4th International Conference on Mechanical Engineering, December 26-28, 2001, Dhaka, Bangladesh/pp. III 127-132

STUDY OF FORCED CONVECTION THROUGH INTERNAL T-SECTION FINNED TUBE

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Abstract Heat transfer performance of a T-section internal fin in a circular tube has been experimentally investigated. Electricity supplied to the specially wrapped Nicrome wire around the tube is the heat source, while fully developed steady state turbulent airflow is the heat sink in this study. Wall temperature, bulk fluid temperature, and pressure drop along the axis of the finned tube were measured for different Reynolds number ranging from $2.0x10^4$ to $5.0x10^4$. From the measured data, heat transfer coefficient and friction factor were calculated and analyzed. The finned tube produces significant enhancement of both heat transfer and friction factor. Enhancement in heat transfer coefficient and friction factor may in part be attributed due to the higher heat transfer area and frictional area of the finned tube, respectively.

Keywords: Heat transfer enhancement, forced convection, finned tube

INTRODUCTION

Enhancement techniques that improve the overall heat transfer coefficient for both laminar and turbulent flow in tubes are important to heat exchanger designers. It is evident that for both laminar and turbulent flow regimes, the finned tubes have exhibited substantially higher heat transfer coefficients when compared with corresponding smooth tubes. Extensive studies have been performed from the beginning of the twentieth century to determine the heat transfer characteristics inside tubes. Sieder and Tate (1936) derived an empirical correlation for turbulent flow of forced convection through both of circular and noncircular tube, as

$$\overline{Nu}_D = 0.027 \mathrm{Re}_D^{0.8} \mathrm{Pr}^{0.3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \qquad (1)$$

where Nu_D is average Nusselt number based on hydraulic diameter, Re_D is Reynolds number based on hydraulic diameter, Pr is Prandtl number, μ is viscosity of flowing fluid and the subscripts b and s indicates at bulk temperature and surface temperature respectively. This correlation is valid for both uniform wall temperature and uniform heat flux for both liquid and gases in the range of $0.7 < Pr < 16{,}000$ and $Re_D > 6000$. The correlation shows that higher heat transfer occurs in finned tube than that of smooth one. Cox et al. (1970) used several kinds of enhanced tube to improve the performance of a horizontal-tube multiple-effect plant for saline water conversion. Prince (1971) obtained 200% increase in heat transfer co-efficient with internal circumferential ribs. Kern and Kraus (1972) used enhanced fins (interrupted and perforated) and demonstrated that finned tubes have substantially higher heat transfer coefficients than continuous finned tubes. Enhancement of heat transfer was also reported by Hu and Chang (1973), Nandakumar and Masliyah (1975), Watkinson et al. (1975) and Glodsterin and Sparrow (1976) in their respective studies.

The first analytical study to predict the performance of tubes with straight internal fin in turbulent airflow was conducted by Patankar et al. (1979). A numerical study of developing fluid flow and heat transfer in a circular tube with internal longitudinal continuous fins have been reported by Chowdhury and Patankar (1985) and Prakash and Liu (1985). Kelkar and Patankar (1987) also analyzed internally finned tubes numerically, whose fins are segmented along their length. The fin segment was done in the flow direction, separated by an equal distance of its length before the next fin. The analysis presents that the inline-segmented fin gives only 6% higher Nusselt number than those of continuous fins. Rustum and Soliman (1988) carried out analyses of thermally developed laminar flow in internally finned tubes for incompressible Newtonian fluid.

Mafiz et al. (1996, 1998) studied experimentally steady state turbulent flow heat transfer performance of circular tubes having six integral internal longitudinal fins and they found an abrupt pressure fluctuation near the entrance region of the tube. Their study indicates

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that significant enhancement of heat transfer is possible by using internal fins without scarifying additional pumping power.

Uddin (1998) studied pressure drop characteristics and heat transfer performance of air through an internal rectangular finned tube and found that the heat transfer co-efficient based on inside diameter and nominal area was about 2 times the smooth tube values. Mamun (1999) studied pressure drop and heat transfer performance of air through an internally in line segmented and non-segmented finned (of rectangular cross section) tube at constant pumping power. The results of the study show that friction factor of in-line finned tube is about 4 times higher than that of smooth tube. Friction factor of in-line segmented finned tube is about 3 times higher than that of smooth tube. Heat transfer for the in-line-finned tube is two times higher than that of smooth tube for comparable Reynolds number. Heat transfer for the in-line segmented finned tube is 2 times higher than that of smooth tube for comparable Reynolds number. The results thus show that both inline finned tube and in-line segmented finned tube results in heat transfer enhancement but in-line segmented finned tube results in the same heat transfer enhancement with less pressure drop and with less pumping power.

In line with the above mentioned studies in heat transfer in finned tube, present geometry was chosen in search of better heat transfer and lower friction factor. A detail of the experimentation is explained in the following section.

EXPERIMENT

Experimental facility shown in Fig. 1(a) could be better explained by dividing it into several systems namely—test section, air supply system, heating system and measurement system.

Test Section

The test section is a brass-tube of 1500mm length. It has two types of geometry. One is smooth of 70mm inside diameter and the other is finned having six internal longitudinal T-section fins as shown in Fig. 1(b). Similar casting methods are employed for both the tubes for getting comparable properties. Different holes are drilled along the axis of the test section for fitting thermocouples and pressure probes.

At first the test section was wrapped with mica sheet and glass fiber tape, and then it was spirally wound by Nichrome wire (of resistance 0.249 ohm/m for smooth tube and 0.739 ohm/m for finned tube) uniformly with spacing of 16mm around the tube. Mica sheet, glass fiber tape, heat insulating tape and asbestos powder were put again sequentially over the Nichrome wire.

An unheated inlet section (sometimes called shaped inlet) casted from aluminium has the same diameter as of the test section. The 533mm-long shaped inlet was made integral to avoid any flow disturbances at upstream of the test section and to get fully developed flow in the test section.

Air Supply System

Air supply system consists of a motor operated suction-type fan fitted downstream the test section. A suction type fan was used here so that any disturbance produced by the fan does not affect the flow through the test section. A 12^o diffuser made of mild steel plate is fitted to the suction side of the fan 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 a butterfly valve. The butterfly valve was used to control the flow rate of air. Airflow was varied by this system in the Reynolds number range of 2.0×10^4 to 5.0×10^5.

Heating System

The Nichrome wire wound around the brass tube provided constant heat flux when it is connected to a power supply. A 5-KVA power supply is used in this experiment with the help of a magnetic connector and temperature controller. The temperature controller was fitted to sense the outlet air temperature to provide signal for switching the heater off or on automatically. Heat input by Nichrome wire was kept constant here and is determined by measuring the current and voltage supplied to the heating element.

Measurement System

Other than power measurement, the inside wall temperature and pressure along the axis of the test section and the airflow were measured. Flow of air through the system was measured at the inlet section with the help of a traversing pitot tube. The manometric fluid used here is high-speed diesel of specific gravity 0.855. Arithmetic mean method is employed to determine the position of the pitot tube for determination of mean velocity.

Pressure tappings for measurement of static pressure were 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 was used for proper fixing of the static pressure tappings. U-tube manometers at an inclination of 30^o were attached with the pressure tapping. Water was used as the manometric fluid in this experiment.

The temperatures at the different axial locations of the test section were measured by K-type thermocouples connected with a data acquisition system. The data acquisition system stored data at five minutes interval. The temperature measuring locations are (i) bulk fluid temperature at the outlet of the test section, (ii) wall temperature at 8 axial locations of the test section, (iii) fin-tip temperature at 8 axial locations of the test section. Measurement (iii) was made when finned tube was set in the test rig. 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 locations of thermocouple for determination of mean temperature arithmetic mean method was applied.

For smooth tube**,** 8 thermocouples were fitted at eight equally spaced axial locations of the test section to measure the wall temperature. Thermal contact between the brass tube and the thermocouple junction was assured by peening thermocouples junction into grooves in the wall. For finned tube system, sixteen thermocouples were fitted at eight locations. The number of thermocouples was two in each location to measure the wall as well as the fin-tip temperature of the test section.

Procedure

The fan was first switched on and allowed to run for about five minutes to have the transient characteristics died out. The flow of air through the test section was set to desired value and kept constant with the help of a flow control valve. Then the electric heater was switched on. The electrical power was adjusted (if necessary) with the help of a regulating transformer or variac. First the variation in wall thermocouples was observed until constant values were attained, then the outlet air temperature was monitored. Steady state condition was attained when the outlet air temperature did not change over 10-15 minutes of time. At the steady state condition thermocouple readings are automatically recorded by the data acquisition system. At the same time, manometer readings were taken manually.

After each experimental run 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.

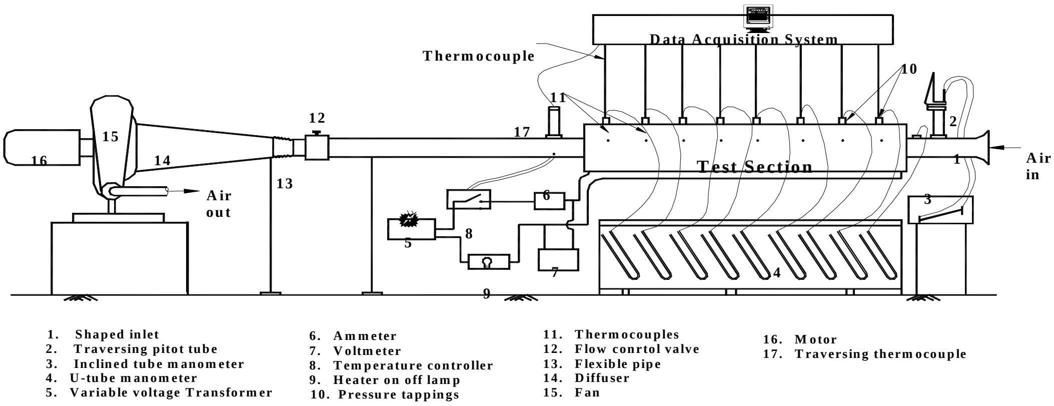

Figure 1(a) : Schamatic of the Experimental Rig

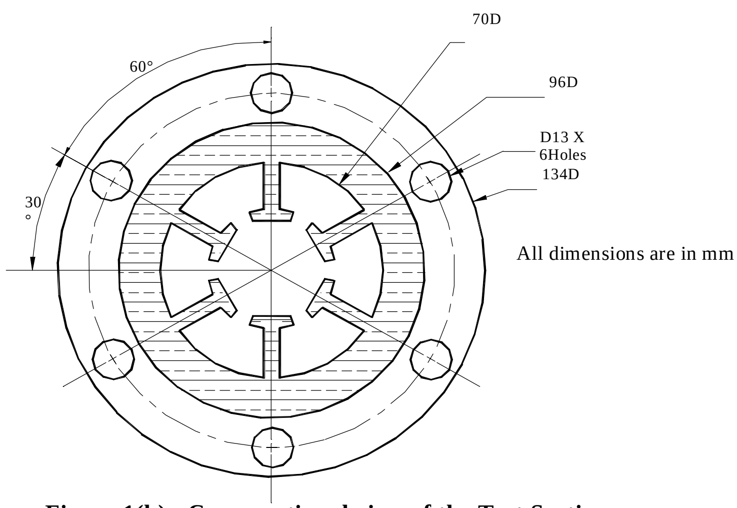

Figure 1(b): Cross sectional view of the Test Section

DISCUSSION

On the basis of experimental data various parameters are presented and analyzed here. Figure 2 represents the variation of wall and fin tip temperature along the length of the finned tube for five different Reynolds numbers at constant heat flux condition. Here wall temperature increases along the axis of the test section. At higher Reynolds number the wall temperature is low because more heat is taken away by air. It is worth mentioning that at the inlet section the temperature gradient is high for both the tubes because the cold entering air takes away much heat. A comparison between tip and wall temperature of finned tube has also been represented in Fig. 2 which demonstrates that wall temperature is always higher than that of fin tip for a particular Reynolds Number.

From Fig. 3, for constant Reynolds number, it is clear that bulk fluid temperature increases linearly as air passes through the test section of smooth tube. Similar behaviour also observed for the case of finned tube. At higher Reynolds number the bulk fluid temperature is lower. In addition, the slope of these curves gradually decreases with the increase of Reynolds number. This is due to the fact that at higher Reynolds number air does not get enough time for being heated while it gets enough at lower Reynolds number. The variation of local heat transfer co-efficient along the axis of the finned tube has been represented in Fig. 4. The coefficient is large at entry of the test section which is also observed for smooth tube. But the developed region starts earlier than observed in smooth tube. Here the developing region ended at X/L = 0.3. It is occurred due to better heat transfer in finned tube.

In Fig. 5 the variation of average Nusselt number with Reynolds Number for both finned and smooth tubes has been shown. Average Nusselt number increases with the Reynolds number. From this figure it is observed that average Nusselt number for finned tube is about 2 times higher than that of smooth tube.

Figure 6 shows the variation of average Nusselt Number as a function of Reynolds number for smooth tube. A comparison has been made here for experimental data with that of Sieder and Tate (1936) and Mamum (1999). The solid upper triangle shows the data collected in this work. The maximum variation of the data with Sieder and Tate (1936) is as high as 22% and as close as 0.64%.

From the curve it is observed that the experimental value of average Nusselt is about two times higher than that of Mamun (1999). The cause of variation of data with Sieder and Tate (1936) may be due to fact that the surface condition of the test section of Sieder and Tate (1936) was perfectly smooth tube, but in this work the tube was smoothed with zero grade emery paper. So this tube contains slightly granular surface. Due to this roughness of surface small higher heat transfer occurred.

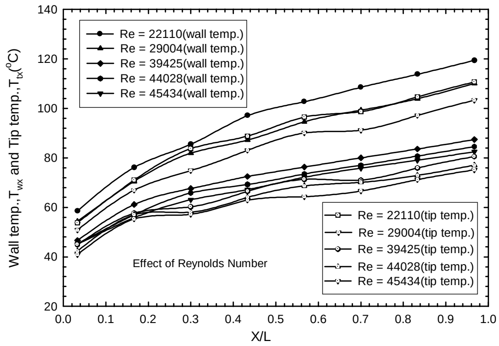

Figure 2: Comparison of Wall and Fin-tip temperature along the axis of the finned tube

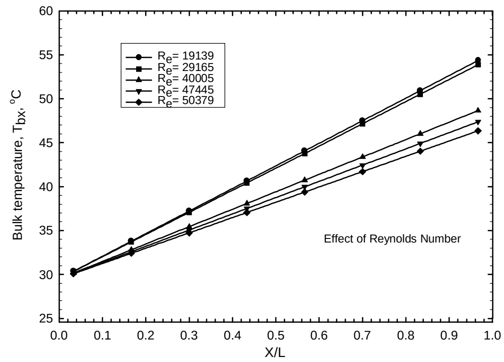

Figure 3: Bulk temperature distribution along the axis of the smooth tube

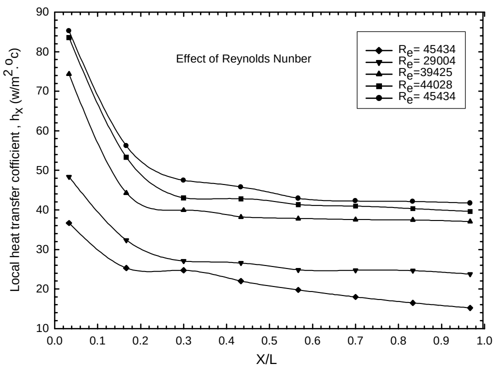

Figure 4: Local heat transfer coefficient along the axis of the finned tube

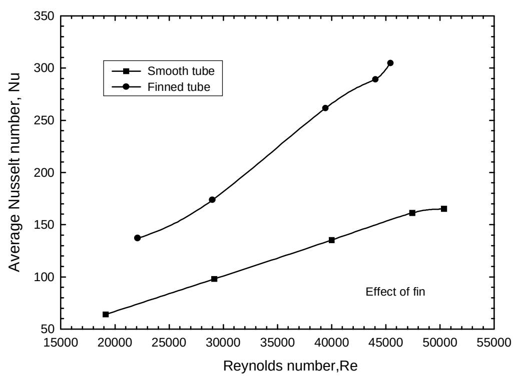

Figure 5: Effect of fin on Nusselt Number

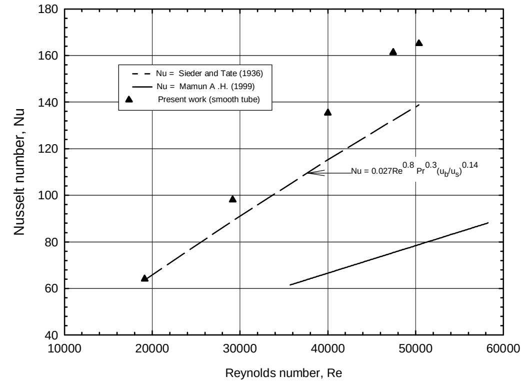

Figure 6: Comparison of Nusselt Number for smooth tube

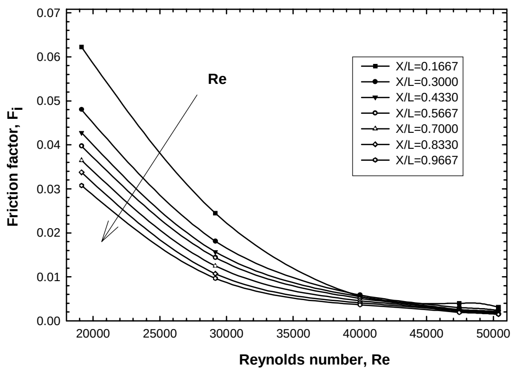

Figure 7: Effect of Reynolds Number on friction factor for smooth tube

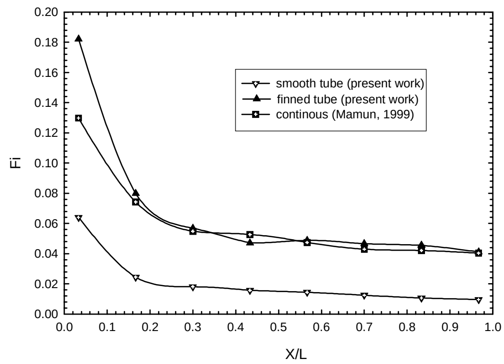

Figure 8: Effect of fin geometry on friction factor

So all the experimental data from this work passes slightly above the line predicted by Sieder and Tate (1936). This comparison concludes that heat transfer increases almost linearly with Reynolds number i.e. disturbance of flow increases the rate of heat transfer.

Figure 7 represents the variation of friction factor with Reynolds number. The friction factor is high near the entrance region, then sharply falls, after which it then remains almost constant. It can be noted that as the Reynolds number increases friction factor decreases. From the figure, it is worth mentioning that the friction factor becomes independent of Reynolds number at higher Reynolds number and remains constant after the value of Re = $4.0x10^4$. At the entrance region friction factor is high which may be due to settings of asbestos plate between the shaped inlet and test section. In this study it is observed that the friction factor is 2.0 to 7.0 times higher in finned tube than smooth tube. The higher frictional area is responsible for higher skin friction factor in finned tube.

A comparison has been made for experimental data for both finned and smooth tube with that of Mamun (1999) for in line finned tube. For all types of test section the friction factor is higher at the entrance section as shown in the Fig. 8. The cause may be due to the settings of asbestos plate explained earlier. The experimental data for finned tube is 3.0 to 4.0 times higher than those for smooth tube and 1.0 to 2.0 times higher for Re = $2.9x10^4$ than that of Mamun (1999). The higher frictional area contributes higher friction factor.

CONCLUSIONS

From the presentation of the experimental data and the subsequent analysis, one can conclude:

1. The friction factor of finned tube is about 4.0 times higher than that of smooth tube for Reynolds number ranges $2.0x10^4$ to $5.0x10^4$.
2. The friction factor is high near the inlet section and drops gradually to the value corresponding to the fully developed flow.
3. The heat transfer coefficient for finned tube is about 2.0 times higher than that of smooth tube.

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