Experimental Evaluation of Performance Intensification of Double-Pipe Heat Exchangers with Rotary Elliptical Inserts

M. E. Nakhchi[a], M. Hatami[b,1], M. Rahmati[a]

[a] Faculty of Engineering and Environment, Northumbria University, Newcastle upon Tyne, NE1 8ST, UK

[b] Department of Mechanical Engineering, Ferdowsi University of Mashhad, P.O.B. 91775-1111 Mashhad, Iran

Abstract

In this experimental work, heat transfer and thermal performance factor of double-pipe heat exchanger pipes with Rotated inclined elliptical (RIE) inserts are examined. The rotation angle of the elliptical turbulators (β), slant angle (α), and the perforation diameters (d_p) were in the range of $0^\circ \le \beta \le 90^\circ$, $15^\circ \le \alpha \le 25^\circ$, and $0.5mm \le d_p \le 1.5mm$, respectively. The rotated inserts with perforations can substantially intensify the flow unsteadiness and disturb the thermal boundary layer to improve the thermal performance without a perceptible effect on the pressure drop. The experimental analysis showed that the heat transfer can be augmented up to 30.7% by utilizing RIE turbulators in comparison with the non-rotated elliptic (NRIE) turbulators. The recirculating flows across the perforations of the elliptical inserts rise the fluid mixing amongst the tube wall and the central region. The highest thermal performance factor of 2.23 is achieved for RIE turbulators with $\beta = 90^\circ$, d_p/b=0.250 and $\alpha = 25^\circ$. The thermal performance factor that is obtained in this study is noticeably higher than that of the previous studies, without increasing the manufacturing costs. The heat transfer coefficient is enhanced by 59.95% by utilizing NRIE vortex generators with d_p=1.5 mm in comparison with the typical non-rotated louvered strips without perforations.

[1] Corresponding author, M. Hatami, Tel/Fax:+98-935-860-2679, E-mail: m.hatami@xjtu.edu.cn

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Nomenclature		**Abbreviations**	
C_p	Specific heat capacity at constant pressure (J/(kg·K))	DPHEX	Double-Pipe Heat Exchangers
d	Diameter of tube (mm)	DPT	differential pressure transmitter
d_p	Perforation diameter (mm)	hBvF	horizontal base vertical fin
f	Friction factor	LMTD	Logarithmic mean temperature difference
h	Convection coefficient (W/(m^{2}K))	MAD	Mean Average Deviation
k	Thermal conductivity (W/(m·K))	NRIE	Non-Rotated Inclined Elliptic
L	Length of Tube (mm)	PREI	Perforated Rotary Elliptic Inserts
$\dot{m}$	Mass flow rate (kg/s)	RIE	Rotated Inclined Elliptic
N	Perforation Number	RMS	Root Mean Square
Nu	Nusselt number	vBhF	vertical base horizontal fin
P	Pitch Distance (mm)	**Subscripts**	
Pr	Prandtl number	c	cold
Q	Heat rate(W)	h	hot
Re	Reynolds number	i	Inner
t	Pipe thichness (mm)	ins	Insulation
T	Temperature (K)	o	Outer
U	Total heat transfer coefficient ($W/(m^2K)$)	rad	Radiation
V	Velocity (m/s)	s	Specific
Greek Symbols			
α	Slant angle (°)		
β	Rotation angle (°)		
μ	dynamic viscosity		
η	thermal performance factor		
ΔP	Pressure difference		

1. Introduction

Louvered strip and perforated inserts improve the heat transfer by disturbing the flow and generating a turbulent flow regime [1-5]. Yaningsih et al. [1] investigated the louvered strip insert slant angles (15, 20 and 25°) on the heat transfer intensification by single-phase turbulent flow. They found that the thermal performance factor using these inserts improved at

the range of 1.00-1.12. Also, Eiamsa-ard et al. [2] used the same louvered strip inserts in backward and forward inclined angles. They showed that the average Nusselt number (Nu number) and friction loss for the forward inclined case were increased by 284% and 413%, respectively. On the other hand, he showed that for the backward inclined case, the averaged Nu number and friction loss were improved by 263% and 233% over the plain tube, respectively. Indeed, louvered strip inserts are not the only inserted devices that enhance heat transfer. Many other devices such as helical surface disc turbulators [6], twisted tapes [7-10], V-cut twisted tapes [11], baffled tapes [12], V-shaped winglet inserts [13], conical rings [14], nanofluids [15], grooved surfaces [16], transversely twisted-turbulators [17], dimpled tube with twisted tape [18], or combination of these methods [19] have been used by researchers to intensify the thermal efficiency of heat exchangers and solar heaters.

Recently, many studies are performed on double-pipe heat exchangers (DPHEX) to intensify their efficiency for various applications [20]. In the numerical study on double-pipe heat exchanger, Xiong et al. [21] investigated the effect of conical and fusiform turbulators on heat transfer. The numerical results for various cases showed that maximum convection coefficient obtained for inner circular pipe and the optimum configuration were for a case with 12 mm fusiform turbulators. In another numerical study on DPHEX, Huu-Quan et al. [22] investigated the effect of the flat inner pipe. They found that the proposed vortex generators are very sensitive to Reynolds number. The results revealed that when the Reynolds number is smaller than 7000, by using small aspect ratio of flat inner pipes, the total heat transfer, thermal efficiency, and performance factor can be improved about 2.9%, 2.7%, and 16.8%, respectively. Córcoles et al. [23], in a numerical simulation, modelled eight inner corrugated tubes for the DPHEX and found a 23% increase in the heat exchanger efficiency. Also, the performance of a novel corrugated tube named sinusoidal hairpin DPHEX filled by MgO-Ag/water hybrid nanofluid was investigated by Li et al. [24]. They examined the effects of the

Reynolds number, nanoparticles concentration and amplitude of sinusoidal tube on the heat transfer enhancement of heat exchangers. Filling the DPHEX with porous media is another way to intensify efficiency, a method used by Chen et al. [25]. They showed that decreasing porosity of porous media and increasing pore density of foam improve the heat exchanger effectiveness significantly. In an experimental study on the DPHEX, Singh and Sarkar [26] used the conical wire coil to enhance the heat transfer from hybrid nanofluids. They investigated the effect of diverging, and converging wire coil turbulators. The analysis showed maximum Nusselt number improvements of about 171%, 152% and 139%, respectively. The comparative study of Kumar et al. [27] revealed that the thermal performance parameter inside a heat exchanger tube equipped by twisted tapes is the highest among various proposed turbulators. Tabatabaeikia et al. [28] evaluated the heat transfer augmentation of heat exchangers equipped by various vortex generators. They compared the heat transfer improvement by using louvered strips, twisted tapes, helical screw-tape inserts, wire coil inserts and other types of turbulators. They concluded that the louvered strips provide better heat transfer in backward flow direction compared with forward one.

Also, Sivalakshmi et al. [29] used helical fins to intensify the DPHEX performance experimentally and revealed 38% and 35% improvements for heat transfer and thermal efficiency, respectively. Thejaraju et al. [30] introduced a novel para winglet tape vortex generator in their experimental study and found a maximum 407% increase in Nusselt number for air-air heat exchanger.

Perforated-type vortex generators not only increase the performance in turbulent flows but also is helpful on laminar flow regimes, as shown by Karlapalem et al. [31], who used perforated fins in laminar natural convection and varied the perforations from 1.1 mm to 6.6 mm at two orientation of fins named vertical base horizontal fin (vBhF) and horizontal base vertical fin (hBvF). They deduced that the maximum total heat flux resulted from the perforated

fins was 1.6 times higher in hBvF orientation. Suri et al. [32] experimentally investigated the heat transfer intensification inside a heat exchanger pipe by using multiple square-perforated twisted tapes. They proposed some correlations for the heat transfer, pressure drop, and thermal performance factor as functions of the geometrical and physical parameters. They observed significant enhancement in thermal efficiency inside the modified heat exchanger compared to the simple plain tube. Nakhchi and Esfahani [33] used double-perforated louvered strip insert to improve the heat transfer. They found that thermal performance of 1.99 could be obtained at the angle of 15°. Furthermore, they improved the heat transfer by using CuO nanoparticle additives up to 15.6%. Liu et al. [34] confirmed experimentally that the perforated fin-tube heat exchanger would increase the heat transfer rate and the heat transfer coefficient by 38.9% and 31.8%, respectively.

Based on the above literature review, it can be noticed that rotating turbulators can substantially increase the thermal performance of heat exchanger pipes. But they have received less attention compared to other turbulators, as can be seen in Zhang et al. [35]. For instance, Goh et al. [36] and Huchet et al. [37] used self-rotating turbulator and rotary kiln wall to enhance the heat transfer, respectively. However, there are very few experimental studies in this field, and no previous numerical or experimental study considered the advantages of perforated turbulators and the rotation effects at the same time. To fill this lack of limited research on rotating inserts, in this study Perforated Rotary Elliptic Inserts (PREI) are employed for the first time to improve the heat transfer in a double pipe heat exchanger experimentally. Furthermore, the effects of inclination angle, rotation angle and perforation diameters on the thermal performance are investigated.

2. Experimental Setup

The picture and the schematic view of the experimental apparatus are shown in Fig. 1. The double-pipe heat exchanger equipment includes two hot-water and cold-water cycles, measuring units and the test section. The hot-flow cycle consists of a hot-water stainless steel tank (35.0 Lit), RS PRO 374-727 heating element with a power rating of 2000 W with 220V supply power, a Wilo Star-RS 25/1-8 circulation pump (151 W), a crystal-conical rotameter with the capacity of 8 Lit/min with the reading accuracy of ± 0.1 Lit/min, 0.5in brass Y-strainer valve with the maximum operating flow characteristics of 6 bar at 110 ºC and 3.8 m connecting pipes. An accurate BEM402-K1220 thermostat controlled the power of the heating element with an uncertainty of $\pm$ 0.5% (Full-scale) to control the water inflow temperature at the desirable point (85.0 ℃ in this study). The cooling loop contains another similar conical rotameter with a capacity of 8 Lit/min and an uncertainly of ± 0.1 Lit/min. Two control valves were utilized to control the inflow rate of the cooling/hot water in the equipment. The inflow temperature of the cooling water (urban water in the present study) was constant at 17.5 ℃ during the experimental measurements. Four control valves were implemented at the inlet and outlet of the hot/water and cold/water loops to adjust the structure of the double-pipe heat exchanger from parallel-flow to counter-flow system. In the present study, all of the experiments and data analysis were performed for the counter-flow heat exchanger adjustment.

TP-101 thermometers with the operational range of -60 ºC to 310 ºC with the measurement precision of ± 0.1 ℃ were used to determine the temperature of the water flow in cold& hot flow loops. A digital differential pressure transmitter (DPT) with the operating range of 0 to 4000 Pa was employed to evaluate the pressure drop throughout the test section with uncertainly of ±1 Pa. The voltage and ampere of the electric current to the heating element were measured by an Omega multimeter with the working range of 0-220 V [±0.2 V] and 1-20 A [±0.002 A], respectively. The dimensions of the geometrical parameters were measured by a digital vernier with an accuracy of ±0.1 mm. The tests were done in steady-state

circumstances. Glass wool is used to insulate the exterior annulus with 60mm thickness to reduce the heat loss from the pipes.

Fig. 2 shows the details of the DPHEX fitted by the rotated orientated elliptical vortex generators within the heat exchanger pipe. The hot water moves across the central pipe whilst the exterior annular duct is chilled with cold water. The hot water internal tube is fabricated of copper with the thickness (t_i), inner diameter (d_i) and length (L_i) of 7.5×10^{-1} mm, 14.3 mm and 855.0 mm, respectively. The external annulus is made of stainless steel with the thickness (t_o), diameter (d_o) and length (L_o) of 1 mm, 23.38 mm, and 840 mm, respectively. The details of the rotation angle (β) on the elliptic turbulators are presented in Fig. 3 for a specific rotation angle of 90°. The rotated inclined elliptic turbulators were constructed by stainless steel and were installed on a linking shaft with a diameter of $\delta = 2$ mm across the central pipe. The specifics of the geometric parameters of the RIE turbulators are shown in Figs. 3-4. To validate the experimental data with the data available in literature, the geometric parameters were selected identical to the equipment of Ref. [1] for typical louvered strips without rotation and holes. The distance (P) among the RIE inserts is selected to be 40 mm. The number of perforations (N) were selected between 0 to 5. The holes diameter varied between (0mm $< d_p <$ 1.5mm). The slant angle (α) among the RIE vortex generators and the connecting rod is selected between 15° to 25°. The other geometrical parameters, including the thickness of the RIE enhancers (t= 1 mm) and the inner and outer diameter of the elliptical strips (b= 6 mm, c=10 mm) kept fixed. Fig. 3 compares the difference between the geometries of non-rotated and rotated inclined elliptic turbulators with the rotation angle of β=90°. The details of the geometric parameters are presented in Table 1.

As described in the above sections, the main goal of this study is the investigation of rotated inclined elliptic (RIE) turbulators on the thermal performance of double pipe heat exchanger.

In this study, it is assumed that three parameters can be varied, namely inclined angle of elliptic (α), the diameter of perforations (d_p) and rotation angle (β) as shown in Figs. 4-5. Fig. 5 demonstrates the four designed geometries for RIE inserts with different parameters. In this depicted geometries, inclined angle (α) is at 25° constant, while the diameter of perforations (d_p) and rotation angle (β) are varied between 0.5-1.5 mm and 45-90°, respectively.

Table 1 Geometrical parameters of the test cases.

Case	Slant angle (α)	Rotation angle (β)	Hole diameter (d_p, mm)	Pitch (P, mm)	Number of Perforations (N)	Type
Validation [1]	15, 20, 25	0	0	40	0	Typical louvered strip
1-3	15	0	0.5-1.5	40	5	NRIE VG
4-6	25	0	0.5-1.5	40	5	NRIE VG
7	15-25	20	1.5	40	5	RIE VG
8-11	15-25	45	0.5-1.5	40	5	RIE VG
12-15	15-25	90	0.5-1.5	40	5	RIE VG

3. Data Reduction and uncertainly analysis

In this section, the equations for the experimental analysis are provided. To evaluate the heat transfer rate inside the heat exchanger pipe fitted by the RIE strip inserts, firstly, the heat transfer from the cold water inside the annulus (Q_c), and the heat gain by the hot water inside the central pipe (Q_h) can be evaluated as [1]:

$$Q_c = \dot{m}_c C_{p,w} \left(T_{c,out} - T_{c,in} \right) \tag{1}$$

$$Q_h = \dot{m}_h C_{p,w} \left(T_{h,in} - T_{h,out} \right) \tag{2}$$

where $\dot{m}_c$ and $\dot{m}_h$ are the mass flow rate of the cold and hot flow loops, $C_{p,w}$ is the specific heat, T_c and T_h are the temperatures of the cold and water, at the inlet and outlet of the pipes,

respectively. It was deduced from the measurements that the total heat loss $\left(Q_{loss}\right)$ via the insulations $\left(Q_{ins}\right)$, radiation $\left(Q_{rad}\right)$ and natural convection were below 2.1%. The mean value of the heat flow rate is chosen to evaluate the heat transfer coefficient (U) of the inner tube by [2]:

$$\overline{Q} = \frac{Q_h + Q_c}{2} \tag{3}$$

$$\overline{Q} = UA_i \Delta T_{LMTD} \tag{4}$$

Where A_i is the surface of the interior pipe and is evaluated by:

$$A_i = \pi d_i L \tag{5}$$

The heat balance between the heat exchanger tube and the environment can be expressed as:

$$Q = \frac{T_{\infty,i} - T_{\infty,o}}{R_{tot}} = UA_i\left(T_{\infty,i} - T_{\infty,o}\right) \tag{6-a}$$

When the heat transfers from the hot fluid to the cold fluid, it is necessary to pass the thermal resistance of the turbulent flow linked to (i) convection in the inner pipe, (ii) heat conduction from the pipe walls, and (iii) convection from the exterior pipe. The thermal resistance is written as:

$$\frac{1}{U\pi d_i L} = \frac{1}{h_i\left(\pi d_i L\right)} + \frac{\ln\left(d_o / d_i\right)}{2\pi k L} + \frac{1}{h_o\left(\pi d_o L\right)} \tag{6-b}$$

The convective heat transfer coefficient $\left(h_i\right)$, which is one of the main parameters in heat transfer enhancement studies, can be evaluated by [1]:

$$h_i = 1 \Bigg/ \left[\frac{1}{U} - \frac{d_i \ln\left(d_o / d_i\right)}{2k} - \frac{d_i}{d_o h_o}\right] \tag{6-c}$$

To calculate h_i, it is essential to get the amount of the heat transfer coefficient from the annular tube (h_o). This coefficient is evaluated by utilizing the Dittus–Boelter correlation [38]:

$$Nu_o = \frac{h_o D_h}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{7}$$

Where $D_h = \dfrac{4\left(\dfrac{\pi d_o^{\,2}}{4} - \dfrac{\pi d_i^{\,2}}{4}\right)}{\pi d_i + \pi d_o} = d_o - d_i$ is the hydraulic diameter, Pr is the Prandtl number of the water at the operating temperature, $\mathrm{Re} = \rho V_{in} d / \nu$ is the Reynolds number, and μ is the dynamic viscosity of the operating fluid. The Reynolds number is changed between 6,000 to 18,000 in the experiments (turbulent flow regime). The average Nusselt number of the inner pipe of the DPHEX fitted with the rotated inclined elliptic (RIE) vortex generators can be calculated by [39]:

$$Nu_i = \frac{h_i d_i}{k} \tag{8}$$

The friction factor, (f) through the inner hot-water pipe is computed by:

$$f = \frac{\Delta P}{0.5 \rho V_{in}^{\,2}} \frac{d_i}{L_i} \tag{9}$$

V_{in} is the velocity in the inlet of interior pipe. The thermal performance factor (η) can be expressed as the ratio of the heat transfer improvement to the friction loss ratio with respect to the plain pipe (Nu_s , f_s) at specific pumping power. The thermal performance factor at constant pumping power is expressed [3]:

$$\eta = \left.\frac{h}{h_s}\right|_{pp} = \left.\frac{Nu}{Nu_s}\right|_{pp} = \frac{Nu}{Nu_s}\left(\frac{f}{f_s}\right)^{-1/3} \tag{10}$$

Before performing the experimental analysis, it is essential to perform an uncertainty analysis to evaluate the accuracy of the experimental facilities and their impact on the outputs of the experiments, such as heat transfer, friction loss and thermal performance factors. The experiments were done after calibrating the measurement instruments and reaching the steady-state conditions. Based on the uncertainty analysis methodology (RMS) of Moffat [40], the

uncertainty of the Reynolds number, Nusselt number, and the friction loss are ±3.5%, ±4.4% , ±4.7%, respectively.

4. Results and Discussion

4.1. Validation

To have a validation study, Fig. 6a and Fig. 6b are depicted for Nusselt number and friction factor, respectively, when the turbulators have not any perforations or rotations (Np=0, β=0). The results of these typical louvered strips are calculated for two angles of inclination (α= 15 and 25°). Outcomes reveal that obtained Nu and f values in the present study are in excellent agreement with those of Yaningsih et al. [1], especially at the Reynolds below 13000.

4.2. Repeatability

The repeatability test is performed at the beginning of the experiments at three different days to verify the accuracy of the measurement tools inside the DPHEX fitted by NRIE vortex generators. Fig. 7 shows the Repeatability results for NRIE inserts with inclination angle of $\alpha = 25^{\circ}$ and $d^{*} = 0.25$. The maximum discrepancy among the experimental data of Nu and *f* are 3.42% and 2.98%, respectively. Therefore, the measuring tools are precisely calibrated and are stable, and can be used for further experimental analysis.

4.3. Parametric study

At the first step of the parametric study, the effect of inclined angle (α=0, 15, 20, and 25°) on the Nu and f numbers for non-rotated inclined elliptic inserts are depicted in Fig. 8 and Fig. 9, respectively. As seen, the maximum value of the Nusselt number has occurred when α=25° around 275.1, which means improved the Nusselt number 223.5% compared to a plain heat exchanger. It is noted that the heat transfer is improved by increasing the slant angle from 15 to 25 degrees. The main physical reason for heat transfer enhancement

is more flow disturbance near the pipe walls with NRIE vortex generators with greater inclination angle, which cause thermal boundary distribution close to the inner tube surface. The flow disturbance increases the fluid mixing among the center of the pipe and the heated wall and increases the heat transfer.

On the other hand, Fig. 9 shows that increasing the inclined angle also increases the friction factor where the maximum friction factor (which is in lower Reynolds number of 6000) increased about 301.2% for α=25° compared to a plain heat exchanger. It is observed that increasing the inclination angle rises the pressure drop and friction loss inside the double-pipe heat exchanger with NRIE turbulators. Physically speaking, increasing the inclination angle have significant impact on flow perturbation and recirculation flows near the walls of the heat exchanger tube. The distorted shear stress because of the NRIE turbulators with higher slant angle leads to the secondary flow production. The fluid is moved toward the pipe surface, which disrupts the axial velocity of the fluid alongside the pipe. The additional flow disturbance increases the pressure drop inside the tube.

As the second step of the parametric study, the effect of rotation angle (β) on the Nu number and friction factor is examined through Fig. 10 and 11, respectively. In these figures, the rotation angles are varied from 0 to 90° when $\alpha = 25^{\circ}$ and $d^{*} = d / b = 0.25$. It can be concluded that increasing the rotation angle enhanced the Nusselt number due to making more turbulent stream inside the heat exchanger and consequently more heat transfer. By expanding the β from 0 to 90°, the Nu has been improved by 30.7%. Fig. 10 confirms that increasing the rotation angle increases the friction factor, but the increase is not very significant.

The effects of the perforation diameter of the rotated inclined elliptic turbulators on the heat transfer and friction factor coefficient of turbulent flow inside heat exchanger pipe are

presented in Table 2. The results are shown for two different rotation angles of $\beta = 45^\circ$ and 90°, respectively and compared with the $\beta = 0^\circ$ (NRIE). It is found that increasing the hole diameter improves the heat transfer significantly. The main reason for the heat transfer intensification is the thermal boundary-layer disturbance near the tube wall in the presence of the perforations. The recirculating flows and vortex generation of turbulent flow amongst the pipe walls and the central area sweep heat from the heated walls to the centre region. Therefore, heat transfer is increased. The results illustrate that the average Nu number is improved by 59.95% by utilizing NRIE vortex generators with d^*=0.250 in comparison with the conventional non-rotated louvered strip inserts with no holes (d^*=0). The maximum Nusselt number of 342.62 is obtained by using RIE turbulators with d^*=0.25, $\alpha = 25^\circ$, and $\beta = 90^\circ$. However, the friction factor is also increased by 14.98%, with raising the d^* from 0 to 0.250 at $\beta = 90^\circ$. As the heat transfer enhancement due to the perforations is much higher than the pressure drop penalty, therefore the proposed RIE turbulators are helpful in improving the thermal performance factor of the double-pipe heat exchangers.

Table 2 The effects of perforated diameter (d_p) on average Nu and f at Re=18,000 and $\alpha = 25^\circ$.

$d^* = d_p / b$		Nu		f		
	$\beta = 0^\circ$ (NRIE)	$\beta = 45^\circ$	$\beta = 90^\circ$	$\beta = 0^\circ$ (NRIE)	$\beta = 45^\circ$	$\beta = 90^\circ$
0 (non-perforated)	180.14	201.94	221.70	0.0726	0.0774	0.0814
0.083	213.05	238.81	262.25	0.0764	0.0813	0.0856
0.167	249.36	279.48	306.87	0.0801	0.0854	0.0897
0.250	288.14	325.85	342.62	0.0835	0.0885	0.0936

The variations of Nu/Nu_p and f/f_p with Re for various NRIE turbulators are presented in Fig. 12. The dimensionless heat transfer and friction factor coefficient compared to the

plain tube help to better understand the impact of the NRIE inserts on the flow characteristics and performance intensification inside DPHEX. As shown in Fig. 12-a, the slant angle (α) has significant impact on heat transfer enhancement. The heat transfer coefficient is 2.73 times higher than a plain tube for the NRIE turbulators with $\alpha = 25^{\circ}$ and $d^{*} = 0.25$. It is observed that using perforations $(d^{*} = 0.25)$ noticeably improve the heat transfer rate compared to conventional RIE inserts $(d^{*} = 0)$. The main physical reason is additional flow disturbance inside the perforations that disrupts the thermal boundary-layer near the tube walls. Similar trend is also observed for the dimensionless friction factor. The friction factor reached 4.27 times of a plain tube for NRIE vortex generators with $\alpha = 25^{\circ}$ and $d^{*} = 0.25$ at Re=6,000. However, the impact of the geometry on friction factor is reduced by raising the Re number. Physically speaking, increasing the fluid velocity intensifies the flow disturbance and perturbations inside the heat exchanger tube. Therefore, the effects of the NRIE vortex generators on the pressure drop increment become less noticeable at higher Re numbers.

Finally, and as the third step of the parametric study, the simultaneous effect of both inclined angle (α) and rotation angle (β) on the thermal performance of the heat exchanger is depicted in Fig. 13. As seen, the maximum thermal performance of HE, which is around 2.2, is occurred when the α=25° and β=90° at the Reynolds of 16,000. Furthermore, it can be observed that increasing each of these parameters improves the thermal performance as well as Reynolds number increment.

To make the provided experimental analysis more general and applicable in heat exchanger industries, two empirical equations of the Nu number and the friction factor for the DPHEX enhanced by NRIE turbulators are presented in Eqs. (11-12). The correlations are generated by the statistical analysis of the data. According to the mean average deviation parameter (

$MAD = \sum_{i=1} \left| \xi_i - \overline{\xi} \right| / n$), the median and overall deviations for Eq. 11 are 1.47% and 3.48%, respectively. The Nu and friction factor correlations are functions of the perforations $\left(0 \le d_p / b = d^* \le 0.25\right)$, rotation angle $\left(0^\circ \le \beta \le 90^\circ\right)$, inclination angle $\left(15^\circ \le \alpha \le 25^\circ\right)$, and Re number $\left(6{,}000 \le \mathrm{Re} \le 18{,}000\right)$, respectively. The governing correlations can be expressed as:

$$Nu = 0.0607\,\mathrm{Re}^{0.896}\,Pr^{0.3}\left(\frac{\alpha}{90}\right)^{0.898}\left(1+\frac{\beta}{180}\right)^{0.512}\left(1+d^*\right)^{2.105} \tag{11}$$

$$f = 82.98\,\mathrm{Re}^{-0.537}\left(\frac{\alpha}{90}\right)^{1.361}\left(1+\frac{\beta}{180}\right)^{0.281}\left(1+d^*\right)^{0.623} \tag{12}$$

Figs. 14-15 compares the predicted Nusselt number and friction coefficient with experimental data. It can be deduced that the correlations can calculate the average Nu number and friction factor with the highest band deviations of ± 11.0%, and 14.2%, compared to the experimental data.

As mentioned in the introduction, many turbulator geometries have been presented by researchers to intensify the heat transfer performance of the heat exchangers. Fig. 16 compares the heat transfer improvement presented in this study with the ones from previous studies. As seen, the current study (RIE vortex generators) with η=2.23 has the maximum thermal performance among other geometries available in the literature. The second-best geometry in terms of thermal performance factor is the V-baffled tape, which has η greater than 1.5. The thermal performance of below unity indicates that the proposed turbulators in those previous studies have lower thermal performance factor compared to the simple plain tube ($\eta = 1$) and does not provide any improvements. The conical rings [15] and delta-wing twisted tapes [14] have the lowest thermal performance, a value below 1.0.

5. Conclusion

In this research, experimental analysis has been performed to investigate the effect of newly designed rotated inclined elliptic vortex generators on thermal performance improvement of the DPHEX under a turbulent flow regime. The design parameters are the Reynolds number $\left(6000 \le \mathrm{Re} \le 18000\right)$, rotation angle $\left(0^\circ \le \beta \le 90^\circ\right)$, inclination angle $\left(15^\circ \le \alpha \le 25^\circ\right)$, and hole diameter ratio $\left(0 \le d_p / b \le 0.25\right)$. The main conclusions are as follows:

- The proposed novel RIE vortex generators can substantially enhance the thermal performance of the heat exchanger compared to the previous studies in this field without raising the manufacturing costs. They can also simply be installed inside the heat exchanger pipes.
- The maximum thermal performance of η=2.23 is achieved at Re=16,000 by using the novel perforated RIE vortex generators with $d^* = 1.25, \alpha = 25°, \beta = 90°$. The thermal performance is considerably improved compared to the recent achievements in this field.
- The heat transfer rate is increased by 30.7% by using RIE turbulators with $\beta = 90^\circ$ compared to the typical louvered inclined vortex generators without rotations ($\beta = 0$). The primary reason for heat transfer improvement is the additional recirculations and flow disturbance in the presence of the RIE turbulators.
- Increasing the diameter of the rotated elliptic turbulators disrupts the thermal boundary layer near the tube walls. Moreover, the heat transfer rate can be increased due to the vortex generation and fluid combination between the heated walls and the central regions of the pipe. The average Nusselt number is enhanced by 59.95% by utilizing

NRIE vortex generators with d^*=0.250 in comparison with the typical non-rotated louvered strip inserts (d^*=0).

- The highest average Nusselt number of 342.62 is obtained by using RIE turbulators with d^*=0.25, $\alpha = 25^\circ$, and $\beta = 90^\circ$.
- Two empirical correlations are provided to predict the heat transfer and friction loss through double-pipe HEs with the newly designed RIE turbulators.

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Figures

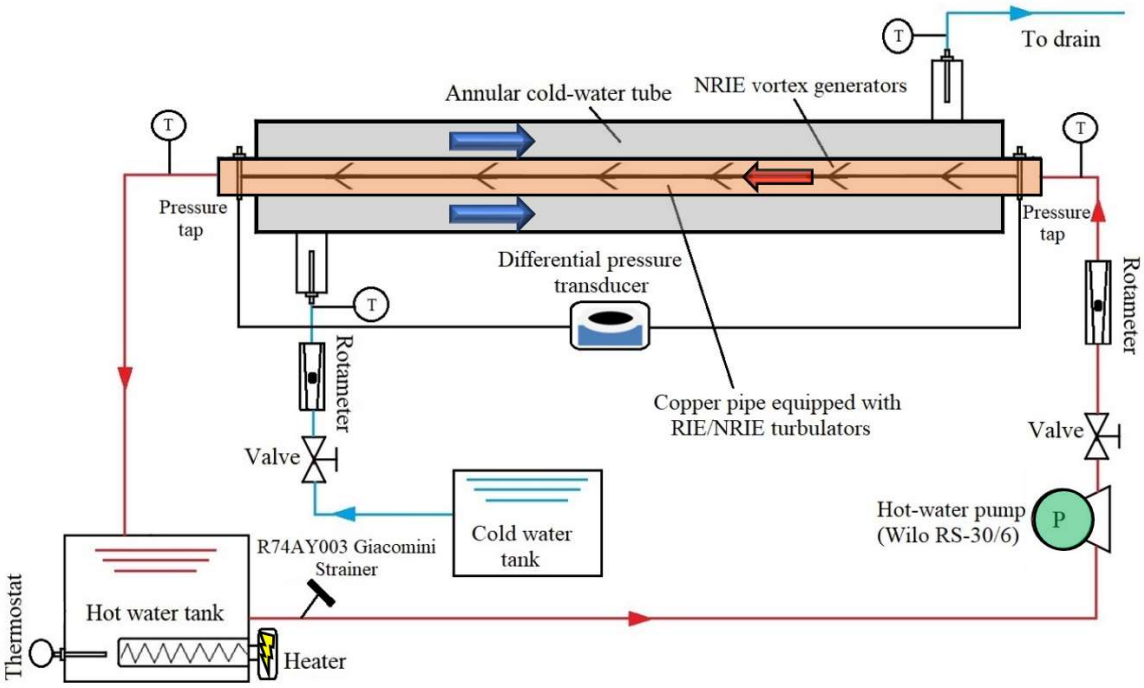


Fig. 1 Schematic view of the experimental setup with the rotated inclined elliptic turbulators.

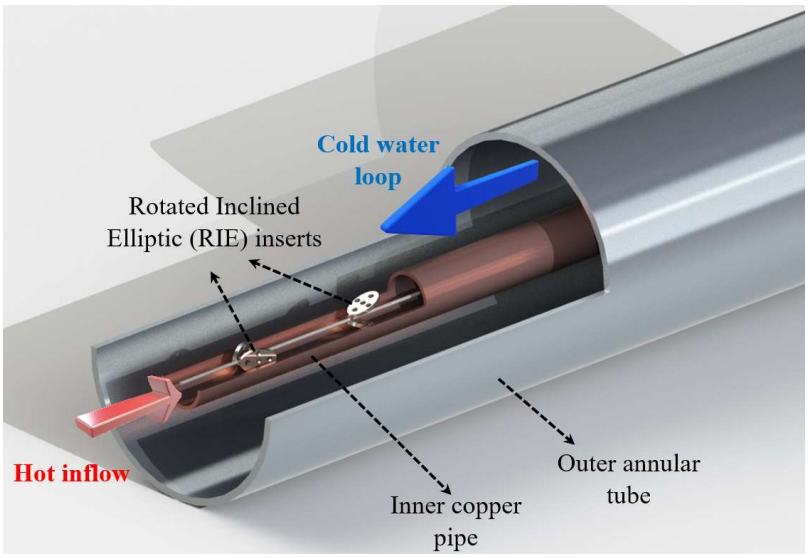


Fig. 2 Detailed view of the double-pipe heat exchanger test section equipped with the RIE inserts.

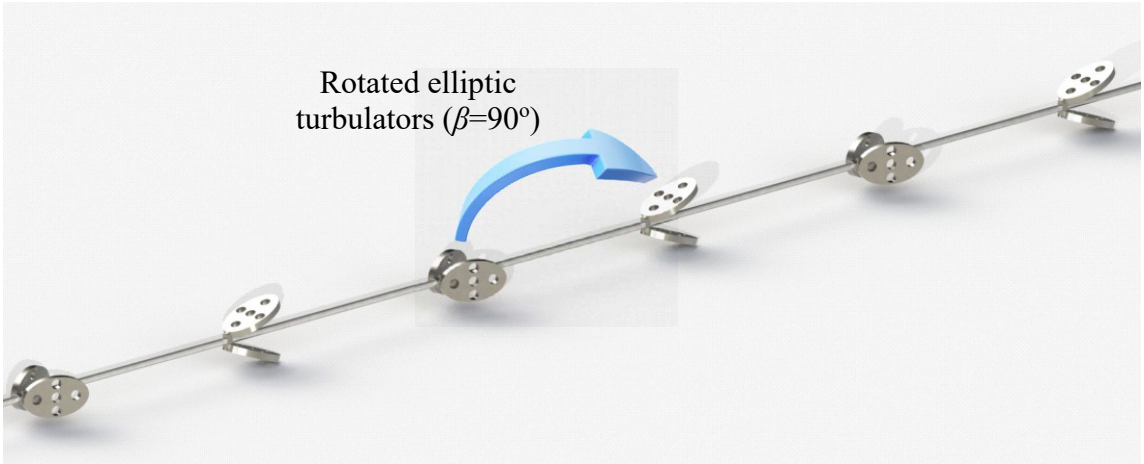
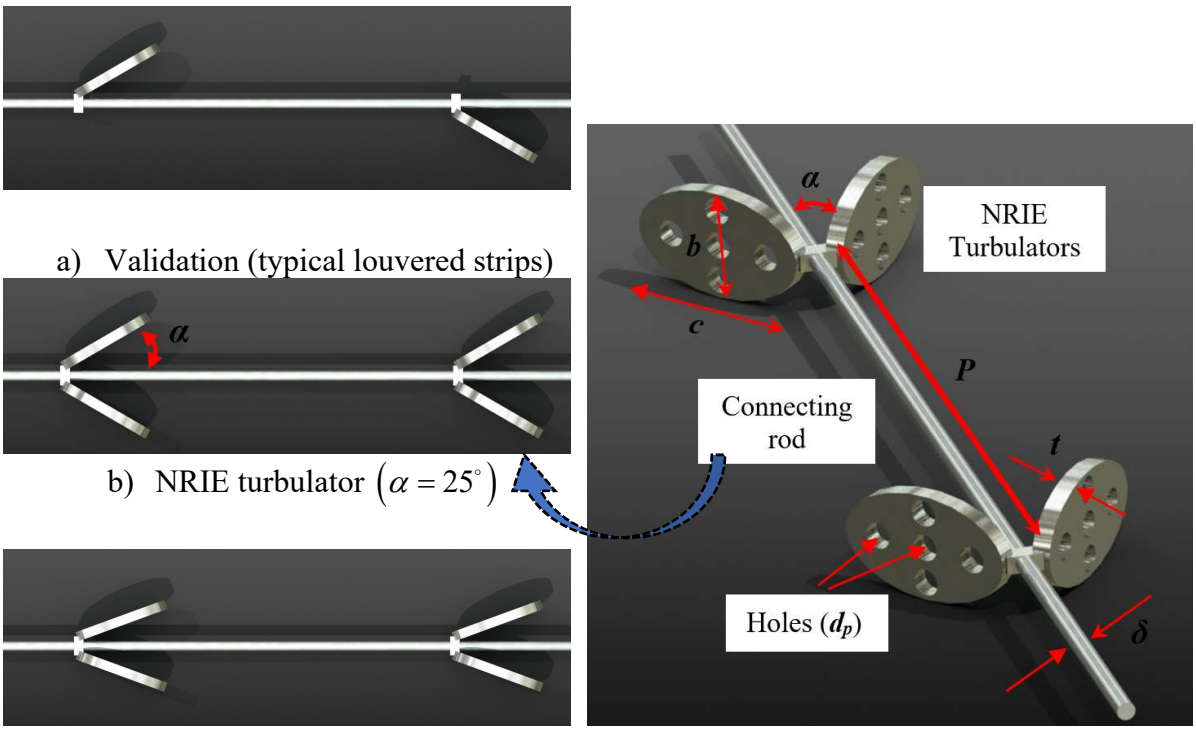


Fig. 3 The Rotated inclined elliptic (RIE) turbulators



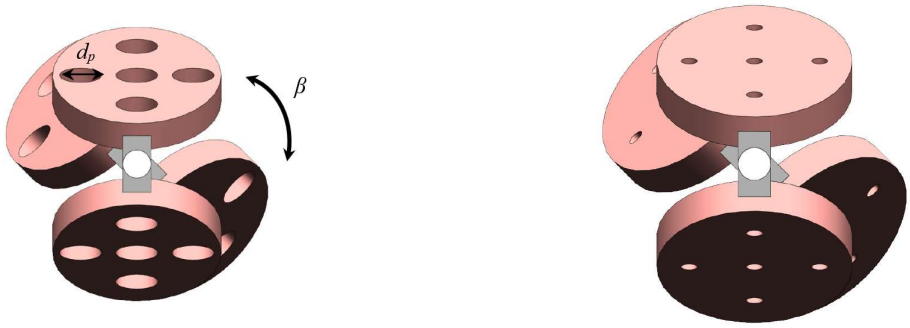
a) Validation (typical louvered strips)

b) NRIE turbulator $(\alpha = 25^{\circ})$

c) NRIE turbulator $(\alpha = 15^{\circ})$

d) Geometrical parameters

Fig. 4 Geometrical parameters of the NRIE inserts with the connecting rod.



a) $\alpha = 25^{\circ}, d_p = 1.5mm, \beta = 45^{\circ}$

b) $\alpha = 25^{\circ}, d_p = 0.5mm, \beta = 45^{\circ}$

c) $\alpha = 25^{\circ}, d_p = 1.5mm, \beta = 90^{\circ}$

d) $\alpha = 25^{\circ}, d_p = 0.5mm, \beta = 90^{\circ}$

Fig. 5 Geometrical parameters of the RIE inserts with the rotation angle (β).

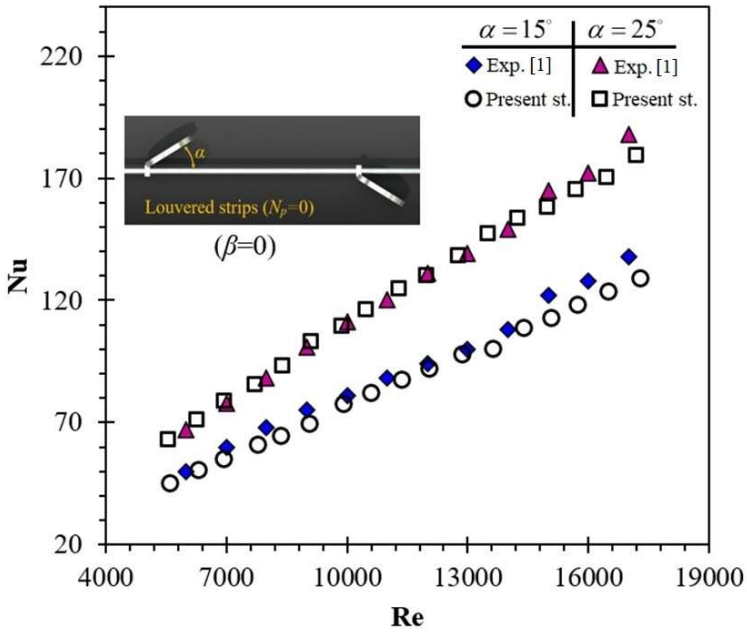


Fig. 6a Validation of the heat transfer coefficient with the experimental data [1] in double-pipe heat exchanger with typical louvered strips.

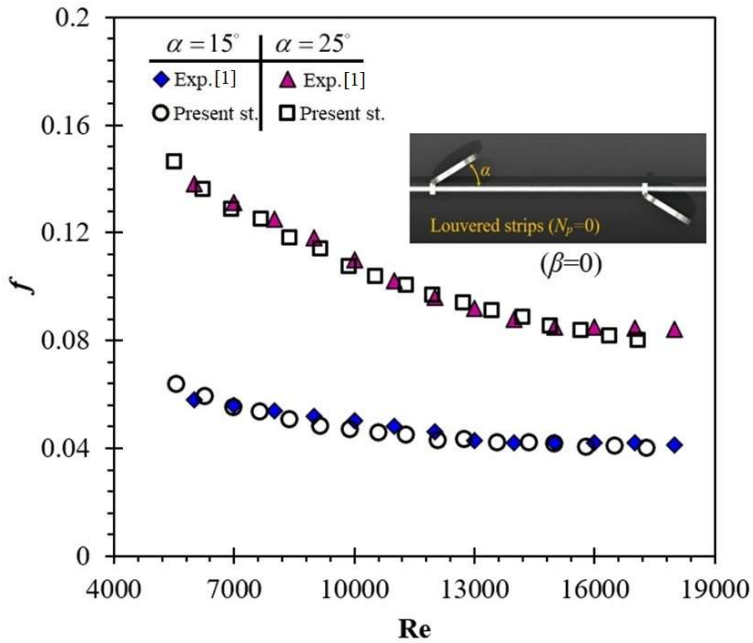


Fig. 6b Validation of the friction loss with the experimental data [1] in double-pipe heat exchanger with typical louvered strips.

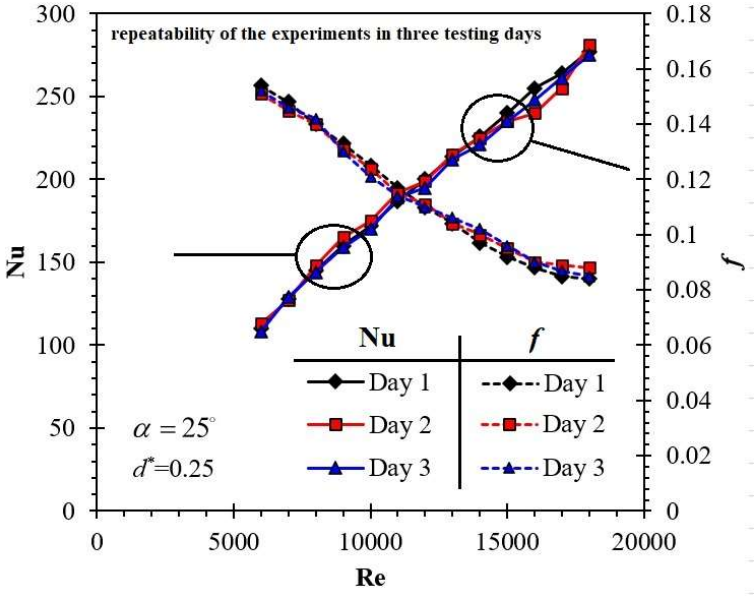


Fig. 7 Repeatability tests of the experimental results in different testing days.

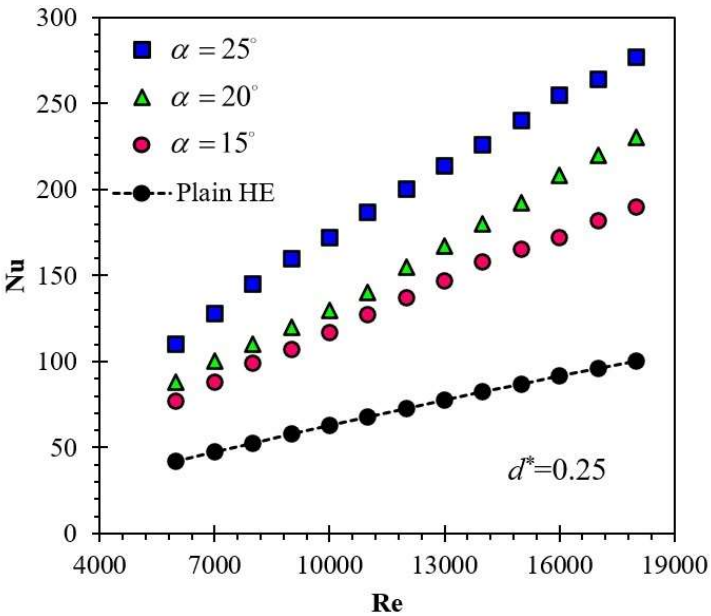


Fig. 8 Impacts of the inclination angle (α) of NRIE turbulators $(\beta = 0)$ on the average Nusselt number.

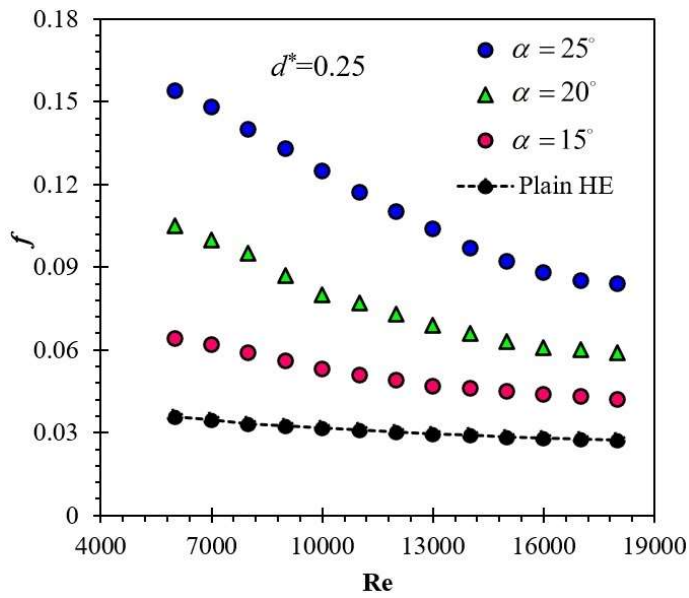


Fig. 9 Impacts of the inclination angle (α) of NRIE turbulators $(\beta = 0)$ on the friction factor in double-pipe heat exchanger tube.

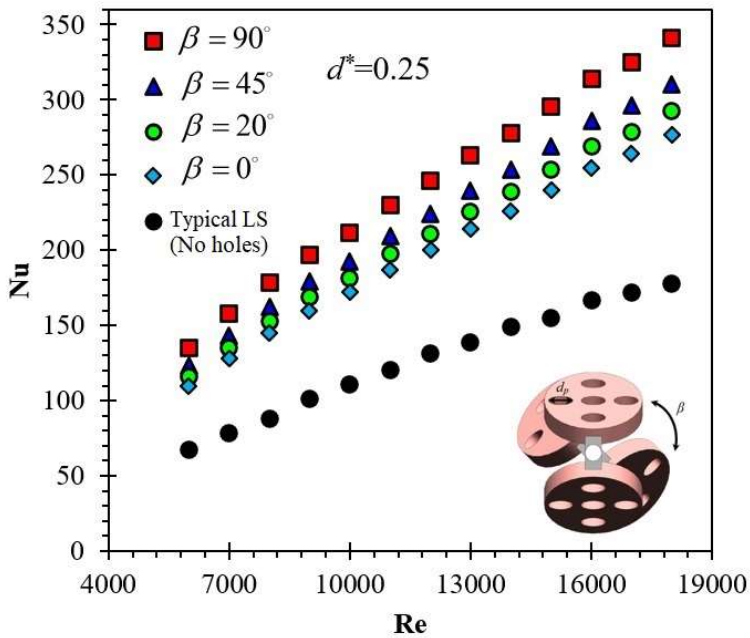


Fig. 10 Impacts of the rotation angle (β) of the RIE turbulators on the average heat transfer coefficient with $\alpha = 25^\circ$ and $d^* = d / b = 0.25$.

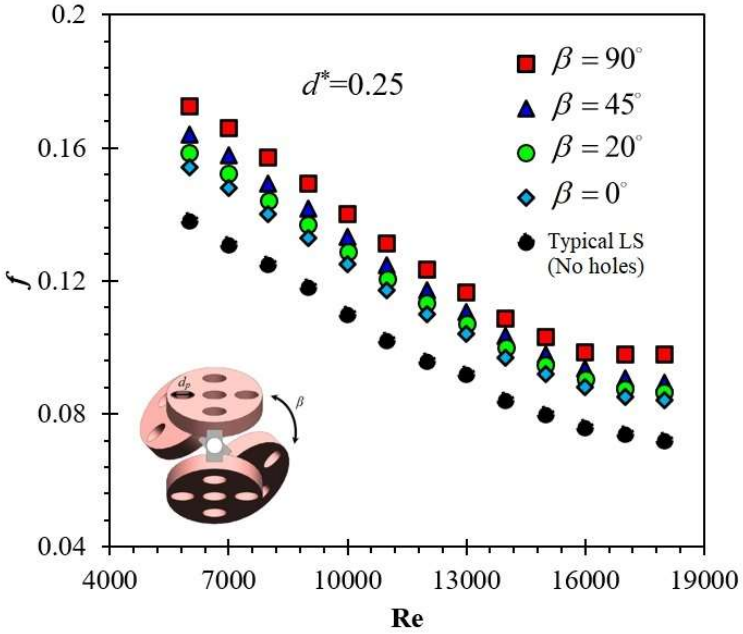


Fig. 11 Impacts of the rotation angle (β) of the RIE turbulators on the friction loss coefficient with $\alpha = 25^{\circ}$ and $d^{*} = 0.25$.

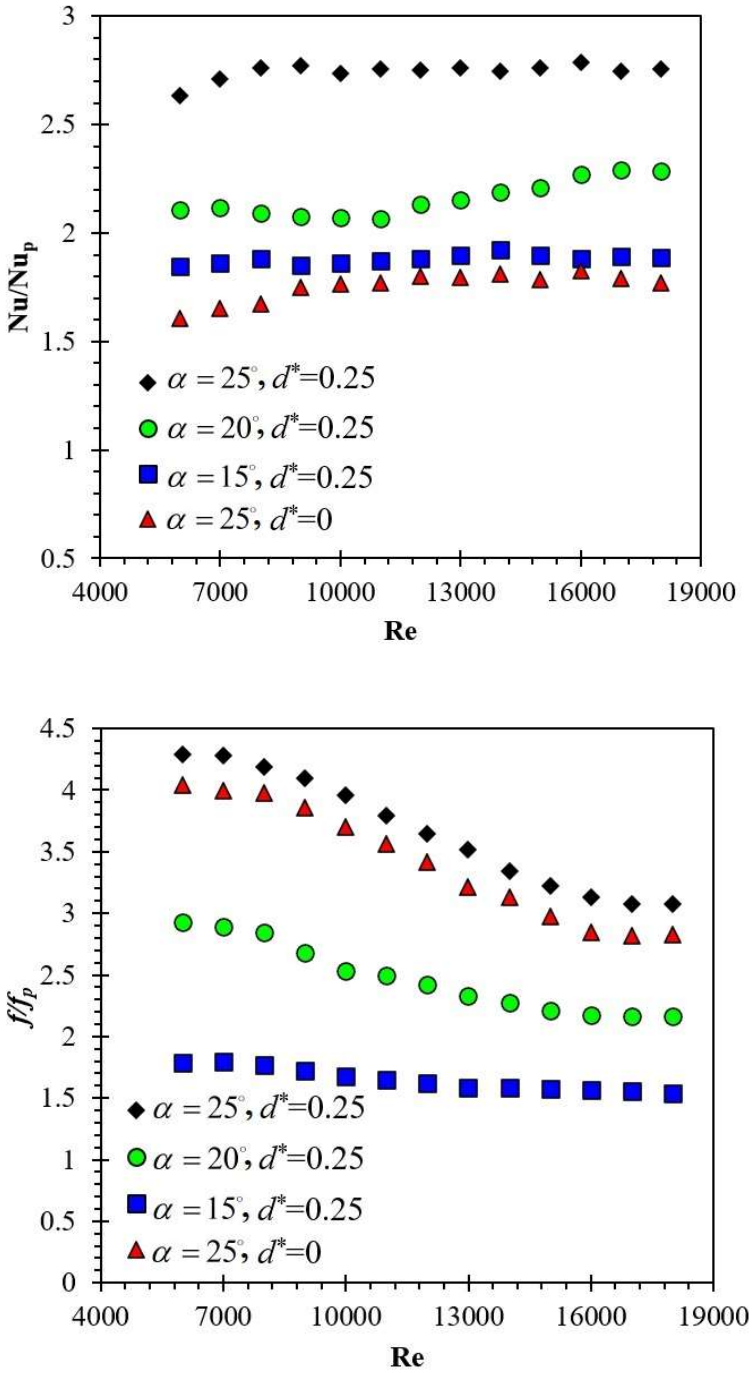


Fig. 12 Variations of Nu/Nu_p and f/f_p with Re for various NRIE turbulators.

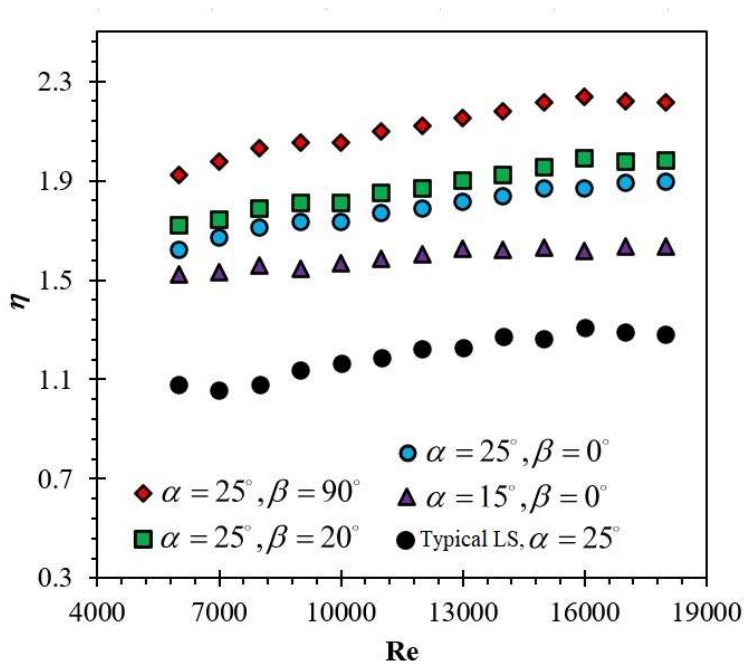


Fig. 13 The effects of the inclination angle (α) and the rotation angle (β) of the RIE vortex generators on the thermal performance intensification at various Re numbers.

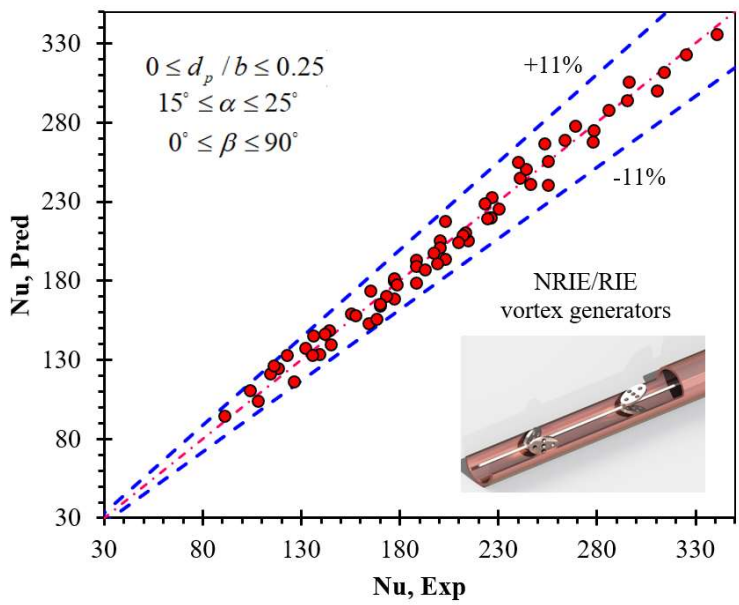


Fig. 14 Comparison among the experimental data and the predicted Nu number.

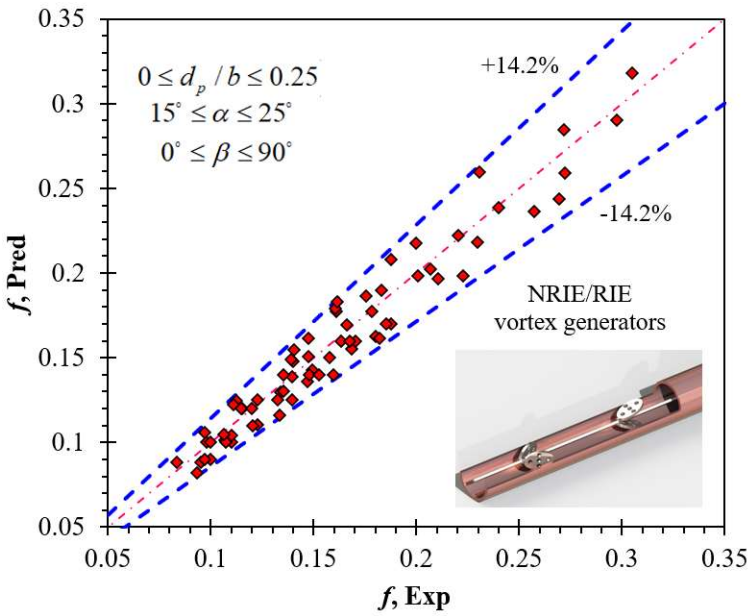


Fig. 15 Comparison among the experimental data and the predicted friction loss coefficient.

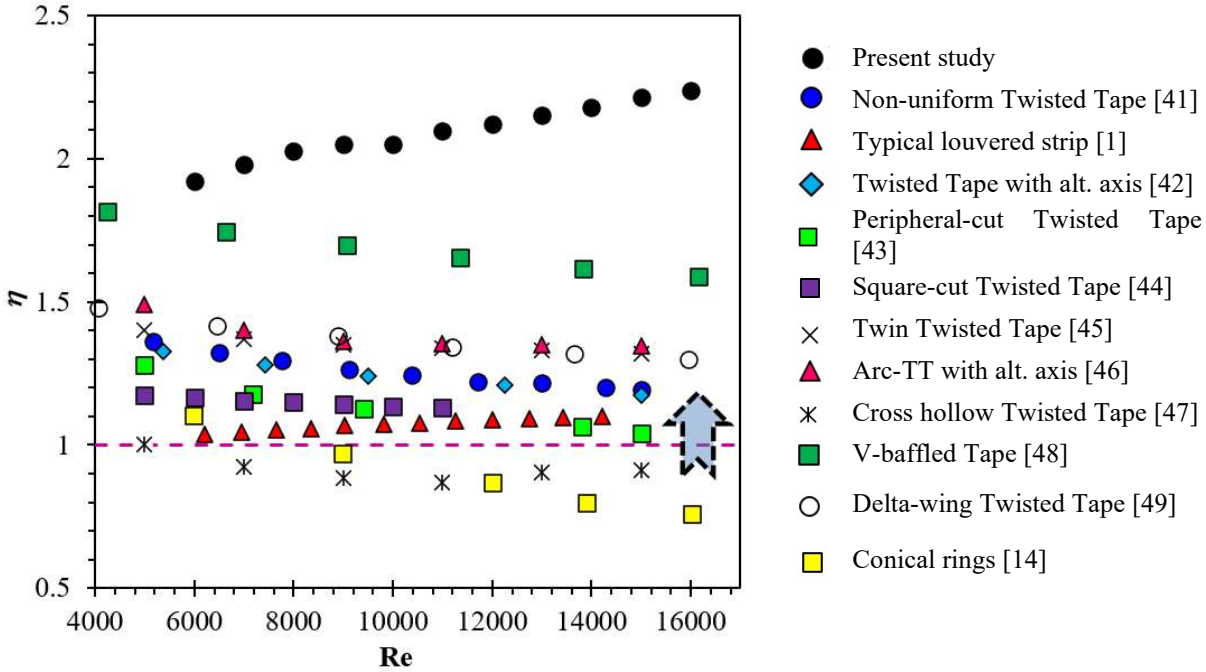


Fig. 16 Comparison amongst the thermal efficiency factor of the current study

$(d^* = 1.25, \alpha = 25^\circ, \beta = 90^\circ)$ with previous other turbulator devices.