# FORCED CONVECTION HEAT TRANSFER PERFORMANCE OF AN INTERNALLY FINNED TUBE

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***Abstract:*** *Heat transfer performance of T-section internal fins in a circular tube has been experimentally investigated. The T-finned tube was heated by electricity and was cooled by fully developed turbulent air. Inside wall temperatures and pressure drop along the axial distance of the test section at steady state condition were measured for different flows having Reynolds number ranging from $2x10^4$ to $5x10^4$ for both smooth and finned tubes. From the measured data, heat transfer coefficient, Nusselt number and friction factor were calculated. From the measured and calculated values, heat transfer characteristics and fluid flow characteristics of the finned tube are explained; the performance of the finned tube is also evaluated. For finned tube, friction factor on an average was 5 times higher and heat transfer coefficient was 2 times higher than those for smooth tube for similar flow conditions. The finned tube, however, produces significant heat transfer enhancement.*

***Key Words:*** *Heat Transfer, Internal Fin, Reynolds Number, Nusselt Number, Pressure Drop.*

## INTRODUCTION

Extended surfaces are commonly used in many engineering applications to enhance heat transfer. A number of studies have been performed in order to increase the heat transfer effectiveness and to reduce the dimensions and weight of heat exchangers. The necessity to reduce the volume and weight of heat exchanger has become more important in many engineering applications like electronic industry, compact heat exchanger sector, power plants, etc. Efficient design of heat exchanger with fins can improve system performance considerably. Among several available techniques for augmentation of heat transfer in heat exchanger tubes, the use of internal fin appears to be very promising method as evident from the results of the past investigations.

**Nomenclature**

| | |
|---|---|
| $A_{core}$ | Core flow area through an internally finned tube (= $A_n(1-H)^2$) ($m^2$) |
| $A_{fin}$ | Cross sectional area of a T-fin ($m^2$) |
| $A_n$ | Nominal flow area of the finned tube (= $\pi D^2/4$) ($m^2$) |
| $A_{xs}$ | Actual flow area of the finned tube (= $A_n$-6$A_{fin}$) ($m^2$) |
| $D$ | Inner diameter of the finned tube (m) |
| $D_h$ | Hydraulic diameter (= 4 ×cross-sectional area/ wetted perimeter) (m) |
| $H$ | Non-dimensional fin height (=2e/D) (-) |
| $L$ | Length of the test section (m) |
| $SA$ | Inside heat transfer area ($m^2$) |
| $T$ | Temperature (°C) |
| $c_p$ | Specific heat capacity of air (J/kg °C) |
| $e$ | Fin height (m) |
| $f$ | Friction factor (-) |
| $h$ | Heat transfer co-efficient based on nominal heat transfer area (W/m$^2$°C) |
| $k$ | Thermal conductivity of air (W/m °C) |
| $m$ | Mass flow rate (kg/s) |
| $p$ | Heated perimeter = $\pi$D (m) |
| $q$ | Input heat flux (W/m$^2$) |
| $v$ | Velocity of fluid (m/s) |
| $x$ | Axial location |
| $\Delta P$ | Axial pressure drop (N/m$^2$) |
| $\mu$ | Dynamic viscosity of air (N.s/m$^2$) |
| $\rho$ | Density of air (kg/m$^3$) |
| *Subscripts* | |
| act | Actual |
| avg | Average |
| b | Bulk |
| bx | Bulk local |
| f | Finned tube |
| ftx | Fin-tip local |
| h | Any quantity based on hydraulic diameter |
| n | Nominal (based on inside diameter) |
| $w$ | Wall |
| $x$ | Local |
| *Dimensionless numbers* | |
| Nu | Nusselt number based on nominal tube diameter = hD/ k |
| Pr | Prandtl Number = $\mu c_p$ / k |
| Re | Reynolds Number based on inside diameter = $\rho vD / \mu$ |

Extensive studies have been performed from the beginning of the twentieth century to understand the heat transfer characteristics inside tubes. A comprehensive summary of useful empirical correlations to determine heat transfer coefficient during forced convection flow inside smooth/rough ducts has been provided by Ozisic[1] and the Nusselt equation (Eq. 1) for low L/D ratio was recommended as follows:

$$Nu = 0.036\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{1/3}\left(\frac{D}{L}\right)^{0.055} \quad \text{for } 10 < \frac{L}{D} < 100 \qquad (1)$$

where Nu is average Nusselt number, Re is Reynolds number, Pr is Prandtl number, L is the length and D is the hydraulic diameter of the duct.

Heat transfer with disturbing the flow geometry has been already investigated thoroughly[2]. The first analytical study to predict the performance of tubes with straight internal fin in turbulent airflow was conducted three decades before[3]. Empirical correlations for predicting friction factor and Nusselt number for fins without helix were proposed as follows[4]:

$$f_h = 0.046\,\mathrm{Re}_h^{-0.2}(A_n / A_{xs})^{-0.5} \qquad (2)$$

$$Nu_h = 0.023\,\mathrm{Re}_h^{0.8}\,\mathrm{Pr}^{0.4}(A_{xs} / A_{core})^{0.1}(SA_{act} / SA_n)^{-0.5} \qquad (3)$$

Compact heat exchangers have large surface-area-to-volume ratios primarily through the use of finned surfaces. An informative collection of articles related to the development of compact heat exchanger has been presented by Shak et al.[5]. A comprehensive literature on principles of enhanced heat transfer has been also presented by Webb[6]. Numerical predictions of developing fluid flow and heat transfer in a circular tube with fins have been reported by many researchers[7-9].

An experimental investigation of fully developed, steady and turbulent flow in a longitudinal finned tube was performed[10]. That investigation used a two-channel, four beams, laser-Doppler velocimeter to measure velocity profiles and turbulent statistics of airflow seeded with titanium dioxide particles. They compared friction factor with different Reynolds numbers to literature results and showed good agreement for both smooth and finned tubes. Most recently, an experimental investigation of steady state turbulent flow heat transfer performance of circular tubes having six integral internal longitudinal fins was performed and found significant enhancement of heat transfer[11]. In the same setup, further investigations for heat transfer performance of internal rectangular finned tubes with in line-segmented[12] and non-segmented[13] fins have been conducted. The results of the study show that friction factor is 2.0 to 3.5 times higher for in line fins and 1.75 to 2.5 times higher for in line-segmented fins compared to smooth tube. Significant heat transfer augmentation was also reported for both types of fins.

In line with the above-mentioned studies in heat transfer in finned tube, present geometry was chosen in search of better heat transfer performance. The detail of the experimentation is explained in the following section.

## EXPERIMENT

### Experimental Set-up

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.

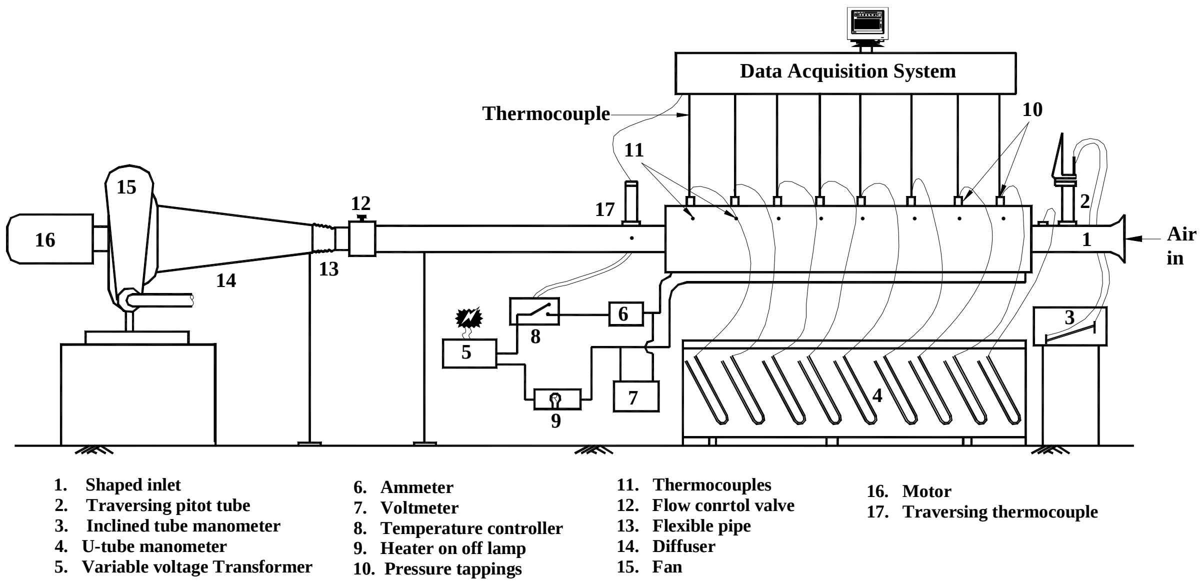


Figure 1(a). Schematic of the Experimental Set-up

**Test Section:** This is a brass-tube of 1500mm length with two types of geometry. One is smooth of 70mm inside diameter and the other is finned having six internal longitudinal T-section fins. A quarter of the cross-section of the finned tube is shown in Fig. 1(b) which clearly exhibits the position of thermocouples, insulation details and above all the dimensions. Similar casting methods are employed for both the tubes for getting comparable surface properties. Different radial holes are drilled along the axis of the test section for fitting thermocouples and pressure probes. 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.

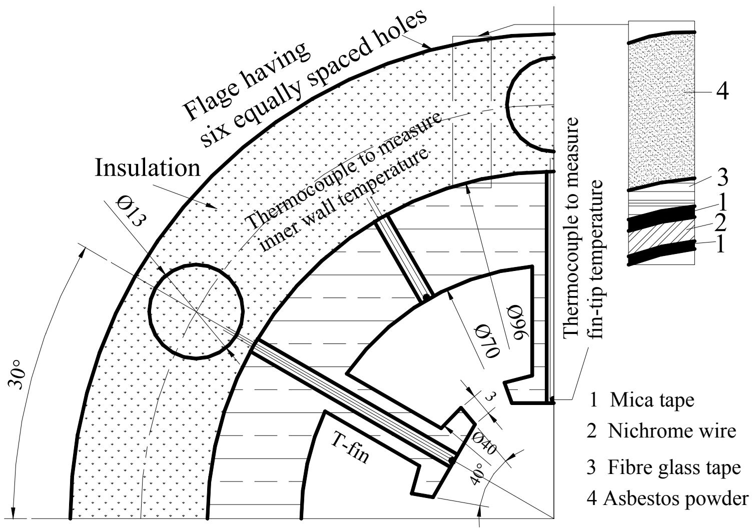


Figure 1(b). Partial cross sectional view of the test section

**Air Supply System:** It consists of a motor (16) operated suction-type fan (15) fitted downstream the test section. The 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 (14) made of mild steel is fitted to the suction side for minimizing head loss. To arrest the vibration of the fan a flexible duct (13) was fitted between the inlet section of the fan and a butterfly valve (12). The butterfly valve was used to control the flow rate of air in the range of Re = $2.0 \times 10^4$ to $5.0 \times 10^5$.

**Heating System:** Nichrome wire wound around the brass tube as shown in Fig. 1(b) supplied constant heat flux when it was connected to a 5-KVA power supply with the help of a magnetic contactor and temperature controller. The temperature controller (marked 8 in Fig. 1a) was set to read the outlet air temperature and 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. It is to mention that the electrical power input is not used to estimated heat taken by the air when it flows through the test section. Better estimation of heat input can be done from the mass flow and temperature measurement.

**Measurement System:** Other than power measurement, the inside wall temperatures and pressure along the axis of the test section and the air flow were measured. Flow of air through the system was measured at the inlet section with the help of a traversing pitot-static tube. Arithmetic mean method is employed to determine the radial position of the pitot-static tube for the determination of mean velocity. Pressure tapings for measurement of static pressure were fitted so carefully that it just touched the inner surface of test section. U-tube water manometers at an inclination of $30^o$ were fitted with the pressure tapings for better readability. The temperatures at the different axial locations of the test section were measured by K-type thermocouples connected with a data acquisition system. The temperature measuring locations are (i) tube center at the outlet of the test section, (ii) 8 axial locations along the inner wall of the test section, (iii) 8 axial locations along the fin-tip of the test section for finned tube. The thermocouple fitted at the center at the outlet of the test section is considered to read the bulk temperature of the air at the outlet. For smooth tube**,** 8 thermocouples were fitted at eight equally spaced axial locations along the inner wall of the test section to measure the wall temperature. For finned tube system, 16 thermocouples were fitted at eight locations having two thermocouples at an axial location: one at the inner wall of the tube and the other at the fin-tip.

**Procedure**

At first, the fan (15) as shown in Fig. 1 was switched on and allowed to run for about five minutes to get the transient effect died out. The flow of air through the test section was set to the desired value and kept constant with the help of a flow control valve (12). The electric heater around the test section was then switched on. The electrical power was adjusted (if necessary) with the help of the variac (5). The wall temperature response in a particular axial location was monitored and we waited for couple of hours until a nearly constant temperature at that location was attained. In addition, the air temperature response at the exit of the test section was monitored. The steady state condition was considered to be attained when the air temperature leaving the test section did not change for about 10 minutes after attaining a nearly constant wall temperature at any axial position. After such a steady state condition, thermocouple readings were recorded by the data acquisition system and manometer (4) readings were taken manually.

After each experimental run, the flow was changed with the help of the flow control valve keeping electrical power input constant. After waiting for steady state condition, desired data were recorded as per procedure narrated above.

**Data Reduction**

**Bulk temperature:** It is well known that bulk temperature is the energy-average temperature of a fluid flowing through a temperature field. It is very difficult to measure quickly. In the present study, the thermocouple set at the exit of the test section is considered to give the apparent bulk temperature at that location. The apparent bulk temperatures at the other axial locations have been calculated using the following relation:

$$T_{bx} = T_i + qp_n x / mc_p \tag{4}$$

where $T_i$, q, $p_n$, x, m and $c_p$ are, respectively, inlet fluid temperature, input heat flux, nominal heated perimeter, axial position, mass flow arte of fluid and specific heat. The input heat flux has again been calculated by Eq. 5.

$$q = mc_p (T_o - T_i) / p_n L \tag{5}$$

where $T_0$ is the thermocouple reading at the exit of the test section and L is its length.

**Heat transfer coefficient:** In this paper both local heat transfer coefficient ($h_x$) and overall heat transfer coefficient (h) are used and calculated by Eqs. 6 and 7 respectively.

$$h_x = q / (T_{wx} - T_{bx}) \tag{6}$$

$$h = q / (T_{w,avg} - T_{b,avg}) \tag{7}$$

The symbols in the above equations have their usual meanings. It is to be noted that for finned tube $T_{w,\,avg}$ has been calculated from the average of the average of $T_{wx}$ and $T_{ftx}$.

**Nusselt number:** In this paper both local Nusselt number ($Nu_x$) and overall Nusselt number (Nu) are used and calculated by Eqs. 8 and 9 respectively.

$$Nu_x = h_x D / k \tag{8}$$

$$Nu = hD / k \tag{9}$$

where D is the nominal tube diameter and k denotes the thermal conductivity of the fluid.

**Friction factor:** Friction factor at any axial location is calculated by the following relation (10) in this study. It gives the total friction factor f when x equals to L.

$$f_x = \Delta P \frac{D}{x} \frac{1}{2\rho v^2} \tag{10}$$

Where ΔP, ρ and v are axial cumulative pressure drop, density of fluid and velocity of the fluid respectively.

It is to be noted that the data were reduced in this study in terms of tube nominal inside diameter and area. This approach is generally recommended as it facilitates direct comparison of finned and smooth tube performance.

**Uncertainty Analysis**

The uncertainty analysis was performed by using the method of Kline and McClintock[14]. By this method the uncertainty of a variable R which is a function of independent variables $x_1$, $x_2$, $x_3$, ….., $x_n$ can be estimated by taking root-sum-square of the contribution of individual variables. The individual uncertainties of different variables measured in this study and the uncertainties of the calculated quantities are provided in Table-1.

Table 1. Uncertainties of the different quantities

| Measured quantities | Uncertainty (%) | Calculated Quantities | Uncertainty (%) |
|---|---|---|---|
| Temperature, T | 1.5 | Cross sectional area, A | 0.3 |
| Manomentric deflection, d | 3.2 | Velocity, v | 1.7 |
| Axial position, x | 0.1 | Input heat flux, q | 4.1 |
| Length, L | 0.0 | Pressure drop, ΔP | 5.8 |
| Diameter of the Tube, D | 0.1 | Friction factor, f | 6.8 |

**RESULTS AND DISCUSSION**

At first the temperature distributions along the tube wall and fin-tip for different flow conditions are presented in the immediate sub-section which is followed by explanations of heat transfer characteristics and fluid flow characteristics. Finally performance evaluation is given at the end of this section.

**Temperature Distribution**

During each experiment, the wall and the fin tip temperatures along the axis of the test section were measured. The measured temperature distributions have been explained in this section.

Figure 2 shows the variation of wall temperatures along the length of the smooth tube for four different Reynolds numbers at constant heat flux condition. From the figure it is clear that the wall temperature increases along the axial distance of the test section. At about x/L = 0.55, the wall temperature gets its maximum value while for x/L = 0.55 to 0.85 the temperature is almost constant, after which the temperature drops slightly at the downstream due to end effect. The end effect may be due to the physical contact between the test section and the down stream unheated tube. Ideally the variation of the wall temperature should be linear for constant heat flux heating as in the present study.

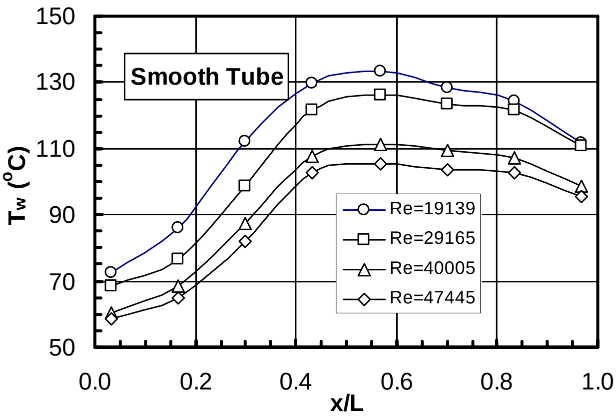


Figure 2. Wall temperatures along the axis of the smooth tube for different Re

Figure 2 also reveals that with the increase of Reynolds number, the wall temperature decreases for any axial position of the test section. Higher Reynolds number is a consequence of higher flow rate and ultimately better mixing of hot and cold fluid during its movement through the test section. All of these provide higher heat extraction capacity of the flowing fluid and result in the reduced wall temperature.

Figure 3 represents the variation of wall temperatures and fin-tip temperatures along the length of the tube for two Reynolds numbers. For a particular air flow, it is clear that the fin-tip temperature is always 6-10$^o$C lower than the wall temperature. It may be attributed due to the conduction resistance of the fin and its exposure to relatively colder fluid. For calculating length averaged wall temperature, local average of wall and fin-tip temperatures are considered.

Figure 4 compares the wall temperatures of smooth and finned tube for two comparable Reynolds numbers. It is worth mentioning that at inlet section the temperature gradient is high for both the tubes because the cold entering air takes away much more heat. It is interesting to note that unlike smooth tube there is a gradual increase in wall temperature for the finned tube with negligible end effect. The end effect refers to as an axial heat loss by conduction from the test section through connectors to the downstream section. It is to be noted again that for a fixed air flow and heat input, wall temperature of the finned tube is always lower than that of smooth tube. As the smooth tube has lower wetted perimeter and less mixing capability due to smaller frictional area, it has no such ability to transfer heat as quickly as the finned tube does. As a result the smooth tube wall gets warmer than its finned counterpart and exhibits significant heat loss due to end effect.

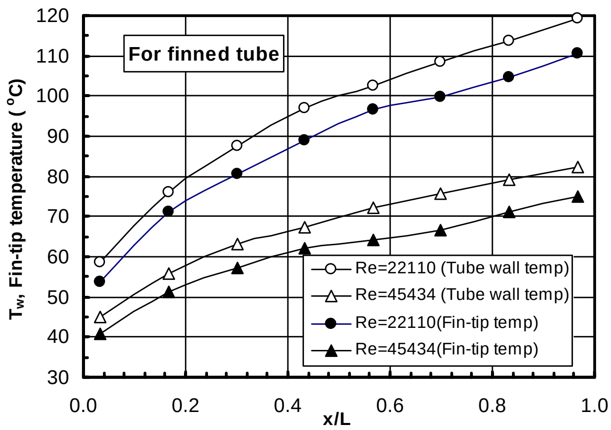


Figure 3. Axial variation of wall and fin-tip temperatures of finned tube

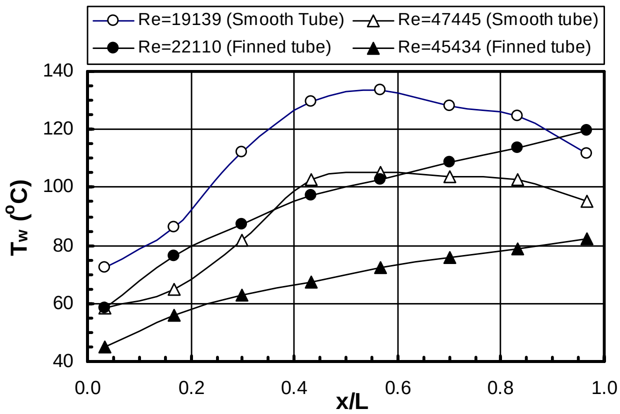


Figure 4. Comparison of wall temperature for smooth and finned tube

Figures 5(a) & 5(b) exhibit the variation of wall and bulk fluid temperatures along the axis of the smooth and finned tubes respectively for nearly same air flow (Re≈20000) and heat input. On the secondary axis, the variations of local heat transfer coefficients for those tubes are also shown for the same condition. The bulk fluid temperature increases linearly as air passes through the test section of tube as assumed by Eq. (4). It is clear that for smooth tube the bulk fluid temperature is lower than that of finned tube. Due to higher heat transfer from the inner surface of the tube, the bulk temperature is higher in finned tube. But smooth tube has lower wetted perimeter and less mixing of fluid compared to finned tube, and has no such ability to transfer heat as quickly as does the finned tube. Therefore, the bulk temperature of the smooth tube is lower. It is generally found that, at lower Reynolds number the bulk fluid temperature is higher. Because at lower Reynolds number the air velocity is low (having low mass flow) and air gets enough time to be heated, but at higher Reynolds number air gets less time to be heated and thus it is relatively colder at the same axial location. It is to be noted that total heat taken by turbulent flow of higher Re is obviously higher even though the bulk temperature is lower.

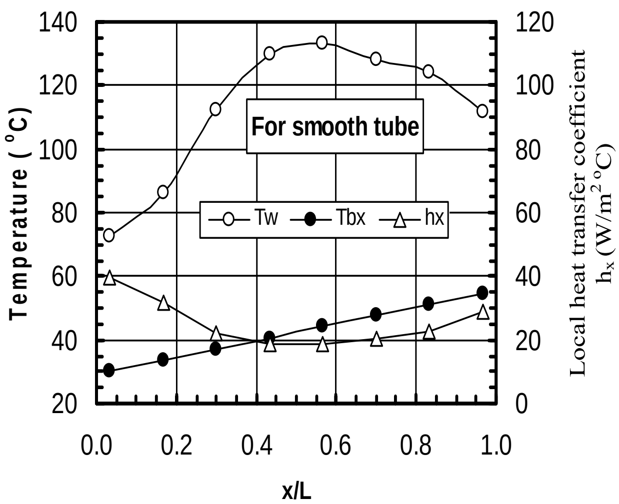


Figure 5a. Relationship between wall temperature, bulk temperature and local heat transfer coefficient for smooth tube

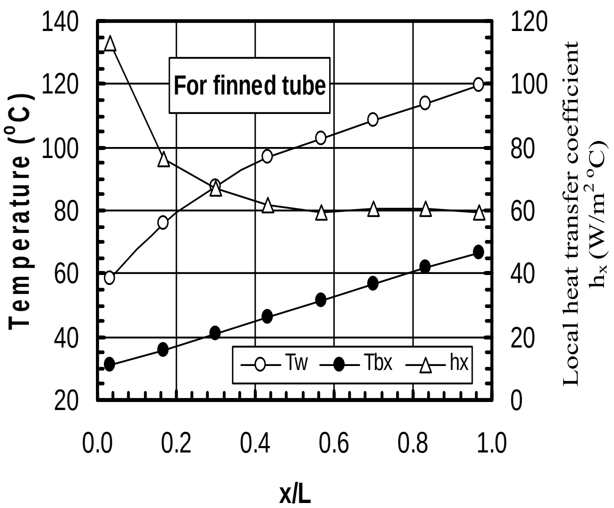


Figure 5b. Relationship between wall temperature, bulk temperature and local heat transfer coefficient for finned tube

Both the figures (Figs. 5a, 5b) also show the variation of local heat transfer coefficient for both smooth and finned tubes for Re≈20000 based on nominal diameter. The local heat transfer coefficient for finned tube is always higher than that for smooth tube. The heat transfer coefficient for finned tube is found to be nearly constant in the last half of the tube which may be because of nearly constant local radial temperature gradient. But for smooth tube, there is remarkable increase in the heat transfer coefficient at the exit of the test section because of lower radial temperature gradient due to end effect as mentioned earlier. For both tubes the coefficient is large at the entry of the test section where the flow can be considered as thermally developing. The thermally developing region seems to be ended for smooth tube at x/L=0.3 having x/D≈6 while for finned tube at x/L=0.4 having x/D≈8. Therefore, it may be mentioned that the thermally developed region in the finned tube starts a little later than that of in the smooth tube.

**Heat Transfer Characteristics**

In this section, the local heat transfer co-efficient and the overall Nusselt number are explained for both the tubes experimented with.

**Heat Transfer Coefficients:** Figure 6 exhibits variation of the local heat transfer coefficient along the axis of the finned tube where different symbols are for different Reynolds number. For any flow of air it is found that the heat transfer coefficient is large at the entrance of the test section due to the development of thermal boundary layer. The coefficient decreases sharply along the axial distance up to x/L = 0.4 after which it remains nearly constant demonstrating the thermal boundary layer be fully developed. At higher Reynolds number, the heat transfer coefficient is higher because of lower radial temperature gradient. For smooth tube, similar variation is observed but the corresponding values of the heat transfer coefficient are lower compared to the finned tube.

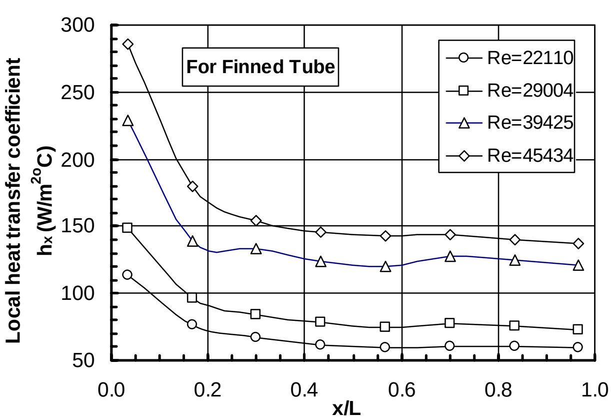


Figure 6. Axial variation of $h_x$ of the finned tube for different Reynolds number

**Nusselt Number:** Figure 7 shows the variation of overall Nusselt number (Nu) with Reynolds number for both finned and smooth tubes where the line-symbol-line curves are for present experimental data, solid line is Nu predicted by Carnavos (1980) where Reynolds number based on nominal diameter has been considered in using Eq. (3) and the dashed line is the prediction (1) for smooth tube. From this figure it is clear that there is an excellent agreement between present data and the Nussetl prediction (1). The figure demonstrates that Nu in the present study increases with Reynolds number with an exponent very close to unity. From the figure it is also clear that the overall Nusselt number for finned tube is 1.75 to 2.0 times higher than that of smooth tube. It is to be noted that the Nu prediction by Carnavos[4] for finned tube is even much lower than that of the smooth tube. Reason for this discrepancy is not clear yet.

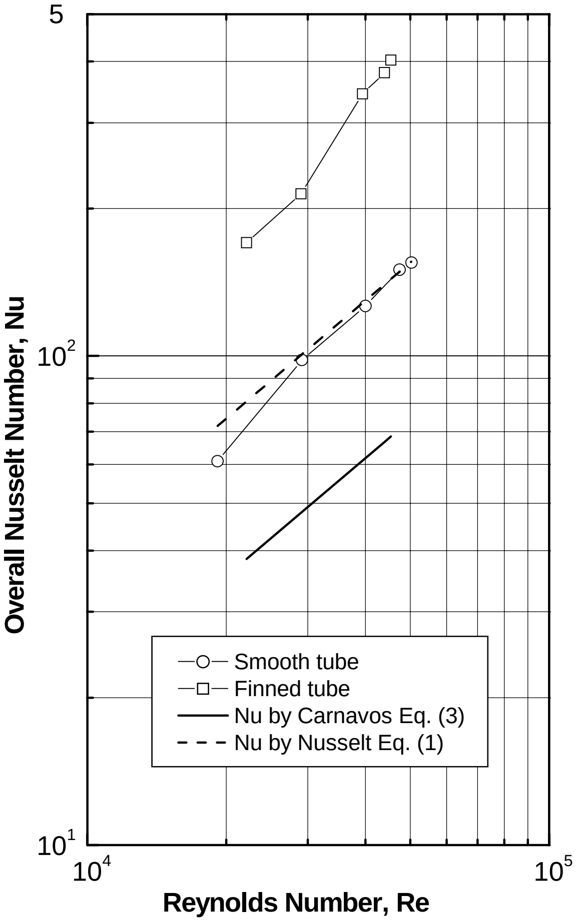


Figure 7. Comparison of overall Nusselt number for both smooth and finned tube

## Fluid Flow Characteristics

Both local and total friction factor for finned tube are presented in this section. Figure 8 shows the variation of friction factor along the axial distance for both finned tube where different symbols are for different air-flow having different Reynolds numbers. As obvious from the figure, the friction factor is high near the entrance region, then sharply falls up to x/L = 0.3, after which it remains almost constant. The higher friction factor at the entrance region may in part be attributed to presence of asbestos plate between the shaped inlet and test section. It is also clear from the figure that as the Reynolds number increases the friction factor decreases. Thin hydrodynamic boundary layer at higher Reynolds number may be responsible for this.

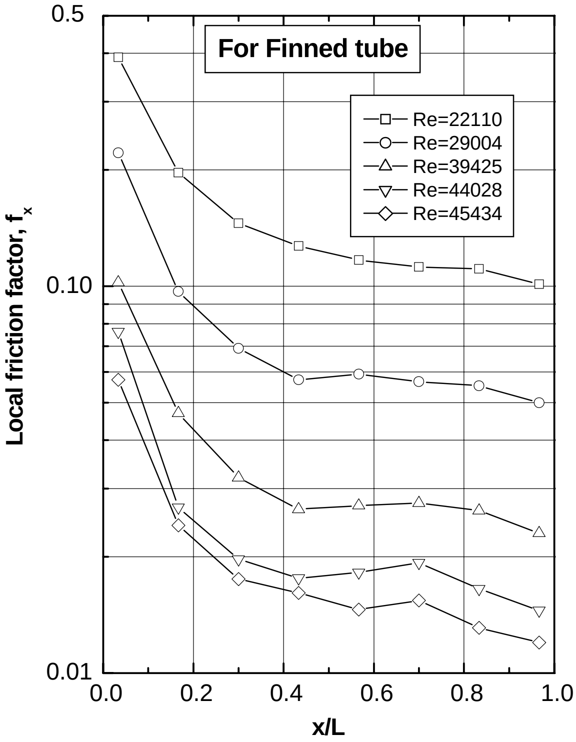


Figure 8. Friction factor along the length of the finned tube

Figure 9 exhibits a comparison of total friction factor for finned tube with other studies. The line-symbol-line data are for present study while solid line is the well known Blasius prediction for smooth tube and the dash-dot-dash line is the prediction (Eq. 2) by Carnavos[4]. Unlike existing prediction, the friction factor for finned tube is found to decrease sharply with flow Reynolds number. The friction factor in the finned tube is about 5 times enhanced compared to the smooth tube as shown by the Blasius equation.

## Performance Evaluation

Several performance criteria to evaluate the thermohydraulic performance of the enhancement techniques have been proposed[15,16]. In this paper, the criterion R3 outlined by Bergles[15] has been calculated and explained to quantify the benefits from the T-section finned tube.

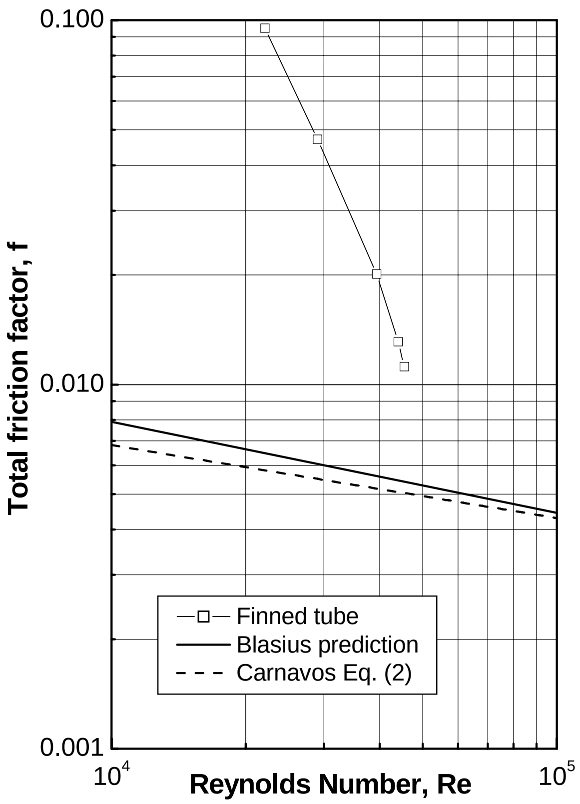


Figure 9. Comparison of total friction factor for finned tube

The criterion R3 is defined by R3 = $Nu_f / Nu_0$ where $Nu_f$ is the heat transfer obtained with T-section fin and $Nu_0$ is the heat transfer obtained with a smooth tube for equal pumping power and heat exchange surface area. The same pumping power requirement, Pm, for both the systems can be expressed as:

$$Pm = (Av)\left(f\frac{L}{D}\frac{\rho v^2}{2}\right) \tag{11}$$

The above equation can be reduced to Eq. 12.

$$Pm \cong cfA\,\mathrm{Re}^3 \tag{12}$$

Therefore, to satisfy the constraint of the criterion R3, $Nu_0$ is evaluated at the equivalent smooth tube Reynolds number $Re_0$ which matches

$$\mathrm{Re}_0{}^3 = \frac{f_f\,\mathrm{Re}_f{}^3}{f_0}. \tag{13}$$

Figure 10 shows the performance parameter R3 for the T-section fin in the present limited Reynolds number experiments, where equivalent Reynolds numbers are calculated using Eq. 13. From the figure it is clear that heat transfer enhancement seems to be about 2 fold using T-section fin in the present flow conditions. A wide range flow of air through the finned tube could be investigated in order to find an optimum Reynolds number that may maximize the heat transfer enhancement.

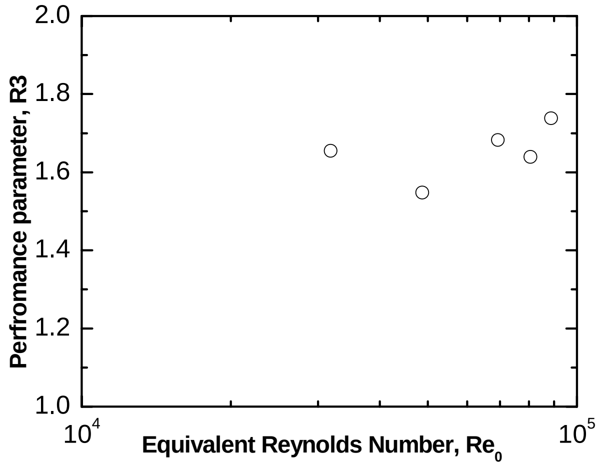


Figure 10. Performance evaluation

## CONCLUSIONS

Hydrodynamically fully developed and thermally partially developed flow through tubes has been experimented in order to test the efficiency of a T-section fin. A quantitative comparison has been made for friction factor and heat transfer for both finned and smooth tube. Performance of the fin has also been evaluated. From the data presentation and their subsequent analysis, following conclusions can be drawn:

(i) The thermally developed region starts at about x/L=0.3 having x/D very close to 6 for smooth tube and at about x/L=0.4 having x/D approaching to 8 for finned tube for a comparable Reynolds number based on nominal diameter.

(ii) The heat transfer is enhanced as high as 2 times in the T-section finned tube compared to smooth counterpart in the Reynolds number range from $2.0\times10^4$ to $5.0\times10^4$.

The friction factor has been enhanced 5 times on an average in the finned tube compared to the smooth tube for the Reynolds number range as mentioned above.

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