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# HEAT TRANSFER IN TURBULENT FLOW THROUGH TUBE WITH LONGITUDINAL STRIP INSERTS

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## ABSTRACT

An experimental investigation has been carried out to study the convective heat transfer in a tube with longitudinal inserts with different shapes of strips in turbulent flow. Three different shapes of strip (Y, X and ✳- configuration) were fabricated from mild steel and inserted into the smooth tube successively. The test section was heated electrically and the flow was varied in the range of Reynolds number $2.0x10^4$ to $5.0x10^4$. Air velocity, air inlet and outlet temperature, longitudinal strip surface temperatures, wall temperature and pressure drops along the axial length of the test section were measured to analyze the friction factor, Nusselt number, and heat transfer co-efficient. With longitudinal strip inserts, heat transfer co-efficient was enhanced with corresponding increase in friction factor and pumping power. 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.

**Keywords:** Longitudinal strip inserts, Friction factor, Pumping power.

## 1. INTRODUCTION

The internal inserts are used to modify the flow characteristics of the fluid to enhance the heat transfer rate. Extended surface inserts are those that are of extruded shape inserted in the tube. The tube is then drawn to provide good thermal contact between the wall and the insert. The insert reduces the hydraulic diameter and acts as an extended surface. 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 ncreased heat transfer rate.

The first analytical study to predict the performance of tubes with straight internal fins in turbulent air flow via a mixing length model was conducted by Patankar et al. [1]. The analytical results showed that the local fin heat transfer co-efficient varied significantly along the 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 co-efficients 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[2]. The strips were crossed-type, rectangular and square in cross-section with different aspect ratio. 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-length strips. They found that friction factor reduces by 8-58 percent and Nusselt number reduces by 2-40 percent for short-length strips. For regularly –spaced strips elements, friction factor increase by 1-35 percent and Nusselt number increases by 15-75 percent.

Hsieh and Huang [3] experimentally studied the heat transfer and pressure drop characteristics of water flow in horizontal tubes with/without longitudinal inserts. Testing was performed on bare tubes and tubes with square and rectangular as well as crossed-strip 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 about a factor of 16 at Re≤ 4000, while the friction factor rise was only about a factor of 4.5 at Re≤ 4000.

Uttarwar and Raja Rao [4] carried out experimentally on isothermal pressure drop and convective heat transfer to servotherm 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 a smooth tube at constant pumping power and 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.

Dipprey and Sabersky [5] have been experimentally investigated heat and momentum transfer in smooth and rough tubes at various Prandtl numbers. From their experiment, increase in heat transfer coefficient due to roughness of as high as 270% has been obtained.

A longitudinal rectangular plate insert in the tubes of heat exchangers 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 [6].

Solanki et al. [7-8] conducted experiment and theoretical studies of laminar forced convection in tubes with polygon inner cores. Forced convection heat transfer in doubly connected ducts bounded externally by a circle and internally by a rectangular polygon of various shapes was analyzed using a finite element method. Hydro dynamically and thermally developed, steady, laminar flow of a constant property fluid was investigated. An insulated outer tube and constant heat flux at the inner core were considered. Temperature profiles as well as Nusselt numbers were presented.

In this paper heat transfer to air flowing in a tube with three different longitudinal strips were studied. Results with longitudinal strip configurations and smooth tube are compared.

## 2. EXPERIMENTAL SET UP AND METHODOLOGY

### 2.1 Experimental Facility

The experimental facility consists of Test section, Inlet section, Air supply system and Heating arrangement. Test section is shown schematically in the Figure 1.

#### 2.1.1 Test Section

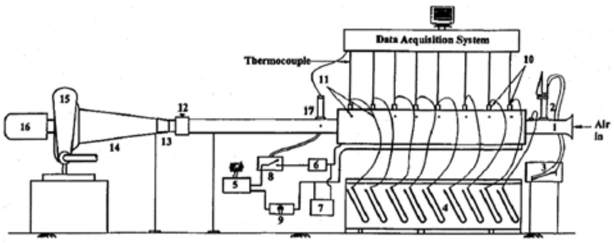
Smooth tube is a circular brass tube having 70mm inside diameter and 1.5m length. The outside diameter of the longitudinal strip was almost same as the inside diameter of the smooth tube so that the insert may tightly fitted against the tube wall. After inserting the strip, the two halves of the smooth tube were clamped and to prevent leakage the appropriate adhesive was pressed into the joint of the tube.

#### 2.1.2 Inlet Section

The unheated inlet section (shaped inlet) cast from aluminium was of same diameter as of test section. The pipe shaped inlet, 533mm long was made integral to avoid any flow disturbances at upstream of the test section to get fully developed flow in the test section.

#### 2.1.3 Air Supply System

Induced draft air flow was created using a electrically operated centrifugal blower. A gate valve placed between the test section and the down stream was used to control the flow rate of air.




| | |
|---|---|
| 1. Shaped inlet | 10. Pressure tappings |
| 2. Traversing pitot tube | 11. Thermocouples |
| 3. Inclined tube manometer | 12. Flow control valve |
| 4. U-tube manometer | 13. Flexible pipe |
| 5.Variable voltage Transformer | 14. Diffuser |
| 6. Ammeter | 15. Fan |
| 7. Voltmeter | 16. Motor |
| 8. Temperature controller | 17.Traversing thermocouple |
| 9. Heater on off lamp | |

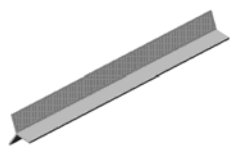
Fig 1. Schematic Diagram of the test rig

#### 2.1.4 Heating System

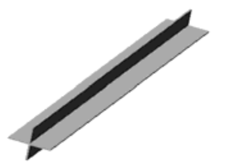
An electric heater (made of Nichrome wire) was used to heat the test section at constant heat flux. Power for heating was supplied by a regulated A-C supply with a 5KVA variable voltage transformer through a magnetic contactor and a temperature controller. A temperature controller was fixed to sense the outlet air temperature to provide signal for switching the heater off or on automatically.

#### 2.1.5 Longitudinal Strip Inserts

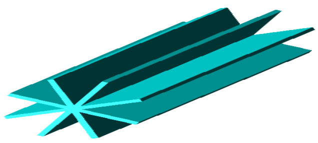
Three different shapes of longitudinal strips were investigated in the experiment as shown in Figure 2



Y- Shaped
($120^0$ angle)



X- Shaped
($90^0$ angle)



✳ - Shaped

($45^0$ angle)

Fig 2. Geometry of Longitudinal Strip

### 2.2 Experimental Data Measurement

Flow of air through the test section was measured using Diesel as monometric fluid at the inlet section with the help of a traversing pitot. The static pressure tapings were made at the inlet of the test section as well as equally spaced 8 axial locations of the test section as shown in the Figure 1. The temperatures at the different axial locations of the test section and the air inlet temperature were measured with the help of K-type thermocouples.

## 3. EXPERIMENTAL RESULTS AND DISCUSSION

**Bulk and Wall Temperature:**

Bulk temperature in the test section at various axial locations measured by the following equation:

$$T_{bx} = T_i + \frac{Q^{/}(\pi D_i x)}{\dot{m} c_p} \quad (1)$$

Where,

$$Q^{/} = Q/A_S \text{ and } A_s = \pi D_i L \quad (2)$$

Figure 3 shows the variations of wall and bulk fluid temperature for smooth tube and also tube with inserts. From figure 3 it is clear that the wall temperature increases along the axial length and reaches its maximum value and then the temperature drops slightly at the downstream due to end effect. The end effect is due to the conductive heat loss between the test section and downstream-unheated tube.The bulk fluid temperature could be explained by the figure 3. Figure show that bulk fluid temperature increases linearly as air passes through the smooth tube as the bulk fluid temperature was calculated using equation 1.

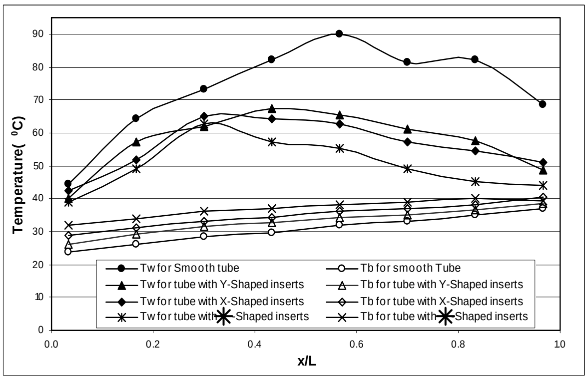



Fig 3. Bulk and Wall Temperature distribution at Re=22000

The vertical distance between the wall temperature and bulk temperature used to calculate the fully developed heat transfer co-efficient. From Fig. 3 one can see that the smooth tube has the largest temperature difference. Keeping the other parameter fixed, the smallest temperature difference in fig.3 indicates the best heat transfer performance. At this stage, the heat transfer increases with a insert for a variety reasons. The blockage of the flow due to the presence of the insert increases the flow velocity. The cross flow pattern of fluid by the presence of insert improves the mixing.

**Heat Transfer:**

Local heat transfer co-efficient computed by the following equation:

$$h_x = \frac{Q^{/}}{(T_w - T_b)_x} \quad (3)$$

Figure 4 shows the variation of local heat transfer co-efficient ($h_x$) along the axial distance (X/L) of smooth tube for different Reynolds number. The figure indicates that the local heat transfer co-efficient is large in the entrance region due to the development of thermal boundary layer. Then it gradually decreases up to a certain point, after which the thermal boundary layer could be considered fully developed. Again from the figure, it is clear that at higher Reynolds number the local heat transfer co-efficient is higher.

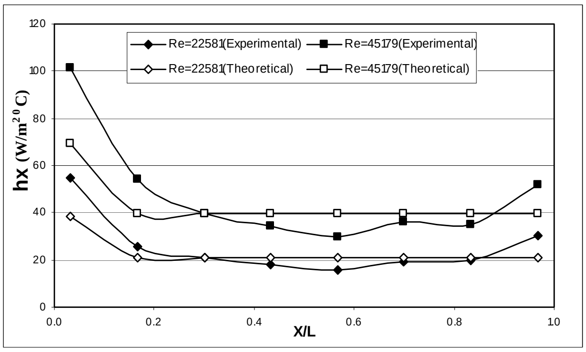



Fig 4. Local Heat transfer co-efficient for smooth tube

Figures 5-7 show the variation of local heat transfer co-efficient ($h_x$) of a tube with longitudinal strip inserts along the axial distance (X/L) at different Reynolds number.

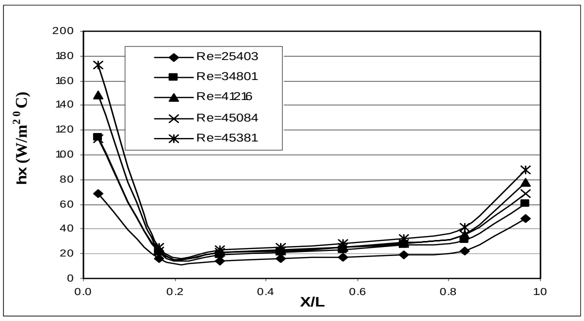



Fig 5. Local Heat transfer co-efficient for tube with Y-Shaped longitudinal strip

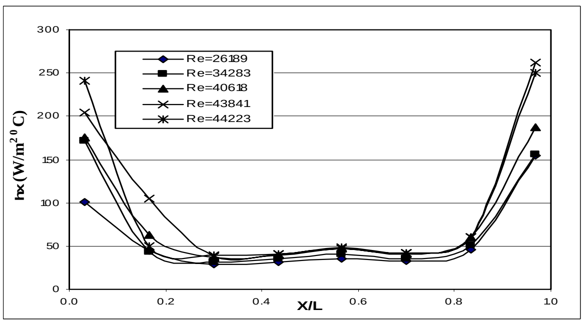



Fig 6. Local Heat transfer co-efficient for tube with X-shaped longitudinal strip

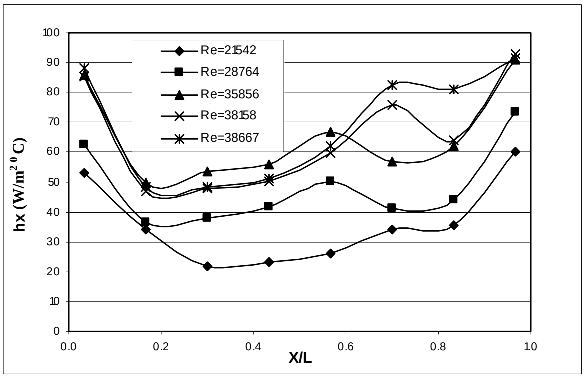



Fig 7. Local Heat transfer co-efficient for tube with ✳- shaped longitudinal strip

The figures indicate that the co-effecients are large at entry of the test section as observed for smooth tube but the developed region starts somewhat earlier than that observed in smooth tube.

Figure 8 shows the variation of average heat transfer co-efficient as function of Reynolds number for different tubes. The figure indicates that the average heat transfer co-efficient increases with the increase of Reynolds number. From the figure it is also clear that the average heat transfer co-efficient increases as the number of strips in the longitudinal inserts increase compared to that of smooth tube.

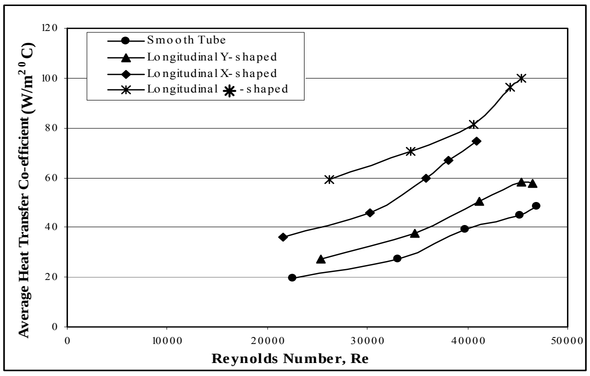



Fig 8. Comparison of average heat transfer co-efficient with Reynolds number for both smooth and longitudinal strip inserted tube

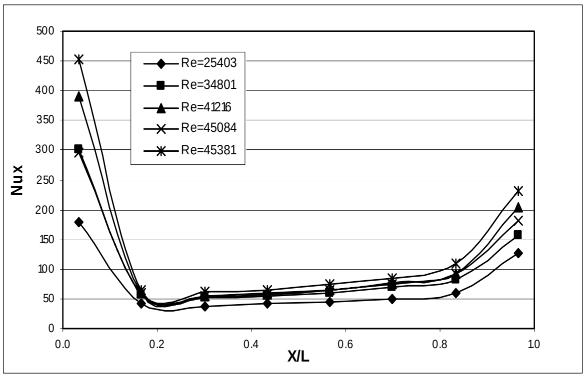



Fig 9. Local Nusselt Number for tube with Y-shaped longitudinal strip

Figure 9-11 show the variation of local Nusselt number ($Nu_x$) of a tube with longitudinal strip inserts along the axial distance (X/L) at different Reynolds number. The figures indicate that the co-efficient are large at entry of the test section as observed for smooth tube but the developed region starts somewhat earlier than that observed in smooth tube. It is occurred due to the presence of longitudinal strips.

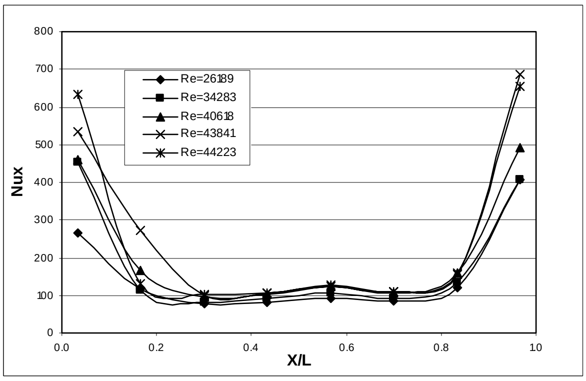



Fig 10. Local Nusselt Number for tube with X-shaped longitudinal strip

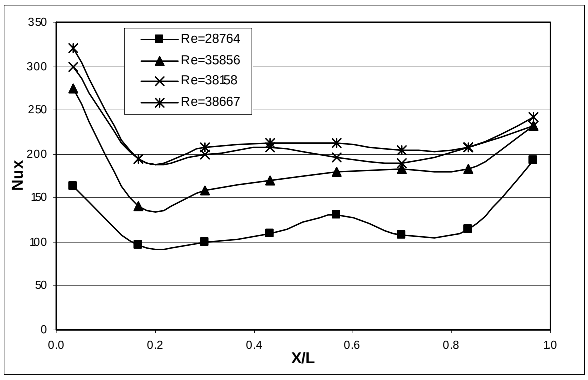



Fig 11. Local Nusselt Number for tube with ✳- shaped longitudinal strip

**Friction Factor:**

The friction factor was calculated from the equation

$$F_i = \frac{(-\Delta P / X) D_i}{\frac{1}{2}\rho V^2} \tag{4}$$

Figure 12-15 show the nature of friction factor with dimensionless distance for both smooth tube and tube with longitudinal strips. From the figures it can be noted that the friction factor is high near the entrance region, then sharply falls up to a certain point and finally remains almost constant. At the entrance region friction factor is high which may be due to settings of asbestos plate between shaped inlet and test section. In this experimental data it is observed that friction factor for tubes with longitudinal strip inserts varies from1.2 to 2.5 times higher than that of smooth tube for comparable Reynolds number. The presence of small vortices behind the inserts may be responsible for higher friction factor.

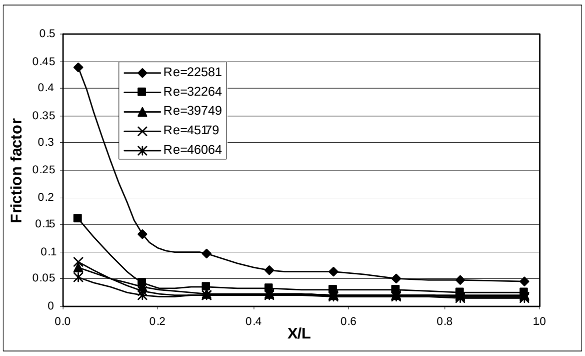



Fig 12. Friction Factor for smooth tube

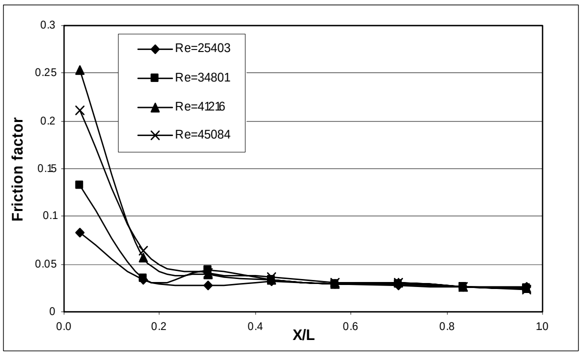



Fig 13. Friction Factor for tube with Y-shaped longitudinal strip

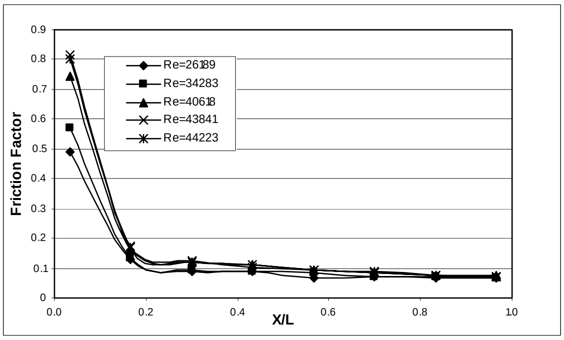



Fig 14. Friction Factor for tube with X-shaped longitudinal strip

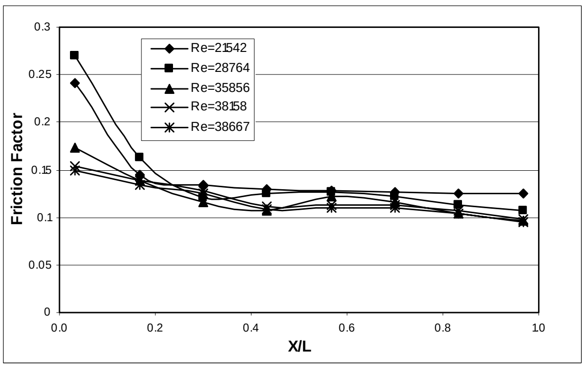



Fig 15. Friction Factor for tube with ✳- shaped longitudinal strip

**Pumping Power:**

Total Pumping power, $P_m$ computed by the following equation:

$$P_m = \frac{\Delta P}{\rho} \dot{m} \tag{5}$$

Figure 16 presents the variation of pumping power with Reynolds number for both smooth tube and a tube with inserts. For all the tubes pumping power increases as the Reynolds number increases. From the figure it can be noted that the pumping power of longitudinal strip inserted tube is higher than that of the smooth tube. The required pumping power for tubes with longitudinal strip inserts varies up to 4 times higher than that of the smooth tube.

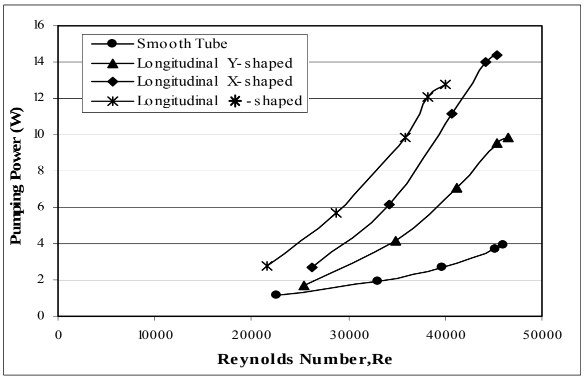



Fig 16. Comparison of Pumping power with Reynolds number for both smooth and strip inserted tube

## 4. CONCLUSIONS

Steady state fluid flow and heat transfer performances of a smooth tube as well as a tube with longitudinal strip inserted were studied experimentally. The study has indicated that the longitudinal inserted tube enhance heat transfer rate at a cost of increased pumping power. From the presentation of the experimental data and the subsequent analysis conclusions of the present study are:

- Heat transfer co-efficient for tube with longitudinal strip inserts increases than that of smooth tube. For comparable Reynolds number, average heat transfer co-efficient increases up to 1.4 times, up to 2 times and up to 3 times than that of smooth tube for tube with Y-shaped, X-shaped and ✳- shaped longitudinal inserts respectively.
- Nusselt number is high at the entrance region and it decreases gradually up to a certain point corresponding to the value of fully developed flow.
- The local friction factor is high near the inlet section and drops gradually to the fully developed flow. For comparable Reynolds number, friction factor increases up to 1.2 times, up to 1.5 times and up to 2.2 times than smooth tube compared to tubes with Y-shaped, X-shaped and ✳- shaped longitudinal inserts respectively.
- The required pumping power increases up to 2 times, up to 3.5 times and up to 4 times than that of smooth

tube for tube with Y-shaped, X-shaped and ⁎-shaped longitudinal inserts respectively.

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## 7. NOMENCLATURE

| Symbol | Meaning | Unit |
|---|---|---|
| $A_s$ | Surface area | $(m^2)$ |
| $C_p$ | Constant Pressure Specific heat of air | (J/KgK) |
| $D_i$ | Inside diameter of tube | (m) |
| $F_i$ | Friction Factor | (-) |
| $h_x$ | Local heat transfer co-efficient | $(W/m^{2}\,{}^{0}C)$ |
| L | Length of the test section | (m) |
| $\dot{m}$ | Mass flow rate of fluid | (Kg/s) |
| $Nu_x$ | Local Nusselt number | (-) |
| $\Delta P$ | Pressure drop along axial length | (Pa) |
| Q | Heat input to the Test section | (W) |
| $Q^{/}$ | Heat flux | $(W/m^2)$ |
| Re | Reynolds number | (-) |
| $T_{bx}$ | Local Bulk Temperature of the fluid | $({}^{0}\,C)$ |
| $T_i$ | Inlet Temperature of the fluid | $({}^{0}\,C)$ |
| $T_w$ | Wall Temperature | $({}^{0}\,C)$ |
| ρ | Density of air | $(Kg/m^3)$ |
| V | Mean velocity of air | (m/s) |
| X | Axial Length | (m) |