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Design, Analysis and Optimization of Centrifugal Pump Impeller using CFD

Prof. Nitin Deshmukh[1] and Mr. Mohammed Ishak Shaikh[2]

Professor, Department of Mechanical Engineering[1]

Student, Department of Mechanical Engineering[2]

Rajiv Gandhi Institute of Technology, Mumbai, India

Abstract: *In this project work, head of centrifugal pump impeller, was obtained by analytical calculation and its validation is done by simulation result. Head results was obtained by keeping different impeller blade angles of 30°, 34°, 38°, 42°. By comparing head results, optimum blade angle for a particular impeller was obtained. Using the dimensions of the impeller, the 3-D model of the Centrifugal pump impeller was created in solidworks. Then Fluid domain is created from impeller model, in solidworks software. The fluid domain model is then converted into a parasolid file, and then imported in ANSYS Workbench. Then the rotation of 3000rpm is applied. Water is given as the rotating fluid. Then meshing is done, and solution is obtained as per CFD analysis.*

Keywords: Centrifugal Pump, Impeller, Head, Pressure, Velocity, Computational Fluid Dynamics

I. INTRODUCTION

PUMP: The Hydraulic Machines which convert mechanical energy consumed by the shaft into pressure energy of the fluid are called pumps.

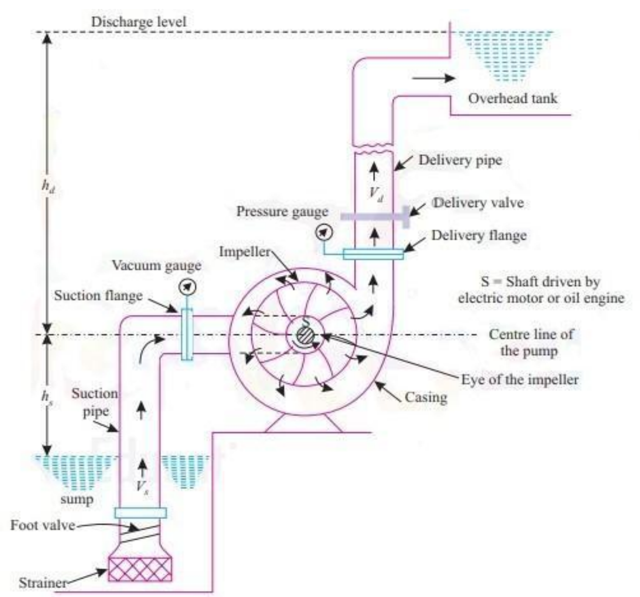

Fig 1. Centrifugal Pump Components

There are two main parts of centrifugal pump namely : 1- Impeller 2- Casing

Impeller: The rotating part of centrifugal pump is called impeller, it consists of vanes. The impeller is mounted on a shaft which is connected to a motor. Impeller imparts kinetic energy and pressure energy, to the fluid by consuming shaft work.

Impeller has two functions to perform:

1. To increase the absolute kinetic energy of the fluid, by imparting velocity to the fluid

2. To increase pressure of fluid, and this is achieved by passing the fluid through the diverging vane passages of the impeller.

In pump, about 50% of pressure increase is achieved in impeller, the remaining 50% pressure increase is achieved in casing.

Casing: The casing is an air tight passage surrounding the impeller, and it has gradually increasing area of flow. In casing the kinetic energy of the liquid, discharged at the outlet of the impeller, is converted into pressure energy. In casing as the area of flow increases, the kinetic energy decreases and the pressure of liquid increases.

II. METHODOLOGY

Methodology:

Methodology is very important to investigate available problem. Therefore it is necessary to define problem domain, physical model, physical properties and boundary conditions. Then it need to solve the problem and compute the results in a correct way. In this present work 3D model is created in Solidworks and the Meshing and solving is carried out in Ansys-CFX Turbo machinery-Solver. Accordingly Moving Reference Frame is selected for the Fluid Domain which is extracted from geometry, because solid-geometry is not directly used for computation and the fluid selected as water. The steps for methodology is shown below.

Geometrical Modelling of impeller in Solidworks

⇩

Meshing in Ansys

⇩

Assigning Boundary Conditions and solving in CFX

⇩

Getting results from Ansys

⇩

Repeating above steps for different blade angles of impeller

Fig 2. Methodology flowchart

III ANALYTICAL AND SIMULATION WORK

3.1 Governing Equations:

Mathematical modeling is the actual representation of any system in mathematical form. Mathematical modeling is used in the simulation. The Continuity equations and Navier-Stokes equation are used. Also for solving the mathematical form boundary conditions are applied to geometry to specify the standard parameters. Boundary conditions have importance in CFD because the applicability of numerical methods and the simultion result depends on it.

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Incompressible continuity equation: $$\frac{1}{r}\frac{\partial(ru_r)}{\partial r} + \frac{1}{r}\frac{\partial(u_\theta)}{\partial \theta} + \frac{\partial(u_z)}{\partial z} = 0$$

r-component of the incompressible Navier–Stokes equation:

$$\rho\left(\frac{\partial u_r}{\partial t} + u_r\frac{\partial u_r}{\partial r} + \frac{u_\theta}{r}\frac{\partial u_r}{\partial \theta} - \frac{u_\theta^2}{r} + u_z\frac{\partial u_r}{\partial z}\right)$$
$$= -\frac{\partial P}{\partial r} + \rho g_r + \mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_r}{\partial r}\right) - \frac{u_r}{r^2} + \frac{1}{r^2}\frac{\partial^2 u_r}{\partial \theta^2} - \frac{2}{r^2}\frac{\partial u_\theta}{\partial \theta} + \frac{\partial^2 u_r}{\partial z^2}\right]$$

θ-component of the incompressible Navier–Stokes equation:

$$\rho\left(\frac{\partial u_\theta}{\partial t} + u_r\frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r}\frac{\partial u_\theta}{\partial \theta} + \frac{u_r u_\theta}{r} + u_z\frac{\partial u_\theta}{\partial z}\right)$$
$$= -\frac{1}{r}\frac{\partial P}{\partial \theta} + \rho g_\theta + \mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_\theta}{\partial r}\right) - \frac{u_\theta}{r^2} + \frac{1}{r^2}\frac{\partial^2 u_\theta}{\partial \theta^2} + \frac{2}{r^2}\frac{\partial u_r}{\partial \theta} + \frac{\partial^2 u_\theta}{\partial z^2}\right]$$

z-component of the incompressible Navier–Stokes equation:

$$\rho\left(\frac{\partial u_z}{\partial t} + u_r\frac{\partial u_z}{\partial r} + \frac{u_\theta}{r}\frac{\partial u_z}{\partial \theta} + u_z\frac{\partial u_z}{\partial z}\right)$$
$$= -\frac{\partial P}{\partial z} + \rho g_z + \mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_z}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 u_z}{\partial \theta^2} + \frac{\partial^2 u_z}{\partial z^2}\right]$$

Fig 3. Continuity Equation and and Navier-Stokes equation in Cylindrical Coordinates

3.2 Boundary Conditions and other parameters:

The boundary conditions are selected as per the application. Here for the pump impeller, only two types of boundary conditions are used, i.e. pressure at inlet and discharge at outlet. The side walls are selected as smooth wall with no slip condition. The no slip condition is used on corrugation wall, to have zero velocity of fluid at surface.

Sr. No	Set Parameter	Set Value
1	Fluid	Water
2	Rotation	3000 rpm
3	Rotation axis	Global Y
4	Inlet pressure	-9810 Pascal
5	Wall	No slip wall
6	Solver	Max. iterations 600
7	Length scale	Aggressive

Table 1. Parameters

3.3 ANALYTICAL CALCULATIONS:

Dimensions and data of existing impeller:

Outlet Diameter of impeller = D_2 = 73mm

Eye Diameter (inlet diameter) of impeller = D_1 = 38mm

Hub diameter = 18mm

Speed of rotation = N = 3000 rpm

Width of impeller at outlet = B_2 = 10mm

Discharge = Q_2 = 0.6 liters/sec = $0.6 * 10^{-3} \frac{m^3}{s}$

Blade angle at outlet (also called as outlet angle) = Φ = 34 degrees

Blade angle at inlet (also called as inlet angle) = θ = 34.29 degrees

Calculations of Theoretical head:

Blade velocity at outlet = U_2

$$U_2 = \frac{\pi . D_2 . N}{60} = \frac{\pi * 0.073 * 3000}{60}$$

U_2 = 11.467 m/s

'Theoretical head' OR 'Work done by the impeller on water' OR 'Head imparted by the impeller to water' is given by

$$H_{theoretical} = \frac{V_{w2} . U_2 - V_{w1} . U_1}{g}$$

But for centrifugal pump, $V_{w1} = 0$

Therefore $H_{theoretical} = \frac{V_{w2} . U_2}{g}$

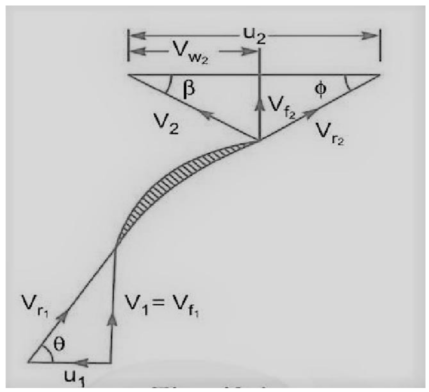

Fig 4. Velocity diagram

From velocity diagram: $V_{w2} = U_2 - V_{f2} . cot\Phi$

Substituting in head equation we get,

$H_{theoretical} = \frac{U_2^2}{g} - \frac{U_2 . cot\Phi}{g} . V_{f2}$

Using, Discharge = Area of flow x Velocity

We get

$$H_{theoretical} = \frac{U_2^2}{g} - \frac{U_2 . cot\Phi}{g} . \frac{Q}{\pi . D_2 . B_2}$$

$$H_{theoretical} = \frac{11.467^2}{9.81} - \frac{11.467 * \cot 34}{9.81} . \frac{0.6 * 10^{-3}}{\pi * 0.073 * 0.01}$$

Substituting all the values in above formula, we get

$H_{theoretical}$ = 12.9506 meter

This is the theoretical head, without accounting any losses.

Similarly, calculating theoretical head for different outlet angles and keeping other dimensional parameters same:

Sr. No.	Outlet angle (deg)	Theoretical Head (m)
1	30	12.8743
2	34	12.9506
3	38	13.0126
4	42	13.0644

Table 2. Theoretical head for different outlet angles

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3.4 MODELLING AND SIMULATION WORK:

3.4.1 Modelling of impeller:

Using the dimensions of the impeller, the 3-D model of the Centrifugal pump impeller was created in solidworks as shown in below figure

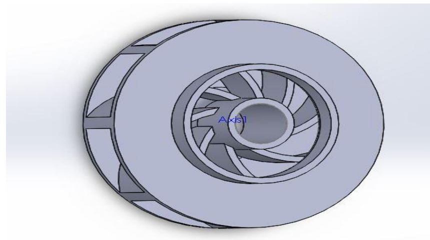

Fig 4. Modelled impeller

3.4.2 Creating fluid domain:

Then Fluid domain is created from impeller model, in solidworks software as shown in below figure

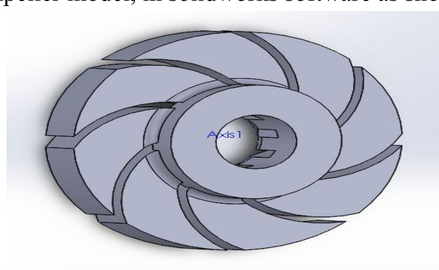

Fig 5. Fluid domain of impeller

3.4.3 Defining inlet, outlet and wall of the impeller:

In Ansys software, while defining the inlet of the impeller it highlights that area with green color as shown in below image.

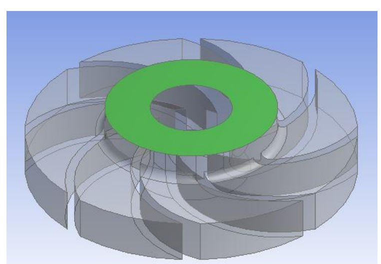

Fig 6. Inlet area of impeller

In Ansys software, while defining the outlet of the impeller it highlights that area with green color as shown in below image.

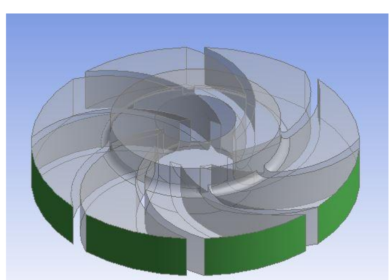

Fig 7. Outlet area of impeller

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In Ansys software, while defining the wall of the impeller it highlights that area with green color as shown in below image.

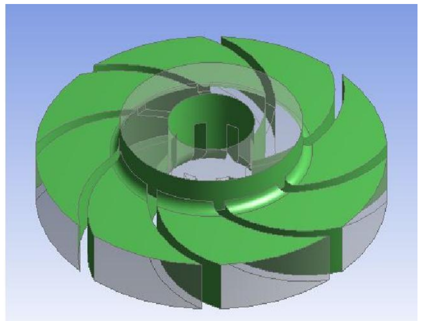

Fig 8. Wall area of impeller

3.4.4 Meshing:

The tetrahedral typed mesh is done. At steeper curves of impeller the tetrahedral shapes of the mesh are small, hence there will be more number of nodes, this can be seen from the meshed image, because of that the results obtained will be more accurate.

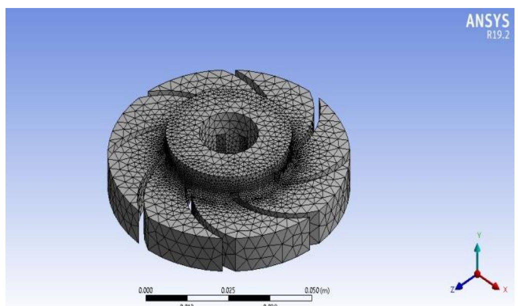

Fig 9. Meshing of impeller

3.4.5 CFD Analysis and Results:

The fluid domain model is then converted into a parasolid file, and then imported in ANSYS Workbench. Then the rotation of 3000rpm is applied. Water is given as the rotating fluid. Then as per CFD analysis meshing is done, and solution is obtained.

In pump impeller, the pressure and velocity should go on increasing from inlet (inner radius) to outlet (outer radius). The 'pressure result at mid-plane' and 'velocity result at mid-plane' shown in fig 5.7 and fig 5.8 for one particular blade angle is proving this point. While performing simulation the mid-plane result was checked for all the blade angles for which simulations are performed, but these mid-plane result is shown only for only blade angle to reduce the unnecessary increase in this report length.

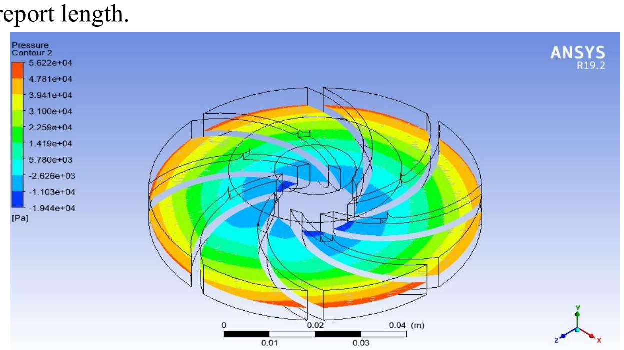

Fig 10. Pressure contour plot at mid plane for impeller outlet angle 34 degree

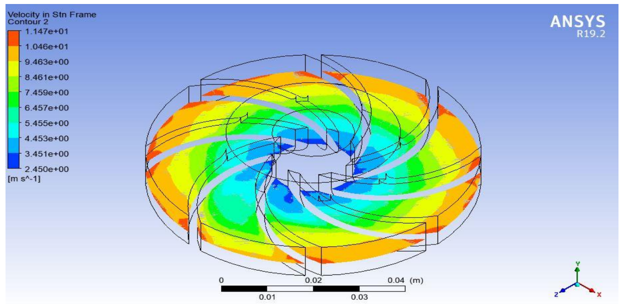

Fig 11. Velocity contour plot at mid plane for impeller outlet angle 34 degree

3.4.5.1 Simulation results for impeller outlet angle 34 degree:

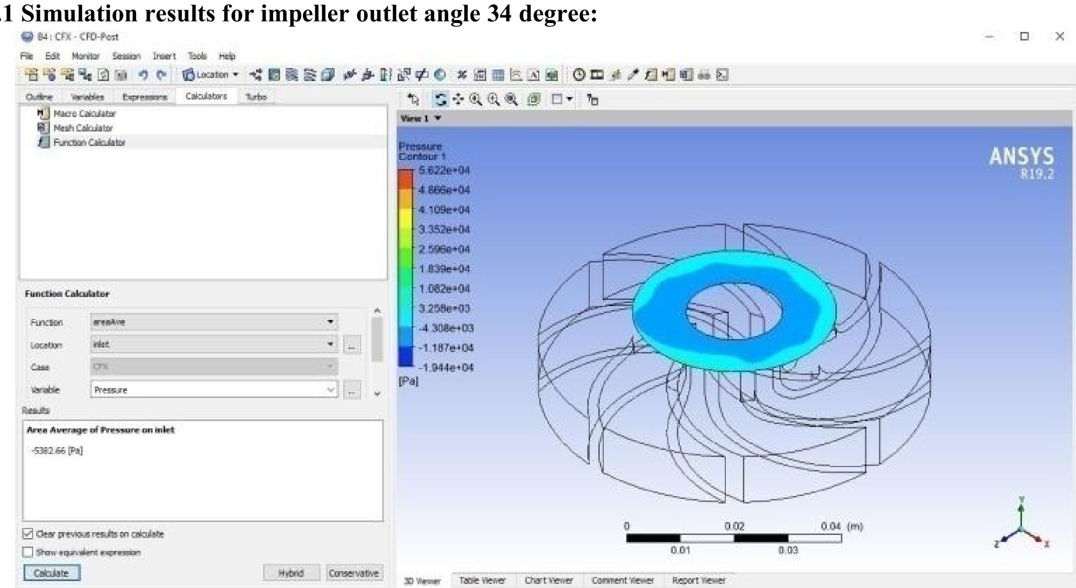

Fig 12. Area average Pressure at inlet = -5382.66 Pa = P_1 for impeller outlet angle 34 degree

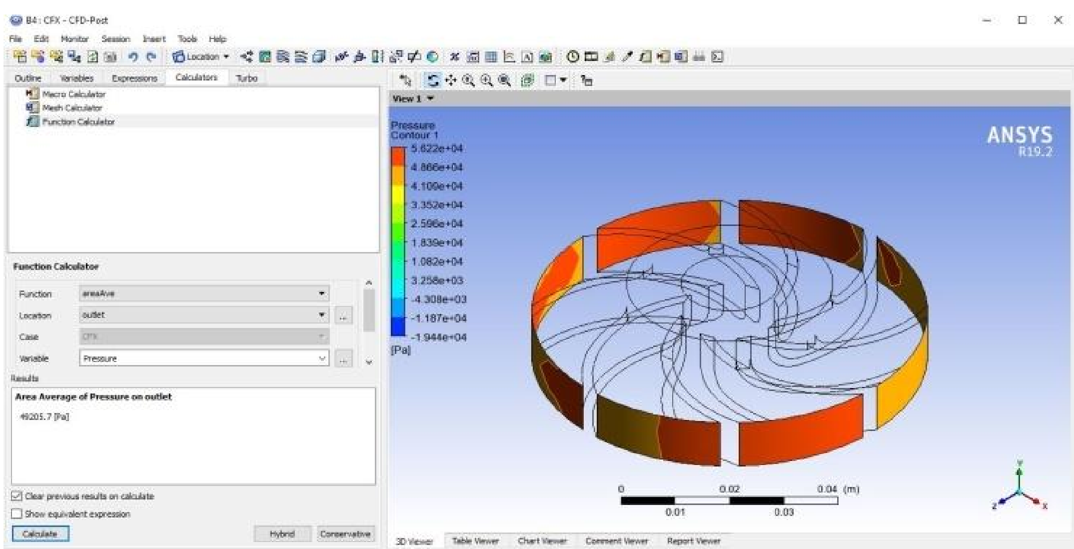

Fig 13. Area average Pressure at outlet = 49205.7 Pa = P_2 for impeller outlet angle 34 degree

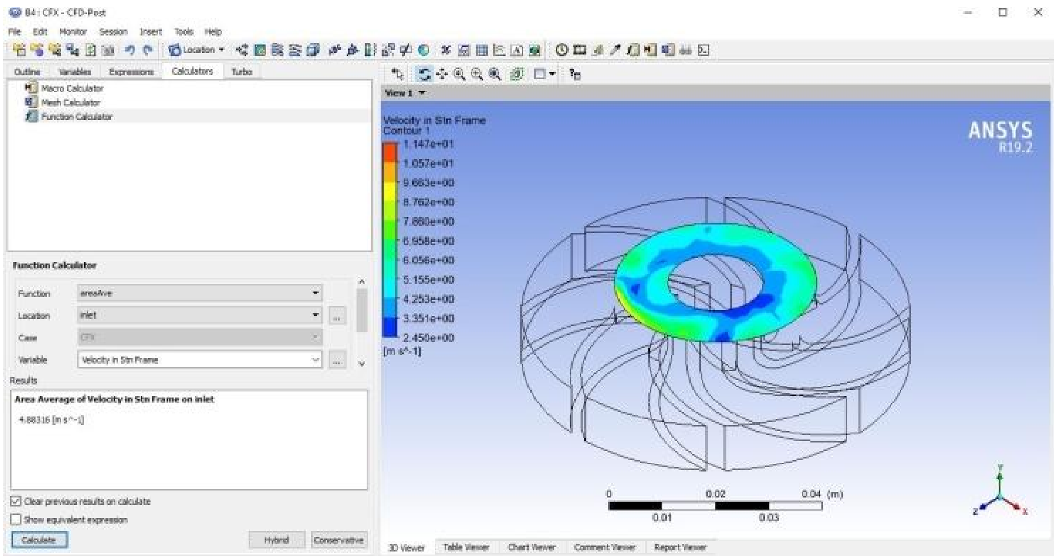

Fig 14. Area average Velocity at inlet = 4.88316 m/s = V_1 for impeller outlet angle 34 degree

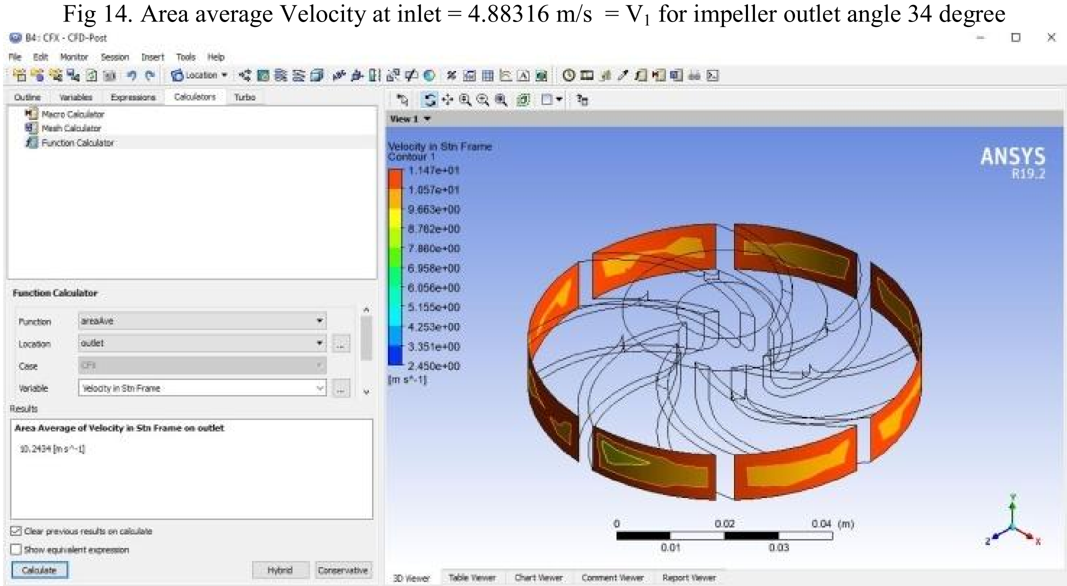

Fig 15. Area average Velocity at outlet = 10.2434 m/s = V_2 for impeller outlet angle 34 degree

Calculating Head from Simulation results for impeller outlet angle 34 degree:

Head= $H = \left(\frac{P_2}{\rho . g} + \frac{V_2^2}{2 . g} + Z_2\right) - \left(\frac{P_1}{\rho . g} + \frac{V_1^2}{2 . g} + Z_1\right)$

For impeller, $(Z_2 - Z_1) = 0$

Head= $H = \left(\frac{P_2 - P_1}{\rho . g}\right) + \left(\frac{V_2^2 - V_1^2}{2 . g}\right) = \left(\frac{49205.7 + 5382.66}{1000 * 9.81}\right) + \left(\frac{10.2434^2 - 4.88316^2}{2 * 9.81}\right)$

Substituting values obtained from simulation results we get

Head= H = 9.697 m

This is the head obtained from simulation result, by accounting losses like inlet loss, exit loss, bend loss, fluid friction loss, skin friction loss, eddy loss etc.

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3.4.5.2 Simulation results for impeller outlet angle 30 degree:

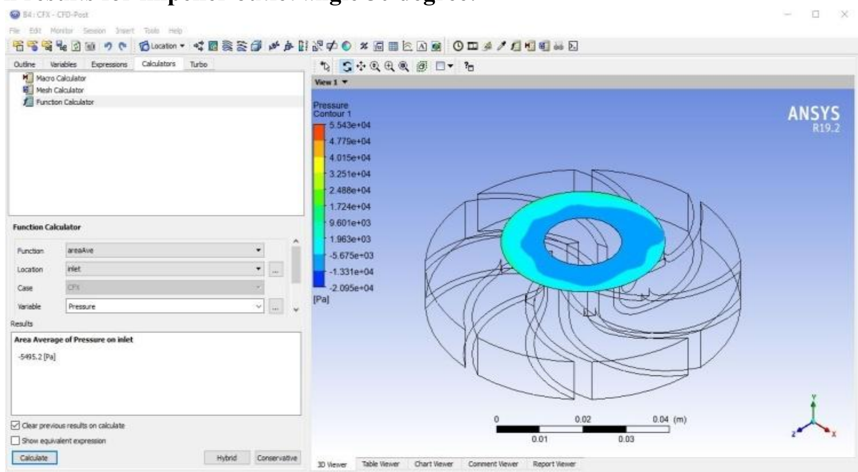

Fig 16. Area average Pressure at inlet = -5495.2 Pa = P_1 for impeller outlet angle 30 degree

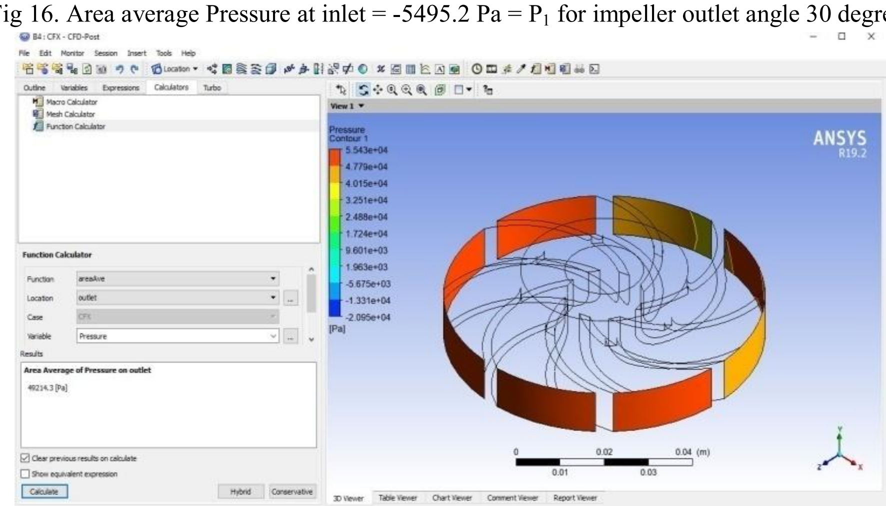

Fig 17. Area average Pressure at outlet = 49214.3 Pa = P_2 for impeller outlet angle 30 degree

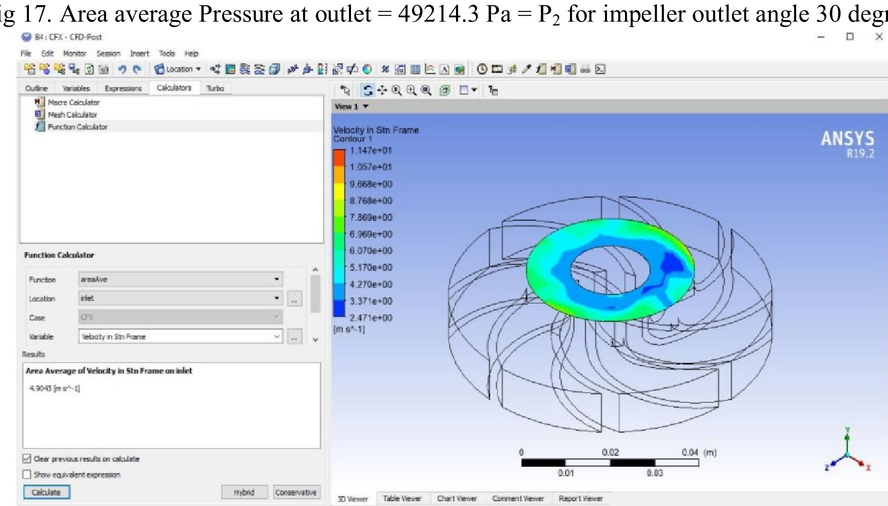

Fig 18. Area average Velocity at inlet = 4.9043 m/s = V_1 for impeller outlet angle 30 degree

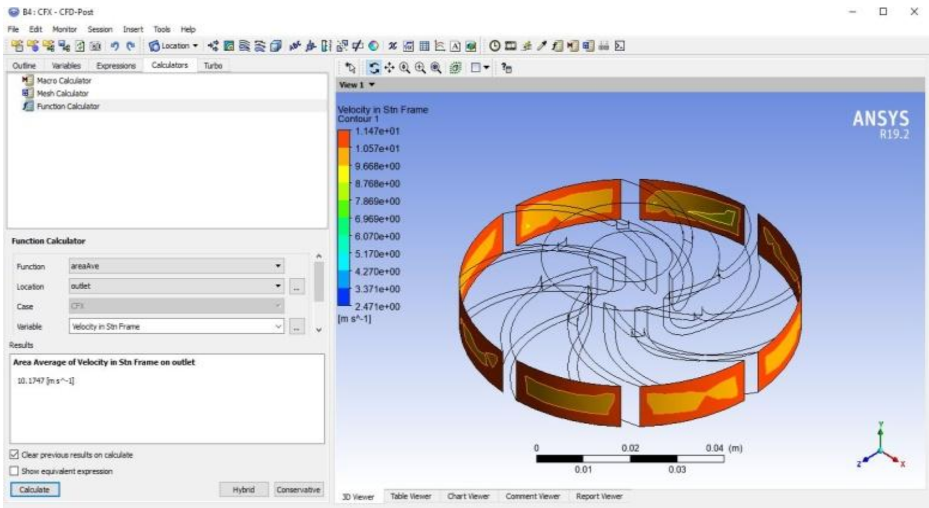

Fig 19. Area average Velocity at outlet = 10.1747 m/s = V_2 for impeller outlet angle 30 degree

Calculating Head from Simulation results for impeller outlet angle 30 degree :

$$\text{Head}= H = \left(\frac{P_2}{\rho .g} + \frac{V_2^2}{2 .g} + Z_2\right) - \left(\frac{P_1}{\rho .g} + \frac{V_1^2}{2 .g} + Z_1\right)$$

For impeller, $(Z_2 - Z_1) = 0$

$$\text{Head}= H = \left(\frac{P_2 - P_1}{\rho .\ g}\right) + \left(\frac{V_2^2 - V_1^2}{2 .g}\right) = \left(\frac{49214.3 + 5495.2}{1000 * 9.81}\right) + \left(\frac{10.1747^2 - 4.9043^2}{2 * 9.81}\right)$$

Substituting values obtained from simulation results we get

Head= H = 9.6275 m

3.4.5.3 Simulation results for impeller outlet angle 38 degree:

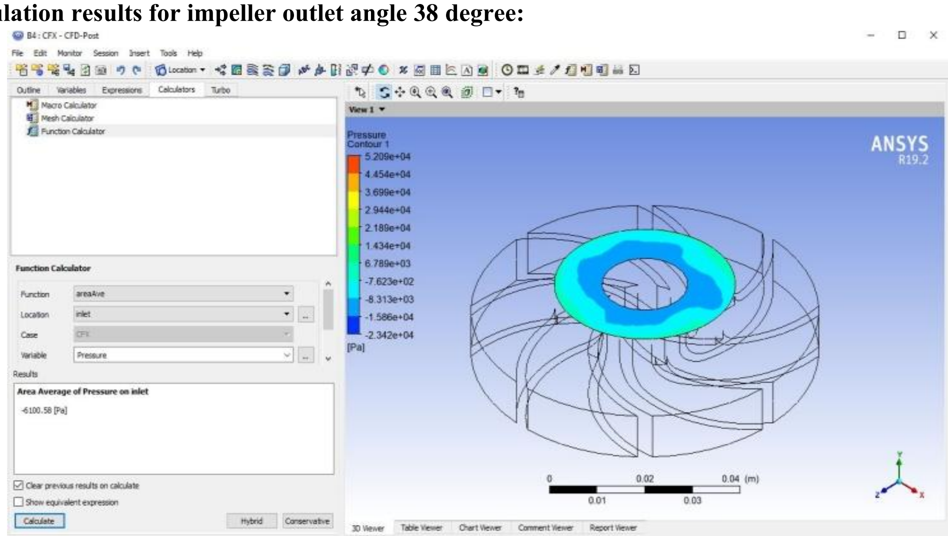

Fig 20. Area average Pressure at inlet = -6100.58 Pa = P_1 for impeller outlet angle 38 degree

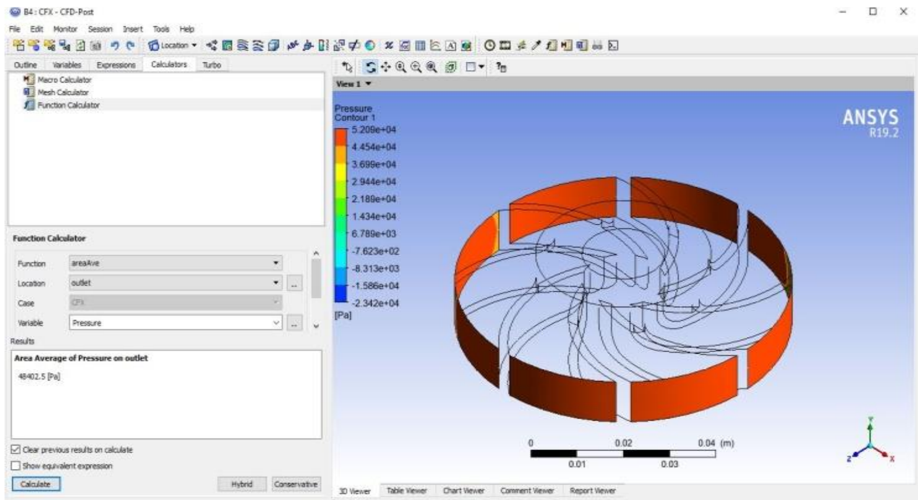

Fig 21. Area average Pressure at outlet = 48402.5 Pa = P_2 for impeller outlet angle 38 degree

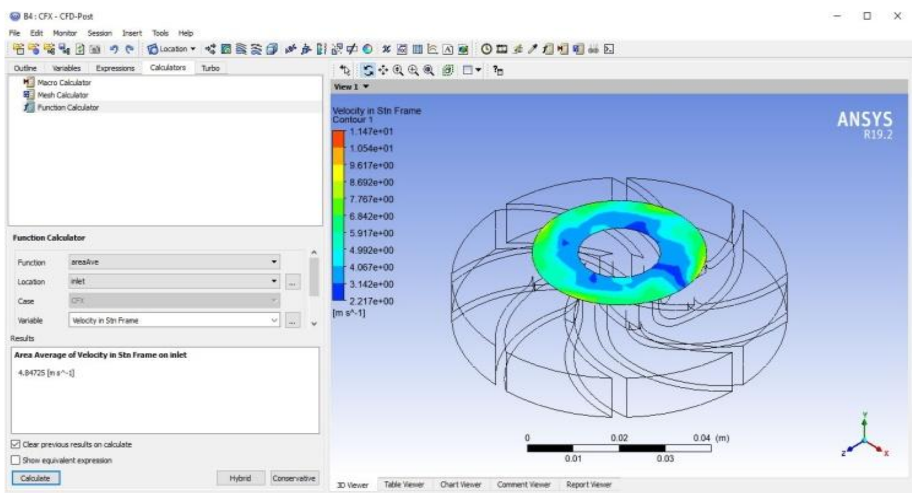

Fig 22. Area average Velocity at inlet = 4.84725 m/s = V_1 for impeller outlet angle 38 degree

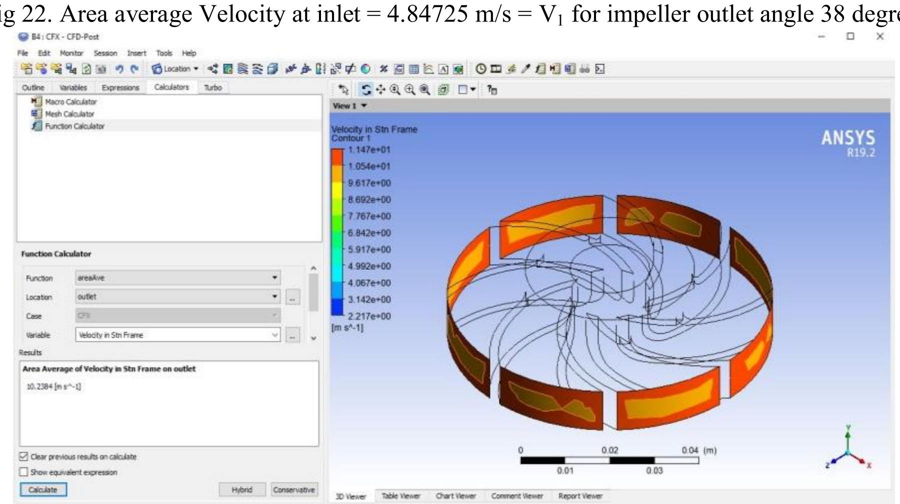

Fig 23. Area average Velocity at outlet = 10.2384 m/s = V_2 for impeller outlet angle 38 degree

Calculating Head from Simulation results for impeller outlet angle 38 degree :

Head= $H = \left(\frac{P_2}{\rho .g} + \frac{V_2^2}{2 .g} + Z_2 \right) - \left(\frac{P_1}{\rho .g} + \frac{V_1^2}{2 .g} + Z_1 \right)$

For impeller, $(Z_2 - Z_1) = 0$

Head= $H = \left(\frac{P_2 - P_1}{\rho .\ g} \right) + \left(\frac{V_2^2 - V_1^2}{2 .g} \right) = \left(\frac{48402.5 + 6100.58}{1000 * 9.81} \right) + \left(\frac{10.2384^2 - 4.84725^2}{2*9.81} \right)$

Substituting values obtained from simulation results we get

Head= H = 9.7011 m

3.4.5.4 Simulation results for impeller outlet angle 42 degree:

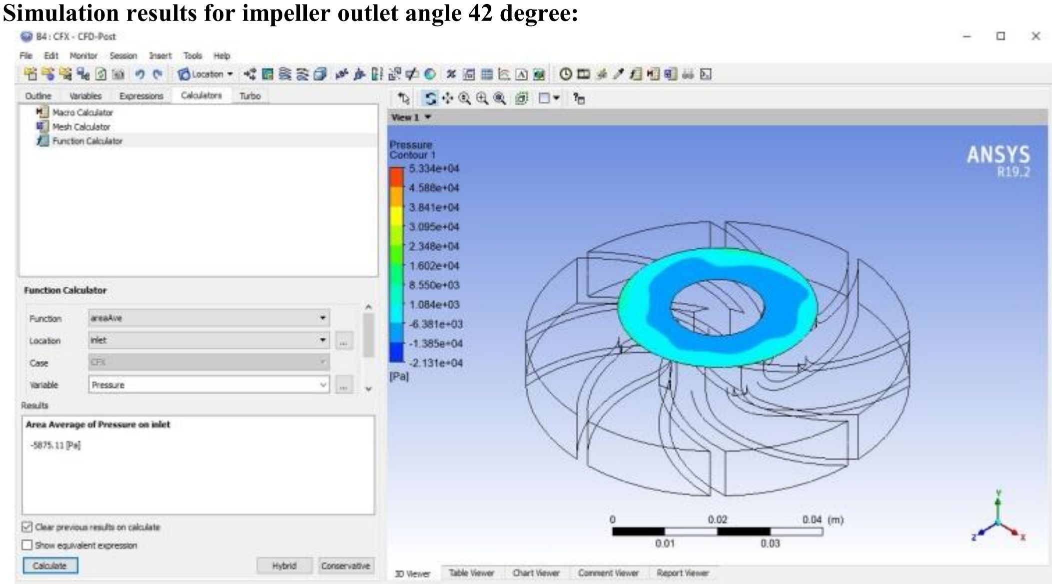

Fig 24. Area average Pressure at inlet = -5875.11 Pa = P_1 for impeller outlet angle 42 degree

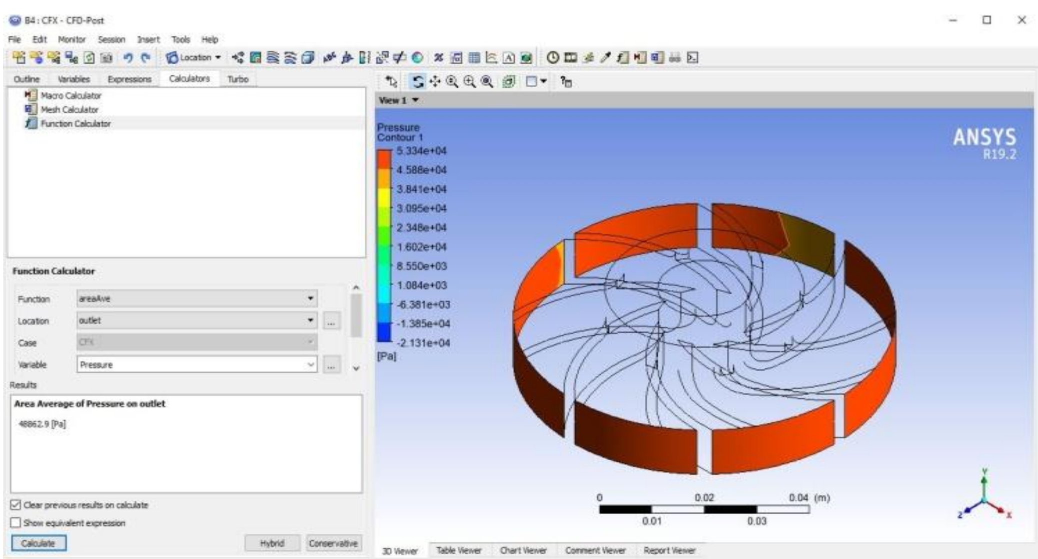

Fig 25. Area average Pressure at outlet = 48862.9 Pa = P_2 for impeller outlet angle 42 degree

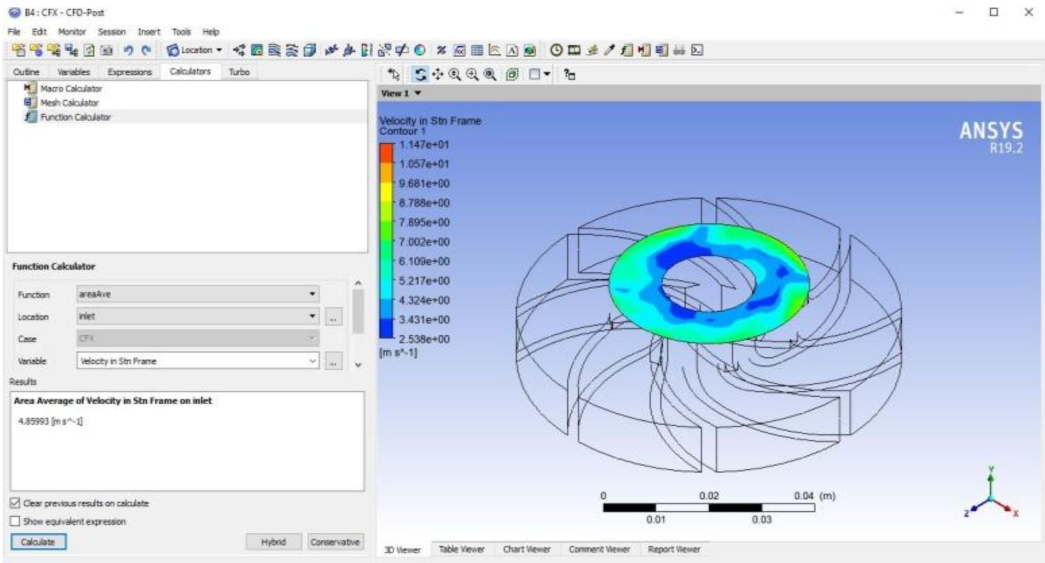

Fig 26. Area average Velocity at inlet = 4.85993 m/s = V_1 for impeller outlet angle 42 degree

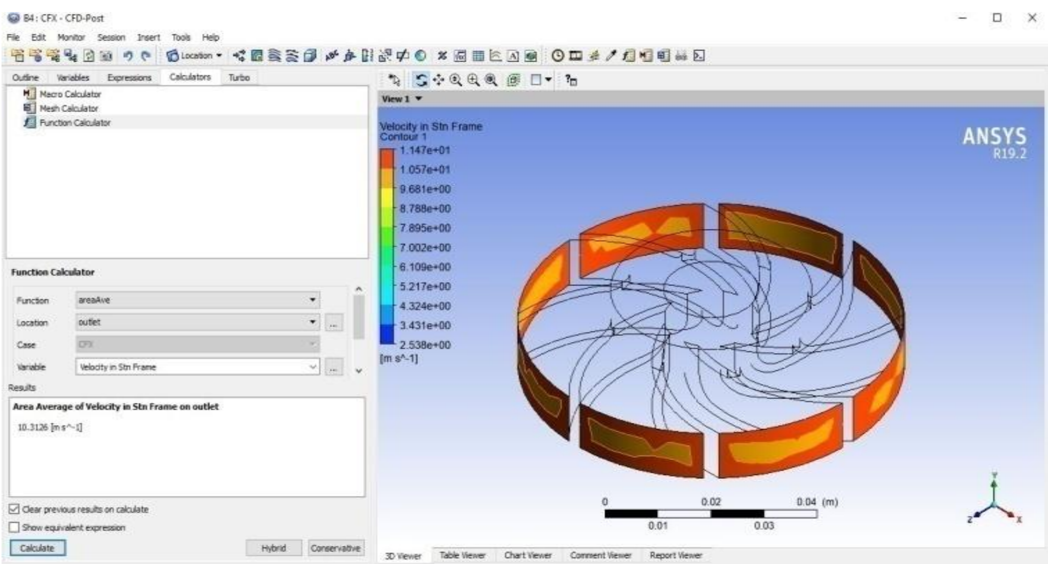

Fig 27. Area average Velocity at outlet = 10.3126 m/s = V_2 for impeller outlet angle 42 degree

Calculating Head from Simulation results for impeller outlet angle 42 degree :

Head= $H = \left(\frac{P_2}{\rho . g} + \frac{V_2^2}{2 . g} + Z_2 \right) - \left(\frac{P_1}{\rho . g} + \frac{V_1^2}{2 . g} + Z_1 \right)$

For impeller, $(Z_2 - Z_1) = 0$

Head= $H = \left(\frac{P_2 - P_1}{\rho . g} \right) + \left(\frac{V_2^2 - V_1^2}{2 . g} \right) = \left(\frac{48862.9 + 5875.11}{1000 * 9.81} \right) + \left(\frac{10.3126^2 - 4.85993^2}{2 * 9.81} \right)$

Substituting values obtained from simulation results we get

Head= H = 9.7964 m

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Sr. No.	Outlet angle (deg)	Head obtained (m)
1	30	9.6275
2	34	9.6970
3	38	9.7011
4	42	9.7964

Table 3. Simulation head results for different outlet angles

IV. RESULTS AND CONCLUSION

The theoretical head, obtained from analytical calculation is 12.9506 metre, without accounting any losses.

The head obtained from simulation result, is 9.697 metre by accounting losses like inlet loss, exit loss, bend loss, fluid friction loss, skin friction loss, eddy loss etc.

Sr. No.	Outlet angle (degree)	Head Obtained (meter)	Theoretical head (meter)
1	30	9.6275	12.8743
2	34	9.6970	12.9506
3	38	9.7011	13.0126
4	42	9.7964	13.0644

Table 4. Head results for different blade angles

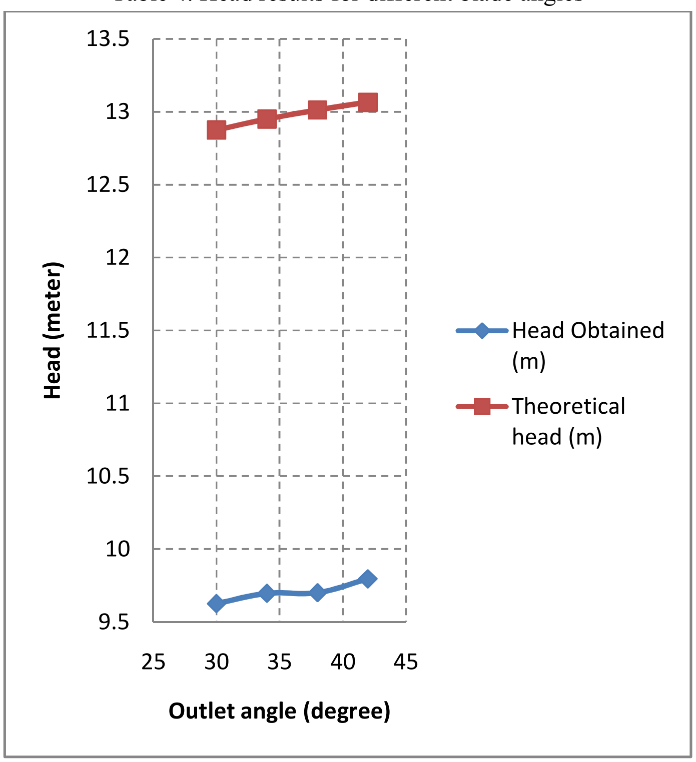

Fig 28. head versus outlet angle

It is observed from the theoretical head calculations and head obtained from Ansys analysis that as the outlet angle increases, the theoretical head and head obtained from Ansys also increases.

At outlet angle of 42 degree, the theoretical head of 13.0644m and head obtained from Ansys of 9.7964m is maximum.

The 'pressure difference of inlet and outlet' and the 'velocity difference of inlet and outlet' goes on increasing as the outlet angle increases.

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