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EXPERIMENTAL INVESTIGATION OF LAMINAR MIXED CONVECTION HEAT TRANSFER IN THE ENTRANCE REGION OF RECTANGULAR DUCT

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ABSTRACT

The present experimental study consider the effect of laminar mixed convection heat transfer in the entrance region of horizontal rectangular duct subjected to uniform heat flux on three sides with adiabatic top. Experimental facility includes ducts of aspect ratio of 2.5 and air as working fluid. Different wall temperatures and flow rates were measured. The range of parameters used were Reynolds number between 1000 to 2300, Grashoff number 10^5 - 10^7 and Prandtl number of 0.7. Results were obtained in terms of wall and fluid temperatures distribution, Richardson number and Nusselt number. Richardson number affects the wall temperature distribution strongly during the flow in the entrance region i.e., wall temperatures steadily increases in the flow direction and after some length, it shows decrease in it. It is also observed that at entrance region. Nusselt number increases with increase in Richardson number.

Keywords: Duct, Grashoff number, Heat flux, Mixed convection, Richardson number.

I. INTRODUCTION

Laminar mixed convection flows has received great interest in last decades in variety of engineering applications, such as solar collectors, compact heat exchangers, geothermal energy system, electronic devices and cooling of nuclear reactors.

A number of numerical as well as experimental investigations have been conducted to study single phase pure natural convection and pure forced convection in different geometries. However, in several cases density variations is always present in pure forced convection problems. Therefore free as well as pure forced convection plays vital role in fluid flow and heat transfer applications.

It is observed that buoyancy effects can significantly enhance the convective heat transfer coefficient over the pure forced convection values [1, 2]. Analytical solution was obtained for different boundary conditions with the application of different solution methods [3-5]. Buoyancy effect are negligible for $Ra \leq 10^3$ [4]. Mahancy et al. [6,7] observed the rise and fall of Nusselt number values depending on free or forced convection effects are dominate in the flow for fully developed flow.

However the assumption of fully developed flow can only be established if the channel is very long. Earlier studies for large Prandtl number fluids with large aspect ratios shows significant effect of buoyancy under thermal boundary condition of uniform heat flux in horizontal and inclined rectangular duct [8-11]. It is also observed in earlier studies that very very low to high Richardson number is used.

The present investigation deals with the experimental study of the problem of laminar mixed convection in the entrance region of horizontal rectangular duct. The experiments reported in the paper were performed with air as working fluid under UHF thermal boundary conditions.

II. EXPERIMENTAL APPARATUS AND PROCEDURE

Experimental set up consists of nozzle with filter, developing section, test section and lastly developed section with arrangement for supply of air.

The flow straightner and nozzle avoids flow separation, turbulence and provide uniform velocity at the entry to test section. This component is followed by developing section which ensures steady and laminar flow. This duct is insulated with Bakelite for certain length. Test section is mild steel rectangular duct of size 40mm×100mm with 500mm length having aspect ratio of 2.5. Total length of set up from bell mouth to test section outlet is such that flow is hydrodynamically and thermally developing. Test section is insulated with mica sheet and Bakelite to avoid heat loss. Test section is subjected to UHF at three faces with top wall adiabatic. Thermocouples are provided on three faces at equal distance all along the test section length. Thermocouples are also arranged on the end of test section to measure end conduction loss. Orifice plate is provided in set up to measure mass flow set of air. Two thermocouples are placed to measure inlet, outlet & third near orifice plate to correct mass flows rate. Micro manometer was placed across the orifice plate to measure pressure difference. The free end of downstream pipe was connected with blower through flexible hose pipe. This has helped to separate test section acoustically and mechanically from vibrations.

First preliminary tests were carried out to check the proper functioning of different components of experimental set up and leakages, if any. Few readings of experiments were repeated to check the repeatedly.

During experimentation, discharge and heat input was varied to desired value and steady state values of various thermocouples were noted down.

III. DATA REDUCTION

The heat input to test section was based on the actual fluid enthalpy rise across the test section. The local fluid bulk temperatures at each axial location were computed from inlet temperature, mass flow rate, and power input. The local heat transfer coefficient was

calculated using the actual heat received by the fluid, local fluid temperature and local average axial wall temperature.

The surface energy balance relates the total energy dissipated per unit area at any point on the surface to heat flux associated with convection, radiation and end conduction. It is given by:

$$\ddot{q} = (Q_{\text{supp}} - Q_{\text{losses}}) / A = Q_{\text{useful}} / A$$

$$Q_{\text{supp}} = V * I$$

$$Q_{\text{losses}} = Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{end}}$$

Fluid axial conduction is neglected as $RePr > 60$ [12]

Heat losses includes convection from sides [13], radiation loss and end conduction losses assuming one dimensional heat transfer.

Bulk mean temperature of air in the test section along the flow direction is determined by:

$$T_{\text{bx}} = T_{\text{fi}} + (Q_{\text{useful}} * X / m * C_p)$$

IV. RESULTS AND DISCUSSION

The experimental results were analyzed for mixed convection in the entrance region of rectangular duct for the aspect ratio $AR=2.5$. The experimental runs were performed in such way that Richardson number value was kept between 0.5 to 5. Results were studied in terms of effect of variation of Richardson number on wall temperature, fluid temperature distribution and Nusselt number.

4.1 Wall temperatures

In mixed convection heat transfer Richardson number plays vital role as it links forced and free convection effects. Experimental runs having Ri of 0.5 and 1.325 are shown in Fig.1. Wall temperatures are lower for lower value of Ri than for its higher value. Fig.1 reveals that higher temperatures are present on bottom wall than sidewalls. This indicates that heat is removed efficiently by sidewalls. Wall temperatures are increases steadily as forced convection is dominant and after some length there is marked decline in temperatures. This indicates the development of buoyancy force which produces secondary motion which helps in removal of heat. Similar trend is found for higher values of Ri as shown in Fig.2.

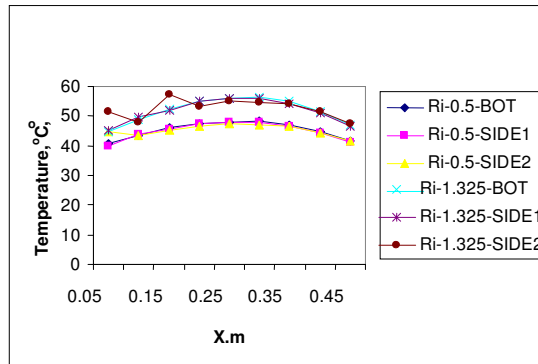


Figure 1 Influence of Ri on wall temperatures distribution

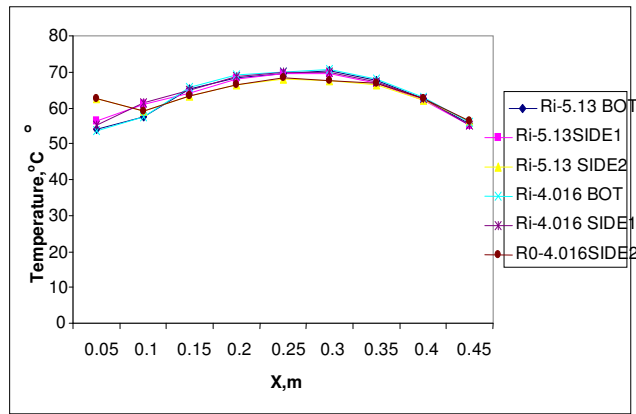


Figure 2 Influence of Ri on wall temperatures distribution

4.2 Wall and Fluid temperatures

Fig.3 represents influence of increase in Ri on average wall and fluid temperatures at given Reynolds number. Higher wall temperatures exist for high Ri, but it is also accompanied with large increase in fluid temperatures which reveals that there is positive replacement of hot fluid by cold fluid. It can also be observed that decline of temperatures begins earlier for higher value of Ri. This indicates that beginning of secondary flow shifts upstream with the increase in the Ri.

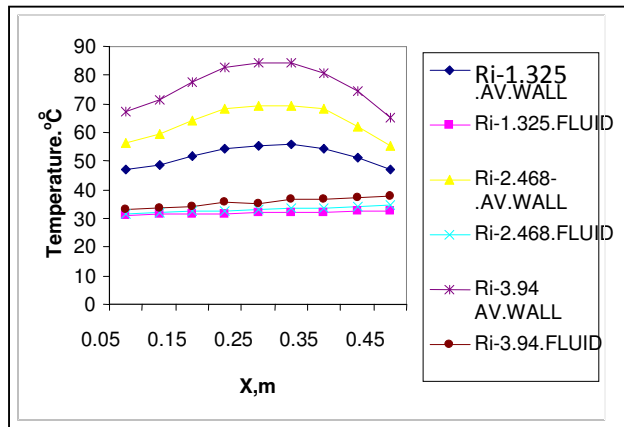


Figure 3 Influence of Ri on average wall temperatures and fluid temperatures distribution

4.3 Nusselt number

Fig.4 indicates the effect of Reynolds number on Nusselt number. It can be observed that for Re of 1035 initially there is decrease in Nu because of strong forced convection effect but as the flow proceeds there is development of secondary flow which disturbs the thermal boundary layer and increases the Nu. But as the Re is increased from 1035 to 2315 there is small rise in the Nu which is about 11% i.e its effect is negligible in mixed convection.

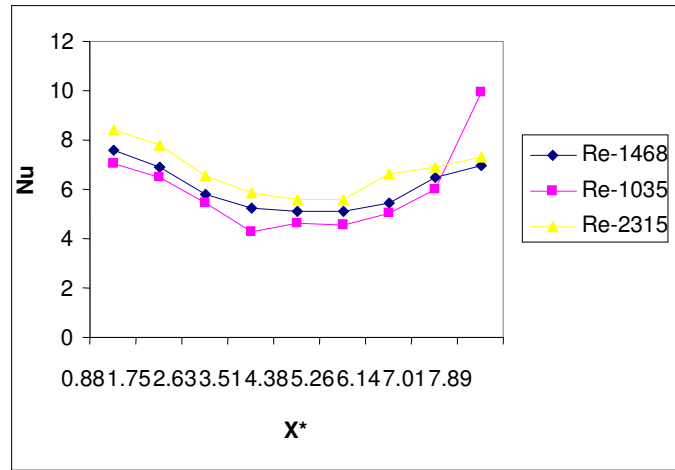


Figure 4 Influence of Re on Nu for same Gr

Fig.5 represents the effect of Ri on Nusselt number. It can be observed that Nu is decreasing along the flow direction which may be due to forced convection effect and then there is rise in Nu which confirms the existence of strong secondary flow motion. Nusselt number increases with increase in Ri. It is about 200% rise for Ri increase from 0.398 to 5.13.

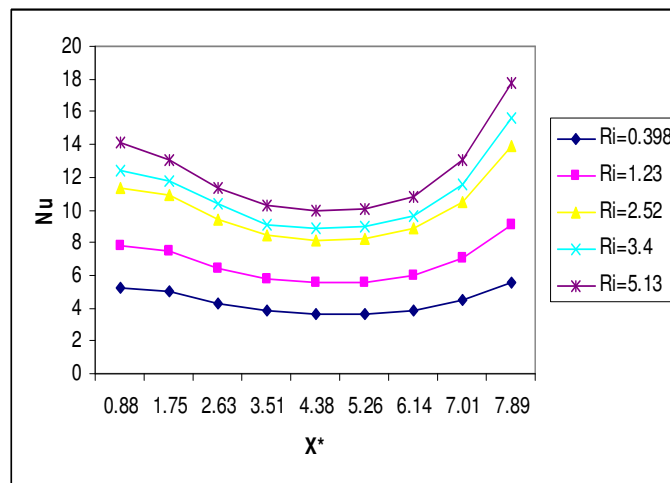


Figure 5 Influence of Ri on Nu for same Re

V. CONCLUSIONS

The wall temperatures for bottom surface are more than sidewalls therefore sidewalls contribute more towards removal of heat. Also wall temperatures are higher for higher Richardson number. The wall temperatures increases to certain length than there is drop in values which is due to increase in buoyancy effect or free convection in flow. It is also observed that rise in Ri tends to shift minimum temperature in upstream side of flow.

The effect of Reynolds number on Nusselt number in mixed convection heat transfer is negligible. The rise of 11% in Nu is observed when rise in Re is from 1035 to 2315.

Initially in the mixed convection heat transfer flow forced convection effects are dominant which is seen by increasing wall temperatures or decrease in Nu but after some length there is drastically decrease in temperature or rise in Nu which is due to onset of buoyancy driven secondary flow or dominance of free convection over forced convection. The rise in value of Nu for increased value of Richardson number is large. It is calculated to be about 200% for the change in Ri from 0.398 to 5.13.

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NOMENCLATURE

AR	: Aspect ratio
A	: Area, m ²
BOT	: Bottom wall
C _p	: Specific heat, kJ/kg-K
D _h	: Equivalent diameter or Hydraulic diameter, 4*A / P, m
Gr	: Modified Grashof number, $g\beta q D_h^4 / \nu^2$
g	: Acceleration due to gravity, m/s ²
h	: Heat transfer coefficient, W/ m ² –K
k	: Thermal conductivity, W/m-K
Nu	: Nusselt number, h D _h /k
P	: Perimeter, m
Pr	: Prandtl number, $\mu C_p/k$
Q _s	: Supplied heat input, W
Q _{loss}	: Total heat loss, W
Q _{rad}	: Heat loss due to radiation, W
Q _{end}	: Heat loss due to end conduction, W
Q _{conv}	: Heat loss due to convection, W
Ri	: Richardson number, Gr/Re ²
Re	: Reynolds number, $u D_h / \nu$
Side1	: Left sidewall
Side2	: Right sidewall
UHF , \dot{q}	: Uniform heat flux , W/ m ²
UWT	: Uniform wall temperature
X*	: Axial Dimensionless length, X/ D _h

Greek symbols

β	: Coefficient of thermal expansion, K ⁻¹
μ	: Dynamic viscosity, kg/m-s
ν	: Kinematic viscosity, m ² /s

Subscripts/Superscripts

f	: Fluid
w	: Wall condition
x	: coordinate system