# MAE 598 - Applied Computational Fluid Dynamics

Project-I

Submitted by:


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# Project 1

***Note***: *No collaboration was done in doing the present Project-1. The student acknowledges Professor Huei-Ping Huang for his insights and help in setting up the problem in FLUENT. For contour plots and profile plots, Tecplot 360 is used.*

## Task 1

**Part a**

To simulate the fluid flow and heat transfer in a water heater, with water as a working fluid. The geometrical parameters are: diameter of the tank $D = 0.6$ m, height of the tank $H = 0.8$ m, diameter of the inlet and outlet side pipes $d = 0.06$ m and length is $L = 0.15$ m, location of the inlet and outlet side pipe from the bottom wall $Z_1 = 0.6$ m and $Z_2 = 0.2$ m respectively. The boundary conditions are: velocity inlet on the inlet side pipe with $V_{in} = 0.05\ \frac{m}{s}$ with temperature $T_{in} = 10^o$ C. The outlet side pipe is assigned outflow condition. The bottom wall of the heater is assigned Dirichlet temperature boundary condition as $T = 55^o$ C. Standard k-epsilon turbulence model is used in the present simulation with full buoyancy effect. The pressure based solver is to be used in the present simulation and steady state solution is desired. The gravitational acceleration is taken as $g = -9.81\ \frac{m}{s^2}$.

The density is set as Boussinesq for water in order to allow density variation with temperature. The operating density is evaluated from below Eq. 1 from the given reference and its value is $\rho = 994.863 \frac{kg}{m^3}$. Based on the density, the thermal expansion coefficient is evaluated from below Eq. 2 and its value is $\beta = 0.0003249\ \frac{1}{K}$. In the given equation $T$ is the operating temperature and its value is average of maximum and minimum of temperatures in the system which is $T = 32.5^o$C.

$$\rho = 999.85308 + 6.3269\times10^{-2}T - 8.523829\times10^{-3}T^2 + 6.943248\times10^{-5}T^3 - 3.821216\times10^{-7}T^4 \tag{1}$$

$$\beta = -\frac{1}{\rho}\frac{d\rho}{dT} \tag{2}$$

**Part b**

To simulate the above given problem with the reduced gravity $g = -1.62\ \frac{m}{s^2}$.

## Solution

**Task 1(a)**

Figure 1 (a) shows the computational domain of the water heater and Fig. 1 (b) shows the grid (on the plane of symmetry) used in the present simulation. The element size for the triangular mesh is taken as 2.5 cm. The inflation is led by 'Program Controlled' with 5 layers having growth rate of 1.2 at the wall. In the present simulation, **"z"** is the direction parallel to the gravity vector. $g = -9.81 \ \frac{m}{s^2}$. Although, symmetry can be invoked, the present simulation is performed on full geometry.

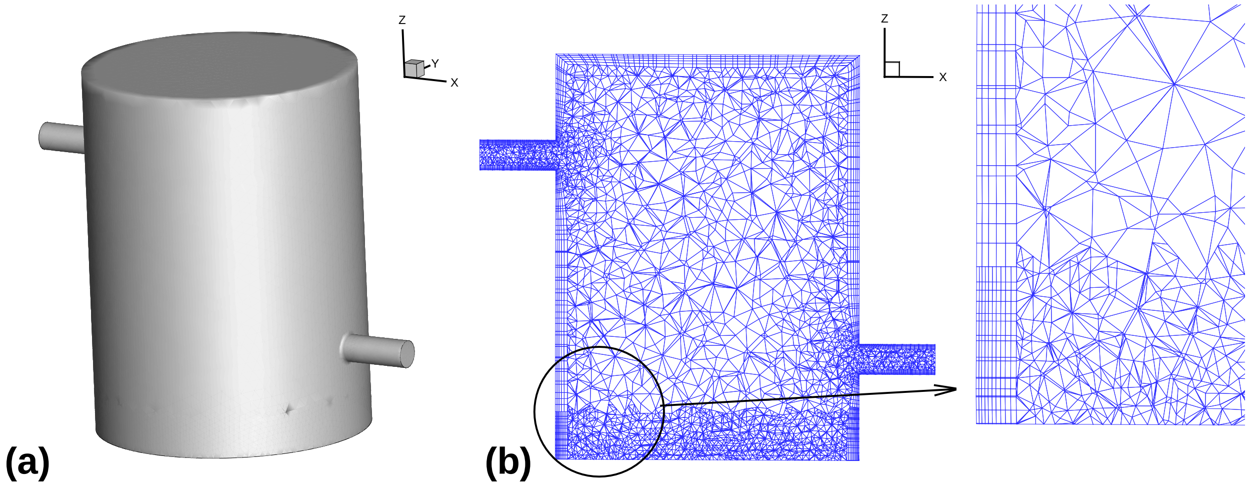


Figure 1: (a) Computational domain of water heater. (b) Grid resolution at plane of symmetry. Finer grid nearer to bottom wall.

### Deliverable (1 & 2)

The velocity and the temperature contour plot is shown in the Fig. 2.

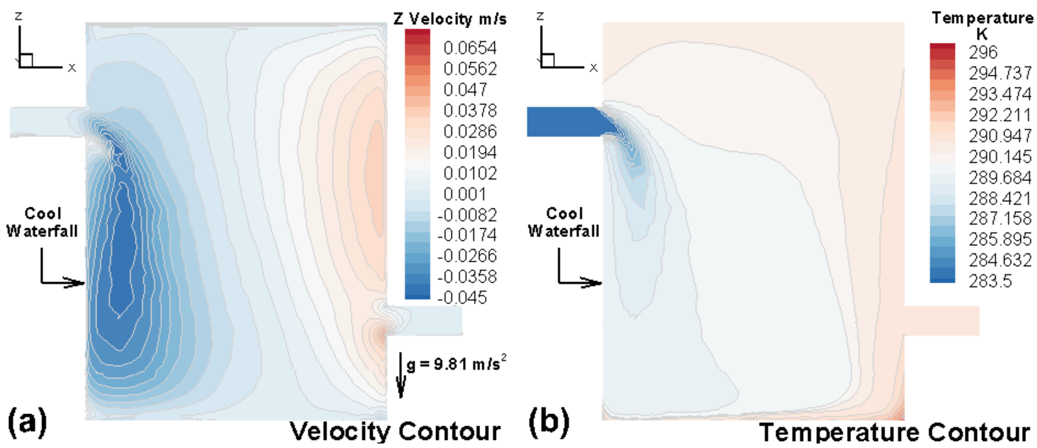


Figure 2: (a) Velocity contour at the plane of symmetry (here z-velocity) and (b) Temperature contour at the plane of symmetry.

### Deliverable (3)

The outlet temperature at the steady state is $T = 290.51$ K which is calculated by the Eq. 3.

$$T_{out} = \frac{1}{A} \iint_A T \, dA \tag{3}$$

The convergence criteria is achieved when the temperature between two iterations with interval of 100 is less than 0.05 K. The same is achieved between iterations 1900 and 2000 as observed in the Fig. 3. The temperature difference is observed as 0.045 K over the given interval of iterations.

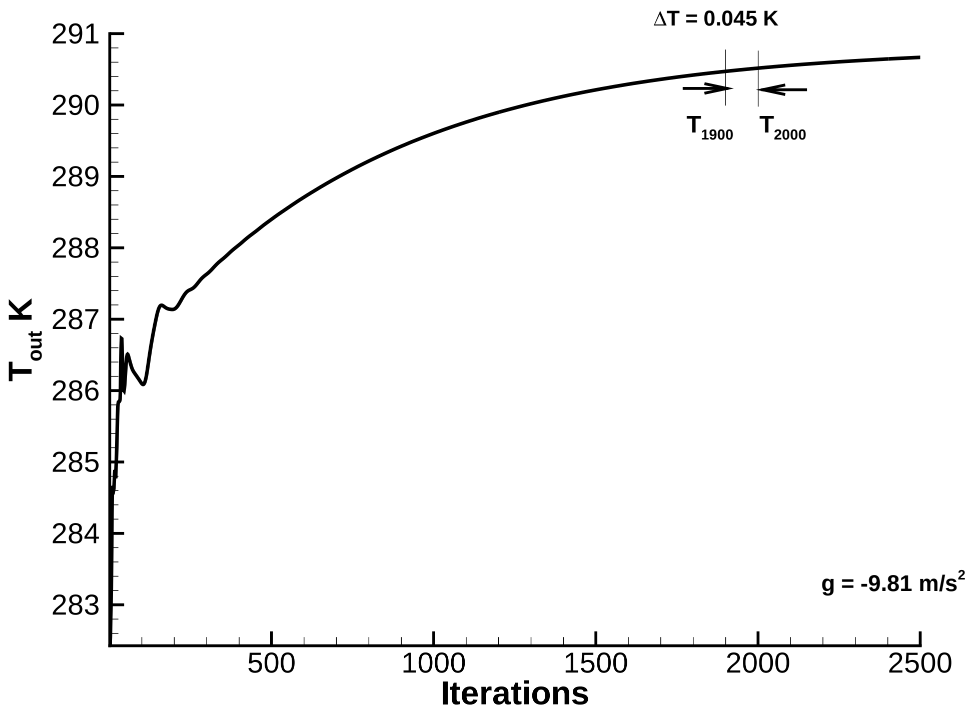


Figure 3: Line plot of the outlet temperature as a function of the number of iterations.

**Task 1(b)**

**Deliverable (1 & 2)**

The velocity and the temperature contour plot is shown in the Fig. 4 when $g = -1.62\ \frac{m}{s^2}$.

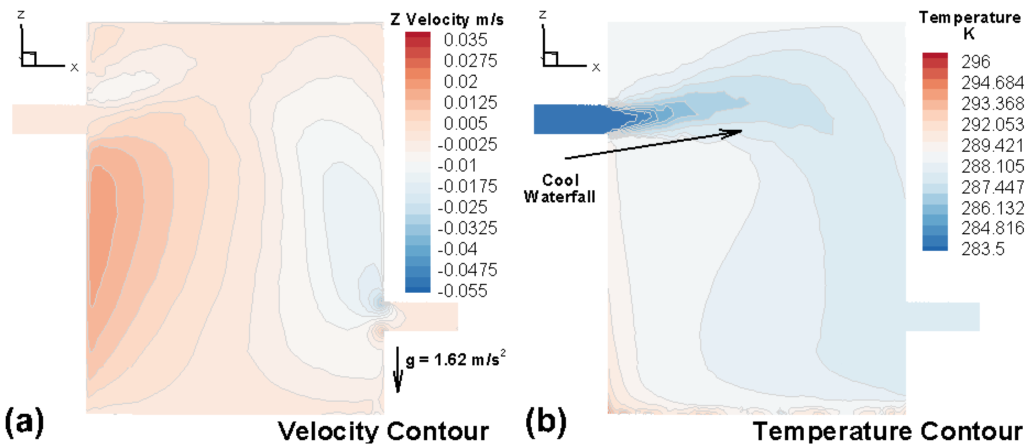


Figure 4: (a) Velocity contour at the plane of symmetry (here z-velocity) and (b) Temperature contour at the plane of symmetry.

**Deliverable (3)**

The outlet temperature at the steady state is $T = 287.82$ K which is calculated using Eq. 3. The convergence criteria is achieved when the temperature between two iterations with interval of 100 is less than 0.05 K. The same is achieved between iterations 2000 and 2100 as observed in the Fig. 5. The temperature difference is observed as 0.0404 K over the given interval of iterations. The nature of temperature plot is somewhat opposite to that expected which is mainly due to some singularity occurring at a significantly nearer grid points causing an initial value of temp. However, the steady state is still achieved.

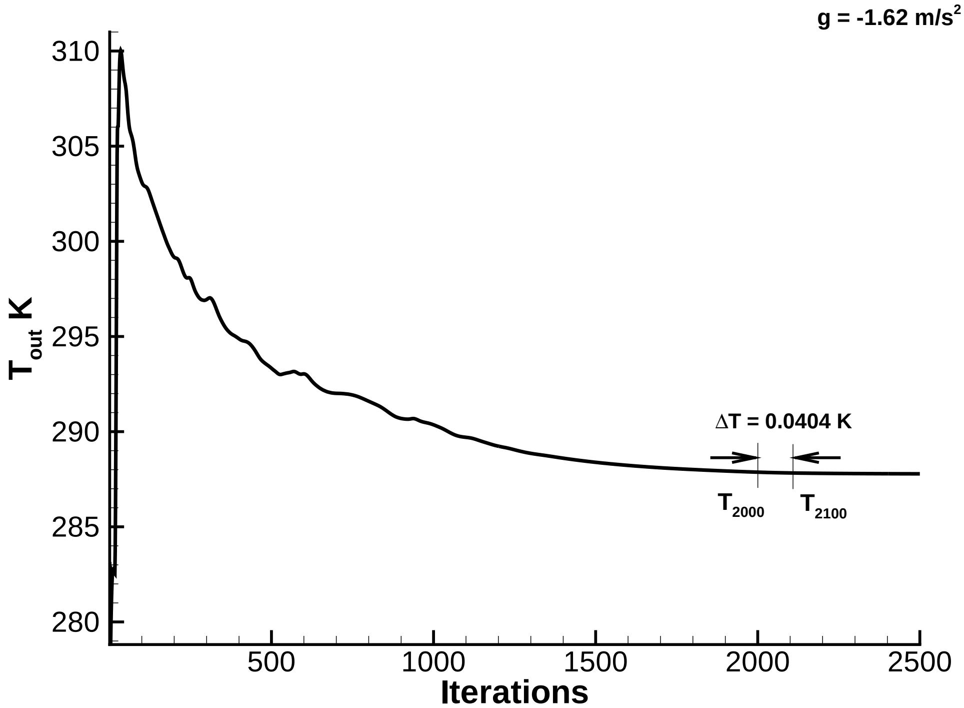


Figure 5: Line plot of the outlet temperature as a function of the number of iterations.

## Task 2

To simulate the fluid flow and heat transfer in a coiled water heater, with water as a working fluid. The helical pipe with its center traced by the equation of the helical curve as follows: $X(t) = R\,cos(t)$, $Y(t) = R\,sin(t)$, and $Z(t) = C\,t$. The curve is traced from $t = 0$ to $t = 10\pi$ with $R = 0.3$ m and $C = \frac{0.15}{2\pi}$ m. The helical pipe has circular cross section with $d = 8$ cm. The boundary conditions are: velocity inlet at inlet with four different values $V_{in} = 0.01, 0.02, 0.04, and\, 0.08\ \frac{m}{s}$ respectively, with temperature $T_{in} = 300$ K for all the cases. The outlet is assigned as outflow condition. The wall of the heater is assigned heat flux boundary condition with $Q = 500\ \frac{W}{m^2}$. Laminar model to be used in the present simulation. Pressure based solver is to be used in the present simulation and steady state solution is desired. No gravity to be considered in the present simulation and density is set to be constant for water.

### Solution

Figure 6 (a) shows the computational domain of the coiled water heater and Fig. 6 (b) shows the grid used in the present simulation and (c) shows the grid at inlet and outlet surface of the helical pipe. The element size for the triangular mesh is taken as 1 cm. The inflation is led by 'Program Controlled' with 5 layers having growth rate of 1.2 at the wall.

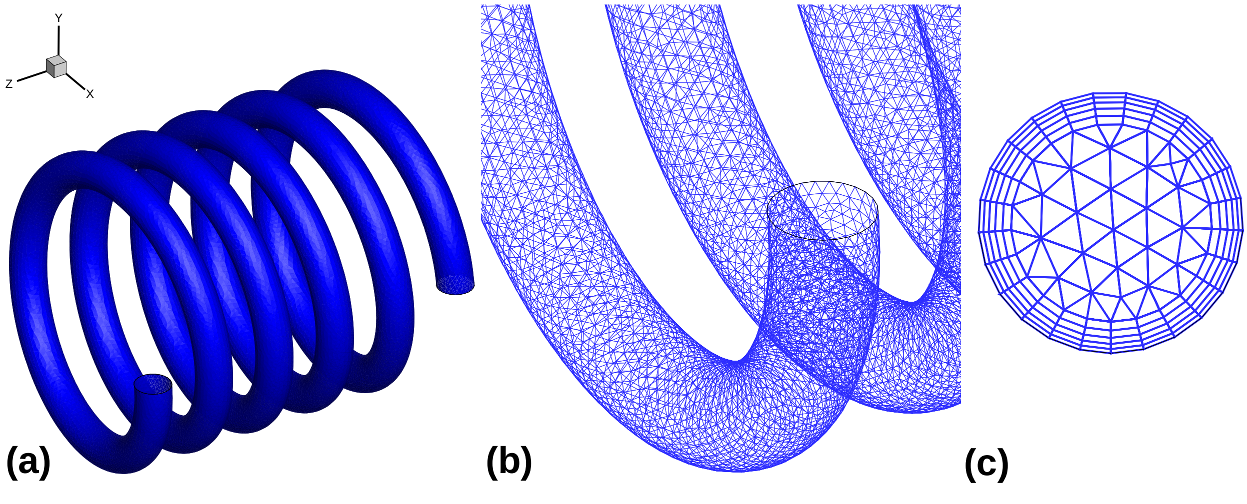


Figure 6: (a) Computational domain of coiled water heater, (b) grid used for the coiled water heater and (c) grid generated at the inlet and outlet surface of the helical pipe with inflation at edges.

#### Deliverable (1)

The values for $\Delta T = T_{out} - T_{in}$ (analytical and numerical) for the four different inlet velocities are given in the Table 1. The analytical $\Delta T$ is evaluated using Eq. 4 where, $A = 2 \cdot \pi \cdot r \cdot l$ which is the surface area, length of the helical pipe is evaluated as $l = \sqrt{R^2 + C^2} \cdot 10\pi$, r is the radius of pipe, $Q$ is the uniform energy input, $V$ is the volume of the helical pipe given as $V = l \cdot \pi \cdot r^2$, $\rho = 998.2\ \frac{kg}{m^3}$ and $C_p = 4182\ \frac{J}{kg \cdot K}$, $\Delta t = \frac{l}{V_{in}}$ s is the residence time for water. Figure 7 shows the plot for $\Delta T$ v/s $V_{in}$ where $\Delta T$ is the numerical value obtained by using Eq. 3.

$$\Delta T_{Analytical} = \frac{A \cdot Q}{V \cdot \rho \cdot C_p} \cdot \Delta t \tag{4}$$

Table 1: Values for $\Delta T$ for different inlet velocities $V_{in}$

| $V_{in} \frac{m}{s}$ | $\Delta T$ K (Analytical) | $\Delta T$ K (Numerical) | Error % |
|---|---|---|---|
| 0.01 | 5.66 | 6.12 | 8.12 |
| 0.02 | 2.83 | 3.14 | 10.95 |
| 0.04 | 1.41 | 1.61 | 14.18 |
| 0.08 | 0.70 | 0.83 | 18.57 |

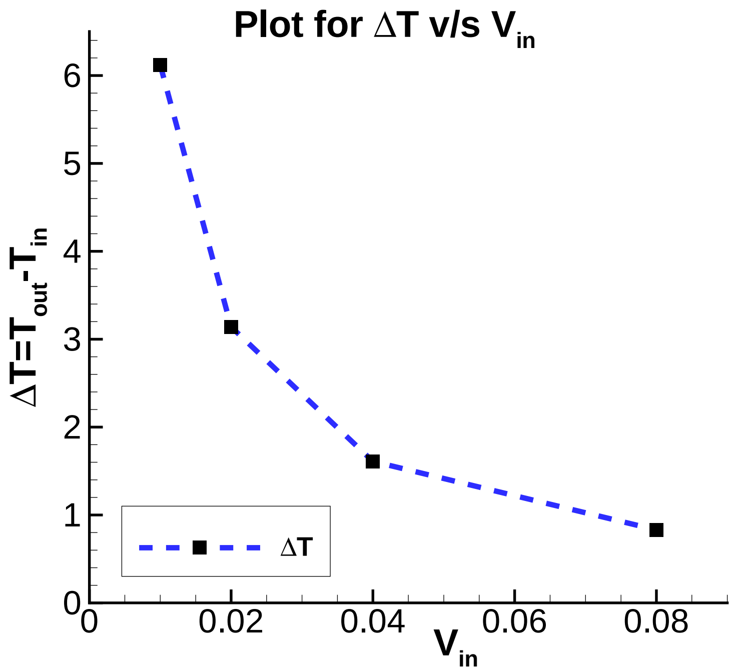


Figure 7: Plot for $\Delta T_{Numerical} = T_{out} - T_{in}$ v/s the inlet velocity $V_{in}$

As observed from the Eq. 4 of $\Delta T_{Analytical}$, there is an inverse relation between the temperature difference $\Delta T$ and the inlet velocity $V_{in}$. The similar relationship is also discernible from the numerical results in Table. 1 and Fig. 7 where the value of $\Delta T_{Numerical}$ is reducing with an increase in the inlet velocity $V_{in}$. This is mainly due to the fact that as the water moves at higher velocity inside the helical pipe, it will have a lesser residence time, which in fact, causes it to gain less amount of heat as compared to the water which moves at slower velocity inside helical pipe of same length and with constant heat flux at the wall.

**Deliverable (2)**

The contour plot for the velocity magnitude and temperature at the outlet surface for the case of $V_{in} = 0.04\frac{m}{s}$ is shown in Fig. 8.

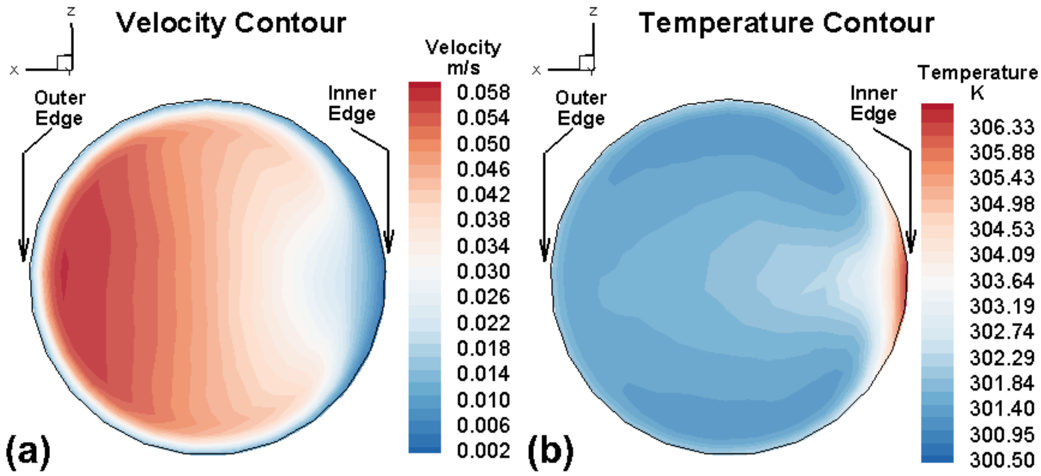


Figure 8: (a) Velocity magnitude at the outlet surface and (b) Temperature contour at the outlet surface.

As observed from the Fig. 8 (a), the velocity of water at the outer edge of the pipe is higher as compared to the velocity of water at the inner edge. This is mainly due to the fact that water particles nearer to the outer edge has to travel more distance in same interval of time as compared to those which travel nearer to the inner edge. Due to this phenomena, the heat gained by water nearer to the outer edge will be less as compared to heat gained by water nearer to inner edge. As a result, the temperature of water nearer to the outer edge will less as compared to the temperature of water nearer to the inner edge. The same is discernible from Fig. 8 (b).

## Task 3

To simulate the fluid flow and heat transfer in a chamber, with air as a working fluid. The geometrical parameters are: diameter of the chamber $D = 20$ cm, height of the chamber $H = 20$ cm, diameter of the outlet side pipe $d = 5$ cm, length of the side pipe is 10 cm. The boundary conditions are: the outlet side pipe is assigned as pressure outlet condition with 0 gauge pressure and $20^oC$ as backflow temperature. The bottom and top wall of the chamber is assigned as wall boundary condition with heat generation as $10\frac{W}{m^2}$. Laminar model is to be used in the present simulation. The density based solver is to be used in the present simulation and transient simulation is to be performed. The density to be set as 'Ideal Gas' for air.
To calculate the area weighted average velocity,$V_{out}$, normal to the surface of the outlet based on Eq. 5

$$V_{out} \equiv \frac{1}{A}\iint_A v\,dA \tag{5}$$

where,$\nu$ is the velocity component normal to the surface of the outlet.

### Solution

Figure 9 (a) shows the computational domain of chamber and Fig. 9 (b) shows the grid used at the plane of symmetry in the present simulation. In the present simulation the element size for grid generation is taken as 1 cm. Programmed controlled inflation of 5 layers with 1.2 growth rate is allowed at boundary of the chamber. Moreover, an additional refinement of the grid is provided nearer to the top and bottom wall of the chamber as observed from the Fig. 9 (b).

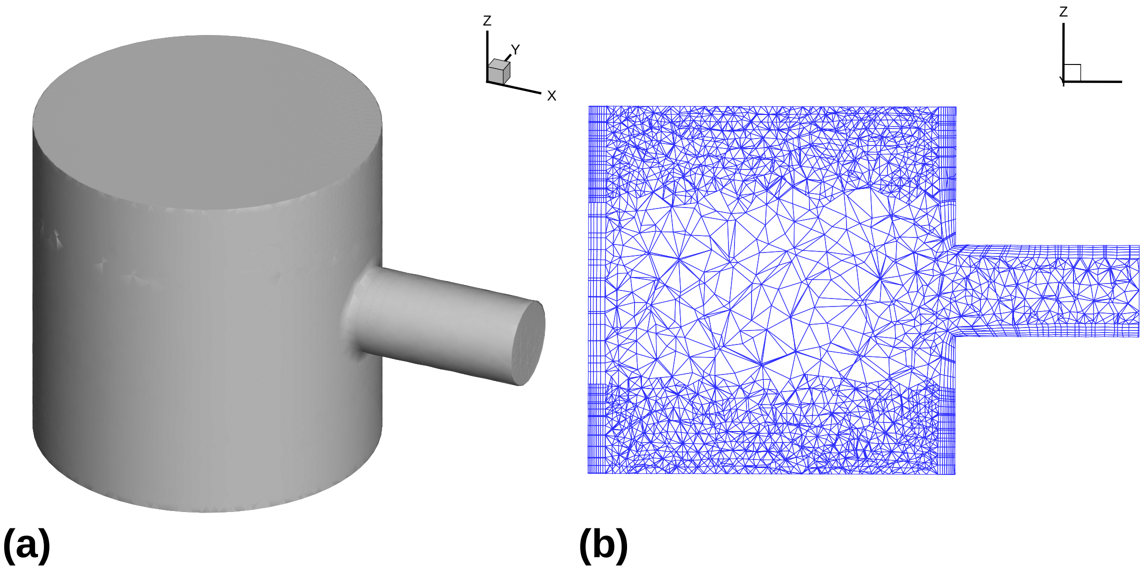


Figure 9: (a) Computational domain for the chamber, (b) grid used for the chamber (shown at plane of symmetry).

**Deliverable (1)**

The line plot for $V_{out}$ v/s time from $t = 0$ s to $t = 10$ s is shown in the Fig. 10. As the temperature of the air increase in the chamber it expands and as a result a continuous increase in the velocity (area weighted average) of the air, normal to the outlet, is observed. In the present simulation, the time step size is chosen as $\Delta t = 0.1$ s. The number of time steps is kept as 100 to achieves the solution till $t = 10$ s. The maximum iterations per time step is chosen as 20. The value of $V_{out}$ is 0.00113 $\frac{m}{s}$ after $t = 10$ s. A detail investigation was carried out by further refinement of the time step size ($\Delta t = 0.05, 0.01\, and\, 0.005$ s), however, the nature of curve for $V_{out}$ was found to be oscillating, which in fact suggested that a further refinement in the grid was required.

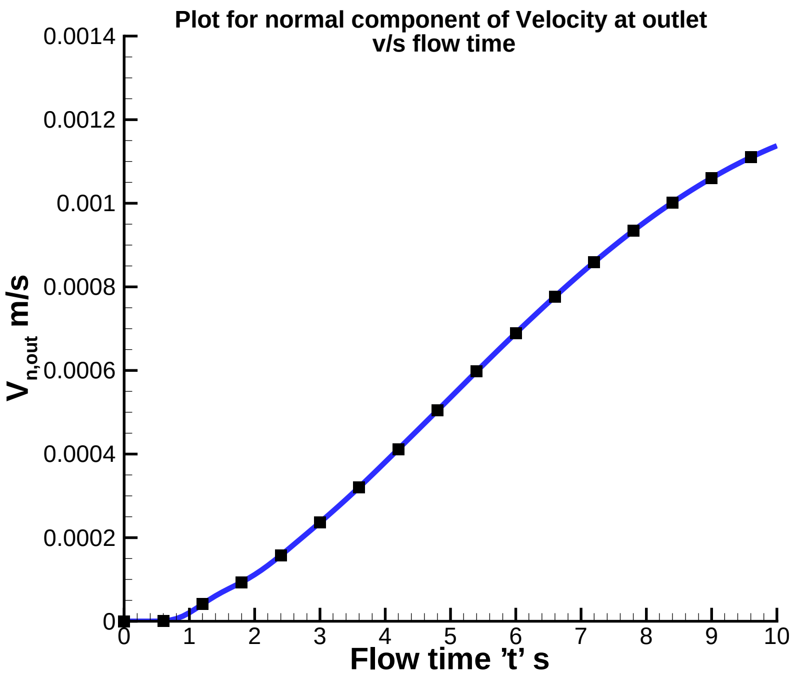


Figure 10: Line plot for $V_{out}$ v/s flow time $t$ from $t = 0$ s to $t = 10$ s.

**Deliverable (2)**

The pressure contour, velocity magnitude contour and the temperature contour are shown at the vertical plane of the symmetry at $t = 10$ s, in Fig. 11 (a), (b) and (c) respectively.

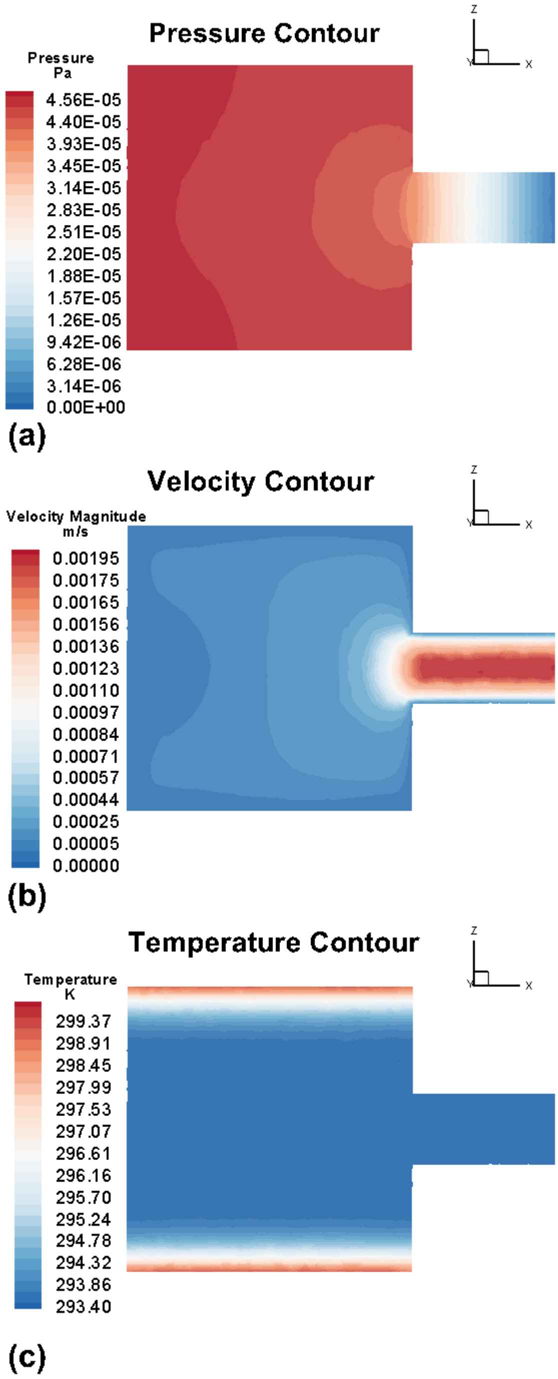


Figure 11: (a) Pressure Contour, (b) Velocity magnitude contour and (c) Temperature contour at the plane of symmetry.

## Task 4

To perform the task 3 invoking the concept of symmetry in 2 planes.

## Solution

### Deliverable (3)

Figure 12 (a) shows the computational domain of the quarter chamber and Fig. 12 (b) shows the grid used in the present simulation. In the present simulation the element size for grid generation is taken as 1 cm. Programmed controlled inflation of 5 layers with 1.2 growth rate is allowed at boundary of the chamber. Moreover, an additional refinement of the grid is provided nearer to the bottom wall of the chamber as observed from the Fig. 12 (b).

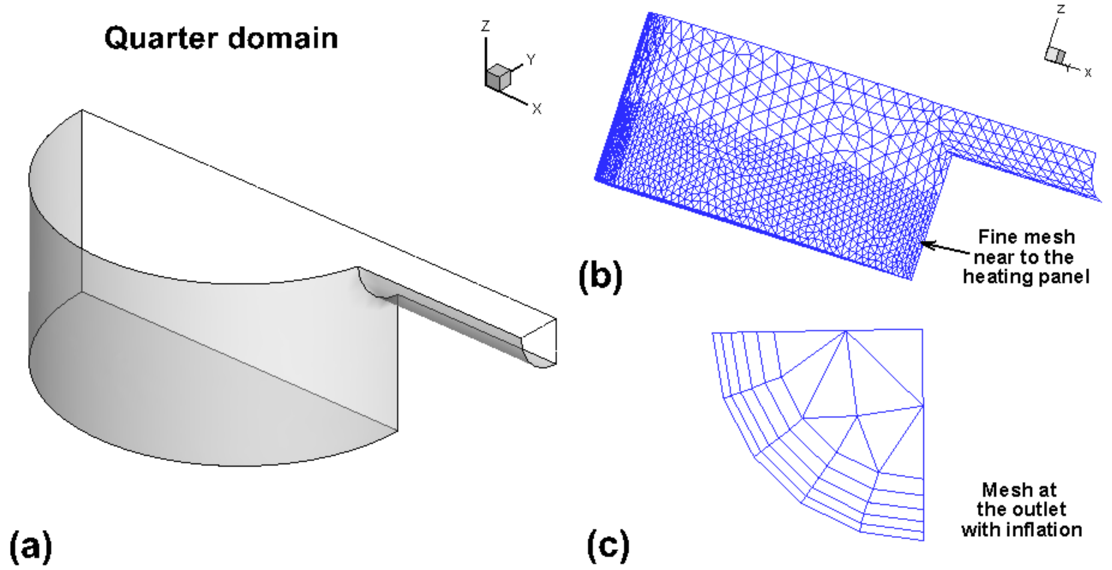


Figure 12: (a) Computational domain for the quarter chamber, (b) grid used for the chamber with refinement at the bottom wall and (c) mesh at the outlet surface of the side pipe.

### Deliverable (1)

The line plot for $V_{out}$ v/s time from $t = 0$ s to $t = 10$ s is shown in the Fig. 13. As the temperature of the air increase in the chamber it expands and as a result a continuous increase in the velocity (area weighted average) of the air, normal to the outlet, is observed. In the present simulation, the time step size is chosen as $\Delta t = 0.1$ s. The number of time steps is kept as 100 to achieves the solution till $t = 10$ s. The maximum iterations per time step is chosen as 20. The value of $V_{out}$ is 0.00107 $\frac{m}{s}$ after $t = 10$ s.

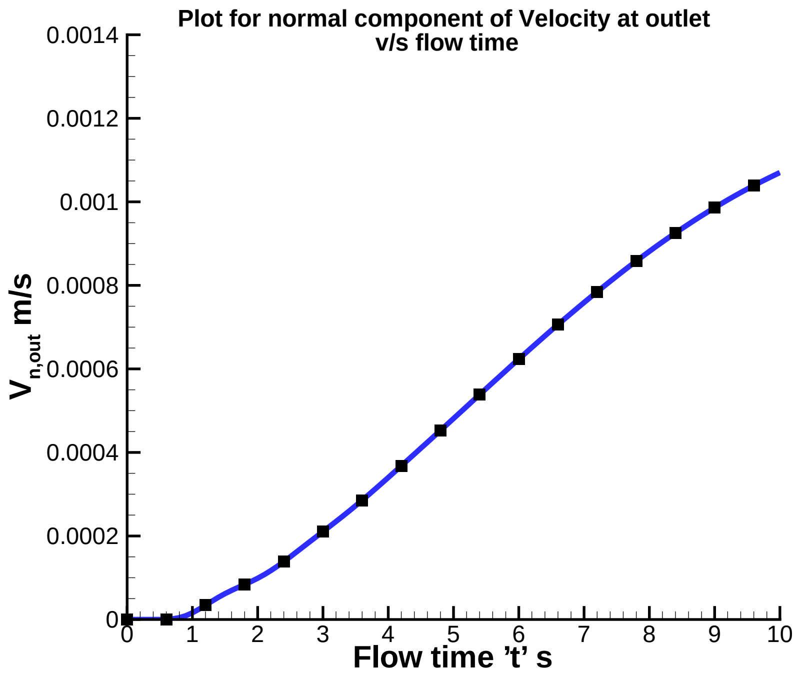


Figure 13: Line plot for $V_{out}$ v/s flow time $t$ from $t = 0$ s to $t = 10$ s.

**Deliverable (2)**

The pressure contour, velocity magnitude contour and the temperature contour are shown at the plane of the symmetry at $t = 10$ s, in Fig. 14 (a), (b) and (c) respectively. It is discernible that the results obtained in the present simulation over a quarter domain do corroborate with that obtained in task 3. However, there are very minor discrepancies in the result which are caused mainly due to difference between the grid generated in original computational domain and the grid generated in the quarter domain, and in addition to that, also due to some numerical error.

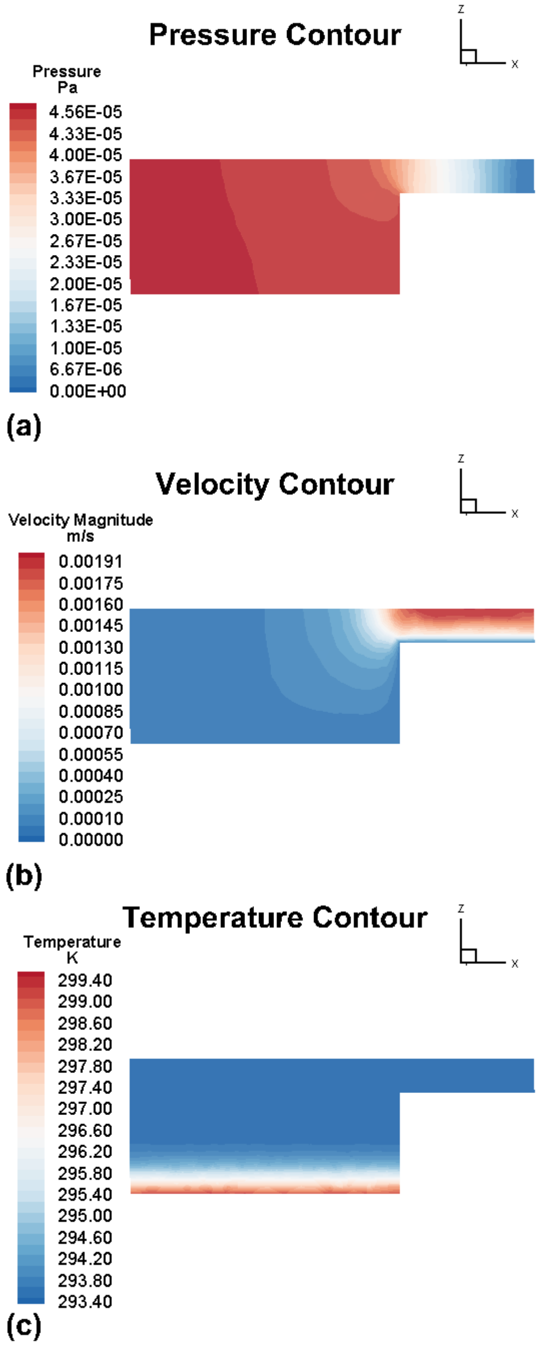


Figure 14: (a) Pressure Contour, (b) Velocity magnitude contour and (c) Temperature contour at the plane of symmetry.