Experimental Investigation the Natural Heat Transfer Coefficient of Vertical Copper Cylinder

Pratik Gupta[1] Yash Kadluva[2] Rishabh Vishwakarma[3] Suraj Wadhe[4]
[1,2,3,4]Student
[1,2,3,4]Department of Mechanical Engineering
[1,2,3,4]Acropolis Technical Campus, Tillore, Indore, Madhya Pradesh-452020, India

Abstract— The present experimental study deals with natural convection through vertical cylinder. The experimental setup is designed and used to study the natural convection phenomenon from vertical cylinder in terms of average heat transfer coefficient. Also practical local heat transfer coefficient along the length of cylinder is determined experimentally and is compared with theoretical value obtained by using appropriate governing equations. The setup consist of copper cylinder of length 500mm and outside diameter 38mm with air as a working fluid. The results indicate the temperature variation along the length of cylinder and the comparative study of theoretically and practically obtain local heat transfer coefficient.

Keywords: Natural Convection, Vertical Cylinder, Boundary Layer, Local Heat Transfer Coefficient

I. INTRODUCTION

While natural convection from vertical heated cylinder has been studied in a variety of ways over the course of nearly 100 years, the interdependent relationship between boundary layer regime transition heat transfer in each regime and curvature.

Free convection or natural convection is a spontaneous flow arising form non homogenous fieldof volumetric (mass) force (gravitational, centrifugal, coriolis, electromagnetic etc.)

$$\Delta\vec{F} = \Delta(\rho\vec{g}) = \Delta\rho\vec{g} + \rho\Delta\vec{g}$$

If density variation $\Delta\rho$ is caused by special non-uniformity of a temperature field, then a flow arising in the Earth gravitational field is called thermal gravitation convection. The density variability may also result from non-uniform distribution of any component in a mixture or from chemical reactions, difference in phase densities or from surface tension forces at from surface tension from at the phase interface (in this case concentration diffusion or convection is implied).

Natural convection heat transfer has always been of particular interest among heat transfer problems. In natural convection, fluid motion is caused by natural means such as buoyancy due to density variations resulting from temperature distribution. Natural convection plays vital role in the heat transfer in case of many applications such as electrical component transmission lines, heat exchangers and many other places. Many experimental studies have been performed during the last three decades and interesting results have been presented.

Y.A.Cengel discussed the natural convection phenomenon in case of vertical cylinders and governing equations to determine heat transfer coefficient. L.Davidson et.al studied the natural convection phenomenon in vertical shell and tube also the effect of different Intel conditions and geometrical dimensions on the developed thermal and velocity boundary layers. Also it was shown that the larger the inlet velocity, the larger Nusselt number. Especially near the transition region this difference is large and gradually vanishes in the fully turbulent region. L.J.Crane studied th natural convection over the vertical cylinder at very large Prandtl number and discussed how, high Prandtl number affect free convection through vertical cylinder C.O.Popiel studied the effect of curvature of the cylinder where the thickness of the boundary layer is considerable i,e thicker.

Also some result of calculations of boundary layer using modified integral method is obtained. Hari.P.Rani studied numerically unsteady natural convection of air and the effect of variable viscosity over an isothermal vertical cylinder and concluded that as the viscosity increases the temperature and skin friction coefficient increases, while velocity near the wall and Nusselt number decrease. P.Ganesaner presented numerical solution for transient natural convection over the vertical cylinder under the combine buoyancy effect also it observed that time taken to reach steady state increases with Schmidt number and decreases as combined buoyancy ratio parameter increases. A. Shiriet studied experimental analysis of natural convection in near wall region of vertical cylinder and measured the mean and turbulence quantities in the near wall region, where the varying thermal properties also effect the flow due to the strong temperature gradient there. A new set of boundary layer equations are established to represent the variable properties of the flow in this region. This experimental investigation also reveals that the strong temperature gradients adversely affect both the steady and unsteady temperature results because of the conduction. H.A.Mohammed studied mixed convection heat transfer inside a vertical circular cylinder for upward and downward flows, for hydro dynamically fully developed and thermally developing laminatr flow under constant wall heat flx boundary conditions. The results slows that the surface temperature values for downward flow were higher than that for upward flow but it was lower than that for horizontal cylinder. J.Wojthowkstudied experimentally the laminar free convective average heat transfer in air from isothermal vertical slender cylinder having circular cross-section using a transient technique. The present experiment of natural convection through the vertical cylinder of brass having specific dimensions gives the analysis of temperature distribution along the length of the cylinder. To measure temperature, at different level the thermocouples are fitted. The average heat transfer coefficient and local heat transfer coefficient are estimated using energy balance in the system and the same are found out by using appropriate governing equations. The result indicate the variation of the heat transfer coefficient with the length of the cylinder and this results are compared for both practically determined and theoretically evaluated heat transfer coefficient.

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II. NOMENCLATURE

- Q_{in} : Heat input to heater coil,(W)
- V: Applied Voltage,(V)
- I: Applied Current,(A)
- h_{avg}: Average surface heat transfer coefficient, (W/m^2K)
- T_{avg}: Average surface temperature,(^0K)
- T_∞: Ambient temperature, (^0K)
- D: Diameter of the cylinder, (m)
- g: Acceleration due to gravity =9.81 m/s^2
- v: Kinematic viscosity, (m^2/s)
- Ra: Rayleigh number
- Gr: Grashoff number
- Pr: Prandtl number
- Nu: Nusselt Number
- β: coefficient of volumetric expansion, (K^{-1})
- T_{mf}: Mean film temperature, (^0K)
- h_1: Local heat transfer coefficient, (W/m^2K)
- K: Thermal conductivity of air, (W/mK)

III. EXPERIMENTATION AND METHODOLOGY

The apparatus consist of a copper tube fitted in a rectangular duct in vertical fashion. The duct is open at top and bottom, and forms an enclosure and serves the purpose of the undistributed surrounding. One side of the duct is made of Perspex for visualization. An electrical heating element is kept in vertical tube which it turns to heat loss bu tube to surrounding air is by natural convection. The temperature of the vertical tube is measured by seven thermocouple which are fixed on the tube by drilling holes along the tube wall. The heat input to the heater is measured by an ammeter and voltmeter and is varied by a dimmerstat. The vertical cylinder with the thermocouple position is shown in Fig.1. The tube surface is polished to minimize the radiation loss.

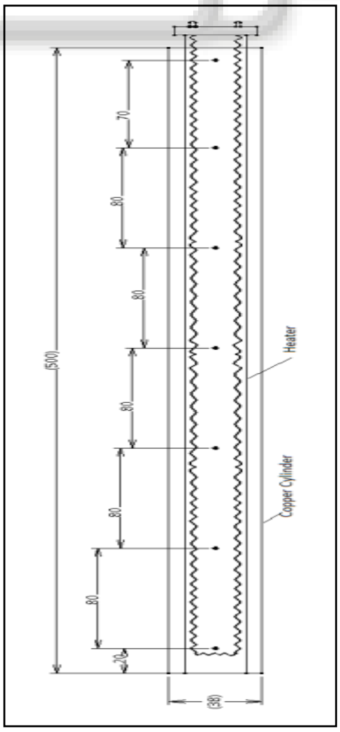

Fig. 1: Heater assembly for vertical cylinder (Dimensions are in mm)

The voltage regulator, ammeter and digital voltmeter have been used to control and measure the input power to the working pipe as shown Fig.2. The apparatus has been allowed to turn on at least for 4 hours before the steady state condition was achieved. The readings of all thermocouple have been recorded. Record the ambient temperature. The input power to the heater could be changed to cover another run in shorter period of time and to obtained stead-state.

The net power input to heater is given as:

$Q_{in}=V*I=h_{avg}A\Delta T$

Where,

$A=\pi DL$

$\Delta T=T_{savg}-T_\infty$

The equation (1) gives practical value of heat transfer coefficient.

Now,

To make comparison of this practical value with the theoretical value, we have to use following governing equations for natural convection through vertical cylinder.

$Gr=\beta g_c{}^3\Delta T/v^2$ (2)

$Ra=Gr*Pr$ (3)

Where;

β= thermal expansion coefficient=$(1/T_{mf})$ K^{-1}

$T_{mf}= (T_{savg}+T_\infty) / 2$

All the other properties of the air are determined at T_{mf} (K)

Hence using free convection correlations

We have, For $10^4<Ra<10^9$

$Nu = 0.59*(Ra)^{0.25} = hL_c/K$

Where, K is thermal conductivity of air at T_{mf}(K)

From equation (4) value of the theoretical heat transfer coefficient is determined.

Thus using this procedure it is possible to find average practical and theoretical heat transfer coefficient.

Procedure to find local practical heat transfer coefficient and local theoretical heat transfer coefficient.

For h_1(Practical):

$Q_{in} = (VI*L_{c1}) / L = h_1A_1\Delta T$

$A_1 = \pi DL_{c1}$

$T_{save}= T_1$

$T_{mf} = (T_{save}+ T_\infty) /2$

Where,

T_1 = Temperature at 1st thermocouple

L_{c1}= Length of cylinder upto 1st thermocouple from base

L = Total length of cylinder

For h_1(Theoratical)

All the properties are determined at T_{mf}(K)

$Gr= \beta gL_{c1}{}^3\Delta T / v^2$ (6)

$Ra= Gr*Pr$(7)

$Nu= h_1L_{c1}/K = 0.59(Ra)^{0.25}$ (8)

From equation (5) local practical heat transfer coefficient and equation (8) local theoretical heat transfer coefficient is determined.

By following same procedure local heat transfer coefficients at various positions are determined.

The first graph (Fig.3) shows variation of the temperature along the length of cylinder. It is observed that as the length of cylinder increases from bottom to top, temperature also on increasing up to length nearly about half the length of cylinder, and then decreases continuously after the maximum temperature point is achieved. The second

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graph (Fig.4) shows comparison between practical and theoretical heat transfer coefficient along the length of cylinder, it is observed that value of practical heat transfer coefficient is less than theoretical heat transfer coefficient of transition region of thermal boundary layer. After this, the value of practical heat transfer coefficient increase which is more than theoretical heat transfer coefficient due to turbulence flow of air.

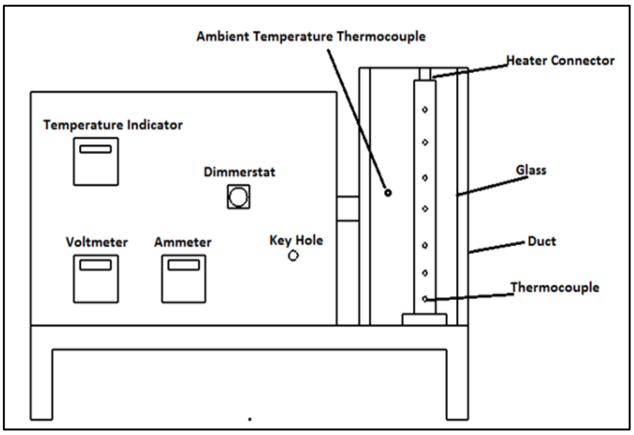

Fig. 2: Schematic Diagram of Experimental Setup

IV. Result and Discussion

The first graph Fig.3 shows variation of temperature along the length of cylinder. It is observed that as the length of cylinder increased from bottom to top, temperature also goes on increasing upto length nearly about half the length of cylinder and then decreases continuously after the maximum temperature point is achieved. The second graph fig.4 shows comparison between practical and theoretical heat transfer coefficient along the length of cylinder.

It is observed that value of practical heat transfer coefficient is less than theoretical heat transfer coefficient upto length nearly half of cylinder when both the lines are intersecting due to existence of transition region of thermal boundary layer. After these the value of practical heat transfer coefficient increases which is more than theoretical heat transfer coefficient due to turbulence flow of air.

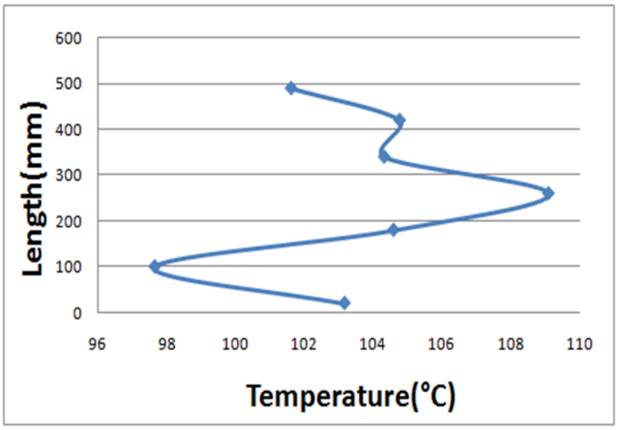

Fig. 3: Temperature Variation along the Length of Cylinder

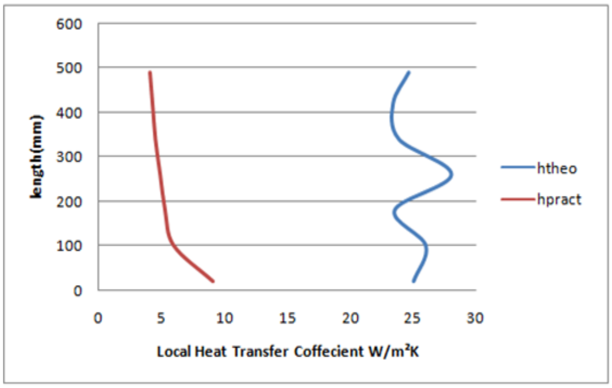

Fig. 4: Variation of Local Heat Transfer Coefficient along the Length of Cylinder

V. Conclusion

The heat transfer coefficient having maximum value at the beginning because starting of development of boundary layer i.e. Thin layer and decrease in upward direction due to thickening of boundary layer. This trend is maintained nearly up to half-length and beyond this there is little variation in the value of local heat transfer coefficient because of the presence of transition and turbulent boundary layer (Fig.5). The last point shows somewhat increase in the value which is attributed to end loss causing a temperature drop.

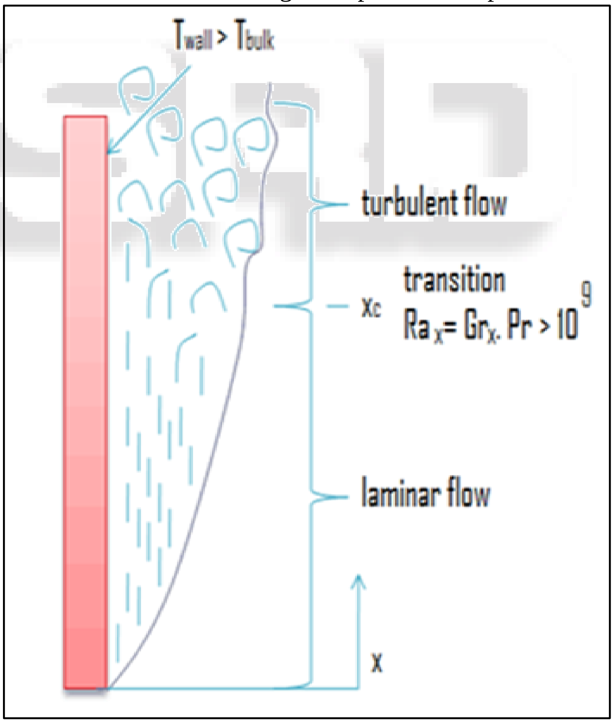

Fig. 5:

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