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HEAT TRANSFER IN A RECTANGULAR ENCLOSURE WITH BAFFLES

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ABSTRACT

A numerical study was carried out to investigate the mixed convective two dimensional flows in a vertical enclosure with heated baffles on side walls. All walls are assumed to be adiabatic, but baffles are considered as isothermally heated. Thus, cold flow is imposed through an opening at the bottom of the left wall and by taking heat from the baffles the fluid becomes heated and exits through outlet at the top of the right wall of the enclosure. Heated baffles are placed both at the left and right wall of the enclosure. The present study simulates a practical system such as a silencer. The consequent mathematical model is governed by the coupled equations of mass, momentum and energy and these equations are discritized. The discritized equations with specified boundary conditions are sought by Successive under Relaxation (SUR) method. A wide range of pertinent parameters such as Reynolds number $50 \leq Re \leq 300$, Richardson number $0 \leq Ri \leq 10$ and Prandtl number $0.01 \leq Pr \leq 2.0$ are considered in the present study. Various results such as the streamlines, isotherms, heat transfer rates in terms of the average Nusselt number and temperature and also heating efficiency in the enclosure are presented for different parameters. It is observed that Maximum heating efficiency is found at higher value of Reynolds and Richardson number.

Keywords: model, heat transfer, mixed convection, Richardson number, baffle, vertical enclosure, heating efficiency.

1. INTRODUCTION

Mixed convective flow in ventilated enclosure with baffle has been studied numerically due to its wide technological application. For its unique advantages like changing the fluid flow direction, increasing the retaining time, providing additional surface area for heat transfer, baffle's are used widely observed in fuel tanks, crankcase, mufflers, radiators and intake system of vehicles; some electronic circuit boards, internally cooled turbine blades, in silencer pipe etc. In ventilated enclosures, the interaction between the external forced stream and the buoyancy driven flow induced by buoyancy forces could lead to complex flow structures. Therefore it is important to understand the heat transfer characteristics of mixed convection in a ventilated enclosure. Three possible cases can be found due to buoyancy-induced motions. If the flow is in the same direction as buoyancy force, it is known as assisting flow; if it is in opposite direction, known as opposing flow and if both directions are perpendicular with each other, the flow is considered as transverse flow.

If an enclosure is equipped with baffles, placed on a very short distance from each other it will give greater surface area for heat transfer, but a smaller heat transfer co-efficient because of the extra resistance in the additional baffle to fluid flow through inter baffle passages. The friction force increases as more and more solid surfaces are introduced, seriously disrupting the fluid flow and heat transfer. Most of the previous researches was conducted in horizontal enclosure and studied the fluid flow and heat transfer characteristics in a normal situation. Here a number of heated baffles have been placed in a small vertical enclosure with little baffle space, which has made a complex geometry for this experiment. This situation has been considered as a research gap and so, this experimental model has been investigated to analyze the fluid flow and heat transfer characteristics.

2. REVIEW OF LITERATURE

Several investigations both numerically and experimentally are carried out to investigate the effect of baffle. Berner, C. *et al.*, [1] presented the effect of baffle spacing and window cuts at the range of Re number from 600 to 10,500. Fung Hing Choi, Lazaridis Anastas [2] showed numerically the heat transfer of baffle on the vertical hot wall of a square cavity having the horizontal walls are kept insulated. The result found that adding baffle on the hot wall can increase the rate of heat transfer by as much as 31.46% compared with a wall without baffle for Ra=104. Bruno Monte Da Silva Miranda [3] studied numerically heat transfer characteristics for parallel plate channel with sixteen porous baffles in a staggered arrangement with different baffle spacing and the baffle aspect ratios. Dermartini L.C. *et al.*, [4] tested both numerically and experimentally the flow in the rectangular section with two rectangular baffles. Moosavy S.S., Hooman K. [5], showed numerically increase of Re and Pr, increase the Nusselt number, as expected for horizontal channel with isothermal walls having staggered baffles. Nasiruddin, Kamran Siddiqui M.H. [6], have found that the average Nusselt number for the two baffles is 20% higher than the one baffle and 82% higher than no baffle. Sivasankaran S. and Kandaswamy P. [7], investigated numerically to analyze the influence of baffle for natural convection in a rectangular partitioned enclosure with isothermal side walls and insulated top and bottom. Chen Han-Taw *et al.*, [8] investigated on a vertical square fin to predict the average heat transfer coefficient h and fin efficiency ηf of finned-tube heat exchangers for various air speeds and fin spacing. Razmi

Amirmahdi, Firoozabadi Bahar, and Ahmadi Goodarz [9], numerically investigated the effect of baffles in settling tanks. A. Bahlaoui, A. Raji, M. Hasnaoui, M. Naïmi, T. Makayssi, M. Lamsaadi [10] investigated numerical mixed convection cooling combined with surface radiation in a portioned rectangular cavity with baffle.

Due to various application of mixed convection, Researchers are also given their focus on mixed convection with heating source without the presence of baffle. Sumon saha, *et al*., [11], showed numerically for mixed convection the best configuration for cooling efficiecncy inside a vented enclosure varying location of inlet and outlet with constant heat flux form uniformly heated bottom wall. Rahman *et al*., [12], investigated numerically for mixed convection in a vented enclosure with various inlet port configurations. Behzad Ghasemi and Saiied Mostafa Aminossadati [13], numerically investigate the cooling performance of electronic devices with an emphasis on the effects of the arrangement and number of electronic components. With the rapid development of digital computers and complexity developed by analytical method at present numerical method is rather used for calculating the heat flux. Influence of the Reynolds number, Richardson number and length of the baffle on the flow field and thermal field has been discussed.

3. PHYSICAL MODEL

A vertical ventilated enclosure with baffles placed at the side walls is considered as the physical problem. The dimensionless length of the enclosure is considered as L=2 and the dimensionless height of the enclosure is considered as H=3. Three baffles are considered in the enclosure for the investigation. The baffle length is extended up to more than half of the enclosure length. Parabolic velocity profile has been specified at the inlet section and isothermally heated baffles have been considered for the investigation. Cold fluid is entered through the opening on the left vertical wall at the bottom and exited through the top of the wall. All walls are assumed to be adiabatic and baffle is considered as the isothermally heated. Thus, the heat transfer process is done by mixed convection. Both the size of the inlet section is the same size as the exit opening which is equal to W = 1. The physical model is shown in Figure-1.

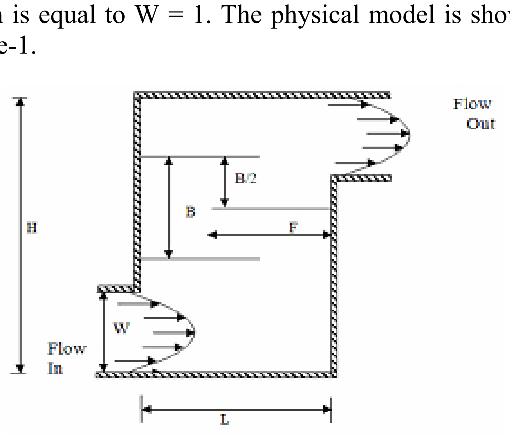

Figure-1. Schematic diagram of the problem.

4. MATHEMATICAL MODEL

4.1 Governing equations

Thermo physical properties of the fluid in the flow model are assumed to be constant except the density variations causing a body force term in the momentum equation. The Boussinesq approximation is invoked for the fluid properties to relative density changes, to temperature changes and to couple the temperature field to the flow field. The governing equations for steady mixed convection flow using conservation of mass, momentum and energy can be written as,

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{2}\frac{\partial p}{\partial x} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{2}\frac{\partial p}{\partial y} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) + Ri\,\theta \tag{3}$$

$$u\frac{\partial \theta}{\partial x} + v\frac{\partial \theta}{\partial y} = \frac{1}{\mathrm{Re}\,.\,\mathrm{Pr}}\left(\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2}\right) \tag{4}$$

Equations (1)-(4) can be converted to the dimensionless forms by definition of the following parameters as:

$$X = \frac{x}{w};\ Y = \frac{y}{w};$$

$$U^* = \frac{u}{u_{\max}};\ V^* = \frac{v}{u_{\max}};$$

$$\mathrm{P}^* = \frac{p}{p_0} = \frac{p}{\frac{1}{2}\rho u_{\max}^2};\ \theta = \frac{T - T\infty}{T_B - T\infty}$$

Therefore using the above parameters leads to dimensionless forms of the governing eqns as below

$$\frac{\partial U^*}{\partial X} + \frac{\partial V^*}{\partial Y} = 0 \tag{5}$$

$$U^*\frac{\partial U^*}{\partial X} + V^*\frac{\partial U^*}{\partial Y} = -\frac{1}{2}\frac{\partial \mathrm{P}^*}{\partial X} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 U^*}{\partial X^2} + \frac{\partial^2 U^*}{\partial Y^2}\right) \tag{6}$$

$$U^*\frac{\partial V^*}{\partial X} + V^*\frac{\partial V^*}{\partial Y} = -\frac{1}{2}\frac{\partial \mathrm{P}^*}{\partial Y} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V^*}{\partial X^2} + \frac{\partial^2 V^*}{\partial Y^2}\right) + Ri\,\theta \tag{7}$$

$$U^*\frac{\partial \theta}{\partial X} + V^*\frac{\partial \theta}{\partial Y} = \frac{1}{\mathrm{Re.Pr}}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \tag{8}$$

4.2 Boundary conditions

At all walls: Since the Velocity Component, u and v both are zero at the walls for no-slip conditions, then stream function ψ = c and ω = c, the walls are considered as adiabatic.

At the baffle: Since the velocity component, u and v both are zero at the walls for no-slip conditions, then, stream function ψ = c and ω = c. The object is assumed to be isothermally heated. So, the dimensionless temperature θ =1.0

Boundary conditions at outlet: Velocity gradients are assumed to be zero for the flow variables as:

$$\frac{\partial u}{\partial x} = 0; \frac{\partial v}{\partial x} = 0; \frac{\partial \psi}{\partial x} = 0; \frac{\partial \omega}{\partial x} = 0; \frac{\partial T}{\partial x} = 0$$

Boundary conditions at the inlet: The parabolic velocity profile is specified at the inlet, which is equal to $u = 4y(1-y)$

And the stream function at inlet section can be specified as,

$$\psi = \left(2y^2 - \frac{4}{3}y^3\right) + \psi_0$$

Where ψo = Constant.

The vorticity at inlet section is

$$\omega = 4(2y-1)$$

And v=0.0.

The fluid entering into the enclosure is cold, therefore temperature at inlet section, T=0.0

4.3 Heat transfer calculations

The heat transfer calculation within the vertical enclosure is measured in terms of the average Nusselt number at the heated baffles as follows,

$$Nu = \frac{1}{L_B}\int_0^{L_B} \frac{H(y)y}{k} dy$$

Here, LB and H(y) are the length and the local convection heat transfer coefficient of the heated baffle respectively. An index of heating effectiveness is the bulk average temperature defined as

$$\theta_{av} = \int \frac{\theta dV}{V}$$

Here V is the volume of the enclosure, which should be minimized.

5. NUMERICAL PROCEDURE

Numerical solutions for the governing equations with the associated boundary conditions were obtained using finite different methods. The governing non-dimensional partial differential equations along with proper boundary conditions have been solved to get stream function, velocities and temperatures at every nodal point in the computational domain. Numerical solution of Singh and Sarif [14], for mixed convection in a rectangular enclosure is observed excellent agreement with the obtained result. The relative tolerance for the error criteria was 10-9.

6. GRID SENSITIVITY TEST

Initially a grid sensitivity check is conducted to choose the proper grid for the numerical prediction. Six different grid sizes were used for grid independence test. Considering the both the accuracy and the computational time a uniform grid system of 34 x 51 grid system were performed throughout the work since it is obvious that more than 34 x 51 grid size does not give more accurate result but increases the computational time only. The result of grid independency is shown in the Figure-2.

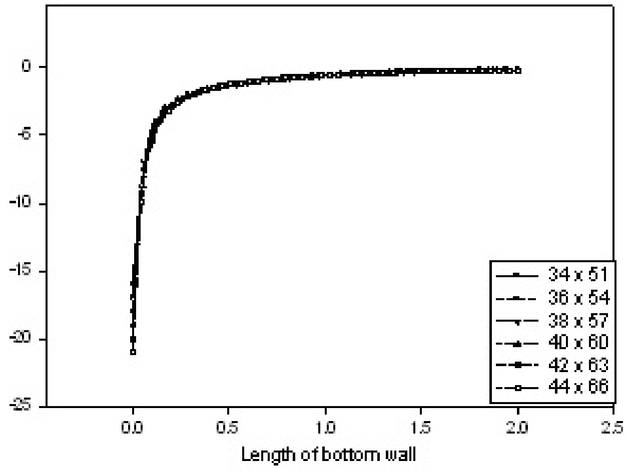

Figure-2. Grid independency test.

7. CODE VALIDATION

A computational model is validated for mixed convection heat transfer by comparing the results of correlation on mixed convection in ventilated cavity performed by Shairf and Shing [14].

Though the difference in the types of the cavity (vertical and horizontal), it is observed that the result from the present work and Sharif and Shing's work are almost similar and obtain good agreement, which is shown above in Figure-3.

From these comparisons it can be decided that the current code can be used to predict the flow field for the present problem.

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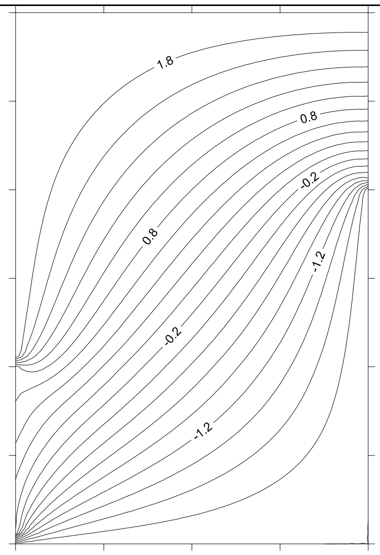
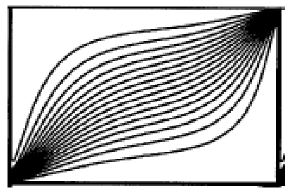
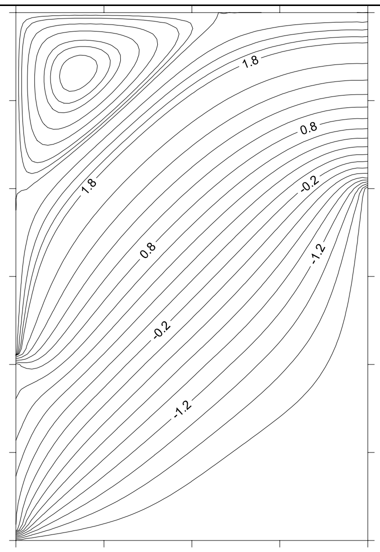
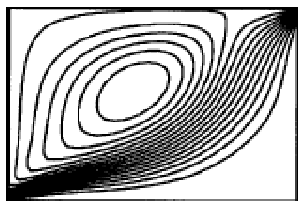
	Ri=0.0	Ri=0.0
Re=50		
Re=300		
	Present work	**Work of sharif and shing** [14]

Figure-3. Comparison with published work.

8. RESULTS AND DISCUSSIONS

In this paper the mixed convective flow inside the vertical enclosure with constantly heated baffles on side walls has been numerically investigated. Since the nature of the flow and thermo-physical properties of fluid strongly influence on the heat transfer rate, the numerical computation was performed for a range of Re, Ri and Pr (Re = 50 - 300; Ri = 0 - 10 and Pr = 0.01 - 2.0). Results have been presented and compared in the form of isotherms, colour spectrums, streamlines, average temperature and Nusselt number, and heating efficiency.

8.1 Flow and thermal field characteristics

Flow field has been expressed by streamlines in Figures 4 and 5, describing the interaction between forced and natural convection.

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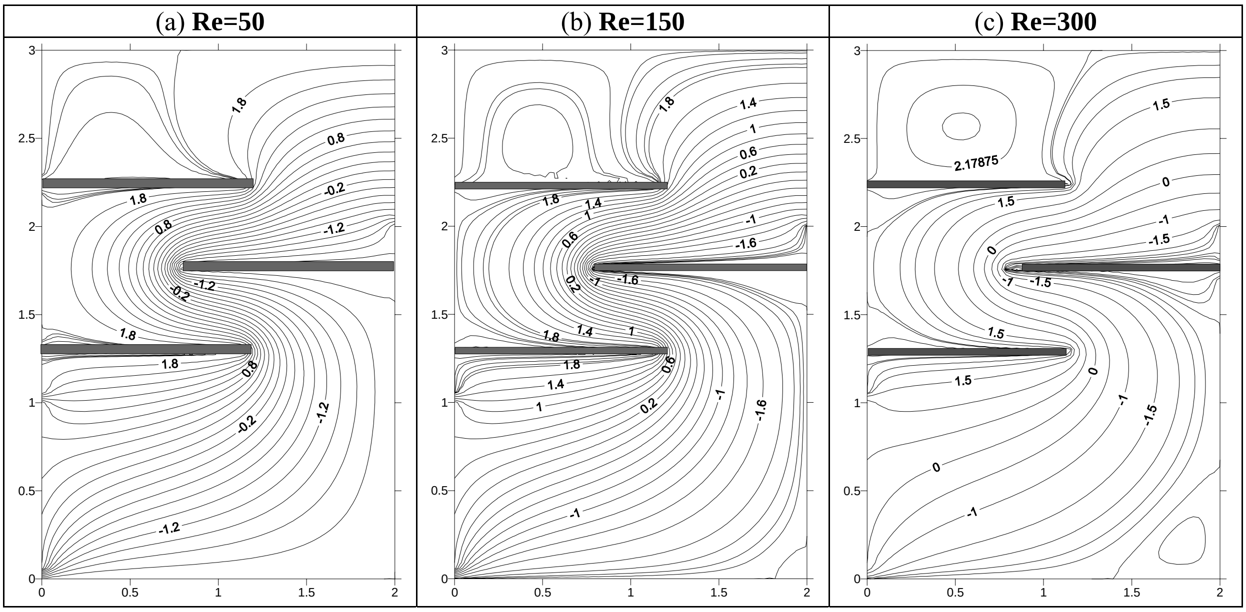

Figure-4. Variations of stream function with Re at Pr=0.71 and RI=0.0.

Figure-4, has described the variation of streamlines with Re = 50 to 300 and Ri = 0.0. For lower Re the induced flow has a little energy and expands suddenly due to pressure rise in to the cavity. Thus the flow nearly occupies whole of the cavity. At higher Re, large recirculation zones have been formed above the main fluid stream at the top-left corner of the enclosure. This is due to at high Re, the flow velocity is high. This has assured more fluid to enter the corner region between third baffle and the top wall of the enclosure which gives enough thrust to form a recirculation at that region.

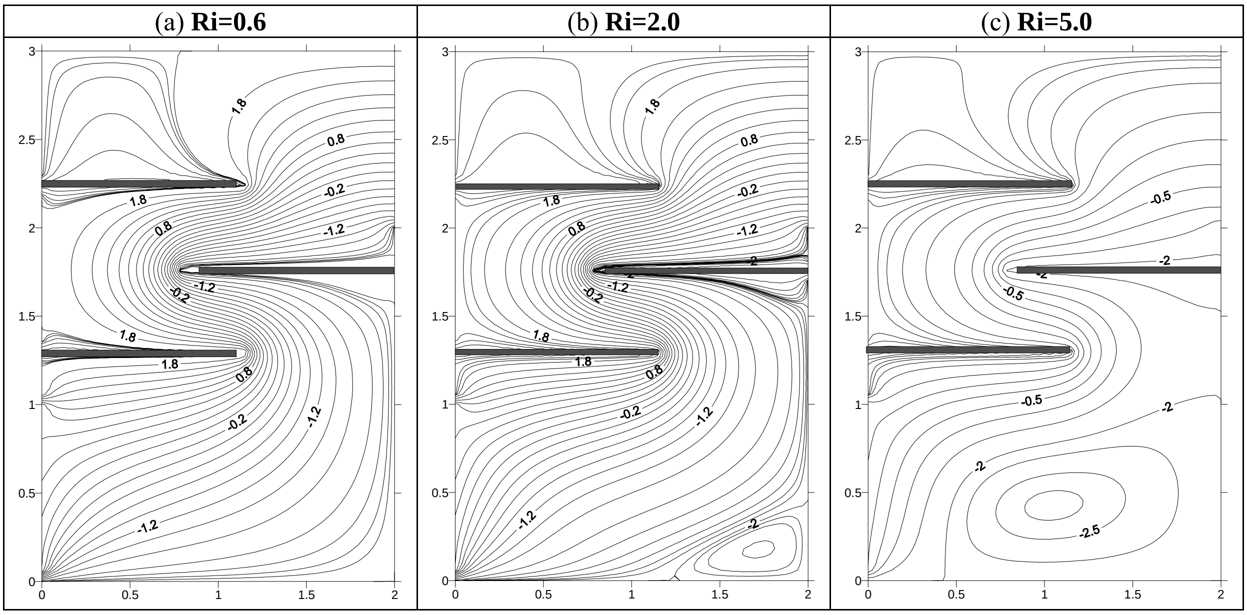

Figure-5. Variations of stream function with Ri at Pr=0.71 and Re=50.

Figure-5, has described the variation of streamlines with Re = 50 and Ri = 0.0 to 5.0. Since increase in Ri augmented the buoyancy effect, so an enlarging recirculation region has been found with increasing Ri at the bottom right corner of the enclosure. It must be noticed that the interaction between recirculation zone and incoming flow gets stronger with the increase of Ri.

Thermal field has been expressed by isotherms and colour spectrums. The isothermal lines along with their corresponding color spectrum have been shown in Figures 6 and 7, respectively for varying Re and Ri.

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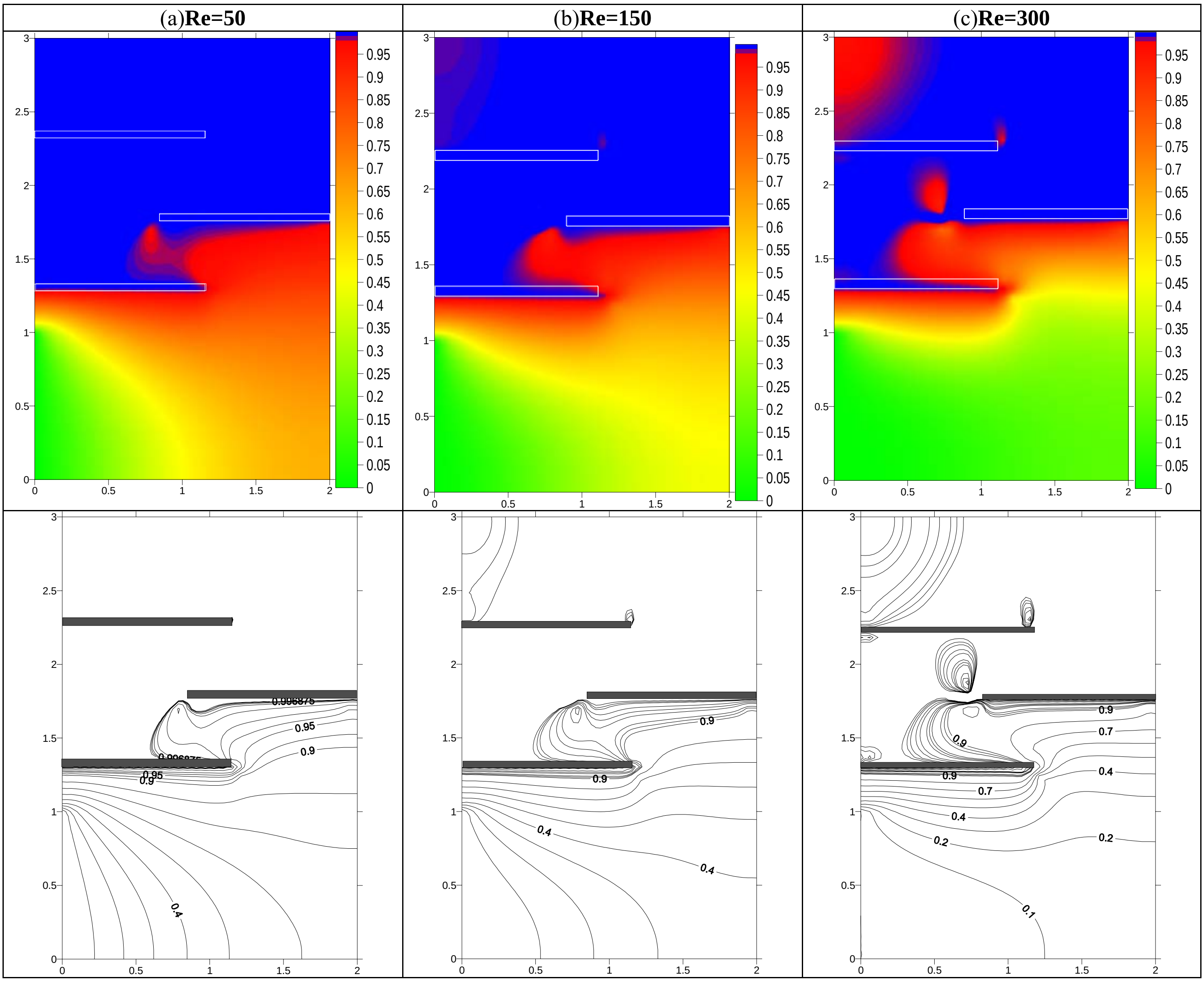

Figure-6. Variation of isotherm and colour spectrum with Re at Pr=0.71 and RI=0.0.

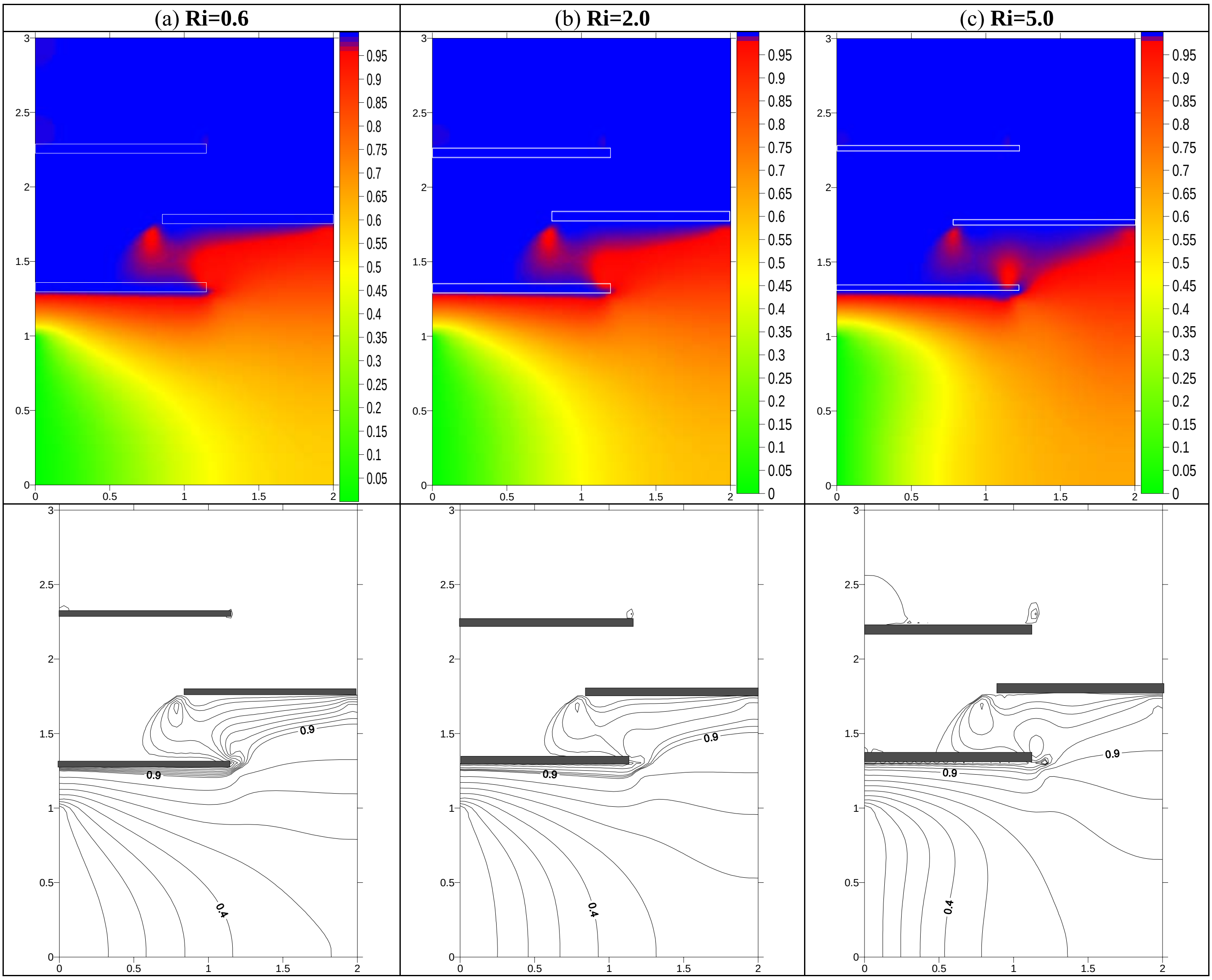

Figure-7. Variation of isotherm and colour spectrum with Ri at Pr=0.71 and Re=100.

It has been clear from the Figure that for low Re and Ri the isotherms are almost uniformly distributed, but with increasing Re and Ri their characteristics have started getting complicated. A recirculation is formed at the top left corner at higher Re which has been shown in Figure-6(c). The thermal boundary layer decreases in thickness slowly as the Ri increases (Figure-7). This has been revealed by the denser concentration of isotherms near the bottom faces of the heated baffles. From color spectrum it is obvious that high temperature region has been found more concentrated near the heated baffles and the cold fluid becomes completely heated after passing the second baffle inside the enclosure. This happens because higher Re contributes to large flow energy which assures more cold fluid to enter into the enclosure and thus decreases the temperature. Again at higher Ri the rotating vortex at the bottom-right corner covers a large portion of the cavity and the inlet flow becomes more pressed or squeezed which improves the contact between flow and heated baffle and enhances the heating rate (Figure-7).

8.2 Heat transfer characteristics

Heat transfer characteristics has been expressed by average Nusselt number and performance of the system has been expressed by heating efficiency. In this experiment three heated baffles have been placed in a very small enclosure with cold incoming fluid. This situation has given greater surface area for heat transfer.

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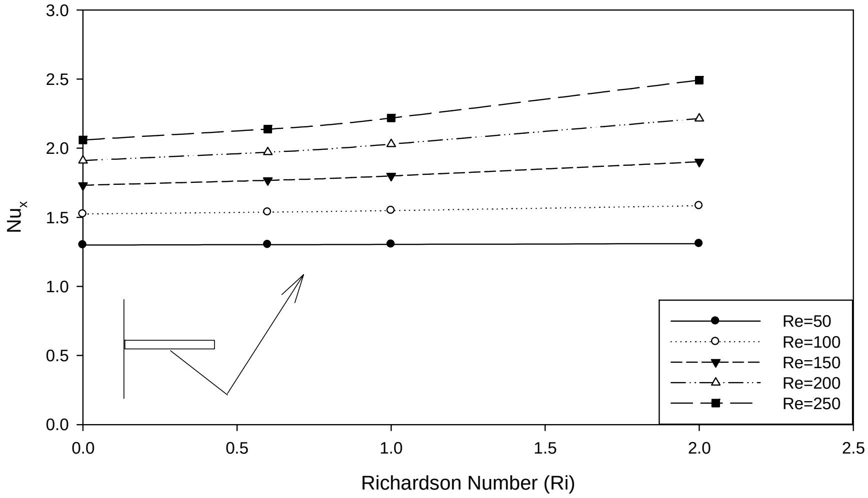

Figure-8. Average Nusselt number at bottom face with the variation of Ri for different Re at Pr=0.71.

Figure-8 shows the variation of average Nusselt number at the bottom face of the baffle with different Re and Ri. It is observed that the average Nusselt number increases with the increase of Re and Ri. It is also observed that at low Reynolds number average Nusselt number is more or less constant. Since the friction force increases as more and more solid surface is introduced, seriously disrupting the fluid flow and heat transfer, so this situation is an example of complex heat transfer characteristics.

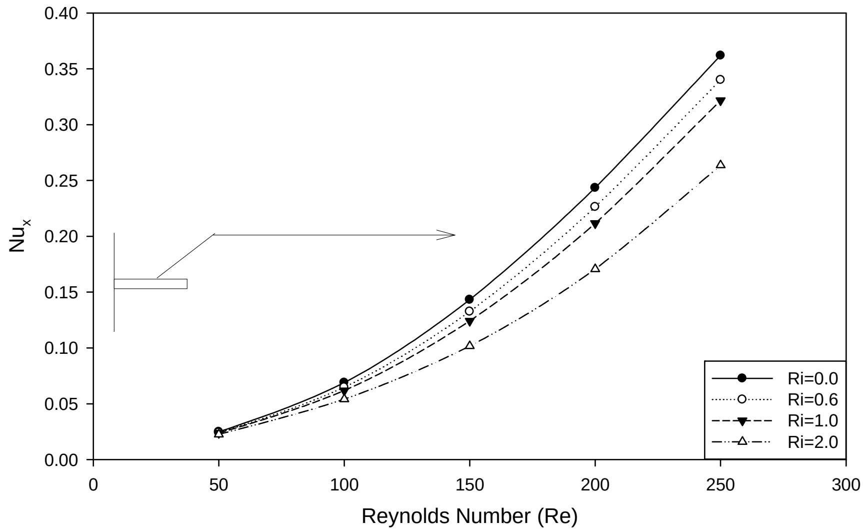

Figure-9. Average Nusselt number at top face with the variation of Ri for different Re at Pr=0.71.

Figure-9 shows the variation of average Nusselt number at the top face of the baffle with different Re and Ri. It is observed that the average Nusselt number at the top surface increases with the increase of Re and decreases with the increase of Ri.

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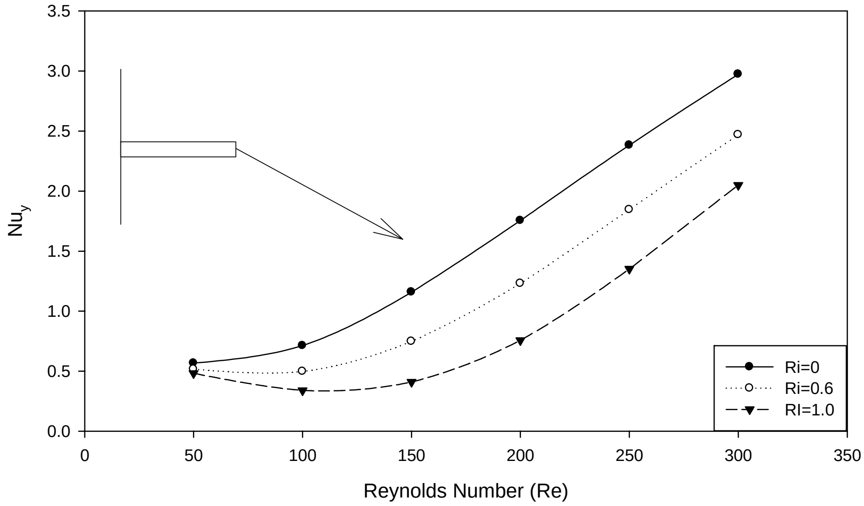

Figure-10. Average Nusselt number at right face with the variation of Re for different Ri at Pr=0.71.

Figure-10 shows the variation of average Nusselt number at the right face of the baffle with different Re and Ri. It is observed that the average Nusselt number at the right surface of the baffle increases with the increase of Re but decreases with the increase of Ri.

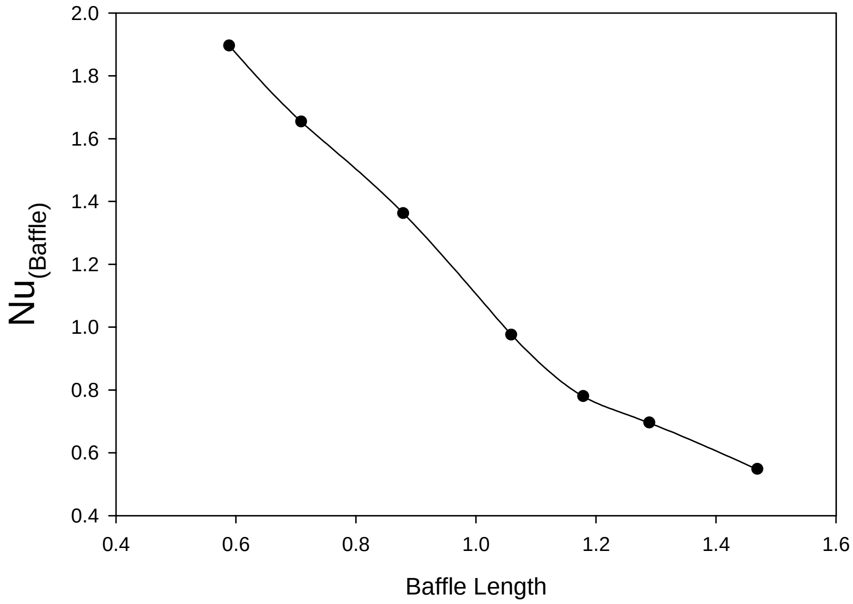

Figure-11. Average Nusselt number of baffle as a function of baffle length for Re=150 and Ri =1.0.

Figure-11 shows the variation of average Nusselt number with the variation of baffle length. It is observed that the average Nusselt number decreases with the increase of baffle length.

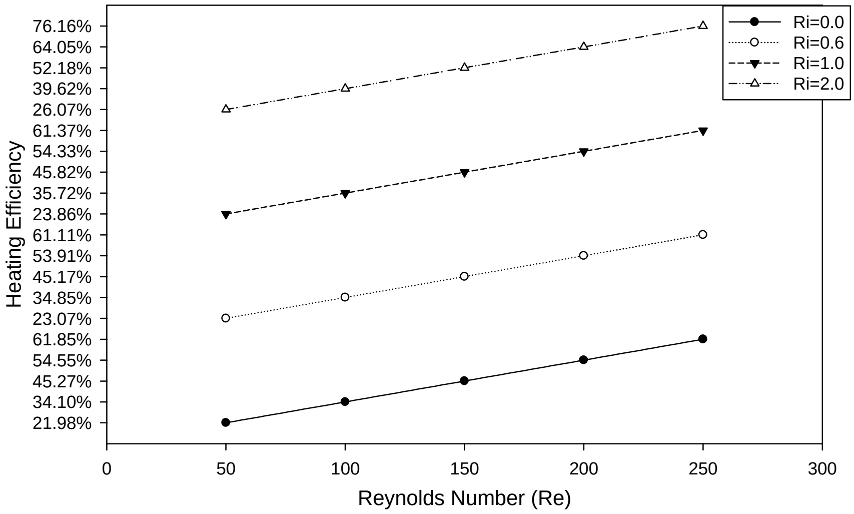

Figure-12. Variation of heating efficiency with Re for Pr=0.71.

Heating efficiency can be defined as a function of Re and Ri which has been shown in Figure-12. It has been observed that heating efficiency increases linearly with the increase of Re and Ri.

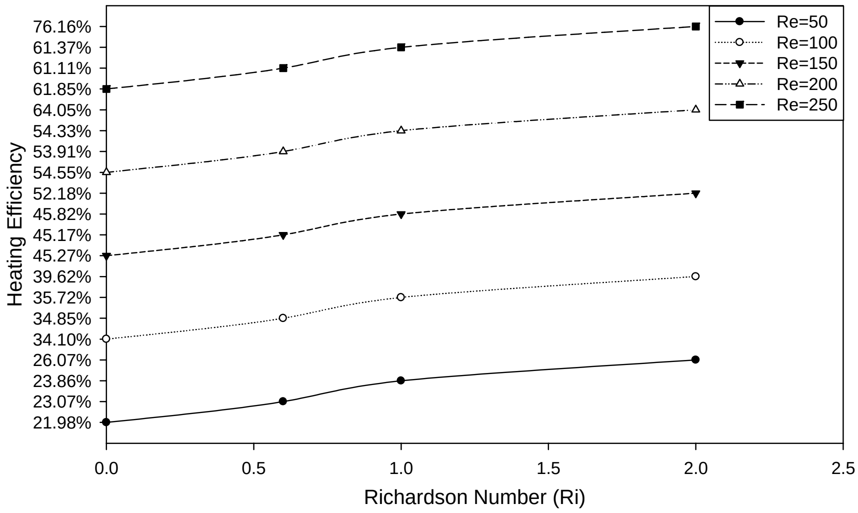

Figure-13. Variation of heating efficiency with Ri for Pr=0.71.

Figure-13 shows the variation of heating efficiency for different Re and Ri. For Re=50, the heating efficiency increases with Ri but the rate of increase is slow. Heating efficiency also increases with the increase of Re, but here the rate of increase is high.

Figure-14 shows the variation of heating efficiency with baffle length. The heating efficiency decreases with the increase of baffle length. Since the experimental model is really complex, cold fluid gets completely heated after passing the second baffle. If the baffle length starts increasing from its intermediate length, the cold fluid starts getting heated just after entering the cavity, but the output temperature remains the same. So the difference between the initial and final temp of the fluid is decreasing with the increase of baffle length, which ultimately reduces the heating efficiency. From here, a critical baffle length can be suggested depending on the size of the experimental model.

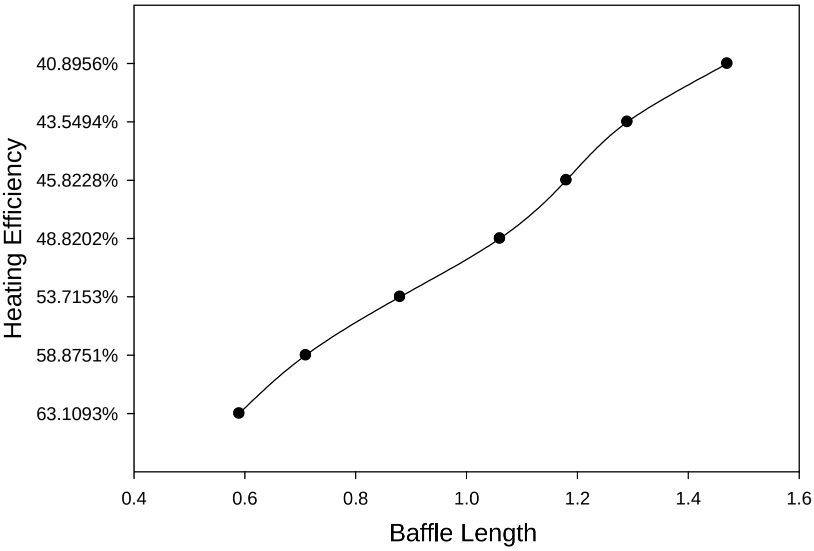

Figure-14. Heating efficiency as a function of baffle length for Pr=0.71, Re=150 and RI=1.0.

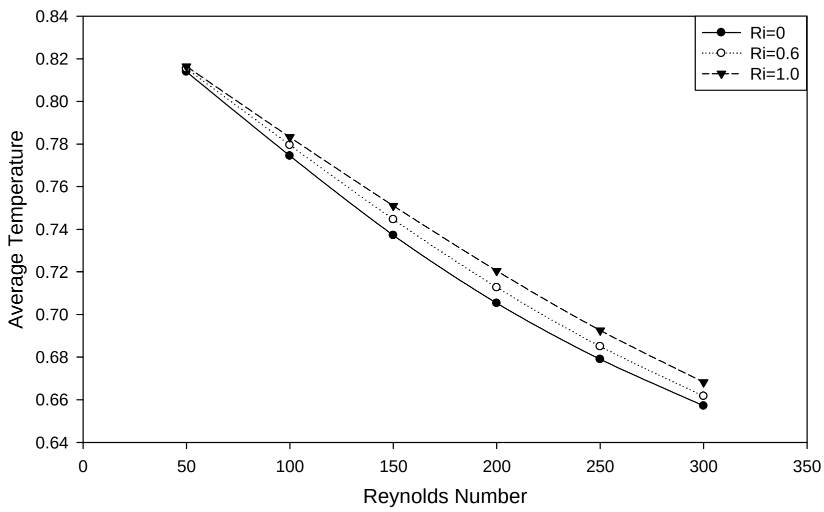

Figure-15. Variation of average bulk temperature with Re at Pr=0.71.

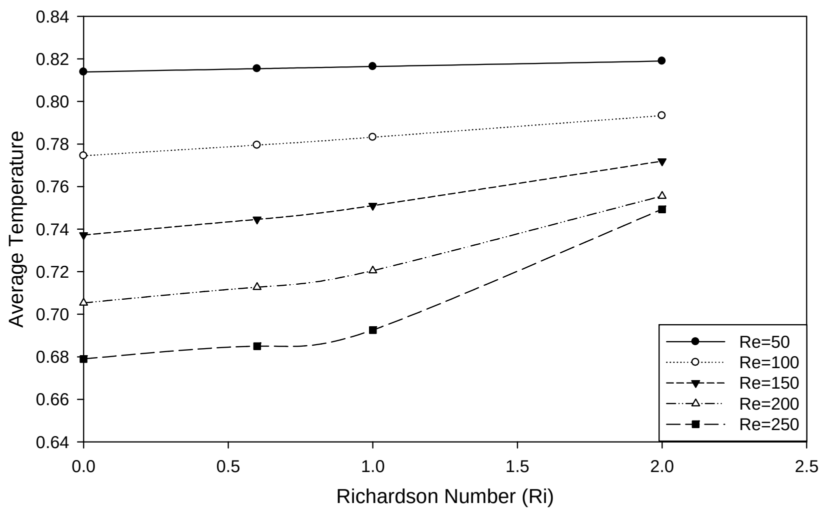

Figure-16. Variation of average bulk temperature with Ri at Pr=0.71.

In Figure-15 and Figure-16, variation of average bulk temperature has been displayed with Re and Ri. For a given Ri and Pr, with the increase of Re average temperature decreases. When Re increases the speed of induced forced flow increases, which does not give sufficient time for the flow to pass along the heated baffles and ultimately get heated, so the average temperature decreases with the increase of Re. But in case of Ri an opposite effect has been found, for increasing Ri the average bulk temperature increases.

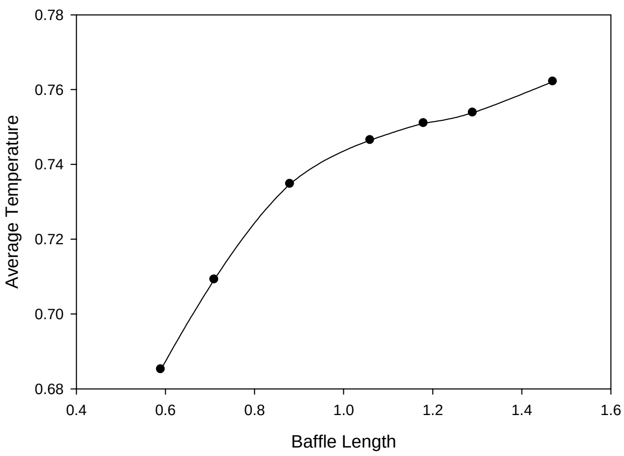

Figure-15. Average temperature as a function of baffle length for Pr = 0.71, Re = 150 and Ri = 1.0

It is interesting to observe that both average Nusselt number and heating efficiency decreases with the increase of baffle length but the average temperature increases with the increase of baffle length which has been shown in Figure-15.

9. CONCLUSIONS

A numerical investigation on mixed convection in a vertical enclosure was carried out using a finite difference method. The present study examined and explained the complex interaction between buoyancy and force flow in a vented vertical enclosure with three heated baffles, where an inlet is situated at the left bottom edge of the vertical insulated wall and the exit port is fixed at the top of the right insulated vertical surface. The study encompasses a range of Reynolds number from 50 to 300, Prandtle Number from 0.01 to 2 and a range of Richardson number from 0 to 10. Details investigation of the heat transfer in terms of fluid temperature and the average Nusselt number has been undertaken for different values of Re, Ri and Pr. The following are the important observations:

a) High temperature region has been found more concentrated near the heated baffles and the cold fluid becomes completely heated after passing the second baffle inside the enclosure.
b) The numerical solution indicates that increasing the value of Re and Ri leads to higher intensity of recirculation and complex fluid flow characteristics.
c) Maximum heating efficiency (up to 76.16%) has been occur with higher value of Re and Ri and it increases gradually with both the increase of Re and Ri.
d) For fixed Re=150 and Ri =1.0 average temperature increases with the increase of baffle length where both average Nusselt number and heating efficiency decreases.

REFERENCES

[1] Berner C., Durst F. and McEligot D.M. 1984. Flow around baffles. ASME Journal of Heat Transfer. 106: 743-749.

[2] Fung H. C. and Lazaridis A. 1996. Finite element analysis of laminar flow in a vertical channel with fin array. Finite Elements in Analysis and Design. 24: 77-93.

[3] Miranda Bruno Monte Da Silva. 2003. A numerical study of convection in a channel with porous baffles. Thesis of Texas A and M University.

[4] Demartini L. C. 2004. Numeric and Experimental Analysis of the Turbulent Flow through a Channel with Baffle Plates. Journal of the Brazil Society of Mechanical Science and Engineering. 26(2).

[5] Moosavy S.S., Hooman K. 2006. Heat and fluid flow in entrance region of a channel with staggered baffles. Energy Conversion and Management. 47(15-16): 2011-2019.

[6] Nasiruddin Kamran Siddiqui M.H. 2006. Heat transfer augmentation in a heat exchanger tube using a baffle. International Journal of Heat and Fluid Flow. 28: 318-328.

[7] Sivasankaran S. and Kandaswamy P. 2007. Effect of Conductive Baffle on Hot-wall in Double Diffusive Convection of Water near Density Maximum. The Arabian Journal for Science and Engineering. 32: 35-48.

[8] Chen Han-Taw, Chou Juei-Che, Wang Hung-Chih. 2007. Estimation of heat transfer coefficient on the vertical plate fin of Finned-tube heat exchangers for various air speeds and fin spacings. International Journal of Heat and Mass Transfer. 50(1-2): 45-57.

[9] Razmi A., Firoozabadi B. and Ahmadi G. 2008. Experimental and Numerical Approach to Enlargement of Performance of Primary Settling Tanks. Journal of Applied Fluid Mechanics. 2(1): 1-12.

[10] Bahlaoui A. Raji, M. Hasnaoui, M. Naimi, T. Makayssi, M. Lamsaadi. 2009. Mixed convection cooling combined with surface radiation in a

partitioned rectangular cavity. Energy conversion and Management. 50: 626-635.

[11] Saha Sumon, Islam Md. Tofiqul, Ali Mohammad, Mamun Md. Arif Hasan and Islam M. Quamrul. 2006. Effect of inlet and outlet locations a transverse mixed convection inside a vented enclosure. Journal of Mechanical Engineering. p. 396.

[12] Rahman M. M., Alim M. A., Mamun M. A. H., Chowdhury M. K. and Islam A. K. M. S. 2007. Numerical Study of Opposing Mixed Convection in a Vented Enclosure. ARPN Journal of Engineering and Applied Sciences. 2(2): 25-36.

[13] Ghasemi B. and Aminossadati Saiied M. 2008. Numerical Simulation of Mixed Convection in a Rectangular Enclosure with Different Numbers and Arrangements of Discrete Heat Sources. The Arabian Journal for Science and Engineering. 33: 1B.

[14] Singh S. and Sharif M. A. R. 2003. Mixed convective cooling of a rectangular cavity with inlet and exit openings on differentially heated side walls, Numerical Heat Transfer, Part A: Applications. 44(3): 233-253.

Nomenclature

B	Distance between two successive baffle	W	Width of inlet opening of the enclosure
Cp	Specific heat of fluid at constant pressure	Re	Reynolds number, $Re = \frac{\rho VW}{\mu}$
F	Length of the baffle	Ri	Richardson number, $Ri = \frac{Gr}{Re^2}$
Gr	Grashsof Number	x,y	Cartesian coordinates
H	Height of the enclosure	u,v	Velocity components
κ	Thermal conductivity	T(x,y)	Local fluid temperature
L	Length of the enclosure	T_B	Baffle temperature
p	Pressure of the flowing fluid	$T\infty$	Free stream temperature
p_0	Initial pressure	U*, V*	Dimensionless velocity components
P*	Pressure in the dimension less form	X, Y	Dimensionless coordinates
Pr	Prandtl number	ρ	Density of fluid
μ	Dynamic viscosity of fluid	θ(x,y)	Dimensionless temperature