COMBINED CONVECTION HEAT TRANSFER THROUGH RECTANGULAR DUCT

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Abstract

A convective mode of heat transfer plays an important role in many engineering applications. Combined convection heat transfer involves with free convection heat transfer effects on forced convection. Sidewalls and bottom surface of the duct were subjected to uniform heat flux while top surface was adiabatic.

Test section was incorporated with 27 copper constantan thermocouple to measure wall temperatures & 12 to determine heat loss. The range of parameters used were – $Gr=10^5$ to 10^7 , Re = 1000-2300 and Pr=0.7.Air was used as working fluid .

From experimental outcome shows that value of Grashof number and Reynolds numbers for air were are considerably greater than the pure forced convection values in the downstream side of flow. It was observed experimentally wall temperature distribution is varying in sidewalls and bottom side. Temperatures are higher on bottomsie than sidewall.

Key words– Combined convection , buoyancy effect , duct,wall.

1. Introduction:

Lot of numerical and experimental findings were carried out to study free convection and forced convection. However, in many cases, density variation may be small, is always present in forced convection. Therefore, both types of convection plays vital role in affecting heat transfer and fluid flow. Such type of convection is called as mixed convection or combined convection. Combined convection is always present for situations where the flow velocity is small and the temperature gredient is large.

Combined convection process may be classified as external flows such as flow over flat plates , cylinders etc and internal flows through pipes, channels etc . Internal flows may through variety of regular or arbitrary shaped geometries such as cylindrical, rectangular, triangular and polygonal. The rectangular duct is commonly used in heat transfer and fluid flow devices like solar collectors , concentrators , compact heat exchangers, modern electronic equipment etc.

2.Literature Survey:

Cheng et al [1] have investigated developing combined convection flow in vertical duct. One vertical wall was subjected to higher temperature and remaining three were at ambient temperature. They have varied buoyancy force term i.e Grashof number to Reynolds number, Gr/Re from 0 to 640 and studied for aspect ratios of 0.5,1 and 2.It was observed that Nusselt number were higher for increase in Gr/Re and aspect ratio .Pressure drop is reduced by the buoyancy force due to the buoyancy assisted flow.

Ichimiya et al [2] has studied performance assessment of combined convection in an inclined square channel which is subjected to uniform wall temperature. They have studied fluid flow by streak lines and found that flow was composed of twisted flow by rising flow along sidewalls and falling flow along centre of flow passage. They have also observed that two recirculating flows were formed whose Centre of recirculation moves upward for positive inclination and downward for negative inclination.

Mixed convection heat transfer in horizontal rectangular channel applied with uniform heat flux at one side &other sides as adiabatic was studied by Gau et at [3] . They have studied heat transfer by visualization technique. Buoyancy parameter Gr/Re^2 was varied from 0 to 20,000. It is observed that due to vertical buoyancy force , the heated buoyant flow moves upward and got accumulated in the upper region of channel. This flow was thermally stable and grew in size with rise of Gr/Re^2. They have observed the rise of Nusselt number when Gr was increased & Re was reduced. They have obtained the correlations for normalized Nusselt number as-

$$Nu=0.94[1+0.0081(Gr/Re^2)]^{0.148}(x/Dh)^{-0.272}$$

Combined convection heat transfer in a top and bottom heated rectangular channel with discrete heat source were studied for air by Dogan et al [4]. Discrete heat sources were subjected to uniform heat flux. Data acquision system was used to collect and process the stored data. It was seen that as the Gr was increased ,Nusselt number first decreased and then increased.

DaoTong et al [5] has studied the effect of duct inclination on the Nusselt number and friction factor in the rectangular duct having composite heater placed in the middle of duct for laminar and transition flow. It was reported that as the Reynolds number was increased Nusselt number first increased with inclination angle to maximum and then decreased. For Reynolds number increased beyond 1800 , Nu was found to be independent of inclination angle.Friction factor was found to be decreasing with inclination angle as the Reynolds number was increased and for Reynolds number above 1300, friction factor was also independent of inclination angle .

3. Experimental set up:

Experimental set up was constructed on the basis of feasibility and simplicity to study mixed convection heat transfer through duct. It is shown in in Fig.1. Experimental set up comprises of following main Components

(i) Bell mouth opening or flow straighter
(ii) Developing duct.
(iii) Test section
(iv) Outlet or Developed section.
(v) Orifice plate with flow control valve.
(vi) Flexible hose pipe and blower.

3.1 Bell mouth opening or flow straightener:

The flow straightener was made of 5mm diameter and 70mm long plastic hoses, 429 in total. At the entry to bell mouth plastic filter used for filtering the air. Flow straighter was followed by nozzle with contraction ratio of 40:1 and length 200 mm properly designed to avoid flow separation , turbulence and provide uniform velocity at the duct entrance .

3.2 Developing duct:

The developing duct having cross section of 40mm by 100mm and length of 500mm .It was made from 2mm mild steel plate. This duct ensured the steady and laminar flow of air at the inlet to test section . Duct was insulated with Bakelite of 12mm thickness for its certain length.

3.3 Test section:

The test section was mild steel rectangular duct having inner dimension of 40mm x 100mm with a wall thickness of 2mm and a total length of 500 mm. Since total length from bell mouth to test section outlet is 1.27m.Therefore flow is hydrodynamic ally and thermally developing.

The aspect ratio, AR of duct was 2.5 . Duct was provided with end flange to connect it to other parts. Uniform electrical heating was provided on three sides by high thermal conductivity nicrohme wire. This was electrically insulated by thin mica sheet. The test section was provided with 27 copper constantans (T type) thermocouples on two sidewalls and bottom of duct at equal nine axial position . The entire test section was first insulated with glass wool and then insulated with Bakelite . Three thermocouples at equal distance were fixed on each outer surface of duct to measure losses and two thermocouples on ends to measure end conduction loss. Two thermocouples measure inlet and outlet and third one near orifice plate to correct mass flow rate.

3.4 Outlet section:

Outlet section was of rectangular inlet and circular outlet of 40mm in diameter and was having length of 200mm . Purpose is to reduce disturbance effect from the ambient surrounding on the flow through test section.

3.5 Orifice plate with flow control valve:

Orifice plate with size of 40mmx20mm used to measure discharge of air. Discharge was found at the temperature of air at orifice plate. Simple flow control valve was used to regulate the flow rate of air.

Fig.1.Experimenta Set Up

4.Experimental Results:

Experimentation was carried for following non dimensional numbers Range. The results are expressed in terms of variation in wall temperatures for different Grashof Number(Gr) and Reynolds Number(Re) . The flow in rectangular duct is controlled and is steady laminar flow , therefore Reynolds number is kept well below critical Reynolds number. Uniformity of the flow is well checked by using flow straightener .Richardson Number(Ri) is kept in a range of 0.3 to 5.0.

The range of parameters covered in experimentation are given below—

Gr = 10^5 to 10^7

Re = 1000 to 2300

Ri(Gr / Re^2) = 0.3 to 5.0

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4.1 Effect of Re on Wall temperatures distribution:

Fig.2. to Fig.4. represents the influence of Re on bottom and sidewall temperatures. In these figures, temperatures in ^oC are presented on Y-axis and linear scale is chosen for dimensional axial length which is scaled along X-axis .Origin of x-axis is inlet of test section.

In Fig.2. for low Re = 1014 and Gr = 3.1E+6 temperatures are ranging between 40 ^oC to 50 ^oC . Initially temperatures are lower due to end effect and then steadily increases in the main flow direction. This increase takes place to axial length of X =0.25m approximately and then it is observed that there is decline in temperatures. This is due to buoyancy effect which is now stronger than forced convection effect. Different walls i.e. bottom and sidewalls are nearly indicating same temperatures.

In Fig.3., Re = 1217and Gr = 7.6E+6 ,temperatures are varying from 52 ^oC to 75 ^oC. Sidewalls temperatures are slightly higher than bottom which indicates faster end effect initially at bottom surface. After certain axial length, temperatures of bottom plates are slightly higher than sidewalls which agrees with earlier authors that heat is effectively removed from sidewalls.

In Fig.4. Re is increased to 2026 it can observed that sidewalls temperatures are lower than bottom. But here Gr is also increased therefore by raising Re earlier forced convection effect is stronger but in comparison with above figures buoyancy effect in that part is also considerable therefore both effects reduces sidewalls temperatures. Bottom surface temperatures are higher than sidewalls, can be observed distinctly.

4.2 Effect of Gr on wall temperatures distribution:

As the Gr is increased, heat flux is also increased which raises the surface temperatures. Gr is increased from 4.41×10^6 to 1.164×10^7 for constant mass flow rate.

In Fig.5. Grashof number is 4.41×10^6 . Surface temperatures are 45 ^oC to 55 ^oC. For small early length sidewalls temperatures are higher than bottom surface. As the flow proceeds temperatures are increased to some length then decrease in temperature occurs. This fall in temperature confirms the presence of secondary flow due to strong buoyancy force.

Fig.6. represents wall temperatures distribution for increased Grashof number $7.482\text{x}10^6$ .Wall temperatures are higher due to increased heat flux. After some length all three surfaces are at nearly same temperatures. Similar trend of wall temperatures can be seen in Fig.7.

4.3 Effect of Ri on wall temperatures distribution:

In mixed convection heat transfer Richardson number plays vital role as it links forced and free convection effects. Ri can be varied by changing value of Gr or Re.

Run-16 and Run-55 are having Ri 0.5 and 1.325 respectively as shown in Fig.8. Wall temperatures are lower for Ri =0.5 than Ri =1.325 .As expected wall temperatures for bottom surface is slightly more than sidewalls. Temperatures are 40 ^oC to 48 ^oC for 0.5 and 45 ^oC to 58 ^oC for 1.325 which is due to high heat flux for larger Ri. It is observed that onset of instability for higher Ri causes more decrease in wall temperature than small Ri.

Conclusions:

- Increase in Reynolds number results in heat dissipation at side wall more than bottom wall .
- Increase in Grashof number causes rise in temperatures but buoyancy effect dominates forced convection hence rapid heat dissipation.
- Increase in Richardson number results in onset of instability to move upstream side of flow.

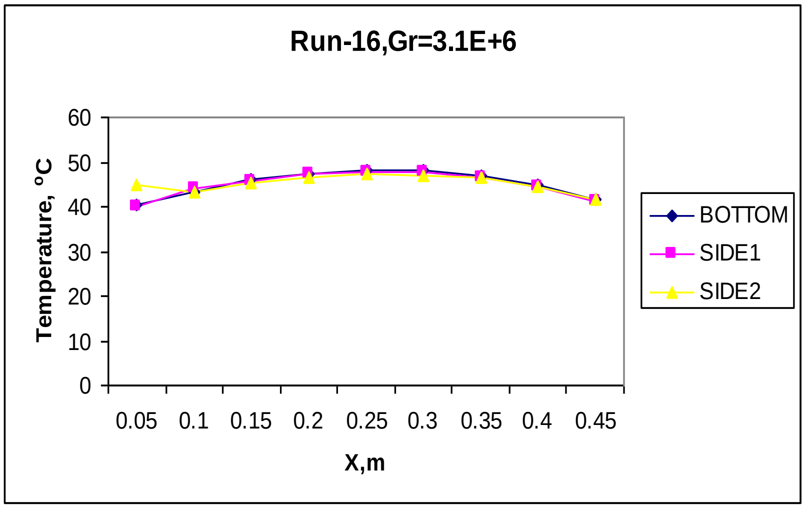

Fig.2. Wall temperatures distribution for Re=1014

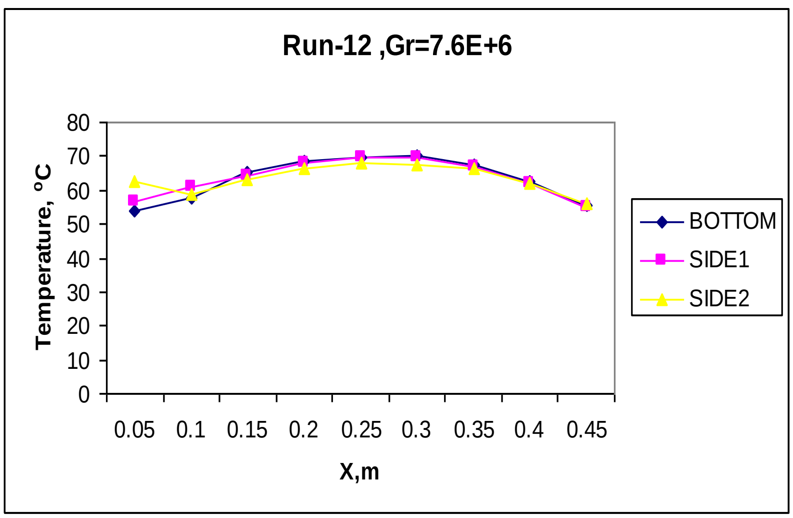

Fig.3. Wall temperatures distribution for Re=1217

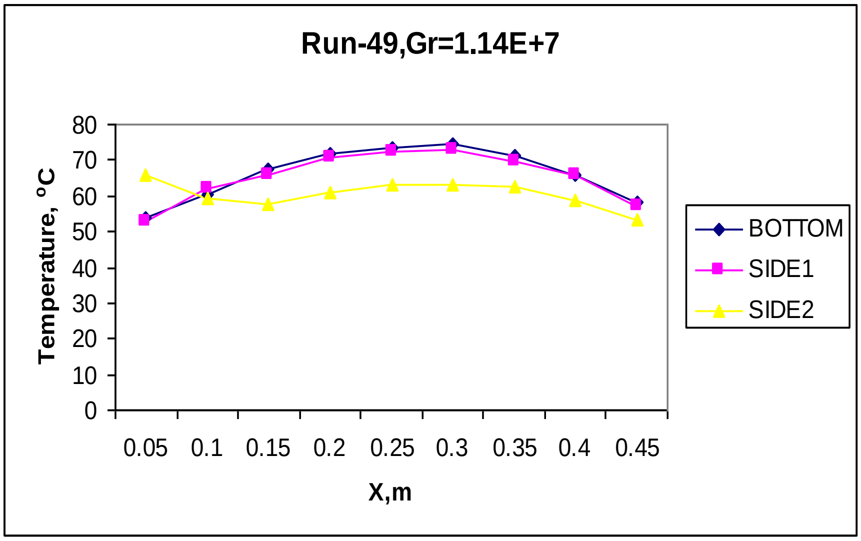

Fig.4. Wall temperatures distribution for ,Re=2026

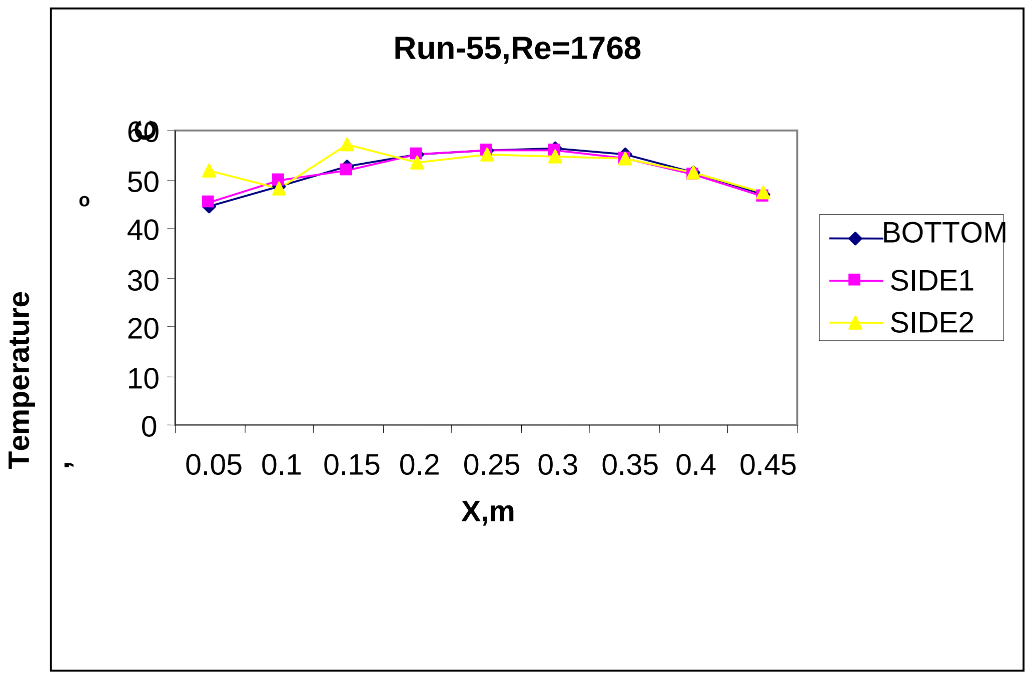

Fig.5. Wall temperatures distribution for Gr=4.41E+6

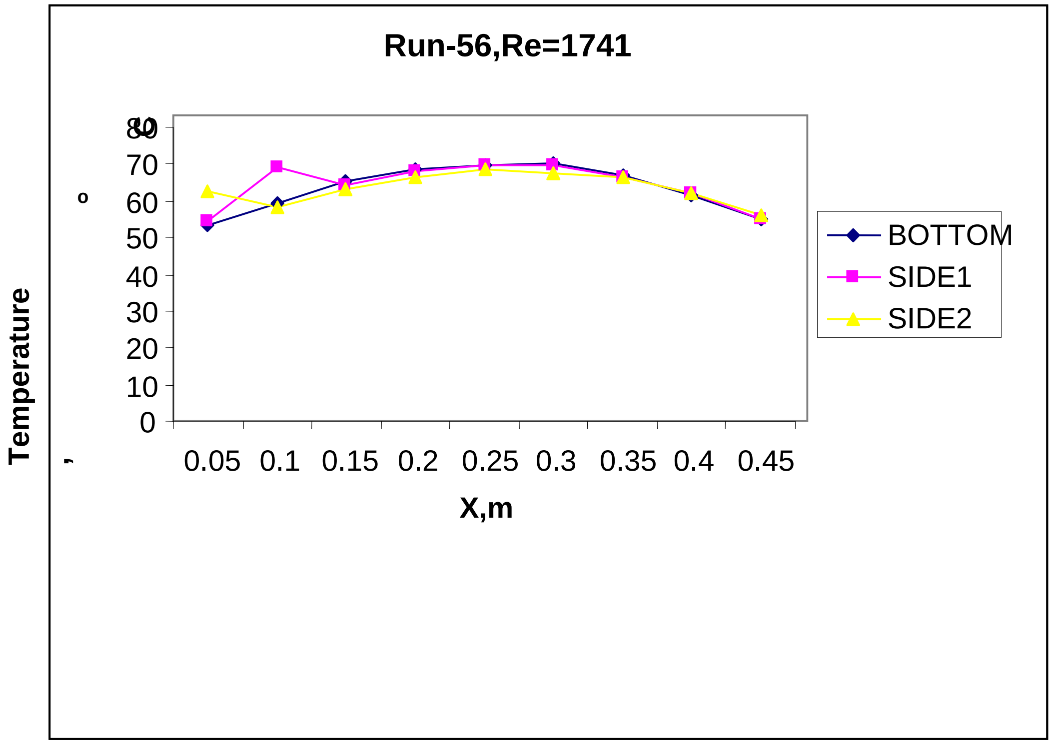

Fig.6. Wall temperatures distribution for Gr=7.482E+6

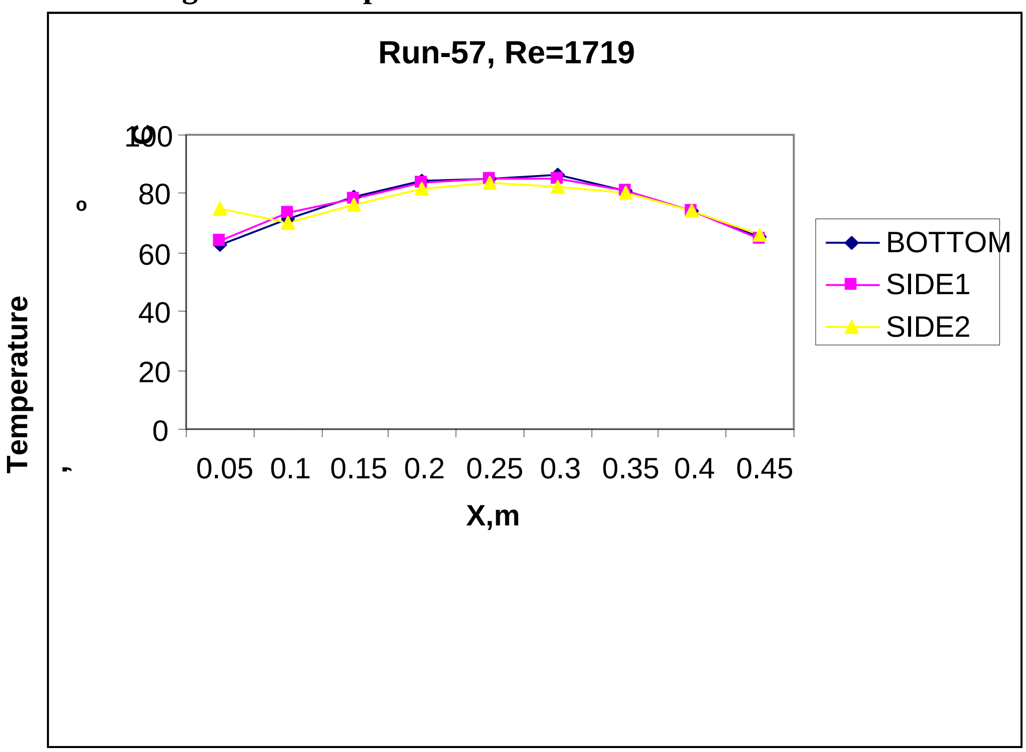

Fig.7. Wall temperatures distribution for Gr=11.64E+6

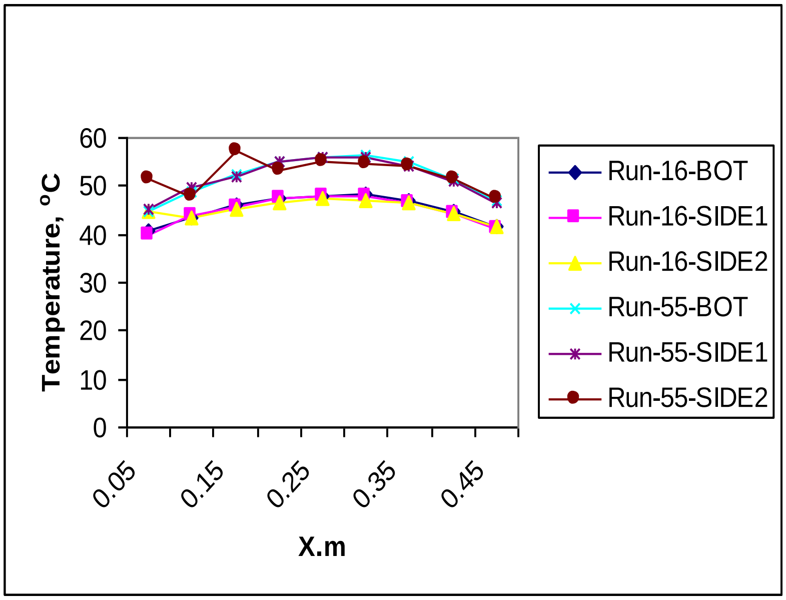

Fig.8. Influence of Ri on wall temperatures distribution

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