ISSN 2395-1621

Convective heat transfer analysis through circular pipes using internal threads of various geometory

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ABSTRACT

a The present study investigated the effect of internal threaded fin configuration on the heat transfer augmentation and pressure drop in a forced convection. For the same, test sections were made of internal threads of various geometry, such as V-threaded, Sq-threaded and U-threaded pipes of pitch 3mm. Experimentations were carried out for the different input conditions like power, mass flow rate etc. The experimentations were performed at different Reynolds number ranges from 2000 to 11000 and results were compared over the plain tube for corresponding identical conditions. All four test sections were identical in dimensions. (i.e. I.D., O.D. and length) except internal fin configuration. All the test sections were made up of aluminium material. Air was used as working fluid. In case of internal V-threaded test section having pitch of 3mm, the heat transfer coefficient increased by the mean range of 0% to 10% over the plain tube, for Reynolds number in the range of 2000 to 11000.In case of internal Square threaded test section having pitch of 3mm, the heat transfer coefficient increased by the mean range of 48% to 69% over the plain tube, for Reynolds number in the range of 2000 to 11000.In case of internal U-threaded test section having pitch of 3mm, the heat transfer coefficient increased by the mean range of 150 % to 260% over the plain tube, for Reynolds number in the range of 2000 to 11000. It was found that Nusselts number and pressure drop increased with the heat transfer coefficient. It was also found that the U-threaded fin performs best over the remaining for three test sections. So it is clear that more turbulence is created by U-threads over plain tube. The use of internal threads (fins) result lesser pressure drop penalty for same heat duty as compared to other passive techniques like twisted tape, louvered fins etc. The limitation of this work is that, manufacturing difficulty to produce internal threads along long length pipe. This technique can be adopted to enhance heat transfer rate to exiting heat exchangers.

Keywords— **a Internal fin configuration, V-thread, Sq-thread, U-thread, Nusselts number, Reynolds number.**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

In recent time, the focus on energy conservation and abatement of environmental degradation generated a renewed urgency for the use of high performance heat transfer equipments. The application of enhancement of technologies in new and conventional heat exchanger can result in substantial energy, material and cost saving. The subject of enhanced heat transfer developed to the stage that it is of enormous interest for heat exchanger applications. Virtually every heat exchanger is a potential element for heat exchanger. The goal of any heat exchanger designer is to do more with less , whether that is to design a more compact heat exchanger for a given heat duty or to increase the heat duty for a given size of heat exchanger, or to accomplish same other specific objectives subject to other specific constraints. One obvious incentive is economic: - if material usage and operating expenses can be decreased, saving may result. These and similar goals are directly related to increase heat transfer performance or efficiency of the heat transfer surface and to the rational implementation of the improved surface. The performance of heat exchangers, for single-phase flows in particular, can be improved by many active and passive augmentation techniques.

The former uses surface modification or an additional device incorporated into the equipment. The existing boundary layer is disturbed and the heat-transfer performance is improved, usually with an increase in the flow friction and pressure drop. The field of heat transfer, which addresses the improvement of the heat transfer rates, is commonly called enhanced or augmented heat transfer. Enhanced tube forced convective vaporization and condensation have received much less attention than other facts of augmented heat transfer, even though these are the areas in which there is high potential for saving. The augmentation techniques used to increase the heat transfer coefficients are classified either active or passive. Typical examples of passive augmentation are surface roughness, displaced promoters, and vortex generators. Surface roughness--introduced through knurling or threading or formed by repeated ribs--promotes augmentation through the disturbance of the laminar sub-layer that is close to the surface. These methods are used to improve heat-transfer coefficients inside tubes or outside tubes or rods. Displaced promoters include inserts that alter flow mechanics near the surface by disturbing the core flow. Examples are baffles and mixing elements. Vortex flow can be created through coiled wires, stationary propellers, or twisted tapes. Active augmentation, which has also been studied extensively, requires the addition of external power to bring about the desired flow modification. Examples include heat-transfer surface vibration, fluid vibration, and electrostatic field introduction of the several methods discussed above. The most popular and successful technique has been augmentation through surface roughness. This is mainly because of its effectiveness in enhancing the heat transfer and its simplicity in application.

A variety of experimental and analytical works have been carried out on enhancement of heat transfer. Especially the heat transfer enhancement by using fins on outer surface of concentric heat exchanger have concerned by many researcher and practitioners. Relevant literature pertaining to enhancement of heat transfer by introducing protrusions mounted on the outer surface of inner tube, reviewed from different points are presented in this chapter.

Paisarn Naphon [1] has experimentally studied the heat transfer characteristics and pressure drop of the horizontal double tubes with coil-wire insert. The effects of the inlet conditions of the working fluids flowing through the test section and coil pitch on the heat transfer characteristics and pressure drop are discussed. It can be seen that the heat transfer rate and heat transfer coefficient depend directly on the mass flow rates of hot and cold water In addition the heat transfer rates at pitch=3.18 mm are higher than pitch =5.08mm.It can be clearly seen that friction factor continues to decrease with the Reynolds number. As expected, the friction factor obtained from the tube with coil wire inserts is significantly higher than that without coil wire inserts.

Ventsislav Zimparov [2] has experimentally obtained results of heat transfer and isothermal friction pressure drop for two and three start spirally corrugated tubes combined with five twisted tape inserts with different relative pitches in range of Reynolds number 3000 to 60000. The characteristics parameters of the tubes are height to diameter ratios e/Di=0.0407 and 0.0569 and relative pitch H/Di=15.3, 12.2, 7.7, 5.8, 4.7. Significantly, higher friction factor and inside heat transfer coefficients than those of smooth tube under the same operating conditions have been observed.

Alberto Garcia et al. [3] have studied the behaviors of helical wire coil inserts. Helical-wire-coils fitted inside a round tube have been experimentally studied in order to characterize their thermo-hydraulic behavior in laminar, transition and turbulent flow. By using water and water–propylene glycol mixtures at different temperatures, a wide range of flow conditions have been covered: Reynolds numbers from 80 to 90,000 and Prandtl numbers from 2.8 to 150. Six wire coils were tested within a geometrical range of helical pitch $1.17 < p/d < 2.68$ and wire diameter $0.07 < e/d < 0.10$. Results have shown that in turbulent flow wire coils increase pressure drop up to nine times and heat transfer up to four times compared to the empty smooth tube. Within the transition region, if wire coils are fitted inside a smooth tube heat exchanger, heat transfer rate can be increased up to 200% keeping pumping power constant. Wire coil inserts offer their best performance within the transition region where they show a considerable advantage over other enhancement techniques.

Leonard D. Tijing et al. [4] have presented the study investigating the effect of internal aluminum fins with a star-shape cross-section on the heat transfer enhancement and pressure drop in a counter flow heat exchanger. A concentric-tube heat exchanger was used with water as the working fluid. The heat transfer rate increased by 12–51% over a plain tube value, depending on internal fin configurations used. However, the pressures drop also increased substantially by 286–399%. The results showed that a straight-fin configuration is the best to produce a heat transfer increase in a counter flow heat exchanger. Twisted fin configurations did not further increase the heat transfer rate.

Yang Dong et al. [5] presented experimental study to determine turbulent friction and heat transfer characteristics of four spirally corrugated tubes, which have various geometrical parameters, with water and oil as the working fluids. Experiments were performed under conditions of Reynolds number in the rage 6000 to 93000 for water, and 3200 to 19000 for oil respectively. The results showed the thermal performance of these tubes superior compared to a smooth tube, but the heat transfer enhancement were not large as the friction factor increases.

S.K.Saha et al. [6] have investigated that there is drastic reduction in pressure drop, which is excess of reduction in heat transfer. Thus it appears experimentally that on the basis of constant pumping power the larger number of turns may yield improved thermodynamic performance compared to single turn or twisted tape module.

Mohamad M. Saadoun et al [7] shown results of an experimental investigation of the heat transfer for turbulent flow in a circular tube with staggered and in-line arrangements of longitudinal continuous and interrupted fins on the inside surface of the tube is investigated experimentally. The values of the module averaged

Nusselt number for the tube with in-line arrangement fins are higher than that of the staggered arrangements at high Reynolds numbers.

George T. Adewoye et al [8] have revealed some important applications of laminar flow in pipes, and has also shown that most analyses carried out on laminar flow in pipes were majorly on circular pipe, while less consideration has been given to elliptic cylindrical pipe, which is another important pipe configuration in industries. More so, it has clearly shown how fluid flow is adversely affected by a deformed circular pipe into an elliptic cylindrical shape, by considering the behavior of some of the important flow parameters in these two different pipe configurations.

Asit Ku. Acharya et al [9] have computed numerically wall temperature of an internally finned tube for different fin number, height, and shape by solving conservation equations of mass, momentum, and energy using Fluent 12.1 for a steady and laminar flow of fluid inside a tube under mixed flow condition. It has been found that there exists an optimum number for fins to keep the pipe wall temperature at a minimum. The fin height has an optimum value beyond which the wall temperature becomes insensitive to fin height. For a horizontal tube, under mixed flow condition, it is seen that the upper surface has higher average temperature than the lower surface.

V. S. Kulkarni et al [10] have determined experimentally heat Transfer and friction factor data at various volume concentrations for flow in absorber/Receiver and with and without twisted tape inserts is determined experimentally (with water and silver Nanofluid). The experiments are conducted in the Reynolds number range 500≤ Re<≤ 6000 with twisted tape inserts of different twist ratios in the range 0.577 ≤H/D/≤1.732. This study shows that twisted tape inserts when used shows great promise for enhancing heat transfer rate in absorber .The heat transfer coefficient and friction factor of 0≤ Φ≤0.1 % volume concentration of silver Nanofliud are higher compared to flow of water in absorber/receiver.

Kashid M. Saqr et al [11] have done numerical investigation of the effect of continuous and discontinuous internal longitudinal fins on heat transfer augmentation for steady state, axisymmetric turbulent compressible flow has been conducted. The numerical model and solution were validated against established empirical correlations. Repetitive discontinuity along the fin profile showed to have radical effect on the internal convective heat transfer coefficient. It was found that the smaller the discontinuity offset distance, the higher the convective heat transfer coefficient. The main contribution of this study is the derivation of a new correlation to express the heat transfer augmentation in terms of temperature drop per unit length as a function of the fin discontinuity offset distance. Future work should verify the accuracy of this correlation, as well as its geometric limitations.

Dr. Sachin L Borse et al [12] have experimentally investigated the hydrodynamic and heat transfer analysis of three different geometries of the tube in tube helical coil. This study was conducted over a range of Reynolds numbers from 2500 to 6700 using cold water in annulus side. The experiments were carried out in counter flow configuration with hot water in tube side and cold water in annulus side. The experimentally obtained overall heat transfer coefficient (Uo) for different values of flow rate in the inner-coiled tube and in the annulus region were reported. It was observed that the overall heat transfer coefficient increases with increase in the inner-coiled tube flow rate, for a constant flow rate in the annulus region. Similar trends in the variation of overall heat transfer coefficient were observed for different flow rates in the annulus region for a constant flow rate in the inner-coiled tube. It was also observed that when wire coils are compared with a smooth tube, at constant pumping power, an increase in heat transfer rate is obtained at Reynolds numbers below 6700. It was also observed that overall heat transfer coefficient is increases with minimum pitch distance of wire coils.

II. THEORY

The engineering cognizance of the need to increase the thermal performance of heat exchangers, thereby effecting energy, material, and cost savings as well as a consequential mitigation of environmental degradation had led to the development and use of many heat transfer enhancement techniques. These methods have in the past been referred as augmentation or intensification, among other terms. Following are the main types of heat transfer enhancement techniques -

i) Active enhancement methods
   a) Mechanical aids
   b) Surface vibration
   c) Fluid vibration
   d) Electrostatic fields
   e) Suction or Injection
   f) Additives for fluids

ii) Passive enhancement methods
   a) Treated Surfaces
   b) Rough surfaces
   c) Extended surfaces
   d) Displaced enhancement devices
   e) Swirl flow devices
   f) Coiled tubes
   g) Surface-tension devices , and

iii) Compound technique

Active Heat Transfer Enhancement-

The basis of any active heat transfer enhancement technique lies in the utilization of some external power in order to permit the mixing of working fluids, the rotation of heat transfer surfaces, and the vibration of heat transfer surfaces or of the working fluids and the generation of electrostatic fields. While mechanical aids (mixing of fluids and rotation of heat transfer surface) are used in appropriate applications such as surface scraping, baking, and drying processes. Electrostatic techniques have been demonstrated on prototype heat exchangers only. It uses electrically induced secondary motions to destabilize the thermal boundary layer near the heat transfer surface, thereby substantially increasing the heat transfer coefficients at the wall. Generally, active heat transfer enhancement methods have not been well established in industrial applications owing to the capital and operating costs and problems associated with vibration or acoustic noise.

1. Mechanical aids are those that stir the fluid by mechanical means or by rotating the surface. The more prominent examples include rotating tube heat exchangers and scraped-surface heat and mass exchangers.
2. Surface vibration has been applied primarily, at either low or high frequency, in single-phase flows to obtain higher convective heat transfer coefficients.
3. Fluid vibration or fluid pulsation, with vibrations ranging from 1.0 Hz to ultrasound (around 1.0 MHz), used primarily in single-phase flows, is considered to be perhaps the most practical type of vibration enhancement technique.
4. Electrostatic fields, which could be in the form of electric or magnetic fields, or a combination of the two, from DC or AC sources, can be applied in heat exchange systems involving dielectric fluids. Depending on the application, they can promote greater bulk fluid mixing and induce forced convection or electromagnetic pumping to enhance heat transfer.
5. Injection, used only in single-phase flow, pertains to the method of injecting the same or a different fluid into the main bulk fluid either through a porous heat transfer interface or upstream of the heat transfer section.
6. Suction involves either vapor removal through a porous heated surface in nucleate or film boiling, or fluid withdrawal through a porous heated surface in single-phase flow.
7. Jet impingement involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface. Single or multiple jets (in clusters or staged axially along the flow channel) may be used in both single-phase and boiling applications.

Passive Heat Transfer Enhancement-

The major heat transfer enhancement techniques that have found widely spread commercial application are those which possess heat transfer enhancement elements. All passive techniques aim for the same, namely to achieve higher values of the product of heat transfer coefficient and heat transfer surface area. A distinguish between the way how the heat transfer enhancement is achieved, is common in the heat transfer community. Hence also in the present work, a terminology similar to the literature is followed.

1. Treated surfaces are heat transfer surfaces that have a fine-scale alteration to their finish or coating. The alteration could be continuous or discontinuous, where the roughness is much smaller than what affects single-phase heat transfer, and they are used primarily for boiling and condensing duties.
2. Rough surfaces are generally surface modifications that promote turbulence in the flow field, primarily in single-phase flows, and do not increase the heat transfer surface area. Their geometric features range from random sand-grain roughness to discrete three-dimensional surface protuberances.
3. Extended surfaces, more commonly referred to as finned surfaces, provide an effective heat transfer surface area enlargement. Plain fins have been used routinely in many heat exchangers. The newer developments, however, have led to modified finned surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.
4. Displaced enhancement devices are inserts that are used primarily in confined forced convection, and they improve energy transport indirectly at the heat exchange surface by “displacing” the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.
5. Swirl flow devices produce and superimpose swirl or secondary recirculation on the axial flow in a channel. They include helical strip or cored screw-type tube inserts, twisted ducts, and various forms of altered (tangential to axial direction) flow arrangements, and they can be used for single-phase as well as two-phase flows.
6. Coiled tubes are what the name suggests, and they lead to relatively more compact heat exchangers. The tube curvature due to coiling produces secondary flows or vortices, which promote higher heat transfer coefficients in single-phase flows as well as in most regions of boiling.
7. Surface tension devices consist of wicking or grooved surfaces, which direct and improve the flow of liquid to boiling surfaces and from condensing surfaces.
8. Additives for liquids include the addition of solid particles, soluble trace additives, and gas bubbles in single-phase flows, and trace additives, which usually depress the surface tension of the liquid, for boiling systems.
9. Additives for gases include liquid droplets or solid particles, which are introduced in single-phase gas flows in either a dilute phase (gas–solid suspensions) or dense phase (fluidized beds).

Compound Techniques-

There are some difficulties in classifying a few techniques, and the somewhat arbitrarily fuzzy distinctions between them should be recognized. A good example to illustrate this is the classification of some of the newer structured surfaces used in boiling (as treated, rough, or extended surfaces. Perhaps a future re-categorization of the enhancement techniques might be needed to sort out such issues in the new expanding database.

Furthermore, as mentioned earlier, any two or more of these techniques (passive and/or active) may be employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by only one technique itself. This simultaneous utilization is termed compound enhancement. Some promising applications, for example, are in heat or mass exchangers where one technique may preexist. For an application to a two tubular heat exchanger, enhancement may be desired for the inner side or both the sides of tube. The enhancement is applied to the inner and outer surfaces, such as:

i) Helical rib roughner on inner surface and an internal fin on the outer surface.

ii) Internal fins on inner surfaces and porous boiling coating on outer surfaces.

iii) Twisted tape inserts on inner surface and integral fins on outer surface.

iv) Corrugated roughness on inner and outer surfaces.

D. Enhancement Mechanism-

The heat exchange between hot surface of the test section tube and cold air are used to obtain the convective heat transfer. Nusselt number, heat transfer rate, and other related parameters for forced convection are considered.

In present dissertation work, the heat transfer characteristics and pressure drop characteristics on the test tubes of internal threads of various geometric configuration such as V-threaded, Square-threaded & U-threaded test sections with same pitch value 3mm are considered.

Internal threads act as fins which increase inside surface area . The Internal threads also act as disturbance to the fluid flow and hence swirl motion of fluid is created which increases turbulence and hence boundary layer thickness is reduced, due to which convective heat transfer rate increases .Newton's Law of Cooling says that the rate of heat transfer between a solid surface and a surrounding fluid is directly proportional to the temperature difference between the two media, the surface area (A) of the solid and the temperature difference between solid surface and bulk mean temperature of the fluid.

Consider the figure, which is for plain tube.

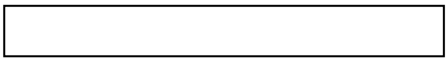


Fig2.1: Plain tube

Then, $Q = h\,A\,(T_{surface} - T_{bulk\ mean})$

So by using internal threads of various geometric fins heat transfer coefficient and pressure drop characteristics can be enhanced in turn augmentation in heat transfer can be achieved.

Now consider the arrangement illustrated in fig is created by machining the tube in different form. As compared with plain tube, the flow characteristics for internal threads of various geometory are completely different because the fins induce more turbulence, better mixing of the fluid, and continuous disruptions in the boundary layer. As a result, the heat transfer coefficient is increased.

Passive enhancement of heat transfer has also been used inside pipes [18] gives an overview of different enhancement mechanisms available in commercial tubes. Figure 3.2 is of great visual help in identifying the roughness type as well as the relevant nomenclature.

In some cases, the heat exchanger operating conditions permit the flow to be tripped from laminar to turbulent flow if subjected to a sufficiently strong perturbation. The surface downstream of flow transition then experiences higher heat transfer coefficients because most resistance to heat transfer occurs across a thin viscous flow layer near the wall instead of across the entire boundary layer. Tripping devices used include surface obstructions (steps, coils, tapes, three-dimensional shapes), surface indentations (cavities, dimples), roughness, as well as upstream turbulence and vorticity.

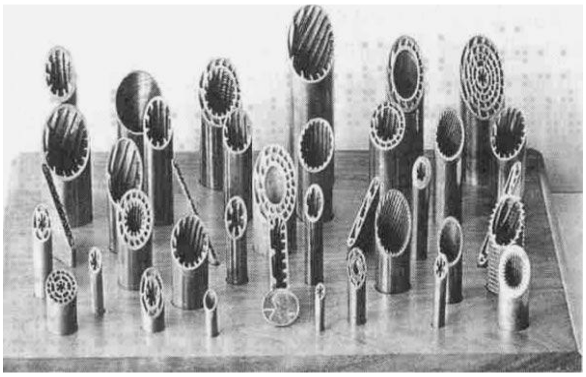


Fig 2.1: Image of various examples of the inner-finned tubes used in heat exchangers

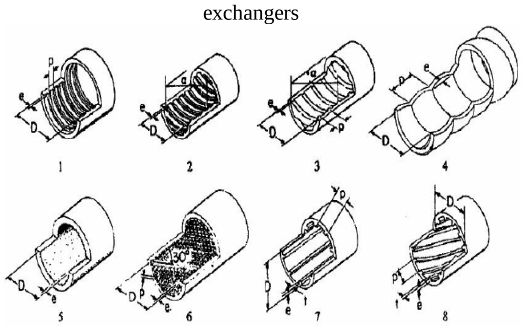


Fig2.3: Sketches of typical roughness configurations

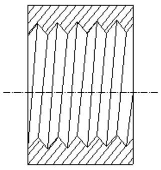
Upstream vorticity does not always cause a flow to become turbulent, but its swirling motions can enhance heat transfer by increasing mixing between the air at the heat exchanger surface and the bulk airflow. Examples include wavy fins, surface winglets, and other elements that protrude from fin surfaces sufficiently to generate vorticity.

E. Use of internal threaded fins for Heat Transfer Enhancement-

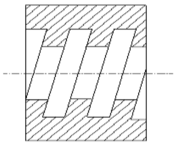
The efficiency of heat transfer equipment is essential in energy conservation. Furthermore, a more efficient heat exchanger can reduce the size of the heat exchanger, thus reducing the costs associated with both material and manufacturing of the heat exchanger. Hence, there have been continuous attempts to improve the efficiency of heat exchangers by various methods. One of the best methods to achieve this is the use of augmented heat transfer surfaces. Improved heat transfer can make heat exchangers smaller and more energy efficient.

An internally-finned tube is one of the most widely used passive heat transfer enhancement methods especially in industrial applications. From the literature study it is found that finned tube has higher Nusselt number compared to the plain tube. In the present study, internal threads of various geometry are manufactured into a circular aluminum test tube sections. The objective of the current experimental study was to determine the effect of internal threads of various geometory on the pressure drop and the heat transfer enhancement in a forced convection. Internal threads act as internal fins which increase inside surface area. The Internal threads also act as disturbance to the fluid flow and hence swirl motion of fluid can be created which increases turbulence and hence boundary layer thickness will be reduced which will result to increase convective heat transfer by reducing convective resistance

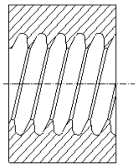
For the internal threaded fin configuration it is found those pressure drop and friction factors are more as compared to the plain tube. Also the internal threads of various geometric fin configuration creates swirling motion of the fluid which increases turbulence of the fluid. The internal threaded fin configuration is shown in the fig(3.3)



V-thread



Sq-Thread



U-Thread

Fig2.4: Internal threads of various threads

III.EXPERIMENTAL SETUP

Experimental tests were performed on the facility shown in fig. below. The air was circulated by the blower and its flow was regulated by a control valve .Air was made to enter in the calming section before entering in the test section for getting fully developed flow .Test sections were made up of internally threaded tube of 36mm inner diameter and 45mm outer diameter and length of 600mm . All test sections were identical in material (i.e. Aluminum) and dimensions such as inner diameter, outer diameter and length.

For conducting experiment first test section was taken. Band heaters were wrapped over it to get constant heat flux. The three band heater each having length 175mm and wattage capacity 200W were taken. These heaters were connected in parallel to get 600W.Heater input was varied by the dimmerstat. The uniform heating of test section had been achieved by passing alternating current from a stabilized single phase variable voltage transformer to the band heaters.

Fig3.1: Band Heaters

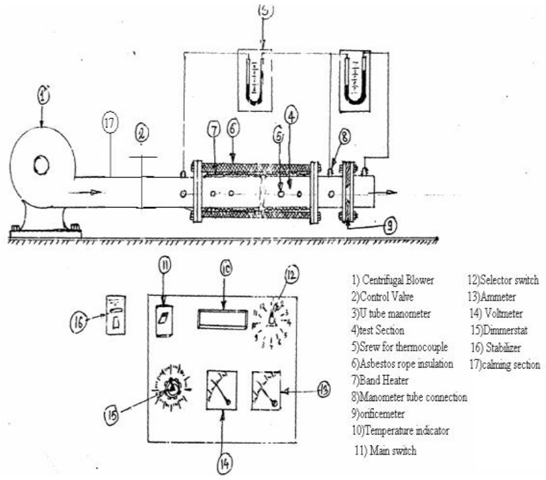


Fig3.2. Experimental Set-up

The test section was insulated properly using asbestos rope material. Ammeter and voltmeter were also connected across the heating element to ensure the proper supply of electrical wattage for preventing the burning of the heating element. The flow rate of air was measured by using orifice meter. The temperatures at the inlet and outlet of the test section were measured by the calibrated Chromel-Alumel thermocouples. The thermocouples were also used to measure the surface temperature of the test section. The pressure drop across the test section was also measured by 'U' tube manometer. All experiments were carried out at steady state conditions. Each test section requires 20 experiments by varying both power heat inputs and fluid flow rates in four steps. Once experimentations on first test section were completed, then the test section was removed by dissembling the flanges .The insulation was unwound, heaters were removed, and thermocouples were disconnected.

Same above process had been repeated to the remaining test sections and experimentation had been carried out. Whole project work containing four test sections required 80 steady state readings. Each reading was finalized after reaching the steady state condition only.The observed readings were used to calculate bulk mean temperatures, average surface temperature and finally the heat transfer coefficients.

The following parameters were included in the present experimental investigation.

1) Heater input was varied up to 600W in 5 steps.
2) Fluid flow rate was varied in four different steps.
3) The experimental study included the four configurations of test sections; first test tube section was a plain tube. Next three test tube sections of internal threads of various geometry such as V-threaded, Square-threaded & U-threaded pipe with 3 mm pitch respectively.

A. Introduction:-

The air was provided by blower and regulated through control valve to ensure constant flow rate. The uniform heating was achieved by passing a.c. current from stabilizer to a band type of heaters .The test section was insulated properly using asbestos rope. Ammeter and voltmeter were also connected across heater.

1. With the help of control valve the mass flow rate of air was adjusted.
2. Band types of heaters were supplied with the

power.

3. Power was adjusted to required level with the help of dimmer stat.
4. One U tube manometer was connected across the test section to measure the pressure drop across test section & another manometer was connected across the orifice plate to measure pressure drop required to calculate the discharge of the air.
5. All the thermocouples were connected to temperature indicator via selector switch. All the readings, voltage, current supplied, temperature at inlet & outlet temperature of surfaces, pressure drop across test section, and pressure drop across orifice plate were recorded in an observation table after steady state condition.
6. For one power input the readings were taken in 4 steps of mass flow rate variation. Likewise Power is varied in 5 steps to have a precise reading. In this way for one Power rating we got readings & for 5 power rating we got 20 steady state readings for one test section.
7. After completing the readings on one test section , test section was removed , insulation was unwound , thermocouples were removed , & were fitted to the next section & same procedure as above was carried out & the readings were taken out.
8. Likewise readings were taken on all seven test sections.
9. In all we got 80 readings. Each reading was finalized after reaching steady state condition only.
10. The observed readings were used to calculate bulk mean temperature, average surface temperature and mass flow rate of air & finally heat transfer coefficients.

IV. RESULT & DISCUSSION

A. Heat Transfer Characteristics:-

The heat transfer coefficient is found out by using Newton's law of cooling, which states that the heat flux from the surface to fluid is proportional to the temperature difference between surface and fluid. The sample calculations in details are given in Appendix A.

B. Experimental Results:-

The experiments were carried out for forced convective heat transfer over all the test sections of different configurations of internal threaded fins and plain tube by varying different parameters such as power input and mass flow rates in four steps each. The range of Reynolds number was taken from 2000 to 11000. The experimental results obtained are presented in graphical forms as shown in figure from 5.4.1.1 to 5.4.4.4 from these graphs different parameters such as Reynolds number, Nusselts number, heat transfer coefficient, pressure drop and friction factor characteristics are studied for all the configuration of the test section for identical conditions of input heat power and mass flow rate. The theoretical values of heat transfer coefficients are calculated by using Dittus-Boelter Correlation. At last by comparing theoretical and experimental results optimum performance of the test section is found out from available configuration of the test sections.

Heat Transfer characteristic curves

1) Reynolds number Vs Nusselt number

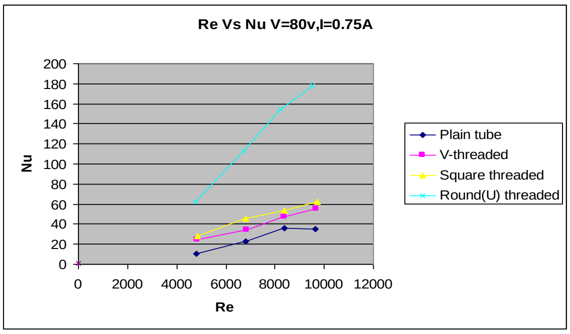


Fig. 5.1

2) Velocity Vs Heat transfer coefficient

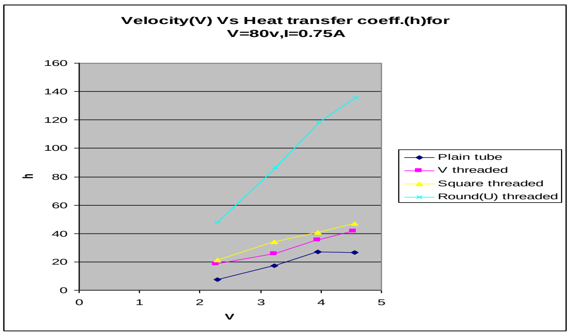


Fig. 5.2

3) Friction factor Vs Reynolds number

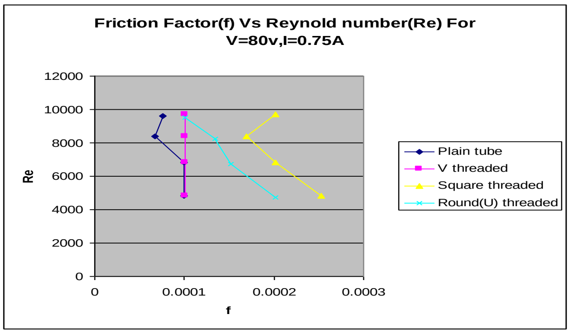


Fig. 5.3

4) Heat supplied Vs Temperature difference:-

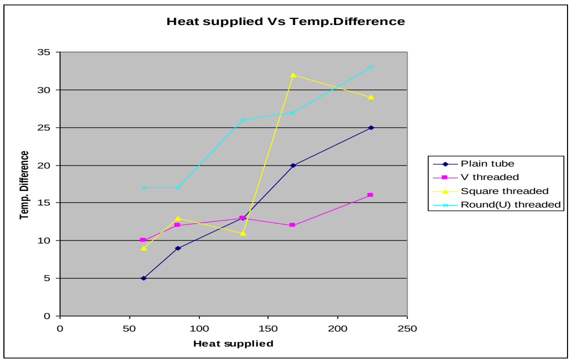


Fig. 5.4

The result obtained and some of them presented as above have been plotted to observe their trends. The results for one heat input are presented in terms of Nusselts numbers verses Reynolds numbers, heat transfer coefficient verses velocity as have been obtained for different fin configurations of the test sections such as V-Threaded, Sq- Threaded and U-Threaded of pitch values of 3mm. It is observed that the experimental observations contain certain uncertainty even with sufficient care.

Looking to the nature of the curves from fig. 5.1 to 5.4, it is found that V threaded test section is having highest average heat transfer coefficient and hence highest average Nusselt number in all the five heater input conditions for 3mm pitch fin configurations for the range of Reynolds numbers from 2000 to 11000.

From the fig. 5.1 to the 5.4, it is found that friction factor goes on decreasing with increase of Reynolds number and hence the mass flow rate. The average friction factor is highest for the double start fin configuration over the triple start , single start fin configurations of both 3mm and 6mm pitch values and over the plain tube.

The friction factor obtained from the tube with internal multi-start threads (fins) is significantly higher than that of plain tube. The results have shown that in turbulent flow multi-start threads (fin profile) increase the friction factor up to times as compared to smooth or plain tube. The use of multi-start threads (fins) on the surface results in heat transfer augmentation in forced convection heat transfer with lesser pressure drop penalty as compared to the inserts like twisted tape, louvered fins etc From the fig 5.1 to 5.4, it is found that temperature difference goes on increasing with Reynolds number. Also with increase in Reynolds number pressure drop increases.

V.CONCLUSION

1. Heat transfer coefficient from the test surface increases with increase in mass flow rate of air for each test section.
2. The use of internal threads of various geometry (fins) on the surface results in heat transfer augmentation in forced convection heat transfer with lesser pressure drop penalty as compared to the inserts like twisted tape, louvered fins etc.
3. In case of internal V-threaded test section having pitch of 3mm, the heat transfer coefficient increased by the mean range of 0% to 10% over the plain tube, for Reynolds number in the range of 2000 to 11000.
4. In case of internal Square threaded test section having pitch of 3mm, the heat transfer coefficient increased by the mean range of 48% to 69% over the plain tube, for Reynolds number in the range of 2000 to 11000.
5. In case of internal U- threaded test section having pitch of 3mm, the heat transfer coefficient increased by the mean range of 150 % to 260% over the plain tube, for Reynolds number in the range of 2000 to 11000.
6. The U threaded test section of 3mm pitch shows best performance over a plain tube.
7. There are two factors surface area in contact with fluid and swirling motion of air, which are responsible for enhancement of heat transfer coefficient.
8. The friction factor goes on decreasing as Reynolds number goes on increasing.
9. The values of heat transfer coefficient and Nusselt number goes on increasing with the power input.
10. In case of internal threaded test sections, U threaded tube shows best performance over V threaded & Sq-threaded test tubes of 3mm pitch and also over a plain tube.
11. The friction factor obtained from the tube with internal threads (fins) is significantly higher than that of plain tube.
12. The results have shown that in turbulent flow threads (fin profile) increase the friction factor up to two times as compared to smooth or plain tube.

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