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HEAT TRANSFER ENHANCEMENT USING DIFFERENT GEOMETRY OF TWIST TAPE TURBULATORS: A REVIEW

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Abstract- The development of high-performance thermal systems has increased interest in heat transfer enhancement techniques. The high thermal performance enhancement of heat exchanger systems is needed to use energy source efficiently due to the sky-rocketing prices of petroleum and coal fuels. Heat exchangers are widely used in industry both for cooling and heating. Insertion of turbulators in the flow passage is one of the favorable passive heat transfer augmentation techniques due to their advantages of easy fabrication, operation as well as low maintenance. The purpose of this review presents the effect of twisted tape turbulators on the heat transfer enhancement, pressure drop, flow friction and thermal performance factor characteristics in a heat exchanger tube. The twisted tape turbulator is a device for increasing the heat transfer rate in the heat exchanger system. The widely employed in several industrial and engineering applications of heat exchanger are automobile, refrigerators, solar collector, heat engine, air conditions, thermal power plant, electronic cooling, milk plant, chemical process industries etc. heat transfer enhancement using different type of the turbulators placed in the tube has been extensively studied for the past decade among the both passive and active technique are compile in this review.

Keywords*-Heat transfer enhancement, flow friction, pressure drop, twisted tape turbulator, swirl flow.*

I. INTRODUCTION

Heat exchanger is equipment which transfers the heat energy between two fluids that are at different temperature while keeping the mixing with each other. Heat exchanger is widely used in several industries. Cost of energy, cost of materials and heat transfer enhancement is a subject of much interest to work on technique of how to increase rate of heat transfer and gain higher thermal efficiency, but in the process pumping power increases ultimately, the pumping cost become high.

Heat transfer enhancement techniques play a vital role for laminar flow heat transfer since the heat transfer coefficients are generally low for laminar flow in plain tubes [5,9,31,32,42]. The heat transfer rate can be improved by introducing a disturbance in the fluid flow, which can be achieved by with the twisted tape/turbulator insert in circular tube.

Insertion of turbulators in the flow passage is one of the favourable Passive heat transfer augmentation techniques due to their advantages of easy fabrication, operation as well as low maintenance [13]. In general, the performance of turbulators strongly depends on their geometries. In earlier investigations, turbulators with several shapes were utilized to promote heat transfer. The inserts [3] studied included coil wire inserts, brush inserts, mesh inserts, strip inserts, twisted tape inserts etc.

Augmentation of convective heat transfer in internal flows with tape inserts in tubes is a well-acclaimed technique employed in industrial practices. Heat transfer enhancement or augmentation techniques refer to the improvement of thermohydraulic performance of heat exchangers. Existing enhancement techniques can be broadly classified into three different categories:

1. Passive Techniques
2. Active Techniques
3. Compound Techniques.

A. Passive Techniques:
These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour (except for extended surfaces) which also leads to increase in the pressure drop[1].

In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques [2] hold the advantage over the active techniques as they do not require any direct input of external power. Heat transfer augmentation by these techniques can be achieved by using:

1) Treated Surfaces: This technique involves using pits, cavities or scratches like alteration in the surfaces of the heat transfer area which may be continuous or discontinuous. They are primarily used for boiling and condensing duties.

2) Rough surfaces: These surface modifications particularly create the disturbance in the viscous sub-layer region. These techniques are applicable primarily in single phase turbulent flows.

3) Extended surfaces: Plain fins are one of the earliest types of extended surfaces used extensively in many heat exchangers. Finned surfaces have become very

popular now a day owing to their ability to disturb the flow field apart from increasing heat transfer area.

4) Displaced enhancement devices: These inserts are used primarily in confined forced convection [19]. They improve heat transfer indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.

5) Swirl flow devices: They produce swirl flow or secondary circulation [7, 30] on the axial flow in a channel. Helical twisted tape, twisted ducts & various forms of altered (tangential to axial direction) are common examples of swirl flow devices. They can be used for both single phase and two-phase flows.

6) Coiled tubes: [7,13,22,36] In these devices secondary flows or vortices are generated due to curvature of the coils which promotes higher heat transfer coefficient in single phase flows and in most regions of boiling.

This leads to relatively more compact heat exchangers.

7) Surface tension devices: These devices direct and improve the flow of liquid to boiling surfaces and from condensing surfaces. Examples include wicking or grooved surfaces.

8) Additives for liquids: This technique involves addition of solid particles, soluble trace additives and gas bubbles added to the liquids to reduce the drag resistance in case of single phase flows. In case of boiling systems, trace additives are added to reduce the surface tension of the liquids.

B. Active Techniques:

These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement [6] in the rate of heat transfer.

It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. Various active techniques are as follows:

1) Mechanical Aids: Examples of the mechanical aids include rotating tube exchangers and scrapped surface heat and mass exchangers. These devices stir the fluid by mechanical means or by rotating the surface.

2) Surface vibration: They have been used primarily in single phase flows. A low or high frequency is applied to facilitate the surface vibrations which results in higher convective heat transfer coefficients.

3) Fluid vibration: Instead of applying vibrations to the surface, pulsations are created in the fluid itself. This kind of vibration enhancement technique is employed for single phase flows.

4) Electrostatic fields: Electrostatic field like electric or magnetic fields or a combination of the two from DC or AC sources is applied in heat exchanger systems which induce greater bulk mixing, force convection or electromagnetic pumping to enhance heat transfer. This technique is applicable in heat transfer process involving dielectric fluids.

5) Injection: [16] In this technique, same or other fluid is injected into the main bulk fluid through a porous heat transfer interface or upstream of the heat transfer section. This technique is used for single phase heat transfer process.

6) Suction: This technique is used for both two phase heat transfer and single phase heat transfer process. Two phase nucleate boiling involves the vapour removal through a porous heated surface whereas in single phase flows fluid is withdrawn through the porous heated surface.

7) Jet impingement: This technique is applicable for both two phase and single phase heat transfer processes. In this method, fluid is heated or cooled perpendicularly or obliquely to the heat transfer surface.

C. Compound techniques:

A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

Different Types of Twisted Tape:

Twisted tapes are the metallic strips twisted with some suitable techniques with desired shape and dimension, inserted in the flow. Following are the main categories of twisted tape which are analysed.

a) Full length twisted tape

b) Varying length twisted tape

c) Regularly spaced twisted tapes

d) Tape with attached baffles

e) Slotted tapes and tapes with holes

f) Tapes with different surface modifications.

LIST OF ABBREVIATIONS

CRT	circular-ring turbulator
DR=d/D	diameter ratios
PR=p/D	pitch ratios
PCR	perforated conical-rings
N	numbers of perforated holes
PR=p/D	pitch ratios
CR	conical-ring
CITSG	conical injector type swirl generator
Ah	cross-section areas
Nux	local Nusselt number
NuER	heat transfer enhancement ratio
NuPR	heat transfer performance ratio
β	director angle
α	CITSG angle
Y	twist ratios
S-DWT	straight delta-winglet twisted tape
y/w	twist ratios
DR= d/w	wing cut ratios
TT	twisted tape
O-DWT	Oblique delta-winglet twisted tape
θ	blade angle of propeller
D_k/D_b	ration of diameter to pipe diameter
CR	coil pitch ratios
Y	twist ratios
X/D	tail length ratio
p/D	pitch ratio
W/D	width ratio
TA	alternate-axis
δ	thickness
Re	Reynolds numbers
TR	twisted-rings
Nu	convective heat transfer
f	friction factor
η	thermal performance factor
ω	energy dissipation in unit volume [1/s]
k	wave number [1/m]
ε	average turbulence dissipation rate [m2/s3]
LES	Large Eddy Simulations
SPH	Smoothed Particle Hydrodynamics
TTP	twisted tape with pins
TTPB	twisted tape with pins bonded

II. REVIEW OF WORK CARRIED OUT

C.B. Sobhan et al. [1] experimentally investigated on a 1-2 shell and tube heat exchanger, to study the spiral turbulators on its performance. Experiments were conducted with various winding wire diameters and pitches and the heat transfer coefficient were evaluated for a wide range of temperature levels and flow rates of the shell side fluid. Nusselt number values for the various winding pitches were used. It is clearly understood that the use of the turbulators have considerable effect on the performance enhancement in the range the Reynolds numbers studied, which was in the laminar regime; improvement as high as 70% when compared to the bare tube has been noticed in the outside Nusselt number. The various pitches used, the best performance was observed for the case with 4 cm winding pitch, indicating the existence of an optimum winding pitch near this value. It was found that an increase in the wire diameter from 1.34 mm to 1.65 mm improves the overall Nusselt number to some extent, that is, the larger diameter wire effects slightly better heat transfer coefficient.

C. Yildiz et al. [2] were placed twisted narrow, thin metallic strips in the inner pipe of a concentric double-pipe heat exchanger and studied their effect on heat transfer and pressure drop for parallel and countercurrent flow. These turbulators were prepared by twisting the strips through certain angles and designed to touch the inside wall at each step. In the system, hot air was passed through the inner pipe, while cold water was flowing through the annulus. The experiments were performed with an empty inner pipe at Re number between 3400 and 6900. The effect of the turbulators on the heat transfer is more pronounced for high Re numbers. The improvements for parallel flow show a parallel trend and are only 10% lower than those for counter current flow. The increase is about 1.3 times that of the empty tube at the highest Re number for 170 mm pitch size.

Kenan Yakut et al. [3] investigated flow-induced vibration characteristics of conical-ring turbulators for heat transfer enhancement in heat exchangers experimentally. The conical- rings, having 10, 20 and 30 mm pitches, were inserted in a model pipe-line through which air was passed as the working fluid. It was observed that as the pitch increases, vortex shedding frequencies also increased and the maximum amplitudes of the vortices produced by conical-ring turbulators occur with small pitches. In addition, the effects of the promoters on the heat transfer and friction factor were investigated experimentally for all the arrangements. It was found that the Nusselt number increased with the increasing Reynolds number and the maximum heat transfer was obtained for the smallest pitch arrangement.

The pressure loss was much higher along a unit experiment element because there is an increase in the friction surfaces of these turbulators that also behave like a sequential loss element and work like a diffuser with respect to the arrangement positions.

V. Kongkaitpaiboon et al. [4] performed an experimental investigation of convective heat transfer and pressure loss in a round tube fitted with circular-ring turbulators. They studied the effect of the circular-ring turbulator (CRT) on the heat transfer and fluid friction characteristics in a heat exchanger tube. The experiments were conducted by insertion of CRTs with various geometries, including three different diameter ratios (DR=d/D=0.5, 0.6 and 0.7) and three different pitch ratios (PR=p/D=6, 8 and 12). During the test air at 27 °C was passed through the test tube which was controlled under uniform wall heat flux condition. The Reynolds number was varied from 4000 to 20,000. According to the experimental results, heat transfer rates in the tube fitted with CRTs were augmented around 57% to 195% compared to that in the plain tube, depending upon operating conditions. Influence of the diameter ratio (DR) and pitch ratio (PR) on the heat transfer rate, friction factor and thermal performance factor behaviors was investigated under uniform wall heat flux condition. The CRTs with different diameter ratios (DR=d/D=0.5, 0.6 and 0.7) and pitch ratios (6, 8 and 12) were employed for the Reynolds number ranged between 4000 and 20,000. Over the entire range investigated CRTs propose heat transfer enhancement around 57% to 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.

V. Kongkaitpaiboon et al. [5] reported an experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings. They have been investigated the influences of the PCR on the turbulent convective heat transfer (Nu), friction factor (f) and thermal performance factor (η) characteristics experimentally. The perforated conical-rings (PCRs) used were of three different pitch ratios (PR=p/D=4, 6 and 12) and three different numbers of perforated holes (N=4, 6 and 8 holes). The experiment conducted in the range of Reynolds number between 4000 and 20,000, under uniform wall heat flux condition and using air as the testing fluid. It was found that the PCR considerably diminishes the development of thermal boundary layer, leading to the heat transfer rate up to about 137% over that in the plain tube. Evidently, the PCRs can enhance heat transfer more efficient than the typical CR on the basis of thermal performance factor of around 0.92 at the same pumping power. Over the range investigated, the maximum thermal performance factor of around 0.92 was found at PR=4 and N=8 holes with Reynolds number of 4000. The effects of the pitch ratio (PR) and number of perforated hole (N) on the heat transfer enhancement in a tube are also considered. The concluding remarks can be described as follows:

(1) At the similar test conditions, the PCRs offers lower heat transfer enhancement than the CRs. However, they generate friction factor only around 25% of that produced by the PCRs.

(2) The heat transfer rate and friction factor of PCRs increase with decreasing pitch ratio (PR) and decreasing number of perforated hole (N).

(3) The mean heat transfer rates obtained from using the PCR with PR=4, 6, and 12 are found to be respectively, 185%, 140%, and 86%, over the plain tube. Over the range investigated, the maximum thermal performance factor of around 0.92 is found at PR=4 and N=8 holes with the Reynolds number of 4000.

Aydın Durmus [6] reported heat transfer and exergy loss in cut out conical turbulators. He investigated the effect of cut out conical turbulators, placed in a heat exchanger tube at constant outer surface temperature, on the heat transfer rates. The air was passed through the exchanger tube, the outer surface of which was heated with saturated water vapor. The experiments were conducted for air flow rates in the range of 15,000 6 Re <60,000. Heat transfer, pressure loss and exergy analyses were made for the conditions with and without turbulators and compared to each other. However, since the turbulators were placed directly in the flow area, they cause pressure losses. Therefore, an optimization should be made for pressure losses because pressure losses cause higher pumping power. Thus, future work may be undertaken on the detailed effect of the number of turbulators and on applications in flue and smoke tube boilers involving very hot gases.

Irfan Kurtbas et al. [7] devised a novel conical injector type swirl generator (CITSG). Performances of heat transfer and pressure drop in a pipe with the CITSG were experimentally examined for the CITSGs' angle (α) of 30°, 45° and 60° in Reynolds number (Re) range of 10,000–35,000. Moreover, circular holes with different numbers (N) and cross-section areas (Ah) were drilled on the CITSG. All experiments were conducted with air accordingly; Prandtl number was approximately fixed at 0.71. The local Nusselt numbers (Nux), heat transfer enhancement ratio (NuER) and heat transfer performance ratio (NuPR) were also calculated. It was found that the NuER decreases with increase in Reynolds number, the director angle (β), the director diameter (d), and with decrease in the CITSG angle (α). Likewise, variation of NuPR and NuER was also essentially similar for the same independent parameters.

P. Promvonge and S. Eiamsa-ard [8] have been investigated heat transfer, friction factor and enhancement efficiency characteristics in a circular tube fitted with conical-ring turbulators and a twisted-tape swirl generator experimentally. Air as the tested fluid was passed both enhancement devices in a Reynolds number range of 6000 to 26,000. Two twisted-tapes of different twist ratios, Y=3.75, and 7.5, were introduced in each run. The experimental results reveal that the tube fitted with the conical-ring and twisted-tape provides Nusselt number values of around 4 to 10% and enhancement efficiency of 4 to

8% higher than that with the conical-ring alone. A maximum heat transfer rate of 367% and enhancement efficiency of around 1.96 was found for using the conical-ring and the twisted-tape of Y=3.75. For all the devices used, the enhancement efficiency tends to decrease with the rise of Reynolds number and to be nearly uniform for Reynolds number over 16,000. It was found that the smaller twist ratio was, the larger the heat transfer and friction factor for all Reynolds numbers. The average heat transfer rates from using both the conical-ring and twisted-tape for Y=3.75, and 7.5, respectively, were found to be 367% and 350% over the plain tube. However, the friction factor from using both devices also increases considerably.

S. Eiamsa-ard et al. [9] have been investigated heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with deltawinglet twisted tape, using water as working fluid experimentally. Influences of the oblique delta-winglet twisted tape (O-DWT) and straight delta-winglet twisted tape (S-DWT) arrangements were also described. The experiments were conducted using the tapes with three twist ratios (y/w = 3, 4 and 5) and three depth of wing cut ratios (DR = d/w = 0.11, 0.21 and 0.32) over a Reynolds number range of 3000–27,000 in a uniform wall heat flux tube. The obtained results show that mean Nusselt number and mean friction factor in the tube with the delta-winglet twisted tape increase with decreasing twisted ratio (y/w) and increasing depth of wing cut ratio (DR). It was also observed that the O-DWT was more effective turbulator giving higher heat transfer coefficient than the S-DWT. Over the range considered, Nusselt number, friction factor and thermal performance factor in a tube with the O-DWT were, respectively, 1.04– 1.64, 1.09–1.95, and 1.05–1.13 times of those in the tube with typical twisted tape (TT). Empirical correlations for predicting Nusselt number and friction factor have been employed. The predicted data were within ±10% for Nusselt number and ±10% for friction factor.

A. Durmus et al. [10] investigated the effect of propeller type turbulators which were located in the laminar pipe of co axial heat exchanger. The blade angle of propeller of turbulator between $10 \leq \theta \leq 40$ the ration of propeller diameter to pipe diameter between $0.87 \leq D_k / D_b \leq 0.94$ and Reynolds number in the range of 10000and 30000.The turbulator place with 10 cm array increased the heat transfer as much as 28% and 39% the pressure loss as much as 17% and 43% compared with the turbulator placed in the arrays of 20 and 30cm. The turbulator with blade angle of 20^0 decrease heat transfer as far as 8%, the ones with a blade angle of 40^0, on the other hand, as far as 35% compared with the turbulators with a blade angle of 10^0. The change of blade angle affected the pressure loss between 15-40%. In the experiment was seen that heat transfer 2-4 times and pressure loses 8.5 times biggest than the values of the empty pipe heat exchanger at Reynolds number of 10000-30000 and different mass flow rate.

Anil Singh Yadav [11] has been studied the influences of the half length twisted tape insertion on heat transfer and pressure drop characteristics in a U-bend double pipe heat exchanger experimentally. In the experiments, the swirling flow was introduced by using half-length twisted tape placed inside the inner test tube of the heat exchanger. The experimental results revealed that the increase in heat transfer rate of the twisted-tape inserts was found to be strongly influenced by tape-induced swirl or vortex motion. The heat transfer coefficient was found to increase by 40% with half-length twisted tape inserts when compared with plain heat exchanger. It was also observed that the thermal performance of Plain heat exchanger was better than half length twisted tape by 1.3-1.5 times.

Piroz Zamankhan [12] has been developed a 3D mathematical model to investigate the heat transfer augmentation in a circular tube with a helical turbulator. Glycol–water blends of various concentrations were used in the inner tube, and pure water was used in the outer tube. Changes in fluid physical properties with temperature were taken into account, and k-ε; k-ω, and large eddy simulations were developed for turbulence modeling. The simulation results showed a nonlinear variation in Reynolds and Prandtl numbers for a long model of a heat exchanger even in the absence of a turbulator. The presence of the turbulator was found to increase the heat transfer, sometimes without inducing turbulence, but also increased the pressure drop. Comparing their numerical results with experimental results, it was found that the LES model predicts the behavior of real systems. Using a multi-objective optimization method such as Genetic Algorithm coupled with the GPU-LES-SPH (which is an implementation of Smoothed Particle Hydrodynamics), a set of solutions will be obtained that satisfy different levels of compromise. From this set the most suitable solution will be selected. This efficiency-enhancing tool developed will be particularly suitable for process intensification (i.e., increasing production capacity per unit volume of equipment) in the chemical process industry.

Pongjet Promvonge [13] has been presented the experimental results on heat and flow friction characteristics in a uniform heat flux tube equipped with the 5 mm wire coil of three different coil pitch ratios (CR = 4, 6 and 8) and the twisted tapes of 2 twist ratios (Y = 4 and 6) are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for the twisted tape and coiled wire with the three ratios.

It was worth noting that the combined wire coil and the twisted tape with CR:Y = 4:4 provides the highest heat transfer rate than those with higher CR:Y values for all Reynolds numbers. The increase in Nusselt number value was in a range of 300–685% over the smooth tube. The friction factor for CR:Y = 4:4 was found to be higher twice of that for CR:Y = 6:4 or 8:4 values while the friction factor for CR:Y = 4:6 was seen to be the mean value between those for CR:Y = 4:4 and 6:4. This means that friction loss mostly comes from using the wire coil with CR = 4. The mean increase in friction factor range of 30 to 68 times over the smooth tube. The heat transfer obtained from both the wire coil and the twisted tape inserts was around 130–250% and 180– 400% depending on the Reynolds number, CR and Y values. The increase in Nusselt number for twisted tape alone was about 20–50% while in wire coil alone was around 100–110%. The use of combined twisted tape and the wire coil inserts also shows a higher heat transfer rate around 150–300%. The friction factor value for both was around 2–4.5 times above that for the wire coil alone or about 6–12.5 times over that for the twisted tape.

Smith Eiamsa-ard and Pongjet Promvonge [14] have been conducted the experiments to investigate the heat transfer and friction factor characteristics of the fully developed turbulent airflow through a uniform heat flux tube fitted with diamond-shaped turbulators in tandem arrangements. Nusselt numbers along the tube fitted with the D-shape turbulator for cone angles $\theta=45^0$ and tail length ratio TR = 1.0, for Reynolds numbers ranging from 3568 to 16,228. Nusselt number was high at the entry region ($7 \leq X/D \leq 22$) and gradually decreased but at exit region ($X/D > 22$) slightly increases. Heat transfer variation in terms of Nusselt number of the inserted tube for different the included cone angle ($\theta=15^0$, 30^0 and 45^0) with Reynolds numbers at TR = 1.0, 1.5 and 2.0, respectively. The larger angle (θ) yields a higher heat transfer rate than the smaller angle. The use of smaller the cone angle of $\theta = 15^0$ and 30^0 leads to the decrease in Nusselt number at 16.3% and 7.4%, respectively, in comparison with $\theta = 45^0$ of all range Reynolds numbers studied. The friction factor variations of using the three included cone angles ($\theta = 15^0$, 30^0 and 45^0) with Reynolds number between 3500 and 16,500 for TR = 1.0, 1.5 and 2.0, respectively. The maximum increase in friction factor was seen at TR = 1.0 which was higher than TR =1.5 and 2.0 around 11.3% and 22.6%, respectively. The improvement of average heat transfer rate was respectively 32.1%, 46.5% and 57.8% higher than those the plain tube while the friction factor was 4.7, 5.25 and 5.67 times of the plain tube.

Panida Seemawute and Smith Eiamsa-Ard [15] have been conducted the experiments for heat transfer in heat exchanger tubes by means of TRs compared to that of CRs at different width and pitch ratio has been investigated for Reynolds number between 6000 and 20,000. At the same width ratio (W/D=0.15) and a given pitch ratio, only TRs with the smallest pitch ratio (p/D) of 1.0 give higher Nusselt numbers than the CRs by around 3 to 4%. TRs and CRs at p/D=1.5 yield comparable Nusselt numbers for the whole range tested. At p/D=2.0, Nusselt numbers attributed to the TRs become lower than those associated by the CRs. The Nusselt number associated by the TR at the largest width ratio (W/D=0.15), are augmented by around 35.7% and 60% over those associated by ones with W/D=0.10 and 0.05, respectively. . By the utilization of TR at the smallest pitch ratio (p/D) of 1.0, an average Nusselt number was found to be around 6.8%and 13.6% higher than those achieved by the use of the ones of larger pitch ratios of 1.5 and 2.0 respectively. Thermal performance factor results Among the TRs tested, it was found that a thermal performance factor increased with decreasing width ratio.

The TRs with W/D=0.05 at p/D=1.0, 1.5 and 2.0 give thermal performance factor in the ranges of 1.02 to 1.24, 1.01 to 1.2, and 1.00 to 1.2 respectively.

Panida Seemawute et al. [16] has been investigated comparatively visualization of flow characteristics induced by twisted tape consisting of alternate-axis (TA) to that induced by typical twisted tape (TT). The visualization was carried out via a dye injection technique. The TAs were made of aluminium strips (for heat transfer setup) and acrylic sheet (for visualization setup) with thickness of 1.0 mm (δ), width of 18 mm and length of 1000 mm. Straight tapes were prepared at three desired twist lengths in 180° rotation (y/W = 3, 4 and 5) by twisting straight tapes, about their longitudinal axis, while being held under tension. The heating test tube was made of copper with thickness of 1.5 mm, inner diameter of 19 mm and length of 1000 mm. A common swirl flow was induced by TTs. with TAs, fluid stream which directly encounters a crosswise edge of the tape at an alternate point, was separated. Effect of TAs at various twist ratios, y/W = 3, 4 and 5 on the Nusselt number. This result was corresponded to the superior chaotic mixing as described above, and consequently results in more violent interruption on thermal boundary layer, and thus more efficient heat transfer though the tube wall.

P. K. Nagarajan et al. [17] have been presented experimental investigation of heat transfer and friction factor characteristics of solar trough collector fitted with full length twisted tapes inserts of twist ratio 6, 8 and 10. The transitional flow regime was selected for this study with the Reynolds number range 1192 to 2534. Friction factor decreased with increase in Reynolds number for a given twist ratio. At Reynolds number 2000 we can able to see the transition where there is a steady friction factor value. The swirl flow caused by the twisted tape effectively increases the heat transfer. The Nusselt Number was increased with decreasing twist ratio. The experimental data obtained were compared with those obtained from plain tube published data. The use of twisted inserts improved the performance of solar collector. The cost involved for manufacturing and inserting twisted tape was very minimal compared to energy efficiency improvement provided by these inserts.

S. Naga Sarada et al. [18] presented the study an experimental investigation of the potential of reduced width twisted tape inserts to enhance the rate of heat transfer in a horizontal circular tube with inside diameter 27.5mm with air as working fluid. The twisted tapes were of three different twist ratios (3, 4 and 5) each with five different widths (26-full width, 22, 18, 14 and 10 mm) respectively. The Reynolds number varied from 6000 to 13500. The percentage increase in Nusselt numbers for reduced width tapes compared to plain tube were about 11–22%, 16–31%, 24–34% and 39–44% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =3.

For full width tapes, the percentage increase was observed to be 58 to 70% compared to plain tube. The percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 5–12%, 9–22%, 13–30% and 23–36% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =4. For full width tapes, the percentage increase was observed to be 36 to 42% compared to plain tube. The percentage increase in Nusselt numbers for reduced width tapes compared to plain tube were about 2–8%, 6– 12%, 9–19% and 14–27% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =5. For full width tapes, the percentage increase was observed to be 22 to 37% compared to plain tube. The overall enhancement ratio of the tubes with full width twisted tape inserts was 1.62 for full width–26mm and 1.39 for reduced width-22mm twisted tape insert.

S.K. Saha et al. [19] have been investigated heat transfer and pressure drop characteristic in a circular tube fitted with regularly spaced twisted tape elements experimentally. For small cases the friction factor is 5-10% less and heat transfer by 20-40%. The local Nusselt number remains almost constant in both the tape and annular sections. For the small diameter negative axial velocity were not predicted even at first axial step in the tape section and thus the swirl was not supported.

The friction factor with the Reynolds number for varied y and s and for each case. The nusselt number decreases for w = 0.727 compared to that for w = 0.909 for all twist ratio and space ratio. For y = 5 and s = 5, the reduction was 25-32%; for y = 5 and s = 2.5, the reduction was 25-45%; for y = 2.5 and s = 5, the reduction was 22-45% and for y = 2.5 and s = 2.5, the reduction was 13-43%. The effect of the phase-angle between two successive tape element on the friction factor and nusselt number increases were 8% and 15% respectively. The variation isothermal and heated friction factor with Reynolds number were 7% and 6%.

S. Eiamsa-ard and P. Promvonge [20] have been conducted an experimental study to investigate heat transfer and friction loss behaviors in a circular wavy surfaced tube with a helical-tape insert using hot air as the test fluid. In general, the average heat transfer rate for the wavy-surfaced tube with the helical tape was found to be 23 to 35% better than that for the wavy-surfaced tube alone. The corresponding increase in the mean Nusselt number of the wavy-surfaced tube with the helical tape was about 330% to 422% over the plain tube. For the wavy surfaced tube with the helical tape, the increase in friction factor was found to be around 50% above one without the tape the average heat transfer rate for employing the tape with rod was found to be 8 to 12% better than that for one without core-rod. Thus, for the tape without rod, the friction factor could be reduced around 50% below one with core-rod. Results of the present correlations reasonably agree well within ±10% in comparison with experimental data for the friction factor, and within ±10% for the Nusselt number. The maximum increase in heat transfer rate and friction factor were found to be about 4.2 and 110 times the plain tube for the flow range studied.

M.R. Salimpour and S. Yarmohammadi [21] have been conducted an experimental investigation to find the influence of twisted tape inserts on the pressure drop during forced convective condensation of R-404A vapor in a horizontal tube. The tube set 5 with twist ratio of 4 has the highest pressure drop. Reduction in twist ratio induces higher turbulence intensity in liquid film and vapor core.

The condensing pressure drop for tube set 5 is up to 239% more than that for plain tube at refrigerant mass velocity 106.8 kg$(m^2\ s)^{-1}$. Tube set 2 has the lowest range of pressure drop increment which increases pressure drop up to 151% compared to the plain tube for refrigerant mass velocity 89 kg $(m^2\ s)^{-1}$. This correlation predicts experimental data with an error range of ±20%.

Shashank S. Choudhari and S.G. Taji [22] have been studied the experimental investigation of the heat transfer and friction factor characteristics of a double pipe heat exchanger fitted with coil wire insert made up of three different material as copper, aluminum and stainless steel and different pitches for Reynolds number in range of 4000-13000. Cu tube causes higher heat transfer enhancement about 1.58, and aluminum and stainless steel causes heat transfer rate enhancement up to 1.41 and 1.31 respectively. Overall heat transfer coefficient was higher for copper coil wire insert than aluminum, stainless steel inserts and plain tube. The friction factor of aluminum coil wire insert of 5 mm pitch was 5.4 to 6.7 times of the plane tube. Stainless steel tube insert causes friction factor of 4.8 to 5.9 times to plane tube and copper insert has friction factor of 4.3 to 5.4 times plane tube.

S. Selvam et al. [23] carried on experiments for different twist pitch to the width of the twisted tape ratios (y/w) for TTP and TTPB. The variation of Nusselt number with Reynolds number for the tube fitted with TTP of three different y/w ratios (3.33, 4.29, and 5.71). The experimental results it was seen that the smaller y/w (3.33) yields the higher values of heat transfer of about 23.86% than plain tube. Similarly for y/w=4.29 and 5.71 the enhancement were 19.9% and 14.4% respectively. It can be seen that the friction factor for y/w=4.29 and 5.71 were less when compared with y/w=3.33.This is due to less contact surface area of the turbulator. The empirical correlations developed relating pitch and Reynolds number were matching with the experimental data within ±7.28, and ±7.16% for Nusselt number and friction factor respectively.

CONCLUSION

Heat transfer enhancement is a subject of many interests to research in focusing on technique of how to increase heat transfer rate and achieve higher efficiency. Heat transfer rate increase due to insertion of twisted tape turbulator in a circular tube is one of the most effective approach. Turbulator is a device that turns laminar flow into a turbulant flow.

Turbulant flow can be desired on parts of the surface of an aircraft wing (air foil) or in industrial application such as heat exchanger and the mixing of fluids. The inserted twisted tape generates swirling flow and increase turbulance intensity which is major influencing factor for heat transfer enhancement.

Variation of Nusselt number with Reynold's number in the tube fitted with typical twisted tape. Nusselt number increase with increasing reynold number. This attributed to the increase of heat convection and also swirl intensity.

The nusselt number, reynold number, prandtl number, pressure drop and friction factor also depend on the geometrics of the twisted tape with different twist ratio, pitch ratio, tape width, space ratio, phase angle ,wire diameter etc. Heat transfer rate with typical twisted tape is higher than that of plain tube.

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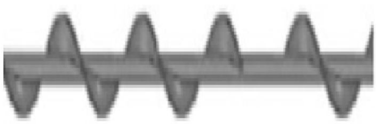
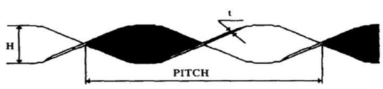
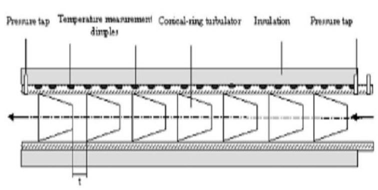
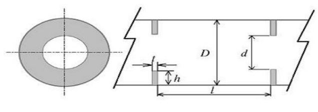
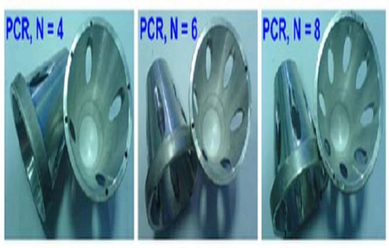
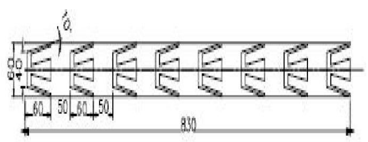
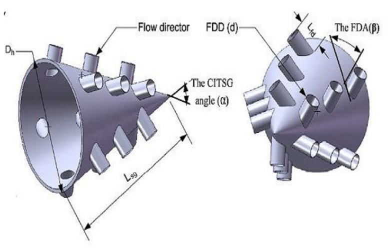
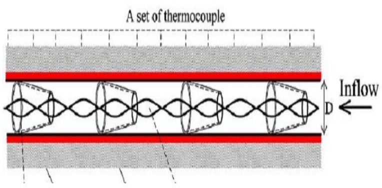
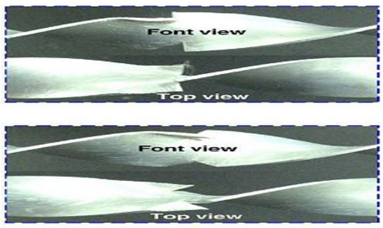
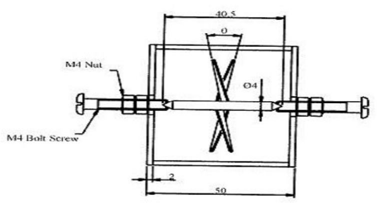
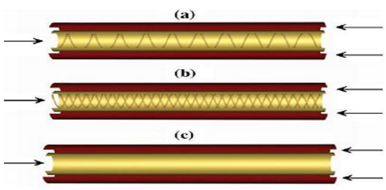
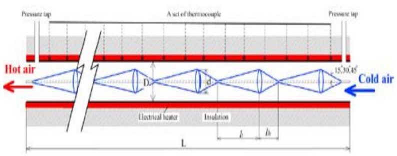
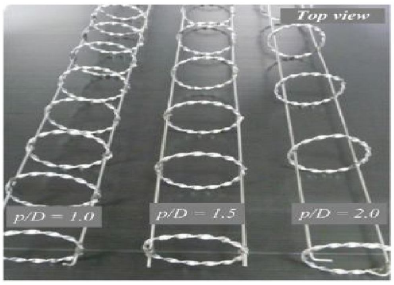
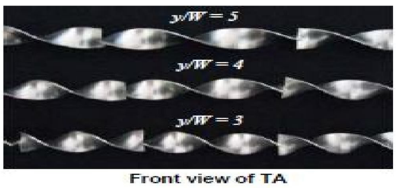
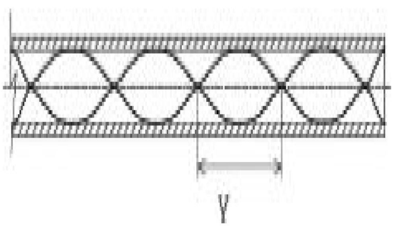
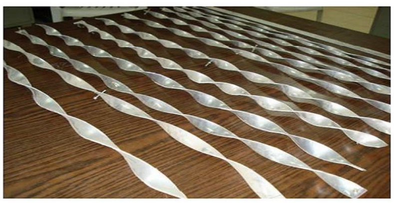
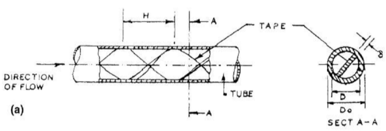
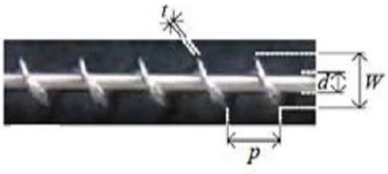
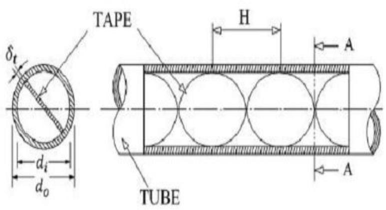
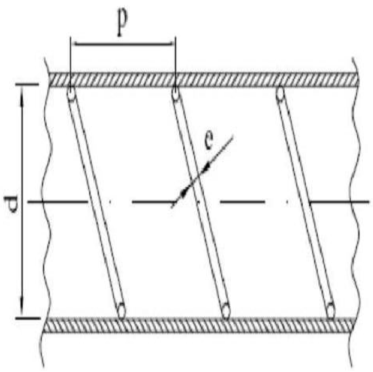
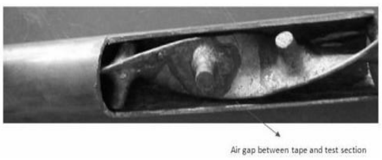
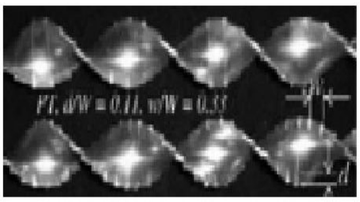
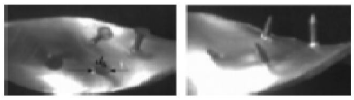
TABLE 1: CONFIGURATION SKETCHES OF VARIOUS TWISTED TAPES

Geometry	Name	Authors Name	Conditions	Principal finding
	Spiral Turbulator	C.B. Sobhan *et al.*[1]	Pitch=2.5,4.0,5.5 ,7.0	➢ Heat exchanger gives substantial enhancement in overall heat transfer, additional expenditure in pumping power, friction losses high.
	Typical twisted tape	Cengiz Yildiz *et al.* [2]	3400≤Re≤6900	➢ The heat transfer rate increase with increasing pitch size ➢ Increase in pressure drop considerable.
	conical-ring turbulators	Kenan Yakut and Bayram Sahin [3]	Pitch=10,20,30, 8000≤Re≤18000	➢ The conical-ring turbulators increase the heat transfer and friction factor, and also produce vortices in the flow. ➢ The turbulator with 10 mm pitches was improve in heat transfer by 250% under constant pumping power.
	circular-ring turbulators	V. Kongkaitpaiboon, *et al.*[4]	Pitch ratio=6, 8, 12. Diameter ratio=0.5, 0.6, 0.7 4000≤Re≤20000	➢ Heat transfer enhancement around 57% to 195% compared to that in the plain tube.
	perforated conical-rings	V. Kongkaitpaiboon, *et al.* [5]	Pitch ratio=4, 6, 12. 4000≤Re≤20000	➢ The PCRs gives lower heat transfer enhancement than the CRs. ➢ The heat transfer rate and friction factor of PCRs increase with decreasing pitch ratio (PR) and decreasing number of perforated hole. Whereas thermal performance factor vice versa

	cut out conical turbulators	Aydın Durmus [6]	$15000 \le Re \le 60000$.	➢ The heat transfer rates increase with turbulator angles too, as well as friction coefficients
	conical injector type swirl generator	I. Kurtbas [7]	Angle (α)= 30^0, 45^0, 60^0. $10000 \le Re \le 35000$.	➢ Better heat transfer rates are found for a pipe with lower CITSG angle (α) and the flow director angle (β) ➢ The heat transfer ratio (NuPR) decreases with increase in Re.
	combined conical-ring and twisted-tape insert	P. Promvonge, and S. Eiamsa-ard, [8]	Twist ratio= 3.75, 7.5 $6000 \le Re \le 26000$.	➢ The Combined conical-ring and twisted-tape is found that the smaller twist ratio gives larger the heat transfer and friction factor for all Reynolds numbers.
	delta-winglet twisted tape	S. Eiamsa-ard [9]	Cut ratio= 0.11, 0.21, 0.32 $3000 \le Re \le 27000$.	➢ The Nusselt number and friction factor in the test tube with delta-winglet twisted tape are noticeably higher than those in the plain tube and also tube with typical twisted tape.
	Propeller type Turbulator	A. Durmuş [10]	Blade angle=10^0, 20^0, 40^0 Diameter=48, 50, 52 $10000 \le Re \le 30000$.	➢ Heat transfer and pressure drop changes with the change in turbulator propeller diameter. A decrease in heat transfer & pressure loss was seen in the decrease in D_k/D_b.
	Half Length Twisted-Tape Turbulators	Yadav Anil Singh [11]	Twist ratio for half length =7, δ = 0.8mm Length= 2m (2-piece)	➢ A significant improvement in heat transfer coefficient by 40% for half-length twisted tape. ➢ On equal mass flow rate basis, the heat transfer of half-length twisted tape is maximum followed by smooth tube.
	helical turbulators	Zamankhan Piroz [12]	$200 \le Re \le 25000$.	➢ The turbulator was found to increase the heat transfer, sometimes without inducing turbulence, but also increased the pressure drop.

	twisted tape and wire coil turbulators	P. Promvonge, [13]	3000≤Re≤18000.	➢ The combined wire coil and twisted tape turbulators are compared with a smooth tube at a constant pumping power, a double increase in heat transfer especially at low Reynolds number. ➢ It was found at lower Reynolds number values for the lowest values of the coil spring pitch and twist ratio.
	diamond-shaped turbulators	S. Eiamsa-ard, and P. Promvonge [14]	Cone angle=150, 300, 450 Tail length ratio= 1, 1.5, 2 3500≤Re≤16500.	➢ The increase in the heat transfer rate with increasing the cone angle and decreasing the tail length ratio.
	twisted-ring turbulators	C. Thianpong [15]	Twisted ring ratio= 0.05, 0.1, 0.15 Pitch ratio= 1, 1.5, 2 6000≤Re≤20000.	➢ For TRs, Nusselt number and friction factor increase as width ratio increases and pitch ratio decreases. ➢ TRs at small width ratios yield lower Nusselt numbers, they give higher thermal factors as the results of their lower friction factors.
	Tape Consisting of Alternate Axis	P. Seemawute, and S. Eiamsa-Ard [16]	Twist ratio=3,4, 5 3000≤Re≤10000.	➢ It is found that the tube with TA provides a better fluid mixing in the tube than those TT which leads to higher heat transfer rate and also friction factor
	full length twisted tapes	P. K. Nagarajan *et al.* [17]	Twist ratio= 6, 8, 10 1192≤Re≤2534.	➢ The heat transfer enhancement and friction factor increases with decrease in twist ratio. ➢ The lower the twist ratio performs much better than the higher twist ratio inserts.
	varying width twisted tape	Naga Sarada *et al.*[18]	Width range=10, 14, 18, 22, 26 Pitch ratio=3, 4, 5 6000≤Re≤13500.	➢ Heat transfer increase with insertion of twisted tape as compared to plain tube. ➢ Reduction in tape width causes reduction in Nusselt numbers as well as friction factors.
	Regularly spaced twisted tape	S. K. Saha *et al.* [19]	Twist ratio=2.5≤y≤5. 45≤Re≤1150.	➢ Heat transfer rate increased regularly spaced twisted tape.

	combined wavy-surfaced wall and helical-tape	S. Eiamsa-ard and P. Promvonge, [20]	3000≤Re≤9200.	➢ The heat transfer rate can be substantially improved by using both the wavy-surfaced wall and the helical tape.
	twisted tape inserts	M.R. Salimpour and S. Yarmohammadi [21]	Twist ratio= 4, 7, 10, 14	➢ The insertion of twisted tape inside horizontal tubes increases the condensing pressure drop. The pressure drop increases with both refrigerant mass velocity and vapor quality.
	Coil Wire Insert	S. Choudhari Shashank and S.G. Taji [22]	Pitch = 5, 10, 15 4000≤Re≤13000.	➢ The maximum Nusselt number is obtained for copper coil wire insert than aluminum and stainless steel coil wire insert. Copper can be used as coil wire insert material for higher heat transfer enhancement than aluminum and stainless steel. ➢ Friction factor found to be increasing with the decreasing pitch of coil wire insert.
	Twisted tape with pins	S. Selvam *et al.* [23]	10000≤Re≤23000 ratio y/w =3.39, 4.29, 5.71	➢ Nusselt number increases with the decrease of the ratio y/w. ➢ The friction factor also increases with the decreasing twist pitch.
	Peripherally-cut twisted tape	S. Eiamsa-ard *et al.* [25]	1000≤Re≤20000, Twist ratio y=3.0	➢ The peripherally-cut twisted tape offered higher heat transfer rate, friction factor and also thermal performance factor compared to the typical twisted tape. An additional turbulence of fluid in the vicinity of the tube wall and vorticity behind the cuts generated by the modified twisted tape are referred as the reason for a better heat transfer enhancement.
	Twisted tape with centre wing	S. Eiamsa-ard, [26]	Twist ratio y=3.0. Angles of attack b=431, 531and 741 5200≤Re≤22000.	➢ WT-A with the largest angle of attack gave the highest Nusselt number (Nu), friction factor (f) as well as thermal performance factor.

	Twisted tapes with alternate axes and triangular, rectangular and trapezoidal wings	K. Wongchare, and S. Eiamsa-ard [27]	Wing-chord ratios (*d*/*W*) of 0.1, 0.2 and 0.3, twist ratio (*y*/*W*) of 4.0.	➢ Nusselt number, friction factor and thermal performance increase with increasing wing-chord ratio.
	Twisted tape consisting wire nails	P. Murugesan, *et al.*[29]	2000≤Re≤12000, Twist ratios y=2.0, 4.4 and6.0	➢ Common swirling flow generated by P-TT. ➢ Additional turbulence offered by the wire nails.
	Twisted tape with trapezoidal-cut.	P. Murugesan, [33]	2000≤Re≤12000, Twist ratios y = 4.4 and 6.0	➢ The mean Nusselt number for trapezoidal -cut twisted tape higher than typical twisted tape.
	Wire coil in pipe	D. Munoz-Esparza, and E. Sanmiguel-Rojas, [36]	500 ≤*Re* ≤600	➢ The increase of the non dimensional pitch, *p*/*d*, decreases the friction factor.

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