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PERFORMANCE ANALYSIS OF NATURAL CONVECTION IN EVACUATED SOLAR WATER HEATER

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ABSTRACT

There has been wide scale adoption of evacuated tubular solar water heaters in China, Europe and recently in India. Factors limiting the performance of this type of product have been evaluated. A numerical model of the heat transfer and fluid flow inside a water-in-glass tube has been developed to investigate the heat extraction from the long collector tubes and the flow structure inside the tube. The effect of diameter of evacuated tube, length of evacuated tube, angle of tilt of evacuated tube and tank diameter on the natural circulation loop and heat extraction has been analysed. An attempt has been made to optimise the dimensions and configuration of the system using Computational Fluid Dynamics too with the help of ANSYS FLUENT® software.

Keywords: Solar water heaters, Heat extraction, Solar Collectors.

NOMENCLATURES

c	specific heat, J kg^{-1} K^{-1}
gy	gravitational acceleration, m s^{-2}
k	thermal conductivity, W m^{-1} K^{-1}
L	length of the evacuated tube, m
r	radius of the evacuated tube, m
Ra	Rayleighs number
T	temperature of the fluid, K
t	time step, s

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u, v, w	components of velocity, m s^{-1}
θ	angle of inclination w.r.t vertical, degrees
ρ	density of the fluid, kg m^{-3}
β	volumetric expansion coefficient, K^{-1}
μ	Viscosity of fluid, Pa-s

1. INTRODUCTION

India's energy challenges are multi–pronged. The excessive consumption of fossil fuels has led to detrimental impact on the environment and has also caused energy poverty in recent years hence, the development of alternative energy has become a widely concerned issue. Solar energy is considered as one of the most promising alternatives. Solar energy can be harvested directly in two forms. The first useful form is electricity, which is obtained by exposing a photovoltaic material to sunlight. The second useful form is heat; here heat is transferred from sunlight to a working fluid such as heat transfer oil or water.

Morrison et al. [1] in his research mentioned that evacuated tube solar collectors have better performance than flat plate solar collectors in particular for high temperature operations. The motive of this work is to study the natural thermosiphon loop that is formed inside the evacuated tube solar water heaters which helps in heat extraction from the tubes and to optimize the configuration and dimensions of the system with the help of Computational Fluid Dynamics (CFD) tool using ANSYS FLUENT® software.

Several researches have been carried out and several devices have been devised for solar water heating applications. W.Elenbaas [2] first studied the natural convection flow through vertical plate channels and pipes. Lighthill [3] presented an analytical study of vertical thermosyphons for laminar and turbulent flow and identified three basic flow regimes for uniform wall temperature. In relation to water-in-glass evacuated tubes a stagnant region would result in an inactive section of the tube. Lighthill defined a parameter P which predicts the existence of a stagnant region as

$$P = Ra\,(Lr)^{-1}\cos\theta \qquad (1.1)$$

According to Lighthill, if P is less than 350, then a stagnant region forms.

G.L. Morrision et al. [4] studied the transient response of thermosyphon solar collectors.

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Measurements of transient flow thermosyphon circuit were obtained using Laser Doppler Anemometry (LDA). They also developed a mathematical model to simulate the transient performance. It was found out that there is long time delays associated with the development of thermosiphon flow; however the energy collection capacity is not affected by such time delays.

F.O. Gaa et al. [5] experimentally studied the flow rates through open inclined thermosyphon. The flow visualization by LDA showed the formation of distinct opposing streams exhibiting the thermosyphon effect in the inclined open cylinder. They also conducted numerical investigations [6] in an open thermosyphon inclined at $45^{0.}$ The investigation shows that flow in a differentially heated inclined open thermosyphon is typically bi-filamental.

In recent decade, a number of studies have been conducted on evacuated tube solar water heater systems. Budihardjo et al. [7-8] investigated the essentials of system such as collector optical efficiency, collector heat loss, storage tank heat loss, and natural circulation flow rate through collector tubes. Kim et al.[9] studied glass evacuated tubes with a co-axial fluid conduit inserted in each tube using a one-dimensional model. Kim et al. [10] examined four different shapes of absorber tube such as finned tubes, a U-tube welded inside a circular fin, a U-tube welded on a copper plate and a U-tube welded inside a rectangular duct using a numerical method and they found the best absorber tube shape for a solar collector.

Budihardjoet al.[11] correlated the circulation flow rate in terms of solar input, tank temperature, collector inclination and tube aspect ratio. Further numerical simulation showed that when the heat input is concentrated on the top circumference of the tube, as is the case with collectors mounted over a diffuse reflector, the effect of circumferential heat flux distribution on the circulation flow rate through the tubes is not significant; therefore, the correlation could be used to predict the flow rate at any time of day.

Many experimental works have been carried out in the past; however, CFD simulations have become a recent trend. The advancement in computer technology in the last two decades had repercussions in research methodology and now the CFD solutions are widely accepted. The consolidated effect of diameter of the tube, length of tube, the distance between the centres of the tubes, angle of inclination on extraction of heat from the tubes have not been analysed and hence need further research. Furthermore, deeper insight is needed on the natural convection loop that is formed inside the evacuated tubes. Since nature of the problem is transient one it requires more calculations compared to steady state simulations. Until the last decade least attempt was made to carry out extensive simulations for various dimensions. With better resources in hand, there is need to present detailed simulation results which can help in further improvement in the design of such thermal systems.

The objective of this work is to analyse the effect of the following parameters on the natural circulation inside the evacuated tube, the stratification inside the tank, heat extraction from tube and performance, using CFD:

- Diameter of evacuated tube.
- Length of evacuated tube.
- Inclination angle of the evacuated tube.
- Diameter of storage tank.

2. SYSTEM DESCRIPTION

The dimension of the solar water heating system in usage, as shown in the above Figure 2.1 is as follows: The diameter of the cylindrical tank in which the water is stored is 370 mm and the length of the tank is 950 mm. There are 11 tubes attached to the tank. Each tube has a length of 1.65 m and is attached to the tank at an angle of inclination of 30^0. The diameter of the inner tube of the solar collector is 40mm.

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Figure 2.1Evacuated Tube Solar Water Heater

Three-dimensional models are developed considering only the water in the shape of the storage and pipe. Since the modelling of glass and other components would result in increased computational cost without adding considerable accuracy to the solution. The heat transfer by radiation is modelled by applying a heat flux incident on the lateral surface of the tube.

Variations in geometry were made in order to check the effect of parameters. In the first case, the diameter (30mm, 34mm, 38mm and 42mm) of the evacuated tube is varied keeping all other dimensions constant. In the second case, length (1000mm, 1400mm, and 2000mm) of the evacuated tube is varied similarly. As a third case the angle of inclination (30^0, 45^0 and 60^0) with respect to horizontal is varied accordingly. Lastly in order to test the effect of diameter of the tank, its diameter is varied keeping the volume constant (as a consequence the length of the storage tank is also changed).

In this study, the initial temperature of the water in the models was set as 300 K, and initially fluid is stationary. No slip condition is applicable at the walls. Uniform heat flux is applied on the tube surface of 1000 [W/m²]. The lower half portion of the tube is kept insulated. This kind of boundary condition aids the bi-filamental flow inside the tube. If somehow heat flux could be provided on both the sides of the tube, practically by using reflecting mirrors, net heat input to the system would be more, but the flow could be impeded by small recirculation regions formed within the tube. With later choice, the heat extraction could become difficult, the temperature inside the tube might rise and decrease the efficiency of the system, yet the losses from the tube is negligible due to maintenance of vacuum outside the tube. Rest of the tank remains insulated and is given insulated boundary condition. However, in real case heat loss takes place from the tank and this lowers the efficiency especially when the ambient temperature is low. Hence in practice heavy insulation for tank and piping network is recommended.

3. GOVERNING EQUATIONS

a) Continuity Equation

$$\frac{D\rho}{Dt} + \rho\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) = 0 \tag{3.1}$$

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b) Boussinesq Approximation

For many natural-convection flows, faster convergence is obtained with the Boussinesq model than by setting up the problem with fluid density as a function of temperature. Even the results obtained by BA model are closer to the experimental data than those obtained using the VPT model.

This model treats density as a constant value in all solved equations, except for the buoyancy term in the momentum equation:

$$(\rho - \rho_0) \sim \rho_0 \beta (T - T_0) g_y \quad (3.2)$$

$$\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p \quad (3.3)$$

Where, ρ_0 is the (constant) density of the flow, $T_0$is the operating temperature, and β is the thermal expansion coefficient whose value was set to 0.0033 K^{-1}. Equation is obtained by using the Boussinesq approximation to eliminate from the buoyancy term. This approximation is accurate as long as changes in actual density are small; specifically, the Boussinesq approximation is valid when, $\beta(T-T_0) \ll 1$. The Boussinesq model should not be used if the temperature differences in the domain are large. As the simulations are run for an hour, the temperature rise is not more than 10^0C. In addition, it cannot be used with species calculations, combustion, or reacting flows. Since here there is no such situation BA model is reasonably applicable.

c) Momentum equations

- Component *x*

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial w}{\partial z} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \quad (3.4)$$

- Component *y*

Eq. (3.2) is combined with the pressure gradient (hydrostatic pressure) and the body force to have a final term in the BA y-momentum Eq. (3.5) that is expressed as $\rho_0\beta(T - T_0)g_y$

(which is highlighted with a different font in Eq. (3.5)) for the buoyancy effects. Eq.

(3.5) couples the temperature field and the flow field.

$$\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = -\rho_0\beta(T - T_0)g_y + \frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) \quad (3.5)$$

- Component *z*

$$\frac{\partial w}{\partial t} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} = -\frac{1}{\rho}\frac{\partial p}{\partial z} + \frac{\mu}{\rho}\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) \quad (3.6)$$

d) Energy equation

$$\rho c\left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) \quad (3.7)$$

4. RESULTS AND DISCUSSION

- **Effect of Diameter of the Evacuated Tube**

The diameter of the inner tube of the solar collector (also called as riser tube) of Sudarshan Saur solar water heater as measured is 40mm. To understand the effect of diameter four different geometries have been created with the diameter of riser tube as 30mm, 34mm, 38mm and 42mm. The other dimensions are kept same as shown in the Figure 1. The angle of inclination is kept at 300. Temperature contours at the mid-section, with four different riser tube diameters is as shown in the Figure 4.1 and 4.2

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As the diameter of the riser tube is increased, the corresponding surface exposed to the solar radiation also increases, thus the net heat input increases. There is corresponding rise in temperatures of the fluid for tubes of larger riser diameter. This is depicted in the temperature contours and the data in Table 4.1. It should also to be noted that as diameter of the riser tube is increased, volume of fluid inside the riser tubes also increases, however the additional heat input is dominant enough to raise the temperature of the fluid.

With the increase in diameter of the riser tube, the temperature stratification inside the tank is disturbed. The average temperature inside the tank also increases. Stratification in tank for large diameter tubes becomes less. Practically, some amount of stratification is an advantage. Usually the usage pipe is connected to the top of the storage tank (where the temperature of the fluid is high) so that hot water can be used for required purpose. If the stratification is disturbed, then the temperature of the fluid at the top of the tank would not be as high as it would be if stratification were present. However higher the average flow velocity, higher is the heat extraction, but the tank stratification is disturbed.

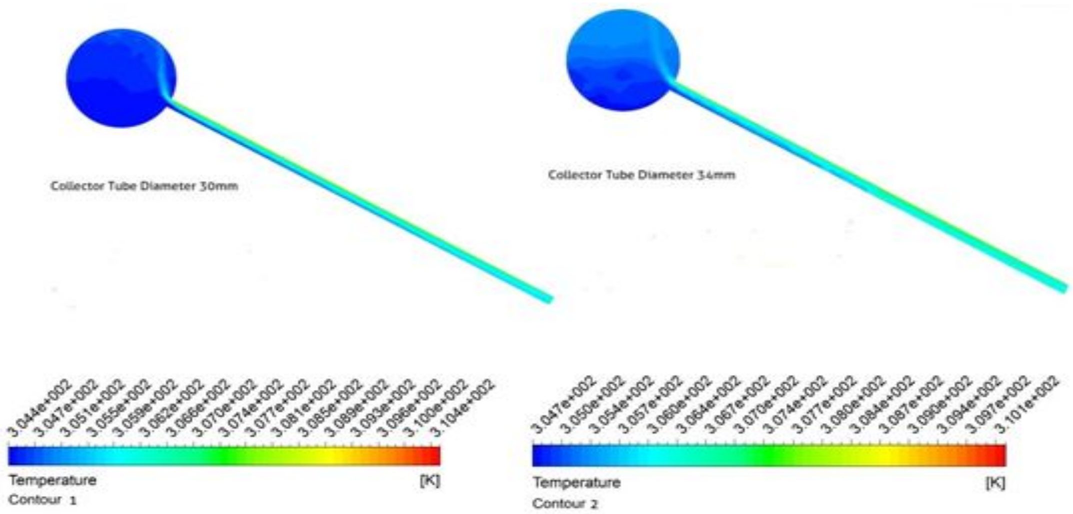

Figure 4.1 Contours of temperature for different diameter riser tubes after 4500 seconds, (a) collector tube diameter 30mm, (b) collector tube diameter 34mm

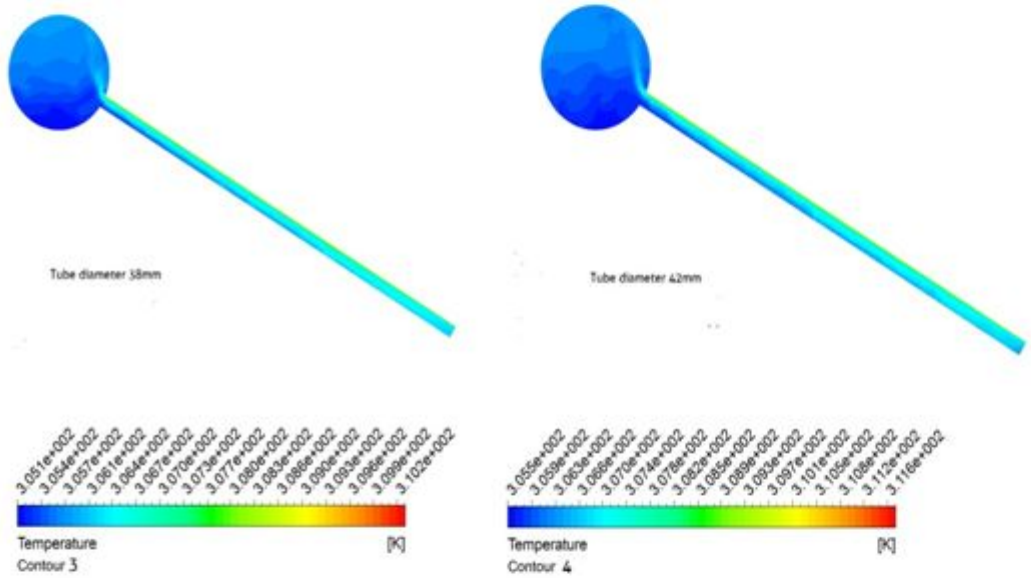

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Figure 4.2 Contours of temperature for different diameter riser tubes after 4500 seconds (a) collector tube diameter 38mm, (b) collector tube diameter 42mm.

Table 4.1 Parametric study for different diameter riser tubes

Diameter of the riser tube (mm)	Aspect ratio	Maximum temp of fluid (K)	Minimum temp of fluid (K)	Maximum heat transfer coefficient, (W/m^{2}K)	Average Temp of fluid in tank (K)	Average surface Temp of the tube (K)	Net Heat Input Q_{input} (W)
30	46.66	309.9	304.2	54.89	304.618	306.973	65.7673
34	41.17	310.1	304.7	53.65	305.119	307.243	74.5449
38	36.84	310.2	305.1	53.60	305.533	307.526	83.3238
42	33.33	311.6	305.5	50.80	305.902	308.154	92.0844

- **Effect of Diameter of the Evacuated Tube**

The length of the riser tube of Sudarshan Saur solar heater is 1.65m. By keeping the diameter as 34mm, the length has been varied. Three different lengths (1m, 1.4m, and 2m) were simulated. All other dimensions are same as shown in figure 3.2 of previous chapter. The angle of inclination is kept at 30^0. Temperature contours at the mid-section, with three different riser tube lengths is as shown in the Figure 4.3 and 4.4

As the length of the tube increases, the surface exposed to the solar radiation increases and thus the net heat input increases which causes the rise in temperature of the fluid. Again here the volume of the fluid inside the tube is also increased but the additional heat input is sufficient to cause rise in temperature. The temperatures and heat input is tabulated in Table 4.2

Table 4.2 Parametric study for different length of riser tubes.

Length of the riser tube (m)	Aspect ratio	Maximum temp of fluid (K)	Minimum temp of fluid (K)	Maximum heat transfer coefficient,h (W/m^{2}K)	Average Temp of fluid in tank (K)	Average surface Temp of the tube (K)	Net Heat Input Q_{input} (W)
1	29.41	307.5	302.9	60.1328	303.214	304.25	45.817
1.4	41.17	310.1	304.7	53.653	305.118	307.24	74.54
2	58.82	312.3	306.0	50.5304	306.439	309.29	99.11

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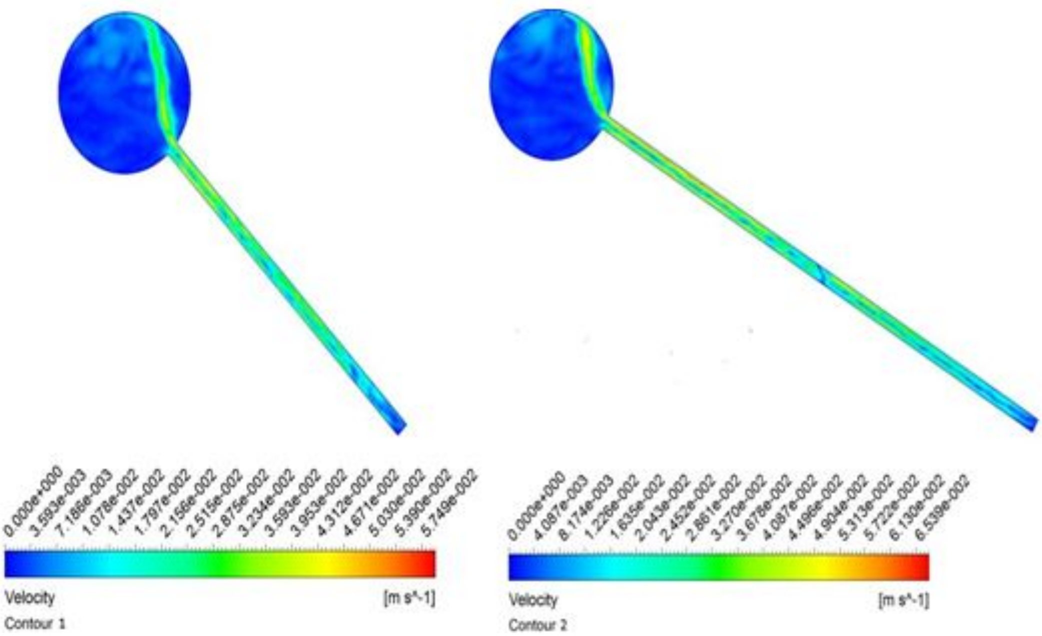

Figure 4.3 Contours of velocity for various tube lengths, (a) collector tube length 1m, (b) collector tube length 1.4m

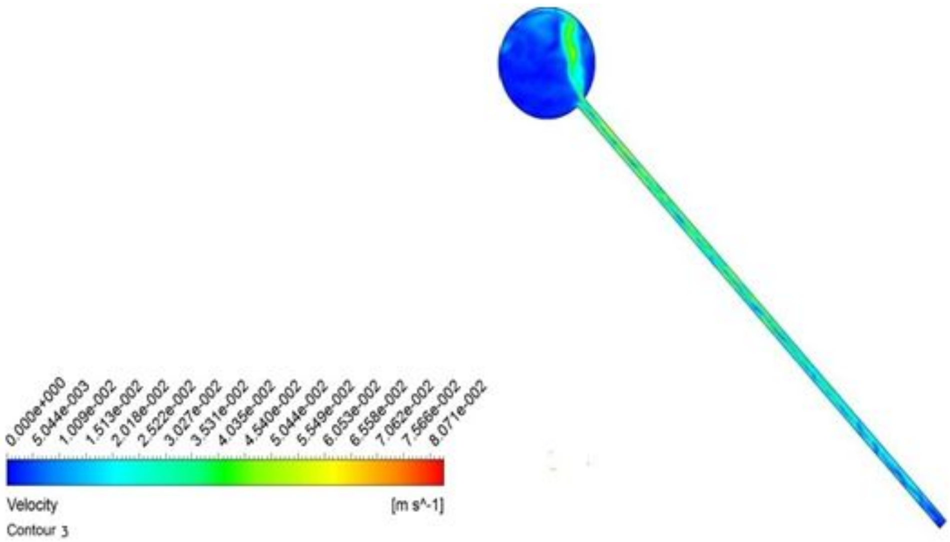

Figure 4.4 Contours of velocity for collector tube length 2m.

- **Effect of Angle of Inclination of the Evacuated Tube**

In order to augment the studies simulations have been carried out for three different angles (30^0, 45^0 and 60^0). It is seen in experiments that the buoyancy effects are more pronounced as the tube becomes vertical. It is recommended that the tilt angle must be chosen according to latitude to gain maximum amount of radiation. The effect of latitude is a dominating factor in selection of inclination angle. The temperature contours for different angle of tilt at the mid-section is as shown in the Figure 4.5 and 4.6. It can be easily seen that as the tilt angle is increased the stratification inside the tank is reduced. The stratification is almost negligible for 60^0 tilt angle.

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The average velocity for tilt angle of 45^0 is low. The temperature at the end of the tube is also high. Correspondingly there is also high amount of recirculation. This needs further investigations. However, in agreement with experimental results the velocities are high at higher inclination angle.

Table 4.3 Parametric study for different angle of inclination of riser tubes

Angle of inclination from horizontal	Maximum temp of fluid (K)	Minimum temp of fluid (K)	Maximum heat transfer coefficient, (W/m^2K)	Average Temp of fluid in tank (K)	Average surface Temp of the tube (K)
300	310	304.4	54.528	305.02	307.099
450	310.5	304.8	54.317	305.11	307.88
600	310	304.7	53.808	305.08	307.38

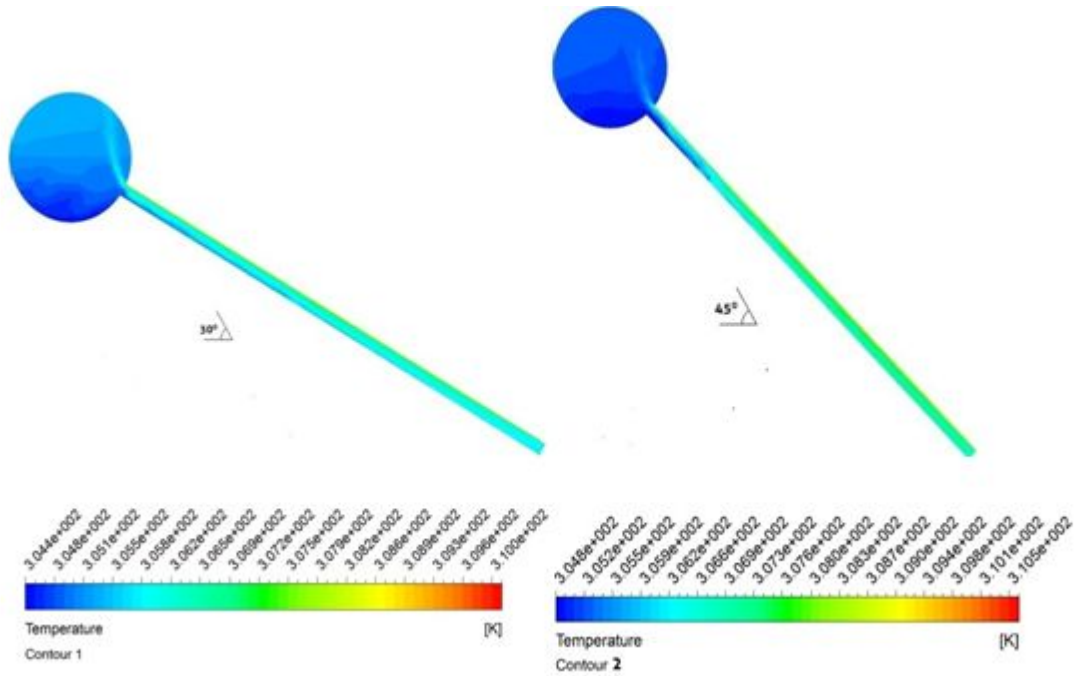

Figure 4.5 Contours of velocity for various collector tilt angles (a) collector tube tilt angle 60^0, (b) collector tube tilt angle 30^0

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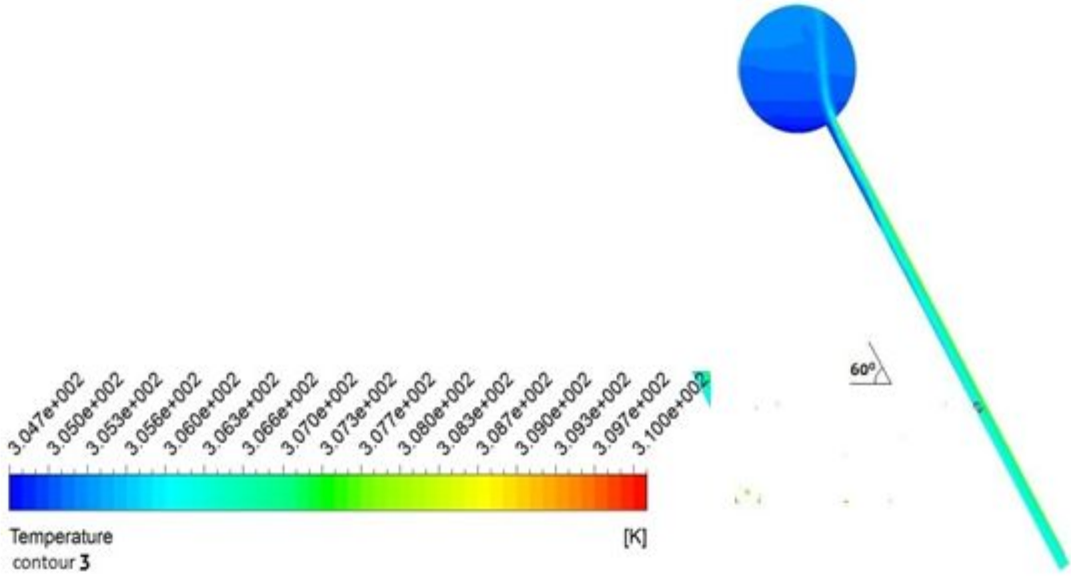

Figure 4.6 Contours of velocity for collector tube tilt angle 45⁰.

- **Effect of Tank Diameter**

The effect of change in spacing can be investigated by changing the length of the cylindrical storage tank (correspondingly the diameter of the tank is also changed to keep the volume constant). Alternatively, the diameter of the tank can be changed the other way round and correspondingly the length is also changed. The geometry has been simplified by considering only one tube in order to save computational costs. Two different tank diameters (350mm and 450mm) have been simulated to see the effect it bears on heat extraction from the riser tubes. The data is tabulated in Table 4.4 As the diameter of the tank is increased stratification has increased. There seems to be not much change in the heat extraction capacity for the larger diameter tank.

Table 4.4 Parametric study for different tank diameters.

Length of the storage tank (mm)	Diameter of storage tank (mm)	Maximum heat transfer coefficient, (W/m^{2}K)	Average Temp of fluid in tank (K)	Average surface Temp of the tube (K)	Average mid-plane velocities (m/s)	Maximum velocity (m/s)
150	350	53.653	305.118	307.24	0.01583	0.0653
114.8	400	52.856	305.375	307.15	0.01409	0.0631

The heat transfer coefficient, the velocities, temperatures are almost the same for both the cases. Inferring from this simplified analysis it can be said that changing the diameter of storage tank might not affect the heat extraction performance of the thermal system. However, the study of spacing of tubes is a complicated task because the effect of other tubes also comes into the picture. The velocity contours for different tank diameters is as shown in the Figure 4.7. One thing can be noted that the stagnation is more at the end of the tube for larger diameter tank. Large diameter could increase the hydrostatic pressure thus make it difficult for the fluid to come out of the tube.

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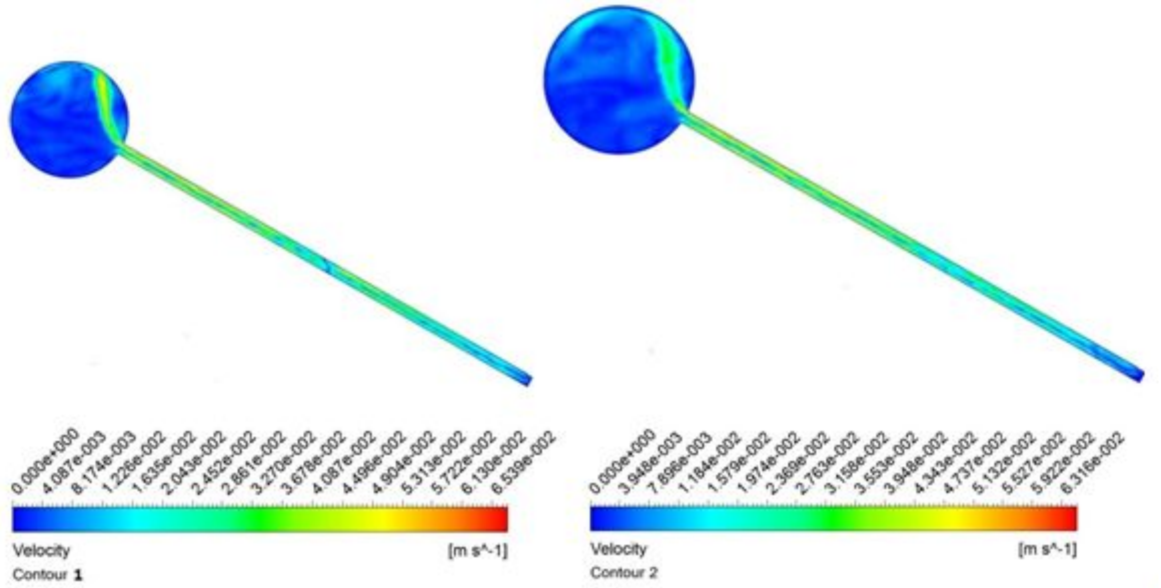

Figure 4.7 Contours of velocity for different tank diameters, (a) tank diameter 350mm, (b) tank diameter 400mm.

5. CONCLUSIONS

The effect of diameter of evacuated tube, length of evacuated tube, angle of tilt of evacuated tube and tank diameter on the natural circulation loop and heat extraction has been analysed using CFD.

- It is found that the shear layer is more pronounced for large diameter tubes. For very large diameter shear layer could impede the flow. Heat transfer coefficients also decrease for large diameter tube.
- For large length of tube/ higher aspect ratios a stagnation region is formed at the extreme end of tube and recirculation zones are formed which impede the flow and heat transfer from the tube to tank. Thus just by making the tube very large we might not get very hot fluid.
- Collector tilt-angle has significant influences on the daily collectible radiation and daily solar heat gain of a system, but insignificant on the heat removal from solar tubes to the water storage tank. Vertical tube/tube with larger inclination to horizontal has a better performance if it could collect maximum amount of radiation.
- Increasing the tank diameter did not have a significant effect on the heat extraction of tube.

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