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Experimental Analysis for Natural Convection Heat Transfer through Vertical Cylinder

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***Abstract* - Convection is the mode of heat transfer which generally takes place in liquid and gases. Consider a fluid flow over a heated surface, the molecules of fluid adjacent to the surface, absorb heat and become hot, on heating the molecules become lighter due to decrease in density, they rise up and the cold molecules of higher density come down in contact of heated surface, in this way, motion of molecules sets up in fluid due to developed density gradient. The thermal protection system in many systems is widely accomplished applying natural convection process due to its low cost, reliability and easy maintenance. Typical applications include the heat exchangers, cooling of electronic equipment and nuclear reactors, solar chimneys and Trombe walls in building industry, etc. Experimental studies of natural convective flows in two dimensional channels, opened to ambient conditions at both end sections, are extensively reported in the literature but most of them are treated severely idealized experimentally.**

***Index Terms* – Natural Convection, vertical cylinder.**

I. INTRODUCTION

In contrast to the forced convection, natural convection phenomenon is due to the temperature difference between the surface and the fluid and is not created by any external agency.

The present experimental set up is designed and fabricated to study the natural convection phenomenon from a vertical cylinder in terms of the variation of local heat transfer coefficient along the length and also the average heat transfer coefficient and its comparison with the value obtained by using and appropriate correlation. Natural convection heat transfer takes place by movement of fluid particles within to solid surface caused by density difference between the fluid particles on account of difference in temperature. Hence there is no external agency facing fluid over the surface. It has been observed that the fluid adjacent to the surface gets heated, resulting in thermal expansion of the fluid and reduction in its density. Subsequently a buoyancy force acts on the fluid causing it to flow up the surface. Here the flow velocity is developed due to difference in temperature between fluid particles. The following empirical correlations may be used to find out the heat transfer coefficient for vertical cylinder in natural convection.

$$Nu = 0.53 \quad (Gr.\Pr)^{1/3} \text{ for } Gr.Pr<10$$

$$Nu = 0.56 \quad \text{for } 10< Gr.\ Pr <10$$

$$Nu = 0.13 \quad (Gr.\Pr)^{1/3} \text{ for } 10 < Gr.Pr<10$$

Where, Nusselt number $Nu = hL/k$

$$Pr = \text{Prandtl number} = \mu C_p/k$$

$$Gr = \text{Grashof number} = \frac{L^3 \beta g (T_s - T_a)}{\nu^2}$$

Properties Of Air

Temp ^{0}C	Density Kg/m^3	Kinematic Viscosity V x 10^6 m2 /s	Prandil Number Pr	Thermal Diffusivity m^2/nr	Specific Heat Cp J/Kg K	Thermal Conduct i vity K x 10^3	Coefficient of Viscosity x 10^6 Ns/m^2 or Kg/m
-50	1.584	9.23	0.728	45.7	1013	20.35	14.61
-40	1.515	10.04	0.728	49.6	1013	21.17	15.20
-30	1.453	10.80	0.723	53.7	1013	21.98	15.69
-20	1.395	11.61	0.716	68.3	1009	22.79	16.18
-10	1.342	12.43	0.712	52.8	1009	23.61	16.67
0	1.293	13.28	0.707	67.7	1005	24.42	17.16
10	1.247	14.16	0.705	72.2	1005	25.12	17.65
20	1.205	15.06	0.703	77.1	1005	25.93	18.14
30	1.165	16.00	0.701	82.3	1005	26.75	18.63
40	1.128	16.69	0.699	87.5	1005	27.56	19.12
50	1.093	17.95	0.698	92.6	1005	28.26	19.61
60	1.060	18.97	0.696	97.9	1005	28.96	20.10
70	1.029	20.02	0.694	102.8	1009	29.66	20.59
80	1.000	21.09	0.692	108.7	1009	30.47	21.08
90	0.972	22.10	0.690	114.8	1009	31.28	21.48
100	0.946	23.13	0.688	121.1	1009	32.10	21.87
120	0.898	25.45	0.686	132.6	1009	33.38	22.85
140	0.854	27.80	0.684	145.2	1013	34.89	23.73
160	0.815	30.09	0.682	158.0	1017	36.40	24.52

II. EXPERIMENTAL SETUP

The apparatus consists of a brass tube fitted in a rectangular vertical duct. The duct is open at the top and bottom and forms an enclosure and serves the purpose of undisturbed surrounding. One side of the duct is made up of perspex for visualization. An electric heating element is kept in the vertical tube which in turn heats the tube surface. The heat is lost from the tube to the surrounding air by natural convection. The temperature of the vertical tube is measured by seven thermocouples. The heat input to the heater is measured by an ammeter and a voltmeter and is varied by a dimmerstat. When a hot body is kept in still atmosphere, heat is transferred to the surrounding fluid by natural convection. The fluid layer in contact with the hot body gets heated, rises up due to the decrease in its density and the cold fluid rushes in to take place. The process is continuous and the heat transfer takes place due to the relative motion of hot and cold fluid particles.

The heat transfer coefficient is given by:

$$h = \frac{Q - Q_R}{A_S(T_s - T_a)} \qquad (1)$$

$$\frac{hL}{k} = A\left[\frac{gL^3\beta\Delta T}{\vartheta^2} \times \frac{C_p\mu}{k}\right]^n \qquad (2)$$

Where:

h = average surface heat transfer coefficient (W/m^2 °C)

Q = Heat transfer rate V. I (watts)

A_S = Area of the heat transferring surface =Π .d L (m^2)

T_S = Average surface temperature = $\frac{T_1+T_2+T_3+T_4+T_5+T_6+T_7}{7}$

T_a = Ambient temperature in the duct = T_8 °C

Q_R = Heat loss by radiation = $\sigma . A . \epsilon (T_S^4 - T_a^4)$

Where

σ = Stefan Boltzmann constant = 5.667x 10-8 W/m2 K4

ϵ = Emissivity of pipe material = 0.06

T_S & T_a = Surface and ambient temperatures in °K respectively.

The surface heat transfer coefficient, of a system transferring heat by natural convection depends on the shape, dimensions and orientation of the fluid and the temperature difference between heat transferring surface and the fluid. The dependence of h on all the above mentioned parameters is generally expressed in terms of non-dimensional groups as follows:

Where

L = A characteristic dimension of the surface.

K= Thermal conductivity of fluid

ϑ = Kinematic viscosity of fluid

μ = Dynamic viscosity of fluid0

C_P = Specific heat of fluid

β = Coefficient of volumetric expansion for the fluid

g = Acceleration due to gravity.

$\Delta T = [T_S - T_a]$

III. CONTROL PANEL AND INSTRUMENTATION SPECIFICATIONS:

- Digital Temperature Scanner: Micro-Controller based 10 Channel Scanner for RTD type Sensor with Thumbwheel Switch. Make: Nutronics, India
- Digital Voltmeter: Single Phase Range 0-300 V AC. Size: 48(H)x 96(W) x 130mm(D) Make Nutronics, India
- Digital Ammeter: Single Phase Range 0-2 Amp AC. Size: 48(H)x 96(W) x 130mm(D) Make Nutronics, India
- Cam Operated Rotary Switch with ON/OFF Switching Position For Heater Operation and Mains Supply
- LED Main Supply Indicator

- Continuously Variable Autotransformer : Single Phase - 2 Amp, Open Type Air Cooled
- Heater: 300 Watt Band Type Heater
- Thermocouple: RTD Type 7 Nos.

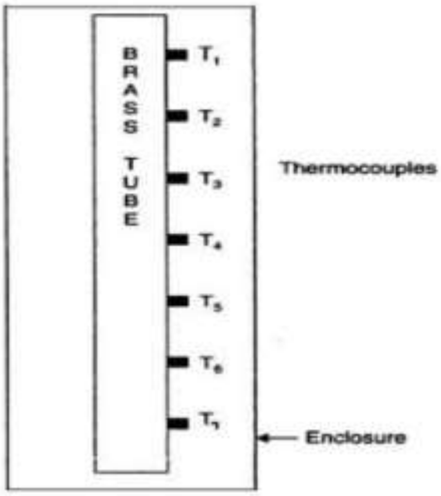

Fig. Natural Convection

IV. OBSERVATIONS

TABLE:

Sr. No.	V Volts	I Amp	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_S	T_8	Q
1	20	1.2	82	83	93	102	99	94	93	92	36	24

$$T_S = \frac{T_1+T_2+T_3+T_4+T_5+T_6+T_7}{7}$$

$$T_S = 92\ °C$$

$$A_S = \Pi .d\ L$$

$$A_S = 0.0596\ m^2$$

Calculate the value of average surface heat transfer coefficient neglecting end losses using equation (1)

Calculate and plot (Fig.4) the variation of local heat transfer coefficient along the length of the tube using:

$$h = Q/[A_s(T-T_a)]$$

$$h = 7.19 \text{ w/m}^2 \text{ °C}$$

$$\Delta T = T_S - T_a$$

$$\Delta T = 56 \text{ °C}$$

$$T_f = \frac{T_S+T_a}{2}$$

$$T_f = 64 \text{ °C}$$

$$\beta = \frac{1}{T_f+273} \text{ K}^{-1}$$

$$\beta = 2.958\times10^{-3} \text{ K}^{-1}$$

$$G_R = \frac{gL^3\beta\Delta T}{\vartheta^2}$$

$$G_R = 53.47\times10^7$$

Compare the experimentally obtained value with the predictions of the correlation equation (3) or (4).

Experimental heat transfer coefficient = 10.19 w/m2

Theoretical heat transfer coefficient = 13 w/m2

V. RESULTS AND DISCUSSIONS

The heat transfer coefficient is having a maximum value at the beginning as expected because of the just staring of the boundary layer and it decreases as expected in the upward direction due to thickening of layer and which is laminar one. This trend is maintained up to half of the lengths (approx.) and beyond that there is little variation in the value of local heat transfer coefficient because of the transition and turbulent boundary layers. The last point shows somewhat increase in the value of heat transfer coefficient which is attributed to end loss causing a temperature drop.

The comparison of average heat transfer coefficient is also made with predicted values are somewhat less than experimental values due to the heat loss by radiation.

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