An experimental analysis on a Double Pipe Heat Exchanger Modified with Turbulators

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Thermal Engineering

Submitted by

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CERTIFICATE

I, Harsimranjot Singh, hereby declare that the thesis entitled "**An experimental analysis on a Double Pipe Heat Exchanger Modified with Turbulators**" is an authentic record of my work carried out as requirements for the award of the degree of **Master of Engineering** in **Thermal Engineering** at **Thapar University, Patiala** under the supervision of **Mr. Kundan Lal**, Assistant Professor , Department of Mechanical Engineering , Thapar University, Patiala during July, 2015 to July, 2016.No part of the matter embodied in this report has been submitted to any other university or institute for the award of any degree.

Date: 15-07-2016

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It is certified that the above statement made by the student is correct to the best of my knowledge and belief.

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Abstract

Heat exchangers find so many applications starting from domestic purposes to industries, automobiles, aviation and many more. The need of the hour is to maximize the efficiency of heat exchangers and at the same time, reduce the size of the heat exchanger. One of the methods to achieve this purpose is introduction of turbulators in the heat exchangers. Turbulators serve the purpose of heat transfer enhancement but they also cause drop in fluid pressure. A variety of turbulators have been proposed by various researchers in the past. Three common turbulators are spring insert, twisted tape insert and twisted wire mesh insert. The present study deals with comparative experimental analysis of a double pipe heat exchanger that has been modified by introduction of turbulators. Different sets of combinations have been used for carrying out experimental study. For spring inserts, pitch has been chosen as the variable. Three spring inserts having pitches 5 mm, 10 mm and 15 mm respectively have been used. For twisted tape inserts, the twist ratio has been taken as variable with twist ratios 5, 3.75 and 2.5 respectively. Similarly, for the twisted wire mesh inserts, twist ratio has been varied for carrying out the experimental analysis. Three twisted wire mesh inserts having twist ratios 5, 3.75 and 2.5 have been used in the heat exchanger. Experimental analysis was carried out for wide range of Reynolds number ranging from 5000 to 30000 for the hot fluid. Experiments have been performed for each set and the results for heat transfer characteristics and pressure drop characteristics have been compared. For the heat transfer characteristics, the comparison has been made between the Nusselt number values from Dittus Boelter equation, values obtained for plain pipe heat exchanger and the results obtained after introduction of various turbulators in the heat exchanger. Similar approach has been used for pressure drop characteristics by using the friction factor values obtained from Blasius Correlation. The whole experimental study is presented in condensed form in this thesis.

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List of symbols

d_i	Internal diameter of inner pipe, mm
d_0	Outer diameter of inner pipe, mm
D_i	Internal diameter of outer pipe, mm
Ac	Cross sectional area, m^2
Ai	Curved Surface area of inner pipe, m^2
Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number
f	Friction factor
μ_w	Dynamic viscosity of water, $kgs^{-1}m^{-1}$
L	Total length of test section, m
k_w	Thermal conductivity of water, $Wm^{-1}K^{-1}$
m	Mass flow rate of water, kgs^{-1}
ρ	Density of water, kgm^{-3}

P	Pressure, Pa
V	Velocity of the fluid, ms^{-1}
g	Acceleration due to gravity, ms^{-2}
Q	Volume flow rate, m^{3}
U	Overall heat transfer coefficient, Wm^{-2}K^{-1}
h	Convective heat transfer coefficient, Wm^{-2}K^{-1}
t_h	Temperature of hot fluid, K
t_c	Temperature of cold fluid, K
h_f	Pressure drop
R_{ov}	Overall thermal resistance
R_i	Thermal resistance corresponding to the internal convection
R_w	Thermal resistance corresponding to the pipe wall
R_0	Thermal resistance corresponding to the external convection
c_{pw}	Specific heat capacity of water, KJkg $^{-1}$K^{-1}
Y	Twist ratio of twisted tape insert or twisted wire mesh insert

Subscripts	
i	inlet
o	outlet
c	cold
h	hot
Abbreviations	
Spring 1	Spring insert having pitch 15 mm
Spring 2	Spring insert having pitch 10 mm
Spring 3	Spring insert having pitch 5 mm
Twisted Wire Mesh 1	Wire Mesh insert having twist ratio 5
Twisted Wire Mesh 2	Wire Mesh insert having twist ratio 3.75
Twisted Wire Mesh 3	Wire Mesh insert having twist ratio 2.5
Twisted Tape 1	Twisted Tape insert having twist ratio 5
Twisted Tape 2	Twisted Tape insert having twist ratio 3.75
Twisted Tape 3	Twisted Tape insert having twist ratio 2.5
LMTD	Log mean temperature difference

lpm	Liter per minute
RTD	Resistance temperature Detector
PID	Proportional Integral Derivative
min	Time in minutes

Chapter 1

Introduction

1.1 Introduction

1.1.1 Heat exchanger

A heat exchanger is a device that is used for efficient heat transfer between two fluids that are at different temperatures. The fluid at lower temperature receives heat from the fluid at higher temperature and its temperature is raised. Heat exchangers find applications in all aspects of life. Following are the few of wide variety application of heat exchangers:

1. Domestic purpose: In domestic purpose, heat exchangers are used in refrigerators, air conditioners and blowers etc.
2. Automotive applications: Heat exchangers finds applications in automotive industry for components such as radiators and air conditioners.
3. Industrial applications: The most wide applications of heat exchangers are found in industries. Heat exchangers are used in power sector for electricity generation, in sugar industries , in fertiliser industries, beverage industries and many more.

1.1.2 Classifications of heat exchangers: Heat exchangers can be basically classified in three main categories

1. **On the basis of operating principle**: In this category, heat exchangers are further classified in to three sub categories.

   First are direct or open heat exchangers in which the energy is transferred from one fluid to the other by the effect of direct physical mixing between the two fluids. Such type of heat exchangers are used in the applications where the physical mixing of two media does not pose any problem. The classic example is jet condensers that are used in the steam power plants

The second sub category is regenerator where the hot fluid is first made to pass through a solid media known as matrix. The hot fluid transfers its heat to the solid matrix. Then the cold fluid is made to flow through the matrix and it absorbs heat from matrix and gets heated. One of the examples of regenerator type heat exchanger is gas turbine.

The third one is known as recuperator and it is the most widely used type of heat exchangers. In these type of exchangers, the two fluids flow simultaneously and they are separated by a wall. There is no direct contact or physical mixing between two fluids. In these type of exchangers, heat is transferred both by convection between fluid and separating wall and also through conduction in the wall. The examples are boilers, super heaters and condensers etc.

2. **On the basis of fluid flow arrangement:** Under this category heat exchangers can be classified into three categories:

   Parallel flow heat exchangers: In parallel flow heat exchangers, both the hot and cold fluids enter the heat exchanger from the same side and they travel parallel and in the same direction. In the due course , heat is transferred from hot fluid to cold fluid.

   Counter flow heat exchangers: In counter flow heat exchangers, fluids enter the heat exchanger from opposite sides . The travel parallel but in opposite directions.

   Cross flow heat exchangers. In cross flow heat exchangers, the two fluids travel at right angles to each other. These heat exchangers are usually used in air or gas heating applications.

3. **On the basis of mechanical design:** The heat exchangers can be categorised on the basis of mechanical design as follows:

   Double pipe heat exchangers – This is the most common type of heat exchangers. It consists of two concentric pipes or tubes. One fluid flow through the inner pipe and the other fluid flows through annulus. The direct of flow can be either parallel or counter. The primary criteria in double pipe heat exchanger is that the material of inner pipe must have very high conductivity in order to achieve the maximum possible heat transfer. These heat exchangers are simple in design and also require very less maintenance.

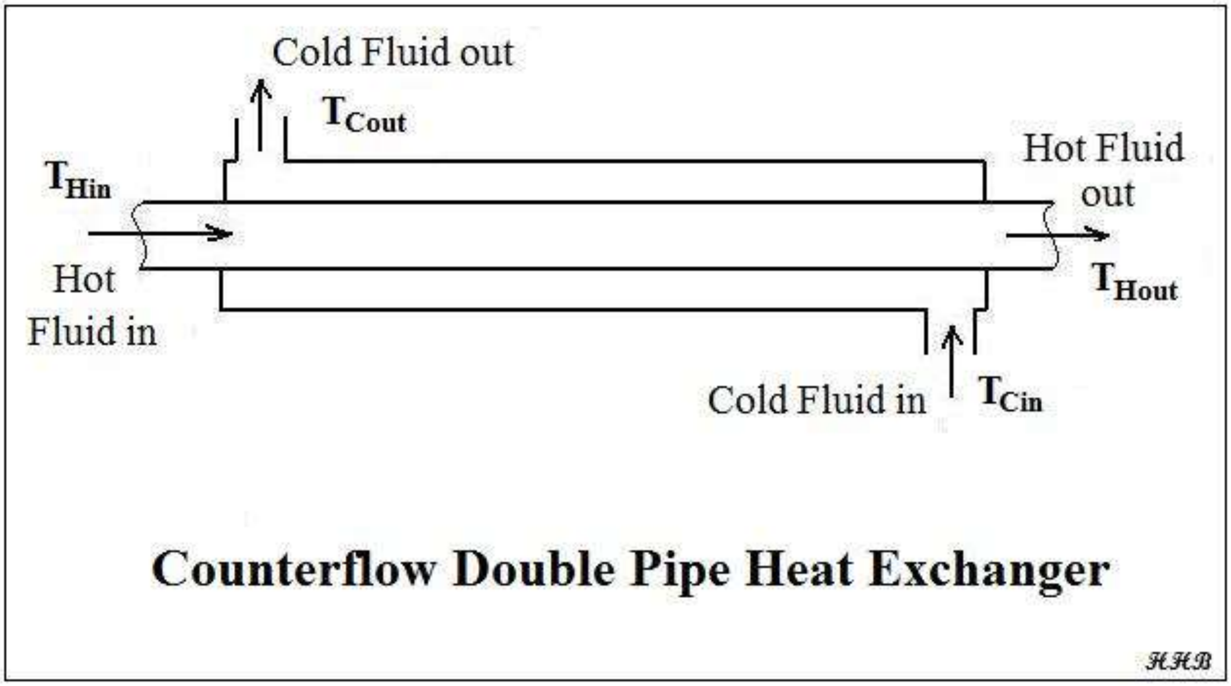

Figure 1.1: Double pipe heat exchanger

(Source: http://www.engineeringexceltemplates.com)

Shell and pipe heat exchanger –Shell and tube heat exchangers consists of a bundle of tubes that are fitted in a shell. One of the fluid flows through the bundle of tubes and the other passes through the shell. The fluids can be either liquids or gases. These exchangers are mostly used in oil refineries and chemical industries. Shell and tube exchangers are the most commonly used heat exchangers because of their simple design and easy fabrication . Also, there are shell and tube heat exchangers where the tube bundles are floating inside the shell and they can be easily removed from the shell for periodic maintenance to remove corrosion or deposits and it reduces the overall maintenance costs . Shell and tube heat exchangers can have different configurations such as U tube heat exchanger, one pass tube side and two pass tube side etc.

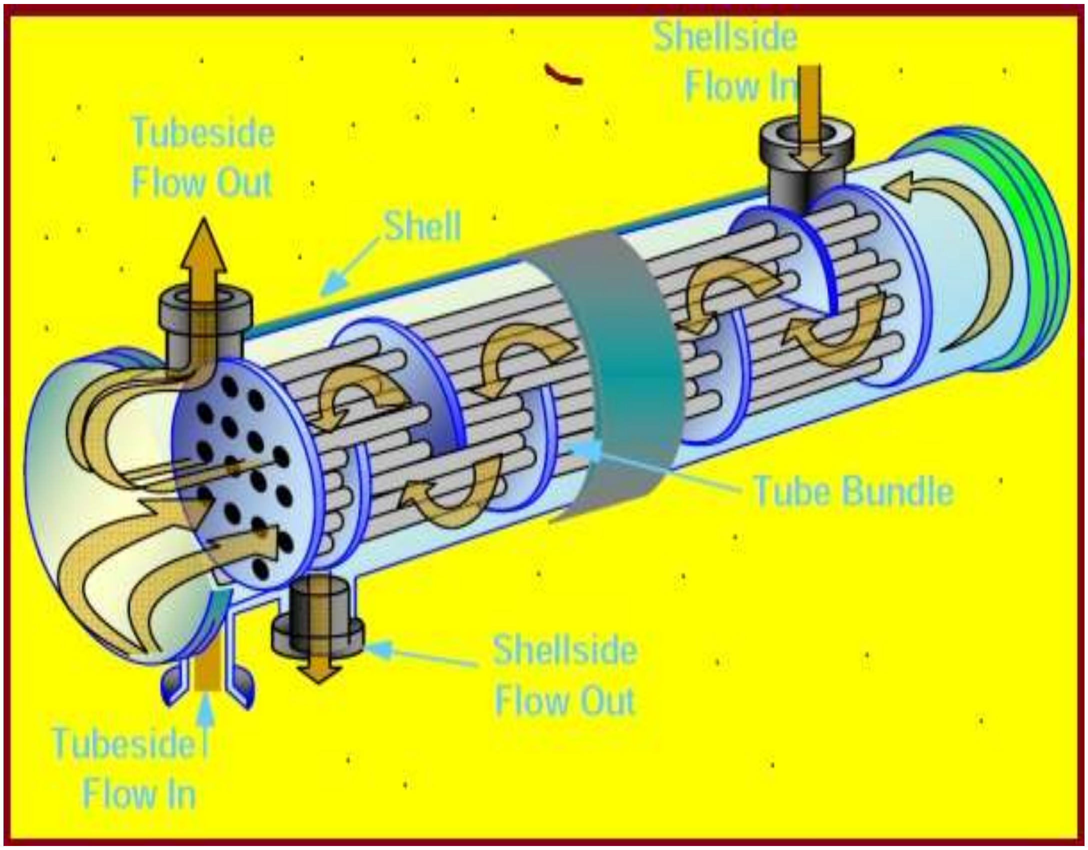

Figure 2.2: Working Principle of Shell and pipe heat exchanger
(Source: http://www.whatispiping.com)

Plate heat exchanger – Plate heat exchangers are fabricated by joining thin metal plated together. A small separation is left between two consecutive plates with the help of a rubber gasket. The two fluid streams flow in the alternate channels of plate heat exchanger. These heat exchangers are more efficient than shell and tube heat exchangers due to their larger surface area and hence find applications in refrigeration and air conditioning applications. The plate heat exchangers are widely used in domestic applications for heating and cooling purposes due to their compact size.In the large scale applications of plate heat exchangers, the gaskets are used between the plates whereas for small exchangers, brazing is used. Steel is the commonly used material for plate heat exchangers due to its corrosion resistance, high strength and ability to withstand high temperatures.

Figure 3.3: A Plate Heat Exchanger installed in industry
(Source: https://www.tranter.com)

Spiral heat exchanger – These heat exchangers are manufactured by coiling the flat plates in the shape of spirals. The two fluids at different temperatures flow in alternate passages. Spiral heat exchangers are compact in size and they are of great use where space is a constraint. Also, spiral heat exchangers cause very high turbulence that results in lesser fouling effects and better heat transfer between two fluids. Due to this property, spiral heat exchangers are widely used for fluids that contain suspended particles or for viscous fluids where chances of depositions are quite high.

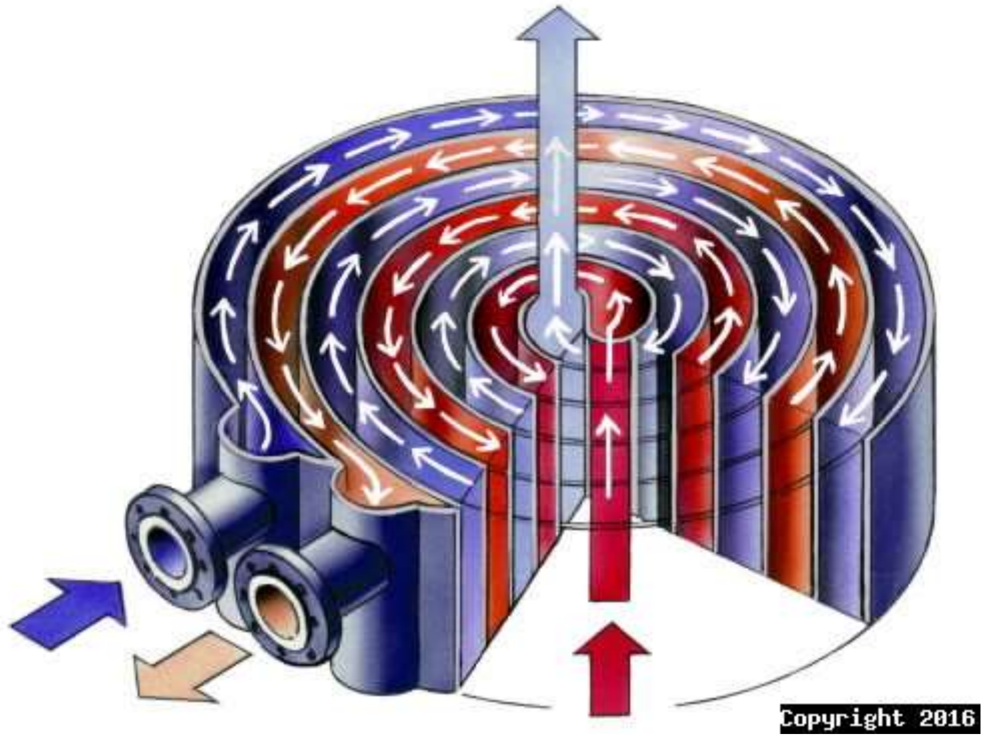

Figure 4.4: Spiral Heat Exchanger (Source: http://www.virginiaheattransfer.com)

Plate Fin heat exchanger- These heat exchangers are fabricated by sand witching fins between plates. They are mostly used for gas to gas applications. One of the benefits of plate fin heat exchangers is their compact design that leads to higher heat transfer rates per unit volume. The two fluids flow in the alternate passages of plate fin heat exchangers. The most common applications of plate fin heat exchangers are nuclear power plants and gas turbines.

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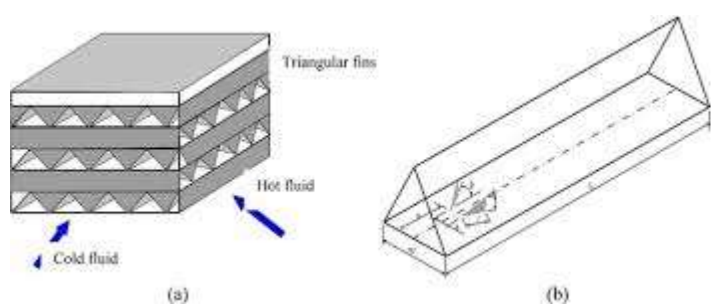

Figure 5.5: Plate Fin Heat Exchanger (Source: https://www.researchgate.net)

Tubular Fin heat exchanger- These heat exchangers are used normally in the applications where one of the fluids is gas and the other fluid is a liquid. The fins are attached to tubes in these heat exchangers. The liquid fluid flows through the tubes and the gas flows over the fins and thereby heat exchange takes place. They find most common applications in condensers and waste heat recovery systems.

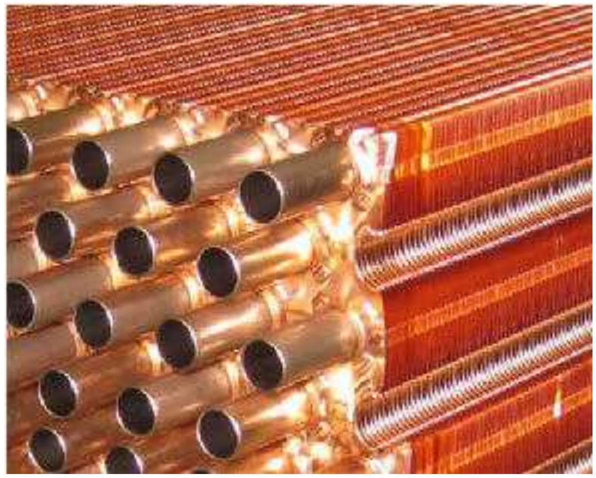

Figure 6.6: Tubular fin heat exchanger

(Source: http://www.alternative-energy-tutorials.com)

1.1.3 Need of Heat transfer enhancement and scope of present study:

In these days of energy crisis, energy resources are depleting and researchers are looking for techniques to utilise every bit of available energy. In heat exchanger industry also, the need of the hour is to maximize the efficiency of the heat exchangers and minimize the losses. Also, in many instances, the need of modification in an existing system arises. For such instances, the concept of turbulators or inserts have been proposed. The techniques by which the efficiency of an existing heat exchanger is increased are known as heat transfer enhancement techniques. Following are the key points why heat transfer methods are required:

1. **Reducing capital cost of new heat exchangers**: If the introduction of an insert is incorporated prior to the design of the heat exchanger, then the overall size of the heat exchanger can be reduced by a fair value for the same amount of heat transfer and it will result in reduction of the capital cost of heat exchanger.

2. **Reducing size of heat exchanger**: In many applications such as automotive applications or aviation industry, the size of the heat exchanger plays a major role in overall performance of the system. The bulkier the heat exchanger, more will be its weight and that will deter the performance of machinery. So, in such cases, with the introduction of inserts, the size of the heat exchanger can be reduced which will result in overall performance improvement of the system.
3. **Increasing efficiency of existing systems**: For an existing system, with the introduction of inserts, the overall heat transfer increases with increase in pressure drop. If pressure drop is of no concern, then the efficiency of existing heat exchanger will be increased with very little input.
4. **Mitigating fouling effects:** In many applications such as steam power plants the fluids contain debris or other particles which under high temperatures and pressures, gets accumulated inside the heat exchangers. These depositions reduce the efficiency of heat exchangers. With the introduction of inserts, the turbulence of the fluid is increased and it leads to lesser depositions which means lesser amount of fouling in the heat exchanger.
5. **Reduction in maintenance costs**: Since increased turbulence means lesser scaling and depositions, heat exchangers will require lesser maintenance. That helps in two manners. Firstly, the maintenance costs are reduced. Secondly, the system goes to breakdown lesser number of times which in turn will mean better profits.

1.1.4 Mechanisms of heat transfer enhancement: There are various methods by which the heat transfer rate can be enhanced in a heat exchanger. Some of the mechanisms that can be followed for heat transfer augmentation are:

1. By using a secondary heat transfer surface
2. By causing disturbance to laminar sub layer
3. By promoting boundary layer separation that causes high turbulence
4. By enhancing the effective thermal conductivity of the fluid
5. By causing a delay in the boundary layer development
6. By increasing the fluid flow rate
7. By causing turbulence in the fluid path that results in better mixing of the fluid

1.1.5 Classification of heat transfer enhancement techniques: A lot of research has been done in past and many researchers are also working these days on the methods that can increase the efficiency of heat exchangers. Various techniques have been proposed and tested numerically as well as experimentally. In the broad scenario, heat transfer augmentation techniques can be divided into two categories:

1. Passive techniques
2. Active techniques

1. Passive Techniques

In the passive techniques of heat transfer enhancement, no external power is required. The heat transfer increase is achieved by altering the geometry of the heat exchanger or by bringing in a change in design of the heat exchanger. This can also involve addition of some additives in the fluids that results in increased thermal conductivity and hence higher heat transfer rates. Various techniques adopted under passive category are:

- **Plain fins:** Under this category, rectangular or triangular fins are attached to the tubes of the heat exchanger. This increases the heat transfer surface are which results in higher heat transfer rate.
- **Surface treatment:** Surface treatment is achieved by either coating the heat transfer surface of the heat exchanger at micro level or by using the machining processes. Grooving can be an option under the category of machining processes. All these methods increase the fluid turbulence that promotes better mixing of the fluid layers and reduction in the thickness of the boundary layer which in turn increases the heat transfer rate. This method also causes lesser pressure drops as compared to some of the other methods such as twisted tape inserts where large amount of obstruction is caused to fluid flow by virtue the geometry of the insert. The coating can also be done at the nano scale or etching can be used to promote turbulence.
- **Corrugated surfaces:** Corrugated surfaces also increase heat transfer by generating turbulence in the fluid flow. Corrugated tubes can be used instead of plain pipes for a double pipe heat exchanger.

Figure 1.7: Corrugated tube for heat exchanger

(Source: http://www.hrs-heatexchangers.com)

- **Swirl generating devices-** These are the methods in which an insert is placed in the flow path that results in turbulence and higher heat transfer rate. Turbulators are of various types and some of them are as under:

➢ **Twisted tape inserts:** Twisted tape inserts are one of the most common inserts in which a lot of research has been carried out. They are mostly used in inner pipe of the double pipe heat exchanger. These inserts are fabricated by twisting a metal strip .The geometry of the twisted tape insert causes fluid turbulence which leads to better heat transfer coefficient for the same Reynolds number at inlet of the hot fluid. For the twisted tape inserts, twist ratio is the most common variable used

The twist ratio is defined as the ratio of the pitch of the twisted tape to the internal diameter of the pipe

$$Y = y/d$$

Where Y is the twist ratio

y is the picth of the twisted tape i.e. distance between two consecutive twists

D is the internal diameter of the pipe in which the twisted tape is inserted

Figure 1.8 : Twisted tape insert (Source: http://www.calgavin.com)

- **Twisted wire mesh insert:** Twisted wire mesh insert is quite similar to the twisted tape inserts. The only difference is that the metal strip here is replaced by a wire mesh. It results in lesser restriction to the fluid flow and hence lower pressure drops. In twisted wire mesh inserts also, the twist ratio is usually the main variable chosen as the base for study. Also, the mesh size of the wire mesh can be chosen as variable to determine its effect on heat transfer and pressure drop characteristics

Figure 7.9 Twisted wire mesh insert used by Veeresh Fuskele and Dr. R.M. Sarviya

- **Louvered Strip inserts:** Another type of swirl generating insert is louvered strip inserts. In the louvered strip inserts, louvers with different geometries are fabricated on a central core rod which is usually inserted in the inner pipe of the double pipe heat exchanger. Various geometries such as square, triangular or trapezoidal have been proposed by researchers for the louvered strip inserts. The main parameters that are used in the louvered strip inserts are the louver arrangement, louver angles and the pitch of louvers. Louvers can be arranged in two methods
  - Forward arrangement in which the louvers are inclined in opposite direction to the fluid in flow
  - Backward arrangement of louvers in which the louvers are inclined in the direction of the flow

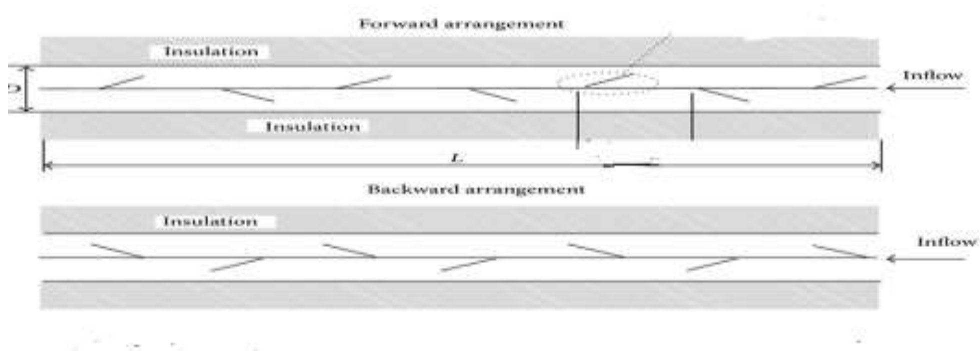

Figure 8.10: Forward and backward arrangements of louvered strips

The angle of inclination of the louvers play an important role in increase in heat transfer and the pressure drop. At the lower angles, pressure drop is lower and % increase in heat transfer is also on the lower side. Higher is the louver angle, higher will be the restriction to the flow of the fluid and it results in better overall heat transfer. The width of the louvers and diameter of the central core can also be the variables in the study for louvered strip inserts

- **Using nano fluids instead of regular fluids:** Nanofluids are fluids containing suspensions of nano particles of high thermally conductive materials like carbon, metals, and metal oxides into heat transfer fluids. These fluids have very high thermal conductivity and this results in higher heat transfer rates. The size of particles in nano fluids is usually of order 100 nm or less. Nano particles can be spherical or cylindrical in geometry.

The suspensions of nanoparticles in nanofluids increase the effective thermal conductivity of the fluid under macroscopically static conditions. Many researchers have carried out numerical and experimental analysis on this aspect of increased thermal conductivity.

The enhancements in thermal conductivity of nanofluids are due to the fact that particles surface area to volume ratio increases with the decrease in particle size. It increases the overall exposed heat transfer surface area for a given concentration of particles as their diameters decreases. Also, the presence of nano particles suspensions in fluids increase the mixing effects within the fluid which produce additional increase in the fluid's thermal conductivity due to thermal dispersion effects as discussed.

The reasons for enhancement of heat transfer by nano fluids can be summarized as:

a) Enhancement of the thermal conductivity

b) Enhancement of thermal conduction under dynamic conditions

c) Reduction in thickness of the boundary layer

e) Delay in the boundary layer development,

f) Particle re-arrangement

g) Higher aspect ratio of carbon nanotube

2. **Active Techniques:** Active techniques involve the use of external power for heat transfer enhancement. These methods are complex and are used in some specific applications. The examples of active methods include pulsation by cams and reciprocating plungers which cause the turbulence in fluid. Also magnetic fields or ultrasonic waves can be used to disturb the fluid flow and thereby increasing the heat transfer rate. These methods are not always possible as in some cases, the movement of walls may not be feasible due to space constraints or some other reason. Under the active methods, one of the techniques can be where the solid wall is made to move by any means such as mechanical ,electrical or magnetic. The ultrasonic waves can also be used which causes nucleation process and thereby increasing the value of heat transfer coefficient.

   The main emphasis in this experimental study is on the use of passive heat transfer augmentation techniques and all the experimentation has been carried out for the passive heat transfer enhancement methods.

Chapter 2

Literature Review

Introduction: The field of research in enhancement of heat transfer of heat exchangers is not a new one. In the past also, many researchers have worked on various methods that can lead to heat transfer enhancement of an existing heat exchanger by bringing some modifications. A lot of work has been done in past for double pipe heat exchangers that have been modified by introduction of inserts. Both numerical and experimental research has been conducted by various researchers to determine the change in heat transfer brought about by these modifications.

S.K.Saha et al. [2001] worked on determining friction and heat transfer characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements. The working media was a viscous fluid with intermediate Prandtl number. Regularly spaced twisted tape elements connected by circular rods were used to cause fluid turbulence and increase the heat transfer rate. The parameters varied were tape width and the rod diameter. The test section was heated electrically under constant heat flux condition. It was proposed that reducing the width of twisted tapes resulted in lower enhancement in heat transfer rate. Tests were also conducted for studying the effect of phase angle. It was observed that higher than zero phase angle does not contribute any how towards increase in heat transfer rate but it involves complex design and hence, its use is not preferable. It was also proposed that twisted elements should be pinched rather than being connected by circular rods, as that would increase the thermo hydraulic performance of the heat exchanger

Akpinar et al. [2004] studied the effect of placing swirl generator in the path of fluid in a double pipe heat exchanger. The experimental analysis was performed for determining heat transfer rates, friction losses and exergy loss. Swirl generators with different number of holes and different diameter were employed. The fluid flowing through the inner pipe was hot air and the one flowing through annulus was cold water. Experimental work was performed for Reynolds number varying from 8500 to 17000 for both the parallel flow arrangement and counter flow arrangements. The results obtained after modification were compared with the results without introduction of swirl generators..The value of Nusslet number increased up to 130 %..The increase in Nusslet number for 20 hole arrangement with hole diameter 6 mm was found to be 113 % and that for arrangement of 20 holes with 9 mm diameter was 109%.. The exergy loss increased with increase in number of holes in the swirl generator and decreased with increase in hole diameter.

Viedma et al. [2005] worked on determination of thermo hydraulic characteristics in laminar, transition and turbulent flow in circular tube with helical wire coils as inserts. The tests were conducted for two fluids, water and ethylene glycol. Six different wire coils were tested. During experiment, the Reynolds number was varied from 80 to 90,000 and Prandtl number was varied from 2.8 to 150. In laminar region, the performance improved with Reynolds number. After introduction of wire coil inserts at Re = 500, heat transfer increased from 30% to 40% as compared to plain pipe values. For the Reynolds number 1500, heat transfer increased from 140% to 160%. Wire coils increased pressure drop up to nine times and heat transfer up to four times compared to the smooth tube independently in turbulent flow. The best performance for the wire coil inserts was obtained in the transition region.

Alberto Garcia et al. [2007] studied enhancement of laminar and transitional flow heat transfer in tubes by means of wire coil inserts. Uniform heat flux condition was used for carrying out the experimental investigation. Three wire coil inserts having different pitches were used to investigate the effect on heat transfer and pressure drop characteristics. The tests were conducted for laminar and transitional regions. The Reynolds number was varied between 500 and 2500 and Prandtl number was varied from 200 to 700. After comparing the results with the values for plain pipe without inserts, it was proposed that at lower Values of Reynolds number lesser than 200, there was not any substantial increase in heat transfer and hence introduction of wire coil inserts is not a preferable option at lower Reynolds number .In the fully laminar region, the % increase in friction factor was found to be lying between 5 % and 40 % compared to plain tube. For the Reynolds number in the range 200 and 1000, the introduction of wire coil inserts have the most pronounced effect and there is a remarkable increase in heat transfer. At the Reynolds number value of around 1000, the increase in heat transfer was found to be 8 times the heat transfer without wire coil inserts. It was also suggested that the wire coil inserts perform better than the twisted tape inserts in the lower Reynolds number range.

Smith Eiasma ard et al. [2008] worked on experimentally studying turbulent flow heat transfer and pressure loss in a double pipe heat exchanger with louvered strip inserts. The louvered strip inserts were introduced in the inner pipe of the double pipe heat exchanger. Hot water and cold water were used as the working media in the inner pipe and annular section respectively. Experiments were conducted for both backward and forward arrangements of louvered strip inserts. Three different louver angles were used for both forward and backward configurations i.e. 15°, 25° and 30. Testing was done in turbulent flow regime by varying the Reynolds number between 6000 and 42000. Since the louvered strip inserts cause turbulence in the fluid, the heat transfer increased at the expense of increased pressure drop. The enhancement in heat transfer was determined based on increase in Nusselt number and increase in pressure drop was determined based on friction factor. The average increase in Nusselt number for forward configuration was 284 % and that for backward configuration was 263 % over the values for plain pipe without any insert. Similarly, the increase in friction factor for forward arrangement of louvered strips was found to be 413 % and that for backward configuration was 233 %. From the average increase in Nusselt number values and friction factor values, it is quite clear that although the forward inclined louver configuration provides higher heat transfer enhancement but it also causes very high pressure drop. On the other hand, although, the heat transfer enhancement for backward inclined louvers is lesser but it also causes low increase in pressure drop relatively. Hence , the backward inclined configuration provides overall better results for heat transfer enhancement purpose.

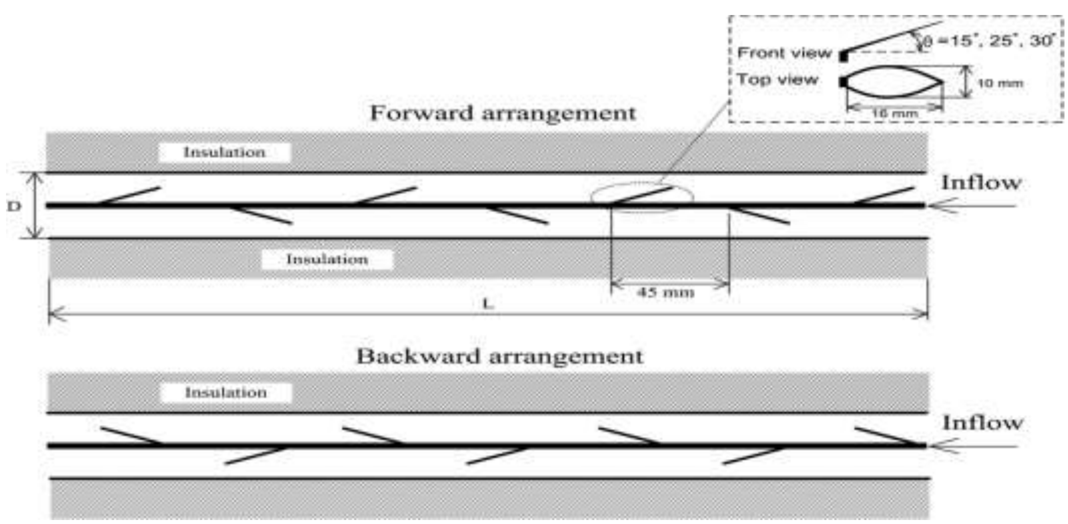

Figure 2.1: Louvered Strip inserts used by Smith Eisma ard et al. in experimental analysis

M.A. Akhavan-Behabad et al. [2010] performed experimental work for determining the Pressure drop and heat transfer augmentation due to coiled wire inserts during laminar flow of oil inside a horizontal tube. The experimental analysis was carried out for counter flow arrangement in the double pipe heat exchanger. The engine oil was made to flow through the inner tube and steam was made to flow through the annular section of the heat exchanger. In the first phase, no insert was used and the experiments were performed only for the plain tube. The results obtained for the plain pipe were used as the reference to compare the results obtained after introduction of coiled wires. Seven different coiled wires were used with their pitches ranging from 12-69 mm. The diameter of wires was also kept as a variable and two different diameters i.e. 2 mm and 3.5 mm were used. The effect of Reynolds number and geometry of wire coil on heat transfer and pressure drop characteristics was studied and the empirical correlations were obtained from the experimental data.

E.Z.Ibrahim [2011] performed experimental analysis for studying the augmentation of laminar flow and heat transfer in flat tubes by means of helical screw-tape inserts. The heat transfer characteristics and friction factor in the horizontal double pipes of flat tubes were determined. The inserts used were helical screw inserts having different twist ratios and different spacer lengths. Cold and hot water were used as working fluid in tube side and shell side respectively. The Reynolds number for experimental analysis was varied between 570 and 1310. It was observed that the Nusselt number and friction factor decreased with an increase in twist ratio and spacer length. In other words, the twist ratio and spacer length must be kept as low as possible in order to maximize the enhancement in heat transfer. The correlations of average Nusselt number and friction factor depending upon Reynolds number, spacer length and twist ratio were obtained.

Paisarn Naphon and Tanapon Suchana [2011] studied heat transfer enhancement and pressure drop of the horizontal concentric tube with twisted wires brush inserts. The inner diameter of the inner tube in heat exchanger was 15.78 mm and the inner diameter of outer tube was 25.40 mm. The wire brush insert was fabricated by twisting the copper wires having diameter 0.2 mm around the iron core rod having diameter 2 mm. Three different wire densities were used to perform the experimental analysis i.e. 100, 200, 300 wires per centimeter. The experiment was performed for two different arrangements. In the first set, the full length wire brush inserts were used to determine the heat transfer and pressure drop characteristics. In the second set, regularly spaced wire brush inserts spaced by 30 mm spacers were used. Hot water was used as the working fluid in the inner tube and cold water was used as the working fluid in the outer tube. The test was conducted for Reynolds number ranging between 6000 and 20000. It was observed that there is a significant increase in the heat transfer by inserting the wire brush insert but it also brought increase in pressure drop. Hence the optimum density depends upon the area of application i.e higher density of wire brush can be used if pressure drop or pumping power are of no great interest. Otherwise, wire brush insert having lower density must be used.

Zen Zhan et al. [2012] conducted experimental study for determination of heat transfer enhancement in double-pipe heat exchanger by means of rotor-assembled strands. Rotor assembled strands having different geometries and lead were used as inserts in the double pipe heat exchanger. First of all, the experiment was performed for the plain pipe without insert in order to calibrate the test setup and standard values were obtained for comparing the results. Then, the experiment was performed for various rotor assembled strands. Significant increase in Nusselt number was found after introduction of the strands. The Nusselt number was found to increase by around 37.4 to 74.8 % as compared to plain pipe and friction factor increased by around 71.5 to 123.1 % for various arrangements. The maximum gain in heat transfer was found for the helical blade rotor where as the maximum increase in friction loss was caused by blade discrete rotor. The Nusselt number and friction factor were found to be increasing with decrease in lead of rotor for helical blade rotor. If the heat transfer criteria is considered, best performance was given by helical blade rotors with ladders followed by helical blade rotor and blade discrete rotor. Considering the thermal performance, the helical blade rotors having ladders with diameter 22 mm and rotor lead 150 mm gave the best results. Based on the experimental results, correlation were proposed using multivariate linear normal regression method.

C. Thianpong et al.[2012] studied the effect of perforated twisted tapes with parallel wings on heat transfer enhancement in a heat exchanger tube. This type of insert was used in order to increase the heat transfer by causing the turbulence by wings and reducing the pressure drop caused by restriction to flow with the help of holes in twisted tapes. The experimental analysis was carried out for Reynolds number varying between 500 and 20500. The variable parameters were wing hole diameter ratio (*d/W* = 0.11, 0.33 and 0.55) and wing depth ratio (*w/W* = 0.11, 0.22 and 0.33). For the twisted tape inserts, increase in heat transfer was found to be 190 % compared to plain pipe and that for Perforated twisted tapes was 209 %.The maximum thermal performance factor was achieved for perforated twisted tape with wings at Reynolds number 5000 for *d/W* = 0.11 and *w/W* = .33. Empirical correlations were developed for heat transfer, friction factor and thermal performance factor. The flow patterns were visualized using dye injection technique.

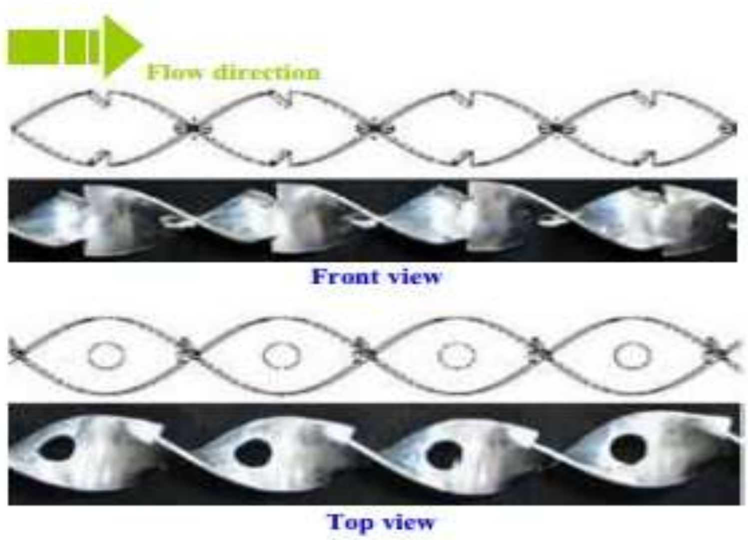

Figure 2.2: Perforated twisted tapes with parallel wings used by C. Thianpong et al.

A.W.Fan et al.[2012] conducted parametric study on turbulent heat transfer and flow characteristics in a circular tube fitted with louvered strip inserts. Numerical simulation was used to determine the effect of louvered strip inserts .The variables chosen for study were slant angle and pitch of the louvered strip inserts. Main attention was paid to the effects of the slant angle and pitch of the turbulators. Through numerical simulation, the Nusselt number was found to increase by around 2.7 to 4.05 times the Nusselt number for the plain pipe. The performance evaluation criteria was found to be lying in the range 1.60 to 2.05 for various arrangements of louvered strip inserts. It was found that the heat transfer increases with decrease in the pitch of louvered strips and also increases with increase in louver angle. But increase in pitch and louver angle also causes higher turbulence and restriction to fluid flow that result in higher pressure drop and friction factor values. It was also observed that the slant angle has more predominant effect on heat transfer rate as compared to the pitch of the insert. Considering both heat transfer criteria and low pressure drop, moderate slant angle with low pitch must be employed for obtaining best results from the louvered strip insets.

N. Targui and H. Kahalerras [2013] carried out numerical investigation for a double pipe heat exchanger in order to study the effect of porous baffles and flow pulsation on a double pipe heat exchanger performance. The hot fluid was made to flow in the inner cylinder, whereas the cold fluid circulated in the annulus of double pipe heat exchanger. The Darcy–Brinkman–Forchheimer model was used to describe the flow in the porous regions .The governing equations involved were solved by used the finite volume method under suitable boundary conditions. The permeability of porous baffles was varied to know its effect on the performance of heat exchanger The effects of the amplitude and frequency of pulsation on heat exchanger efficiency were also analyzed. It was proposed that there is a significant increase in heat transfer rate under pulsating flow in comparison to the steady non pulsating flow. The maximum increase in heat transfer was found for the arrangement in which hot fluid was made to pulsate.

M.M.K. Bhuiya et al. [2013] carried out experimental analysis for heat transfer and friction factor characteristics in turbulent flow through a tube fitted with perforated twisted tape inserts . This porosity of the perforated twisted tape was chosen as the variable and whole experiment was performed for four different porosities of the perforated twisted tape i.e. 1.6, 4.5, 8.9 and 14.7%. Reynolds number ranging from 7200 to 49800 was used in the turbulent flow regime. Air was used as the working fluid under uniform wall heat flux boundary condition. It was observed and proposed that both the heat transfer rate and friction factor increases by the introduction of perforated twisted tapes in the heat exchanger. The increase in Nusselt number as compared to the plain pipe was found to be approximately 110-340 % and the increase in friction factor was around 110 – 360 % depending upon various porosities of the twisted tapes. Overall, the gain in the thermal performance factor was around 28 -59 %. Based upon the experimental data, empirical correlations were derived for the Nusselt number

M.M.K. Bhuiya et al. [2014] worked on studying the effects of the double counter twisted tapes on heat transfer and fluid friction characteristics in a heat exchanger tube. The double counter twisted tapes were installed as counter-swirl flow generators in the test section. Four different twist ratios were used to carry out the experimental analysis i.e. 1.95, 3.85, 5.92 and 7.75. Air was used as the working fluid in the experimentation. The work was carried out for turbulent flow by varying the Reynolds number from 6950 to 50050. It was observed that that the Nusselt number, friction factor and thermal enhancement efficiency were inversely proportional to the twist ratio of the twisted tape insert and the values of these entities increased by decreasing the twist ratio. It was found that the heat transfer rate in the tube fitted with double counter twisted tape was significantly increased. Since the introduction of twisted tape inserts cause a restriction to the fluid flow, hence pressure drop also increased. The heat transfer rate was found to be increased by around 60 to 240% and increase in friction factor was approximately 91 to 286% as compared to the plain without twisted tape inserts. The maximum thermal enhancement efficiency of 1.34 was achieved by the use of double counter twisted tapes at constant blower power. By using the experimental data, the empirical correlations for the Nusselt number, friction factor and thermal enhancement efficiency were proposed.

Darzi et al. [2014] determined the effect of nanoparticles and helical corrugation on turbulent heat transfer friction factor characteristics.. Thermal conductivity and viscosity of nanofluids with various volume concentrations were experimentally measured at different temperatures. Reynolds number and nano particle concentration was varied from 5000 to 20000 and 0% to 1% by volume, respectively. Nusselt number and friction factor of nano fluids were obtained for different concentrations as well as various Reynolds and Prandtl numbers. Results show that the heat transfer and friction factor increase by increasing nano fluid concentrations in plain and helical corrugated tube while its effect are more significant in helically corrugated pipes. The effect of nano particles on heat transfer is intensified at higher corrugation height and smaller corrugation pitch. For the same amount of pumping power, heat transfer rate was improved by increasing the nano particles concentration and by increasing the corrugation height or reducing the corrugation pitch. By using nano-particles and helical corrugation simultaneously , the heat transfer increased by 3.2 times the heat transfer for the plain tube.

Prasant W. Deshmukh and Rajinder P. Vedula[2014] studied heat transfer and friction factor characteristics of turbulent flow through a circular tube fitted with vortex generator inserts. The vortex generator was fabricated by attaching curved delta wing vortex generators on a rod. The vortex generators were placed axially on the opposite sides of the central rod. The effect of pitch to projected length ratio, height to pipe inner diameter ratio and angle of attack on the heat transfer performance was determined experimentally. Air was used as the working fluid and the experiment was performed for turbulent region. The Reynolds number was varied from 10000 to 45000. For the same value of Reynolds number, the increase in Nusselt number was found to be around 1.3 to 1.5 times for various arrangements as compared to plain tube without inserts. Considering same amount of pumping power, the increase in Nusselt number varied from 1 to 1.8 times as compared to the tube without vortex generators. Empirical relations were derived for the dependence of Nusselt number on pitch to projected length ratio, height to tube inner diameter ratio and angle of attack.

Hamed Sadighi Dizaji et al. [2015] performed experimental analysis for studying the effect of corrugated outer and inner pipes of a double pipe heat exchanger on the heat transfer and pressure drop characteristics. The heat exchanger was fabricated by obtaining the corrugations on both inner and outer pipes with the help of a special machine. Both convex and concave corrugated arrangements were fabricated to carry out the experimental investigation.. Wilson plots were used to determine the heat transfer coefficient. Water was used as the working media in both the inner and outer pipes of the heat exchanger. In the inner pipe, hot water was made to flow at the temperature 40 °C and in the annulus, cold water as made to flow at the temperature 8 °C. Reynolds number was varied in the range of 3500–18000 based on the hydraulic diameter of the annular space between the two pipes. The Reynolds number for hot water flowing through the inner pipe was kept constant at 5500. Findings indicated that the outer pipe corrugations and arrangement type of corrugated pipes have significant effect on thermal and frictional characteristics. Maximum effectiveness was obtained for heat exchanger made of concave corrugated outer pipe and convex corrugated inner pipe.

Kumbhar D.G. and Sane N.K[2015] conducted experimental work for exploring heat transfer and friction factor performance of a dimpled tube equipped with regularly spaced twisted tape insert. Water was used as the working media in the experimental setup and Reynolds number of fluid was varied from 4200 to 16000. Three different twist ratios of twisted tapes were used i.e. 2, 3 and 4. Also regularly spaced twisted tapes with same twist ratios and two different free space ratios i.e. 4.5 and 9 equipped with dimpled tube with two pitch ratios (P=1.5, 3.0) were tested in heat exchanger. The benchmark values were obtained and then the results were compared for twisted tapes and combination of twisted tapes and dimpled tube. It was found that although, there is always heat transfer enhancement brought about by modifying the heat exchanger, but the maximum heat transfer enhancement is found for the combination of twisted tapes and dimpled pipe.

Zaid S. Kareem et al. [2015] conducted experimental and numerical study for Heat transfer enhancement in three-start spirally corrugated tube. The study was conducted in the region of low Reynolds number .The tube that was used for the study was spirally corrugated. Firstly, the results were obtained using numerical study for the corrugated tube. After that, the experimental analysis was performed to determine the actual values of Nusselt number and friction factor for the spirally corrugated tube. On comparison with the standard data for plain tube, it was observed that the Nusselt number increased by 2.4-3.7 times the plain pipe. The increase in friction factor was found to be 1.7-2.4 times the plain tube values. The best thermal performance of 1.8-3.4 in the Reynolds number ranging between 100-1300 was found for the tube with severity index 36.364×10^{-3}. Thus, it was concluded that the change in the profile of the corrugated tube can substantially increase the heat transfer rates.

Suriya Chokphoemphun et al. [2015] carried out experimental investigation on enhanced heat transfer and pressure loss characteristics by using single, double, triple, and quadruple twisted-tape inserts in a round tube having a uniform heat-fluxed wall. Air was used as the working media and the experiments were performed for various twisted tape inserts for co and counter twisted arrangements by varying the Reynolds number from 5300 to 24000. The typical single twisted-tape inserts at two twist ratios, 4 and 5, were used as the base case, while the other multiple twisted-tape inserts had twist ratio of 4 only. It was observed that the values of Nusselt number increased with increase in Reynolds number and the increase in number of twisted tapes. The pressure drop also followed a similar trend and friction factor also increased with increase in number of twisted tapes. The increase in values of Nusselt number was found to be 1.15 times compared to plain tube and increase in friction factor was around 1.9–4.1 times. The thermal enhancement factor of the inserted pipe under similar pumping power was found to be above unity for all the arrangements unity except for the single and the double co-twisted tapes. The maximum enhancement in heat transfer characteristics was found for quadruple counter-twisted tape inserts.

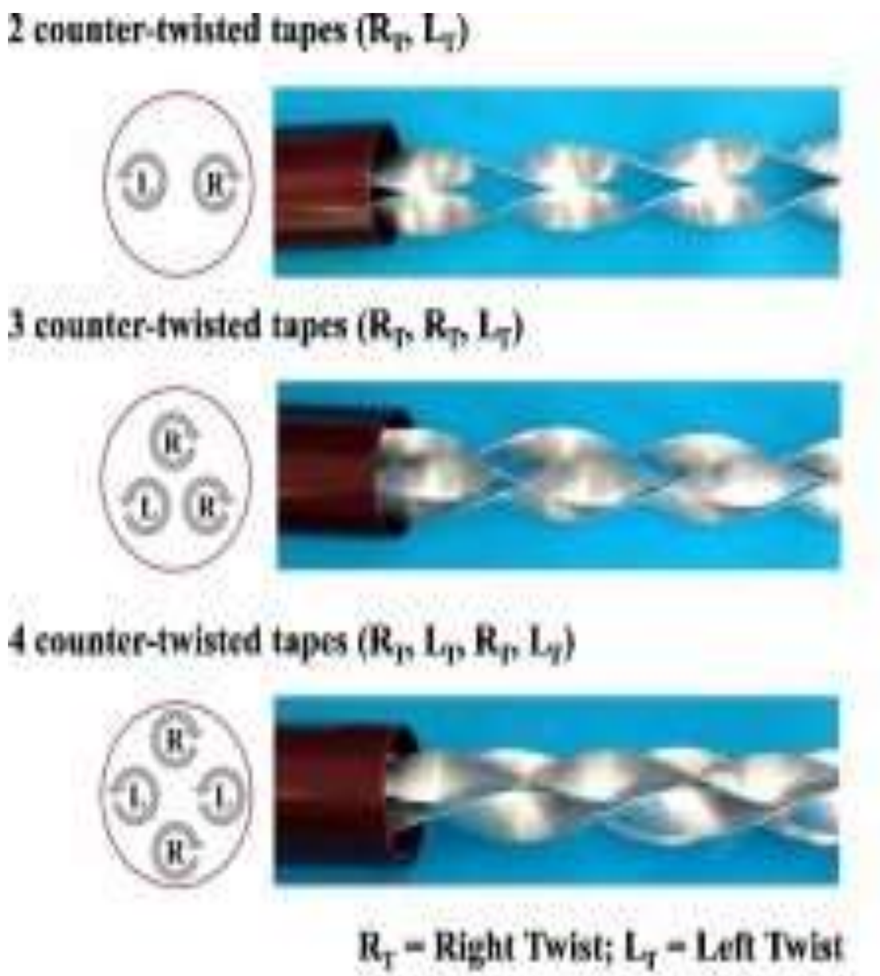

Figure 2.3: Different arrangements of twisted tape inserts used by

Suriya Chokphoemphun et al.

Wenbin Tu et al.[2016] carried out numerical investigation for turbulent flow in a circular tube modified with pipe inserts. The Reynolds Number was varied in the range 2892–28915. Water was used as the working fluid. It was proposed that the heat transfer rate increases substantially after introduction of pipe inserts. It was suggested that the heat transfer rate decreases with the increase in length of spacer and flow resistance increases with the decrease in length of spacer. Hence a suitable spacer length must be considered depending upon the application. Comparison was also made between three pipe inserts and four pipe inserts and four pipe inserts performance was found to be better than three pipe inserts. For the four pipe inserts, the values of performance evaluation criterion were proposed to be lying in the range approximately 1.4–3.0. It was also suggested that the performance of pipe inserts also depends upon installation angle between insert nodes. The installation angle has very little effect on increase in pressure drop and it can be used accordingly to increase the heat transfer rate.

Chaitanya Vashistha et al. [2016] carried out the experimental investigation for determining the performance of various inserts in a circular tube that was fitted with multiple inserts arranged in co-swirl and counter-swirl orientations. The experimental analysis was carried out to determine the change in heat transfer brought by single, twin and four twisted tapes inserts by keeping the twist ratios as 2.5, 3 and 3.5 .The Reynolds number was varied in the range 4000 to 14000.It was found that the there was a significant increase in heat transfer after introduction of twisted tape inserts along with increase in pressure drop due to the restriction caused by twisted tapes to the fluid flow. The maximum enhancement in the heat transfer and friction factor was found to be 2.42 and 6.96 times as compared to the smooth tube while the maximum value of the thermo-hydraulic performance factor was found to be 1.26 for a set of four counter-swirl twisted tapes with the twist ratio of 2.5. Correlations were obtained for Nusselt number and friction factor as function of Reynolds number and twist ratio for different twisted tape configurations, friction factor and thermal performance factor showing their dependence upon the porosity..

Varun et al.[2016] have reviewed the research work of various researchers conducted in the field of heat transfer augmentation of heat exchangers by introducing twisted tape inserts. They have compared the results of various researchers that were obtained experimentally as well as analytically. The empirical relations proposed by previous researchers have been compared in order to know about the similarity between them. It has been proposed that research needs to be continued in field of optimization of shapes of twisted tape inserts in order to maximize the heat transfer and minimize the friction losses after introduction of twisted tapes.

P V Durga Prasad et al. [2016] performed experimental investigation in order to enhance the rate of heat transfer in a U-pipe heat exchanger modified with twisted tape insert with Al_2O_3 nanofluid flowing through the heat exchanger. The experimental investigation was performed for two different volume concentrations of the nano fluid , 0.01 % and 0.03 %. Two twist ratio swere used for the twisted tape insert i.e. 5 and 20 respectively. The Reynolds number was varied from 3000 to 30000. The increase in Nusslet number was found to be 31.28 % for the volume concentration of 0.03 % and twist ratio of 5. Correlations have been proposed in order to estimate the Nusselt number and friction factor. It has been concluded that there is significant increase in heat transfer with twisted tape inserts and friction factor also increases.

Chapter 3

Gap study and Research Objectives

3.1 Gap study: The research has been going on in the field of heat transfer enhancement of heat exchangers for several years. Different researchers have proposed so many different methods to enhance the heat transfer of an existing heat exchanger or to bring some modifications in the design of new heat exchanger. Talking about the field of introduction of turbulators, a lot of literature is available regarding numerical and experimental analysis of various turbulators such as twisted tape inserts, louvered strip inserts and wire mesh inserts. After going through literature, following key points were identified which needs further investigation :

1. Different researchers have used different turbulators in their studies but a comprehensive analysis of comparison between various turbulators is not available

2. Since, there are no standard equations available for tabulators, the most of the literature is based on the experimental analysis and variation in results is found for different researchers.

3. The comparison has not been made for different turbulators at the same platform i.e. on the same setup under same experimental conditions

3.2 Research Objectives: The present study deals with the experimental investigation into performance characteristics of the double pipe heat exchanger modified with introduction of turbulators. These turbulators come under the category of passive heat transfer enhancement techniques. The main objectives of the study can be summarized as under;

a) To perform the experimental investigation on double pipe heat exchanger without any turbulator and then with introduction of turbulators and compare the results obtained from two studies to determine the % increase in heat transfer enhancement

b) To study the effect of Reynolds number on heat transfer rate with introduction of turbulators such as spring inserts, twisted wire mesh inserts and twisted tape inserts

c) To compare the performance of various turbulators on the basis of increase in heat transfer enhancement

d) Comparison of turbulators on the basis of pressure drop values obtained from experimental investigation as the turbulators such as twisted tape inserts cause a substantial amount of pressure drop. This pressure drop must be considered before employing such an insert in real life problems. The increased pressure drop will mean that more amount of pumping power will be required which in turn will increase the operating cost and on the whole, there may be no use of introduction of the insert.

e) Comparison among the different variations of a single turbulator. For example spring inserts were compared on basis of different pitches i.e. 15 mm, 10 mm and 5 mm. Similarly, twisted tape insert were compared on basis of twist ratio as 5, 3.75 and 2.5 respectively. The effects of twisted wire mesh inserts compared on the basis of twist ratios 5, 3.75 an 2.5 respectively.

Chapter 4

Methodology

There are no standard results available for the heat transfer augmentation techniques as it is still a new field and also different researchers propose different geometries . In order to determine the exact effect of heat transfer enhancement methods, the only option left is to determine the heat transfer and pressure drop characteristics experimentally. Then, the results obtained from the experimental investigation can be compared with theoretical values. Same procedure has been used in this study. Experimental analysis has been carried out for the following heat transfer augmentation methods

1. Heat exchanger modified with spring
   inserts having different pitches
2. Twisted wire mesh inserts with different twist ratios
3. Twisted tape inserts with different twist ratios

The experimental studies are carried out on a setup which was fabricated in the heat transfer laboratory and the details of the setup are given as under:

4.1 Experimental setup

To perform the experimental investigation , the test setup was fabricated at the Laboratory of Heat transfer, Thapar University Patiala. Following are the component details of the fabricated test setup and other utilities that have been involved in this study:

1. Support structure: Due to involvement of large number of components such as pumps, rotameters, temperature display, hot and cold water reservoirs etc, a separate support structure was required to mount each of these components at their respective position. Also, some of the components were very delicate such as U tube manometer and resistance temperature detectors and their care full mounting was utmost necessary as they could not be moved at time to time as that would have lead to unnecessary disruptions. The support structure was fabricated using iron channels and steel sheet and all the components were mounted on the support structure.

2. Double pipe heat exchanger: The simplest form of heat exchanger i.e. double pipe heat exchanger has been used in the present study as the selection of a different heat exchanger would have involved lot more complexities due to difficulty in introduction of turbulators. The heat exchanger was fabricated after finalising the dimensions and material. The material of the outer pipe of heat exchanger is Galvanized iron and it has been chosen because of its resistance toward corrosion as the experimental procedure involves long time study and corrosion would have affected the performance of heat exchanger. Also, since the outer pipe has no constraints for thermal conductivity, hence galvanized iron pipe was selected. For the inner pipe, the material is copper. Copper is chosen because it has very high thermal conductivity and the inner pipe surface is solely responsible for the whole heat transfer phenomenon. Copper also is the most widely used material in heat exchanger industry and has good resistance against fouling. Hence inner pipe material is chosen as copper. The dimensions of pipes of heat exchanger are as under

Inner diameter of copper pipe: 12 mm

Outer diameter of Copper pipe: 16 mm

Figure 4.1: Inner copper pipe used in heat exchanger

The dimensions of the galvanized iron outer pipe are:

Inner diameter of outer pipe: 25.4 mm

Outer diameter of outer pipe:28.4 mm

Figure 4.2: Galvanized iron outer pipe of heat exchanger

The length of the double pipe heat exchanger is 1500 mm. Connectors are provided at both ends of inner and outer pipes for easy dismantling so that inserts can be easily introduced into the exchanger. The heat exchanger is insulated from surroundings using an asbestos rope so that there are no heat losses which can cause an error in the experimental readings. Also, the radiation heat losses are minimised by providing the outer surface of galvanized iron pipe with aluminium tape. The fabricated heat exchanger is shown as under:

Figure 4.3: Fabricated heat exchanger

3. Hot fluid supply: The hot water to the test section is supplied from the hot water tank which is a stainless steel tank that was fitted with a 3 KW power heating element. The heating element was also connected to a PID temperature controller in order to keep control on the maximum temperature of the hot water as we are to provide the water at inlet of heat exchanger at 80℃. The hot fluid tank is insulated from the surroundings first with the help of asbestos rope and then ,it is covered with aluminium tape to minimize the heat losses .The hot fluid flows in a closed cycle i.e. hot fluid is pumped from the hot fluid tank into the test section and once the heat transfer has taken place, the hot fluid again returns to the hot fluid tank.

4.Heatingcoil: A heating coil has been provided in the hot fluid tank to attain the temperature of water that is to be supplied at the inlet of inner pipe of heat exchanger. The capacity of heating coil was calculated as

Amount of Fluid (water)to be heated = 4 kg

Specific heat of water = 4.18 KJ/Kg K

Temperature difference i.e. (temperature at outlet of hot fluid tank –ambient temperature of water) = 80- 25 = 55 ℃ approx.

Amount of heat required =m * c_p * Δt = 4 * 4.18 * 55 = 920 KJ

Assuming that water has to reach the desired temperature in 5 minutes time,

Capacity of heating coil required = 920/5*60 = 3.06 kW

Hence a heating coil having capacity 3 KW is provided in the hot fluid tank.

Figure 4.4: Heating coil provided in hot fluid tank

5. Cold fluid supply: The cold fluid i.e. water is supplied at the ambient temperature from the supply line. Monoset pump is used to circulate the cold fluid. The cold fluid is not circulated as there is no provision of a chiller and once the cold fluid passes through the exchanger, it is diverted towards the drain side.

6. PID temperature controller: In order to maintain the constant temperature of the hot fluid, PID temperature controller has been installed along with the equipment. Since, the hot fluid to the heat exchanger is to be supplied at the constant temperature of 80 ℃ . Once the temperature of water reaches the set limit, the heater gets cut off automatically .

Figure 4.5: PID temperature controller used in test setup

7. Pumps: For pumping the hot and cold fluids into the test section two self priming monoset pumps have been provided in the test setup. The pumps were connected to the setup through the bye pass valves to control to mass flow rate to the test section .The technical specifications of two pumps are as under:

Power of the pump: 0.125 HP

RPM: 2780

Head:6 m, 12 m

Disharge: 1400 LPH , 600 LPH

Figure 4.6: Self priming monoset pump used for circulating fluids

8. Rotameters: 2 Rotameters have been provided in the test setup for measuring the mass flow rates of the hot and cold fluid flowing through inner pipe and annular sections respectively. The whole experiment was performed for constant mass flow rate for the cold fluid. The mass flow rate of the hot fluid was varied in order to achieve the desired value of Reynolds number

Flow range of hot fluid rotameter:0-10 lpm

Flow range of cold fluid rotameter:0-5 lpm

Rotameters were provided with the control valves that could be used to alter the mass flow rate and thereby controlling the Reynolds number of the hot fluid flowing through the inner pipe. Rotameters were calibrated before each set of the experiments to apply the corrections required if any.

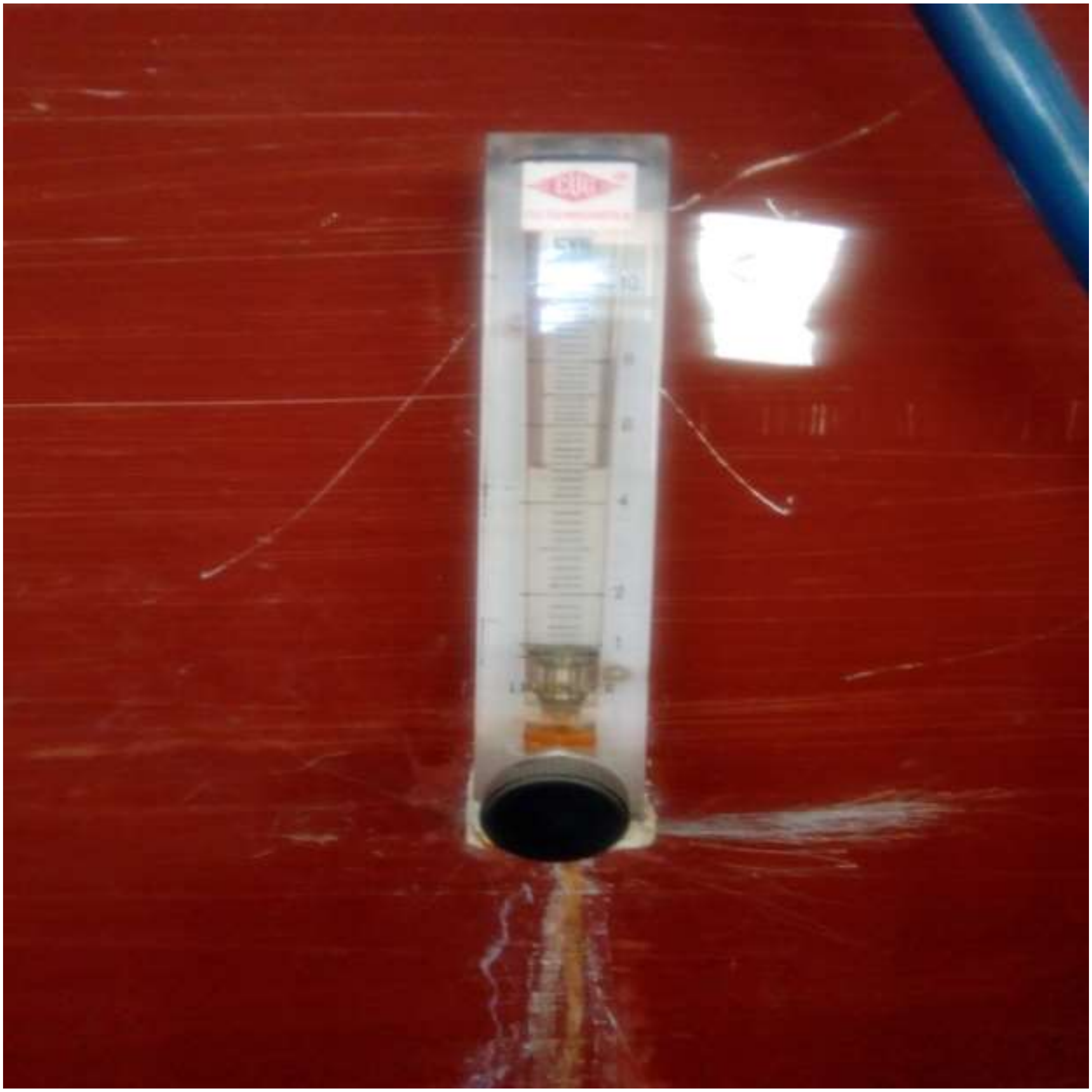

Figure 4.7: Rotameter used in experimental investigation

9. **Resistance temperature detectors**: In order to measure the inlet and outlet temperatures of hot and cold fluids flowing through the heat exchanger, four RTD were installed by providing the fixtures for installing the RTD. Thermal paste was used around the RTD to obtain the exact results. RTD were used for experimental purpose as the range in which RTD can function with great accuracy is -50 to 300 ℃ where as the other temperature measuring equipments such as thermocouples give poor repeatability which would have posed certain problems in this study.

10. Temperature display Screen and Selector switch: The output from the RTD is displayed on a digital screen. Since we are using four RTD in this experiment, the temperature to be displayed on the screen must be chosen manually. Hence a digital panel meter with selector switch was used in the experimental setup. The RTD on inlet of hot fluid was marked as 1, that on outlet of hot fluid was marked as 2. Similarly, the RTD on inlet and outlet of cold fluid were marked as 3 and 4 respectively. We could choose the temperature to be shown on the screen from the selector switch.

Figure 4.8: Temperature display and selector switch in digital panel meter

11.Measurement of Pressure Drop across hot fluid: Since the pumping power is also of great interest in heat exchanger industry , the pressure drop across the hot fluid must be kept in mind before bringing any sort of modification in an existing heat exchanger. Since the present study deals with introduction of external components ie. turbulators or inserts, it is obvious that that pressure drop across the hot fluid will vary from the one that occurs in the plain pipe. To measure this change in pressure drop for different turbulators, a U tube manometer has been provided in the test setup across the hot fluid section. The manometric fluid is chosen as mercury.

Figure 4.9: U-tube manometer with mercury as manometric fluid for pressure drop measurement.

12. Twisted tape inserts: Twisted tape inserts are the type of turbulators which induce turbulence along the fluid path due to their spiral shape. Twisted tape inserts were fabricated in the machine shop at Thapar University, Patiala. The material for the twisted tape was chosen as steel as it does not break during the twisting operation. The steel strips having width 10 mm were cut fom the steel sheet having thickness 1 mm. The width is intentionally kept as 10 mm to avoid any direct contact between the twisted tape insert and the inner copper pipe of the heat exchanger. The cut strips were held in the tool post of lathe machine. The one end of the strip or tape was kept stationery and the other end was rotated slowly by keeping the strip under adequate tention to avoid the distortion .The strips were twisted till the desired pitches and desired twist ratios were obtained. The parameters of twisted tape inserts can be summarized as :

Width of twisted tape: 10 mm

Thickness of twisted tape: 1mm

Pitch of twisted tape insert 1: 60

Twist ratio of twisted tape insert 1: 5

Pitch of twisted tape insert 2: 45

Twist ratio of twisted tape insert 2: 3.75

Pitch of twisted tape insert 3: 30

Twist ratio of twisted tape insert 3: 2.5

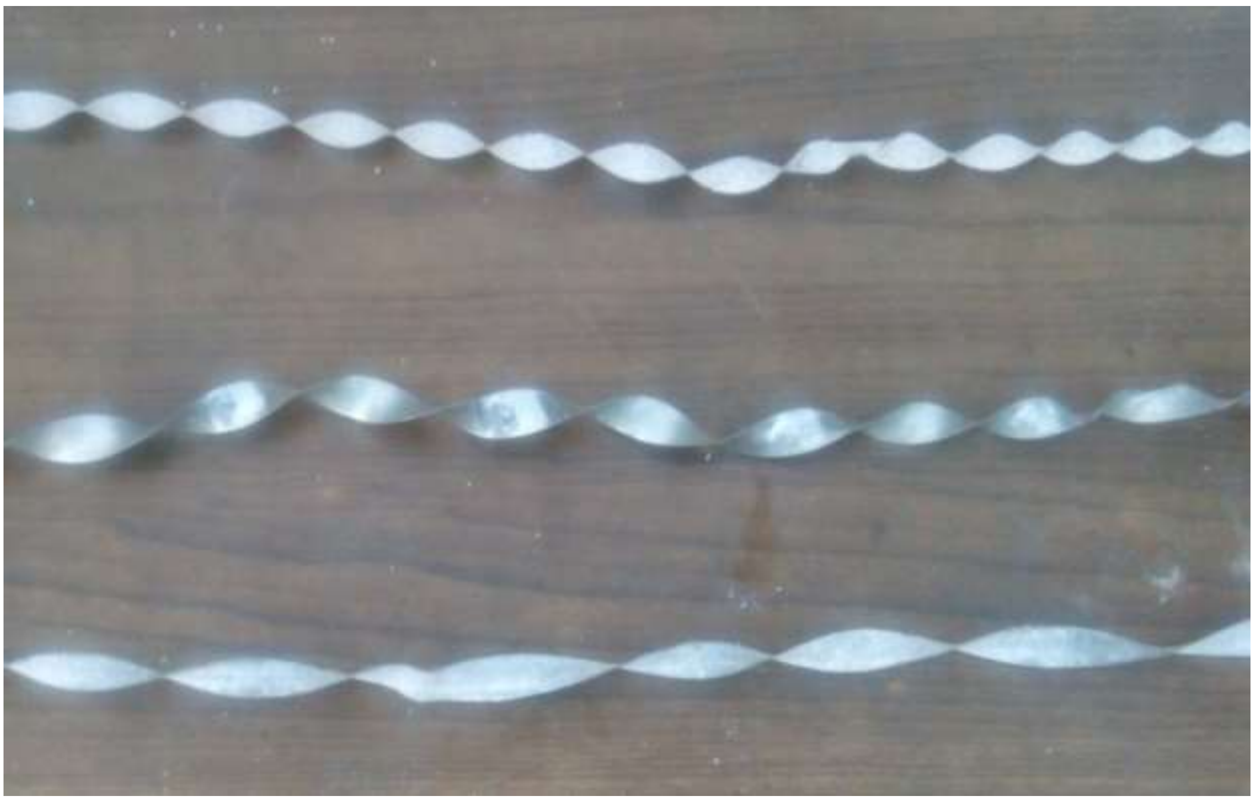

Figure 4.10: Twisted tape inserts

13. Spring inserts: Three different springs inserts are used in the experimental analysis. The spring inserts were procured from the market by specifying the diameter of the spring and its pitch. The diameter of all the spring inserts was kept constant i.e. 10 mm as the inner pipe diameter of heat exchanger is 12 mm and there must be a sufficient gap between the pipe and insert so that conduction heat transfer does not affect the experimental results. The wire diameter of the springs for all three pitches was 1 mm. Following are the pitches of three spring inserts

Spring 1: 15 mm

Spring 2: 10 mm

Spring 3: 5 mm

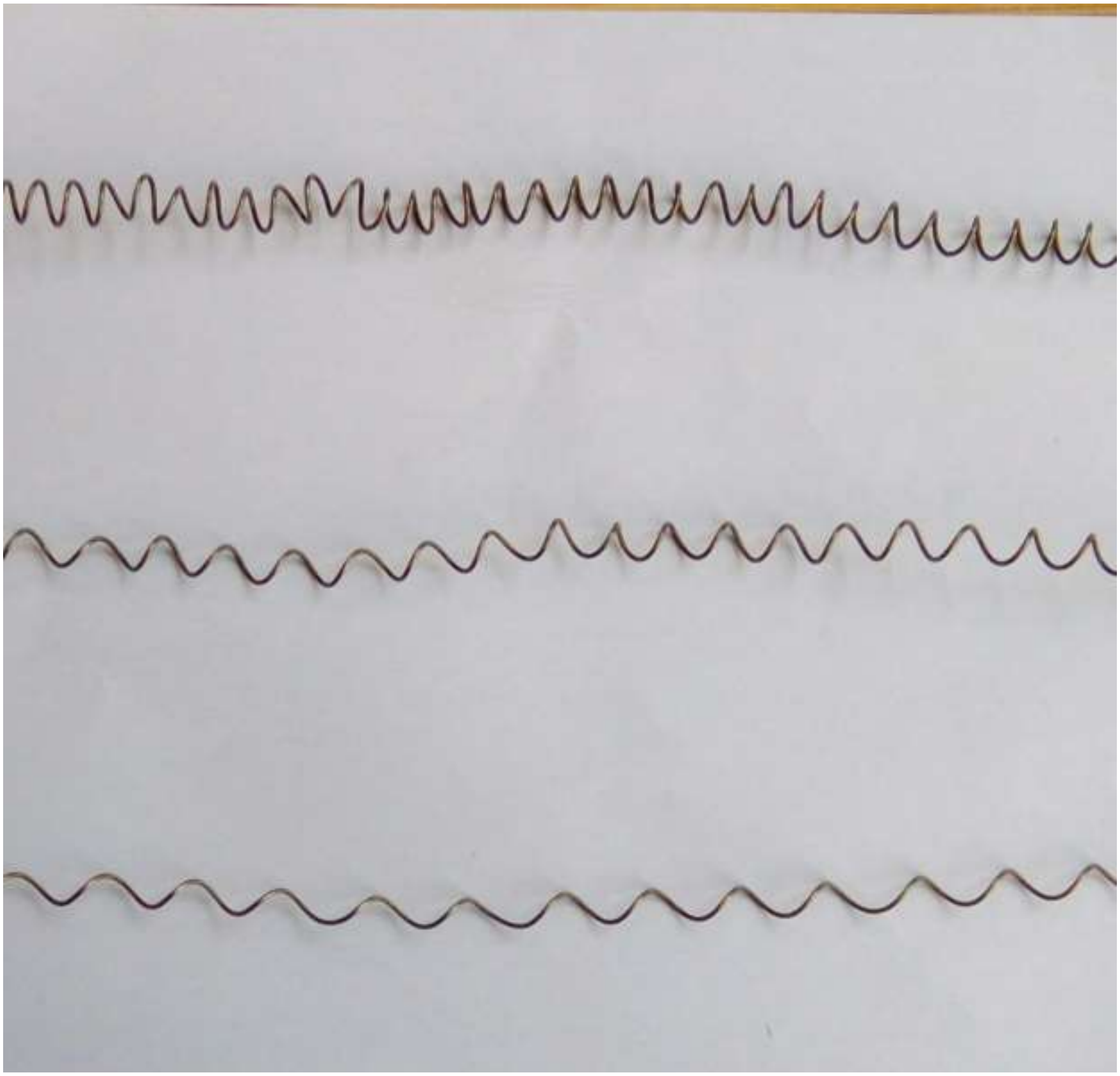

Figure 4.11: Spring inserts

14. Wire mesh insert: The third type of insert used in the experiment were twisted wire mesh inserts. Steel wire mesh having mesh size 2 mm has been used in the experiment. Twisted wire mesh inserts were fabricated by cutting the mesh strips having width 10 mm and length 1.5 m. The mesh strips were twisted similar to the twisted tapes in order to achieve the following specifications:

Pitch of twisted wire mesh 1: 60 mm

Twist ratio of twisted wire Mesh 1: 5

Pitch of twisted wire mesh 2: 45 mm

Twist ratio of twisted wire Mesh 2: 3.75

Pitch of twisted wire mesh 3: 30 mm

Twist ratio of twisted wire Mesh 3: 2.5

Figure 4.12 : Twisted wire mesh insert

15. Final Experimental Setup: After arrangements of all the components of the test setup the final experimental setup was obtained for carrying out the experimental analysis.The setup was initially tested for any sort of leakages before actual experimental work. The layout of the experimental setup and the actual setup obtained after fabrication at Thapar University, Patiala are shown as under :

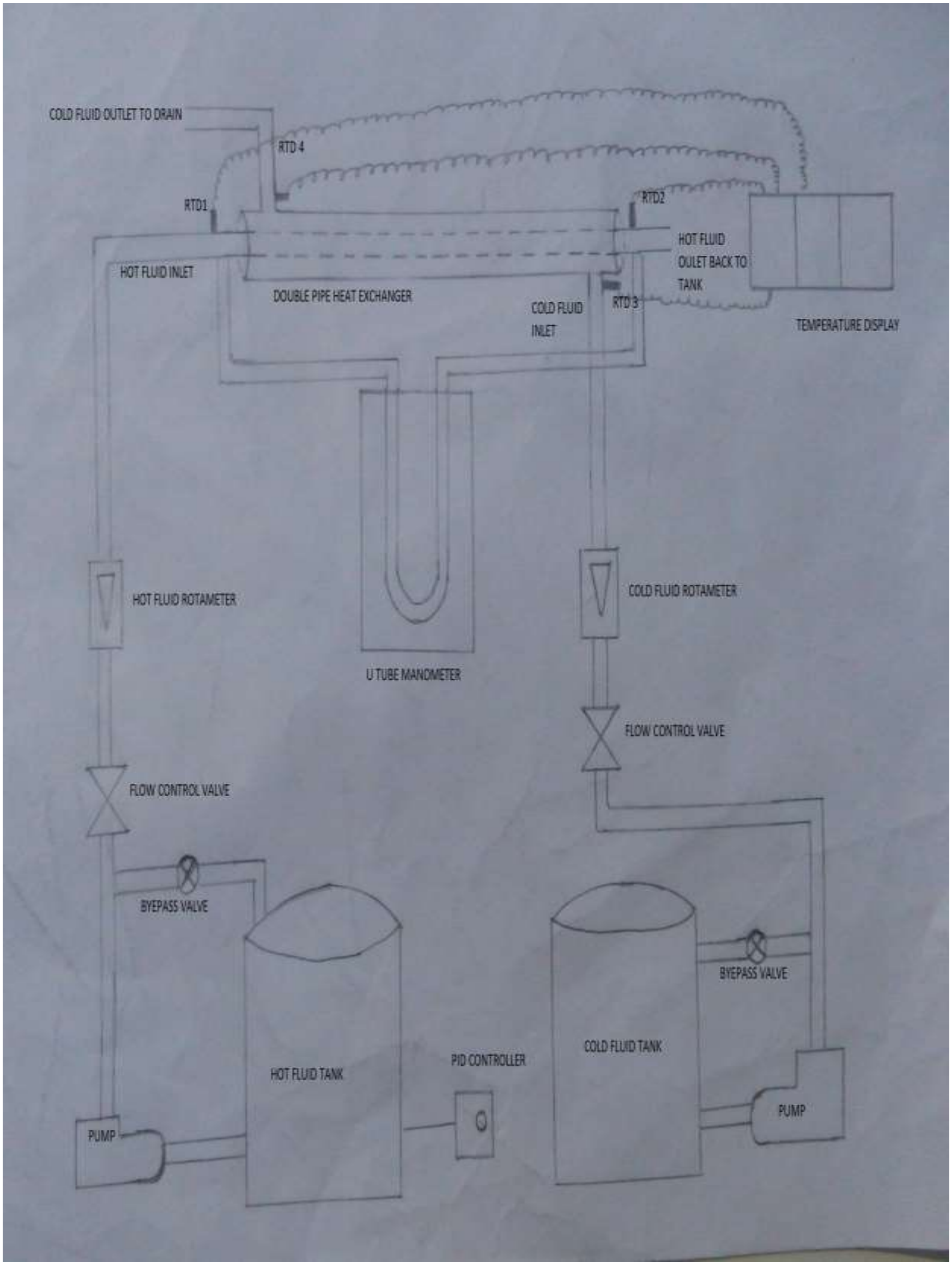

Figure 4.13: Experimental setup layout

Figure 4.14: Experimental setup fabricated at Thapar University, Patiala

Parameters in experimental investigation: Following are the parameters that were involved in the experimental procedure:

1. Reynolds number of hot fluid: The Reynolds number of hot fluid was varied between 5000 and 30000 for each type of insert. The desired values of Reynolds number was obtained by controlling the mass flow rate through the rotameter with the help of control valves and bye pass valve.

2. The cold fluid was supplied at constant rate of 5 lpm

3. Three types of spring inserts were used . The pitches of three spring insets were 15 mm, 10 mm and 5 mm respectively. The wire diameter of the spring inserts were kept constant for all the inserts.

4. The twist ratio of twisted tape inserts were varied .Three twist ratios i.e. 5, 3.75 and 2.5 were used.

5. The wire mesh inserts used also had three different twist ratios i.e. 5. 3.75 and 2.5.

Calibration of components: The experimental procedures included many sensitive components and it was of great importance to properly calibrate those components in order to ascertain that there are no errors in the final results that are based on experimental values. Both the hot fluid and cold fluid rotameters as well as four RTD were calibrated as under:

1.Calibration of rotameters: Rotameters were calibrated by measuring the discharge from the rotameter at a certain value in the beaker on which volume marking were written. The discharge was measured five times for each calibration. The average was obtained from the five observations to obtain the final value. The corrections were then applied during all the experiments to compensate for the error .

2. Calibration of RTD: For calibration of RTD , a constant temperature water bath was used. The RTD were dipped in the water bath and were kept inside the water bath for sufficient period to obtain the steady state. After that, the temperatures were noted from the display for all the four RTD. One of the values was kept as the reference and corrections were applied to rest of the RTD during all the experiments. This calibration was done periodically to cover any sort of damage occurred to the RTD while conducting the experiment.

Experimental Procedure followed: The experimental analysis was carried out at the Heat transfer Laboratory , Thapar University, Patiala. The details of experiment are as under:

1. After fabrication of experimental setup, the preliminary test run was conducted in order to eradicate any sort of leakages in the pumps, pipes and joints. Once, all the components, were running fine, the components such as rotameter and RTD were calibrated. The rotameters for both the hot and cold fluids were calibrated by measuring the discharge in a beaker having volume markings. The RTD were calibrated as discussed above necessary corrections were applied to adjust the errors in the equipments.

2. Standardization of the setup: Since the whole results of this study are based on the experimental analysis, it was necessary to test the performance of the test setup in order to determine whether the results obtained are in good correlation with the results available in literature. To standardize the test setup, the experimental procedure was carried out for the heat exchanger without any turbulator. The results of the heat transfer characteristics of double pipe heat exchanger are available according to Dittus Boelter equation. Similarly, we can get friction factor values from Blasius correlation. These theoretical values were compared with the results obtained from experimental procedure and the error was found to be within permissible limits

3. In the beginning of experiment, the heater was turned on to achieve the desired inlet temperature of the hot fluid. One the hot fluid reached the desired temperature, the pumps were turned on. Hot fluid was pumped in to the heat exchanger by the monoset pump. The discharge of hot fluid was controlled by the valves to achieve the certain value of Reynolds number. The hot fluid after passing through the heat exchanger was again made to flow into the hot fluid tank and it formed a closed cycle for the hot fluid. The cold fluid was pumped into the annulus of the heat exchanger by the monoset pump The cold fluid was made to flow at same discharge for all the experimental analysis. After exchanging the heat with the hot fluid, the cold fluid was not brought into the cold fluid tank as the chiller arrangement was not installed in the test setup. Hence , the cold fluid was ejected into the drain and the fresh cold fluid was circulated for each set of experiment.

4. Once both the hot and cold fluids were in circulation, the experiment was allowed to run so in order to achieve the steady state conditions. It usually took 25-30 minutes for the test section to reach the steady state conditions.

5. Once the steady state conditions used to occur, the temperatures of hot fluid at inlet and outlet of the heat exchanger were noted from the display screen by selecting the desired RTD from the selector switch. Similarly, temperatures of cold fluid at inlet and outlet were noted.

6. The pressure drop was only measured for the inner pipe of the heat exchanger by installing a U tube manometer across its ends. The pressure drop was obtained and it as further used to calculate the friction factor.

7. After each run, the flow rate of the hot fluid was varied and the experiments were conducted for Reynolds Number varying from 5000 to 30000 .

8. Similar approach was followed to carry out the experimental analysis for spring inserts, twisted tape inserts and twisted wire mesh insert. The calibration activity was performed at certain periodic instances in order to identify any component that could cause the error in the reading and hence the final results.

Validation of test setup and testing for repeatability: The whole experimental analysis in this study is based upon experimental values and an error in any of the instruments would mean wrong results. The validation of test setup was done by comparing the values obtained for the plain pipe with the theoretical values. The values of Nusselt number were compared with the values obtained from Dittus Boelter equation and values of friction factor were compared with friction factor values obtained from Blasius correlation. The parity was found between two results. In order to test the setup for repeatability, the tests were performed thrice for spring inserts having pitch 15 mm. The following graph was obtained for two test runs

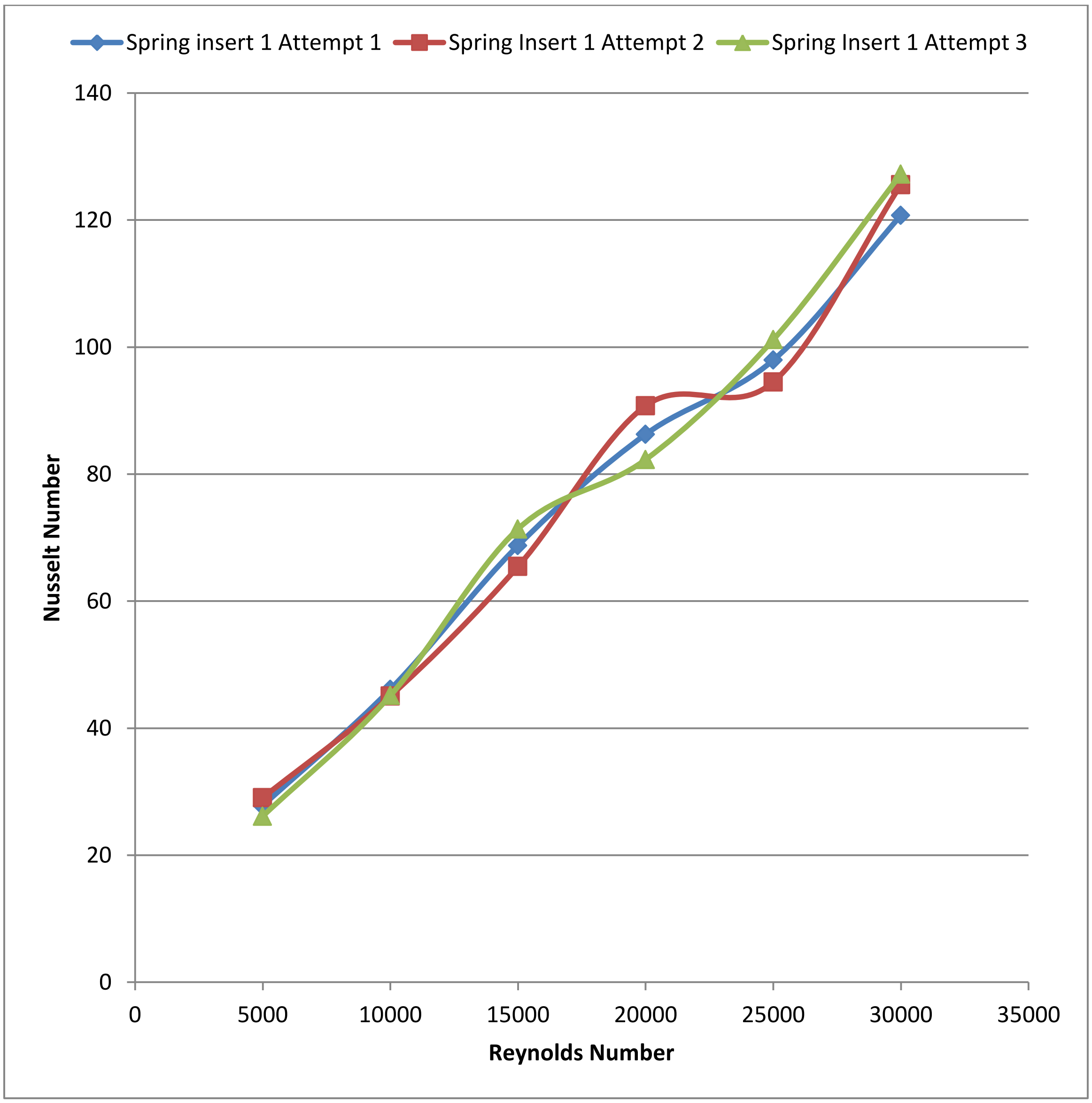

Figure 4.15 : Result of testing for repeatability of test setup

Precautions to be followed during experimental investigation:

1. All the inserts must be carefully fabricated to achieve exact pitch for spring insert and correct twist ratio for twisted tape inserts and twisted wire mesh inserts. During fabrication of twisted tape, care must be taken to avoid the distortion of twisted tape as that can lead to direct contact between the insert and pipe surface.
2. All the instruments such as rotameter and RTD must be calibrated before performing the actual experiment. The corrections required if any, must be applied prior before actual execution phase.
3. The manometric line should be free from any sort of air bubbles, otherwise the pressure drop readings will not be correct.
4. The system takes a certain minimum time before achieving steady state. All the readings must be noted once the system is in steady state and there are no variations in temperature sensor readings.
5. Spring inserts, twisted tape inserts and twisted wire mesh inserts should be carefully placed inside the inner pipe of the double pipe heat exchanger so that there is no direct contact between the insert and the pipe surface as any contact will mean heat transfer through conduction that will affect the results thus obtained.
6. RTD wires and other wiring connections should be periodically tested for firm joint. A loose connection can lead to inaccurate result.
7. The outer pipe of heat exchanger must be properly insulated in order to avoid the heat loss to the surroundings and leakages in hot and cold line must be avoided.

Chapter 5

Results & Discussions

The soul of the present study is based upon actual experimental investigation for the comparison of performance of various turbulators. The experimental analysis was performed for the plain pipe heat exchanger and the heat exchanger modified with various turbulators under similar conditions. Following results are obtained from the study:

1. Variation of Nusselt number for Spring Inserts:

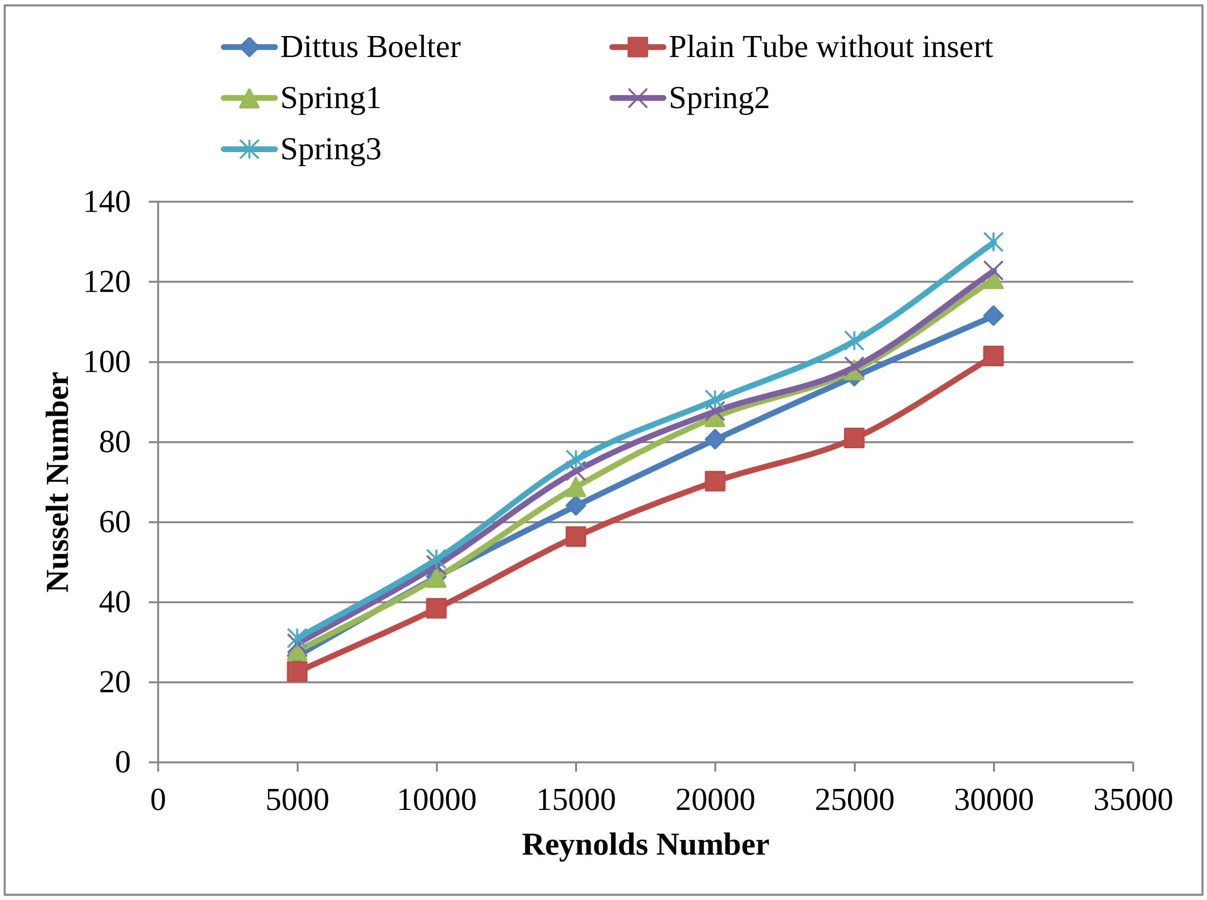

Figure 5.1: Nusslet number Plot for Dittus Boelter, Plain pipe without insert and spring inserts having pitches 15 mm, 10 mm and 5mm respectively

The graph shows the comparison between Nusselt number obtained from Ditttus boelter equation, Nusselt number for plain pipe without any insert and the experimental Nusselt numbers obtained for each spring insert. From the plot, it is clear that the Nusselt number increases with decrease in pitch of the spring i.e. lesser is the pitch, more will be the % increase in heat transfer. The variation in Nusselt number is between 19 % and 37 %. For the spring insert having pitch 5 mm for Reynolds number 5000, the increase in heat transfer is found to be 37 % as compared to the plain pipe without insert.

2. Variation of Friction factor for spring inserts:

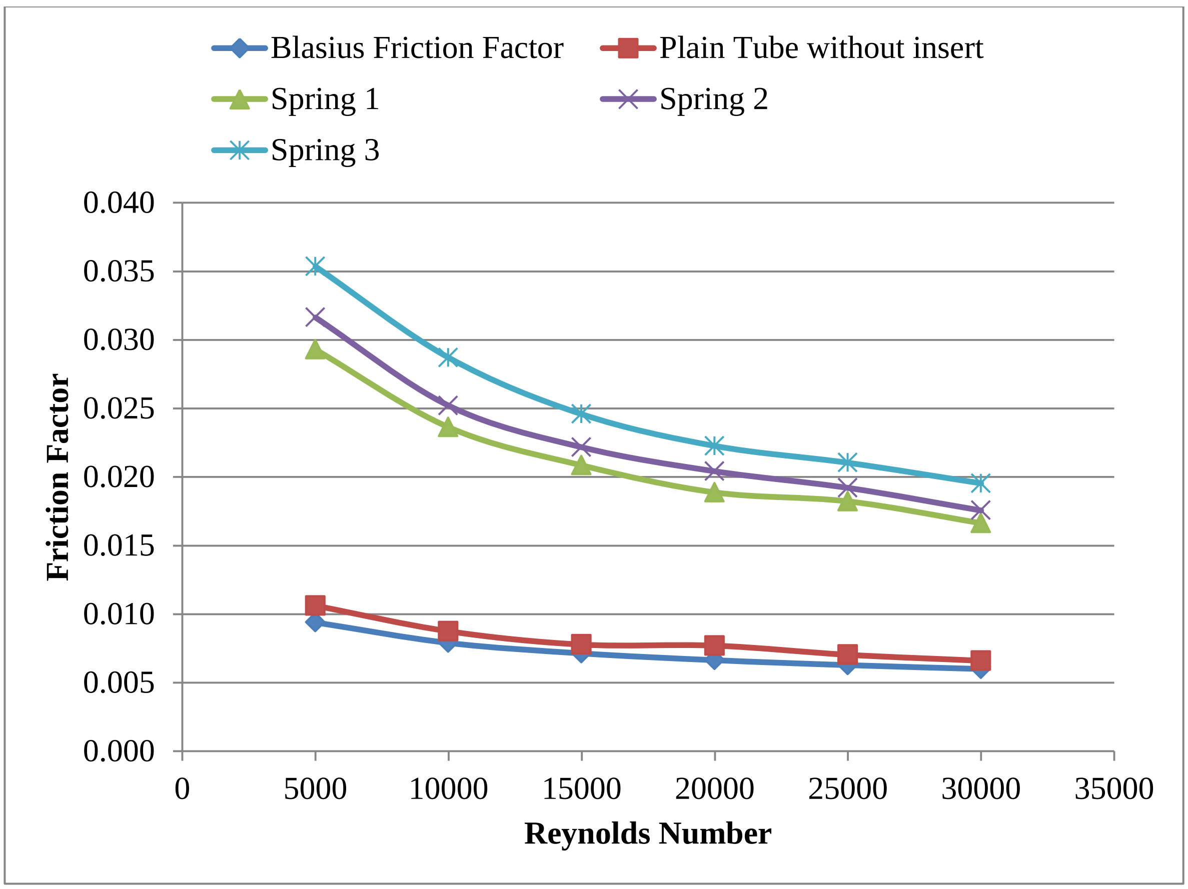

Figure 5.2: Variation of friction factor for Blasius correlation, plain pipe heat exchanger and heat exchanger modified with spring inserts having pitches 15 mm, 10 mm and 5 mm respectively

The plot represents the variation of friction factor for the double pipe heat exchanger based upon Blasisus correlation, friction factor values obtained experimentally for the plain pipe without spring inserts and for the heat exchanger modified with spring inserts. From the trend it is clear that the value of friction factor decreases with increase in Reynolds number. It can also be observed that friction factor increases substantially after introduction of turbulators. It is clear that increase in friction factor is inversely proportional to the pitch of the spring insert used. Lesser is the pitch of the spring insert, higher is the pressure drop and higher is the value of friction factor. Overall, the increase in friction factor varies from 152 % to 233 %. The maximum increase i.e. 233 % is found for the spring insert having pitch 5 mm at the Reynolds number 5000.

3. Variation of Nusselt number for twisted wire mesh inserts:

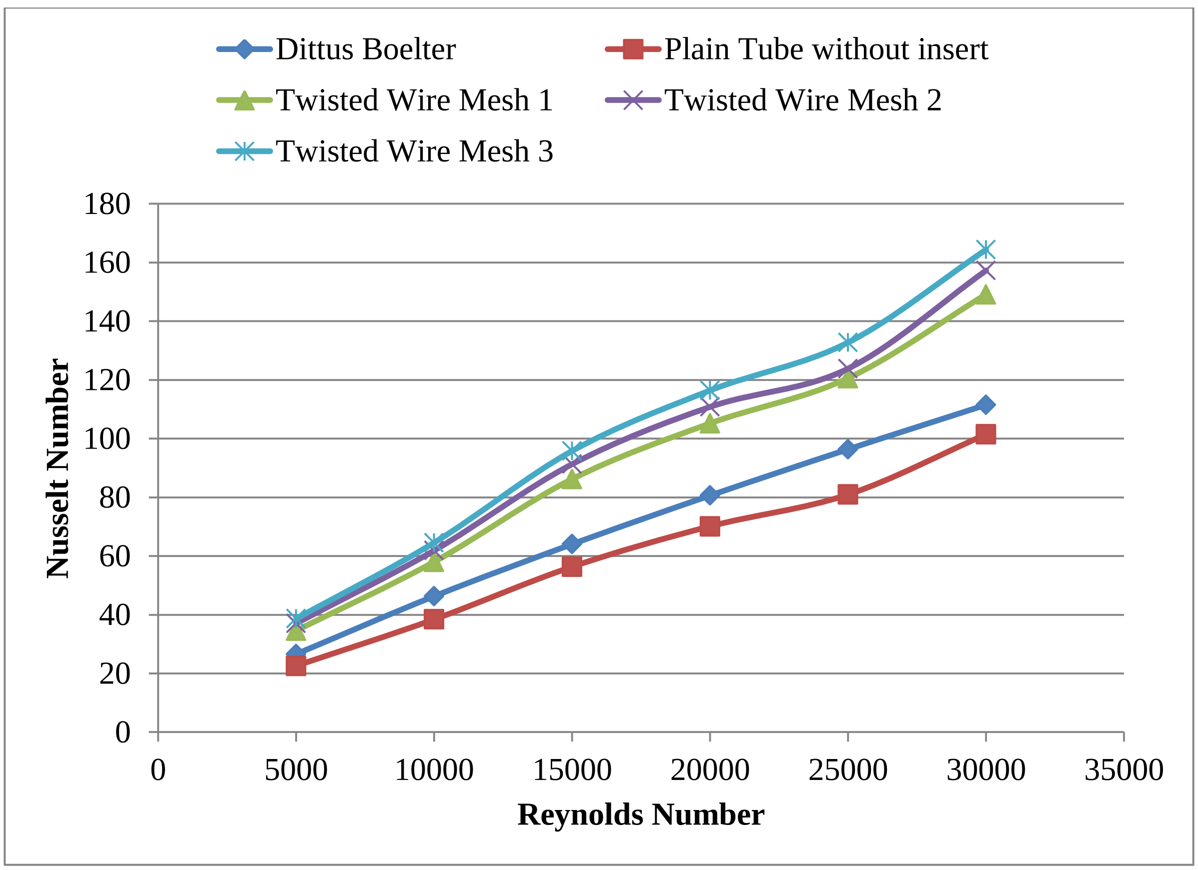

Figure 5.3: Nusslet number Plot for Dittus Boelter, Plain pipe without insert and twisted wire mesh inserts having twist ratios 5, 3.75 and 2.5 respectively

The graph shows the variation of Nusselt number obtained from Ditttus Boelter equation, Nusselt number for plain pipe without any insert and the experimental Nusselt numbers obtained after inserting twisted wire mesh inserts inside the heat exchanger. It is evident from the plot that Nusselt number increases with decrease in twist ratio of the wire mesh insert. The variation in Nusselt number is between 47 % and 71 %. For the twisted wire mesh insert having twist ratio 2.5 for Reynolds number 5000, the increase in heat transfer is found to be 71 % as compared to the plain pipe without insert

4.Variation of Friction factor for twisted wire mesh inserts:

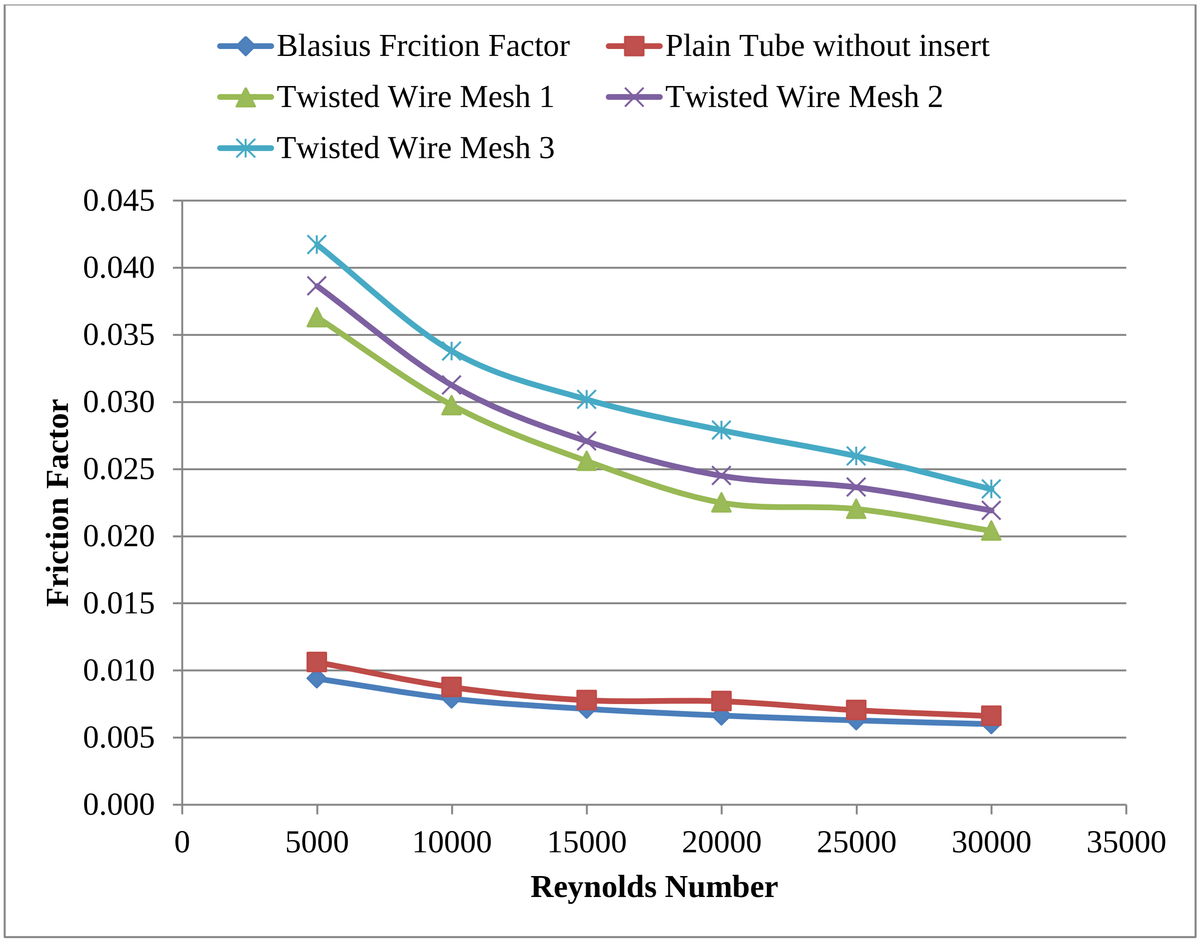

Figure 5.4: Friction factor Plot for values obtained from Blasius correlation, experimental values for plain pipe heat exchanger and heat exchanger modified with twisted wire mesh inserts having twist ratios 5, 3.75 and 2.5 respectively

The plot represents the variation of friction factor for the double pipe heat exchanger based upon values obtained from Blasisus correlation, results obtained experimentally for the plain pipe without twisted wire mesh insert and for the heat exchanger modified with twisted wire mesh inserts. A similar trend as obtained for spring inserts can also be observed in this plot. More is the pitch and twist ratio of the twisted wire mesh insert, lesser will be the increase in friction factor. Although on the basis of heat transfer enhancement, the twist ratio must be kept as low as possible but in case, the pumping power is an important criteria, the twisted wire mesh having higher twist ratio must be used. The increase in friction factor ranges between 209 % to 293 %. For the twisted wire mesh insert having twist ratio 2.5 at Reynolds number 5000, the increase in friction factor is found to be maximum. The minimum increase is found for twisted wire mesh insert having twist ratio 5 at Reynolds number 30000.

5.Variation of Nusselt number for twisted tape inserts:

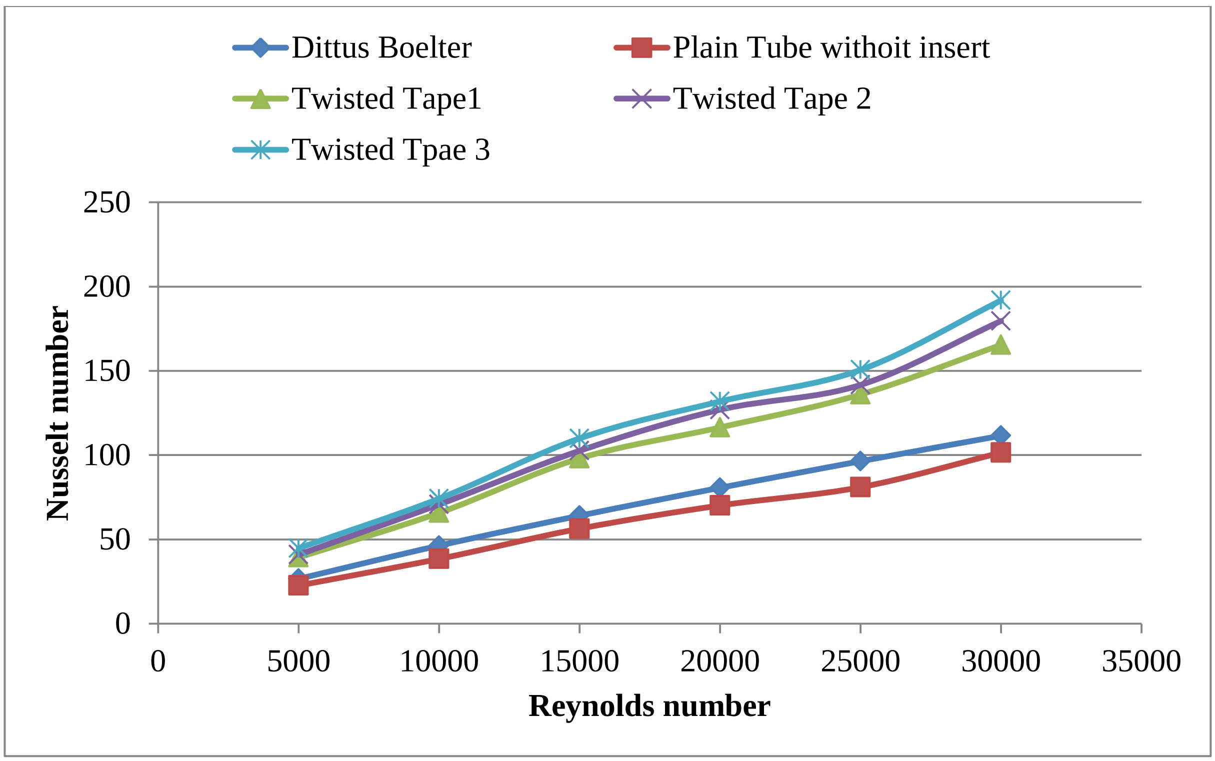

Figure 5.5: Variation of Nusselt number based on Dittus Boelter euation, Nusselt number determined experimentally for plain pipe heat exchanger and heat exchanger modified with twisted tape inserts having twist ratios 5, 3.75 and 2.5 respectively

The plot represents the variation of Nusselt number obtained from Ditttus boelter equation, Nusselt number for plain pipe without any insert and the experimental Nusselt numbers obtained for the test setup when different twisted tape inserts have been introduced in the heat exchanger. Again, it is clear that the Nusselt number increases with decrease in twist ratio of the twisted tape insert i.e. lesser is the twist ratio, more will be the % increase in heat transfer. The variation in Nusselt number is between 63% and 98% for different twisted tape inserts in comparison with the plain pipe. For the twisted tape insert having twist ratio 2.5 for Reynolds number 5000, the increase in heat transfer is found to be 98 % as compared to the plain pipe without insert which is the maximum gain in value of Nusselt Number.

6.Variation of Friction factor for twisted tape inserts:

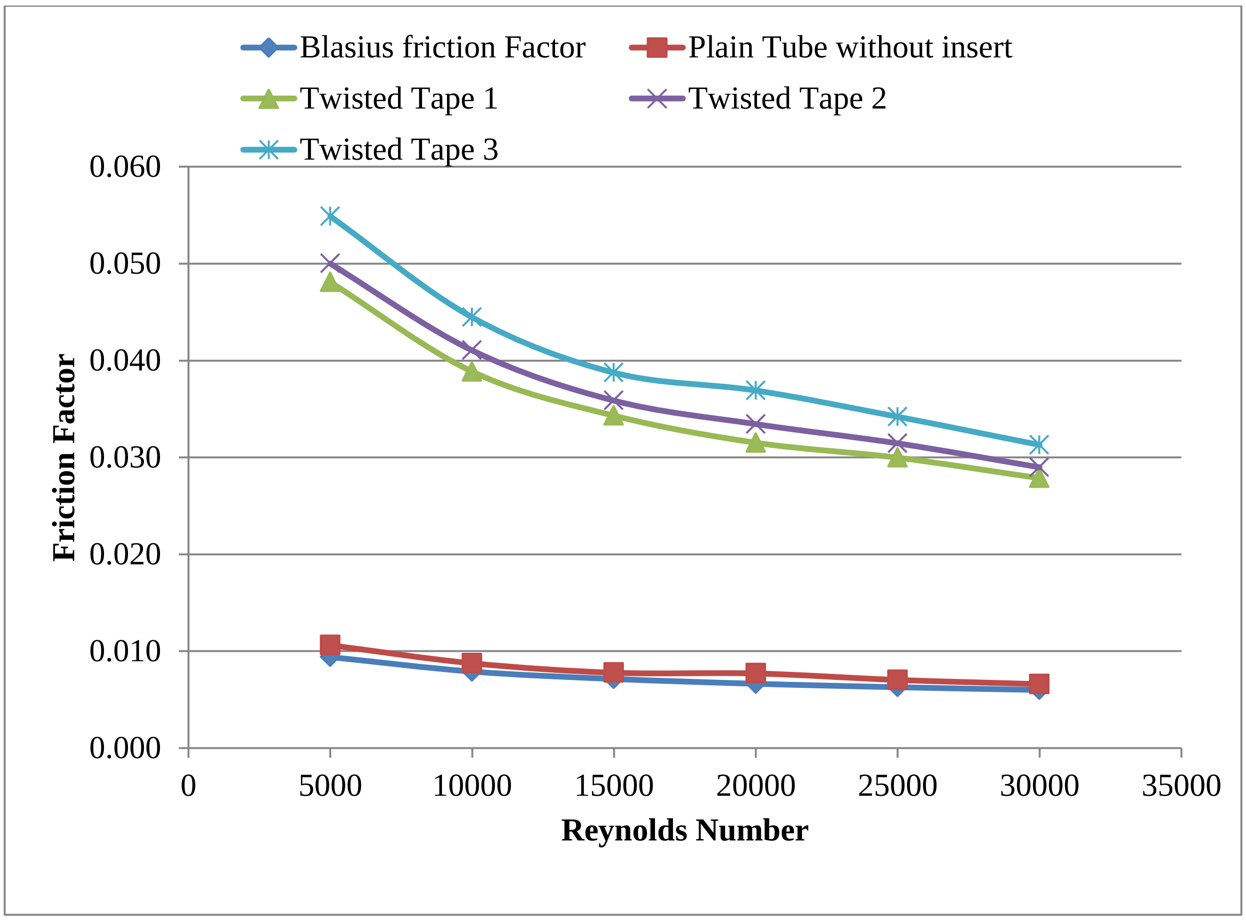

Figure 5.6: Variation of friction factor for Blasius correlation, plain pipe heat exchanger and heat exchanger modified with twisted tape inserts having twist ratio 5, 3.75 and 2.5 respectively

The graph shows comparison of friction factor for the double pipe heat exchanger based upon Blasisus correlation , friction factor values obtained experimentally for the plain pipe without twisted tape insert and for the heat exchanger modified with twisted tape inserts. It can be observed clearly that friction factor increases after introduction of turbulators. It is also evident that increase in friction factor is inversely proportional to the twist ratio of the twisted tape insert used. Lesser is the twist ratio of twisted tape insert , higher is the pressure drop and higher is the value of friction factor. Overall, the increase in friction factor varies from 322 % to 417 %. The maximum increase i.e. 417 % is found for the twisted tape insert having twist ratio 2.5 at the Reynolds number 5000 and minimum increase is found for twisted tape insert having twist ratio 5 at Reynolds Number 30000.

7. Relative comparison of increase in Nusselt Number for Various Turbulators

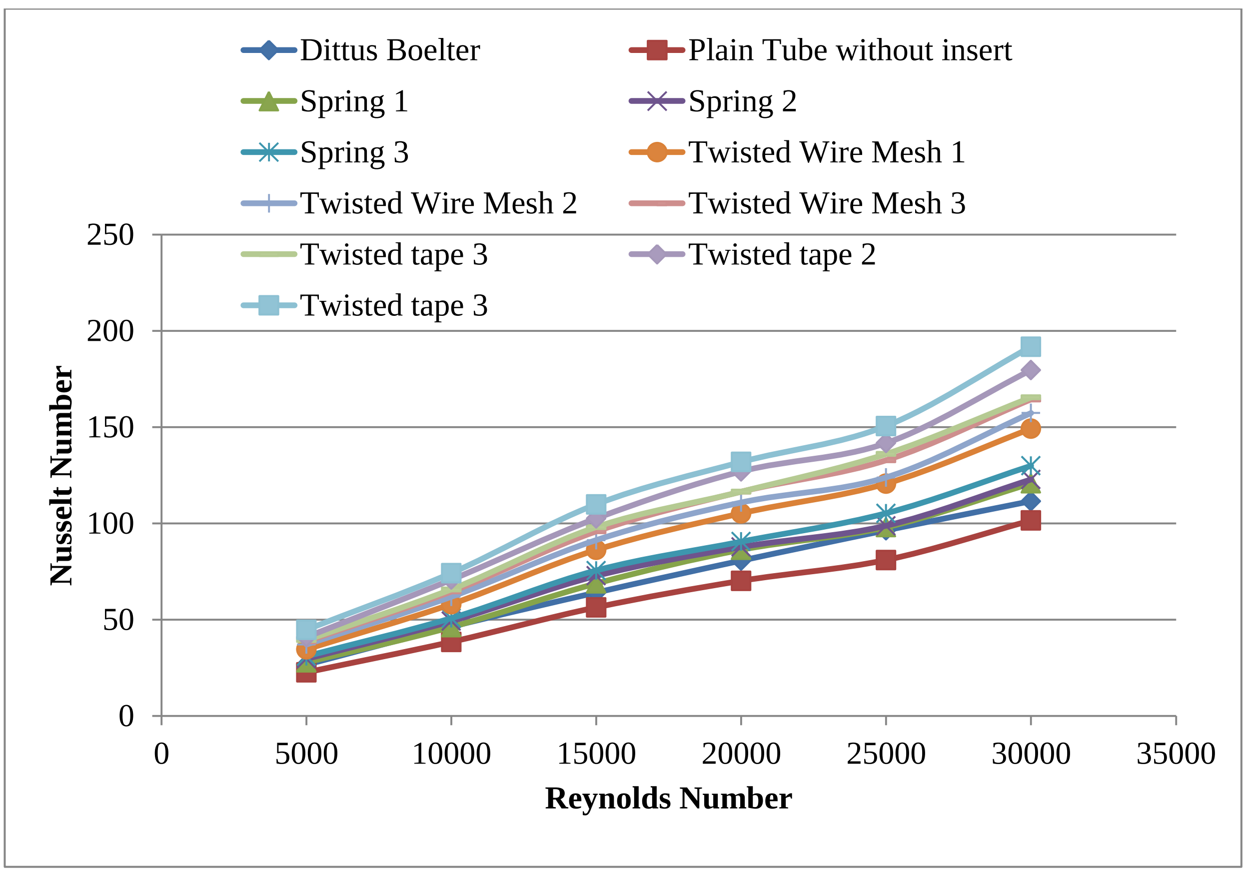

Figure 5.7 :Comparison of increase in Nusselt Number for various turbulators

This graph represents the overall comparison between various turbulators that have been used in this study. All the results have been plotted in a single graph to compare the relative enhancement in Nusselt number. Plots for Dittus Boelter values and for plain pipe without turbulator have also been included. From the plot we can observe that there is always an increase in value of the nusselt number after the introduction of inserts but the % increase varies from insert to insert. For all the inserts, the variation in Nusselt number is between 19 % and 98 %. The minimum increase in Nusselt number is found for the spring inserts and the maximum increase is found for twisted tape inserts. This can be attributed to the high turbulence created by twisted tape inserts in the double pipe heat exchanger. Also, we can observe the trend that % increase in Nusselt number is more for lower values of Reynolds number. This is due to the fact that turbulence has more pronounced effects for lower Reynolds number.

8. Relative comparison of increase in friction factor for Various Turbulators

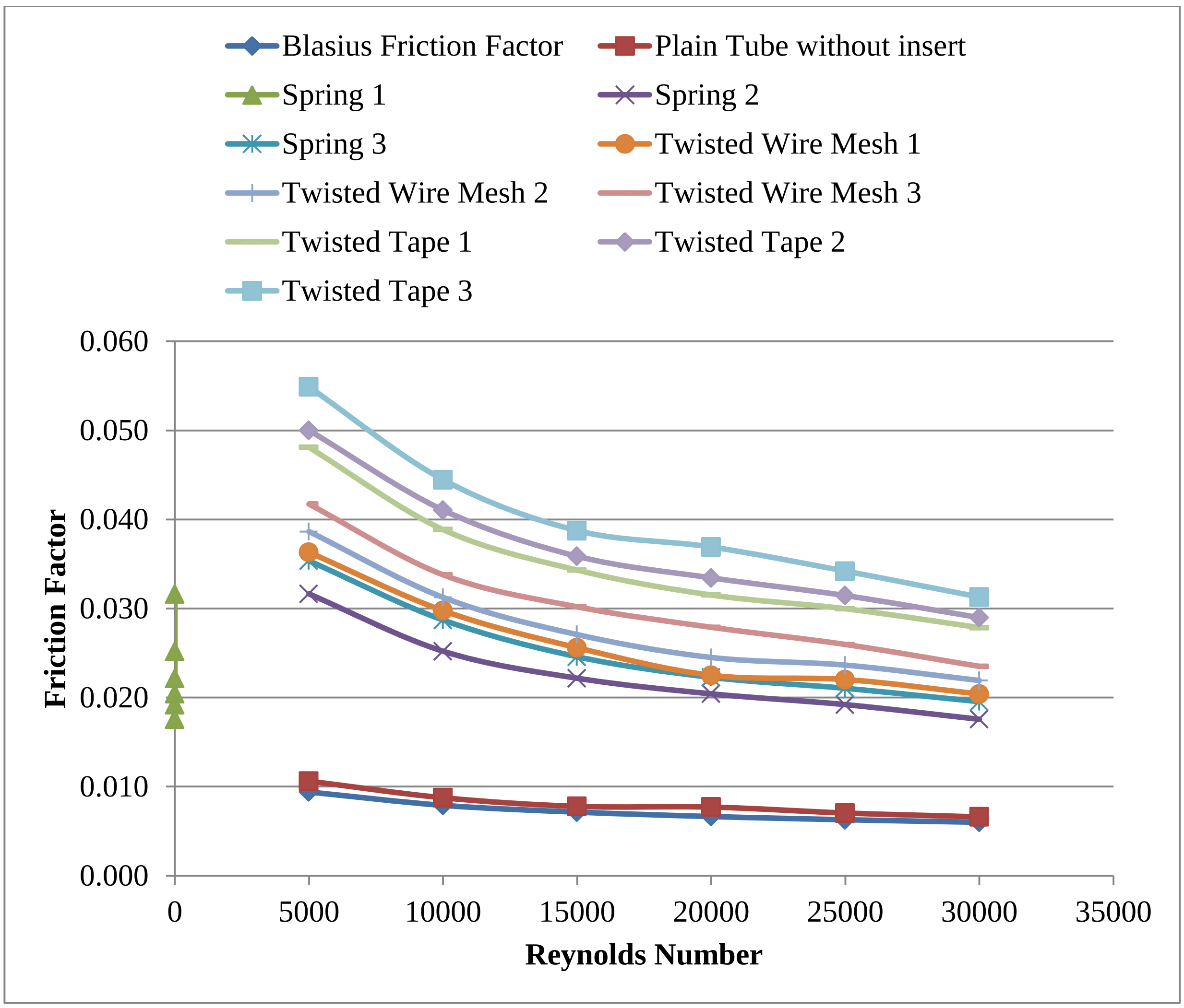

Figure 5.8 :Comparison of increase in friction factor for various turbulators

This plot shows the overall comparison between various turbulators for % increase in friction factor. Since the introduction of turbulators cause a restriction to fluid flow, there is always pressure drop and hence increase in friction factor for all types of turbulators. The increase in friction factor lies between 152 % and 417 % for various turbulators. The minimum increase in friction factor is found for the spring inserts and the maximum increase is found for twisted tape inserts where as the Wire Mesh insert lies some where between the two. This can be attributed to the high turbulence created by twisted tape inserts in the double pipe heat exchanger. If we see from the point of pumping power, then spring inserts are ideal turbulators because they cause minimum amount of increase in pressure drop friction factor. On the other hand, if the pumping power criteria is of lesser importance and the main aim is to maximize the heat transfer, then twisted tape inserts with minimum twist ratio can be said to be ideal inserts. In the essence, we can say that the real life application of these inserts depends on the particular condition. If both heat transfer enhancement and pumping power criteria are considered, then spring inserts or Wire Mesh inserts must be introduced in the heat exchanger. If the sole aim is to increase the heat transfer at any expense, then twisted tape inserts must be used to enhance the heat transfer.

Chapter 6

Conclusions

The experimental investigation was carried out for each of the combinations and the calculations were made for the heat transfer characteristics and pressure drop. The results thus obtained were used to generate the plots and compare the relative performance of various turbulators. The following conclusions can be drawn from the results:

1. For a given insert, be it be spring insert, twisted wire mesh insert or twisted tape insert, the Nusselt number and convective heat transfer coefficient increases with introduction of insert as compared to the plain double pipe heat exchanger without inserts. The maximum gain in heat transfer is found for twisted tape inserts and minimum gain is found for spring inserts. Twisted wire mesh insert lies somewhere between the other two inserts.
2. The pressure drop and friction factor increases in each type of insert although there is large variation in relative comparison. Pressure drop is minimum for spring inserts and maximum for the twisted tape inserts.
3. For spring inserts, the heat transfer increases with decrease in pitch of the spring and friction factor also increases for same Reynolds number. With increasing Reynolds number, heat transfer % enhancement decreases relatively. With the increase in Reynolds number, the value of friction factor decreases for all the arrangements of spring inserts. The variation in Nusselt number is between 19 % and 37 % for various spring inserts. The increase in friction factor for three spring inserts ranges between 152 % to 233 % as compared to the plain pipe heat exchanger without inserts. Thus , to maximize the heat transfer enhancement, the pitch of the spring must be kept as low as possible unless it poses a problem where pumping costs become so high that the % gain in heat transfer becomes non fruitful on the expanse of increased pressure drop .
4. For the twisted wire mesh insert, the heat transfer increases by decreasing the twist ratio of the insert. Pressure drop also increases with decrease in twist ratio of the insert. The % increase in heat transfer for twisted wire mesh insert lies between 47 % and 71 % for

different twist ratios. The % increase in friction factor comes out to be lying between 209 % and 293 %. Thus, the twisted wire mesh insert provides better heat transfer enhancement as compared to the spring inserts. The effect of introduction is more predominant for lower values of Reynolds number that can be attributed to high level of turbulence caused by twisted Wire mesh which results in better mixing of fluid streams and hence higher heat transfer rates.

5. For twisted tape inserts, maximum heat transfer enhancement was found for tape with twist ratio 5 mm and in general, both heat transfer and friction factor increases with decrease in twist ratio for a constant value of Reynolds number. The % increase in heat transfer for twisted tape inserts lies between 63 % and 98 % and the % increase in pressure drop lies between 322 to 417 %. Since, the lower twist ratios results in higher heat transfer enhancement, hence twist ratio of twisted tape insert must be kept as low as possible if so allowed under the circumstances.

In the end, it can be concluded that although, there is always a gain in heat transfer after introduction of inserts, but it does not come without any cost. It is obtained on the expense of increase in pressure drop which in turn alters the pumping cost. Although the experimental results ascertain that twisted tape insets are the best inserts among these for maximum heat transfer enhancement, but still, the selection of a particular insert depends upon the very particular condition for which the heat exchanger is being used.

Chapter 7

Future Scope

The augmentation of heat transfer in heat exchangers is quite a wide field. The work has been done in this field from so many years and still, new techniques are proposed each year that keeps this field afresh. If we talk about the turbulators, different researchers take different turbulators and perform the experimental or numerical analysis by choosing some given number of parameters relevant to their study. Also, there are no standard results available in case of turbulators and some variation is found between researches of various people working in this field. If we talk about the twisted tape inserts, it can itself become a large area of research. The experiments can be conducted for large number of pitches for wide range of Reynolds number. Some research has been carried out for modification in twisted tapes. Then, the width of twisted tape and its thickness can also become one of the parameters that can be varied in the research. Thus, a lot of work can be still carried out in the field of twisted tape inserts in order to determine some standardize results that have all been obtained on the same apparatus and under uniform conditions. Similar is the case with so many other inserts and other heat transfer enhancement methods under uniform conditions. Talking about the present research in this thesis, the following aspects can be covered in future to further investigate heat transfer enhancement methods as:

- The application of turbulators for real life problems can be experimentally verified. For example, in some cases the fluids have properties that lead to settlement of particles and that leads to deposition of layers on heat exchanger surface. That deposition leads to decrease in efficiency of heat transfer. In such cases, use of inserts such as twisted tape insert can come handy as the fluid will always be agitated. It leads to lesser depositions, which does not only mean better heat transfer, but also lesser maintenance requirements for the heat exchanger.

- Experiments can be conducted to compare the performance of turbulators based on their physical dimensions. In the current study diameter of spring wire and width of twisted tape insert has been keep constant. In future, we can vary these parameters and the resulting effects on the performance parameters can be studied further.

- CFD simulation can be used by creating a model of double pipe heat exchanger and various inserts. The results obtained from the software and the one obtained from the actual experimental investigation can be compared to see whether there is match between two. In addition, if the parity is found, the dimensions of turbulators can be designed and analysed in the simulation software itself before conducting the actual experimental testing.

- From the experimental analysis carried out for various set of conditions, empirical relations can be derived and proposed that can help in further research on this topic.

- Similar analysis can be carried out for other turbulators such as louvered strip inserts or using other heat transfer enhancement techniques.

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Appendix 1

Calculations

Sample calculations: In the present study, the effects of various turbulators have been studied experimentally by performing actual experimental study for each set of inserts. The benchmark values for comparing the relative effectiveness of each type of inserts were obtained by first conducting the experimental analysis for the plain pipe heat exchanger without using any insert. In order to test the setup for consistency, it is necessary to compare the valued obtained for the plain pipe heat exchanger with the values available from standard equations. For the heat transfer characteristics, Nusselt number is used and for the plain pipe, Nusselt number can be obtained from Dittus Boelter correlation. Similarly, the theoretical friction factor for the plain pipe can be calculated from the Blasisus correlation. The sample calculations for one of the readings, say Reynolds number =10000 are shown as under:

Internal diameter of inner pipe of heat exchanger , di =12 mm

External diameter of internal pipe of heat exchanger , d_o = 16 mm

Internal diameter of external pipe of heat exchanger, D_i =25.4 mm
length ofheat exchanger , l =1.5 m

The values of various parameters involved in the calculations are

$$\mu_{water} \text{at } 80°C = 3.52 \times 10^{-4} \text{pas}$$

$$\rho_{water} \text{ at } 80°C = 927 \text{ kg/m}^3$$

$$\text{We know that, Mass flow rate, } m = \rho A v$$

$$= \rho A \times \frac{R_e \mu}{D\rho}$$

$$= \mu \times \frac{\pi}{4} \times d \times R_e$$

$$m = 3.3175 \times 10^{-6} R_e$$

$$Q = \frac{m}{\rho} = 3.413 \times 10^{-9} R_e$$

R_e	m(kg/s)	Q(Lpm)
5000	0.0165875	1.0239
10000	0.033175	2.0478
15000	0.0497625	3.0717
20000	0.06635	4.03567
25000	0.0829375	51195

Table A.1: Mass flow rates of hot fluid for different Reynolds number

Sample calculation for Re= 10000

For inner pipe where hot water is used as the working media,

$$Pr = \frac{\mu c_p}{k} = \frac{3.52 \times 10^{-4} \times 4.19 \times 10^3}{0.67} = 2.22$$

$$N_u = 0.023 \, Re^{0.8} Pr^{0.3} = 46.31$$

$$N_u = \frac{h_i D_i}{k}$$

$$h_i = 2585.64 w/m^2 k$$

For Annular section containing cold water as the working media

Q= 4 Lpm (constant)

$$m = \rho Q = \frac{4 \times 10^{-3}}{60} \times 997 = 0.0664 \text{ kg/s}$$

$$m = e \times \frac{\pi}{4} \left(D_i^2 - d_o^2\right) \times \frac{Re \times \mu}{e \times (D_i - d_o)}$$

$$= \frac{\pi}{4} (25.4 + 16) \times 10^{-3} \times Re \times 1.01 \times 10^{-3}$$

$$= 3.28406 \times 10^{-5} Re = 0.0664 kg/s$$

Therefore,

$$\mathrm{Re} = 2021.88$$

Assuming isothermal inside and adiabatic outside

By interpolation,

for ,$\frac{d_i}{d_o} = 0.63$, $\quad \mathrm{Nu} = 5.5112$ (Source: Cengel, Y. A. (2011). *Heat and Mass Transfer.* 638)

$$h_0 = \frac{\mathrm{Nu}\,k}{D_h} = \frac{5.5112 \times 0.61}{9.4 \times 10^{-3}} = 357.6417\ {}^{W}/_{m^2k}$$

$$\text{Thermal Resistance}, R_{th} = \frac{1}{h_i A_l} + \frac{\ln\left(\frac{D_i}{d_o}\right)}{2\pi l k_{cu}} + \frac{1}{h_0 A_0}$$

$$= \frac{1}{2585.64 \times \pi \times 0.012 \times 1.5} + \frac{\ln\left(\frac{16}{12}\right)}{2\pi \times 4.01 \times 1.5} + \frac{1}{357.6417 \times \pi \times 0.016 \times 1.5}$$

$$R_{th} = 0.0515609$$

$$R_{th} = \frac{1}{U_i A_i}$$

$$U_i = \frac{1}{R_{th} A_i} = 343.1447\ {}^{W}/_{m^2k}$$

Wilson Plot method was used then to calculate the values of the Nusselt number on experimental basis.

Now for calculating the value of friction factor, the theoretical value of the friction factor can be obtained by using the Blasius correlation which is defined as:

Blasius correlation for friction factor

$$f = \frac{0.079}{R_e 0.25}, R_e \leq 10^5$$

for $R_e = 10000$, *f=0.0079*

We have got the value of head loss from the experimental data and that can be used for obtaining the actual friction factor as

$$f = \frac{12.1 \times di^5 \times h_{f1}}{LQ^2} = \frac{12.1 \times (12 \times 10^{-3})^5 \times 0.508 \times 10^{-2}}{1.5 \times (3.4130 \times 10^{-5})^2} = 0.008753$$

From the above calculation done for Nusselt number and friction factor based on both empirical relations and also on the basis of actual experimental investigation, we can clearly see that there is close proximity between the results obtained theoretically and practically.