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Investigation on Heat Transfer Enhancement for Fluid Flow through a Tube with Internally Fitted Pin Fins

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Abstract: Heat transfer by convection between a surface and the fluid surrounding can be increased by attaching to the surface called fins. In many engineering applications large quantities of heat needed to be dissipated from small areas. The fins increase the effective area of a surface thereby increasing the heat transfer by convection. Hence the fins have practical importance because it gives maximum heat flow per unit mass with ease of manufacture. This research paper presents an experimental study of effect of internally fitted pin fins in a tube on the mean Nusselt number, friction factor and thermal enhancement factor characteristics. In the experiments, measured data are taken at Reynolds number in range of 7,000 to 14,000 with air as the test fluid. Heat transfer analysis is carried out on circular tube with internally fitted pin fins of constant diameter. Analysis is carried out by varying temperatures on the surface of the tube from 200 ºC to 600ºC and varying fins length from 6 cm to 14 cm. Input parameters such as density, heat transfer coefficient, thermal conductivity and length of fins are taken into consideration and output parameters such as rate of heat flow, pressure drop and effectiveness are determined. The heat transfer and friction factor data obtained is compared with the data obtained from a plain circular tube under similar geometric and flow conditions. It is observed that at all Reynolds number, the Nusselt number and thermal performance increases for a tube with internal fins as compared with a plain tube. These are because of increase in surface area and intensity of vortices ejected from the internally fitted fins.

Keywords: Internal pin fins, turbulent flow, heat transfer and pressure drop.

I. INTRODUCTION

Heat exchangers are used in different processes ranging from conversion, utilization & recovery of thermal energy in various industrial, commercial & domestic applications. Some common examples include steam generation & condensation in power & cogeneration plants; sensible heating & cooling in thermal processing of chemical, pharmaceutical & agricultural products; fluid heating in manufacturing & waste heat recovery etc. Increase in Heat exchanger's performance can lead to more economical design of heat exchanger which can help to make energy, material & cost savings related to a heat exchange process. The need to increase the thermal performance of heat exchangers, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as Heat transfer Augmentation. These techniques are also referred as Heat transfer Enhancement or Intensification. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger. Use of Heat transfer enhancement techniques lead to increase in heat transfer coefficient but at the cost of increase in pressure drop. So, while designing a heat exchanger using any of these techniques, analysis of heat transfer rate & pressure drop has to be done. Apart from this, issues like long-term performance & detailed economic analysis of heat exchanger has to be studied. To achieve high heat transfer rate in an existing or new heat exchanger while taking care of the increased pumping power, several techniques have been proposed in recent years.

Generally, heat transfer augmentation techniques are classified in three broad categories: active methods, passive method, and compound method. A compound method is a hybrid method in which both active and passive methods are used in combination. The compound method involves complex design and hence has limited applications.

M. Sozen and T.M. Kuzay [1] numerically studied the enhanced heat transfer in round tubes filled with rolled copper mesh at Reynolds number range of 5,000-19,000. With water as the energy transport fluid and the tube being subjected to uniform heat flux, they reported up to ten fold increase in heat transfer coefficient with brazed porous inserts relative to plain tube at the expense of highly increased pressure drop. Q. Liao and M.D. Xin [2] carried out experiments to study the heat transfer and friction characteristics for water, ethylene glycol and ISOVG46 turbine oil flowing inside four tubes with three dimensional internal extended surfaces and copper continuous or segmented twisted tape inserts within Prandtl number range from 5.5 to 590 and Reynolds numbers from 80 to 50,000. They found that for laminar

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flow of VG46 turbine oil, the average Stanton number could be enhanced up to 5.8 times with friction factor increase of 6.5 fold compared to plain tube. D. Angirasa [3] performed experiments that proved augmentation of heat transfer by using metallic fibrous materials with two different porosities namely 97% and 93%. The experiments were carried out for different Reynolds numbers (17,000-29,000) and power inputs (3.7 and 9.2 W). The improvement in the average Nusselt number was about 3-6 times in comparison with the case when no porous material was used. Fu et al. [4] experimentally demonstrated that a channel filled with high conductivity porous material subjected to oscillating flow is a new and effective method of cooling electronic devices. The experimental investigations of Hsieh and Liu [5] reported that Nusselt numbers were between four and two times the bare values at low Re and high Re respectively. Bogdan and Abdulmajeed et al. [6] numerically investigated the effect of metallic porous materials, inserted in a pipe, on the rate of heat transfer. The pipe was subjected to a constant and uniform heat flux. The effects of porosity, porous material diameter and thermal conductivity as well as Reynolds number on the heat transfer rate and pressure drop were investigated. The results were compared with the clear flow case where no porous material was used. The results obtained lead to the conclusion that higher heat transfer rates can be achieved using porous inserts at the expense of a reasonable pressure drop. Smith et. al. [7] investigated the heat transfer enhancement and pressure loss by insertion of single twisted tape, full length dual and regularly spaced dual twisted tapes as swirl generators in round tube under axially uniform wall heat flux conditions. Chinaruk Thianpong et.al. [8] Experimentally investigated the friction and compound heat transfer behavior in dimpled tube fitted with twisted tape swirl generator for a fully developed flow for Reynolds number in the range of 12000 to 44000. Whitham [9] studied heat transfer enhancement by means of a twisted tape insert way back at the end of the nineteenth century.

Date and Singham [10] numerically investigated heat transfer enhancement in laminar, viscous liquid flows in a tube with a uniform heat flux boundary condition. They idealized the flow conditions by assuming zero tape thickness, but the twist and fin effects of the twisted tape were included in their analysis. Saha et al. [11] have shown that, for a constant heat flux boundary condition, regularly spaced twisted tape elements do not perform better than full-length twisted tape because the swirl breaks down in-between the spacing of a regularly twisted tape. Rao and Sastri [12], while working with a rotating tube with a twisted tape insert, observed that the enhancement of heat transfer offsets the rise in the friction factor owing to rotation. Sivashanmugam and Sundaram [13] and Agarwal and Rao [14] studied the thermo-hydraulic characteristics of tape-generated swirl flow. Peterson et al. [15] experimented with high-pressure (8–16 MPa) water as the test liquid in turbulent flow with low heat fluxes and low wall–fluid temperature differences typical of a liquid–liquid heat exchanger.

The present experimental study investigates the increase in the heat transfer rate between a circular tube heated with a constant uniform heat flux with air flowing inside it using internally fitted pin fins with constant diameter. As per the available literature, the enhancement of heat transfer using internal fins in turbulent region is limited. So, the present work has been carried out with turbulent flow (Re number range of 7,000-14,000) as most of the flow problems in industrial heat exchangers involve turbulent flow region.

II. EXPERIMENTAL WORK

The apparatus consists of a centrifugal blower unit fitted with a circular tube, which is connected to the test section located in horizontal orientation. Nichrome bend heater encloses the test section to a length of 50 cm. Input to heater is given through dimmer stat. Three thermocouples T_2, T_3 and T_4 at a distance of 11 cm, 22 cm, 33 cm and 44 cm from the origin of the heating zone are embedded on the walls of the tube and two thermocouples are placed in the air stream, one at the entrance (T_1) and the other at the exit (T_5) of the test section to measure the temperature of flowing air. The digital device is used to display the temperature measured by thermocouple at various position. The temperature measured by instrument is in ^{0}C.

The test tube of 110 mm thickness is used for experimentation. A micro-manometer measures the pressure drop across the test section, with double reservoir (range = 0.002–5 mbar) filled with benzyl alcohol and water. The pipe system consists of a valve, which controls the airflow rate through it and an orifice meter to find the volume flow rate of air through the system. The diameter of the orifice is 1.4 cm and coefficient of discharge is 0.64. The two pressure tapings of the orifice meter are connected to a water U-tube manometer to indicate the pressure difference between them. Display unit consists of voltmeter, ammeter and temperature indicator. The circuit is designed for a load voltage of 0-220 V; with a maximum current of 10 A. Difference in the levels of manometer fluid represents the variations in the flow rate of air. The velocity of airflow in the tube is measured with the help of orifice plate and the water manometer fitted on board.

A. Procedure

Air was made to flow though the test pipe by means of blower motor. A heat input of 90 W was given to the nichrome heating wire wound on the test pipe by adjusting the dimmerstat. The test tube was insulated in order to avoid the loss of heat energy to the surrounding. Thermocouples 2 to 4 were fixed on the test surface and thermocouples 1 and 5 were fixed inside the pipe. The readings of the thermocouples were observed every 5 minutes until the steady state condition was achieved. Under steady state condition, the readings of all the five thermocouples were recorded. The experiments were repeated for four different test tubes with internally fitted pin fins of varying lengths of constant diameter with constant airflow rate. The fluid properties were calculated as the average between the inlet and the outlet bulk temperature. Experiments were carried out at constant heat input and constant mass flow rate, for all the three test tubes with different lengths of internal fins.

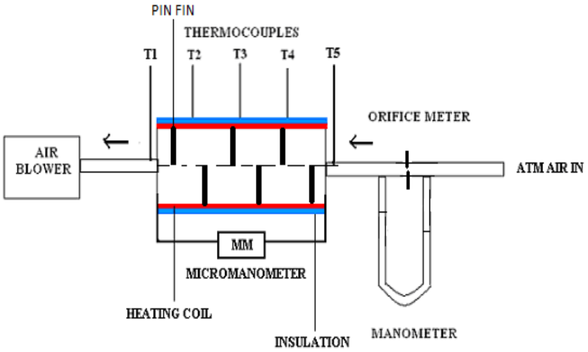

Fig. 1: Schematic diagram of the Experimental Set-up

Fig.2: Photograph of Test tube with internally fitted pin fins

B. Data reduction

The data reduction of the measured results is summarized in the following procedures:

$T_s = (T_2 + T_3 + T_4)/3$

$T_b = (T_1 + T_5)/2$

Discharge of air,

$Q = C_d * a * \sqrt{(2gh_{air})}$

Equivalent height of air column,

$h_{air} = (\rho_w * h_w) / \rho_w$

Velocity of air flow, V = Q/A

$Re = \rho V d / \mu$

d = inner diameter of tube

L= Length of tube

Convective heat transfer area without fins

$A_b = \pi dL - \{\frac{\pi}{4} * (d_1)^2 * n\},$

n = No. of fins

d_1 = Fin diameter

$Q_{base} = hA_b * (T_s - T_b)$

$Q_{fin} = nKA_s . m(t_o - t_i) * \tanh(ml)$

Where, $m = \sqrt{\frac{hp}{kA}}$

l = Fin length

$Q = Q_{fin} + Q_{base}$

Q = Qc + Qr

$Q = m * C_p * (T_1 - T_5)$

m= mass flow of air

$Qr = \sigma A \varepsilon (T_s^{\,4} - T_b^{\,4})$

$h = (Q-Qr)/(A (T_s - T_b))$

Nu = h D/ K

Equation gives experimental Nusselt number.

Nusselt numbers calculated from the experimental data for plain tube were compared with the correlation recommended by Dittus-Boelter.

$Nu = 0.023\ Re^{0.8}\ Pr^{0.4}$

Equation gives gives theoretical Nusselt number.

In straight pipe lengths, Pressure drop (ΔP) can be calculated using the Darcy Equation

$$\Delta P_{Friction} = \frac{f.\ L_{Pipe}}{d_{Pipe}} \frac{\rho . u^2}{2}$$

f = Darcy friction factor

Friction factor correlation

Correlation of Petukhov

$f = (0.790 \ln Re - 1.64)^{-2}$ for $3000 \leq Re \leq 5 \times 10^6$

The enhancement efficiency (ψ) is defined as the ratio of the heat transfer coefficient for the tube with internal fins (h_f) to that for the plain tube without internal fins (h_p) at a constant Reynolds number (CR) as follows (Yakut et al,2004):

$$\psi = h_f / hp$$

III. RESULTS AND DISCUSSION

Experimentally determined Nusselt number values for plain horizontal circular tube (without internal fins) are compared with Dittus-Boelter correlation.

Fig. 3 shows the comparison between Nusselt numbers obtained experimentally and by using Dittus-Boelter equation for plain tube. It is observed that the value of Nu (experimental) is less than Nu (Dittus-Boelter). Actual heat carried away by air passing through the test section is the combination of convective and radiative heat transfers. As the heat transferred by convection alone is considered while performing experimental and numerical calculations, it can be expected that Nu (experimental) is less than Nu (Dittus-Boelter).

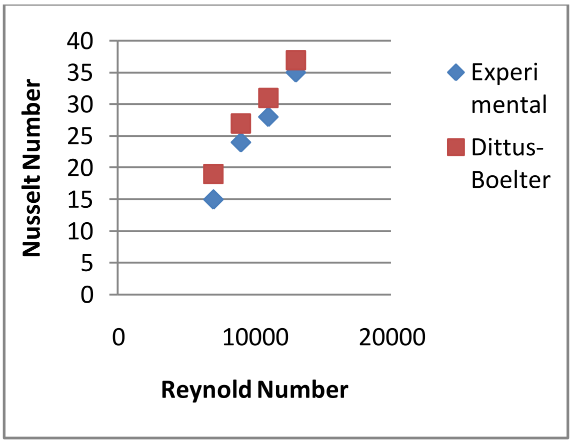

Fig.3: Comparison between the measured results and the calculated values of air in plain tube

Fig. 4 shows the variation of friction factor with Reynolds number for plain tube. The data obtained by the experiment is compared with Petukhov equation with the deviation of ± 7% for the friction factor.

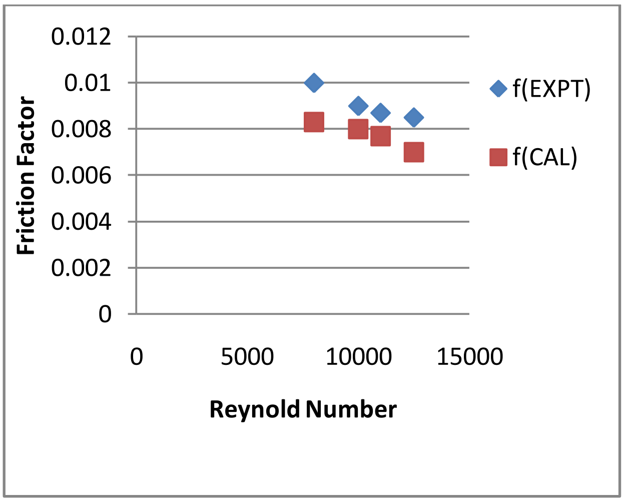

Fig.4: Comparison between the measured results and the calculated values of air in plain tube

Fig.5 shows the variation of Nusselt number with Reynolds number in the plain tube and tubes with internally fitted pin fins of varying lengths of constant diameter. It is observed that for all cases, Nusselt number increases with increasing Reynolds number. It is observed that for tube with internal fins the heat transfer rates are higher than those from the plain tube and also for a test tube with fin length 1/2 of the diameter of tube the heat transfer rate is higher than those for a test tube with fin length 1/3 of the diameter of tube. Heat transfer enhancement in a tube flow with internal fins is mainly due to the increase in heat transfer area, separation of the flow and secondary flow. An internally fitted fins also increases the flow velocity and leads to a significant secondary flow.

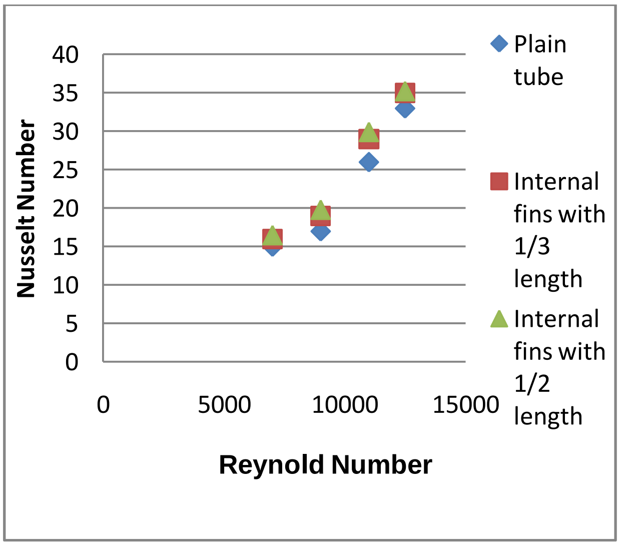

Fig. 5: Comparison Nusselt number of air as a function of Reynolds number in plain tube and tube with internal fins

Secondary flow further provides a better thermal contact between the surface and the fluid because secondary flow creates swirl and the resulting mixing of fluid improves the temperature gradient, which ultimately leads to a high heat transfer coefficient. It was observed that the mean Nusselt numbers for test tubes with internal fins of length 1/2 and 1/3 of the diameter of tube are respectively, 3.96 and 3.27 times better than that for the plain tube.

Fig.6 shows the variation of friction factor vs Reynolds number for the plain tube and tubes with internally fitted pin fins of varying lengths of constant diameter. The friction factor for the test tube using internal fins is more than that for plain test tube. Also friction factor decreases with increase in Reynolds number for all the three different test tubes. Friction factor is a measure of the pressure losses in a system to the kinetic energy of the fluid. In the present work, the pressure losses include losses due to friction and due to drag force exerted by obstacles. It is noticed that the increase in Reynolds number leads to decrease in the friction factor, because the friction factor is proportional with pressure drop and inversely proportional to the square root of flow speed. These figures also indicate that the larger the length of internal fins, higher is the pressure drop because increase in the length of internal fins means increase in the size of obstacles and hence the pressure drop also increases.

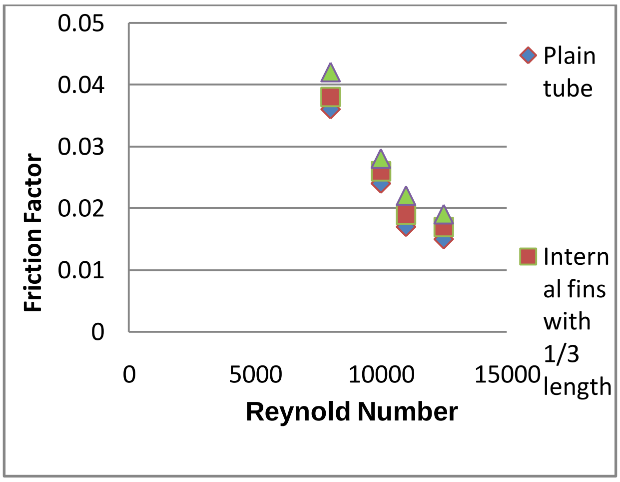

Fig. 6: Comparison Friction factor of air as a function of Reynolds number in plain tube and tube with internal fins

IV. CONCLUSIONS

Experimental investigations of heat transfer, friction factor and pressure drop of a plain circular tube and circular tubes with internally fitted pin fins of varying lengths of constant diameter are described in the present report. The conclusions can be drawn as follows:

1. With increases in Reynolds number Nusselt number and friction factor also increases.
2. The heat transfer enhancement can be achieved with internal fins and decrease in length of internal fins can results in decreases in heat transfer.

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3. The friction factor increases with increase in the length of internal fins due to increase in pressure drop.
4. The performance of circular tube can be improved by the use of internal fins.

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