52–77% above that of the plain tube. As expected, the f of the V-ribbed twisted tape with larger B_R is higher than that with smaller B_R and is about 31–68% higher than that of the twisted tape (TT) alone, depending on B_R.

In Fig. 6b, it is visible that the f/f_0 for the inserted tube tends to increase considerably with rising B_R and Re values. The V-ribbed twisted tape with B_R=0.19 provides the f/f_0 higher than the one with lower B_R. This is because the V-ribbed twisted tape with B_R=0.19 gives rise to higher flow resistance, larger surface area and stronger vortex flow, leading to a substantial increase in pressure drop. The f/f_0 for the V-ribbed twisted tapes with B_R=0.19, 0.14, 0.09 and 0.07 is about 4.36–4.47, 3.33–3.41, 2.53–2.59 and 2.06–2.12 times, respectively, while that for the double twisted tapes alone is around 1.4–1.47 times.

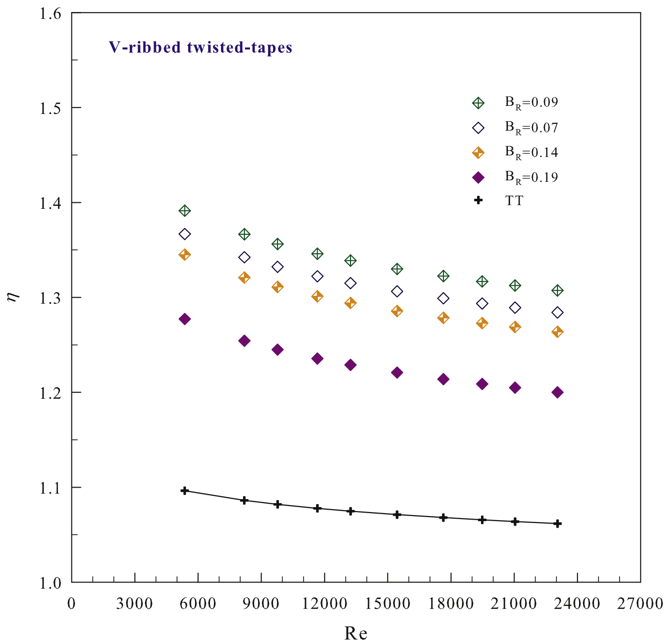


Fig. 7. Variation of η with Re for V-ribbed twisted tapes.

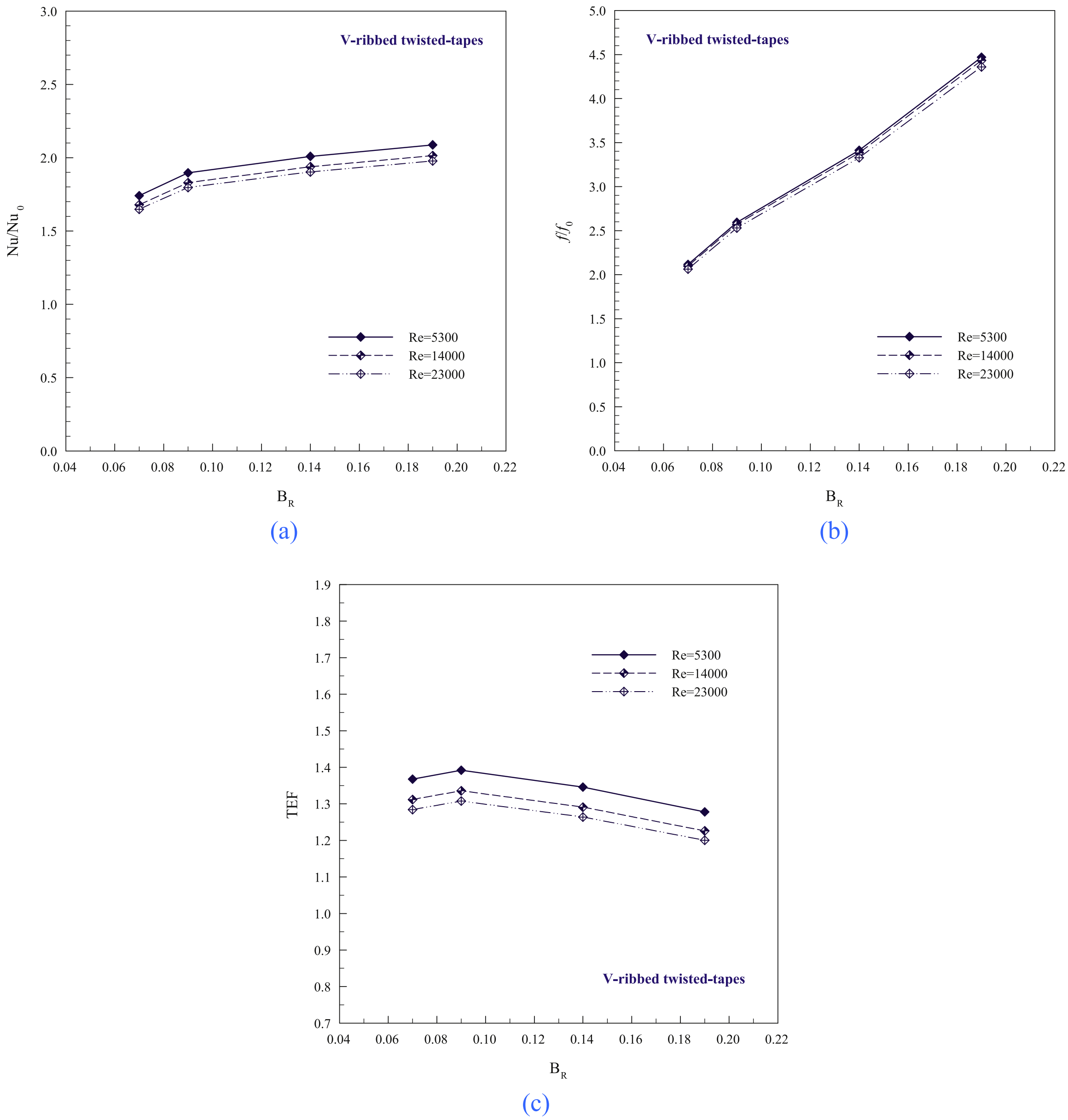


Fig. 8. Effect of B_R on (a) Nu/Nu_0, (b) f/f_0 and (c) η at similar pumping power.

4.4. Thermal performance

Fig. 7 depicts the variation of the thermal enhancement factor (η) with Re for the V-ribbed twisted tape with various B_R values. In the figure, the measured Nu and f values of the tube insert and the plain tube are compared at the same pumping power. It is seen that η tends to decrease with increasing Re for all the cases and is about 1.1–1.4 depending on Re and B_R values. The maximum η of about 1.4 is achieved for the V-ribbed twisted tape with B_R=0.09 and is higher than the other V-ribbed twisted tapes around 1.7–8.2% while higher than the twisted tape alone at some 18.8–21.2%. A close examination reveal that the V-ribbed twisted tape with B_R=0.09 performs better than the twisted tape alone around 20%. Therefore, if the choice of an insert device is the employ of double twisted tapes, V-ribs should be incorporated to obtain higher thermal performance.

4.5. Effect oF B_R

The thermal performance factor (η), indicating the practical benefit of the combined devices (V-ribbed twisted tape) is obtained from Eq. (12), in which heat transfer rate and friction factor in the tube with and without V-ribbed twisted tape,

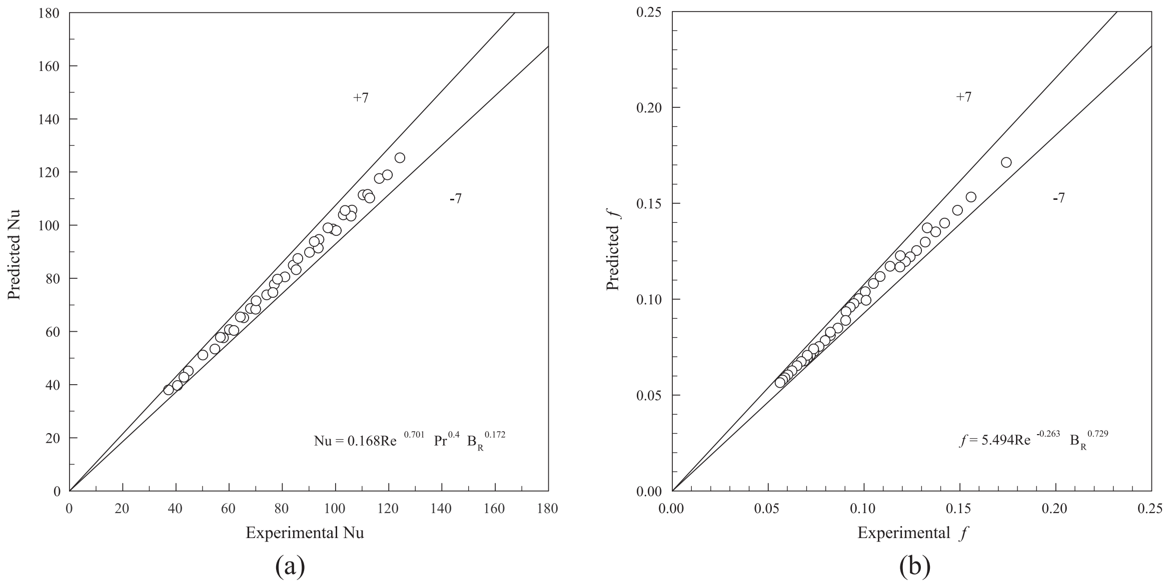


Fig. 9. Validation test of (a) Nu and (b) f correlations with measured data.

are simultaneously determined at the same pumping power. Fig. 8a, b and c portray an effect of rib blockage ratio, B_R on Nu/Nu_0, f/f_0 and η, respectively. It is visible in the figure that the Nu/Nu_0 increases with the increment of B_R, especially for $B_R=0.19$, but with the reduction of Re. For V-ribbed twisted tapes, the average Nu/Nu_0 and f/f_0 values for the $B_R=0.19$ are, respectively, about 3.8% and 23.7%; 9.5% and 55%; 16.6% and 52.7% above those for the $B_R=0.14$, 0.09 and 0.07. The V-ribbed twisted tapes with $B_R=0.09$ provides the maximum η of about 1.4 at the lowest Re.

In addition, the empirical correlations for Nu and f developed by relating the Re and B_R together are compared with experimental data within $\pm 7\%$ deviation each, as can be seen in Fig. 9a and b, respectively. These correlations are valid for the double V-ribbed co-twisted tapes with twist ratio of 4, Re=5300–24,000, $P_R=1.9$ and $B_R=0.07$–0.19.

Correlations for double V-ribbed twisted tapes:

$$\mathrm{Nu} = 0.168\,\mathrm{Re}^{0.701}\,\mathrm{Pr}^{0.4}\mathrm{B_R}^{0.172} \quad (15)$$

$$f = 5.494\,\mathrm{Re}^{-0.263}\,\mathrm{B_R}^{0.729} \quad (16)$$

5. Conclusions

An experimental investigation on thermal characteristics in a constant heat-fluxed round tube fitted with double V-ribbed twisted-tape for turbulent flow, Re from 5300 to 24,000 has been carried out. The highest heat transfer and pressure loss from the V-ribbed twisted tape inserts is found at the largest B_R. The Nu is in the range of 1.56–2.3 times while the f is 2.06–4.94 times above the plain tube for the V-ribbed twisted-tape. The inserted tube with V-ribbed twisted tape at $B_R=0.19$ gives the highest Nu and f. However, the V-ribbed twisted tape at $B_R=0.09$ yields the highest η around 1.4. Therefore, the use of the V-ribbed twisted tape is a promising enhancement device in the heat transfer improvement in a heating/cooling tube system.

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