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A review on recent development in heat transfer augmentation techniques in circular tube

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ABSTRACT

In the recent years, high cost of energy and material availability has resulted in an increased efforts aimed at producing more efficient heat exchange equipment. Heat exchangers are extensively used in several industries, such as thermal power plants, air conditioning equipment chemical processing plants, refrigerators, radiators for space vehicles as well as automobiles etc. The design of heat exchanger requires a consideration of different modes of heat transfer, pressure drop, sizing, long term performance estimation as well as economic aspect. The present paper is a review of research work of recent development in heat transfer augmentation techniques. Many researchers have used different techniques to improve heat transfer. Worldwide energy use is projected to increase by almost 1.6 times, from 4.03 x 10^{20} J in 1999 to 6.4 x 10^{20} J in 2020. Hence, if the effectiveness of energy usage could increase by 10 per cent, by means of various heat transfer enhancement techniques, this will result in 6.4 x 10^{19} J of benefit in terms of energy consumption to the society. In present paper emphasis is given on different techniques of heat transfer augmentation, developed recently.

Keywords: heat transfer, augmentation, effectiveness, economic aspect, etc

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I. INTRODUCTION

Conventional resources of energy are depleting at an alarming rate, which makes future sustainable development of energy use very difficult. As a result, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices [1, 2]. The study of improved heat transfer performance is referred to as heat transfer enhancement, augmentation or intensification. In general, this means an increase in heat transfer coefficient [3]. Energy and materials-saving considerations, as well as economic incentives, have led to efforts to produce more efficient heat exchange equipment. Common thermal-hydraulic goals are to reduce the size of a heat exchanger required for a specified heat duty, to upgrade the capacity of an existing heat exchanger, to reduce the approach temperature difference for the process streams, or to reduce the pumping power [4].

The need to optimize and conserve these expenditures has promoted the development of efficient heat exchangers.

1.1 Terminology Used

1. **Thermo Hydraulic Performance**: For a particular Reynolds number, the thermo hydraulic performance of an insert is said to be good if the heat transfer coefficient increases significantly with a minimum increase in friction factor [5, 6].
2. **Overall Enhancement Ratio**: The overall enhancement ratio is defined as the ratio of the heat transfer enhancement ratio to the friction factor ratio [5, 6].
3. **Nusselt Number**: The Nusselt number is a measure of the convective heat transfer occurring at the surface and is defined as hd/k, where h is the convective heat transfer coefficient, d is the diameter of the tube and k is the thermal conductivity [5, 6]

4. **Prandtl Number**: The Prandtl number is defined as the ratio of the molecular diffusivity of momentum to the molecular diffusivity of heat [5, 6].
5. **Pitch**: The Pitch is defined as the distance between two points that are on the same plane, measured parallel to the axis of a Twisted Tape [5, 6].
6. **Twist Ratio**: The twist ratio is defined as the ratio of pitch length to inside diameter of the tube [6, 7].
7. **Modified Twist Ratio**: The twist ratio is defined as the ratio of pitch length to width of the insert [7].
8. **Space Ratio**: the ratio of the distance between drilled conical rings and the inner diameter of tube [8].

1.2 Techniques Augmentation [9, 10]

Heat transfer augmentation techniques are generally classified into three categories namely: Active techniques, Passive techniques and Compound techniques.

1.2.1 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 in the rate of heat transfer and finds limited 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.

In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer. Augmentation of heat transfer by this method can be achieved by: Mechanical aids, Surface vibration, Fluid vibration, Electrostatic fields, etc…

1.2.2 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 behavior (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power; rather they use it from the system itself.

Heat transfer augmentation by these techniques can be achieved by using; Treated Surfaces, Rough surfaces, Extended, Swirl flow devices, etc…

1.2.3 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. This technique involves complex design and hence has limited applications. Some examples of compound techniques are given below:

1. Rough tube wall with twisted tape
2. Rough cylinder with acoustic vibrations
3. Internally finned tube with twisted tape insert
4. Finned tubes in fluidized beds
5. Externally finned tubes subjected to vibrations
6. Gas-solid suspension with an electrical field
7. Fluidized bed with pulsations of air

1.5 Applications of Heat Transfer Enhancement [11]

The petrochemical and chemical industries are under economic pressure to increase the energy efficiency of their processing plants to compete in today's global market. Hence, these industries must invest in innovative thermal technologies that would significantly reduce unit energy consumption in order to reduce overall cost Important applications of heat transfer enhancement are listed below:

1. Heating, Ventilating, Refrigeration and air conditioning
2. Automotive Industries
3. Power sector
4. Process Industries
5. Industrial Heat Recovery
6. Aerospace and others.

II. REVIEW

2.1 Passive Heat Transfer Techniques Improved by the Different Researchers

Shrirao, et al. [12] had done experimental study on the mean Nusselt number, friction factor and thermal enhancement factor characteristics in a circular tube with different types of internal threads of 120 mm pitch under uniform wall heat flux boundary

conditions. In the experiments, measured data are taken at Reynolds number in range of 7,000 to 14,000 with air as the test fluid. The experiments were conducted on circular tube with three different types of internal threads viz. acme, buttress and knuckle threads of constant pitch. Results shows Fig.1 shows test tube with internal threads.

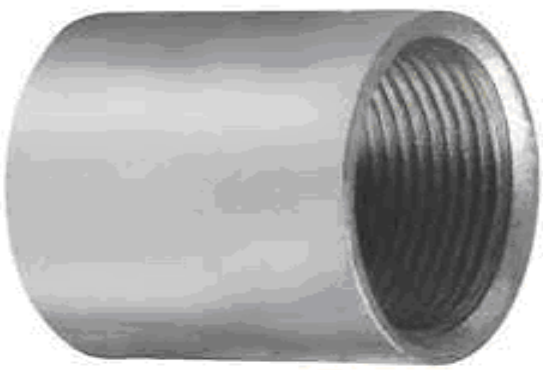

Fig.1 Photograph of Test tube with Internal Thread [12].

Mean Nusselt numbers for test tubes with internal threads such as buttress, acme and knuckle threads are respectively, 1.46, 1.30 and 1.19 times better than that for the plain tube. The thermal enhancement factors are in a range between, 1.12 - 1.04, 1.1 -1.03 and 1.08 - 1.02 respectively for the test tubes with buttress, acme and knuckle threads. Fig.2 shows the variation of enhancement ratio with Reynolds Number.

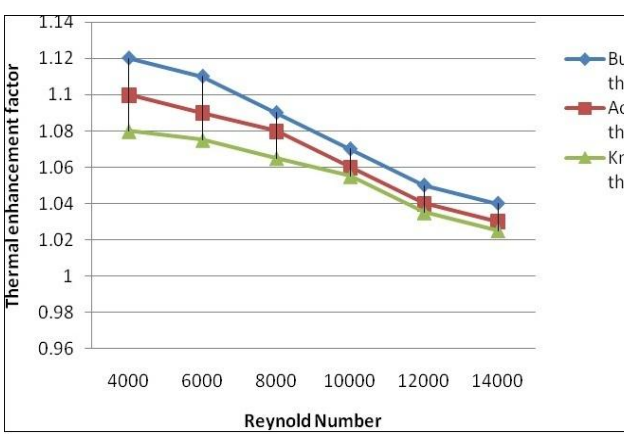

Fig.2 Variation of Enhancement Ratio with Reynolds Number [12].

Sarada, et al. [7] has done experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube by means of varying width twisted tape inserts with air as the working fluid. In order to reduce excessive pressure drops associated with full width twisted tape inserts, with less corresponding reduction in heat transfer coefficients, reduced width twisted tapes of widths ranging from 10 mm to 22 mm, which are lower than the tube inside diameter of 27.5 mm are used. Experiments were carried out for plain tube with/without twisted tape insert at constant wall heat flux and different mass flow rates. The twisted tapes are 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. Both heat transfer coefficient and pressure drop are calculated and the results are compared with those of plain tube. It was found that the enhancement of heat transfer with twisted tape inserts as compared to plain tube varied from 36 to 48% for full width (26mm) and 33 to 39% for reduced width (22 mm) inserts. Maximum friction factor rise was about 18% for 26 mm and only 17.3% for reduced width inserts compared to plain tube Fig.3 shows photo varying width twisted tape.

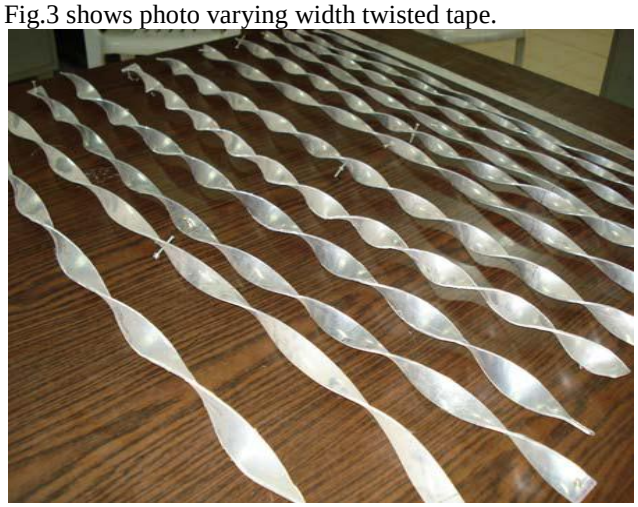

Fig.3 Photo Varying Width Twisted Tape [7].

The experimental results of the Nusselt number and friction factor are correlated in terms of Reynolds number and modified twist ratio as follows

$$Nu = 0.4141\times10^{-4}\times Re^{0.9591}\,[(0.001+H/w)^{-0.04645}]\times(Dh/L)^{-1.411} \quad (1)$$

$$f = 0.01391\times Re^{-0.1374}\,[(0.001+H/w)^{-0.003}]\times(De/L)^{-0.2097} \quad (2)$$

Average deviation = 2.216% and standard deviation = 2.692%

Eq. (1) and (2) are applicable when $3.17 < H/w < 61$, $0.02 < D_e/L < 0.03$ and $6000 < Re < 13500$

When D_h = Dand H/w=0; Eq. (1) and (2) are applicable to plain tube also.

The heat transfer rate and pressure drop characteristics of turbulent flow of air through uniformly heated circular tube fitted with drilled cut conical rings with three space ratios (X=5.4, 6.4, and 8.4) have been studied experimentally. The flow characteristics are governed by space ratio (the ratio of the distance between drilled conical rings and the inner diameter of tube), Reynolds number, and drilled conical ring diameter to inner diameter of tube. Fig. 2.3 shows Geometry of drilled conical-ring turbulator [8].

Jadoaa [8] investigated the effect of drilling of the cut conical ring turbulators (with constant ring to tube diameter ratio) and space ratio on heat transfer, friction factor, and enhancement efficiency under ranging of Reynolds number from 5000 to 23500. In addition, correlation for Nusselt number, friction factor and performance evaluation criteria to assess the real benefits in using the drilled-conical ring turbulator of the enhanced tube are determined. Fig.4 shows Geometry of Drilled Conical-Ring Turbulator.

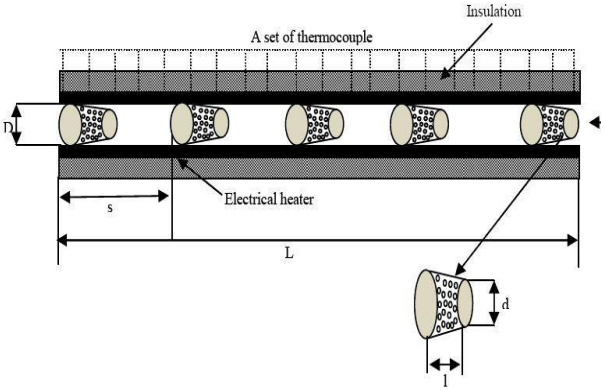

Fig.4 Geometry of Drilled Conical-Ring Turbulator [8].

The results show that the process of drilling of the conical ring inside tube gives high rates of heat transfer more than that in the conical ring without drilling. Fig.5 shows Variation of Enhancement Efficiency with Reynolds Number.

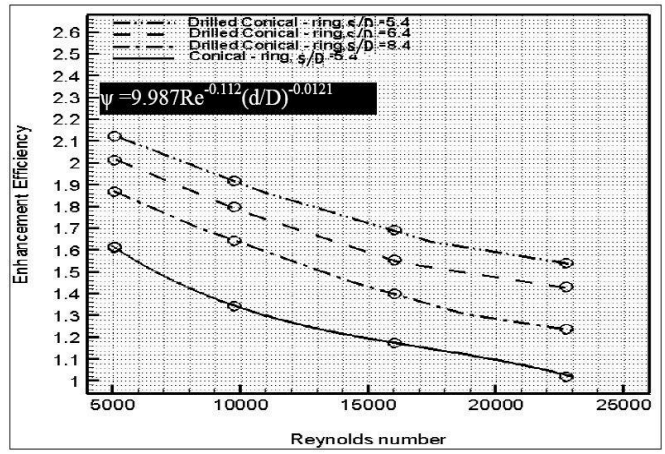

Fig.5 Variation of Enhancement Efficiency with Reynolds Number [8].

Based on the present data of the tube fitted with drilled conical ring, Nusselt number, friction factor and enhancement efficiency correlations are derived in the following forms, respectively,

$$Nu = 1.725\times Re^{0.891}\times(d/D)^{-1.012}\times(s/D)^{0.011} \quad (3)$$

$$C_f = 18Re^{-0.11}\times(d/D)^{-1.11}\times(s/D)^{-0.121} \quad (4)$$

$$\psi = 9.987\times Re^{-0.112}\times (d/D)^{-0.0121} \quad (5)$$

Sarada, et al. [1] had done experimental and numerical investigations of the augmentation of turbulent flow heat transfer in a horizontal circular tube by means of mesh inserts with air as the working fluid. Sixteen types of mesh inserts with screen diameters of 22 mm, 18 mm, 14 mm and 10 mm for varying distance between the screens of 50 mm, 100 mm, 150 mm and 200 mm in the porosity range of 99.73 to 99.98 were considered for experimentation. The horizontal tube was subjected to constant and uniform heat flux. The Reynolds number varied from 7,000 to 14,000. Fig.6 shows Porous Medium manufactured from Copper Screens.

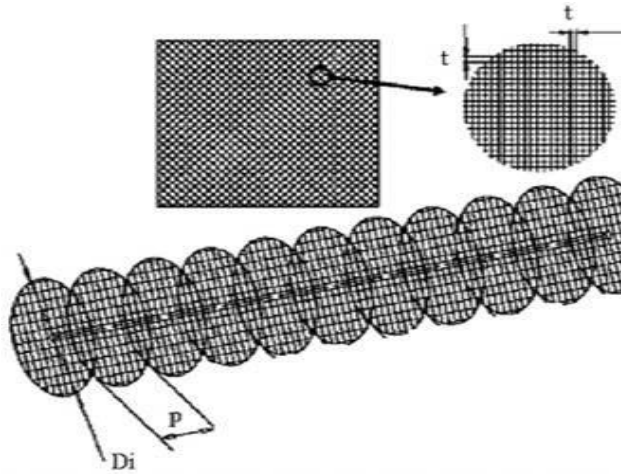

Fig.6 Porous Medium Manufactured From Copper Screens [1]

The maximum Nusselt number obtained at smallest pitch (2.5 mm) of larger mesh diameter (26mm) using CFD analysis is 2.15 times that of plain tube. The maximum increase in Nusselt number and pressure drop approximately 1.86 and 1.23 times was obtained through experimental investigation than plain tube for Rp = 0.8 with the distance between screens equal to 50 mm. Fig.7 shows variation of Nusselt number with Reynolds number.

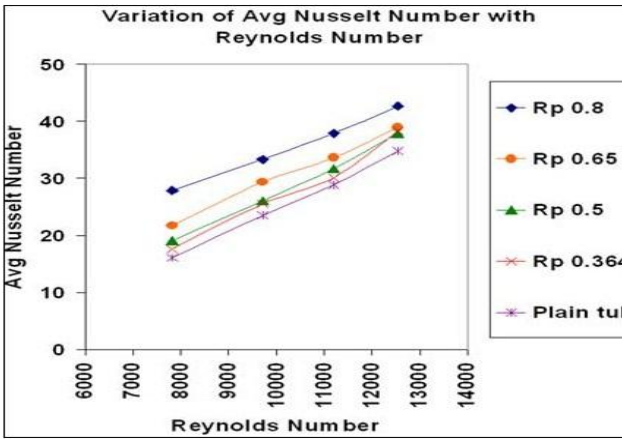

Fig.7 Variation of Nusselt number with Reynolds Number [1]

Sarada, et al. [2] investigated the heat transfer enhancement by square leaf inserts. Overall enhancement ratio is observed highest for BW insert. The maximum overall enhancement ratios are 1.30, 1.32, 1.34, 1.02 and 0.79 for 30^0 BW, 60^0BW and 90^0, 30^0FW, 60^0FW inserts respectively. Although Nusselt number increases with respect to plain tube is less for 60^0 BW insert compared to that of 90^0 insert, as the obstruction to air flow is less in this case, it might have caused the increase of overall enhancement ratio. Fig.8 shows Louvered Square Leaf Inserts 30^0 BW, 60^0 BW and 90^0, 30^0 FW, 60^0 FW (left to right) Counter co swirl generator respectively.

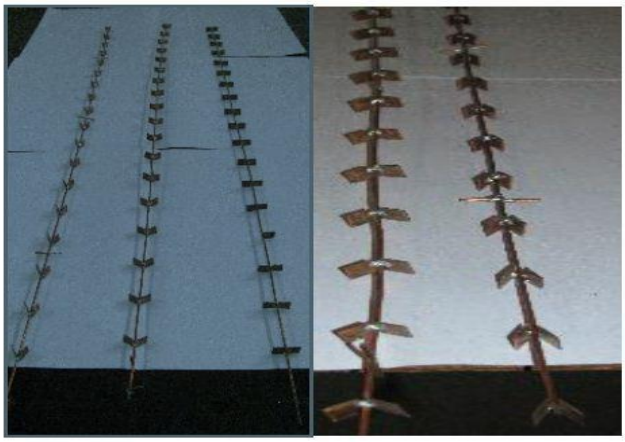

Fig.8 Louvered Square Leaf Inserts 30^0 BW, 60^0 BW and 90^0, 30^0 FW, 60^0 FW (left to right) Counter co swirl Generator Respectively [2]

Nusselt number increased by a maximum of 128.39, 121, 81.31, 30.03 and 32.72 % in the presence of 90^0, 60^0 forward, 60^0 backward, 30^0 forward, 30^0 backward square leaf inserts respectively. Friction factor increased by a maximum of 441.31, 369.17, 143.43, 116.48 and 80.39% in the presence of 90^0, 60^0 forward, 60^0 backward, 30^0 forward, 30^0 backward square leaf inserts respectively. Maximum overall enhancement ratios were 1.30, 1.32, 1.34, 1.02 and 0.79 for 90^0, 60^0 forward, 60^0 backward, 30^0 forward, 30^0 backward inserts respectively. Fig.9 shows variation of enhancement ratio with Reynolds Number.

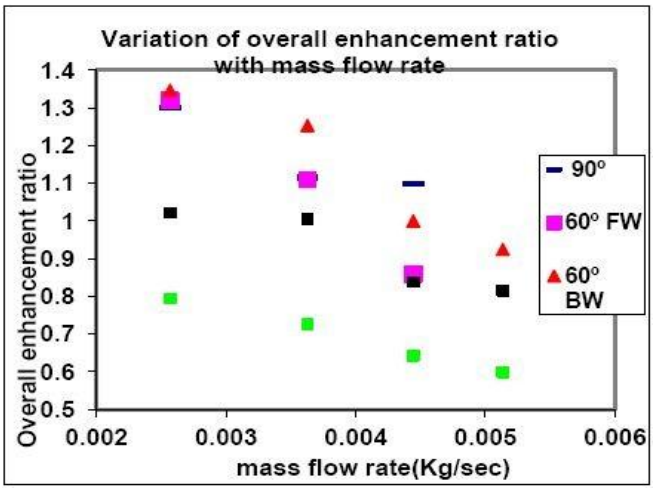

Fig.9 Variation of Enhancement Ratio with Reynolds Number [2]

Kore et al. [13] carried out experimental investigation to study heat transfer and friction factor by dimpled surface. The aspect ratio of rectangular channel is kept 4:1 and Reynolds number based on

hydraulic diameter is varied from 10000 to 40000. The ratios of dimple depth to dimple print diameter is varied from 0.02 to 0.04 to provide information on the influences of dimple depth. The ratio of channel height to print diameter is 0.5. The heat transfer and friction factor data obtained is compared with the data obtained from smooth plate under similar geometric and flow conditions. It is observed that at all Reynolds number as depth increases from 0.2 to 0.3, the normalized Nusselt number and thermal performance increases and then after when depth increase from 0.3 to 0.4 normalized Nusselt number and thermal performance decreases. These are because of increase in strength and intensity of vortices and associated secondary flows ejected from the dimples. Fig.6 shows Geometry of Dimpled Surface.

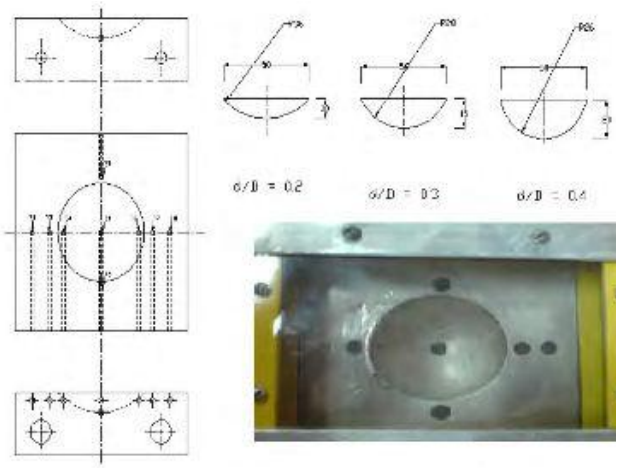

Fig.10 Geometry of Dimpled Surface [13]

Heat transfer improvement with dimples seems to have a maximum value of approximately 2.88 and overall maximum thermal performance of about 2.63 for a depth of 0.3. Fig.11 shows variation Enhancement ratio with Reynolds number.

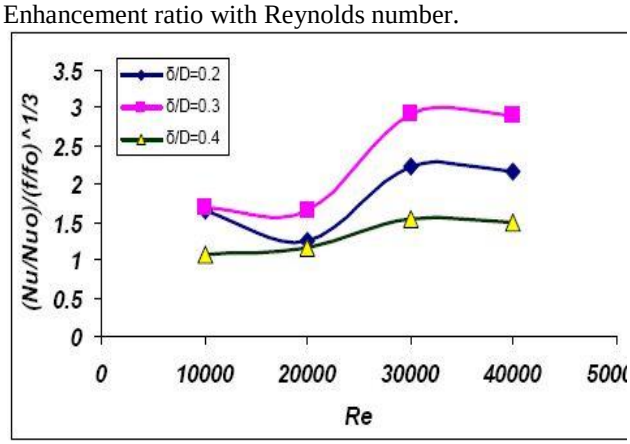

Fig.11 Variation Enhancement Ratio with Reynolds number [13].

2.2 Active Heat Transfer Techniques Improved by the Different Researchers.

Shewale et al. [14] had done experimental investigation of double-pipe heat exchanger with helical fins on the inner rotating tube.In this work, to improve the heat transfer characteristic of the double pipe heat exchanger, the helical fins were installed on the outer surface of the inner tube and the level of turbulence increased by the rotating the inner tube. The length of heat exchanger was 1m and the pitch of helical fins kept constant equal to 17 mm. The convective heat transfer coefficients were obtained for the stationary as well as rotating inner tube for the counter flow mode using water as cold fluid in the tube side and glycerol as hot fluid in the shell side. The flow rate of cold fluid was kept constant and that of hot fluid was varied. The Nusselt number was calculated for the each speed of the rotation and compared with standard values obtained from Dittus-Boelter equation. The helical fins increases heat transfer area and rotation of the inner tube increases the mixing of fluid particles which is necessary for the convection mode of heat transfer. Fig.12 shows the model of Inner Tube with Helical Fins on Outer Surface.

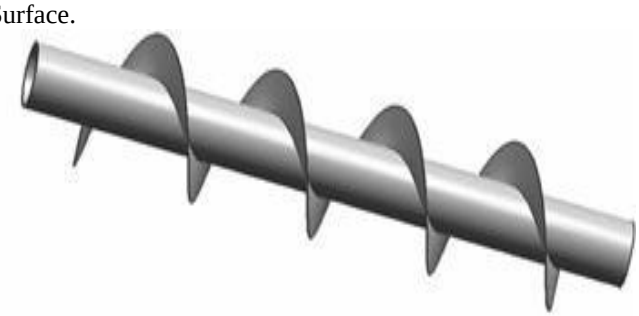

Fig.12 Inner Tube with Helical Fins on Outer Surface [14]

Results shows that Nusselt numbers at the speed 50 rpm and 100 rpm are 36% and 64% more than that of stationary inner tube respectively. The Nusselt numbers at 100 rpm are 21 % higher than that of 50 rpm. Fig.13 shows variation Nusselt Number with Reynolds number

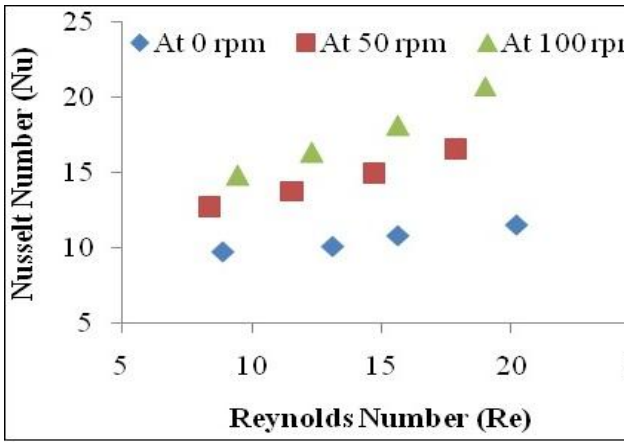

Fig.13 Variation Nusselt Number with Reynolds number [14].

Zhang et al. [15]carried out investigation on heat transfer characteristics of a helically baffled heat exchanger combined with one three-dimensional finned tube. The experiments were carried out in counter mode operation with hot oil in the shell side and cold water in the tube side. Overall heat transfer coefficients were calculated and heat transfer coefficients in the shell and tube side were determined using modified Wilson plot. A commercial computation fluid dynamics (CFD) program called Fluent 6.0 was used to predict the flow and heat transfer performance in the heat exchanger. The numerical results agree well with the measurements. The maximum differences between the present numerical results and the experimental data are approximately 6.3% for Nusselt number and 9.8% for pressure drop, respectively. Fig.14 shows Photography of PF (petal shaped finned) tube enwound with helical baffle. Fig.15 shows Cross-sectional diagram of PF (petal shaped finned) tube.

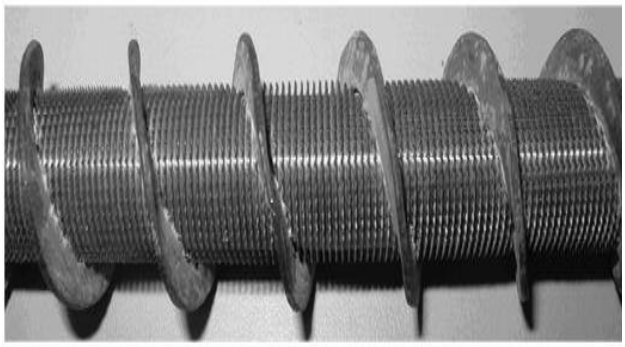

Fig.14 Photography of PF tube enwound with helical baffle [15].

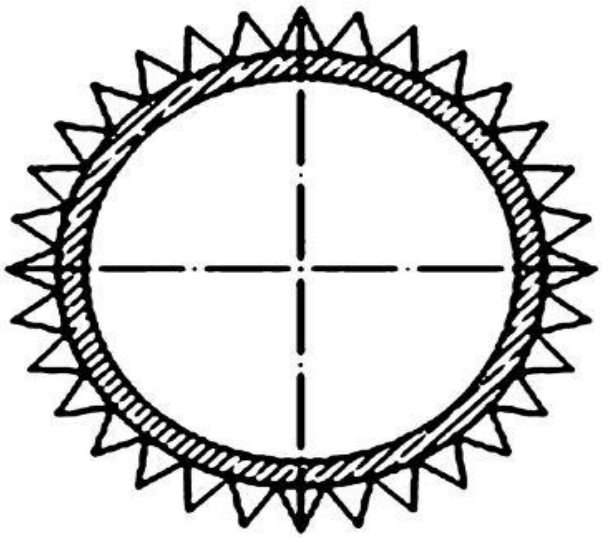

Fig.15 Cross-sectional diagram of PF tube [15]

III. CONCLUSIONS

This review has considered heat transfer and pressure drop investigations of the various passive techniques like tubes with internal threading, drilled conical ring turbulator, porous mesh inserts, Louvered square leaf inserts, combined twisted tape and wire oil inserts, Twisted tape, Dimpled Surface and active techniques like Inner Tube with Helical Fins on Outer Surface, PF(petal shaped finned) tube.From this review, various ways of enhancing the heat transfer rate by generating the swirl flow by active or passive method can be observed by using various types of inserts.

Mechanisms used for the enhancement of heat transfer for above inserts are as follows:

1. Disruption of laminar sub-layer in the turbulent boundary layer by internal threading.
2. Use of swirl flow devices like drilled conical ring turbulator, twisted tape, combined twisted tape and wire coil insert.
3. Use of flow obstruction devices like mesh inserts, Louvered square leaf inserts.
4. Introducing Secondary flows by devices like dimpled surfaces.
5. By creating turbulence by providing helical fins on the inner rotating tube.

Passive methods can easily manufacture and applicable. Few researchers used active techniques of heat transfer, as they are complex. We need to work on compound technique, surly it will augment more heat transfer than other methods.

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