HEAT TRANSFER ENHANCEMENT USING INSERT: EXPERIMENTAL SET UP WITH SOME PRELIMINARY RESULTS

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ABSTRACT— This research intends to enhance the heat transfer in a turbulent flow of air through tube using inserts. A special geometry insert called rod pin insert is designed for this purpose. A suitable experimental set up is developed for accurately conducting the experiment. The test section of the set up is electrically heated, and air is allowed to flow as the working fluid through the tube by means of a blower. Air velocity, air inlet and outlet temperature, local wall temperature and the pressure drop are measured to determine the friction factor, the Nusselt number, and the heat transfer coefficient. The same experiment is carried out to determine heat transfer through the same tube without any insert. Comparing the results from these two conditions, the heat transfer enhancement due to the use of insert can easily be calculated. From some preliminary results, it is found that heat transfer through tube can be enhanced by using inserts inside the tube up to 9.8 times than tube without insert.

KEY WORDS: Heat Transfer Enhancement, Insert, Experimental Methodology

1. INTRODUCTION

Turbulent flow heat transfer in tubes with inserts has found wide application in heat exchanger used in aerospace, vehicles, refrigeration and air conditioning, cooling of electronic equipments and so on. Improvement in the performance of the above may result in the reduction of the size of heat exchangers or increased heat transfer rate.

The analytical results showed that the local fin heat transfer coefficient varied significantly along fin height, with the smallest value (essentially zero) at the base and the largest value at the tip. Lesser and more gradual variations were exhibits by the local heat transfer coefficient on the wall of the tube or annulus. In general, the fins were found to be as the wall (per unit area).

Heat transfer and pressure drop characteristics in a circular tube fitted with full length strip, short-length strip, and regularly spaced strip elements connected by thin circular rods under uniform heat flux was investigated experimentally by Saha and Langille (2002). The strips were crossed-type, rectangular and square

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Laminar flow of water and other viscous liquids was considered. The short-length strip and regularly spaced strip elements had been found to perform better than the full strips. They found that friction factor reduces by 8-58 percent and Nusselt number by 2-40 percent for short length strips. For regularly-spaced strips elements friction factors increase by 1-35 percent and Nusselt number increased by 15-75 percent.

Hsieh and Huang, (2000) experimentally studied the heat transfer and pressure drop characteristics of water flow on horizontal tubes with/without longitudinal inserts. The Reynolds number ranged from 1700 to 4000. The enhancement of heat transfer as compared to a conventional bare tube at the same Reynolds number based on the hydraulic diameter was found to be a factor of 16 at Re ≤ 4000, while the friction factor rise was only about a factor of 4.5 at Re ≤ 4000.

Uttawar and Raja Rao (1985) carried out experimentally on isothermal pressure drop and convective heat transfer to servo herm medium grade oil in laminar flow in one smooth and seven wire coil –inserted tubes of varying pitch of wire coil. Their investigation has indicated that, When compared with smooth a plain tube at constant basic geometry, an improvement as high as 350% has been obtained in heat capacity and a reduction in heat exchanger area of about 70 to 80 percent may be achieved using wire –coil-inserted tubes.

Lin and Wang (2009) investigate heat transfer enhancement in circular tube with twisted tape insert. They reported that for fully developed laminar convective heat transfer, the thermal conduction in the tape obviously affects the overall heat transfer performance for the UWT condition. It has great influence on the local Nusselt number on the tube surface.

The effect of Prandtl number on heat transfer and friction in smooth and rough tubes was studied by Dippery and Sabersky (1963). The results showed that the heat transfer increased by a factor to 2-3 when rough tubes replaced smooth tubes at the expense of increase in friction. However at high Prandtl number the improvement in heat transfer was achieved without any loses by roughening.

The variation of pressure drop and average heat transfer along a square duct for Reynolds number ranging from 7,000 to 90,000 and for various rib configurations was studied by Han et al. (1985) The results showed that for (α = 90°) the average Nusselt number was twice the average Nusselt number for a smooth channel and the friction factor increase about four to six times.

Lau et al. (1989) studied the turbulent heat transfer and friction in square channel with discrete rib turbulators for Reynolds number ranging from 10,000 to 80,000. The studies revealed that the heat transfer in 90° discrete ribs were 10-15% higher than that of the 90° transverse ribs. The overall thermal performance of oblique ribs at (α = 60°, 45° and 30°) was found to be 20% more than that of 90° discrete ribs.

A longitudinal rectangular plate insert in the tubes of heat exchanger is a good displaceable device and this enhances tube side heat transfer rate. The buoyancy effect on laminar forced convection in a circular tube with longitudinal thin rectangular plate insert was studied by Chen and Hseih (2000).

Compact heat exchangers of the plate-fin, or tube and plate-fin, or tube and center variety use several types of augmented surfaces: offset strip fins, louvered fins, and perforated fins or wavy fins. The flow (usually gases) in these channels is very complex and few generalized correlations or predictive methods are available. Overall heat exchanger information is often available from manufacturers of surfaces for the automotive, air conditioning, and power industries; however, the air-side coefficients cannot be readily flat fins. The improvements are the result of flow separation, secondary flow, or periodic starting of the boundary layer. The latter is illustrated in above figure. It should be emphasized that the tube geometry and arrangement strongly affect

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the heat transfer and pressure drop. For example, heat transfer coefficients are increased with staggered tubes and pressure drop is reduced with flattened tubes (in the flow direction). Design data for augmented compact heat exchanger surfaces are given by Kays and London (1984).

The augmentation is attributable to several effects: increased path length of flow, secondary flow effects, and in the case of the tapes, fin effects. Phenomenological, these devices are part of the general area confined swirl flows, which also includes curved and rotating systems. The recent survey by Razgaitis and Holman (1976) provides a comprehensive discussion of the entire field.

Sarkar et al. (2005) investigated the convective heat transfer in a tube with longitudinal inserts. At comparable Reynolds number, heat transfer co-efficient in tube with longitudinal strip inserts is enhanced by 1.4 to 3 times, friction factor increased by 1.2 to 2.2 times while the pumping power increased up to 4 times compared to that of smooth tube.

The surveys show heat transfer through tube can be significantly enhanced using inserts. However, the thermo hydraulic performance of tubes with rod-pin inserts is yet to be performed to best of our knowledge. This research focus on the heat transfer performance of tube with rod pin insert. This paper presents the experimental facility for studying turbulent flow heat transfer and fluid friction in a tube with rod-pin inserts along with some preliminary results. In this paper tube without insert and smooth tube has the same meaning.

2. EXPERIMENTAL SET UP AND METHODOLOGY

This research includes developing an experimental set up. Figure 1 shows the experimental setup for doing the experiment. The experimental facility consists of Test section, Inlet section, Air supply system and Heating arrangement. The rig is provided with inlet section for getting fully developed airflow in the test section. The experiment has been conducted between the range of Reynolds number of $2.0 \times 10^4$ and $4.05 \times 10^4$.

![Experimental Setup Diagram](image)


Fig 1: Experimental setup

2.1. Test Section

The test was designed, fabricated of Mild Steel (1500 mm long and 68.5 mm inside diameter) with circular tube to avoid contact resistance.

The test section was wrapped with mica sheet, glass fiber taped and insulation tape. Over mica sheet Nichrome wire (of resistance 0.610 ohm/m) was spirally wound uniformly with spacing of 4 mm. Two coils have been used to increase heating. The Nichrome wire was covered with mica sheet, glass fiber tape and insulation tape to make it electrically insulated by covering with asbestos.

The test section was placed in the test rig with the help of the bolted flanges, between which asbestos sheets were installed which act as heat guards in the longitudinal direction.

The test section electric heater was supplied power by 5 KVA variably voltage transformer connected to 220V AC power through a magnetic contactor and temperature controller.

The temperature controller was fitted to sense the air outlet temperature and give signal to
heater for switching it off or on automatically. It protects the experimental set up from being excessive heated which may happen at the time of experiment when the heating system is in operation continuous for hours to bring the system in steady state condition. It also controls the air outlet temperature.

2.2. Inlet Section

The unheated inlet section (shaped inlet) cast from aluminum is of same diameter as of test section. An open end of the pipe would probably act as a sharp edge orifice and airflow would contract and fill the pipe completely within a short distance from the end. This effect was avoided here in the experiment by fitting a shaped inlet; 533 mm long was made integral to avoid any flow disturbances at upstream of the test section to get fully developed flow in the test section as well.

2.3. Air Supply System

A motor operated suction type fan was fitted downstream the test section to supply air that will cool the test section for ascertaining the heat transfer performance. A suction type fan was used here that any disturbance produced by the fan does not effect on the test section flow, as a suction type fan is always fitted at the downstream side of the test section. A 12° diffuser made of 1.5875 mm mild steel plate is fitted to the suction side of the fan. The diffuser was used for minimizing head loss at the suction side. To arrest the vibration of the fan a flexible duct was installed between the inlet section of the fan and the gate valve as shown. The gate valve is of butterfly type and was used to control the flow rate of air. It is fitted at the suction side before the flexible duct.

2.4. Heating System

An electric heater (made of Nichrome wire) was used to heat the test section at constant heat flux. The heater was supplied power by a 5 KVA variable voltage transformer connected to 220 V AC power through a magnetic contactor and temperature controller.

The electrical power to the test section was determined by measuring the current and voltage supplied to the heating element. The voltage was measured with a voltmeter and current by an AC ammeter.

2.5. Relay System

The electric heater was provided with a relay system. It consists of a magnetic contactor, a temperature controller and a thermostatic sensor. The temperature controller was fixed at a certain temperature. The sensor receives signal from outlet air temperature and sends it to the magnetic contactor which is operated by a relay switch. It helps to perform the experiment at a constant temperature.

2.5. Measuring System

Different variables were measured by different types of instruments; some of those were manually operated and some were automatically. The detail description of all the systems used in the study the described in the following sections.

2.5.1. Flow measuring system

![Fig.2. Location of Pitot tube](image)
Flow of air through the test section was measured at the inlet section with the help of a traversing pitot. The traversing pitot was fitted at a distance of 4D from the inlet. The manometric fluid used here is high-speed diesel gravity 0.855. A schematic of the micrometer traversing pitot is shown in Figure 2. Arithmetic mean method is employed to determine the position of the Pitot tube for determination of mean velocity.

2.5.2. Pressure measuring system

The static pressure tapping were made at the inlet of the test section as well as equally spaced 8 axial locations of the test section. Pressure tapping for measurement of static pressure was fitted so carefully that it just touches the inner surface of test section. The outside parts of the tappings were made tapered to ensure an airtight fitting into the plastic tubes, which were connected to the manometer. Epoxy glue (Araldite) was used for proper fixing of the static pressure tapping. U-tube manometers at an inclination of 30° were attached with the pressure tapping. Water was used as the manometric fluid in the experiment.

2.5.3. Temperature measuring system

The temperature at the different axial locations of the test section was measured with the K-type thermocouples.

The temperature measuring locations are –

i. Fluid bulk temperature at the outlet of the test section.
ii. Wall temperature at 8 axial locations of the test section.

The bulk temperature of the air at the outlet of the test section was measured using a thermocouple at the outlet of the test section to determine the location of thermocouple for determination of mean temperature.

For smooth tube heat transfer system, 8 thermocouples were fitted at eight equally spaced axial location of the test section one at each position to measure the wall temperature. Thermal contact between the brass tube and the thermocouple junction was assured by penning thermocouples junction into grooves in the wall. Thermocouples were inserted into the holes and pinned into the grooves of the tube walls.

2.6. Experimental Procedure

The fan was first switched on and allowed to run for few minutes to have the transient characteristics died out. The flow of air through the test section was to set desired value and kept constant with the help of a flow control valve. Then the electric heater was switched on. The electric power was adjusted with the help of a regulating transformer or variac. Steady state condition for temperature at different locations of the test section was by two measurements. First the variation in wall thermocouples was observed until constant value was attained, and then the outlet air temperature was monitored. Steady state condition was attained when the outlet air temperature did not deviate over 10-15 minutes time. At the steady state condition thermocouple and manometer readings were taken manually.

After each experiment the Reynolds number was changed with the help of the flow control valve keeping electrical power input constant. And after waiting for steady state condition, desired data are recorded as per procedure narrated above.

The same procedure was repeated for tube with inserts.

2.7. Inserts

Four inserts with different longitudinal pin spacing (50mm, 100 mm, 150 mm, and 200 mm) were used as inserts for this study. The diameter of the rod was 20 mm and that of pin was 1.6 mm. The pins were 15 mm long and the number of pin in radial direction was 8. The insert is shown in Figure 3.
3. RESULTS AND DISCUSSION

Some preliminary results obtained from the heat transfer measurement of turbulent flow through tube with and without inserts show great potential of the research. Four different rod-pin inserts were used. The inserts differ themselves by the pin spacing.

Figure 4 compares Nusselt number for smooth tube and tube with rod-pin inserts. This curve is plotted for Reynolds Number between $1.4 \times 10^4$ and $4.4 \times 10^4$. For tube with rod-pin inserts (pin distance 50 mm) Nusselt number is about 4 times higher than that of smooth tube at Reynolds number $1.4 \times 10^4$ and is 1.16 times higher at Reynolds number $3.9 \times 10^4$. For inserts pin distance 100 mm this ratio becomes 3.5 times at Reynolds number $1.4 \times 10^4$ and 1.36 times at Reynolds number $3.9 \times 10^4$. Similarly the ratio for 150 mm and 200 mm pin distance are about 4 times at $1.4 \times 10^4$, 1.32 times $3.9 \times 10^4$ and about 3.63 times at Reynolds number $1.4 \times 10^4$, 1.46 times at Reynolds number $3.9 \times 10^4$. The comparison of total heat transfer between tube without insert and with insert is shown in Figure 5. For tube with rod pin inserts, heat transfer is up to 9.8 times higher than that of smooth tube (tube without insert).
4. CONCLUSIONS

An experimental set up has been developed to investigate the heat transfer augmentation in a tube by means of rod-pin inserts. The study has revealed that the rod-pin inserts in the tube enhances heat transfer rate at the cost of increased pumping power. The following conclusion can be made from some preliminary results.

- The Nusselt number is high in the entrance region and it decreases gradually up to a certain point corresponding to the value of a fully developed flow.
- It was observed that tube with rod-pin insert having pin distance 150 mm performs better.

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REFERENCES


