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SHORT COMMUNICATION

Comparative study for thermal-hydraulic performance of circular tube with inserts

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Abstract Several researchers have worked on the passive approach of heat transfer enhancement in tube heat exchangers. Some of them tried to modify the surface by creating dimple or using wire coil of different cross-section, while some worked on core fluid disturbance by using some insert geometries such as twisted tapes. But the ultimate aim of all was to create some disturbance in the flow in order to obtain enhanced heat transfer. This paper focuses on comparison of some of the most commonly used insert geometries. Insert geometry selected for this comparison is collection of core fluid disturbance, surface modification and combination of both. Different geometries taken in this study include twisted tape, twisted tape with ring, circular band, multiple twisted tape, twisted tape with conical rings, and so on and used air under turbulent flow regime as working fluid. On the basis of comparison made, it is observed that, in case of "single twisted tape insert" the thermal performance factor was maximum and in the event of "twisted tape with circular ring" the overall heat transfer rate is maximum. Future aspect is also proposed, which includes perforation in circular ring, and causes decrease in friction factor value because of less flow blockage.

1. Introduction

Heat transfer enhancement using turbulent promoters as inserts in heat exchangers has become a prime area of research for the researchers. Use of the different passive approach of heat transfer enhancement for improving thermo-hydraulic performance of heat exchanger, in order to cut down its size and price is the main motive for such kind of work. In the past two decades, several works have been carried out in this field in order to produce more efficient heat exchange devices for our thermal and mechanical systems. Different insert geometries with different parameter ranges have been used by researchers as turbulence promoters for the heat exchanger device. Some of the major recent studies include, Bas and Ozceyhan [1], who used single twisted tape as insert geometry separated from tube wall, and found very significant effect on the heat transfer. Similarly, Bhuiya et al. [2,3] used double counter twisted tape and triple twisted tape respectively in their study and found that, by decreasing the twist ratio of the twisted tapes, heat transfer rate increases to a great extent. Eiamsa-ard et al. [4], used circular ring with twisted tape as insert geometry

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Nomenclature

Abbreviations		η	thermal performance factor
TT	twisted tape	PR	pitch ratio
CRT	circular ring tuberator	DR	diameter ratio
Re	Reynolds number	CR	clearance ratio
Nu	Nusselt number	f_s	friction factor of plane tube
f	friction factor of tube with insert	Pr	Prandtl number
Nu_s	Nusselt number of plane tube		

for experimentation, and they found that by decreasing the spacing between circular ring, heat transfer increases. Promvonge [5] used twisted tape with uniform wire coil, Promvonge and Eiamsa-ard [6] used conical ring with twisted tape as their insert geometry, and they also found significant effect of ring spacing on the heat transfer in tube heat exchangers. Similarly, Gunes et al. [7] used triangular cross section wire coil as their insert geometry. Kongkaitpaiboon et al. [8] used circular ring tuberator in experimentation; significant effect of diameter ratio and pitch ratio was observed, which shows that as the value of diameter ratio increases heat transfer rate decreases, and as the value of pitch ratio decreases heat transfer increases. Promvonge et al. [9] used inclined vortex ring, and Eiamsa-ard and Promvonge [10,11] used double sided delta wing and serrated twisted tape respectively. Similarly some other important geometries used for such research are perforated twisted tape [12], conical nozzle [13], perforated conical ring [14], twisted ring [15], short length twisted tape [16], non uniform wire coil with twisted tape [17] and Protruded surface [18] with several geometrical and flow parameters respectively using air as working fluid. Chamoli et al. [19] shows several other geometries of turbulence promoters used in passive heat transfer enhancement technique in several solar thermal systems. In all of the above study, one thing is very common that all the researchers tried to bring turbulence in the fluid flow by creating surface modification or core fluid disturbance or both at the same time.

This comparative study focused on some of the most effective work in the recent year which includes, surface modification, core fluid disturbance and both of it, [1–9]. Comparison of heat transfer, friction factor and thermal performance factor is made on the basis of correlation developed by these researchers in their studies. This comparison also aims at developing some future aspect for the passive method of heat transfer enhancement.

2. Methodology

The study is borne out by comparing results and correlations of very similar variety of experimental study. Nine different insert geometries have been used up, which are single twisted tape, double twisted tape and triple twisted tape, TT with ring, CRT, etc. Each of the experiments has been performed on similar kind of experimental setup and because of their different geometries, different results and correlation have been obtained by different researchers. In this paper all the nine works had been compared in order to get optimal effect and future prospect. The different insert geometries and parameters used with their results are presented in Table 1. All the correlations for heat transfer, friction factor and thermal performance factor are presented in Table 2. Table 3 shows the values of parameters on which the correlation in each case shows the maximum result for heat transfer, friction factor, and thermal performance factor, respectively.

3. Results and discussion

3.1. Effect of twist ratio

3.1.1. Heat transfer

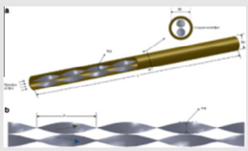
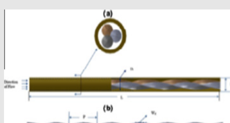
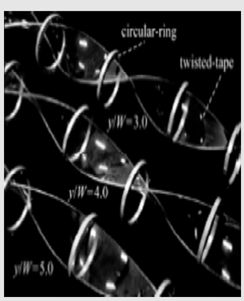
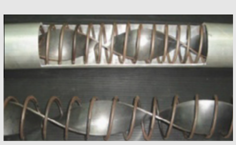
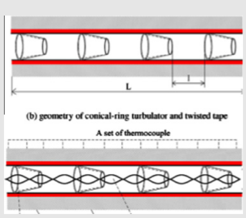
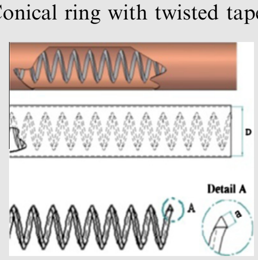
In each experimental run, it is observed that the heat transfer rate increases with decrease in twist ratio values. The results of heat transfer rate in terms of the Nusselt number for roughned tube and their enhancement over smooth tube are presented in Figs. 1(a) and (b) and it is observed from the graph that the heat transfer is maximum in case of twisted tape with circular ring insert [4], and minimum in case of single twisted tape insert [1]. In case of twisted tape with circular ring, the ring which is placed over the twisted tape plays an important role in the heat transfer. The insertion of ring modified the fluid flow, which accommodates to high heat transfer rate. It is also observed that for the higher Reynolds number, the heat transfer has higher values and as the value of the Reynolds number decreases the heat transfer rate also decreases.

3.1.2. Friction factor

In each experiment, it was found that the friction factor increases with a decrease in the twist ratio. Here along the basis of observation and the graph obtained by comparing each experiment which is presented in Figs. 2(a) and (b), it is found that in case of twisted tape with wire coiled insert [5] there is significant impact along the friction factor, only the maximum frictional factor can be understood in the case of the circular ring insert [7] and the minimum in case of single twisted tape [1], double twisted tape [2] and triple twisted tape insert [3]. It is also determined that as the value of the Reynolds number increases the frictional factor decreases and as it decreases the frictional factor increases. Hence, for the low Reynolds number the value of the frictional factor is more gamey.

And it can also articulate that in the case of core fluid disturbance which is induced by the twisted tape the value of frictional factor is more down as compared to the surface disturbance in the fluid which is done in the sheath of the circular ring insert [7], or combines both core and surface disturbance which is insured in case of twisted tape with circular ring [4].

Table 1 Insert geometry and parameters used.

Reference no.	Insert type	Insert parameter	Result
[1]	 Twisted tape	• Twist ratios (y/D = 2, 2.5, 3, 3.5 and 4) • Clearance ratios (c/D = 0.0178 and 0.0357) • Reynolds number from 5132 to 24,989	The highest heat transfer enhancements are obtained at 1.756 for c/D = 0.0178, as 1.744 for c/D = 0.0357 and use 1.789 for the typical twisted tape (c/D = 0) at y/D = 2 of all twist ratios
[2]	 Double twisted tape	• Twist ratios (y = 1.95, 3.85, 5.92 and 7.75) • Reynolds number from 6950 to 50,050	In the range of the present work, heat transfer rate and friction factor were obtained to be around 60–240% and 91–286% higher than those of the plain tube values, respectively
[3]	 Triple twisted tape	• Twist ratios (y = 1.92, 2.88, 4.81 and 6.79) • Reynolds number from 7200 to 50,200	The Nusselt number and friction factor of using the triple twisted tape inserts were found to be increased up to 3.85 and 4.2 times and heat transfer performance was achieved to be 1.44 as compared to plain tube
[4]	 Twisted tape with circular ring	• Pitch ratios (l/D = 1.0, 1.5, and 2.0) • Twist ratios (y/W = 3, 4, and 5) • Reynolds number between 6000 and 20,000	The increases of mean Nusselt number, friction factor and thermal performance, in the tube equipped with combined devices, respectively, are 25.8%, 82.8% and 6.3% over those in the tube with the CRT alone and thermal performance factor of 1.42 is found
[5]	 Wire coil with twisted tape	• Reynolds number from 3000 to 18,000 • Coil pitch ratios (CR = 4, 6 and 8) • Twist ratios (Y = 4 and 6)	Highest heat transfer and performance of about 200–350%, leading to the more compact heat exchanger
[6]	 Conical ring with twisted tape	• Twist ratios, Y = 3.75 and 7.5 • Reynolds number 6000–26,000	A maximum heat transfer rate of 367% and enhancing efficiency of around 1.96 are found for using the conical-ring and the twisted-tape of Y = 3.75
[7]	 Coiled wire with triangular cross-section	• Pitch ratios (P/D = 1, 2 and 3) • Two different ratios of equilateral triangle length side to tube diameter (a/D = 0.0714 and 0.0892) • Reynolds number from 3500 to 27,000	The highest overall enhanced efficiency of 36.5% is achieved for the wire with a/D = 0.0892 and P/D = 1 at Reynolds number 3858

(*continued on next page*)

Table 1 (*continued*)

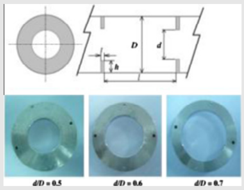
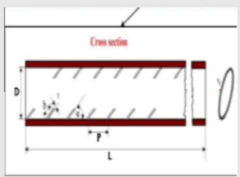
Reference no.	Insert type	Insert parameter	Result
[8]	 Tube with circular ring	• Diameter ratios (DR = d/D = 0.5, 0.6 and 0.7) • Pitch ratios (PR = p/D = 6, 8 and 12) • Reynolds number from 4000 to 20,000	Heat transfer enhancement around 57–195% compared to that in the plain tube. The maximum thermal performance factor of 1.07 is found by the use of the CRT with DR = 0.7 and PR = 6
[9]	 Circular ring at some angle	• Pitch ratios (PR = P/D = 0.5, 1.0, 1.5 and 2.0) • Width ratios (BR = b/D = 1/4 0.1, 0.15 and 0.2) • Reynolds number from 5000 to 26,000	The presence of the VRs at BR = 0.2, PR = 0.5 causes a much higher pressure drop increase, $f/f0$ = 35.1–36.5 but also provides a considerable heat transfer augmentation in the tube, Nu/Nu0 = 3.6–4.3

Table 2 Correlations obtained in case of different geometries.

Ref. no.	Correlations	Uncertainty in measurement (%)
	Nu: Nusselt number, **f**: frictional factor, **η**: thermal performance factor	
[1]	1. $\mathbf{Nu} = 0.406903\ \mathrm{Re}^{0.586556}\ (y/D)^{-0.443989}\ (c/D)^{-0.055072}\ \mathrm{Pr}^{0.38}$ 2. $\mathbf{f} = 6.544291\ \mathrm{Re}^{-0.452085}\ (y/D)^{-0.730772}\ (c/D)^{-0.1579}$ 3. $\boldsymbol{\eta} = 9.750184\ \mathrm{Re}^{-0.177983}\ (y/D)^{-0.183513}\ (c/D)^{-0.009558}\ \mathrm{Pr}^{0.38}$	±8.2 ±6.1
[2]	1. $\mathbf{Nu} = \{-0.0007y^3 + 0.0077y^2 - 0.0385y + 0.4777\}\ \mathrm{Re}^{\{0.0002y3-0.0021y2+0.0047y+0.5894\}}\ \mathrm{Pr}^{0.33}$ 2. $\mathbf{f} = \{-0.0009y^3 - 0.1015y^2 + 1.0842y + 8.685\}\ \mathrm{Re}^{\{-0.00004y3+0.0015y2-0.0165y-0.4722\}}$ 3. $\boldsymbol{\eta} = 41.176\mathrm{CC}_1^{-0.6802}\ \mathrm{Re}_p^{\{0.0002272y3-0.0031203y2+0.015923y-0.13103\}}$	±3.2 ±3.4
[3]	1. $\mathbf{Nu} = \{-0.0017y^3 + 0.0179y^2 - 0.0962y + 0.7734\}\ \mathrm{Re}^{\{0.00002y3+0.0013y2-0.0094y+0.5746\}}\ \mathrm{Pr}^{0.33}$ 2. $\mathbf{f} = \{-0.0388y^3 + 0.2484y^2 - 0.8462y + 17.685\}\ \mathrm{Re}^{\{0.00005y3+0.0017y2-0.0164y-0.5193\}}$ 3. $\boldsymbol{\eta} = 41.176\mathrm{CC}_1^{-0.6802}\ \mathrm{Re}_p^{\{-0.000014y3+0.000144y2+0.001755y-0.11379\}}$	±3.2 ±3.4
[4]	1. $\mathbf{Nu} = 0.326\ \mathrm{Re}^{0.724}\ \mathrm{Pr}^{0.4}\ (l/D)^{-0.475}\ (y/W)^{-0.406}$ 2. $\mathbf{f} = 13.99\ \mathrm{Re}^{-0.202}\ (l/D)^{-0.927}\ (y/W)^{-0.619}$ 3. $\boldsymbol{\eta} = 4.63\ \mathrm{Re}^{-0.111}\ (l/D)^{-0.166}(y/W)^{-0.199}$	±5 ±7
[5]	1. $\mathbf{Nu} = 4.47\ \mathrm{Re}^{0.5}\ \mathrm{Pr}^{0.4}\ \mathrm{CR}^{-0.382}\ Y^{-0.38}$ 2. $\mathbf{f} = 338.37\ \mathrm{Re}^{-0.367}\ \mathrm{CR}^{-0.887}\ Y^{-0.455}$ 3. $\boldsymbol{\eta} = (\mathrm{Nu}/\mathrm{Nu}_s)/((f/f_s)^{1/3})$	±7.6 ±9.5
[6]	1. $\mathbf{Nu} = 1.356\ \mathrm{Re}^{0.433}\ \mathrm{Pr}^{0.4}\ (d/D)^{-1.23}\ Y^{-0.053}$ 2. $\mathbf{f} = 24.87\ \mathrm{Re}^{-0.43}\ (d/D)^{-3.99}\ Y^{-0.16}$ 3. $\boldsymbol{\eta} = 14.9\ \mathrm{Re}^{-0.277}\ (d/D)^{-0.129}\ Y^{-0.01}$	±10 ±15
[7]	1. $\mathbf{Nu} = 0.598417\ \mathrm{Re}^{0.745064}\ (P/D)^{-0.268374}\ (a/D)^{0.813205}\ \mathrm{Pr}^{0.39}$ 2. $\mathbf{f} = 83.70924\ \mathrm{Re}^{-0.305268}\ (P/D)^{-0.388}\ (a/D)^{1.319018}$ 3. $\boldsymbol{\eta} = (\mathrm{Nu}/\mathrm{Nu}_s)/((f/f_s)^{1/3})$	±8.6 ±6.5
[8]	1. $\mathbf{Nu} = 0.354\ \mathrm{Re}^{0.697}\ \mathrm{Pr}^{0.4}\ \mathrm{DR}^{-0.555}\mathrm{PR}^{-0.598}$ 2. $\mathbf{f} = 0.715\ \mathrm{Re}^{-0.081}\ \mathrm{DR}^{-4.775}\ \mathrm{PR}^{-0.846}$ 3. $\boldsymbol{\eta} = 5.315\ \mathrm{Re}^{-0.078}\ \mathrm{DR}^{1.031}\ \mathrm{PR}^{-0.317}$	±7 ±10
[9]	1. $\mathbf{Nu} = 0.165\ \mathrm{Re}^{0.698}\ \mathrm{Pr}^{0.4}\ (\mathrm{BR}+1)^{3.063}\ (\mathrm{PR}+1)^{-0.549}$ 2. $\mathbf{f} = 1.709\ \mathrm{Re}^{-0.209}\ (\mathrm{BR}+1)^{10.753}\ (\mathrm{PR}+1)^{-1.433}$ 3. $\boldsymbol{\eta} = (\mathrm{Nu}/\mathrm{Nu}_s)/((f/f_s)^{1/3})$	±7.6 ±9.5

Table 3 Result showing peak values of correlations.

Ref. no.	Parameters at which the result gives the maximum value		
	Nu	**F**	**η**
[1]	Twist ratio $(y/D) = 2$ Clearance ratio $(c/D) = 0.0178$	Twist ratio $(y/D) = 2$ Clearance ratio $(c/D) = 0.0178$	Twist ratio $(y/D) = 2$ Clearance ratio $(c/D) = 0.0178$
[2]	Twist ratio $(y) = 1.95$	Twist ratio $(y) = 1.95$	Twist ratio $(y) = 1.95$
[3]	Twist ratio $(y) = 1.92$	Twist ratio $(y) = 1.92$	Twist ratio $(y) = 1.92$
[4]	Pitch ratio $(l/D) = 1.0$ TT with $y/W = 3$	Pitch ratio $(l/D) = 1.0$ TT with $y/W = 3$	Pitch ratio $(l/D) = 1.0$ TT with $y/W = 3$
[5]	CR = 4 and $Y = 4$	CR = 4 and $Y = 4$	CR = 4 and $Y = 4$
[6]	$Y = 3.75$ and $d/D = 0.5$	$Y = 3.75$ and $d/D = 0.5$	$Y = 3.75$ and $d/D = 0.7$
[7]	$a/D = 0.0892$ and $P/D = 1$	$a/D = 0.0892$ and $P/D = 1$	$a/D = 0.0892$ and $P/D = 1$
[8]	DR = 0.5 and PR = 4	DR = 0.5 and PR = 4	DR = 0.7 and PR = 4
[9]	BR = 0.2 and PR = 0.5	BR = 0.2 and PR = 0.5	BR = 0.1 and PR = 0.5

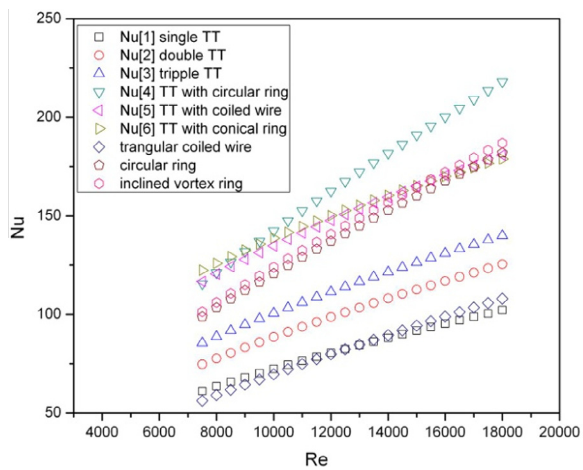


Figure 1a Nusselt no. (Nu) versus Reynolds no. (Re).

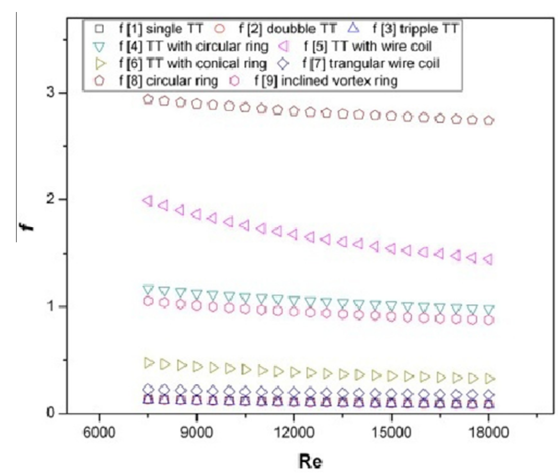


Figure 2a Friction factor (f) versus Reynolds no. (Re).

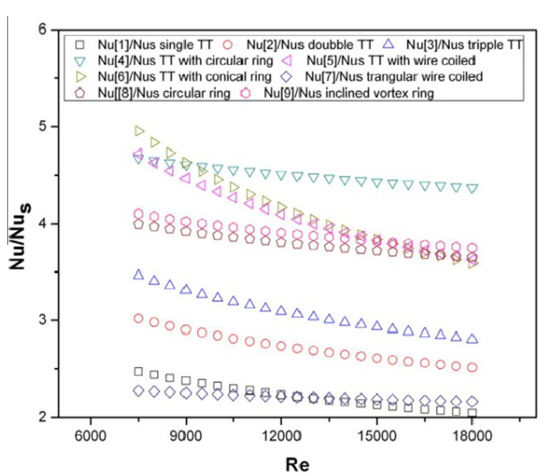


Figure 1b Nusselt no. (Nu/Nu_s) versus Reynolds no. (Re).

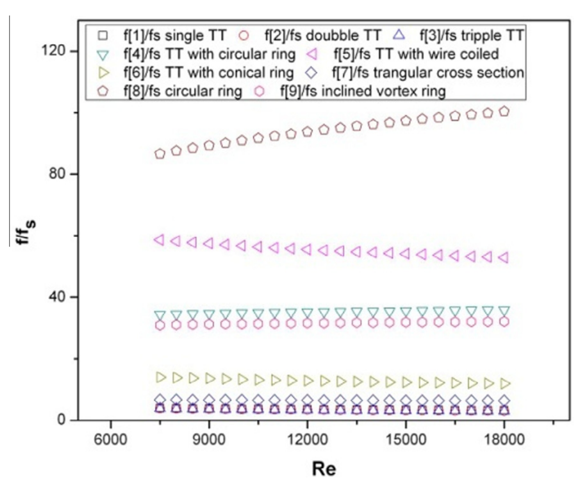


Figure 2b Friction factor (f/f_s) versus Reynolds no. (Re).

3.2. Effect of pitch ratio

3.2.1. Heat transfer

For all the investigated geometries it is observed that heat transfer is higher in the case of the lower pitch ratio. On the basis of observation and correlation graph obtained from the correlation which is presented in Figs. 1(a) and (b), it is seen that heat transfer is maximum in case of twisted tape with circular ring insert [4], and minimum in case of wire with triangular cross section [7]. In case of twisted tape the effect of pitch ratio is not seen and heat transfer is minimum in such event. It is also discovered that the heat transfer is higher for the higher

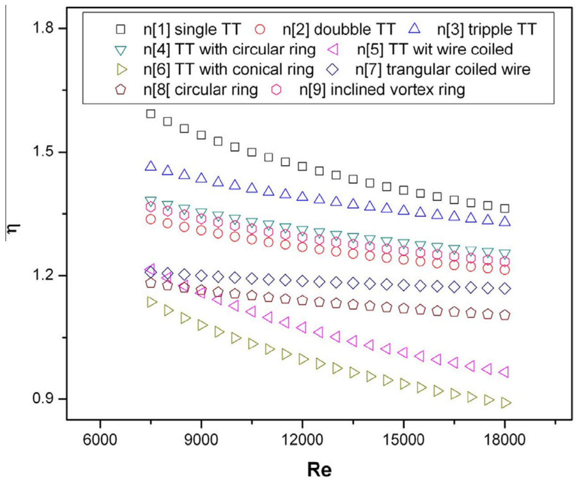


Figure 3 Thermal performance factor (η) versus the Reynolds number (Re).

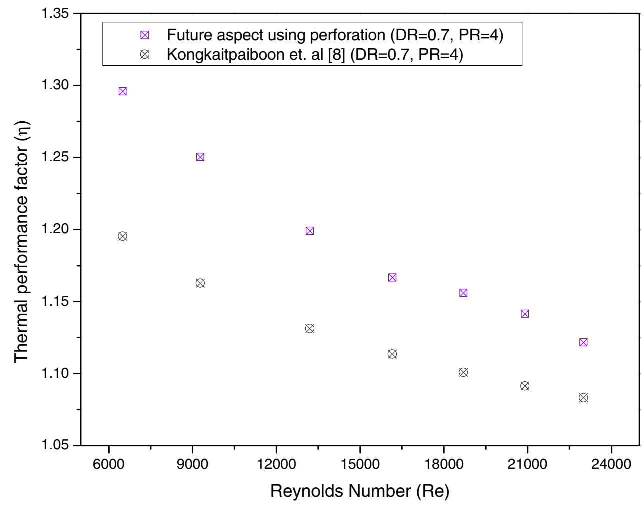


Figure 6 Effect of perforation on thermal performance factor as compared to CRT.

Reynolds number and lower in case of lower Reynolds number.

3.2.2. Friction factor

On the basis of observation and graph obtained from Figs. 2 (a) and (b), it is seen that frictional factor is higher in case of circular ring insert [8] and lower in case of coiled wire with triangular cross section [7]. It is also found that frictional factor increases with a reduction in the value of pitch ratio in each of the experimentations. It is also found that frictional factor is higher in case of lower Reynolds number and as the Reynolds number increases the friction factor also increases.

3.3. Thermal performance factor

This helps to compare the enhanced performance of the heat exchangers with respect to smooth tube heat exchanger. Comparing all the correlations with respect to their peak value in the graph obtained is presented in Fig. 3. On the basis of observation and graph obtained, it can be determined that the thermal performance factor is maximum in case of the single twisted tape insert [1] and minimum in case of twisted tape with conical ring. It is also found that at the low Reynolds number the value of the thermal performance factor is higher and as the Reynolds number increases the thermal performance factor decreases respectively. The thermal performance factor of a heat exchanger by the mathematical relation used is

$$\boldsymbol{\eta} = (\mathrm{Nu}/\mathrm{Nu}_s)/((f/f_s)^{1/3}) \tag{1}$$

4. Conclusion

On the basis of observation and graphs obtained, it can be concluded that in the above comparison the thermal performance factor in case of single twisted tape [1] is higher

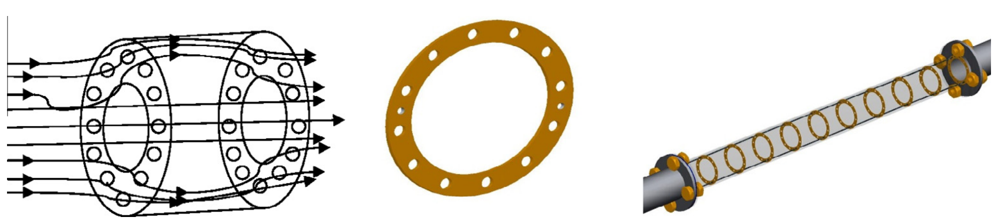


Figure 4 (a) Fluid flow behavior, (b) proposed insert geometry, and (c) heat exchanger with perforated disk insert.

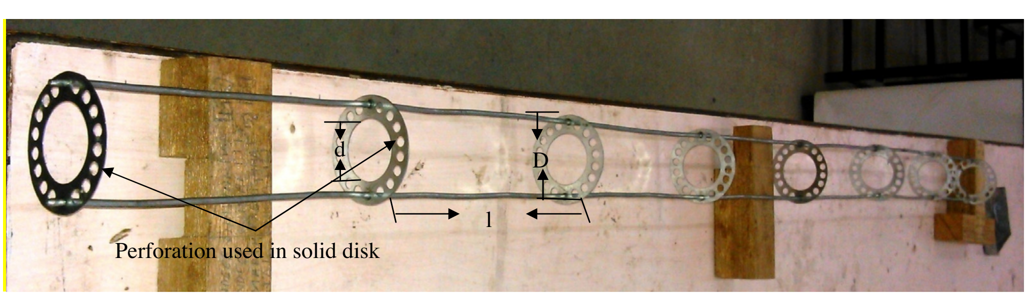


Figure 5 Perforated circular disk insert.

followed by triple twisted tape [3] and twisted tape with circular ring [4]. For the better thermal performance factor in a heat exchanger core disturbance and surface disturbance of fluid flow both play a significant role. In case of twisted tape with circular ring both the disturbances can be determined. But in case of twisted tape only core disturbance was observed. In case of circular ring insert [8] the friction factor can be seen maximum because of the surface disturbance in the fluid layer, while the maximum heat transfer can be seen in case of twisted tape with circular ring [4] in which both surface disturbance and core disturbance of the fluid can be seen. So, it is needed to investigate some new insert geometries which can have less flow blockages in order to control friction factor and improve thermal performance factor. At the same time there should be sufficient improvement in heat transfer rate.

5. Future aspects

For future aspect, if the similar work is carried out on ''perforated circular ring'' as insert geometry, then there can be a significant increase in the thermal performance factor of heat exchanger tube. This increase is predicted because of the perforation, which will significantly decrease the value of frictional factor because of the fluid layer detachment and reattachment and less flow blockages. The effect of perforation on thermal performance factor can easily be seen in Fig. 6, which shows comparative study of perforation for the same parameter and flow rate used by Kongkaitpaiboon et al. [8]. It is seen that on the implementation of perforation on circular disk, there is significant reduction in the friction factor, as there is decrease in the flow blockages. Detachments and reattachments of fluid streams can also be observed which causes proper fluid mixing, which finally improved thermal performance factor of heat exchanger tube. In Fig. 4, it is observed the flow behavior of the proposed model of perforated circular disk and its assembly used as an insert in the heat exchanger tube.

The photographic view of the insert model used for comparison experimentation is shown in Fig. 5, and the insert model comprises of the perforated circular disk.

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