EFFECT OF NATURAL CONVECTION HEAT TRANSFER FROM HORIZONTAL SQUARE CHANNEL

By

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ABSTRACT

Experiments have been conducted on square duct to investigate natural convection heat transfer from the outer surface of axially horizontal square ducts. Steady State Heat Transfer characteristics are studied under laminar and transition regions. Constant heat flux heating element is used to heat the duct from the center and 10 equidistant points are located on each side of the duct to measure temperature from all the four sides using IR-Thermometer. Correlations were developed in the form of circumferential averaged Nusselt number, modified Rayleigh number.

Keywords: Square Ducts, Natural Convection, Experimental Heat Transfer, IR-Thermometry.

INTRODUCTION

The operation of many engineering systems results in the generation of heat. This can cause serious overheating problems that lead to reduced performance and at sometimes even failure of the system. The heat generated within a system must be dissipated to its surrounding in order to maintain the system at its recommended working temperatures for effective and reliable functioning. Convection heat transfer strongly depends on the fluid properties such as dynamic viscosity, thermal conductivity, density, and specific heat, as well as fluid velocity. It also depends on the geometry and roughness of the solid surface, in addition to the type of fluid flow (such as streamlined or turbulent). During the past few decades, many researchers investigated the influence of various parameters on natural convection heat transfer consisting of vertical and horizontal plates. Natural convection from horizontal circular cylinder for different boundary conditions [1] and [2] for wide range of Prandtl numbers were studied. Correlating equations were proposed [3, 4] to perform numerical analysis for investigating the natural convection heat transfer phenomenon from a vertical plate enclosed in a horizontal cylinder. Numerical analysis using finite difference method to identify unsteady natural convection heat transfer in a square cavity under laminar conditions was reported [5]. This extensive study includes the variation of heating element position and temperature as parameters for investigation. Observations were made from various configurations with reference to vertical surfaces for different fluid medium [6,7]. The natural convection phenomenon was analyzed for various surfaces subjected to porous media [8,9]. Experimental and numerical correlations subjected to laminar and turbulent regions were outlined [14,12], Experimental investigations on vertical cylinders were reported by [10,11,15]. An experimental study on natural convection explains heat transfer from square cylinders [13]. However, data on natural convection in respect to non-circular ducts are limited. The present study is claimed at the development of natural convection and regression equation in respect to natural convection around non-circular ducts.

1. Experiment Setup and Procedure

Figure 1 describes the experimental setup which consists of square duct (50 mm X 50 mm) made from aluminum extrusion (6063-T6). Aluminum (emissivity =0.17) was chosen to reduce the radiation heat transfer such that the generated heat will dissipate in convection mode only. Axial conduction is significantly influenced by the thermal conductivity of end cap material. Temperatures on either side of the end caps are tabulated for evaluating heat transfer via conduction. A temperature measuring instrument IR-Thermometer (fluke instruments), voltmeter and ammeter are used to measure power supplied to the

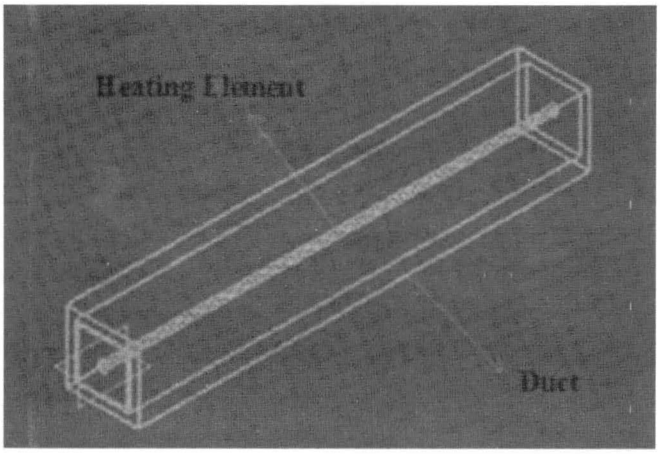

Figure 1. Arrangement of Heating Element into the Duct

system and dimmerstat. The emissivity of the duct material is 0.17. A long-straight heating element 19 mm OD is kept at the center of duct to heat the duct from inside to out. The continuous operating temperature is maintained below 110°C to investigate laminar and transition regions. Temperatures are measured at 45 locations (40 on the surface of duct, two are on inner surface of end cap, four are on outer surface of end cap and one is for measurement of ambient temperature). Two end caps made from fiber glass with thermal conductivity k=0.1 W/m-K are used to close the duct end in order to prevent heat loss in this direction. Electrical energy supplied to the heating element is regulated using 1-ϕ dimmer stat (10A) and the regulated power supply is measured using an ammeter connected in series and a voltmeter connected across the power supply. A Tailor-made arrangement is made to orient the duct in horizontal orientation.

2. Analysis of the Experiment

Under steady conditions, five experiments in total were conducted at various heat loads that correspond to laminar and transition regions and, all the experiments were repeated twice for consistency. IR thermometer is calibrated using K-type digital temperature indicator and adjusted its emissivity value to be 0.17. Regulated Electrical energy supplied to the heating element located at the center of the duct and the heat generated over the duct surface will be dissipated in terms of convection, radiation and conduction from the ends of the duct. The arrangement is such that the heat generated dissipates by convection and radiation via the duct surface, while by conduction through the end caps of the duct. The sample data generated in terms of circumferential averaged temperature and overall averaged temperature are presented in Table 1.

The data have been subjected to analysis by calculating the circumferential averaged temperature at equidistant locations and overall averaged temperature around the duct surface (6). The following equations were employed to evaluate non dimensional numbers such as Nusselt and Rayleigh numbers.

$$E = \text{Electrical Power Input} = V * I$$

$$= Q_{conduction} + Q_{convection} + Q_{radiation} \quad (1)$$

Radiation heat transfer from the duct is,

$$Q_R = \epsilon \sigma A_s (\bar{T}^4 - T_a^4) \quad (2)$$

Conduction through fiber glass from the two ends of the duct is given as,

$$Q_f = KA_c \frac{T_{in} - T_{out}}{L} \quad (3)$$

Heat transfer by convection from the duct will be,

$$Q_c = E - Q_f - Q_R = hx A_s (T_s - T_a) \quad (4)$$

Circumferential averaged local temperature and, overall averaged temperature were evaluated from,

$$T_s = \frac{Ts1 + Ts2 + Ts3 + Ts4}{4}, \quad (5)$$

$$\bar{T} = \frac{T_s}{10} \quad (6)$$

Local Values of Nusselt number and modified Rayleigh numbers were obtained using,

$$Nu_x = \frac{hx}{K} \quad (7)$$

$$Ra_x^* = \frac{g\beta q_c x^4}{\nu K \alpha} \quad (8)$$

S.no	X/l	T_{s1}	T_{s2}	T_{s3}	T_{s4}
1	0.1	104	90	105	89
2	0.2	104	93	104	94
3	0.3	108	95	110	97
4	0.4	111	97	110	97
5	0.5	114	96	114	95
6	0.6	108	97	107	97
7	0.7	109	94	110	94
8	0.8	105	93	106	94
9	0.9	103	93	105	93
10	1	102	89	102	90

Voltage	42
Current	0.96
E	40.32
Tin	67.6
Tout	38
Ta	28.6

Table 1. Typical experimental data in the form of temperature around eh circumferences at various locations along the duct for a sample heat load

3. Results and Discussions

Circumferential averaged temperatures at various locations for different heat loads are presented in Figure 2, from which it may be noted that, the temperature in general, increases from the inlet end of the duct up to the center of the duct. Thereafter, the temperature is observed to decrease from the center towards the outlet end of the duct. The temperatures are higher at the higher heat loads; this trend is more pronounced at elevated heat loads. The heat transfer coefficients were evaluated (4) for the conditions explained in Figure 2. The data are plotted in Figure 3. It is observed from the plot that at higher temperature, the local convection heat transfer coefficients are observed to be minimum.

The observation explained in Figures 2 and 3 are elaborated in Figures 4 and 5. In Figure 4, the non-dimensional temperatures at various locations along the duct are presented. While Figure 5 shows the local heat transfer coefficients at various locations along the duct. From Figures 4 and 5, it is evident that the convective heat transfer coefficient exhibits decreasing trend with an increase in the temperature.

Local Nusselt numbers and modified Rayleigh numbers were evaluated from the local heat transfer coefficient data (7) and (8) explaining the relation between local Nusselt number (Nu_x) and modified Rayleigh number (Ra^*)

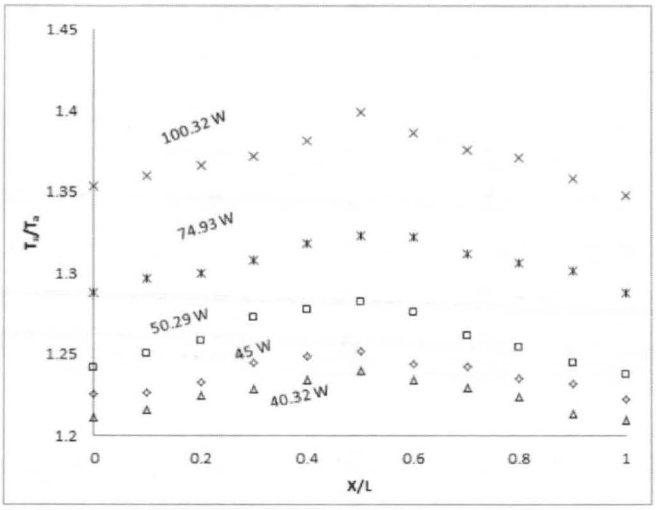

Figure 2. Circumferential averaged Non-Dimensional Temperature (T_x/T_a) correspond to non-dimensional distance from the leading edge of the duct

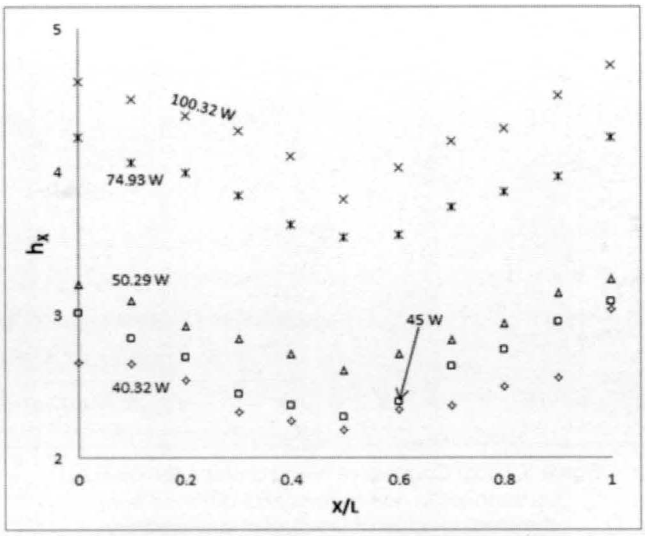

Figure 3. Local Convective heat transfer coefficient (h_x) corresponds to non-dimensional distance from the leading edge of the duct

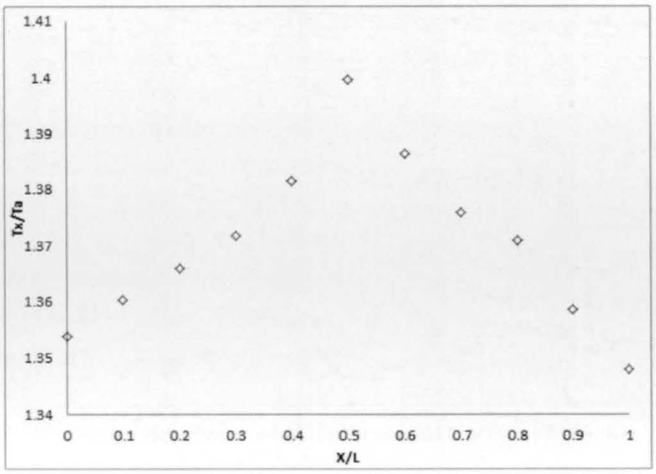

Figure 4. Circumferential averaged Non-Dimensional Temperature (T_x/T_a) correspond to non-dimensional distance from the leading edge of the duct at arbitrary heat load

in Figure 6, it is seen that as Ra^* increases, Nu_x increases. Regression analysis was performed on the data in order to obtain correlation between these non-dimensional numbers. The experimental data provides best fit with R^2 value of 98.93 with a power law fit ($y = a\,(x)^n$) curve. The constant values of 'a' and 'n' from the regression fit equation are obtained to be a= 0.108 and n = 0.2526, and the final equation is obtained in the form of,

$$Nu_x = 0.1084\,(Ra_x)^{0.2526}$$

$$10^7 < Ra_x < 10^{11}$$

The above regression fit is in good agreement under laminar and transition regions.

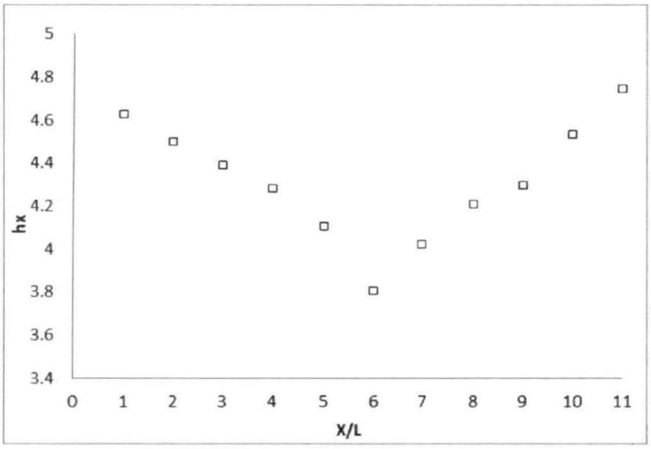

Figure 5. Local Convective heat transfer coefficient (h_x) corresponds to non-dimensional distance from the leading edge of the duct at any arbitrary heat load same as that of figure 3

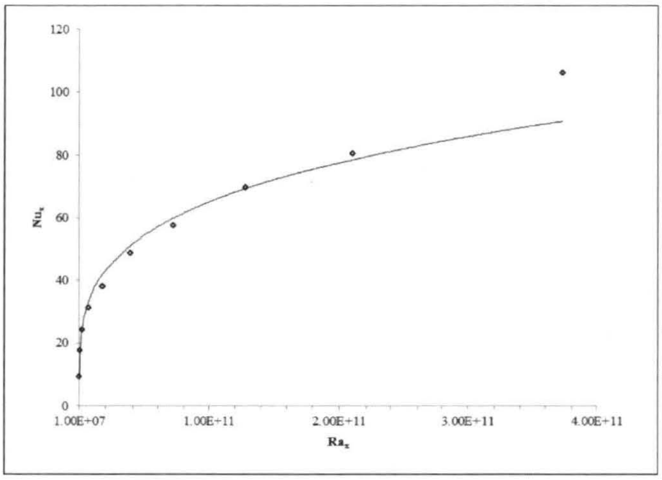

Figure 6. Local Nusselt number (Nu_x) corresponds to modified Rayleigh number (Ra_x^*)

Conclusions

Experiments were conducted on square duct to investigate natural convection phenomenon around the duct under laminar and turbulent regions. At any constant heat flux condition, it is observed that the maximum temperatures were recorded at the center (from the leading edge) of the duct. The reverse trend is observed with respect to convective heat transfer coefficient also. It is observed that at lower heat fluxes the end effects due to conduction losses are almost negligible, edge effects are significant at elevated heat fluxes. At any particular length of the duct, increment of temperatures correspond to the increment of length of the duct during first half, but the reverse trends are reported during the second half (next to the middle of the duct). Regression analysis is performed to generate correlations in terms of Nusselt number and Rayleigh number, power law fit trend line with R^2 value of 98.93 provided with the regression constants.

List of Symbols

- V Voltage (V)
- I Current (A)
- E Electrical input power (w)
- Q_f Heat conducted through fiber glass.
- Q_C Heat transferred from the duct by convection.
- Q_R Heat transferred from the duct by radiation.
- σ Stefan – Boltzmann constant. (5.67×10^{-8} W/m^{2}k^4)
- € Emissivity of aluminum (0.17)
- A_s Surface area of the duct, m^2
- Ac Cross sectional area of the fiber glass, m^2
- T_s Surface temperature on the duct. °C
- T_a Ambient temperature. °C
- K Thermal conductivity of fiber glass =0.04 W/m^2–k
- T_{in} Inside temperature of the fiber glass, °C
- T_{out} Outside temperature of the fiber glass. °C
- W Width of the fiber glass, m
- L Length of fiber glass, m
- h Convective heat transfer coefficient, W/m^{2}k
- T_s Surface temperature on the duct, °C
- T_{s1} Average temperature at top surface, °C
- T_{s2} Average temperature at bottom surface, °C
- T_{s3} Average temperature at front surface, °C
- T_{s4} Average temperature at bottom surface, °C

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