Numerical simulation of turbulent mixed convection flow with and without moving plate in a channel.

Neelapu Satish

ME12M1017

Under the guidance of

Dr.K.VENKATASUBBAIAH

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Department of Mechanical & Aerospace Engineering

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Declaration

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N. Satish

Signature

Neelapu Satish

Student Name

ME12M1017

Roll No

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This thesis titled on " Numerical simulation of turbulent mixed convection flow with and without moving plate in a channel" by Mr. Neelapu Satish is approved for the degree of Master of Technology from IIT Hyderabad.

(Dr.Kirti Chandra Sahu)External Examiner
Department of Chemical Engineering
IIT H

(Dr.Pankaj kolhe)Internal Examiner
Department of Mechanical and Aerospace Engineering
IIT H

(Dr.K.Venkatasubbaiah)Adviser
Department of Mechanical and Aerospace Engineering
IIT H

(Dr.Vinod Janardhanan)Chairman
Department of Chemical Engineering
IIT H

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Satish.N

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Abstract

The effect of buoyancy on turbulent mixed convection flow through vertical and horizontal channels has been numerically studied. The effect of heated plate velocity relative to fluid velocity in a vertical channel has been studied. Turbulence is modeled by Reynolds averaged Navier -Stokes equations (RANS) with standard k-ε turbulence model. The present analysis is valid when the buoyancy force effects are small compared with the forced convection effects. The mixed convection flow problem is formulated by two dimensional unsteady incompressible flow with the buoyancy term modeled by Boussinesq approximation. The governing equations are solved by high accuracy compact finite difference schemes with four stage Runge-Kutta method for time integration. Results are reported for Reynolds number of 6000 with Richardson number varied from 0 to 0.5. The velocity, temperature profiles and average Nusselt number values are presented. The heated plate moves at a constant velocity varied from 0.05 to 0.5 in the same and opposite directions of bulk flow. The buoyancy shows significant effect on the flow and heat transfer characteristics in vertical channel in comparison with the horizontal channel. The buoyancy enhances the heat transfer rate in assisting mixed convection flow and opposite trend for opposing mixed convection flow in a vertical channel. A small recirculation zone forms near the inlet due to motion of heated plate. The significant changes of flow and heat transfer characteristics are observed at higher plate velocities. The present results are matching well with the experimental results available in the literature.

NOMENCLATURE

L Width of the channel

H Height of the channel

u, v velocity components in x and y directions respectively

T temperature

k turbulent kinetic energy

Ɛ dissipation rate of turbulent kinetic energy

α thermal diffusivity

υ_t Turbulent eddy viscosity

P pressure

Re Reynolds number

Gr grashof number

Ri Richardson number

ω, ψ Dimensional form of vorticity and stream function

U, V Non dimensional velocity components in x and y directions respectively

θ Non dimensional temperature

Ψ, Ω Non dimensional stream function and vorticty function

k_n Non dimensional kinetic energy

ε_n Non dimensional dissipation rate of kinetic energy

Contents

Chapter 1

INTRODUCTION

The study on turbulent mixed convection flow through channel is important as it has practical engineering applications such as cooling of electronic components, internal cooling system of blades of gas turbine, heat exchanger, solar panels. These devices have become part of human life. The efficiency of these systems is of great importance to the industry. The main function of the thermal control system of any electronic component is to maintain a relatively constant temperature, which should be below the maximum allowable temperature of the component. As the component dissipates more amount heat then the cooling system has to accommodate which means fluid velocity has to be increased to maintain the component below the prescribed limits. The temperature difference causes generation of buoyancy which causes the fluid flow in free convection and forced convection. The buoyancy force acts vertically, but the forced flow may be in any direction. In vertical, internal flows, the forced flow may be upward or downward, and the heat transfer may be to or from the fluid in the conduit. The upward forced flow is termed as assisted mixed convection flow because the natural convection created by buoyancy is in the same direction as the bulk flow. In contrast, the downward flow is called opposing mixed convection flow based on its direction opposite to the natural convection. The study of fluid flow and heat transfer characteristics with a moving plate in a channel is important as it has several engineering applications like rolling, extrusion, continuous casting, melt spinning and production of glass fiber. The quality of the product is mainly depends on the temperature distribution and flow characteristics hence the present study would be very useful.

1.1 Literature survey

Studies on mixed convection flows has received considerable attention in literature. Jackson et al. [1] presented a comprehensive review of experimental and theoretical studies on mixed convection flow in vertical tubes. Experiments [2, 3] were performed to investigate the effect of buoyancy on turbulent mixed convection flows in a vertical passages. The fully developed, turbulent combined forced and natural convection between two vertical parallel plates kept at different temperatures was investigated through a series of direct numerical simulations [4]. Sekimoto [5] numerically investigated the effect of buoyancy on turbulent mixed convection flow in a horizontal square duct and they found that the secondary flow regimes appears with increase in Richardson number. Conjugate turbulent mixed convection heat transfer in a vertical channel with a flush-mounted discrete heat source in each channel wall [6] were studied using k-ε turbulence model, effects of thermal conductivity of wall, heat source and Reynolds number have been studied. Turbulent mixed convection of air flow in vertical tubes [7] under constant heat flux have been studied and the effect of buoyancy force on the heat transfer coefficient is investigated using a zonal turbulence model. Low Reynolds number mixed convection flows through vertical tube [8] were studied. Mixed convection flow in convergent nozzle has been studied with heated smoke wire experimentally by Huang [9], flow reversal observed for both assisting and aiding flow. Sommer [10] derived the turbulent heat flux model for buoyant flow which is very useful for wall bounded flows based on algebraic techniques. Cheng studied the buoyancy induced recirculation bubbles [11]. Turbulent flow and heat transfer in pipes with buoyancy effects [12] has been investigated with consideration of parabolic flow using turbulence model. Correlations for mixed turbulent natural and forced convection heat transfer in vertical tubes with consideration of influence of length to diameter and heat, mass fluxes direction on heat transfer were given by Aicher [13]. Experiments were conducted on mixed convection of air flow through horizontal and inclined channel by Maughana [14]. Assunta Andreozz numerically studied mixed convection in air due to the interaction between a buoyancy flow and the flow induced by a moving plate in a vertical channel, the effect of channel spacing ,heat flux and plate velocity are carried out [15]. The numerical solution for the boundary layer flow field of a stretched surface has been investigated by Sakiadas [16]. The heat transfer characteristics for a boundary layer convective flow over continuous flat on isothermal surface moving in a presence of heat source have been studied by Swati Mukhopadhyay [20]. The steady boundary layer flow and heat transfer of a viscous and incompressible fluid in the stagnation point towards a non-linearly moving flat plate in a parallel free stream with a partial slip velocity studied by Ioan Pop [21]. Magyari calculated the heat transfer coefficient over continuous moving surface by using direct method [22]. The effect of temperature dependent viscosity on laminar mixed convection boundary layer flow and heat transfer on a continuously moving vertical surface is studied [23].

1.2 Motivation

Studies in the past have not emphasized the effect of buoyancy on turbulent forced convection dominated mixed convection flows. The effect of buoyancy on turbulent mixed convection flow through vertical and horizontal channels using high accuracy numerical methods are not available in literature. The effect of heated plate velocity relative to the fluid velocity on flow and heat transfer characteristics is not available in the literature. This has been the motivation for present investigation

1.3 Objectives of present study

The objective of present study is to understand the flow and heat transfer characteristics of turbulent mixed convection flow with and without moving plate in a channel. The problem statement(s) of the current study is summarized as follows:

- To study the effect of buoyancy on assisting and opposing turbulent mixed convection flows in a vertical channel.
- To study the effect of Reynolds number on flow and heat transfer characteristics in a channel
- To study the effect of buoyancy on turbulent mixed convection flows in a horizontal channel.
- To study the effect of heated plate velocity relative to the fluid velocity in a vertical channel.

1.4 Outline of thesis

Numerical simulation of turbulent mixed convection flow with and without moving plate in a channel

- Chapter 2 deals with the governing equation of two dimensional incompressible mixed convection flow through channel
- Chapter 3 deals the numerical methods
- Chapter 4 discusses the results of mixed convection flow through channel
- Chapter 5 deals with the results of mixed convection flow with moving plate in a channel

Chapter 2

Governing equations and boundary conditions

Turbulent mixed convection flow through vertical channel and horizontal channel is described by Reynolds averaged Navier-Stokes equations (RANS) in which the momentum fluxes appears as stress terms which leads to turbulence closure problem which could be solved by any turbulence model. For this problem Turbulence is modeled with standard k-ε model including the contribution of buoyancy force in the turbulent kinetic energy generation and dissipation. Reynolds time averaging is suitable for stationary turbulence. In this analysis at any instant of time flow variable can be expressed as the sum of mean and fluctuating quantities. The buoyancy term is modeled by Boussinesq approximation that treats density as a constant value in all equations, except for the buoyancy term in the momentum equation. Two dimensional unsteady RANS equations are obtained in the stream function ψ and vorticity ω formulation. Advantages of this formulation are added accuracy due to exact satisfaction of mass conservation and reduction in the number of unknowns to two as compared to three unknowns for primitive variable formulation. Pressure is eliminated by taking the curl of the momentum equation to give the vorticity transport equation (VTE).

2.1 Dimensional form governing equations

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

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{\partial p}{\rho\partial x}+\upsilon[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}]+2\frac{\partial}{\partial x}[\upsilon_t\frac{\partial u}{\partial x}]+\frac{\partial}{\partial y}[\upsilon_t\frac{\partial u}{\partial y}]+\frac{\partial}{\partial y}[\upsilon_t\frac{\partial v}{\partial x}]$$

$$\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{\partial p}{\rho\partial y}+\upsilon[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}]+\frac{\partial}{\partial x}[\upsilon_t\frac{\partial u}{\partial y}]+\frac{\partial}{\partial x}[\upsilon_t\frac{\partial v}{\partial x}]+2\frac{\partial}{\partial y}[\upsilon_t\frac{\partial v}{\partial y}]+\mathrm{g}\,\beta(\mathrm{T}-\mathrm{T}_m)$$

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{\partial}{\partial x}[(\alpha+\frac{\upsilon_t}{\Pr_t})\frac{\partial T}{\partial x}]+\frac{\partial}{\partial y}[(\alpha+\frac{\upsilon_t}{\Pr_t})\frac{\partial T}{\partial y}]$$

$$\frac{\partial k}{\partial t}+u\frac{\partial k}{\partial x}+v\frac{\partial k}{\partial y}=\frac{\partial}{\partial x}[(\upsilon+\frac{\upsilon_t}{\sigma_k})\frac{\partial k}{\partial x}]+\frac{\partial}{\partial y}[(\upsilon+\frac{\upsilon_t}{\sigma_k})\frac{\partial k}{\partial y}]-\frac{g\beta\upsilon_t}{\Pr_t}\frac{\partial T}{\partial y}$$
$$-\varepsilon+\upsilon_t[2(\frac{\partial u}{\partial x})^2+2(\frac{\partial v}{\partial y})^2+(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})^2]$$

$$\frac{\partial \varepsilon}{\partial t}+u\frac{\partial \varepsilon}{\partial x}+v\frac{\partial \varepsilon}{\partial y}=\frac{\partial}{\partial x}[(\upsilon+\frac{\upsilon_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x}]+\frac{\partial}{\partial y}[(\upsilon+\frac{\upsilon_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial y}]-\mathrm{c}_{2\varepsilon}\ f_2\frac{\varepsilon^2}{k}$$
$$+c_{1\varepsilon}f_1[\upsilon_t\{2(\frac{\partial u}{\partial x})^2+2(\frac{\partial v}{\partial y})^2+(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})^2\}-c_{3\varepsilon}\{\frac{g\beta\upsilon_t}{\Pr_t}\frac{\partial T}{\partial y}\}]$$

$c_{1\varepsilon}=1.44, c_{2\varepsilon}=1.92, c_{3\varepsilon}=0.486$, $\upsilon_t=c_\mu k^2/\varepsilon$, $c_\mu=0.09$, $\sigma_k=1.0, \sigma_\varepsilon=1.3$

2.2 Boundary conditions

Uniform velocity and temperature are specified at the inlet of the channel. No slip velocity boundary conditions are applied on the walls and prescribed velocity applied to moving plate according to direction. At the outlet of the channel, fully developed flow condition is applied for both velocity and temperature fields. The left wall of the channel is considered to be insulted and adiabatic boundary condition is applied to it. Constant wall temperature is applied at the right wall. The turbulent kinetic energy and normal gradient of dissipation is set to zero at the walls. The initial conditions for velocity and temperature fields are considered as inviscid solution.

2.2.1 Mixed convection flow through the channel

- At inlet uniform velocity and temperature are specified

  $u=u_i,\ T=T_i$

- Left wall is adiabatic and no slip boundary condition applied

  $$u=0, v=0, \frac{\partial T}{\partial y}=0$$

- Right wall is at constant wall temperature and no slip boundary conditions applied

  $T = T_s, u = 0, v = 0$

- At out let fully developed boundary conditions applied

  $\frac{\partial u}{\partial x} = 0, \frac{\partial T}{\partial x} = 0$

2.2.2 Continuous moving plate

- At inlet ,uniform velocity and temperature are specified

  $u = u_i, T = T_i$

- Left wall is adiabatic and no slip boundary condition applied

  $u = 0, v = 0, \frac{\partial T}{\partial y} = 0$

- Right wall is at constant wall temperature and having constant velocity less than to inlet velocity

  $T = T_s, u = nu_i, v = 0$

  Where n=0.1, 0.5,-0.1,-0.5 (different fractions of velocities)

- At out let fully developed boundary conditions applied

$\frac{\partial u}{\partial x} = 0, \frac{\partial T}{\partial x} = 0$

2.3 Stream function Vorticity formulation

The two dimensional incompressible governing equations are solved in stream function and vorticity formulation which gives accuracy and reduces number of unknowns to two as compared to three in primitive variable formulation.

2.4 Stream function

It is the function of spatial coordinates and time. At particular instant stream function gives the picture of flow field with stream lines to which drawn tangent at particular point gives the velocity field at that point.

$$u = \frac{\partial \psi}{\partial y}; v = -\frac{\partial \psi}{\partial x}$$

2.5 Vorticity

It is the function of space and time, specifies if the flow is rotational or irrotational and defined as curl of velocity.

$$\omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = -\frac{\partial^2 \psi}{\partial x^2} - \frac{\partial^2 \psi}{\partial y^2}$$

The dimensionless governing equations are obtained by the following variables

$$X = \frac{x}{L}, \mathrm{Y} = \frac{y}{L}, U = \frac{u}{u_i}, \mathrm{V} = \frac{v}{u_i}, \tau = \frac{tu_i}{L}, \theta = \frac{T - T_i}{T_s - T_i}$$

$$\Omega = \omega \frac{L}{u_i}, \Psi = \frac{\psi}{u_i L}, \mathrm{k}_n = \frac{k}{u_i^{\,2}}, \varepsilon_n = \frac{\varepsilon L}{u^3{}_i}$$

The governing equations are

$$\frac{\partial^2 \Psi}{\partial X^2} + \frac{\partial^2 \Psi}{\partial Y^2} = -\Omega$$

$$\frac{\partial \Omega}{\partial \tau} + U\frac{\partial \Omega}{\partial X} + V\frac{\partial \Omega}{\partial Y} = \frac{1}{\mathrm{Re}}\frac{\partial^2}{\partial X^2}[(1+\frac{\upsilon_t}{\upsilon})\Omega] + \frac{1}{\mathrm{Re}}\frac{\partial^2}{\partial Y^2}[(1+\frac{\upsilon_t}{\upsilon})\Omega] + \frac{2}{\upsilon\,\mathrm{Re}}\frac{\partial U}{\partial Y}\frac{\partial^2 \upsilon_t}{\partial X^2} - \frac{2}{\upsilon\,\mathrm{Re}}\frac{\partial V}{\partial X}\frac{\partial^2 \upsilon_t}{\partial Y^2} + Ri*\frac{\partial \theta}{\partial X}$$
$$+\frac{2}{\upsilon\,\mathrm{Re}}[\frac{\partial V}{\partial Y} - \frac{\partial U}{\partial X}]\frac{\partial^2 \upsilon_t}{\partial X \partial Y}$$

$$\frac{\partial \theta}{\partial \tau} + U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{\partial}{\partial x}[(\frac{1}{\mathrm{Pr}} + \frac{\upsilon_t}{\upsilon\,\mathrm{Pr}_t})\frac{\partial \theta}{\partial x}] + \frac{\partial}{\partial y}[(\frac{1}{\mathrm{Pr}} + \frac{\upsilon_t}{\upsilon\,\mathrm{Pr}_t})\frac{\partial \theta}{\partial Y}]$$

$$\frac{\partial k_n}{\partial \tau} + U\frac{\partial k_n}{\partial X} + V\frac{\partial k_n}{\partial Y} = \frac{1}{\mathrm{Re}}\frac{\partial}{\partial x}[(1+\frac{\upsilon_t}{\upsilon\sigma_k})\frac{\partial k_n}{\partial X}] + \frac{\partial}{\partial Y}[(1+\frac{\upsilon_t}{\upsilon\sigma_k})\frac{\partial k_n}{\partial Y}] - \frac{Ri*\upsilon_t}{\mathrm{Re}\,\upsilon}\frac{\partial \theta}{\partial Y}$$
$$-\varepsilon_n + \frac{\upsilon_t/\upsilon}{\mathrm{Re}}[2(\frac{\partial U}{\partial X})^2 + 2(\frac{\partial V}{\partial Y})^2 + (\frac{\partial U}{\partial Y} + \frac{\partial V}{\partial X})^2]$$

$$\frac{\partial \varepsilon_n}{\partial \tau} + U\frac{\partial \varepsilon_n}{\partial X} + V\frac{\partial \varepsilon_n}{\partial Y} = \frac{1}{\mathrm{Re}}\frac{\partial}{\partial X}[(1+\frac{\upsilon_t}{\upsilon\sigma_\varepsilon})\frac{\partial \varepsilon_n}{\partial X}] + \frac{\partial}{\partial Y}[(1+\frac{\upsilon_t}{\upsilon\sigma_\varepsilon})\frac{\partial \varepsilon_n}{\partial Y}] - \mathrm{c}_{2\varepsilon}\, f_2\frac{\varepsilon_n^{\,2}}{k_n}$$
$$+\frac{c_{1\varepsilon} f_1}{\mathrm{Re}}[\frac{\upsilon_t}{\upsilon}\{2(\frac{\partial U}{\partial X})^2 + 2(\frac{\partial V}{\partial Y})^2 + (\frac{\partial U}{\partial Y} + \frac{\partial V}{\partial X})^2\} - c_{3\varepsilon}\{\frac{Ri}{\mathrm{Pr}_t}\frac{\partial \theta}{\partial Y}\}]\frac{\varepsilon_n}{k_n}$$

2.6 Non dimensional boundary conditions

2.6.1 Mixed convection flow through the channel

- At inlet Uniform velocity and temperature are specified

  $$U = 1, \theta = 0 \quad \frac{\partial \Psi}{\partial Y} = 1$$

- Left wall is adiabatic and no slip boundary condition applied

  $$\frac{\partial \theta}{\partial Y} = 0, U = 0, V = 0, \Psi = 1, \Omega = -\frac{\partial^2 \Psi}{\partial Y^2}$$

- Right wall is at constant wall temperature and no slip boundary conditions applied

  $$\theta = 1, U = 0, V = 0 \quad \Psi = 0, \Omega = -\frac{\partial^2 \Psi}{\partial Y^2}$$

- At out let fully developed boundary conditions applied

  $$\frac{\partial \theta}{\partial X} = 0, \Omega = -\frac{\partial^2 \Psi}{\partial Y^2}, V = 0$$

Chapter 3

Numerical methods

The governing equations are discretized using finite difference schemes. Finite difference method proceeds by replacing the derivatives in the differential equations by finite difference approximations. The finite difference method optimizes the approximation for the differential operator in the central node of the considered patch and provides the numerical solution for differential equation and the solver is developed in Fortran 90. The stream function equation (SFE) is discretized using a second order central difference scheme CD2. An iteration method of Biconjugate gradient is used to solve the stream function equation. The vorticity transport equation (VTE), energy equation (EE),Kinetic energy and dissipation equations are solved by discretizing the diffusion terms using CD2.Time integration is performed explicitly by the four stage RungeKutta (RK4) scheme and nonlinear convection terms are discretized by high accuracy compact schemes, whose details are given in [17,18]. A small time step of 10^{--4} is used to avoid numerical instability.

Chapter 4

Results and Discussion of mixed convection flow through channel

The turbulent mixed convection flow through vertical and horizontal channels is numerically studied and the heat transfer characteristics are reported for different Richardson numbers. The Prandtl number (Pr) for air is taken as 0.72. Here the buoyancy force effects are small compared with the forced convection effects. The vertical channel has height to width aspect ratio of 9.0. The right wall is maintained at constant surface temperature T_s and it is greater than the inlet temperature T_i of air.

4.1 Grid independence and validation

The grid independence test has been conducted with different grids for Re = 6000.it is noticed that velocity and temperature are shown grid independent with grid size 805×80.All simulations are reported for grid size 805×80. The results are validated for pure forced convection flow (Ri =0) through a channel and average Nusselt number for such flow is 14.58 calculated by using experimental correlation reported in [12] .The value of average Nusselt number of present model is 14.39.

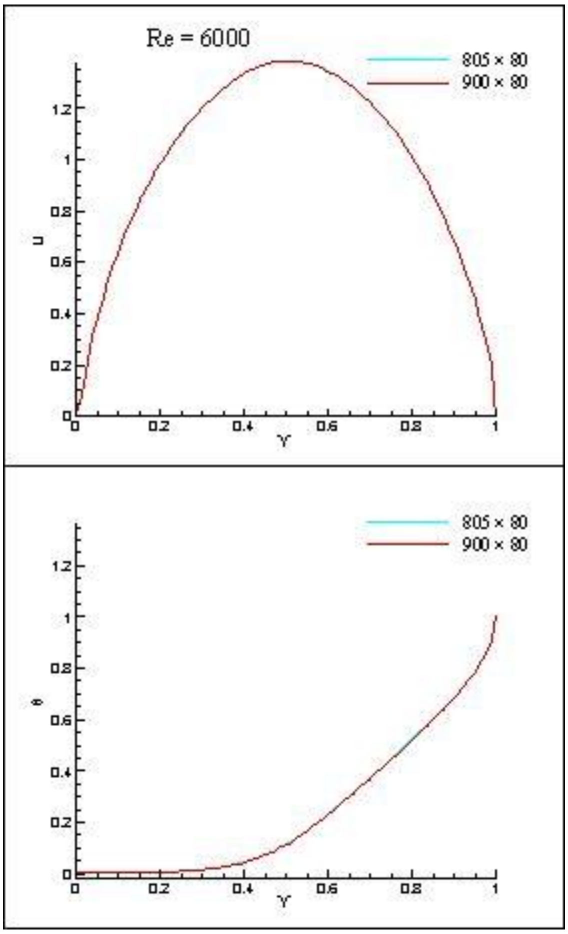


Figure 4.1: grid independency for Re=6000

4.2 Effect of Reynolds number

To know the effect of Reynolds number, results of steady velocity and temperature profile at the outlet of the channel is shown in Figs. 4.2 and 4.3. for forced convection (Ri = 0). There is no significant variation of velocity profile with increases in Re from 3000 to 6000 because the velocity is normalized with inlet velocity. The temperature is decreases with increase in Re which indicates that heat transfer rate increases with increase in velocity of fluid. The average Nusselt number increases from 13.9 to 14.39 with increase in Re from Re = 3000 to 6000.

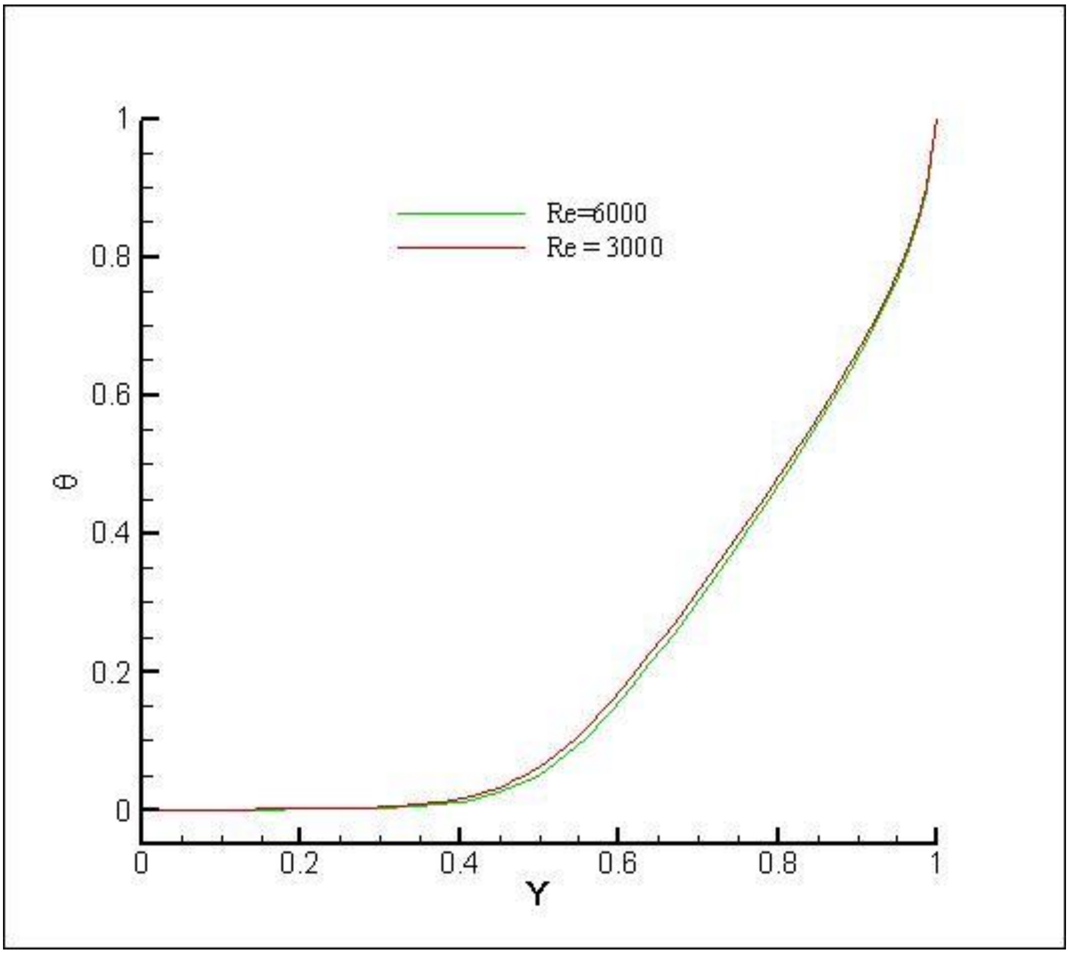


Figure: 4.2 Effect of Re on Temperature profile

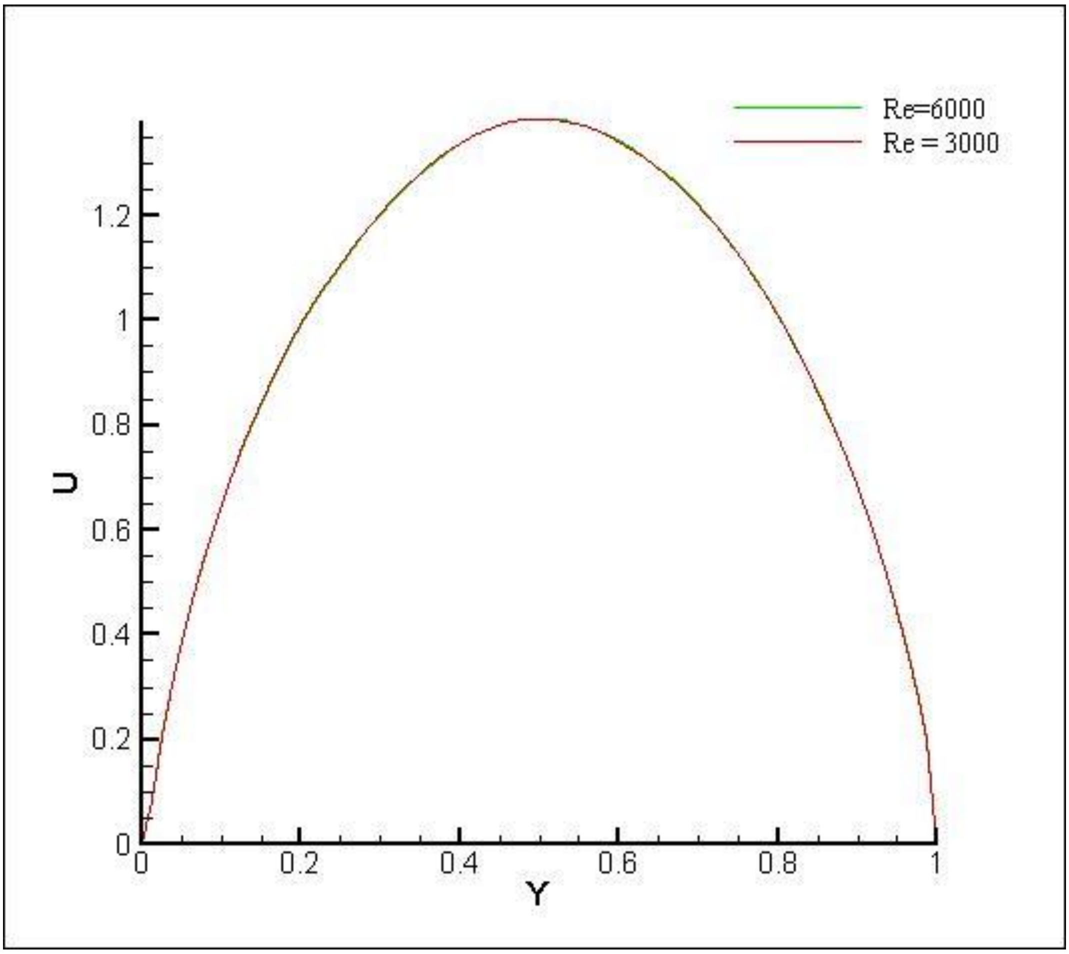


Figure 4.3: Effect of Re on Velocity profile

4.3 Vertical channel

4.3.1 Aiding flow

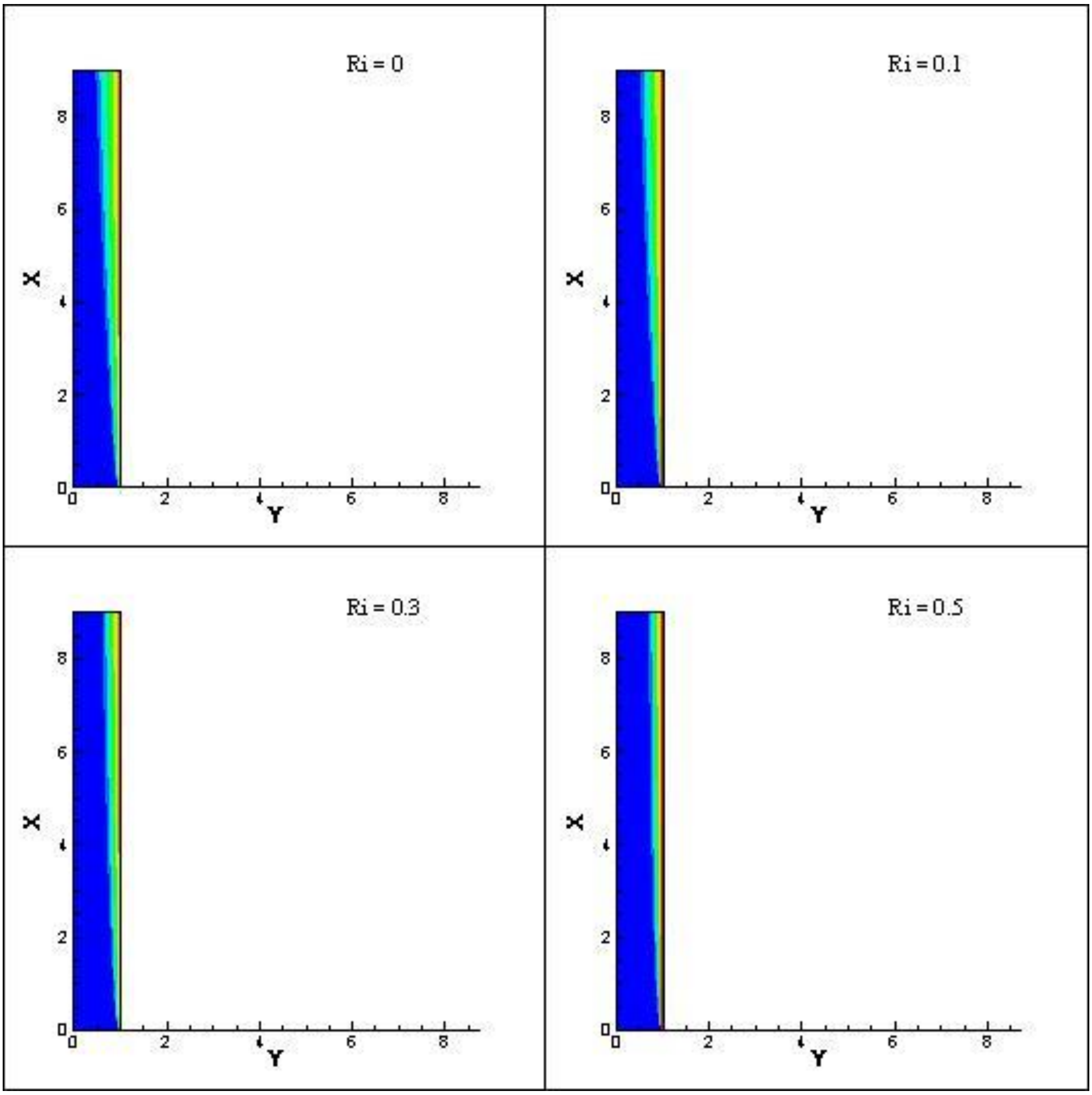


Figure 4.4: Temperature contours for aiding flow at various Richardson numbers

Figure 4.4 illustrates the temperature distribution of aiding flow through channel for different Richardson numbers. The temperature gradients increases as buoyancy increases which results in increase in velocity near the wall and decrease in temperature as shown in figure 4.6. Table 2 shows the average nusselt number for aiding flow. The buoyancy improves the heat transfer as it aids the flow.

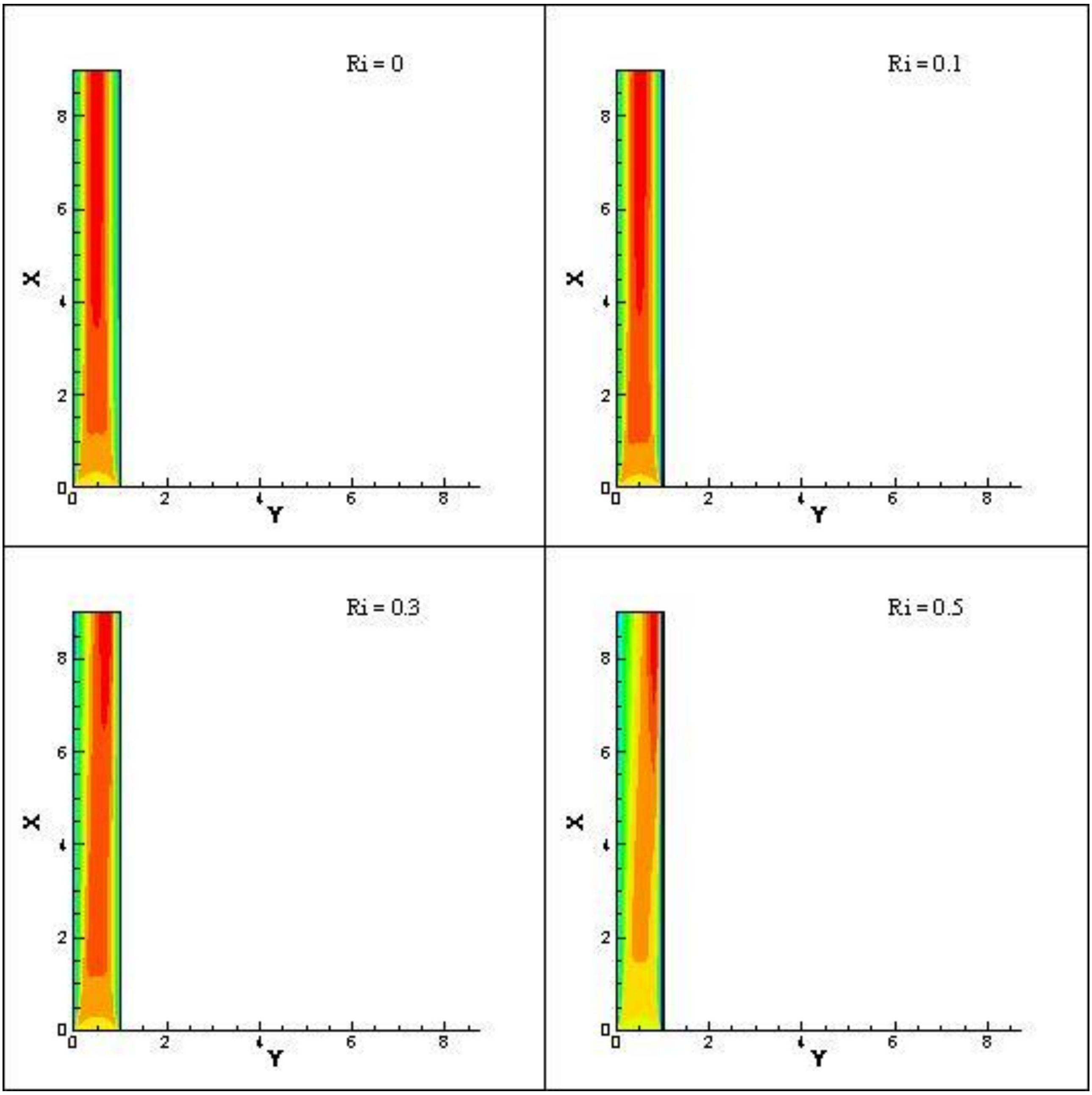


Figure 4.5 : velocity contours for aiding flow at various Richardson numbers

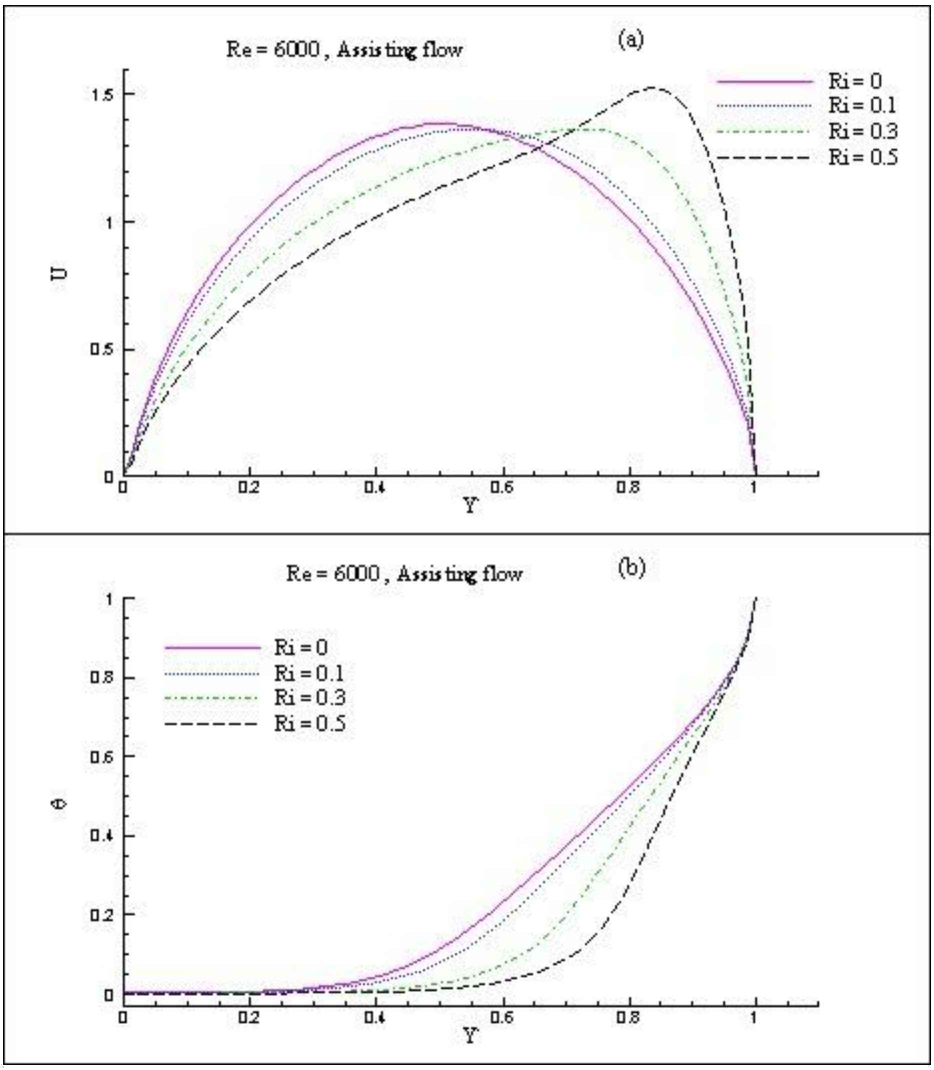


Figure 4.6: Velocity and temperature plots for aiding flow

Table 1 : Average Nusselt numbers for aiding flow

Type of flow	Re	Ri	Nu_{avg}
Forced convection	6000	0	14.39
Assisting flow	6000	0.1	14.63
Assisting flow	6000	0.3	15.05
Assisting flow	6000	0.3	15.45

4.3.2 Opposing flow

The buoyancy has great influence on opposing flow through vertical channel as compared to assisting flow.

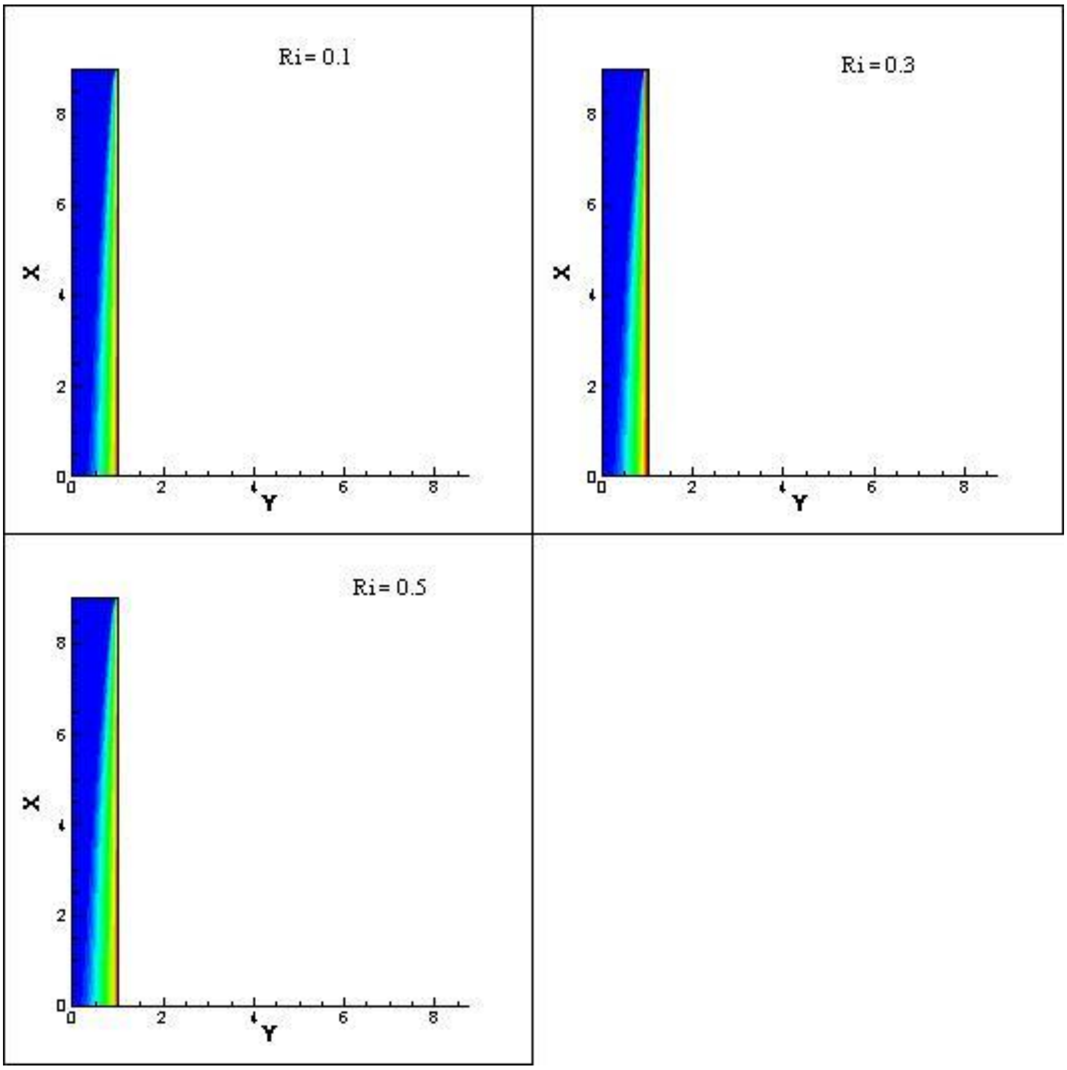


Figure 4.7: Temperature contours of opposing flow at various Richardson number

The buoyancy has significance effect on temperature distribution in opposing flow which can be seen from figure 4.7. As buoyancy increases the temperature and velocity near the heated wall decrease as shown in figure 4.8. Buoyancy also alters the flow pattern i.e. velocity contour can be seen from the figure 4.9.

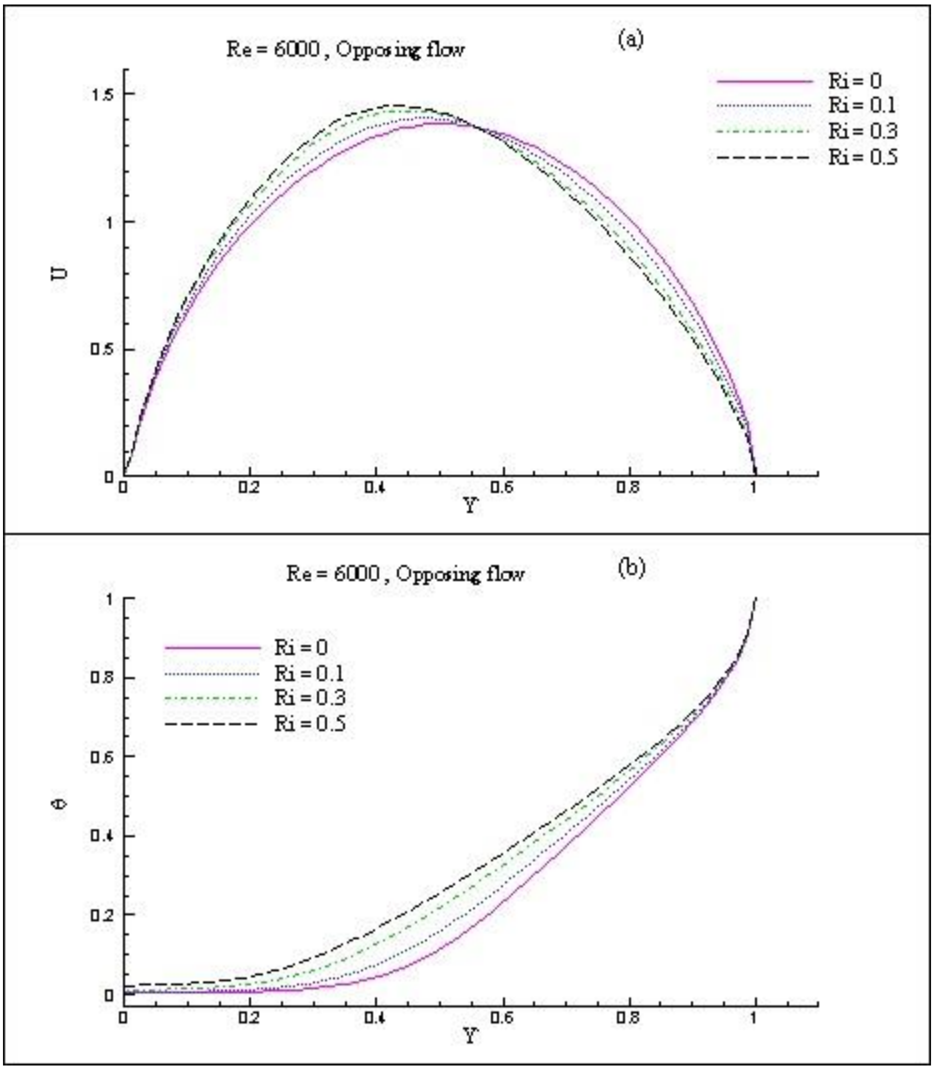


Figure 4.8: velocity and temperature plots for opposing flow

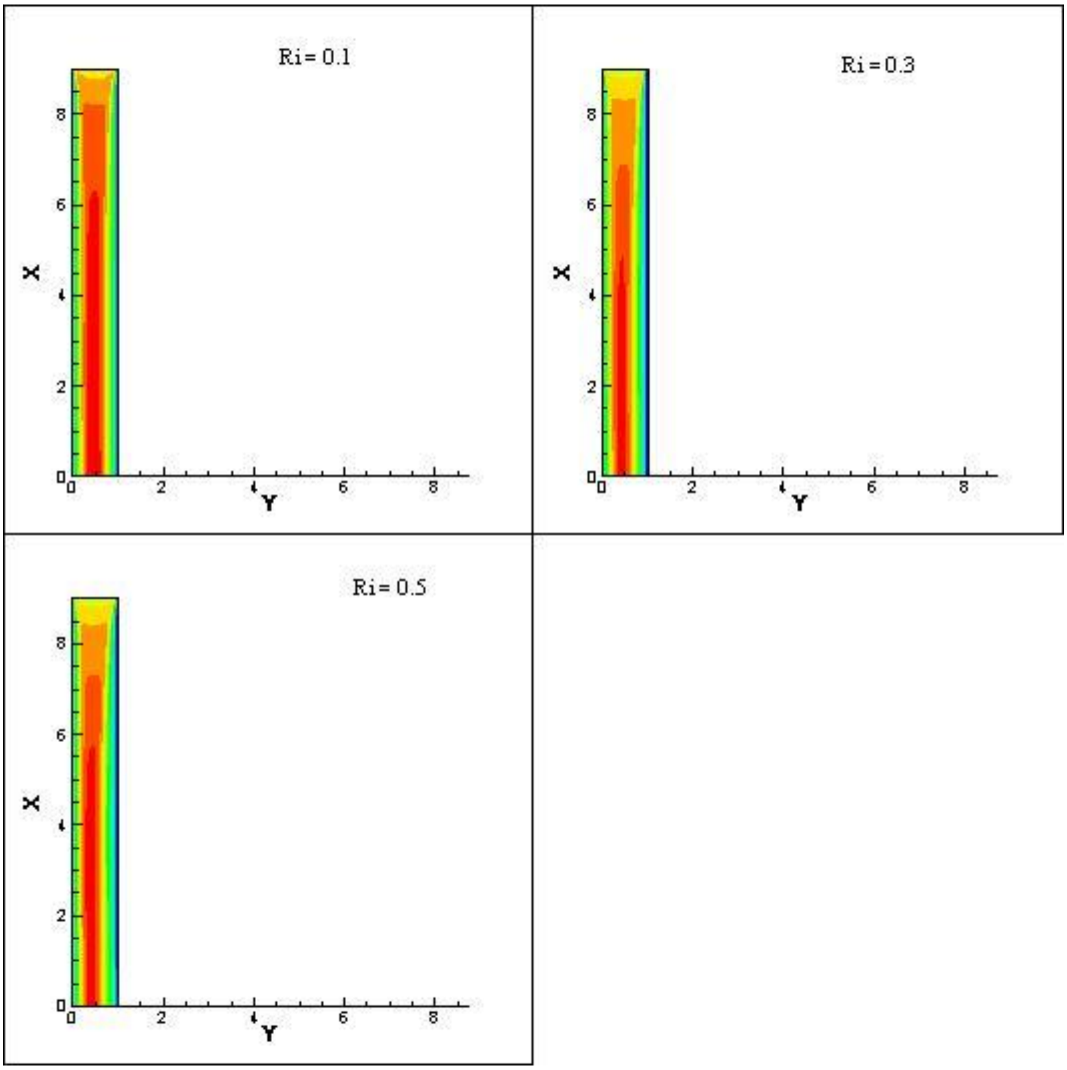


Figure 4.9: Velocity contours for opposing flow

Velocity reduction near the wall due to buoyancy opposes the flow can be seen from figure 4.8. Hence the temperature gradients fall down with respect to increase in buoyancy force in opposing flow, hence reduction in Nusselt number can be seen from Table 2.

Table 2 Average Nusselt numbers for opposing flow

Type of flow	Ri	Nu_{avg}
Forced convection	0	14.39
Opposing flow	0.1	14.22
Opposing flow	0.3	13.84
Opposing flow	0.5	13.51

4.3.3 Horizontal channel:

The buoyancy has significant effect on the temperature distribution in horizontal channel as the buoyancy increases the distribution of temperature gradient decreases, which impairs the rate of heat transfer hence reduce in heat transfer coefficient which can be seen from figure 4.10 and 4.11

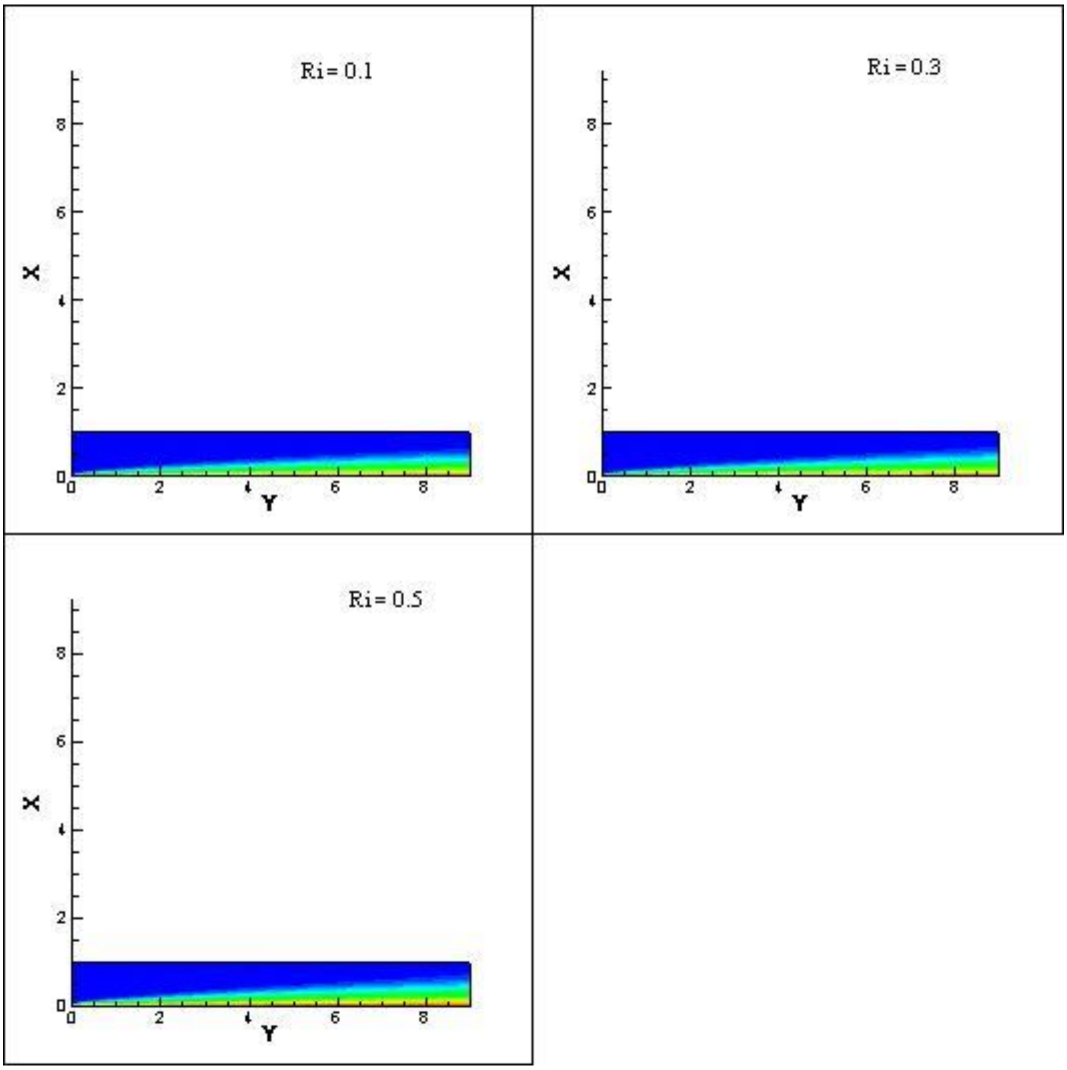


Figure 4.10: Temperature contours for horizontal channel

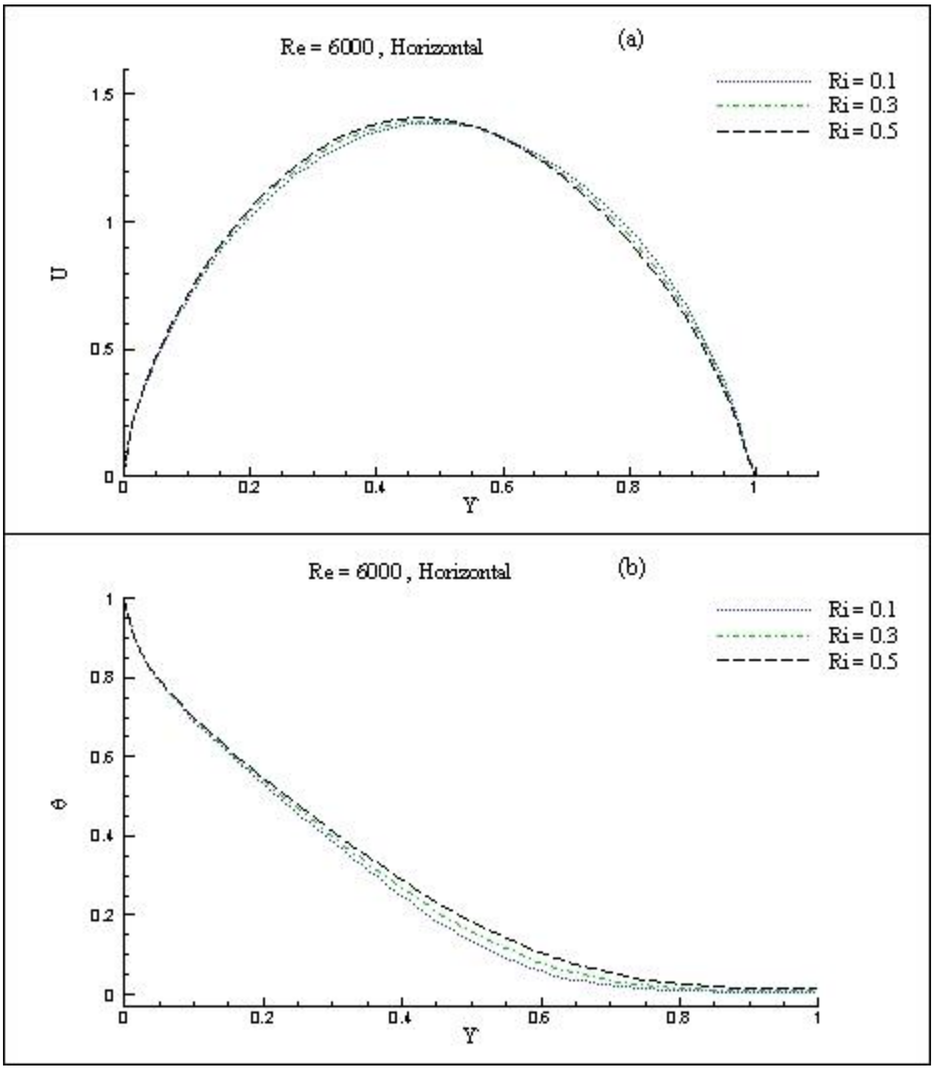


Figure 4.11: temperature and velocity plots for horizontal channel

Table 3 Average Nusselt numbers for horizontal channel

Re	Ri	Nu_{avg}
6000	0	14.39
6000	0.1	14.35
6000	0.3	14.23
6000	0.5	14.11

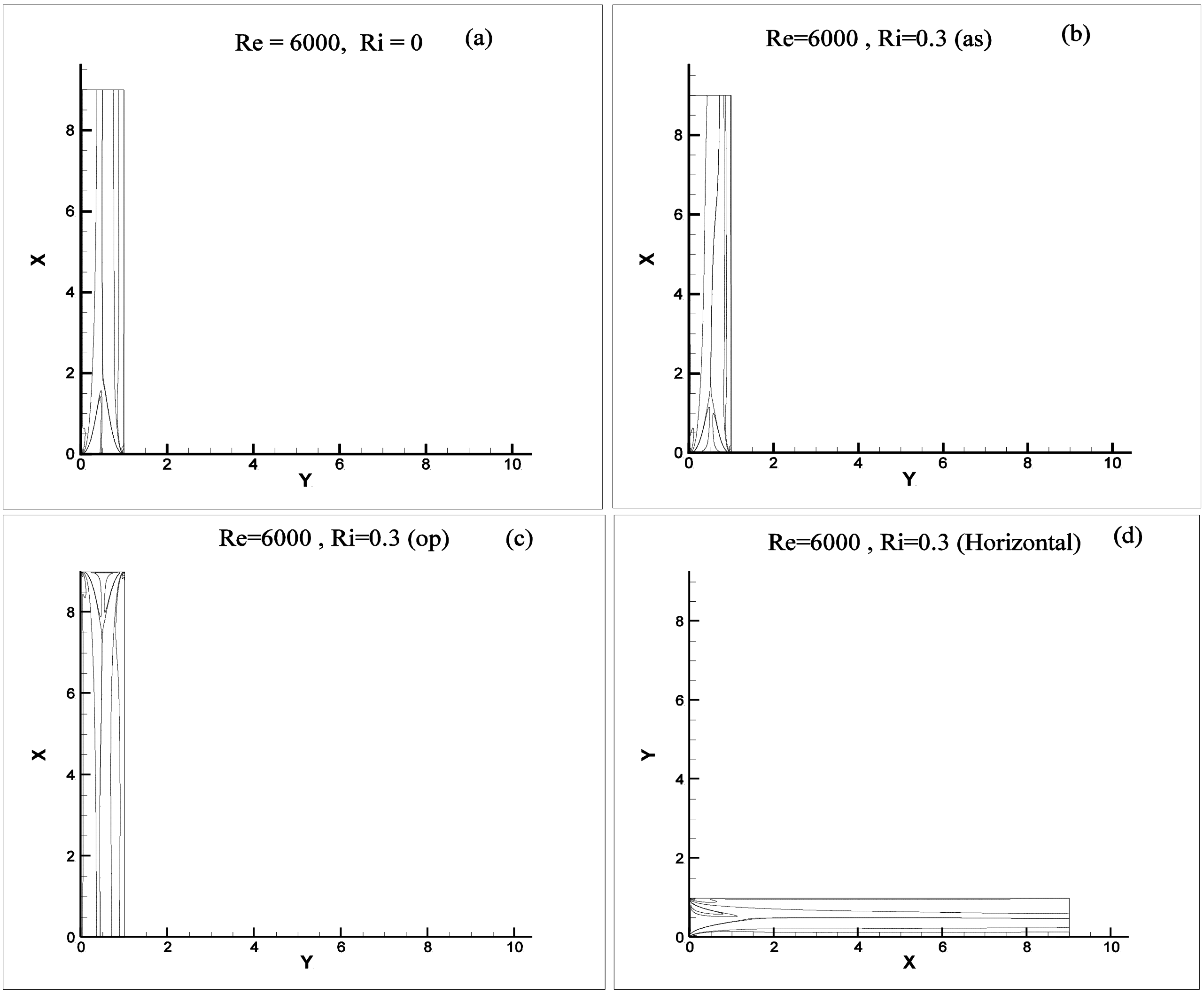


Figure 4.12: vorticity contours in vertical and horizontal channel

Steady state vorticity contours for vertical channel are shown in figure 4.12 for Ri =0.3. Fig.4.12 (a) represents the pure forced convection flow (Ri =0) which depicts the boundary layer formation near the surfaces, the center core vortex and leading edge vortex formations near the inlet of the channel due to turbulence and viscous effects.Fig.4.12 (b) illustrates the reduction of the vortex formation strength in a vertical channel for assisting mixed convection flow with Ri= 0.3 compared to Ri = 0 due to stabilizing the flow by buoyancy. In a vertical channel, for Ri = 0.3 the strength of the leading edge vortex increases for opposing mixed convection flow in comparison with the assisting mixed convection flow since buoyancy destabilizes the flow as shown in Fig.4.12(c). The leading edge vortex strength has increased in a horizontal channel with Ri =0.3 compared to Ri =0 due to buoyancy as shown in Fig.4.12 (d). the temperature distribution for vertical and horizontal channel at Ri=0.3 is shown in figure 4.13.the buoyancy has significant effect on mixed convection flow for horizontal channel and vertical channel as well compared to forced convection.

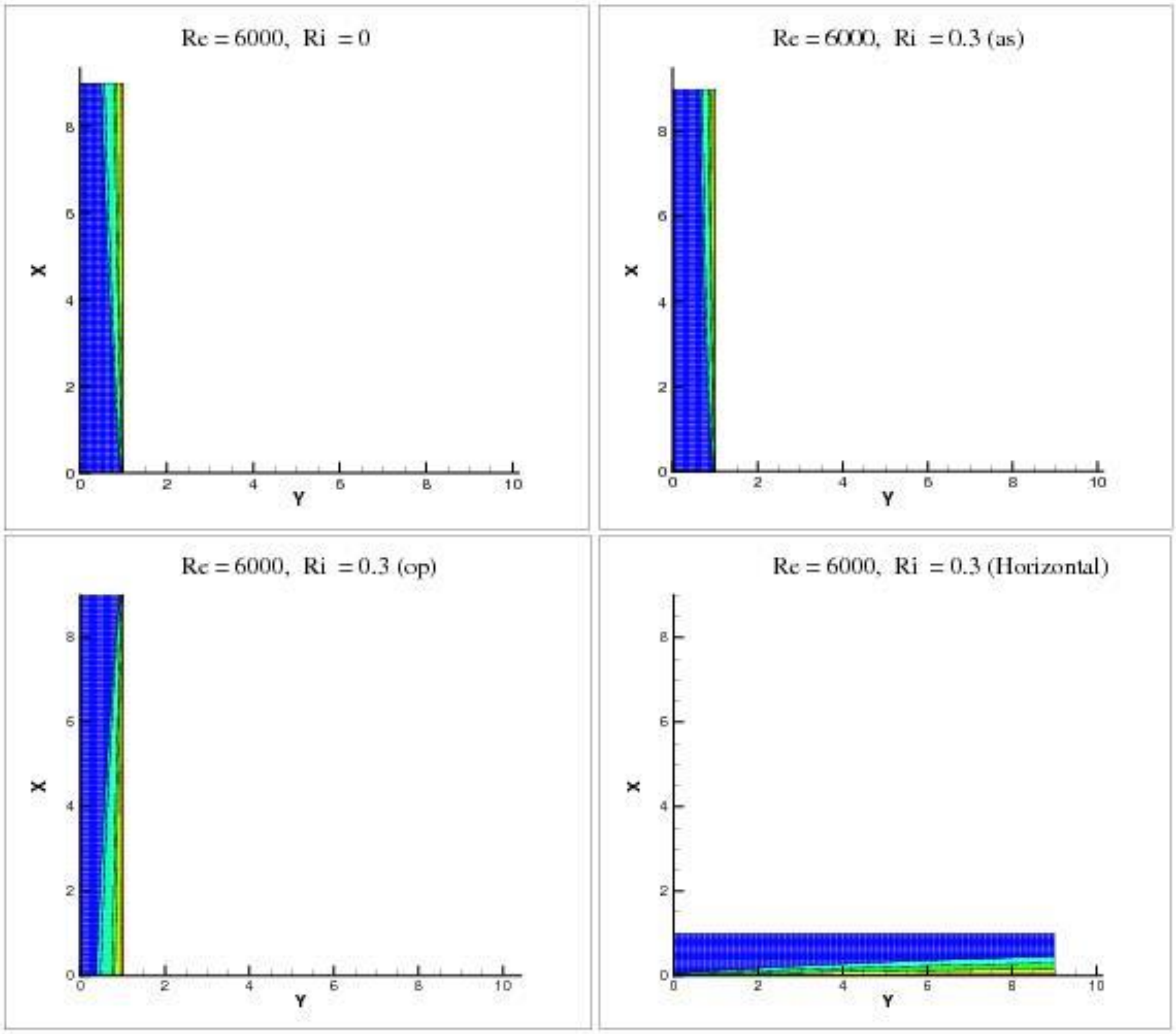


Figure 4.13: steady state temperature contours at Ri =0.3 for different orientation

Chapter 5

Results of moving plate in a channel

Turbulence mixed convection flow through channel with considering the motion of plate inside the channel has been studied numerically. The plate velocity is negative when the moving direction of plate is opposite to flow direction and vice versa. Problem has been solved by considering the axi symmetric conditions. An extra domain has been taken at inlet to get the qualitative characteristics near the leading edge of the moving plate. Flow characteristics has been reported at Ri = 0 for various velocities.

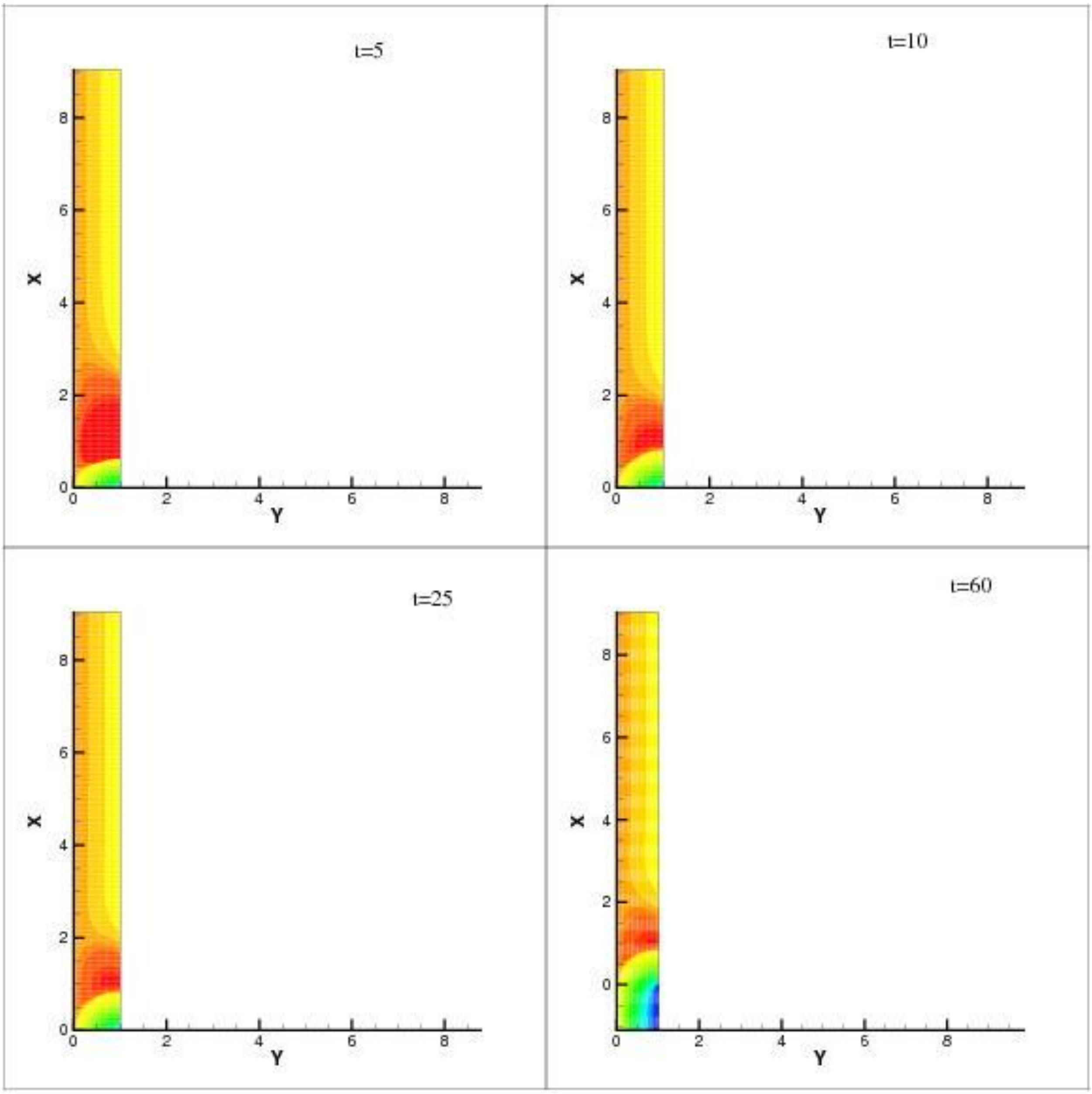


Figure 5.1: stream line contours for moving plate velocity $U = 0.01$

The positive velocity of plate causes the flow to rotate near the plate at inlet which can be observed from figure 5.1. Hence the rate of heat transfer is more at this location. The temperature contour is shown in figure 5.2 which depicts the temperature growth with respect to time.

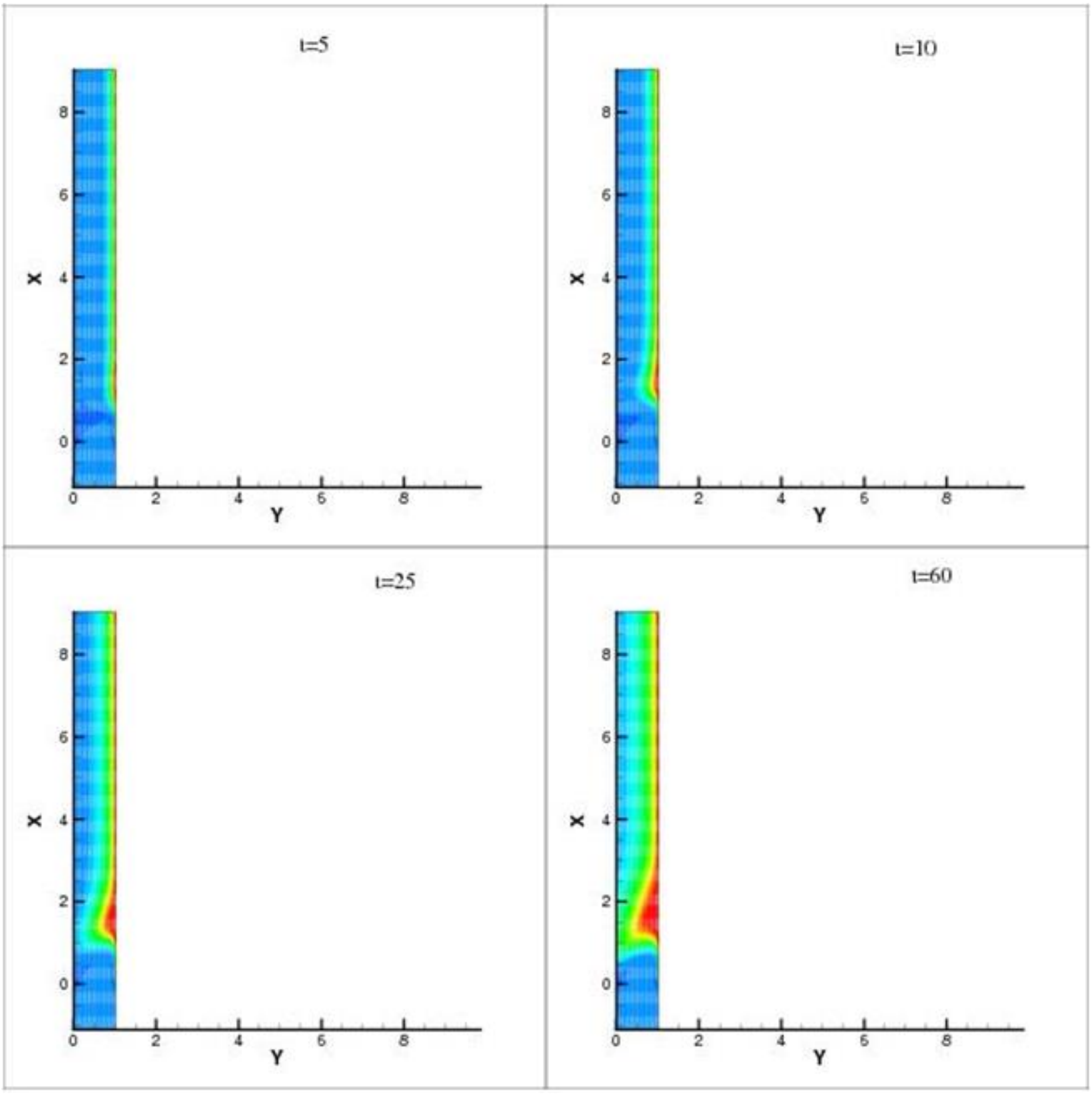


Figure 5.2: Temperature contours for moving plate velocity $U = 0.01$

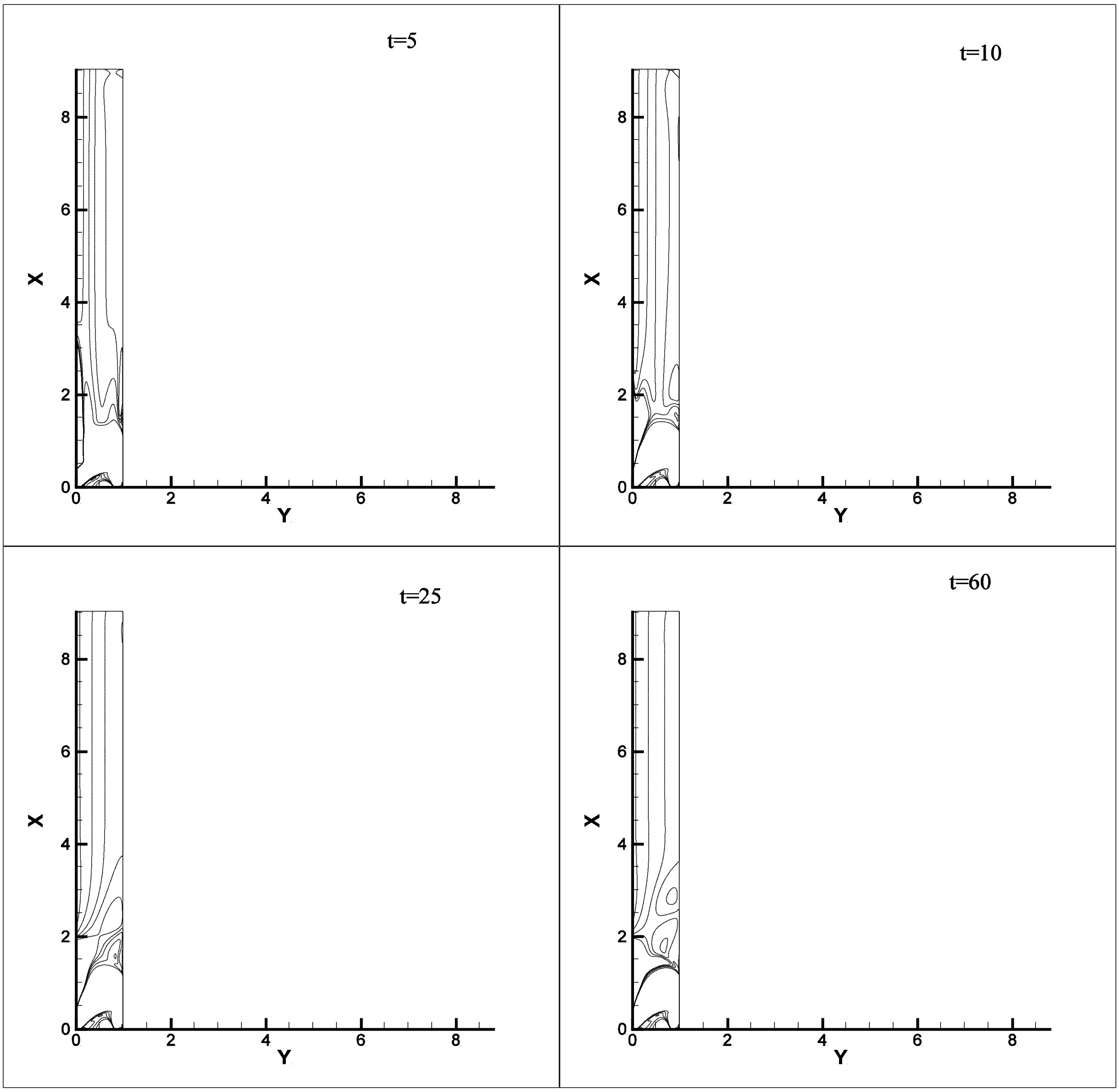


Figure 5.3: Vorticity contours for moving plate velocity $U = 0.01$

Figure 5.3 illustrates the evolution of vorticity contours for the case of plate moving at a constant velocity of U= 0.01. At the inlet, small vertical structures formed near the moving plate due to motion of the plate. This causes to increase the local convective heat transfer coefficient near the inlet but the overall convective heat transfer coefficient decreases compared to the case of stationary plate as shown table 4.

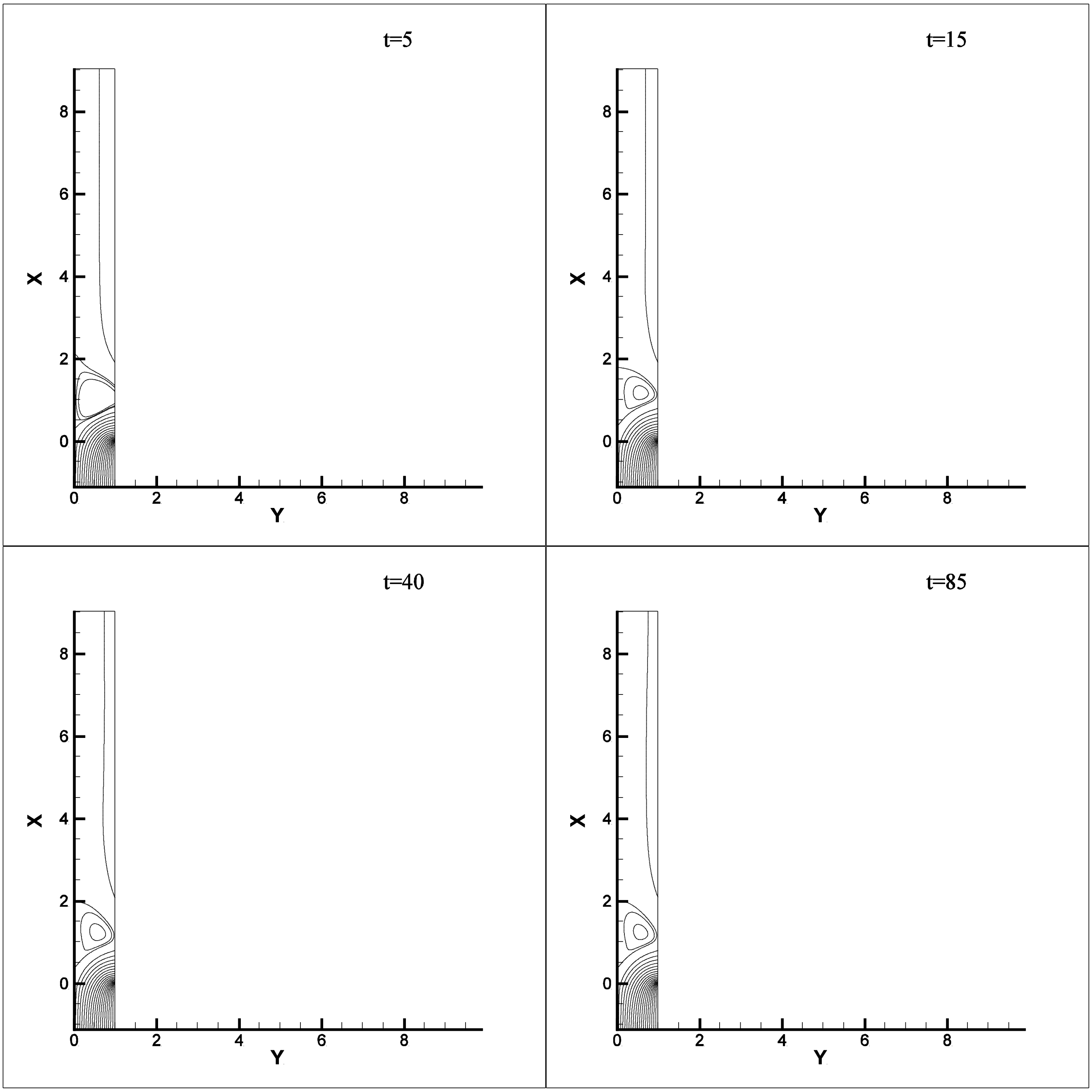


Figure 5.4: Stream line contours for moving plate velocity $U = 0.1$

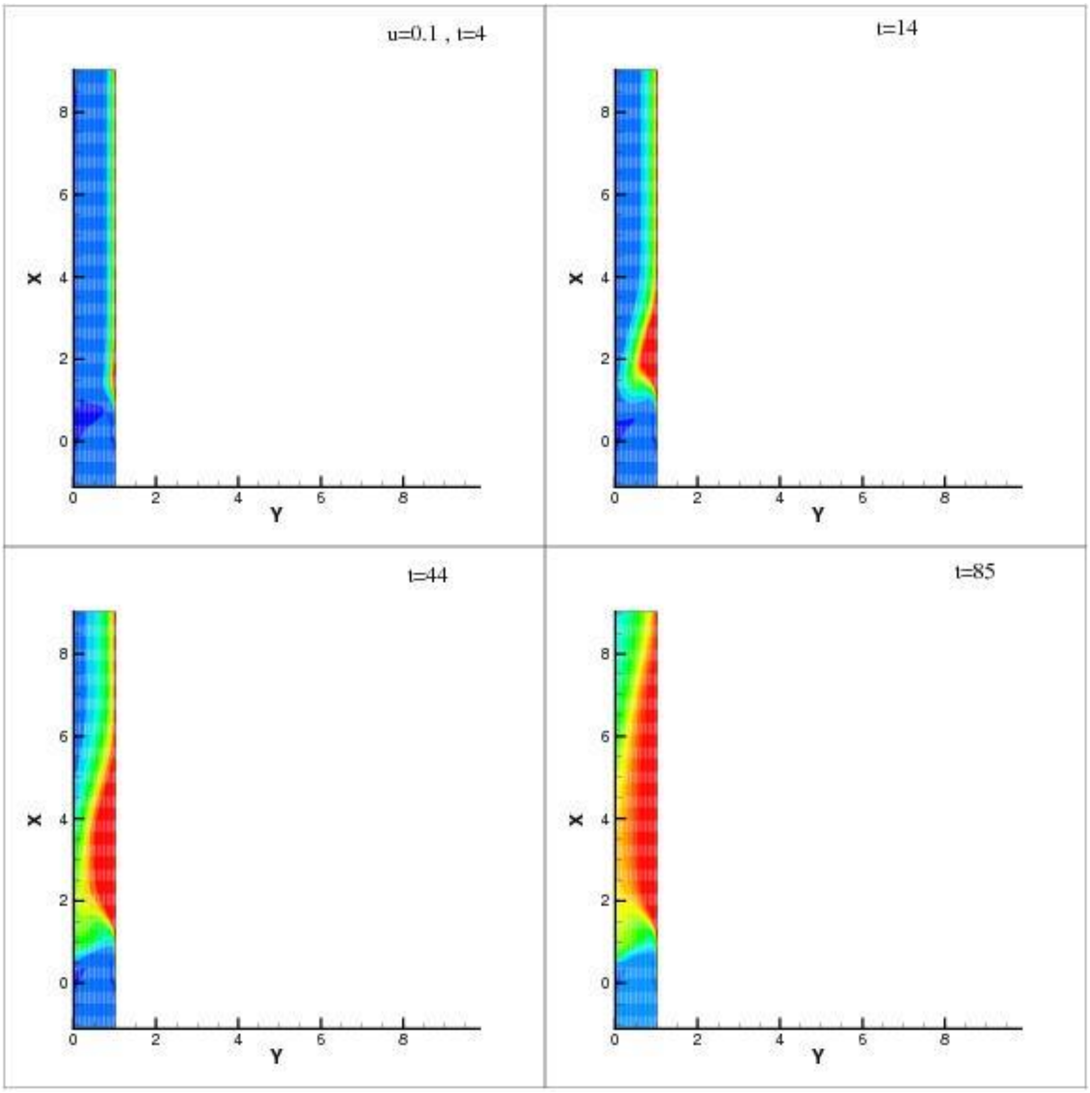


Figure 5.5 : Temperature contours for moving plate velocity $U = 0.1$

As plate velocity increases from 0.01 to 0.1 there is huge variation in stream line pattern can be seen from figure 5.4, also recirculation increases near the plate at inlet hence the rate of heat transfer increases at that location, boundary layer growth improved and over all heat transfer coefficient decreases .figure 5.6 shows how vorticity develops with respect time and its effect on recirculation near the moving plate.

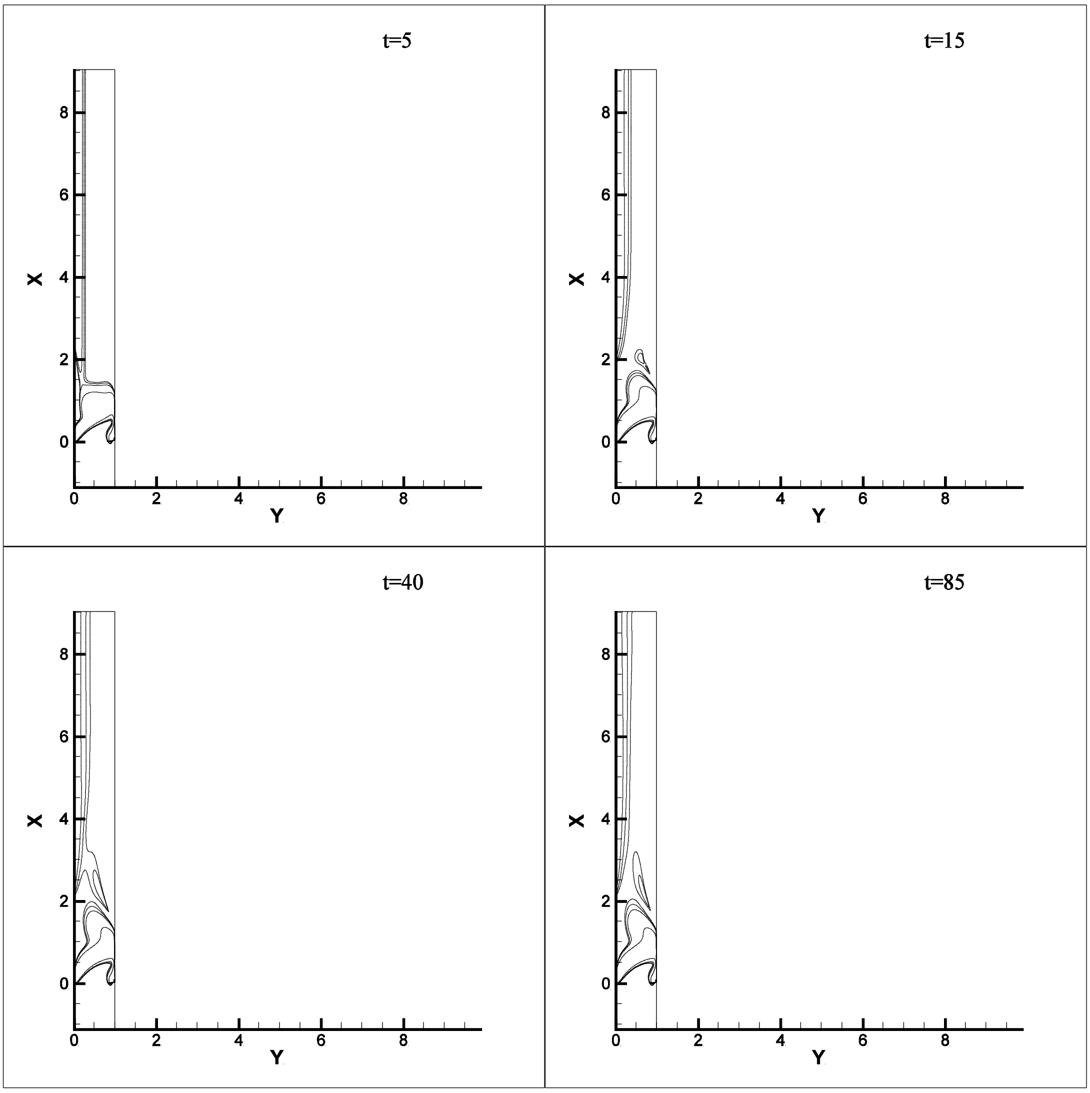


Figure 5.6: Vorticity contours for moving plate velocity $U = 0.1$

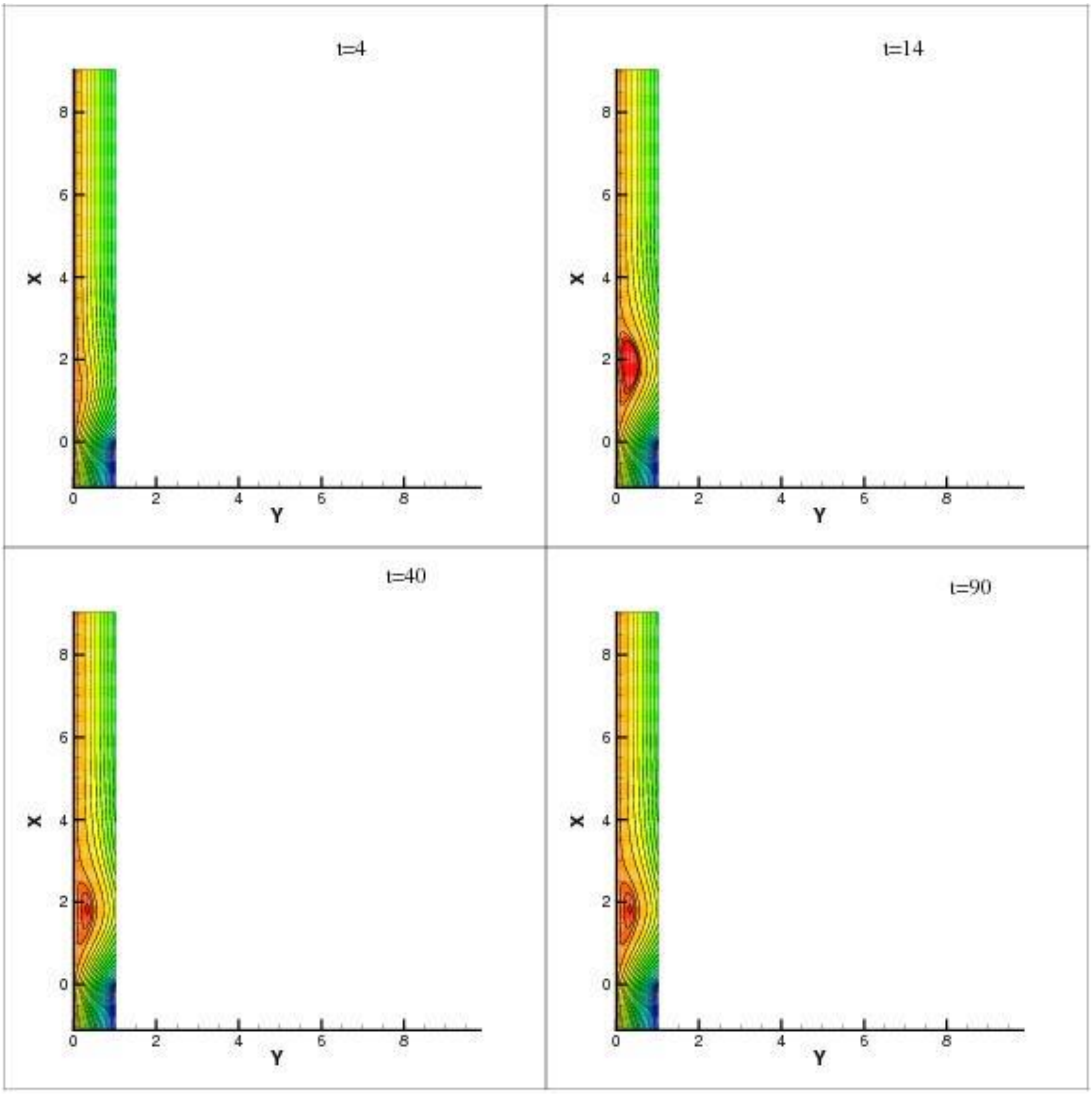


Figure 5.7: stream line contours for plate velocity $U = 0.5$

Figure 5.7 show the shifting of recirculation zone from right to towards left side as plate velocity increases. Hence the rate of heat transfer near this zone little decreases as compared to either plate velocity 0.01 or plate velocity 0.1 and over all heat transfer coefficient increases because of intensity of rotational flow is more near the wall as compared to case of plate velocity 0.1.the average Nusselt number values are presented in table 4.

The steady state temperature growth shown in figure 5.8 is more for plate velocity of 0.5 compared to plate velocity of 0.1 because of vorticty near the wall is more.

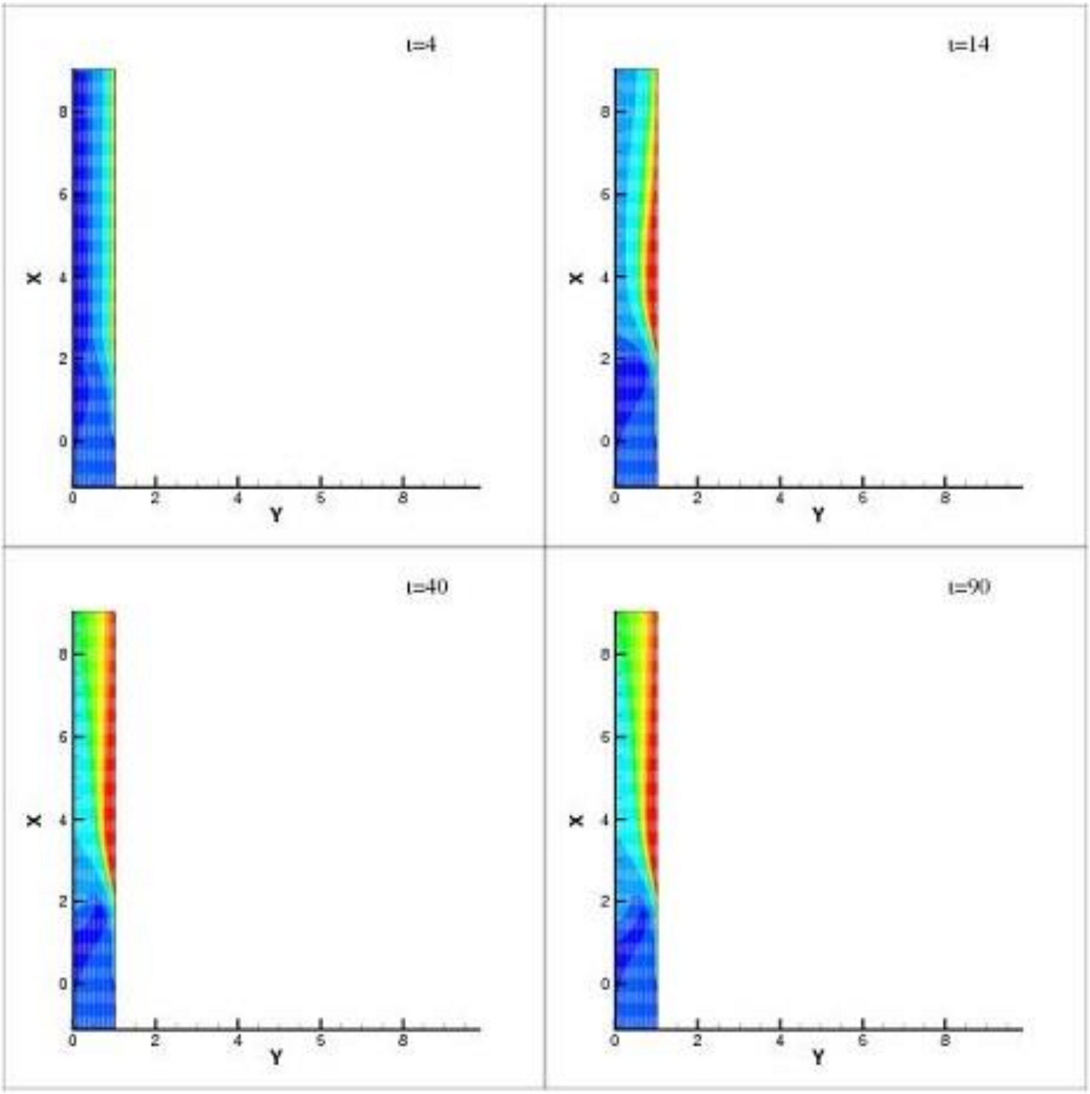


Figure 5.8 : temperature contours for plate velocity $U = 0.5$

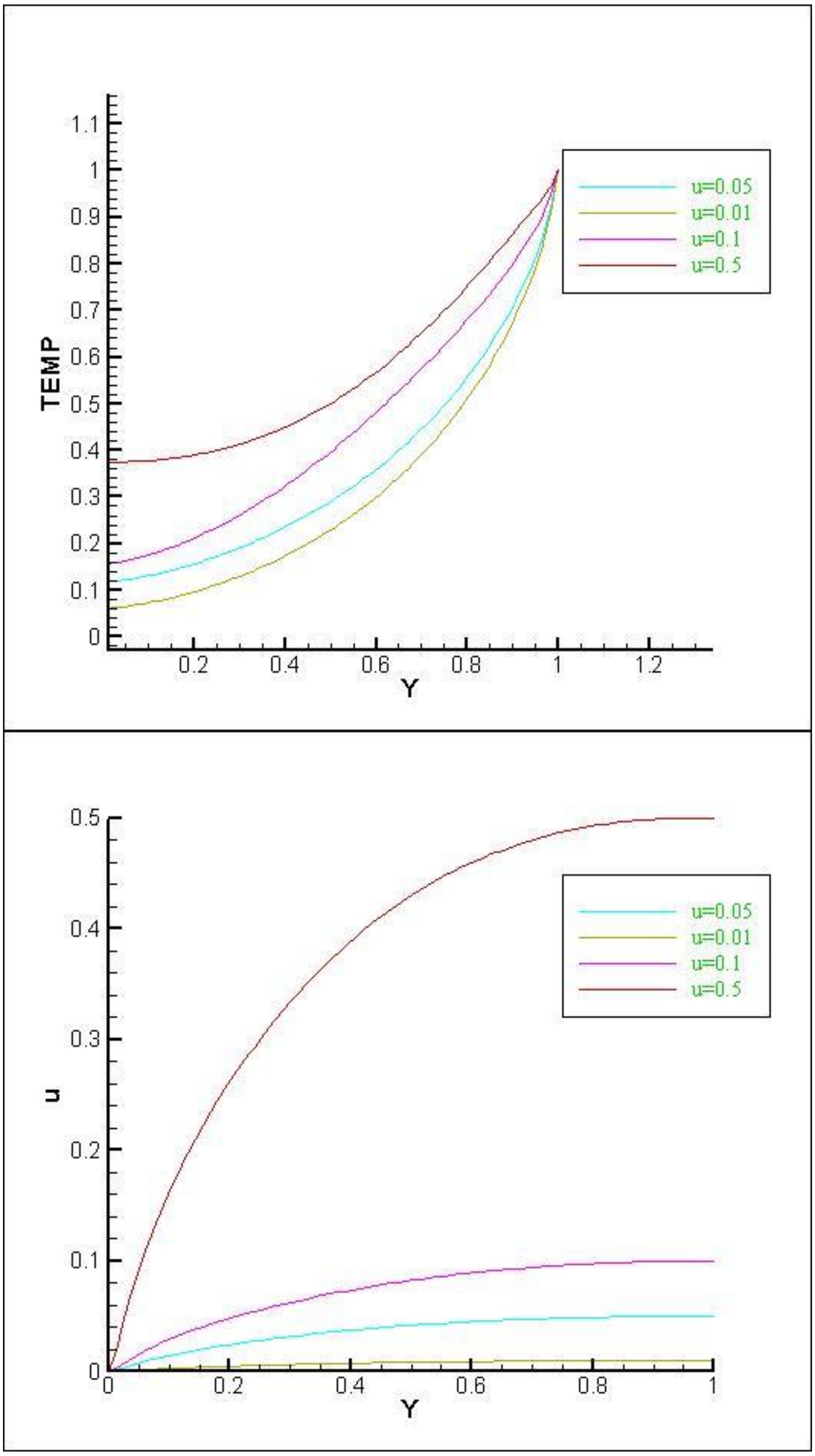


Figure 5.9: velocity and temperature plots for moving plate positive velocities

The velocity and temperature profiles at the outlet of the channel are shown in Fig. 5.9. The velocity and temperature has increased with increase in plate velocity. The velocity profile is different compared to the case of stationary plate.

Table 4 Average Nusselt number for moving plate (positive velocity)

Plate velocity	Ri	Nu_{avg}
0	0	14.39
0.01	0	14.007
0.05	0	12.07
0.1	0	10.813
0.5	0	18.256
0.5	0.3	25.59

Table 5 Average Nusselt number for moving plate (negative velocity)

Plate velocity	Ri	Nu_{avg}
0	0	14.39
-0.01	0	14.097
-0.05	0	15.238
-0.1	0	17.113
-0.5	0	13.22

The local nusselt number variation shown in figures for moving plate cases. The effect of moving plate velocity magnitude and its direction is very significant over the convective heat transfer, as velocity of plate increases from 0 to 0.01, 0.05, 0.1 and 0.5 the recirculation zone observed near to moving plate at inlet and this bubble growth increases as velocity increases. Overall convective heat transfer decreases from velocity of plate 0 to 0.1even though this bubble increases local convective heat transfer at the location of bubble in contrast to moving plate velocity 0.5, because the convective heat transfer drop after the location of bubble to outlet is more than the increment of convective heat transfer at bubble location, this phenomenon can be seen from figure 5.10

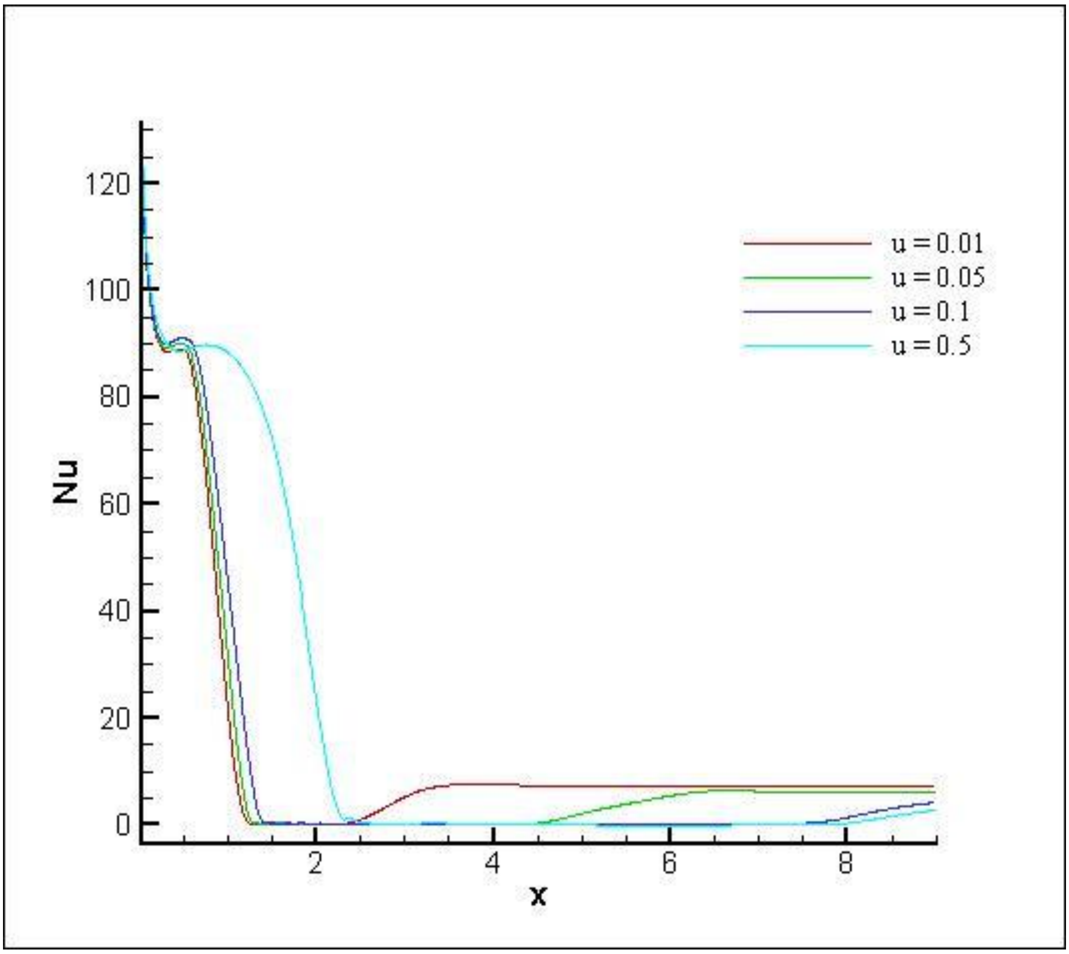


Figure 5.10: Local Nusselt number variation for positive velocities of moving plate

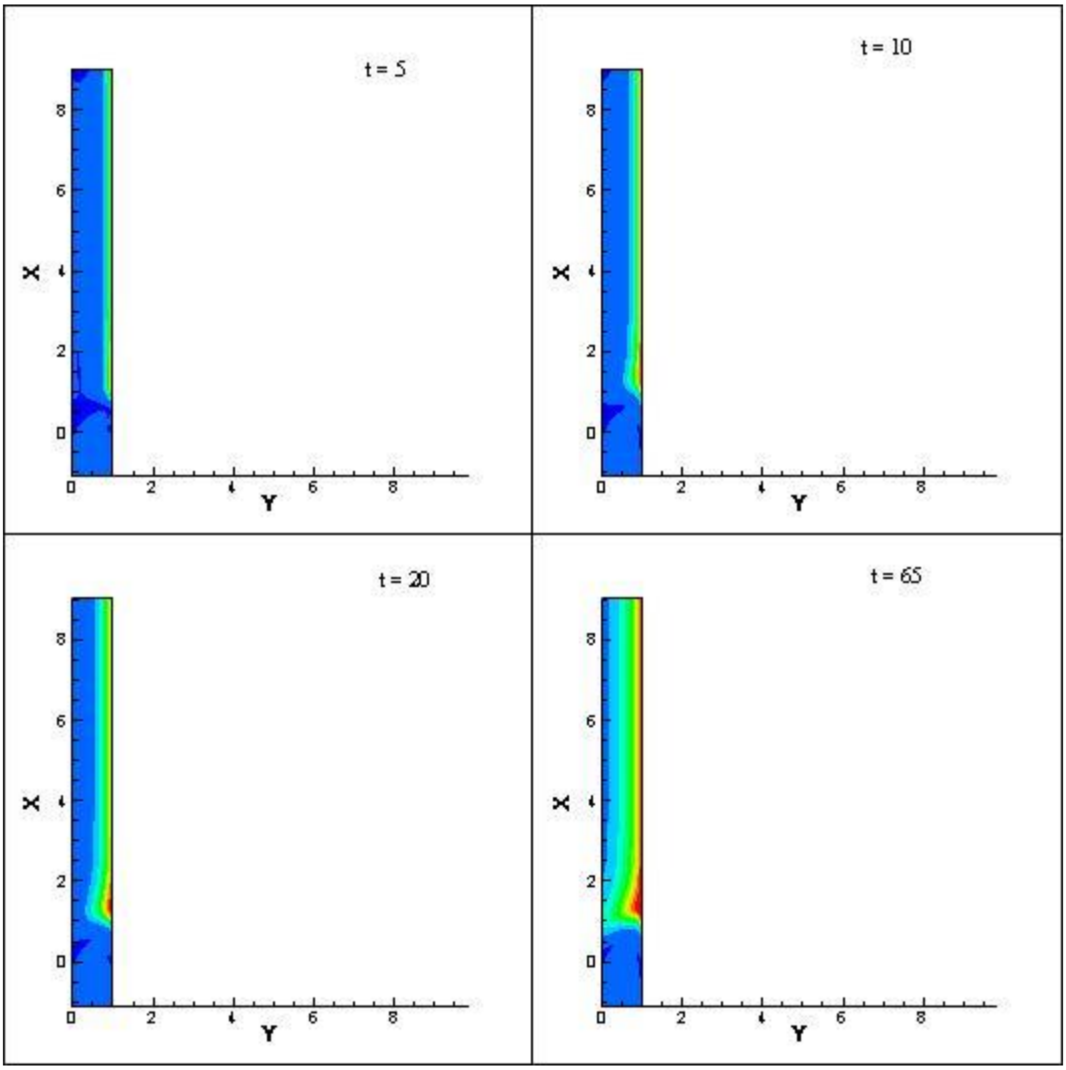


Figure 5.11: temperature contours for plate velocity $U = -0.01$

When the plate velocity becomes negative i.e. the direction of the plate motion opposite to the direction of the flow. The fluid attracted towards moving plate as it passes at certain cross section of the channel. The reversal flow takes place at out let when flow reaches steady state. The further incremental velocity causes the reversal flow towards to inlet which can be seen from velocity plots.

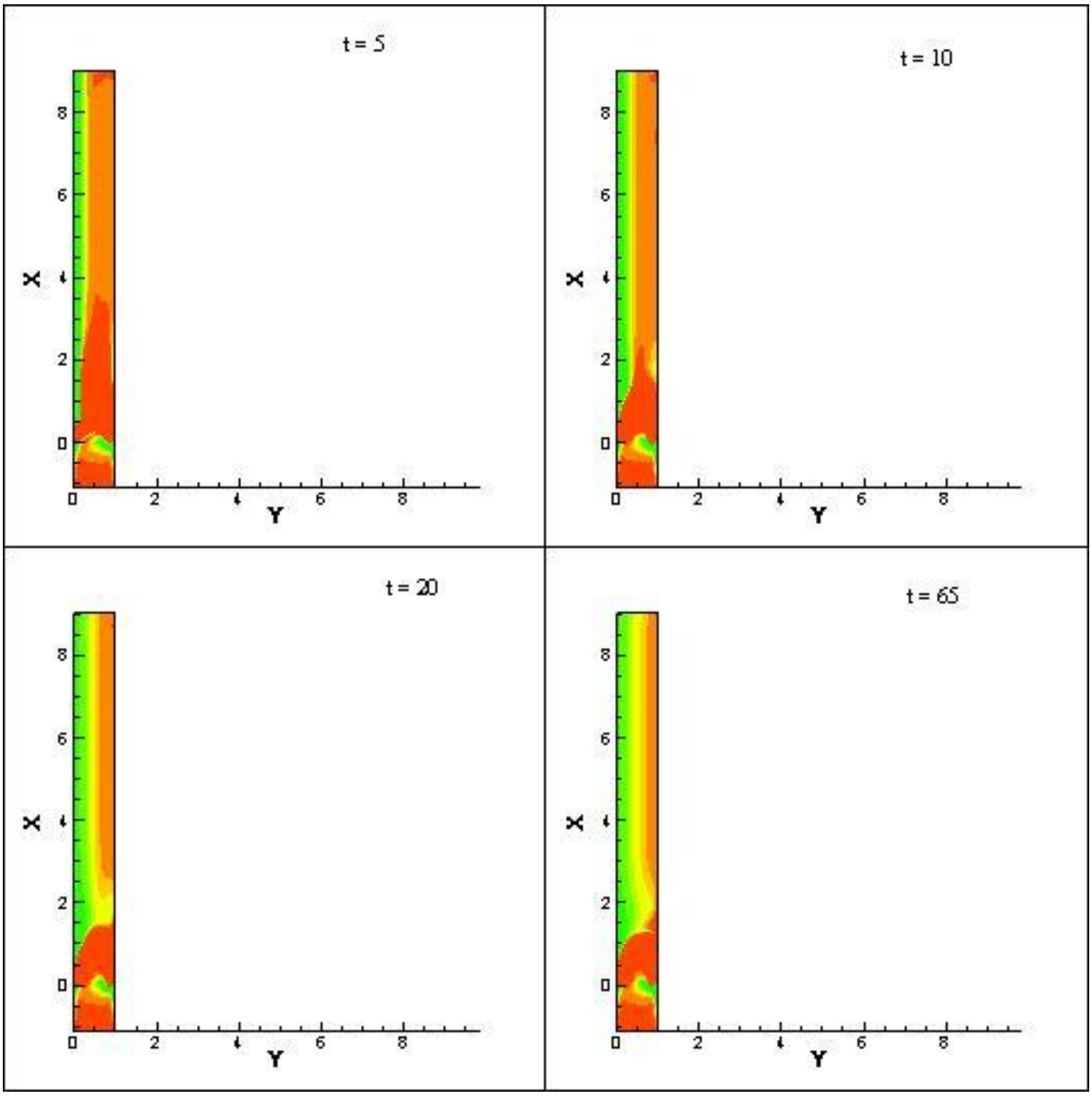


Figure: 5.12 vorticity contours for plate velocity $U = -0.01$

The intensity of rotational components near to wall decreases as plate velocity changes from 0 to -0.01, which leads to decrease of heat transfer rate. The evolution of temperature contours are shown in figure 5.12.

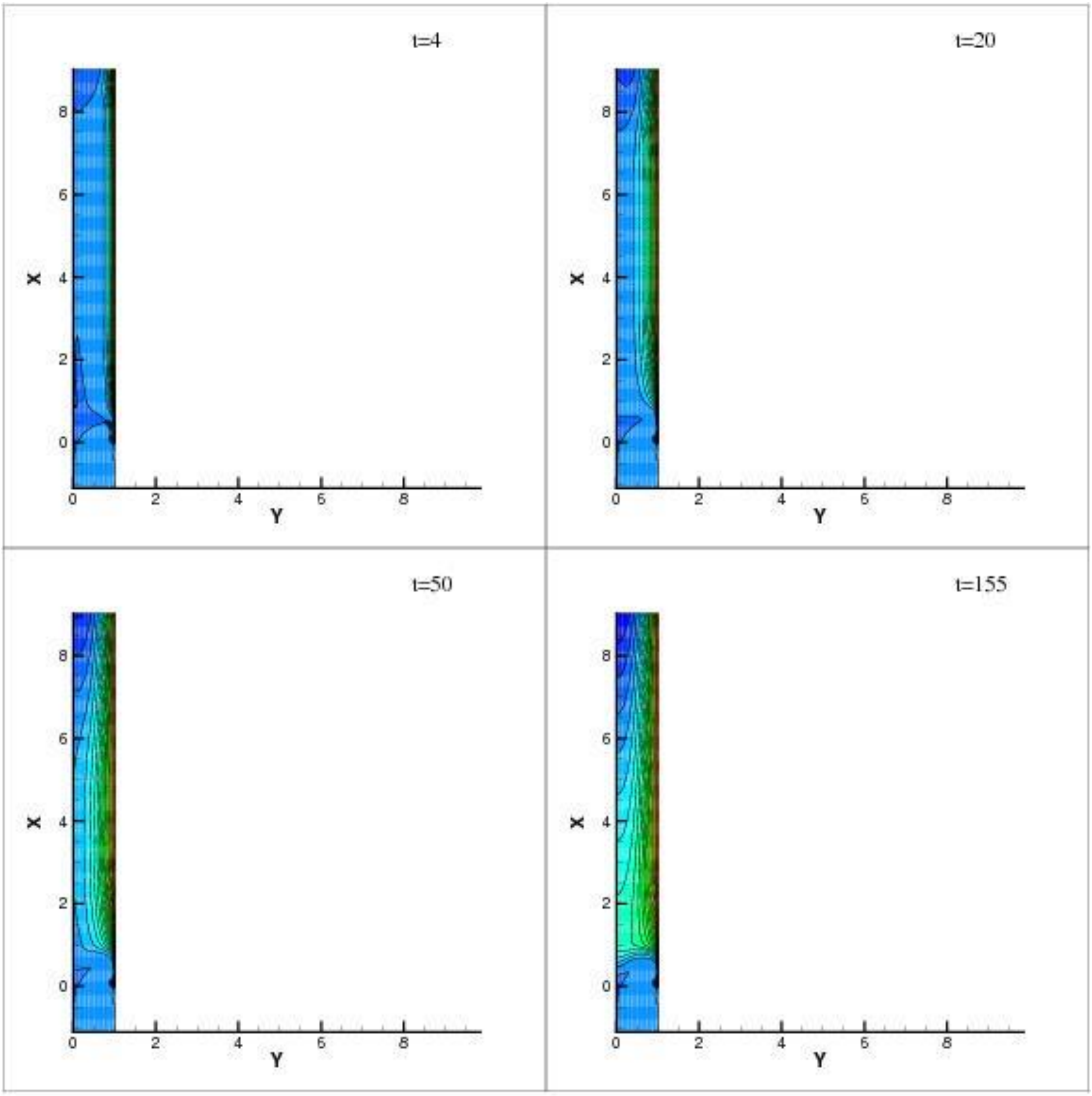


Figure: 5.13 Temperature contours for plate velocity $U = -0.1$

As the velocity of plate increases from -0.01 to -0.1,the rate of heat transfer increses due to rigorous mixing occurring near the moving plate. The temperature boundary layer growth is shown in figure 5.13.

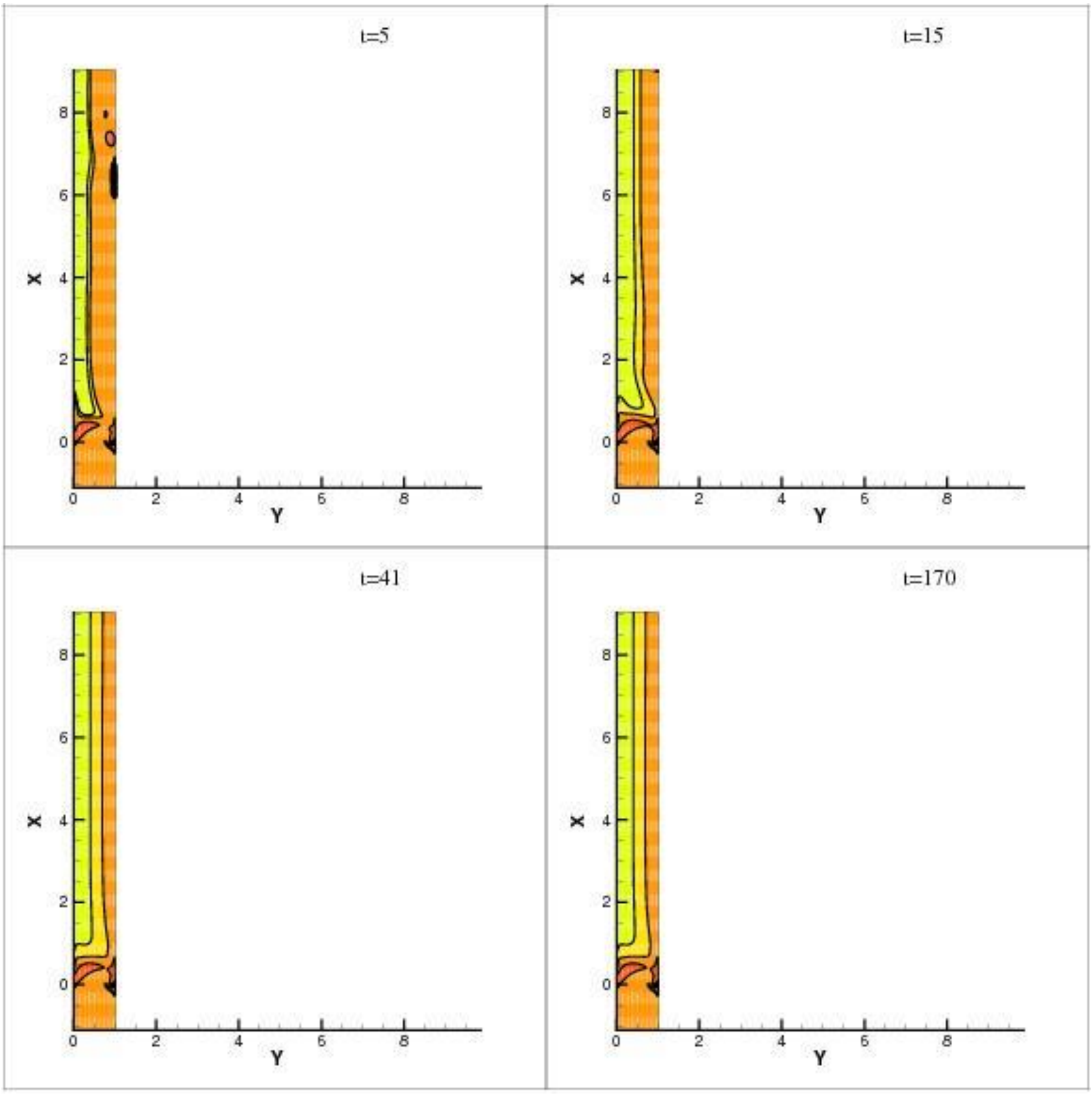


Figure 5.14: vorticity contours for plate velocity $U = -0.5$

When plate velocity further increases to -0.5 the overall heat transfer coefficient decreases because of the intensity of reversal flow across the section increases and fluid flows along with plate ,whatever the heat transferred to the adjacent layer of the plate is carried away with the fluid along the plate, result in decrement of heat transfer. Figure 5.15 shows the temperature growth. The average nusselt number values shown in table 5.

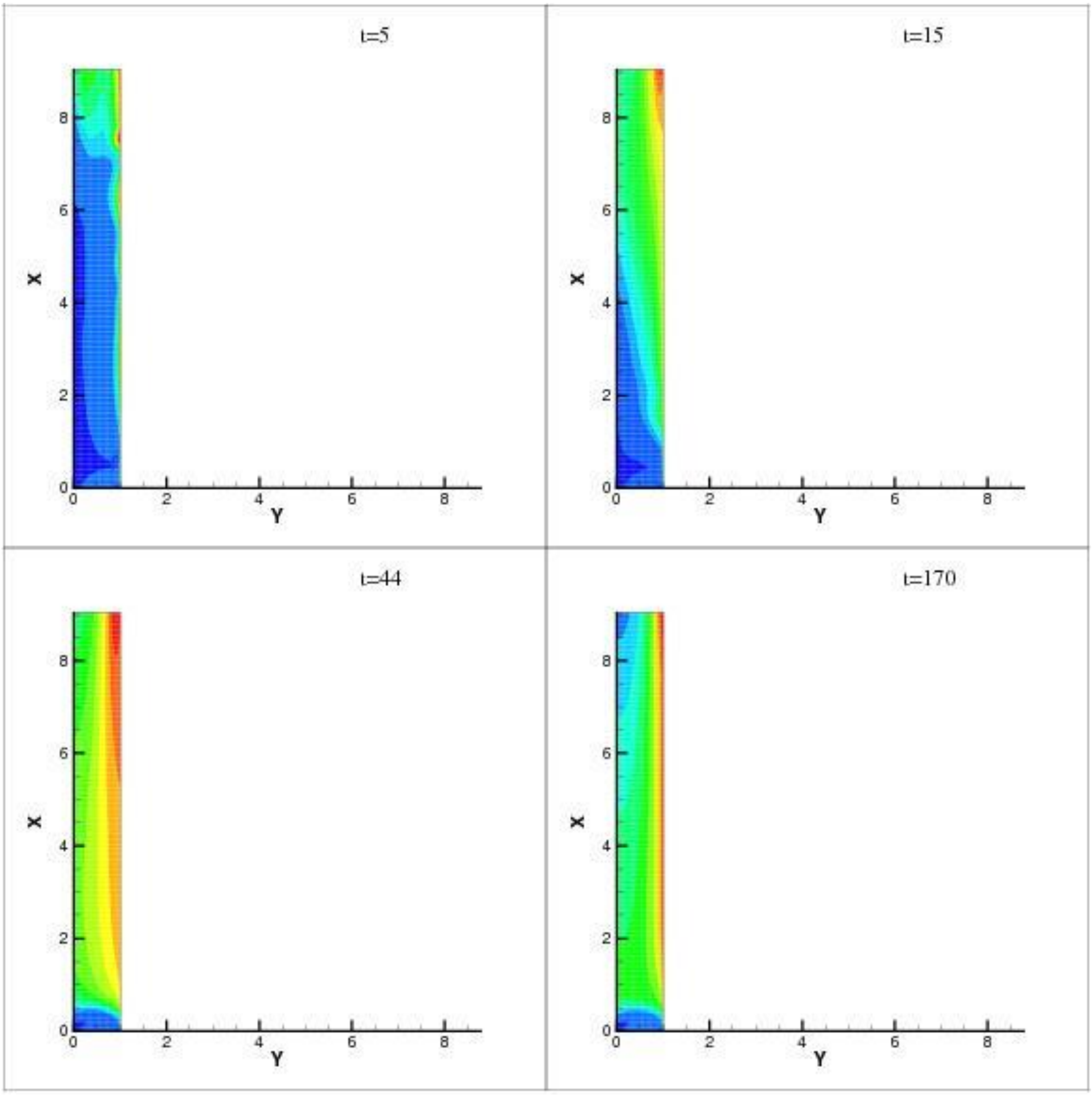
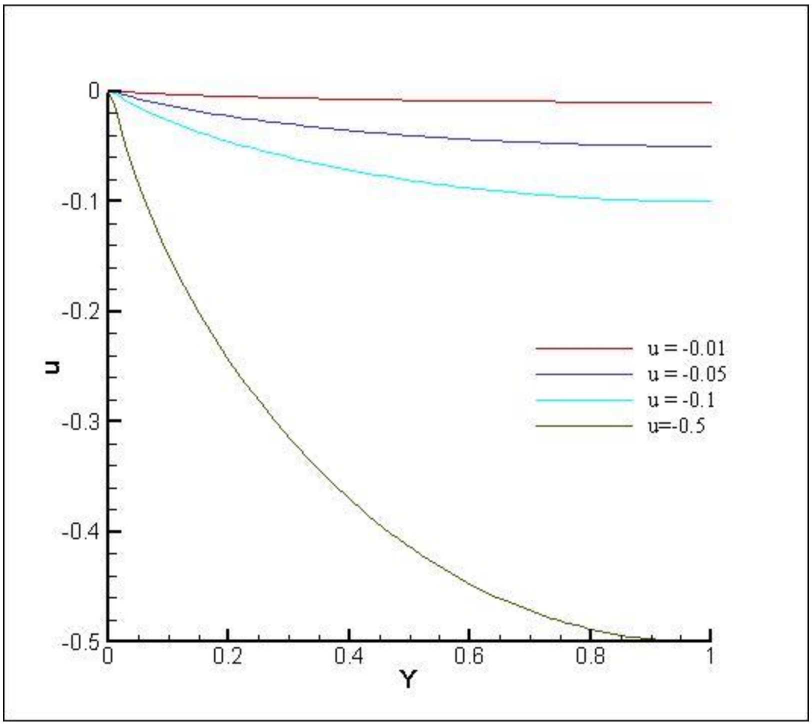


Figure 5.15: Temperature contours for plate velocity $U = -0.5$



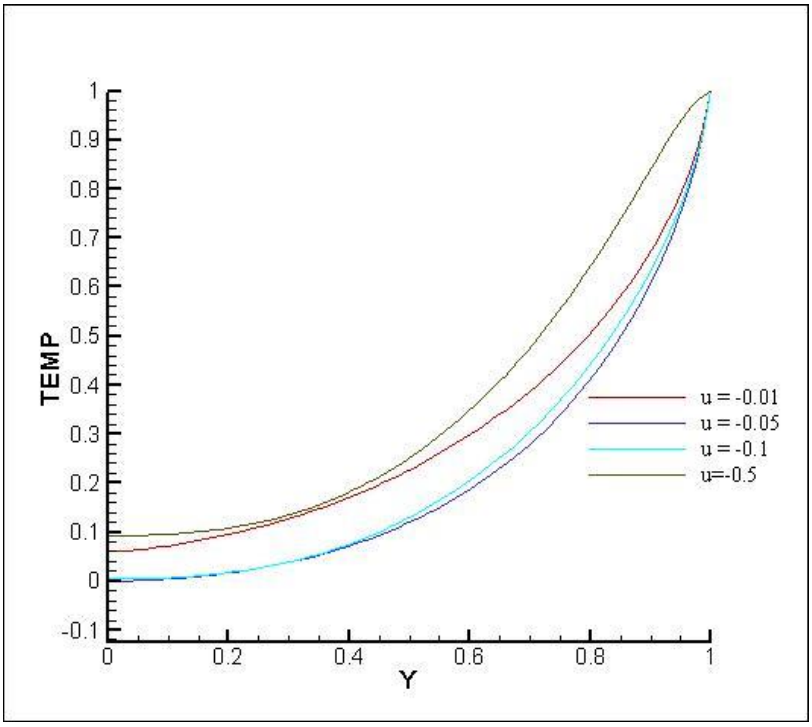


Figure 5.16: Velocity and Temperature plots for plate velocity $U = -0.5$

The velocity and temperature profiles at the outlet of the channel are shown in Fig. 5.16. The velocity has increased with increase in plate velocity.

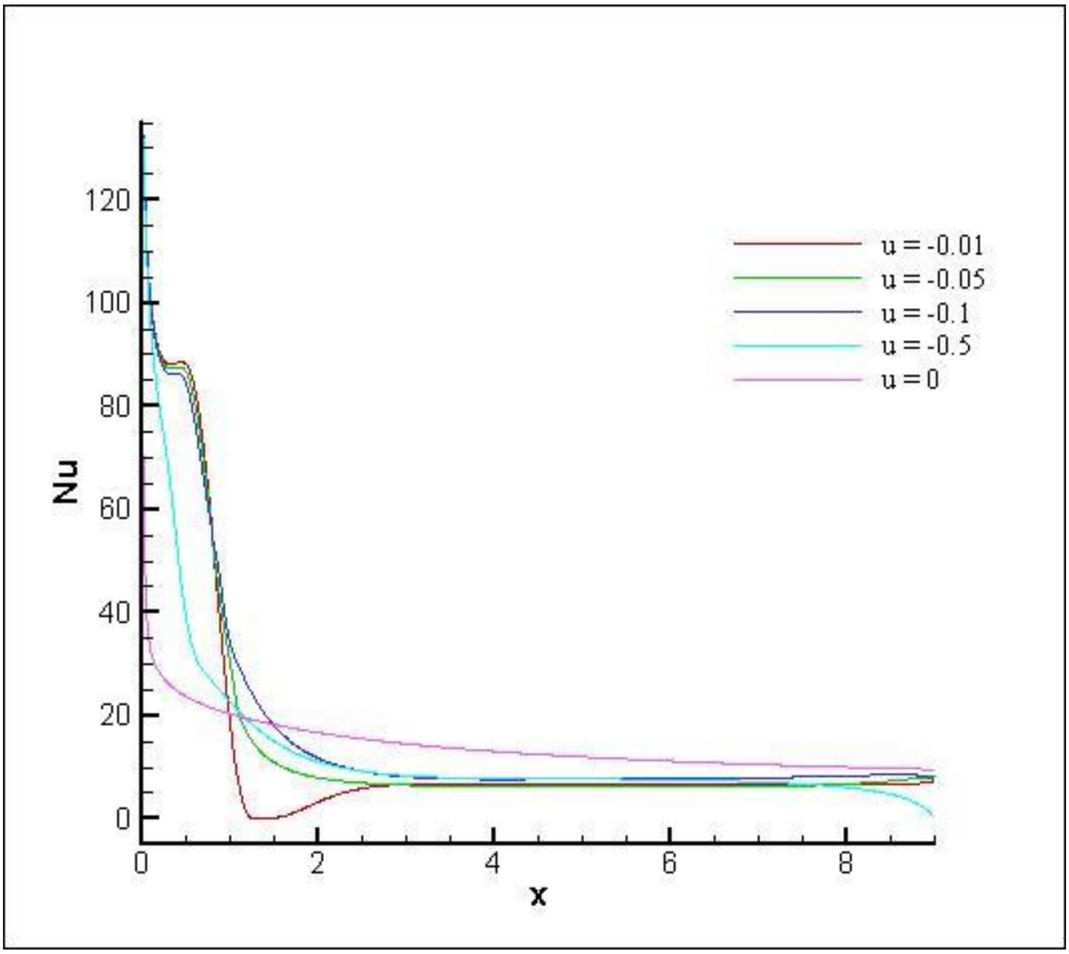


Figure 5.17: Local Nusselt number variation for negative velocities of moving plate

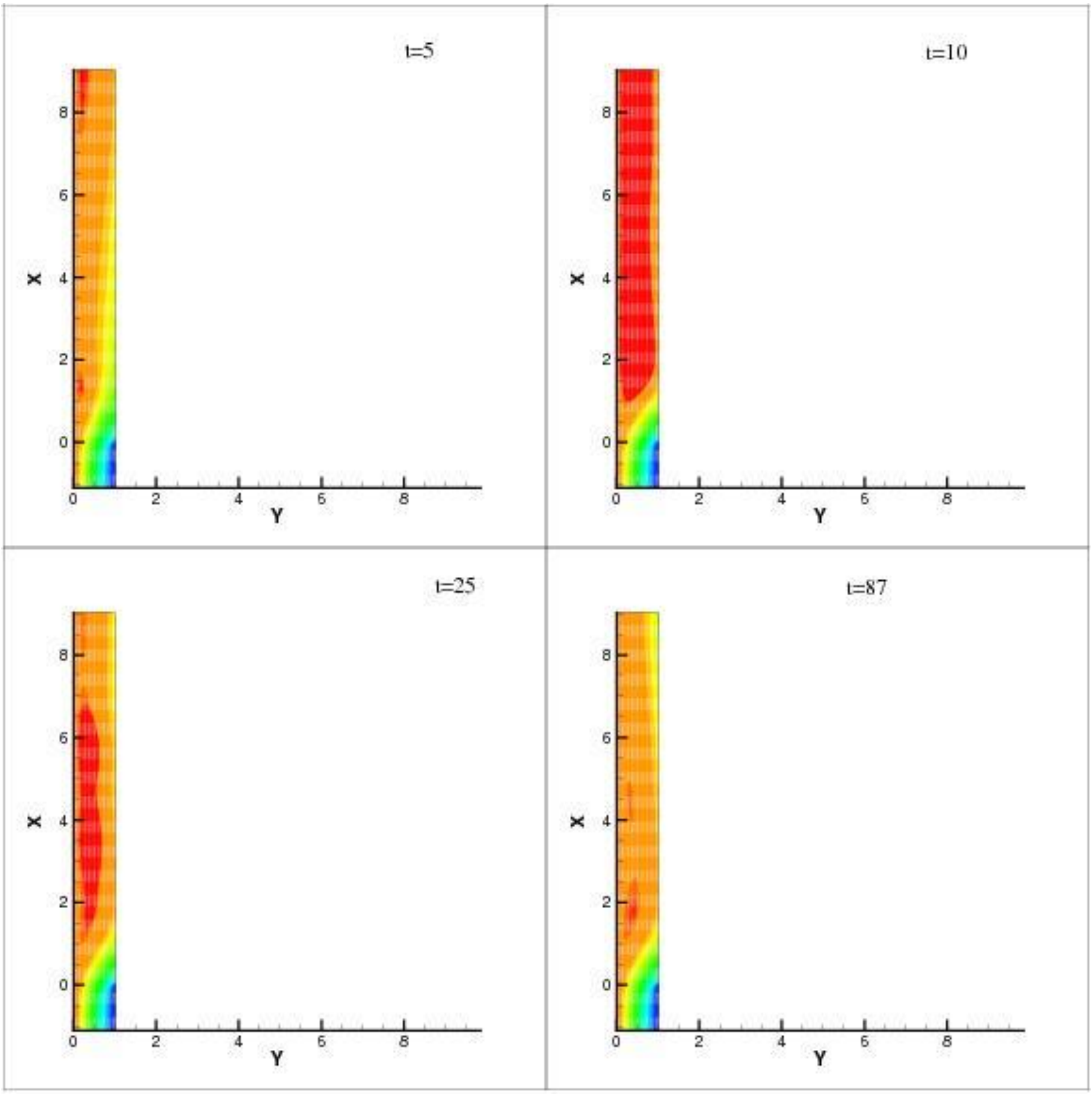


Figure 5.18: Stream line contours for moving plate velocity of $U = 0.5$ at Ri=0.3

The buoyancy effect including moving plate causes more heat transfer, the buoyancy cause one more recirculation zone which can be seen from figure 5.18, the two circular zones changes the temperature distribution as compared to the case without buoyancy can be seen from 5.19.

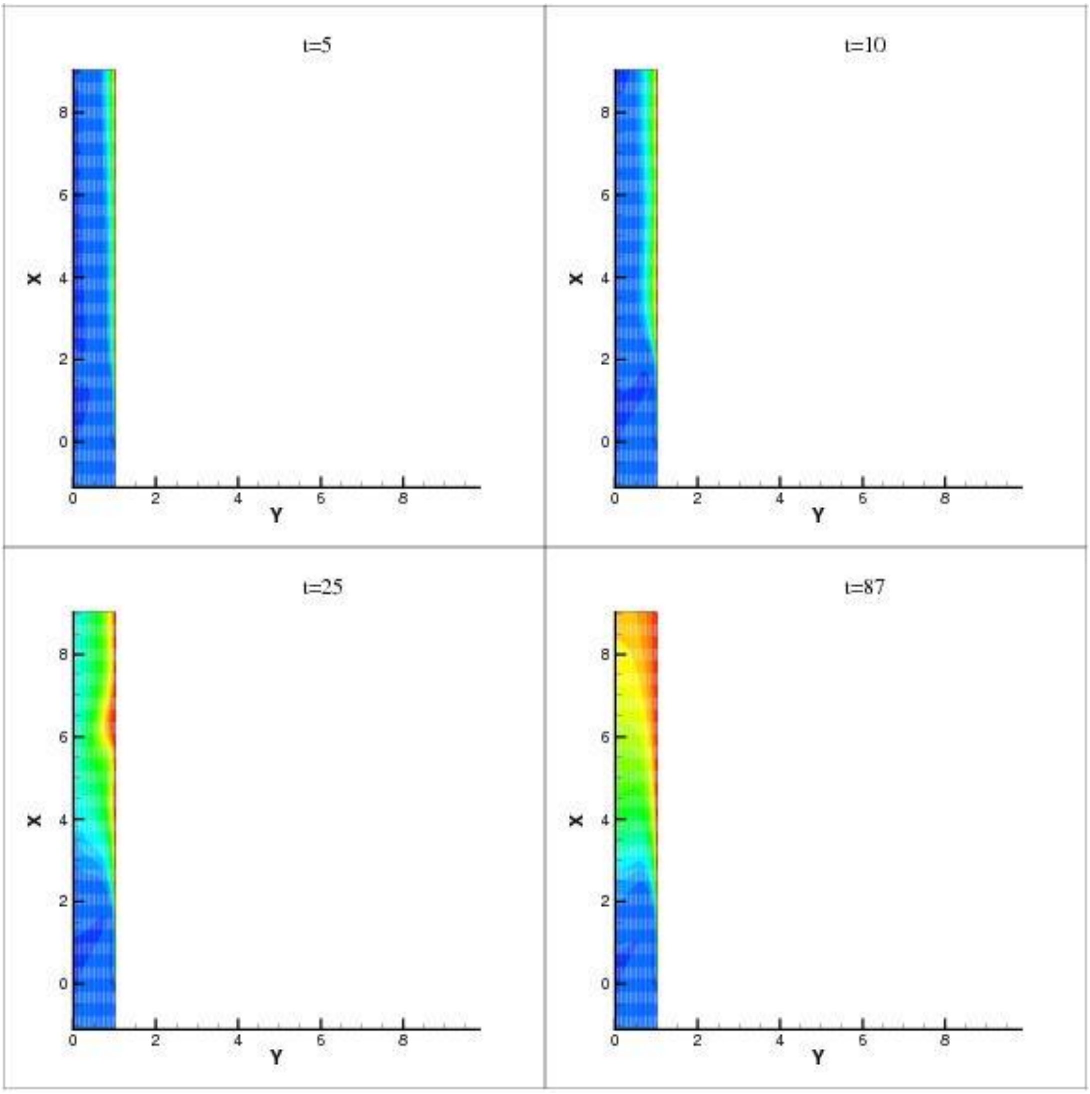


Figure 5.19: Temperature contours for moving plate velocity of $U = 0.5$ at Ri=0.3

Chapter 6

Conclusions

The effect of buoyancy on flow and heat transfer characteristics of assisting and opposing mixed convection flows in a vertical channel and mixed convection flow in a horizontal channel has been investigated numerically. The effect of heated plate velocity relative to fluid velocity in a vertical channel has been reported here. The following conclusions are derived from present investigation.

- The heat transfer rate increases with increase in Reynolds number due to increase in velocity of fluid.
- In a vertical channel, the velocity increases near the heated wall and the convective heat transfer rate increases with increase in Richardson number for assisting mixed convection flow due to buoyancy aiding the flow.
- For opposing mixed convection flows in vertical channel, the significant decrease in convective heat transfer rate with increase in Richardson number due to buoyancy opposing the bulk flow.
- The convective heat transfer coefficient slightly decreases with increase in Richardson number for mixed convection flow in a horizontal channel due to buoyancy acts normal to the flow.
- Buoyancy significantly altered the flow and heat transfer characteristics in a vertical channel compared to horizontal channel.
- A small recirculation zone forms near the inlet and its size increases with increase of plate velocity moves in the direction of bulk flow.
- The local convective heat transfer coefficient suddenly increases near the inlet due to motion of the plate but the average heat transfer coefficient decreases for plate velocity of $U_p \leq 0.1$
- At higher plate velocity of 0.5 the average heat transfer coefficient increases due to growth of recirculation zone.
- Disappearing of recirculation zones near the inlet if the plate moves opposite to the bulk flow.
- Present results would be useful to understand the flow and heat transfer characteristics of mixed convection flows in vertical and horizontal channels.

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