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Numerical Investigation of Mixed Convective Flow inside a Straight Pipe and Bend Pipe

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Abstract:*The present study deals with a numerical investigation of steady laminar and turbulent mixed convection heat transfer in a horizontal pipe and bend pipe using air as the working fluid.The thermal boundary condition chosen is that of uniform temperature at the outer wall. Computations were performed to investigate the effect of inlet Rayleigh number and Reynolds number in the velocity and temperature profile at inside of the pipe. The secondary flow is more intense in the upper part of the cross-section. It increases throughout the cross-section until its intensity reaches a maximum, and then it becomes weak at far downstream. For the horizontal pipe the value of the L/D ratio becomes more than 10 the secondary flow effects are neutralized and the velocity profile almost become constant throughout.*

Keywords: *Mixed Convection Heat Transfer, Laminar, Turbulent, Straight and Bend Pipe.*

I. Introduction

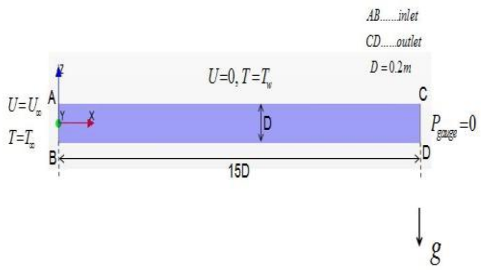
In heated flows in horizontal pipes, gravity-induced body forces may result from density variations within the fluid. Mori et al [1, 2] have shown flow visualization results for mixed convection in laminar flows of Newtonian fluids in horizontal pipes, demonstrating that the local variations in the fluid density lead to counteracting transverse vortices (or secondary flow patterns), which are superimposed to the main axial flow. The conditions under which flow in a circular pipe can be assumed to be purely forced, purely free and mixed convective have been presented in graphical form by B. Metais and ERG Eckert[3,4]. The Rayleigh number and Reynolds number was selected based on this reference .Anu Nair et al [5] conducted an review of Natural Convective Heat Transfer From Horizontal Heated Plate Facing Upward In Vertical Channel. He compared correlation for Nussult number in terms of Rayleigh number with different working fluid in different shapes of plates in detailed manner. Anu Nair et al. [6,7] Conducted new Optimization method (i. e, Least square method and Bayesian Approach) to find out the Value of C in the Nusslet Number.. Anu Nair et al [8,9] conducted a numerical investigation of natural convection flow and heat transfer in a square enclosure due to heat source or sources on its left wall has been investigated. The paper presents the effect of Rayleigh number and geometrical parameters such as location of heat source and gap between heat sources.The convective flow regimes are characterized the Richardson number (Ri) defined as the ratio of Rayleigh number to the square of the Reynolds number. The value of Richardson number is in the order of greater than one then the flow is free convection, and the order is less than one then the flow is forced convection, hence combined convection exist when the value becomes the order of one.

The present study deals with a numerical investigation of both laminar and turbulent mixed convection heat transfer in a horizontal and bend pipe. The special attention has been paid to the straight section in the bend pipe. The combination selected in this study is isothermal wall condition at the surface of the pipe and uniform temperature inside the flow domain. The numerical method adopted in this work involves the solution of the non- linear, pressure based segregated momentum and energy Equations with Boussinesq model. The k – ε model (model with two equations) was used in the turbulent flow calculations.

II. Mathematical Formulation

2.1 Governing Equations

The schematic diagram is depicted in Fig .(1) and Fig. (2). They represent for the horizontal and bend pipe respectively. The diameter of the pipe is 0.1 meter. The meshing was done filling complete domain with polyhedral cells and prism layer at the wall surface. After the grid dependence studies the number of cells in horizontal pipe to be fixed to $N \approx 2.4\times10^5$ and for the bend pipe $N \approx 3\times10^5$.



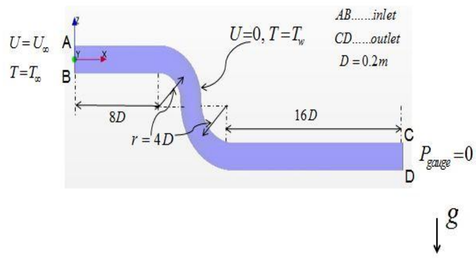


Fig .1 Geometry and Computational domain of horizontal pipe. **Fig.2** Geometry and Computational domain of bend pipe.

The following assumptions are adopted for this Numerical investigation

- Steady.
- Dissipation term is neglected.
- Constant temperature at the pipe surface.

The physical properties of the fluidare assumed constant except for the density in the body force,which change linearly with the temperature changes (Boussinesq's hypothesis). Dimensional conservation equations for steady state condition in integral form are as follows:

Continuity equation:

$$\iiint_V \vec{\nabla}\cdot\rho\vec{U}\,dV = 0 \qquad (1)$$

Momentum equation:

$$\frac{\partial}{\partial t}\iiint \rho\vec{U}dV + \iint_s \rho\left(\vec{U}\cdot\vec{n}\,dS\right)\vec{U} = \underbrace{\iiint_V \rho\,\vec{g}\,\beta\,(T-T_\infty)\,dV}_{Boussinesq\ Approximation} - \iiint_v \vec{\nabla}p\ dV + \vec{F}_{viscous} \qquad (2)$$

Energy equation:

$$\frac{\partial}{\partial t}\iiint \rho E\,dV + \iint_s (\rho\vec{U}\cdot\vec{n}\,dS)E = -\iint_S p\vec{U}\cdot dS + \iint_s k\vec{\nabla}T\,dS + W_{body} + W_{viscous} \qquad (3)$$

2.2 Initial Conditions

Initially the computational domain was simulated with inlet velocity as $V=V_\infty$ and find the developed velocity profile at the entry length. And this velocity profile is initiated in the complete domain. The profile is in parabolic shape with following conditions

$$V_x = V_{\max}(r=0)\ ,\quad V_x = 0(r=R),\ V_y=0,\ V_z=0,\ T=To \qquad (4)$$

2.3 Boundary conditions

The set of all the above non- linear equation can be solved by the following boundary conditions:

At the pipe inlet(X=0)

$$V_x = V_{\max}(r=0)\ ,\quad V_x = 0(r=R),\ V_y=0,\ V_z=0,\ T=T\infty \qquad (5)$$

At the pipe wall

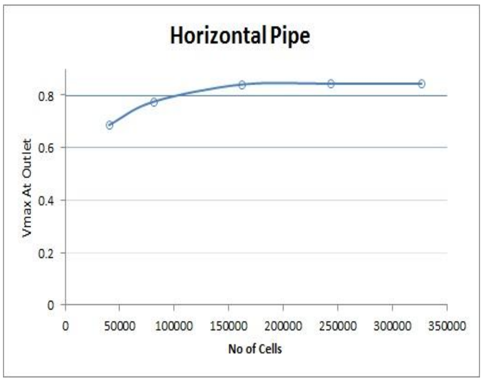
$$V_x = 0,\ V_y=0,\ V_z=0,\ T=T_W \qquad (6)$$

At the pipe outlet

$$p_{gauge}=0,\ \frac{\partial V_x}{\partial r}=0,\ V_y=0,\ V_z=0 \qquad (7)$$

III. Computational Methodology

The mesh was generated in the Star CCM+ from coarse to finer and grid independence study was done. For the horizontal pipe the maximum velocity at the outlet section is considered and for bend pipe maximum velocity at the longitudinal central plane was considered. The Fig.3 and Fig.4 represent the grid convergence result of horizontal and bend pipe.



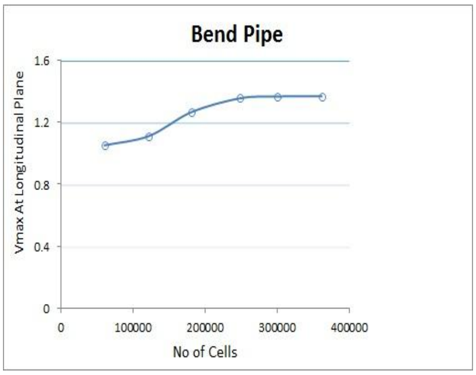


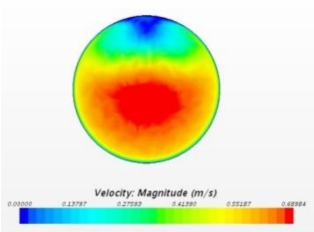
Fig.3 Grid convergence (horizontal pipe) **Fig.4** Grid convergence (bend pipe)

For the horizontal pipe from $N \approx 2.4 \times 10^5$ the property values are become almost constant and for bend pipe from $N \approx 3 \times 10^5$. so the computational domain can be fixed according to the above convergence criteria. Polyhedral cells with prism layers at the pipe wall surface are created in the Star CCM+ and this domain was taken for the analysis.

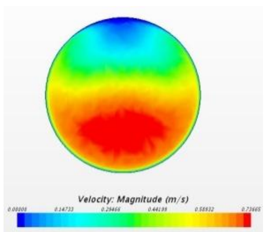
The basic numerical procedure adopted in the present study is the SIMPLE algorithm. The above sets of non- linear differential equations are discretized with the finite volume method.

IV. Results And Discussions

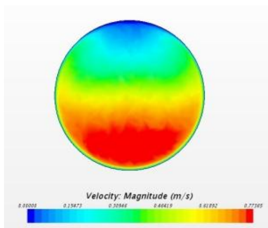
The numerical investigations are carried out with $Ra = 10^4, 10^5, 10^6, 10^7, 10^8$ and Re = 1000, 7500 the velocity and temperature variation are analyzed here.



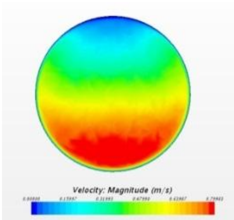
$(a) L/D = 2.5$



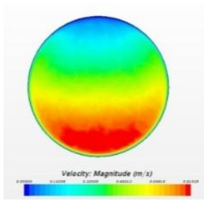
$(b) L/D = 5$



$(c) L/D = 7.5$



$(d) L/D = 10$

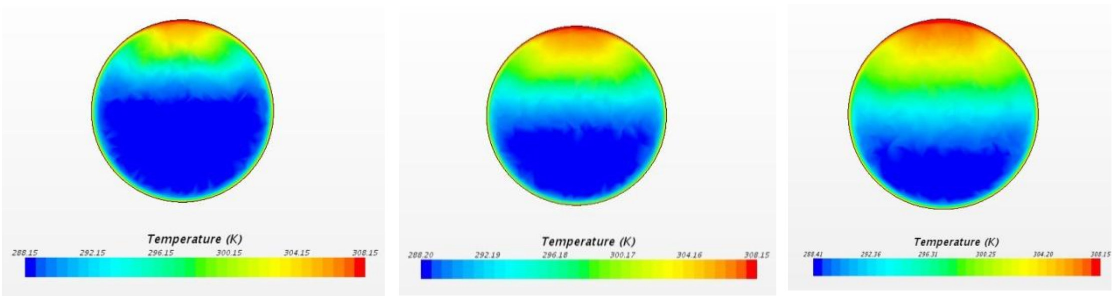


$(e) L/D = 12.5$

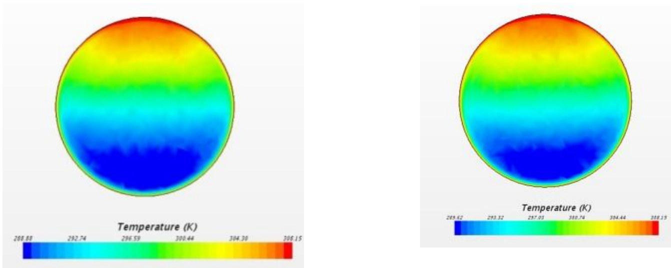
Fig. 5 Velocity contours at various cross section Ra=10^8, Re=7500(horizontal pipe)

The above Fig.5 represent the velocity contours at the various cross section of the horizontal pipe such that the length to diameter ratio going from 2.5 to 12.5 with a step of 2.5. Initially the velocity is maximum at the center of the pipe because of the parabolic fully developed profile. As the L/D ratio increasing the local variations in the fluid density lead to counteracting transverse vortices which are superimpose to the main axial flow. By this effect the secondary flow is generated inside the pipe, which produces maximum velocity at the bottom area (buoyancy force) of the pipe. The value of the L/D ratio becomes more than 10 the secondary flow effects are neutralized and the velocity profile almost become constant throughout.

The following Fig.6 shows the variation in the temperature contour with respect to the increasing value of L/D ratio.



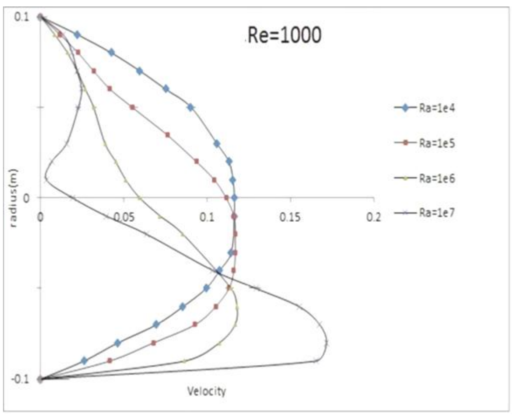
$(a)\,L/D=2.5$ $(b)\,L/D=5$ $(c)\,L/D=7.5$



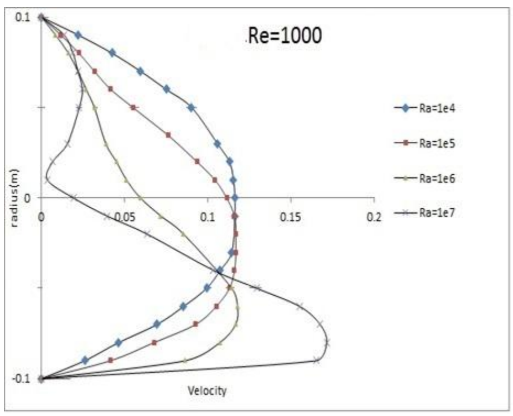
$(d)\,L/D=10$ $(a)\,L/D=12.5$

Fig.6 Temperature isotherms at various cross section Ra=10^8, Re=7500(horizontal pipe)

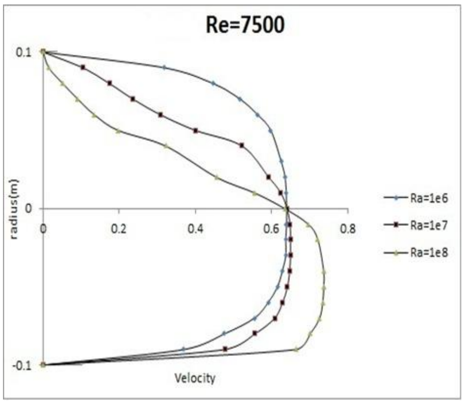
The secondary flow is more intense in the upper part of the cross-section. It increases throughout the cross-section until its intensity reaches a maximum, and then it becomes weak at far downstream. The above contours clearly shows, due to the buoyancy force the direction of heat flowing from top to bottom part. The numerical study was carried out with the isothermal wall temperature and uniform temperature air throughout the domain with a temperature difference of 20K, Rayleigh number 10^8, with Reynolds number as 7500 which is in turbulent mixed convection region according to B. Metais and ERG Eckert[3,4].



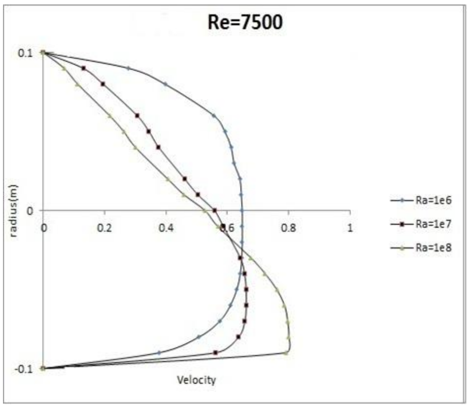
$(a)\,L/D=5,\ \mathrm{Re}=1000$



$(b)\,L/D=10,\ \mathrm{Re}=1000$



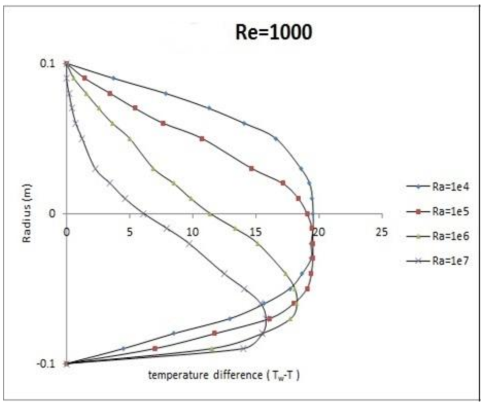
$(c)\, L/D = 5,\ \mathrm{Re} = 7500$



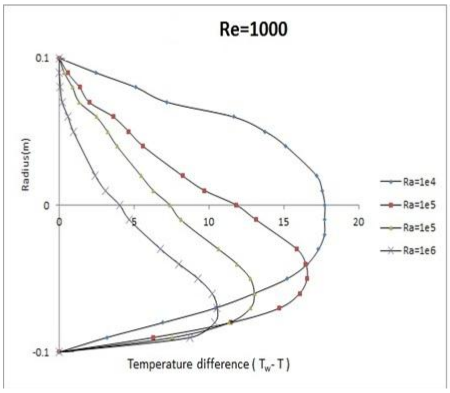
$(d)\, L/D = 10,\ \mathrm{Re} = 7500$

Fig. 7 Velocity profile at various cross sections (horizontal pipe)

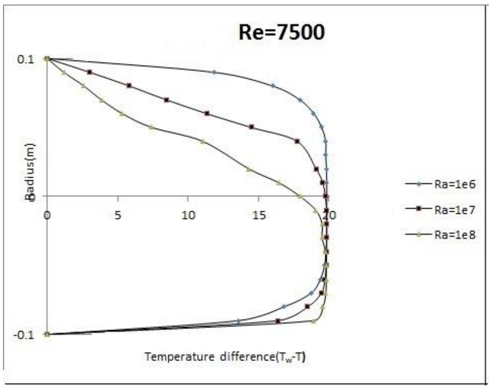
The Fig.7 shows the velocity profile for both laminar and turbulent flow in the various section of the horizontal cylinder at L/D value of 5 and 10 with respect to the various Rayleigh number. As the value of the Rayleigh number increases the buoyancy force effects are presents. For the low value of the Rayleigh number the flow is forced convection and for high values flow becomes free convection and there are some regions were the both free and forced convective flows are significant. From the above figures explains that as the Rayleigh value increases the velocity magnitude at the lower area is dominating compared with the upper area which is due to the effect of secondary flow. This effect becomes weaker at the far downstream.



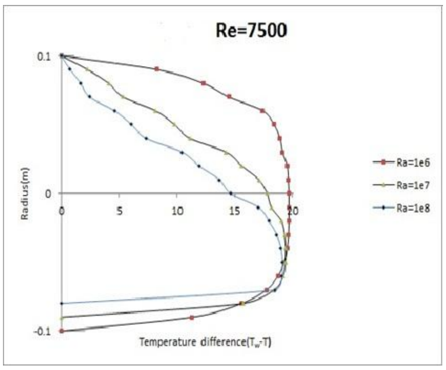
$(a)\, L/D = 5,\ \mathrm{Re} = 1000$



$(b)\, L/D = 10,\ \mathrm{Re} = 1000$



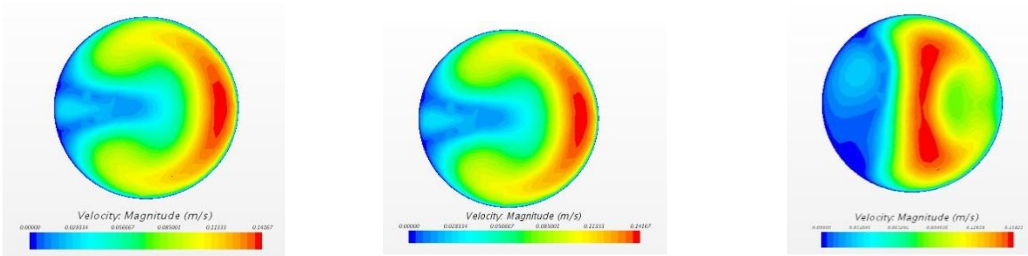
$(c)\, L/D = 5,\ \mathrm{Re} = 7500$



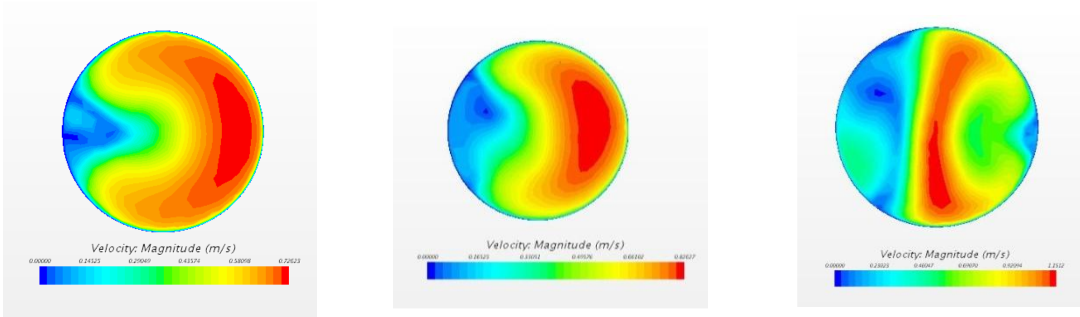
$(d)\, L/D = 10,\ \mathrm{Re} = 7500$

Fig.8 Temperature difference profile at various cross sections (horizontal pipe)

The convection effects in the temperature profile as shown in the Fig.8. For the low Rayleigh numbers the temperature profile is parabolic and when the value of Rayleigh number greater than some value the buoyancy forces become significant and it plays role on the temperature profile. Fig. (a) and (b) for laminar case and (c) and (d) for turbulent case.



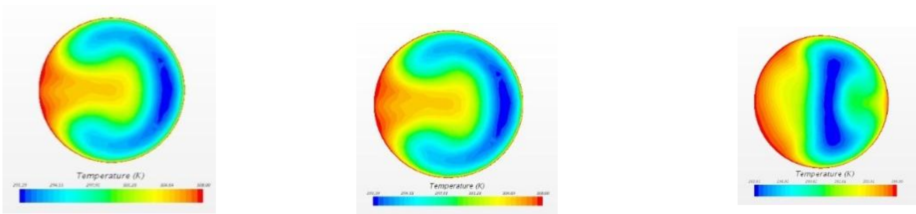
$(a) Ra = 10^4,\ \mathrm{Re} = 1000$ $(b) Ra = 10^5,\ \mathrm{Re} = 1000$ $(c) Ra = 10^6,\ \mathrm{Re} = 1000$



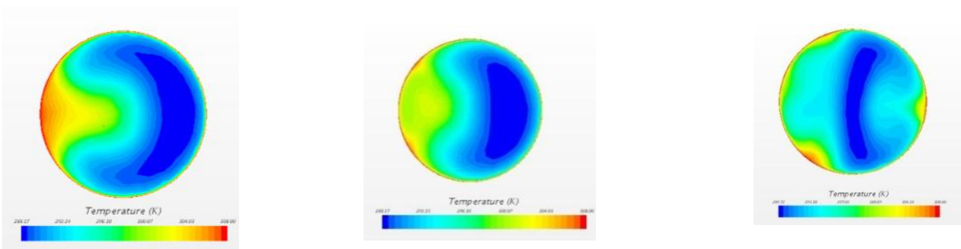
$(d) Ra = 10^6,\ \mathrm{Re} = 7500$ $(e) Ra = 10^7,\ \mathrm{Re} = 7500$ $(f) Ra = 10^8,\ \mathrm{Re} = 7500$

Fig. (9) velocity contours at cross-sectional central plane (bend pipe)

The velocity contours at the cross sectional central plane of the bend pipe as shown above. Out of these Fig.(a), (b), (c) is for laminar flow with different Rayleigh number and Fig. (d), (e), (f) is for turbulent flow with different Rayleigh number. As the Rayleigh number values going up the convective flow transition occur from forced to free convection. For the pipe bend the mixed convective flow consists of two effects. One is due to the secondary flow generation in the horizontal region and due to density variation in the vertical region. Both the effect superimposes each other and it is shown in the Fig. (c), (f).



$(a) Ra = 10^4,\ \mathrm{Re} = 1000$ $(b) Ra = 10^5,\ \mathrm{Re} = 1000$ $(c) Ra = 10^6,\ \mathrm{Re} = 1000$



$(d) Ra = 10^6,\ \mathrm{Re} = 7500$ $(e) Ra = 10^7,\ \mathrm{Re} = 7500$ $(f) Ra = 10^8,\ \mathrm{Re} = 7500$

Fig. 10 Temperature Isotherms at Cross-Sectional Central Plane (Bend Pipe)

The temperature isotherms at the cross sectional central plane of the bend pipe as shown above. Out of these Fig.(a), (b), (c) is for laminar flow with different Rayleigh number and Fig. (d), (e), (f) is for turbulent flow with different Rayleigh number. As the Rayleigh number values going up the convective flow transition occur from forced to free convection. For the pipe bend the mixed convective flow consists of two effects. One is due to the secondary flow generation in the horizontal region and due to density variation in the vertical region. Both the effect superimposes each other and it is shown in the Fig. (c), (f).

V. Conclusion

Numerical investigation of mixed convective flow was analyzed here. The effect of secondary flow due to the buoyancy forces is found to be significant. The secondary flow in the horizontal pipe is more intense in the upper part of the cross-section. It increases throughout the cross section until its intensity reaches a maximum, and then it becomes weak at far downstream. The development of axial flow and temperature field are strongly influenced by the buoyancy. The buoyancy influence is more pronounced near the inlet section. This influence increases with increasing Rayleigh number. At far downstream, the interaction between the axial flow and the secondary flow becomes weak.

Inside the pipe bend the buoyancy force in the horizontal and vertical region are affects the heat transfer. From the computation it found to be for the laminar (Re=1000) case with Rayleigh number between 10^5 and 10^6 the effect of both free and forced convection starts and for turbulent (Re=7500) case the value of Rayleigh number is between 10^7 and 10^8.

Nomenclature

C_p — *specific heat*, J/kgK

D — *diameter of the pipe*, m

L — *length of the pipe*, m

g — *acceleration due to gravity*, m/s^2

Gr — *Grashoff number*, $(g\beta\Delta TD^3)/\nu^2$

k — *thermal conductivity*, W/mK

$\Pr$ — *Prandtl number*, $\mu C_p/k$

Ra — *Rayleigh number*, $Gr\Pr$

Re — *Reynolds number*, VD/ν

T — *Temperature*, K

V — *velocity*, m/s

N — *number of cells*

Ri — *Richarson number*, Ra/Re^2

Greek symbols

β — *coefficient of thermal expansion*, $1/K$

Δ — *difference*

ν — *kinematic viscosity*, Ns/m^2

ρ — *density*, kg/m^3

Subscripts

∞ — *free stream*

w — *wall*

X — *axial component*

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