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PERFORMANCE OF DIFFERENT TWISTED TAPE INSERTS IN TURBULENT TUBE FLOW – CRITICAL REVIEW

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ABSTRACT

There are different kinds of inserts employed in the heat exchanger tubes such as helical/twisted tapes, coil wires, ribs/fins/baffles, and winglets. This paper presents performance evaluation of some twisted tape inserts, using a simple evaluation criterion to assess the possible energy benefit. Twisted tape inserts, classical and modified, with different geometrical parameters with working fluids as air or water in the range $R_e = (0.2 - 5.0) \times 10^4$ *have been taken into consideration.*

INTRODUCTION

Increasing energy demand caused high cost of energy as well as material, which has resulted in an increased effort to produce high performance heat exchanger equipment. Heat transfer augmentation techniques are frequently used in heat exchanger systems in order to enhance heat transfer and increased the thermal performance.

Heat transfer augmentation can be classified broadly into two main techniques: active and passive methods. In the active methods, heat transfer is improved by supplying extra energy from external sources. On the other hand, the passive enhancement can be attained without any external energy source but an additional power needed to enhance the heat transfer can be taken from the available power in the system.

In general, the passive method involves the modifications of flow surfaces (such as turbulators and swirl generations). Among the passive techniques, insertion of swirl generator is one of the most promising techniques. The major function of swirl generator device is producing recirculation of an existing axial flow, leading to an improvement of fluid mixing and thus an efficient reduction of the thickness of a boundary layer.

Among various techniques, insertion of twisted tape swirl generator is one of the most promising techniques, which has been widely adopted for heat transfer augmentation. The presence of swirl generator i.e., twisted tape causes reduction of the hydrodynamic or thermal boundary layer thickness which leads to greater convective heat transfer rate. It can be explain that such tape induce turbulence and superimposed vortex motion (swirl flow) causing a thinner boundary layer.

Thus, twisted tape inserts have been widely used as the continuous swirl flow devices for enhancing the heat transfer performance in heat exchanger systems and applied in many engineering applications; for example, heat recovery processes, air conditioning and refrigeration systems, and chemical reactors.

It is obvious that associated with heat transfer augmentation the friction in the tube equipped with twisted tape insert and the pumping power is inescapably increase, which leads to a considerable increase of pumping cost. Therefore, the proper design of twisted tapes is a challenging task to meet the requirements of satisfactory heat transfer enhancement with a reasonable pressure drop, resulting in effecting energy saving.

The studies on heat transfer enhancement by means of twisted tape inserts have been extensively carried out for many years. Conventional twisted tapes are usualy installed in a heat exchanger tube to promote the fluid transfer between the tube core and the tube wall, Fig. 1. As reported in several research papers, twisted tapes can be combined with several enhancement devices for further improvement of heat transfer rate, for example: twisted with spirally grooved tube, converging-diverging tube, corrugated tube, dimpled tube, or twisted tape with wire-coil inserts.

Another approach to improve their performance is modifying their geometries to induce extra fluid low disturbing, such as peripherally cut twisted tape, broke twisted tape insert, serrated twisted tape, twisted tape with oblique teeth, delta-winglet twisted tape, trapezoidal-cut twisted tape, square- cut twisted tape, twisted tape with wire-nail, twisted tape with alternate-axes and triangle, rectangular and trapezoidal wings, and perforated twisted tape.

Mostly, the heat transfer rate associated by the combined techniques or the modified twisted tapes are higher than those given by typical twisted tapes. However, an improve heat transfer is unavoidably accompanied by an increase of pressure drop. It is a challenging task of

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researchers to modify twisted tapes with proper geometries in order to achieve an excellent heat transfer with reasonable pressure drop penalty.

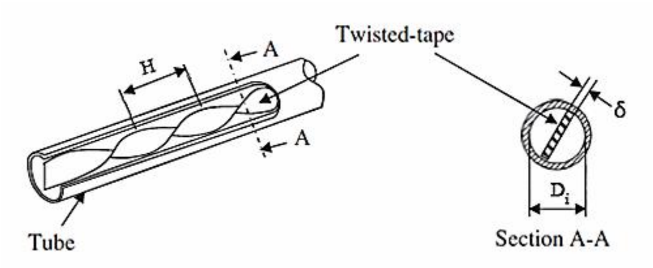


Fig. 1. The illustration of tube with conventional twisted tape insert

Wongcharee and Eiamsa-ard [1] studied the effect of twisted tapes with alternate-axes and wings on heat transfer, flow friction and thermal performance. The influence of wing shape including triangle, rectangle and trapezoid is also studied. The results show that both heat transfer and friction factor associated by all twisted tape are consistently higher than those without twisted tape. Under the similar operating conditions, Nusselt number, friction factor as well as thermal performance factor given by the tape with alternate-axes and trapezoidal wings are higher than those given by the others. Eiamsa-ard et al. [2] describe heat transfer enhancement attributed to helically twisted tapes. The conventional helical tape was also tested for a comparison. The obtained result reveals that at similar conditions, helically twisted tapes give lower Nusselt number and friction factor but higher thermal performance factor than conventional helical tape. Bhuiya at al. [3] explored the effect of perforated double counter twisted tapes on heat transfer and fluid friction characteristics in a heat exchanger tube. The experimental results demonstrated that the Nusselt number, friction factor and thermal enhancement efficiency were increased with deacreasing porosity except porosity of 1.2%. Murugesan at al. [4] investigated the effect of V-cut twisted tape insert on heat transfer, friction factor and thermal performance factor characteristics in a circular tube. The results show that the mean Nusselt number and the mean friction factor in the tube with V-cut twisted tape increased with deacreasing twist ratios, width ratios and increasing depth.

EXPERIMENTAL RESULTS

The twisted tapes used by Wongcharee and Eiamsa-ard [1] in their study are shown in Fig.1.

All of the tapes used were made of aluminum strips with thickness δ and width W. The modified twisted tapes were prepared at three different wing-chord ratios d/W, constant twisted ratio y/W and wing span ratio b/W. The geometrical parameters of the modified twisted tapes are presented in Table 1.

The experimental data of study [1] are fitted by the following empirical equations in the range of Reynolds number $Re = (5.5 - 20.2) \times 10^3$ and $Pr = 7.0$.

- Correlations for twisted tapes with trapezoid wings:

$$Nu = 0.625\,Re^{0.547}\,Pr^{0.4}\left(\frac{d}{W}\right)^{0.113}, \tag{1}$$

$$f = 64\,Re^{-0.587}\left(\frac{d}{W}\right)^{0.189}, \tag{2}$$

- Correlations for rectangular wings:

$$Nu = 0.506\,Re^{0.562}\,Pr^{0.4}\left(\frac{d}{W}\right)^{0.103} \tag{3}$$

$$f = 43.7\,Re^{-0.562}\left(\frac{d}{W}\right)^{0.181} \tag{4}$$

- Correlations for triangular wings:

$$Nu = 0.404\,Re^{0.58}\,Pr^{0.4}\left(\frac{d}{W}\right)^{0.096}, \tag{5}$$

$$f = 36\,Re^{-0.553}\left(\frac{d}{W}\right)^{0.172}. \tag{6}$$

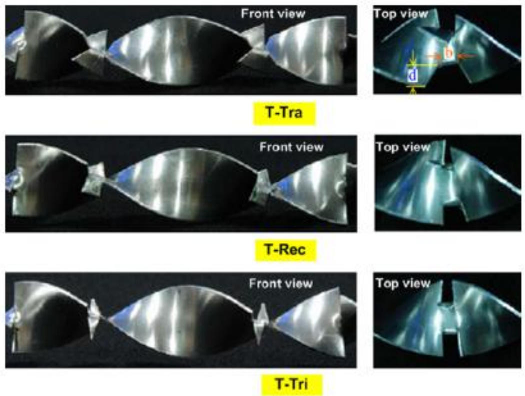


Fig. 1. The tape inserts used by Wongcharee and Eiamsa-ard [1]

Table 1 Geometrical parameters of triangular, rectangular and trapezoidal wings [1]

No	W mm	d/W	y/W	b/W
1	20	0.1	4	0.2
2	20	0.2	4	0.2
3	20	0.3	4	0.2
4	20	0.1	4	0.2
5	20	0.2	4	0.2
6	20	0.3	4	0.2
7	20	0.1	4	0.2
8	20	0.2	4	0.2
9	20	0.3	4	0.2

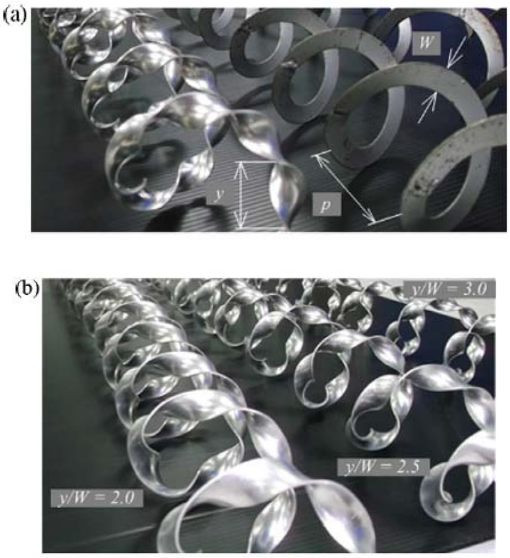
Fig. 2 shows photographs of helically twisted tapes (HTT) and conventional helical tapes (CHT) studied by Eiamsa et al. [2]. The geometric details of the helically twisted tapes are presented in Table 2. The experiments were performed using helically twisted tapes with constant width W and an inner diameter of the tube D, three different twist ratios y/W and helical pitch ratios p/D.

The correlations for helically twisted tapes

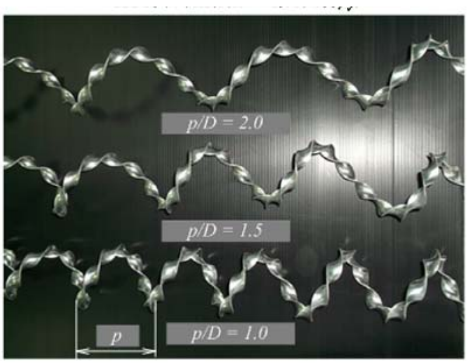
$$Nu = 0.053\,Re^{0.796}\,Pr^{0.4}\left(\frac{y}{W}\right)^{-0.127}\left(\frac{p}{D}\right)^{-0.188} \tag{7}$$

$$f = 12.653\,Re^{-0.295}\left(\frac{y}{W}\right)^{-0.652}\left(\frac{p}{D}\right)^{-1.513} \tag{8}$$

have been used to calculate the ratios Nu/Nu_s and f/f_s at $Pr = 0.7$ and $Re = (6.0 - 20.0) \times 10^3$.



HTTs at different twist ratios, y/W



HTTs at different pitch ratios, p/D

Fig. 2. Photographs of helically twisted tapes (HTT) and conventional helical tapes (CHT): (a) geometries of HTT and CHT and (b) geometric details of helically twisted tapes , Eiamsa et al. [2]

Table 2 Geometrical parameters of helical twisted tapes [2]

No	D	W	y/W	p/D
	mm	*mm*		
1	64	10	2.0	1.0
2	64	10	2.0	1.5
3	64	10	2.0	2
4	64	10	2.5	1.0
5	64	10	2.5	1.5
6	64	10	2.5	2.0
7	64	10	3.0	1.0
8	64	10	3.0	1.5
9	64	10	3.0	2.0
10	64	10	-	1.0
11	64	10	-	1.5
12	64	10	-	2.0

The next experimental results that have been assessed in this study are those of Bhuiya at al. [3]. Fig. 3 depicts the geometry of the test section fitted with perforated double counter twisted tape inserts. The geometrical parameters of the circular tube studies outlined in [3] are presented in Table 3 where twisted tapes with four different porosities R_p are introduced: with constant twist ratio p/W and four different pore diameters of the tape d_s.

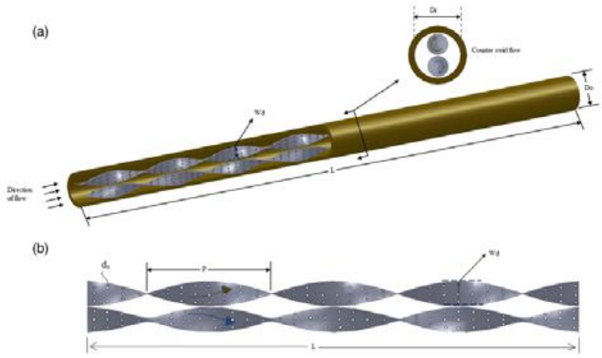


Fig. 3. (a) Geometry and test section fitted with perforated double counter twisted tape insert; b) Geometric parameters of the perforated double counter twisted tape insert

Table 3 Geometrical parameters of perforated double counter twisted tape [3]

No	R_p	d_s	p/W
	%	*mm*	
1	1.2	2	1.92
2	4.6	4	1.92
3	10.4	6	1.92
4	18.6	8	1.92

Heat transfer and fluid friction for air were measured in the range of $R_e = (7.2 - 50.2) \times 10^3$ and correlated as follows:

Correlations for perforated double counter twisted tapes:

$$Nu = \{0.0003y^3 - 0.0093y^2 + 0.0556y + 0.6194\} \times Re^{\{-0.000004y^3+0.0002y^2-0.0001y+0.546\}} Pr^{33} \tag{9}$$

$$f = \{0.0008y^3 - 0.0231y^2 - 0.1877y + 16.659\} \times Re^{\{-0.00004y^3+0.0012y^2-0.0108y+0.5579\}} \tag{10}$$

The last study investigated in this research is that of Murugesan at al. [4]. The geometries of V-cut twisted tapes insert in a circular tube are shown in Fig. 4 and Table 4 for three twist ratios y and three different combinations of depth DR and width ratios, where d_e is depth of V cut, W is tape width and respectively w is width of V cut.

Thermal performance characteristics of V-cut twisted tapes inserts in a circular tube in the range of Reynolds number $Re = (2.0 - 11.0) \times 10^3$ and $Pr = 7.0$ have been evaluated using the next correlations for V-cut twisted tapes

$$Nu = 0.0296\, Re^{0.853}\, Pr^{0.33}\, y^{-0.222} \times \left(1 + \left[\frac{d_e}{W}\right]\right)^{1.148} \left(1 + \left[\frac{w}{W}\right]\right)^{-0.751} \tag{11}$$

and

$$f = 8.632\, Re^{-0.615}\, y^{-0.269} \times \left(1 + \left[\frac{d_e}{W}\right]\right)^{2.477} \left(1 + \left[\frac{w}{W}\right]\right)^{-1.914} \tag{12}$$

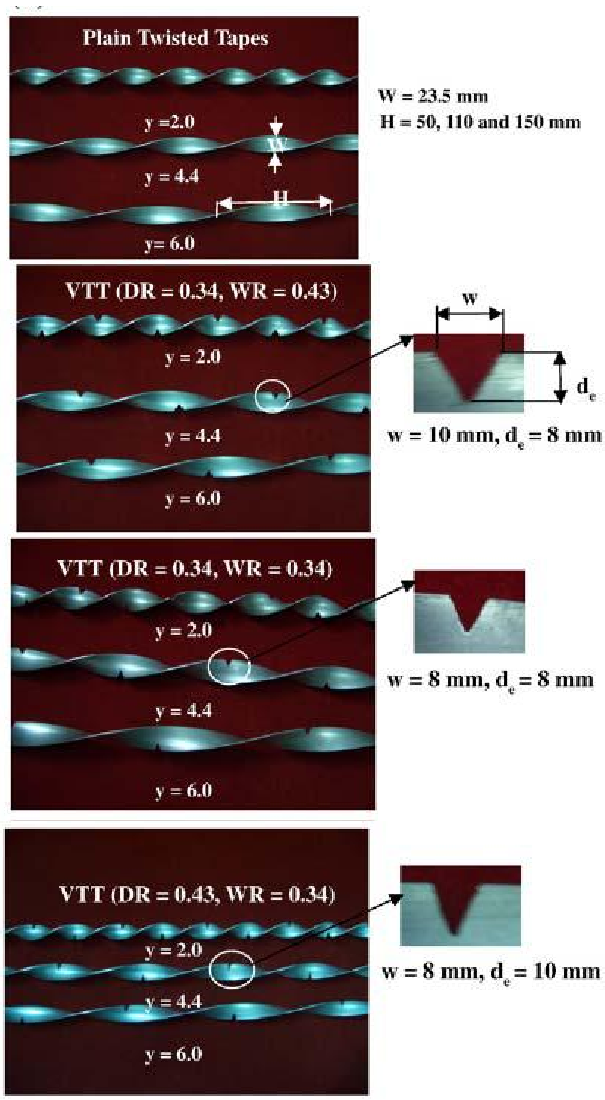


Fig. 4. Geometries: (a) plain twisted tapes (b) V-cut twisted tapes

Table 4. The geometries of V-cut twisted tapes insert [4]

No	y	DR	WR	W mm	w mm	d_e mm
1	2.0	0.43	0.34	23.5	8	10
2	2.0	0.34	0.34	23.5	8	8
3	2.0	0.34	0.43	23.5	10	8
4	4.4	0.43	0.34	23.5	8	10
5	4.4	0.34	0.34	23.5	8	8
6	4.4	0.34	0.43	23.5	10	8
7	6.0	0.43	0.34	23.5	8	10
8	6.0	0.34	0.34	23.5	8	8
9	6.0	0.34	0.43	23.5	10	8
10	2.0			23.5		
11	4.4			23.5		
12	6.0			23.5		

PERFORMANCE EVALUATION AND DISCUSSION

Many performance evaluation criteria (PEC) have been developed for evaluating the performance of heat exchangers. They may be categorized as criteria based on the first law of thermodynamics and criteria based on the second law of thermodynamics.

A widely used method to evaluate the benefit of an enhanced heat transfer surface is to compare the performance of the enhanced surface with that of the corresponding plain (smooth) surface. The benefit depends on the goal to be achieved and the constraints imposed of the comparison. In general, the performance evaluation includes three considerations: the performance objective, operation conditions and constraints. The potential objectives could be: increased heat transfer rate, reduced pumping power, or reduced size of the heat exchanger. Possible constraints are: fixed mass flow rate, heat flow, pumping power, size of the heat exchanger, etc. The major operational variables include the heat transfer rate, fluid pumping power or pressure drop, flow rate, and fluid velocity. A PEC is established by selecting one of the operational variables for the performance objective subject to design constraints on the remaining variables.

The most common PEC, used by many researchers for easy evaluation of the practical application of heat transfer enhancement, are those proposed by Bergles et al. [5]. When the PEC are directed to evaluate the improvement of existing exchanger, then the basic geometry is fixed, and relationships are obtained related to the increase of heat flow or decrease in pumping power. If the objective is more heat flow to be transferred, this criterion is known as R_3 [5]

$$R_3 = \left(\frac{h}{h_s}\right)_{D,L,N,P,T_{in},\Delta T} = \frac{\dot{Q}}{\dot{Q}_s} \tag{13}$$

In this case, the process constraints are: fixed pumping power P, inlet fluid temperature T_{in}, and driving temperature difference ΔT. The constraint of equal pumping power requires different Reynolds numbers for the working fluid in reference and augmented channels, $\mathrm{Re} < \mathrm{Re}_s$, and

$$f(Re)Re^3 = f_s(Re_s)Re_s^3 . \tag{14}$$

Consequently, the corresponding heat transfer coefficients in Eq. (13) should be calculated at these Reynolds numbers and Eq. (13) yields

$$R_3 = \frac{Nu(Re)}{Nu_s(Re_s)} = \frac{\dot{Q}}{\dot{Q}_s} . \tag{15}$$

It must be emphasized that the criterion R_3 has been developed with the assumptions of negligible external thermal resistance, $R_{ext} = 0$, and equal temperature difference ΔT in the comparative heat exchangers. In general, however, the ΔT will decrease due to the increased rate of heat transfer [5].

Sano and Usui [6] suggested evaluation of the heat transfer promoters by fluid dissipation energy, developing a criterion based on the correlation of the heat transfer coefficient as a function of the energy dissipation per unit mass of fluid (ε). For turbulent flow, this criterion takes the form [6]

$$i_E = \frac{Nu/Nu_s}{(f/f_s)^{0.291}} = f(Re). \tag{16}$$

It is important to note that the criterion i_E is identical to R_3 criterion of Bergles et al. [5], but is more convenient to use since the Nusselt numbers Nu, Nu_s, and friction factors f, f_s are defined at one and the same Reynolds number Re. It should also be emphasized that the two heat exchangers must work in turbulent regime, $\mathrm{Re} > 3\times10^3$, and $\mathrm{Re}_s > \mathrm{Re}$.

Fig.5 presents the variation of i_E with the Reynolds number in the augmented channel for the experimental results of Wongcharee and Eiamsa-ard [1]. The tubes with twisted tapes with triangular wings are for 1-3, rectangular wings are for 4-6 and trapezoidal wings are for tubes 7-9.

As seen, all twisted tapes 1-9 performed with $i_E > 1$ and the variation with the Reynolds number is large enough. The benefit of some of them is very similar, tapes 2, 4, 7. The greatest profit can be obtained by using the tapes 3, 6 and 9 as all of them have one and the same d/W = 0.3.It should be noted that the value of i_E decrease with the increase of Reynolds number, $0.5.\times10^4 > \mathrm{Re} > 2.0\times10^4$

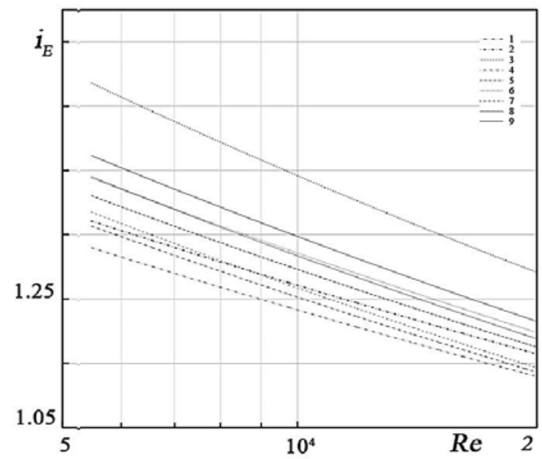


Fig. 5. The variation of i_E with Reynolds number. Experimental result of Wongcharee and Eiamsa-ard [1]

The equation of thermal performance factor used in [2] can be written as

$$\eta = \left.\frac{h}{h_s}\right|_{pp} = \left.\frac{Nu}{Nu_s}\right|_{pp} = \left(\frac{Nu}{Nu_s}\right)\left(\frac{f}{f_s}\right)^{-1/3} . \tag{17}$$

The comparison with the results published in [1] using the criterion and those in this study showed that the benefit assessed by criterion i_E is bigger than this one calculated through the criterion η . As a result, it can be concluded that tube 3 is the most efficient from the all nine tube studied with performance value $i_E = 1.60 - 1.62$.

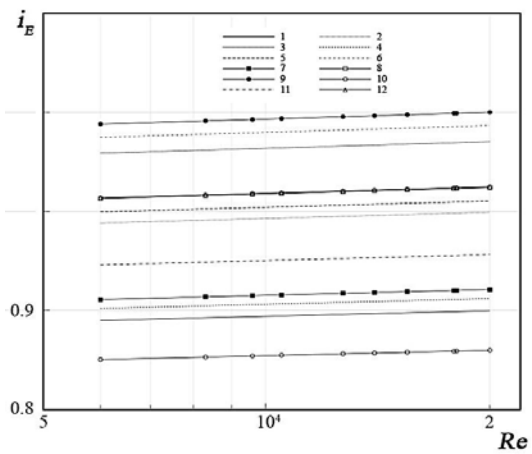


Fig. 6. The variation of i_E with Reynolds number. Experimental result of Eiamsa-ard et al. [2]

Fig. 6 shows the variation of the criterion i_E with the Reynolds number for the experimental results of Eiamsa-ard et all [2]. As seen from the figure, the helical twisted tapes with helical pitch ratio $p/D = 2$ have the higher value of i_E . The helical twisted tape 9 has the highest value of i_E . the tapes 3, 6, 8, 9 and 10 have $i_E > 1$. The benefit remains negative, $i_E < 1$ for the rest as for the convention helical tape with $p/D = 1$ the criterion range is $i_E = 1.85 - 1.87$.

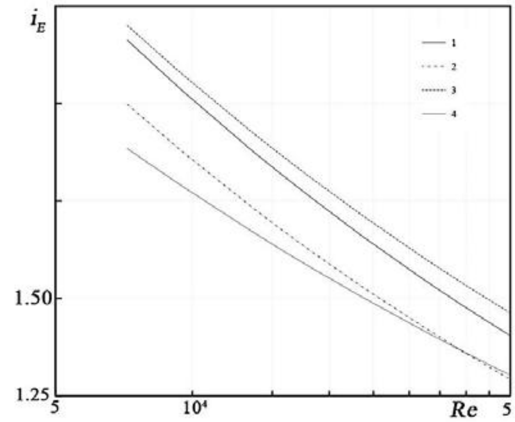


Fig. 7. The variation of i_E with Reynolds number. Experimental result of Bhuiya et al. [3]

The comparison between η and i_E shows that the thermal performance factor η used in [2] is more convenient and sophisticated than criterion i_E with range $\eta = 1.1 - 1.3$ for tapes 3, 6 and 9.

The evaluation of the experimental results of Bhuiya et al. [3] through the performance parameter i_E is shown in Figs. 7. As seen from the figure, tape 3 shows the highest value of i_E for porosity $R_p = 10.4$. All four perforated double counter twisted tapes performed with $i_E > 1$ which shows a significant enhancement for heat transfer. It should be noted that the value of i_E decrease rapidly with the increase of Reynolds number $0.7\times10^4 > \mathrm{Re} > 5.0\times10^5$. The comparison between the values of performance factor η obtained by the experimental results of Bhuiya et al. [3] and performance evaluation criteria i_E shows a huge advantage of using i_E . It increase in the range of $i_E = 1.27 - 2.20$ in comparison with $\eta = 1.1 - 1.4$.

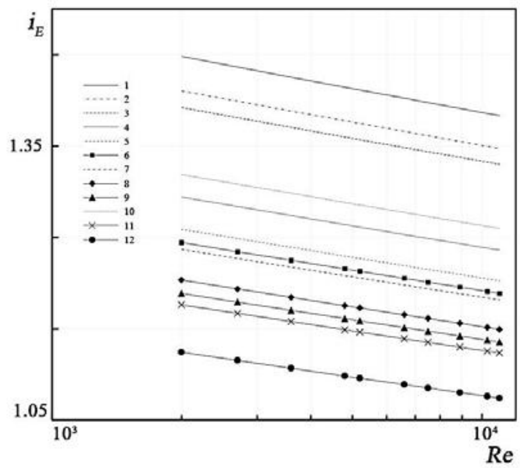


Fig. 8. The variation of i_E with Reynolds number. Experimental result of Murugesan et al. [4]

Fig. 8 presents the variation of i_E with Re, calculated by means of the experimental results of Murugesan et al. [4]. As seen, the V-cut twisted tapes 1, 2 and 3 with twisted ratio $y = 2.0$ performed better compared to the other tapes studied. However, the values of i_E are dependent on Reynolds number and regularly decrease with the increase of Re in the range of study.

The plain twisted tape 12 has the lowest value of i_E as at the end of the region, $i_E = 1.07 - 1.09$. The benefit between the criterion i_E which has been applied in fig.4 and the values of thermal performance factor η obtained by the experimental results of Murugesan et al. [4] is found to be greater using i_E as the benefit is about 15%.

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AUTHORS' INFORMATION

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