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Research Article

Experimental Investigation on Natural Convection Heat Transfer from Inclined Square Ducts

T.V.S.M.R.Bhushan[†*] and K.Vijaya Kumar Reddy[‡]

[†]Department of Mechanical Engineering, Nalla Malla Reddy Engineering College, Hyderabad. India.
[‡]Department of Mechanical Engineering, J.N.T.U. Hyderabad, India

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Abstract

Natural convection heat transfer from inclined square ducts is studied experimentally. Experiments have been conducted in still air with duct oriented at an angle of 45^0. In the present study two ducts having different area ratio are considered and uniform heat flux conditions are maintained. Variable heat inputs are applied to the centrally located resistance heating element in order to understand convection phenomenon in laminar and transition regions. Experimental data have been reduced in terms of Nusselt and modified Rayleigh numbers, Local values are evaluated by considering axial distance and average values by using square side length. Correlations were proposed in terms of local and modified Nusselt and modified Rayleigh numbers. Comparison of Experimental results with those reported in the literature pertaining to vertical cylinders is observed to be in close conformity.

Keywords: *Experimental Investigation, Natural Convection, Inclined surfaces, External Heat Transfer.*

1. Introduction

In the design of electronic components, solar energy collectors and heat exchangers it is important to consider the heat dissipation and its containment, such that the temperature does not exceed permissible limits that deteriorate the function of such systems. Therefore it is pertinent to understand the heat dissipation characteristics in relation to geometry and orientation of components as well as the influence of heat input on such characteristics. During the past many researchers have reported the natural convection phenomenon from vertical plates by (Churchill and Chu) , (T.Y.Na and J.P.Chiou) used an improved integral method to study effect of transverse curvature on natural convection phenomenon over a circular slender cylinder for a wide range of prandtl numbers under turbulent conditions. (T.Cebeci *et. al.*) studied free convective heat transfer from slender cylinders by solving boundary layer equations under constant heat flux conditions. (A.Kalendar and P.H.Oosthuizen) examined the effect of inclination on natural convection heat transfer from square cylinders under isothermal conditions. Correlations were developed between Nusselt and Rayleigh numbers for various dimensionless width and Rayleigh numbers. On the other hand (G.D.Rathiby *et. al*) developed correlations for natural convection heat transfer from spheroids at a wide range of eccentricity in air for all Rayleigh numbers under uniform temperature. The curvature effects and turbulent heat transfer phenomenon were detailed in this study. Natural convection heat transfer studies from the outer surface of the rectangular ducts by (M.E.Ali) are the latest that are reported. This study used the approximation methods suggested by (G.D.Rathiby and Hollands) for various cross sections. (F.Kimura *et. al.*) conducted experiments to understand fluid flow and natural convection heat transfer phenomenon around vertical cylinders by varying cylinder diameters. Flow visualization and liquid crystal thermometry were used to report turbulent transition effects. (O.Zeoitunan and M.E.Ali) simulated free convection heat transfer around rectangular ducts using commercial CFD software. (J.Li and J.D.Tarasuk) performed an interferometry study to analyze free convection around inclined cylinders. Numerical investigations were reported by (J.J.Wei *et. al*) on natural convection from uniformly heated inclined plates; both sides of the plate were considered for analysis. The prime aim of the present study is to develop correlations for square ducts of different area ratio subjected to uniform heat flux under laminar and transition regions. In addition the impact of orientation of the duct has also been the emphasis of the study.

2. Experiment Description

Investigations are made on two Aluminum ducts of different square cross section (with square side length

*Corresponding author: **T.V.S.M.R.Bhushan**

of 28 and 38 mm) of length 1 m. The inner volume of the duct has been heated by a centrally located electrical resistance heating element with Inconel sheading. The heating element extends from one end to the outer end along the length of duct. Heat losses from the end of ducts were reduced by end caps made of teak wood. J-type thermo couples with a time response of 0.1 sec were positioned at the center of the four faces at each location. The thermocouples were attached to the duct employing a polyamide adhesive tape. In addition four thermocouples were positioned at the ends (two in each end) to measure the temperature both within and outside the end caps. The ambient temperature was measured by using mercury thermometer located away from the experimental setup.

The power input was varied using a dimmer stat. The voltage and current were measured using a voltmeter and ammeter from which the power input was calculated. The experimental setup showing the location of thermocouples and instruments used for voltage and current measurement as well as for varying voltage is shown in fig-1. The orientation of duct was varied by a Tailor made arrangement. To connect the duct to the Taylor made arrangement. In order to avoid conduction loss through the bolts, wooden beads were used in between the bolts and duct surface. Temperature measurements at 44 locations were carried out by multi-channel digital temperature indicator.

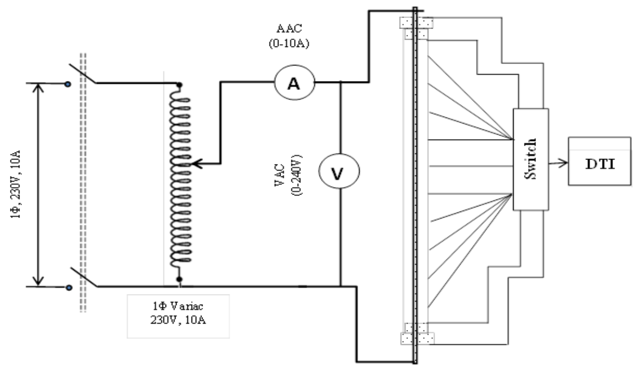

Fig.1 Description of Experiment setup with duct oriented vertically

The temperature measurements were made under steady state conditions and the input voltage was varied in steps of 10 Volts up to a maximum voltage of 60 V. For the above input voltages the power was in the range of 9 W to 130W. The experiments were conducted on each duct for seven power inputs at an angle of 45°. Thus the total numbers of experiments were 14. These number of experiments were to be conducted to identify demarcation between laminar and transition region based on Rayleigh number. In order to verify the correctness of the data experiments were repeated at randomly selected power inputs. To validate the data, experiments were also conducted in vertical orientation. All the experiments were conducted in still air (Pr $\simeq$ 0.7) from the application point of view. In all the experiments the data collection was in steady state conditions.

3. Experiment Procedure and Analysis of Data:

3.1 Experiment Procedure

Total Heat supplied to the duct is from the centrally located resistance heating element under constant heat flux conditions. The heat generated inside dissipates from the surface of the duct in the form of convection, radiation and conduction via end caps.
Total heat supplied

$$Q = V\,I = Q_c + Q_{tk} + Q_r \tag{1}$$

Heat transfer via radiation and conduction from the end caps are calculated by

$$Q_r = \varepsilon \sigma A_s (\bar{T}^4 - T_\infty^4) \tag{2}$$

$$Q_{tk} = K_{tk} \frac{(T_{in} - T_{out})}{l} \tag{3}$$

$\bar{T}$ is the overall circumferential averaged temperature defined from equation (6). Surface emissivity of the duct material was evaluated as 0.17. It is observed from the experimental data that a maximum 25% of the total heat is dissipated in the form of radiation. T_{in} and T_{out} are the temperatures of Teak wood (end caps) inside and outside respectively. Thermal conductivity of the teak wood is considered to be 0.015 W/m-K, which is the standard reference value. Heat transfer via conduction from the end of the duct is calculated using equation (3). Conduction heat losses are around 5-7%.

$$T_{bm} = \frac{(\bar{T} + T_\infty)}{2} \tag{4}$$

The thermo physical properties of air are evaluated at bulk mean temperature which is calculated from equation (4)

$$T_x = \frac{\sum_{i=1}^{4} T_{xi}}{4} \tag{5}$$

'T_x' is the circumferential average of the temperature at any point x varying from 0.1 to 1 m from the leading edge to the trailing edge of the duct. 'i' is the temperature at any ' x ' on all four sides (front, top, back and bottom) of the duct.

Overall averaged temperature on the surface of the duct is evaluated by averaging the circumferential averaged temperatures from the following equation

$$\bar{T} = \frac{\sum_{x=1}^{10} [T_x]}{10} \tag{6}$$

3.2 Analysis of Data

3.2.1 Local circumferential averaged values

After obtaining local circumferential averaged temperature and the convective heat flux the local convective heat transfer coefficient for each power input was calculated from equation (7) and the non-dimensional numbers such as modified Rayleigh number and Nusselt numbers were evaluated form equations (8) and (9).

$$h_x = \frac{q_c}{(T_x - T_\infty)} \quad , \quad x = 1,2,3,\dots,10 \tag{7}$$

$$Ra_x^* = \frac{(g cos\theta)\, \beta\, q_c x^4}{\vartheta\, k\, \alpha} \tag{8}$$

$$Nu_x = \frac{h_x\, x}{k} \tag{9}$$

3.2.2 Overall averaged values

The overall averaged heat transfer coefficient is obtained by averaging the local convective heat transfer coefficients from the starting to the end of the duct using equation (10). Similarly average Nusselt number and modified Rayleigh numbers are evaluated by equations (11) and (12) at a characteristic length L_c= (D).

$$\bar{h} = \frac{\sum_{x=1}^{10}[h_x]}{10} \tag{10}$$

$$\overline{Ra_{L_c}^*} = \frac{(g cos\theta)\ \beta\, q_c\, {L_c}^4}{\vartheta\, k\, \alpha} \tag{11}$$

$$\overline{Nu_{L_C}} = \frac{\bar{h}\, L_C}{k} \tag{12}$$

4. Result Analysis

Experiments have been conducted on two square ducts oriented at 45^o to the horizontal for different heat inputs. The variation of temperature at various locations for different power inputs is presented in fig. 2(a). While fig. 2(b) represents local convective heat transfer coefficient at different locations and heat inputs. The temperature and distance are non-dimensional. The temperatures obtained are the maximum at the center of the duct while they are minimum at the two entrance ends. The difference between the temperature at the center and at the end increases with an increased heat flux.

From fig. 2(b) it is observed the convective heat transfer coefficients are higher at the ends and lower in the mid portion of the duct. The convective heat transfer coefficient at each location exhibit higher values with an increase in heat flux.

The experimental data have been analyzed by obtaining Nusselt number as well as modified Rayleigh number for different experimental conditions. The Nusselt and Rayleigh number are calculated at each point based on circumferential average temperature (Proposed by Rathiby and Hollands) as well as average Nusselt Number and Rayleigh Number based on Overall averaged temperature on the duct and characteristic length (Lc = D). Gravitational component (g) is replaced by the corresponding tangential component "$gcos\theta$" to understand the natural convection phenomenon for inclined ducts.

From Fig.3 it is observed that, increase in Rayleigh number results in increase in Nusselt number. The relation is linear in the laminar region (Ra*<10^9), while the same is nonlinear in transition region (10^9< Ra^*_x <10^{12}). Regression analysis is performed to correlate Nusselt and Rayleigh numbers, least square power law (y= a (x) ^{n}) fit trend line with R^2=97.41 % is obtained with regression constants a=0.6294 and n=0.2177. The proposed correlation between the local values of Nusselt number and the modified Rayleigh number is as per the following equation.

$$Nu_x = 0.6294\, (Ra_x^*)^{0.2177} \tag{13}$$

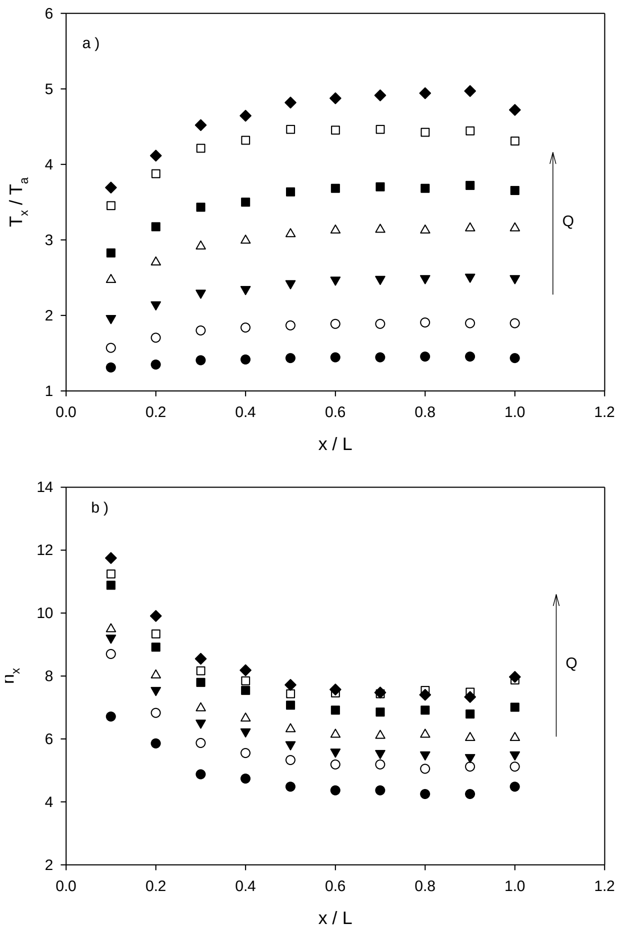

Fig.2. a) Variation of Circumferential averaged and Non-Dimensional temperature with normalized distance from the leading edge to the end of the duct. b) Variation of Local convective heat transfer with normalized distance from the leading edge of the duct to the end of the duct

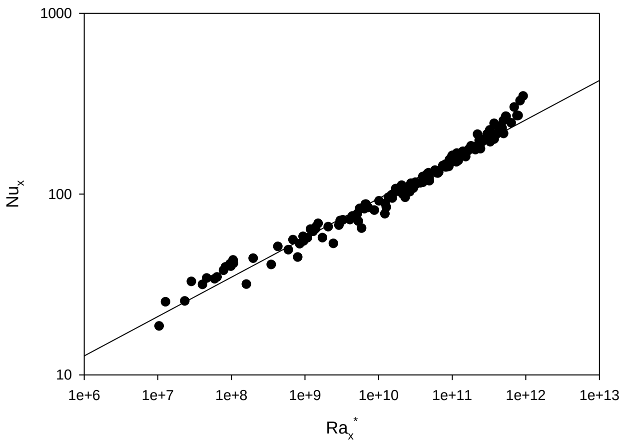

Fig.3 Local Nusselt number variation with modified Rayleigh number for all ranges of heat loads of both the ducts

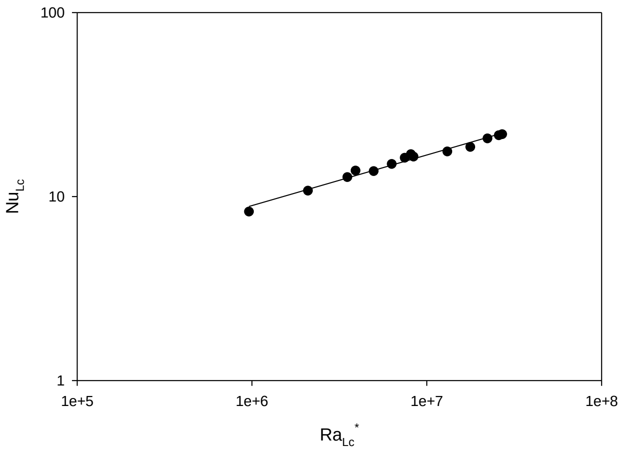

Fig.4 Variation of Average Nusselt number with average modified Rayleigh number for both the ducts

From the application point of view it is important to understand the relation between average non dimensional numbers, in view of this fig. 4 is constructed between average Nusselt and modified Rayleigh numbers irrespective of the area ratio. Higher values of Rayleigh numbers are observed at the low heat fluxes for duct 2 compared to duct 1 since the area ratio of the duct 2 is higher. It is observed that increase in Rayleigh number results in an increase in Nusselt number. Regression analysis is performed and a power law fit with R^2=97.07 % is obtained with regression coefficients (a=0.1567, n= 0.2893) and the observed correlation is in the following form.

$$Nu_{L_c} = 0.1567\ (Ra^*_{L_c})^{0.2893} \qquad (14)$$

The present experimental data at 45^0 have been compared with vertical circular cylinders with (Vliet and Liu) at $10^{12}<Rax^*<10^{14}$ and (Fujii) $10^6 < Rax^* <10^{14}$, shows relation between local values of Rayleigh number and Nusselt number, for duct oriented at 45^0 angle, It may be noted from the figure that the relation between the Nusselt and Rayleigh number for Vertical cylinders and for inclined square duct from the present study follow similar trends in that as Rayleigh number increases Nusselt number also increases. However the values of Nusselt number are on the lower side for the vertical cylinders.

Conclusions

Natural Convection heat transfer phenomenon around square ducts inclined at an angle of 45^o is experimentally investigated. The study parameters include the duct area ratio, orientation and flow regime. All the experiments are conducted in still air. Two ducts were used to vary area ratio and, experiments were conducted at various heat inputs to analyze flow regime. From the experimental investigation the following are the salient observations.

- Convective heat transfer coefficients are on the lower side in the mid portion of the duct while they are on the higher side at the entrance ends that is heat transfer coefficient decreases from the end to the center of the duct.
- Modified Rayleigh number is calculated by considering the tangential component of the gravitational component and the correlations were proposed between local Nusselt number and modified Rayleigh number with a power law fit.
- Correlations are proposed in terms of average Nusselt number and modified average Rayleigh number.

Present study indicates that the inclined square ducts exhibit higher values of Nusselt numbers as compared to that for vertical cylinders. However present trends follow the established correlations pertaining to vertical cylinders.

List of Symbols

As = Surface area of Duct, m^2
D = Side Length of Square, m
I = Current, A
$\bar{h}$ = Average heat transfer coefficient , W/m^2K
hx = Local heat transfer coefficient, W/m^2K
k = Thermal conductivity of air, W/m-K
ktk= Thermal conductivity of Teak wood,
L = Length of duct, m
Lc = Characteristic length, D, m
Nu = Nusselt Number, (hx/k)
qc = Convective heat flux, h_x (T_x-T_∞) , W/m2
Ra* = Modified Rayleigh number,
T = Temperature, ^oC
$\bar{T}$ = Overall averaged Temperature, ^oC
V = Voltage, V

Subscripts

In = End cap inside temperature, ^oC
Out = End cap outside temperature, ^oC
bm= Bulk mean temperature, ^oC
∞ = Ambient temperature, ^oC

Greek Symbols

α = Thermal diffusivity, m^2s^{-1}
β = Coefficient of Thermal Expansion, K^{-1}
ϑ = Kinematic Viscosity, m^2s^{-1}
θ = Angle of orientation, deg
σ = Stephen Boltzmann Constant,
ε = Emissivity of the duct material

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