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DESIGN AND CFD ANALYSIS OF SUBMERSIBLE PUMP IMPELLER OF MIXED FLOW TYPE FOR PRFOMANCE IMPROVEMENT

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***Abstrac*t-** When we deal with any mechanical engineering system, it needs great consumption of electricity and in the era of energy crisis, effective use of energy sources is must. Country like China and India are the biggest users of energy sources and mechanical equipments like pump has very wide range of applications in industry as well as domestically. In order to save energy one of the most preferred ways is to improve performance of the device for the same energy consumption. In this work focus is on the impeller of the mixed flow type submersible pump. In order to improve the efficiency of the pump need to work at the design level and find the parameters that affects the design and performance of the pump. Here the design of impeller of the mixed flow type submersible pump with the best efficiency point is compared with the existing impeller layout and it is found that the change in inlet and exit angle of blade and reducing number of blade have significant improvement on the performance of the pump. This improved design impeller will be checked by CFD analysis with existing impeller with same boundary condition.

Keywords-*Mixed flow pump; efficiency; design parameters; inlet and outlet blade angle; number of blades; CFD analysis.*

I. INTRODUCTION

Pump is classified according to its mechanism based on reciprocating and rotary. A mixed flow pump is a centrifugal type of pump working on the rotary mechanism. Centrifugal is classified based on the impeller imparts energy to water to transform velocity in to head or liquid enters and leaves the casing. Based on the impeller configuration and specific speed the pump is classified traditionally as radial flow, axial flow and mixed flow pump.

Figure 1. Types of pump [1]

1.1 Mixed flow pump:

Mixed flow pump where Head is developed partially by centrifugal action and partially by lift of the vane on liquid. The liquid enters axially and discharges in radial and axial direction. The casing of the pump may be volute type of diffusion type. They may have single or double suction and may be multistage. The mixed flow pump is applicable for medium head application (7-16m) and for medium or large application.

1.2 Impeller:

Impeller is important parts of any submersible pump as whole performance of the pump is depend upon geometrical parameters of an impeller. Performance parameters like head, discharge, hydraulic and volumetric efficiency can be improved by changing the design of pump in order to change the design of impeller for same bowl design.

1.3 Submersible Pump:

Produced liquids, after being subjected to great centrifugal forces caused by the high rotational speed of the impeller, lose their kinetic energy in the diffuser where a conversion of kinetic to pressure energy takes place. This is the main operational mechanism of radial and mixed flow pumps.

II .ANALYTICAL DESIGN OF IMPELLER LAYOUT

As efficiency of the selected impeller that is of submersible pump YCQ 125,of mixed flow type chosen for study, does not give required performance, in addition to that we need to redesign the impeller at best efficiency point for given value of input data's.

2.1 Input data for impeller design

QUANTITY	SYMBOL	VALUE	UNIT
HEAD	H	46	Meter
DISCHARGE	Q	0.0078	m^{3}/ sec.

SPEED	n	2880	Rpm
STAGES	m	6	No.

For selection of new impeller layout we have to first calculate the specific speed in u.s customary unit of the pump for above values of head, discharge and speed.

$$N = \frac{\omega\sqrt{Q}}{(gH)^{0.75}} \qquad (1) \qquad\qquad N_{(u.s)} = \frac{n\sqrt{Q}}{H^{0.75}} \qquad (2)$$

From above equation specific speed of the pump is found as N=0.9243 and $N_{(u.s)}$=2529.21

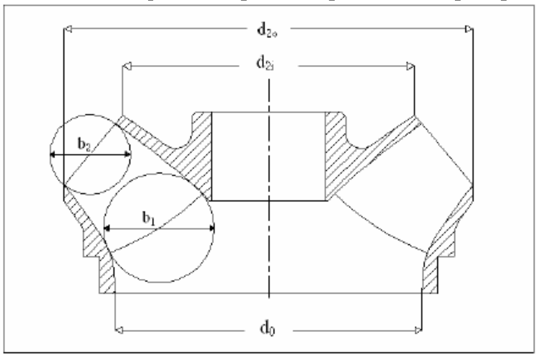


Figure 2.Impeller layout selected from[2] layne bowler library

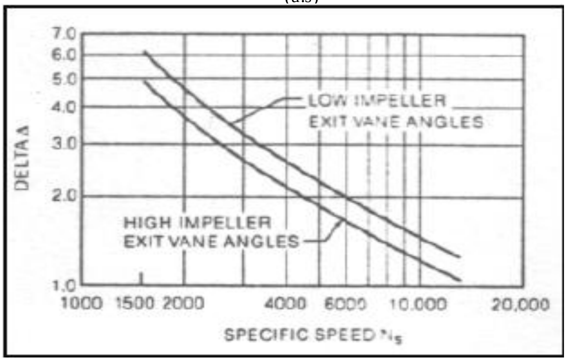


Figure 3. Cordier diagram[2]

for above value of $N_{(u.s)}$, we get Δhigh = 3.2 and Δlow= 3.8 from cordier diagram shown in figure 3.
Putiing this value in equation 3 ,we get d_{2m} high = 92.205and d_{2m} low = 109.49 and hence d_{2m} av. =100.85 .from this value we get d_{2m}/d_{2o} =0.92.Putting this value in equation 4,we get d_{2i}=91.25.

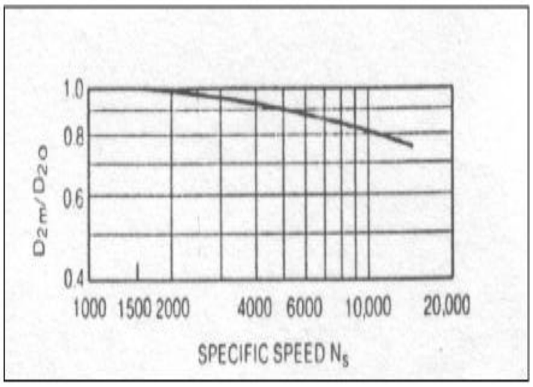


Figure 4. Design chart showing d_{2m}/d_{2o} ratio[2]

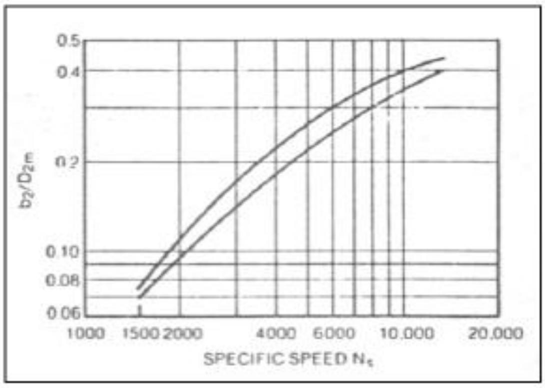


Figure 5.Design chart showing b_2/d_{2m} ratio[2]

$$\Delta = \frac{(gH)^{0.25} d_{2m}}{\sqrt{Q}} \qquad (3) \qquad\qquad d_{2m} = \sqrt{\frac{d_{2o}^{\,2} + d_{2i}^{\,2}}{2}} \qquad (4)$$

Considering n=2880rpm, P_m=5595watt, ζ torsion=40Mpa for steel we get diameter of shaft ds=18mm from equation 5.And power can be calculated from equation 6.

$$d_s = \sqrt[3]{\frac{360000\, p_m}{\tau_{Torsion}\, .n}} \qquad (5) \qquad\qquad P_m = \frac{\rho GQHm}{\eta} \qquad (6)$$

K0 term in Equation (7) varies between 4 and 4.5. K0 is selected close to the 4 where better pump efficiency is desired and K0 is selected close to the 4.5 where better Cavitation performance is desired during the design. The K0 term is selected close to 4 for the pump to be designed in terms of concerning better efficiency in the design. Impeller inlet eye diameter will be do=55.75mm

$$d_o = K_0 \sqrt[3]{\frac{Q}{n}} \qquad (7)$$

The volumetric and mechanical efficiencies are assumed to be 96% for the pump to be designed. The pump efficiency is taken as 76% we can find hydraulic efficiency from the equation 8. ***[2]***

$$\eta = \eta h\,.\,\eta v\,.\,\eta m \qquad (8)$$

The flow rate, Q_i which is mainly defined as the volumetric flow rate passing through the impeller blade passage is calculated using the volumetric efficiency, η_v and the design point flow rate Q. We get Qi = 0.008125 m^3/sec

$$Q_i = \frac{Q}{\eta_v} \qquad (9)$$

The net area in the impeller inlet eye, A_0, is calculated with considering the shaft in the impeller inlet eye as

$$A_0 = \frac{\Pi(d_0^2 - d_s^2)}{4} \qquad (10)$$

Putting appropriate value in the above equation we get the value A_0=2188.3 m^2The net impeller inlet eye diameters, d_o, is taken from the designed impeller layout. The axial velocity in the impeller inlet eye which is also named as meridional velocity is given by

$$V_{m0} = \frac{Q_i}{A_0} \qquad (11)$$

Above equation will give the value of meridional velocity V_{mo}=4.56 m/sec. The meridional velocity at leading edge, V_{m1}, of the blade is calculated as follows,

$$V_{m1} = \frac{Q_i}{A_1} \qquad (12)$$

Peripheral velocity of the impeller will be,

$$U = \frac{\omega d1}{2} \qquad (13)$$

Inlet and exit angle of blade can be calculated from equation (14) and (15)

$$\beta_1 = \tan^{-1}\frac{Vm1}{U1} \qquad (14)$$

$$\beta_2 = \tan^{-1}\frac{Vm2}{U2} \qquad (15)$$

From the above equations the geometrical parameters of the impeller can be obtained as follows

2.2 Output data obtained from analytical design

Quantity	Symbol	Value	Unit
Shaft diameter	d_{sh}	`15	mm
Hub diameter	d_{hi}	25.06	mm
Inlet diameter	D_{2i}	91.25	mm
Outlet diameter	D_{2o}	107.28	mm
Blade inlet angle	β_1	25.56	degree
Outlet blade angle	β_2	11.6	degree

New impeller design obtain from above analytical relation will give efficiency assumed in the equation (8).Now this impeller design will be compared with existing design by cfd analysis to evaluate the perfomance of both the impeller .

III .CFD ANALYSIS OF THE IMPELLERS

3.1 Steps of the CFD analysis:

3.1.1 Modelling of impeller

Solid modelling of the impeller is one option to model with software like creo; solidworks etc., but is better to use Ansys blade modeller available under Ansys workbench component system, vista CPD. By giving the input parameters like inlet and outlet angle, tip diameter we can get below profile corresponding to the given efficiency.

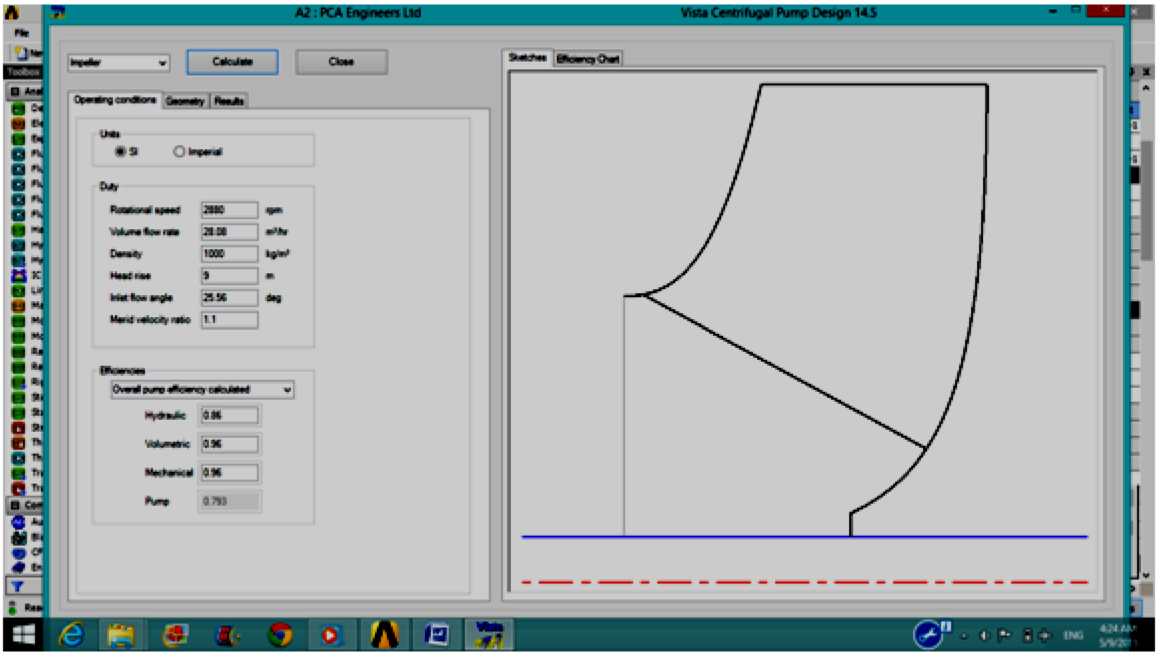


Figure 6 Blade modelling in VISTA CPD

Properties of the impeller model can be defined in the Ansys blade design module. As vista CPD is for centrifugal pump design we need to enter the flow as mixed flow type and direction of rotation. This can be done for existing design and new designed impeller by simply changing the inlet and outlet blade angle and number of blades. We have chosen the number of blade as 6 in the new designed impeller and 7 in existing design impeller

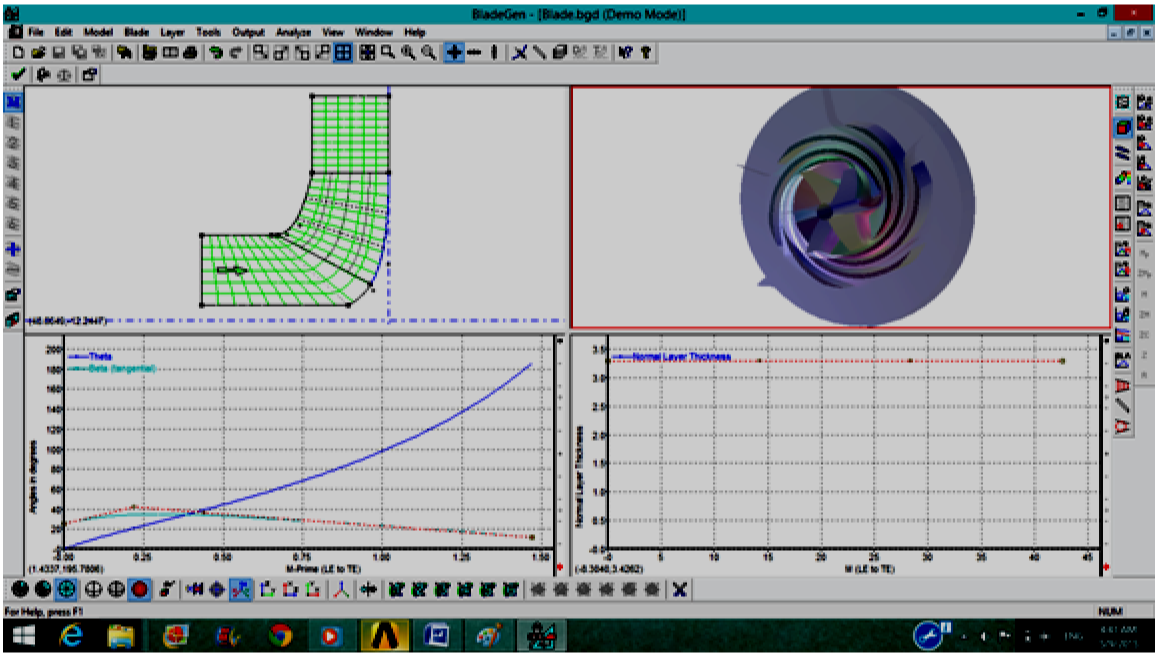


Figure 7 impeller models with leading to trailing edge blade profile

3.1.2 Mesh Generation

Meshing is done by checking global mesh method for all regions and then sizing and refinement at curves has been done. Total 20940 nodes and 13788 elements are generated. After meshing the component in ICEM CFD file with .cgns format will be exported to ansys CFX pre and boundary condition is defined in it.

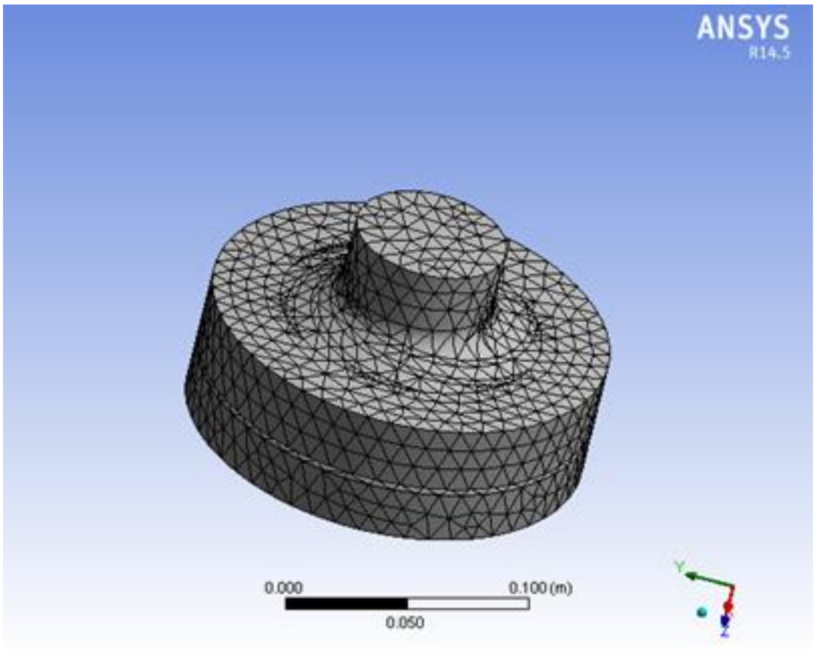


Figure 8 Meshed impeller

3.1.3 Definition of Boundary Conditions

The pressure rise across the inlet and outlet of the pump is aimed to be found throughout the analyses. In order to find the pressure rise, the volumetric flow rate is defined at the outlet of the pump and 1atm. pressure is defined at the inlet of the pump. In real life experiments the pump is submerged into the large reservoir. In cfx pre basic fluid domain is selected with material properties of water and shear stress model. After defining the boundary condition at inlet and outlet solver control is defined in which high resolution with number of iteration up to 1000 marked.

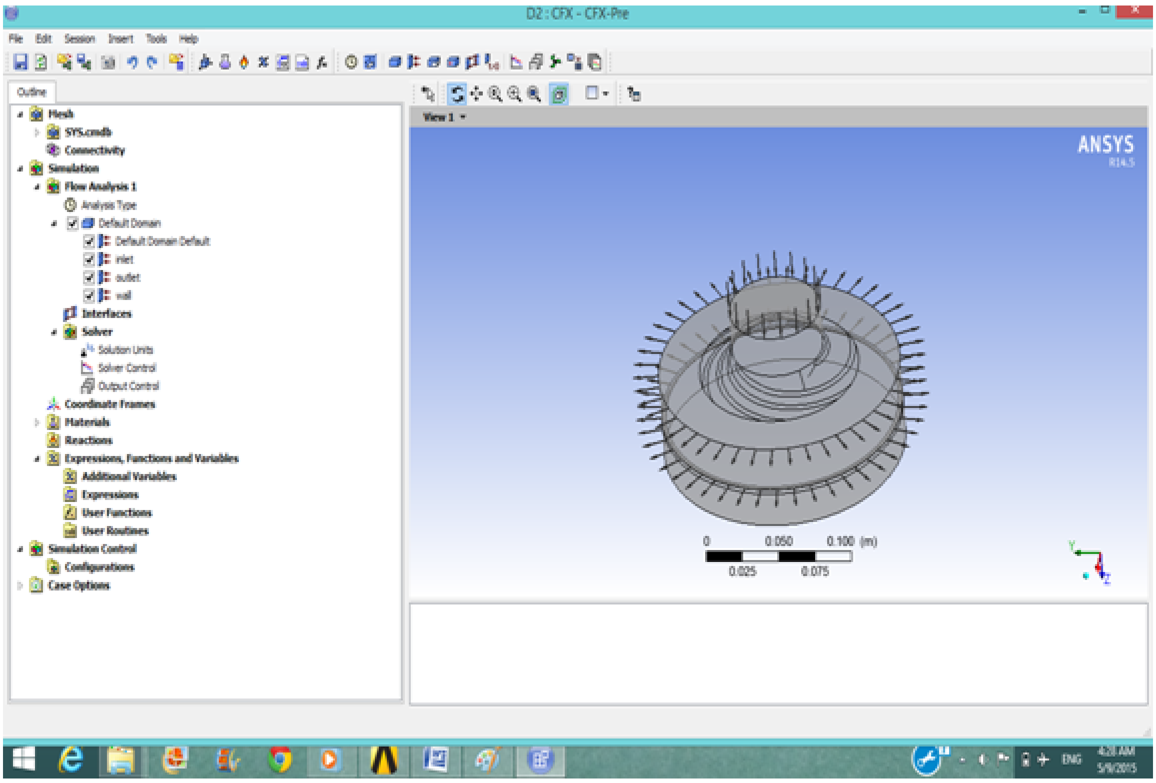
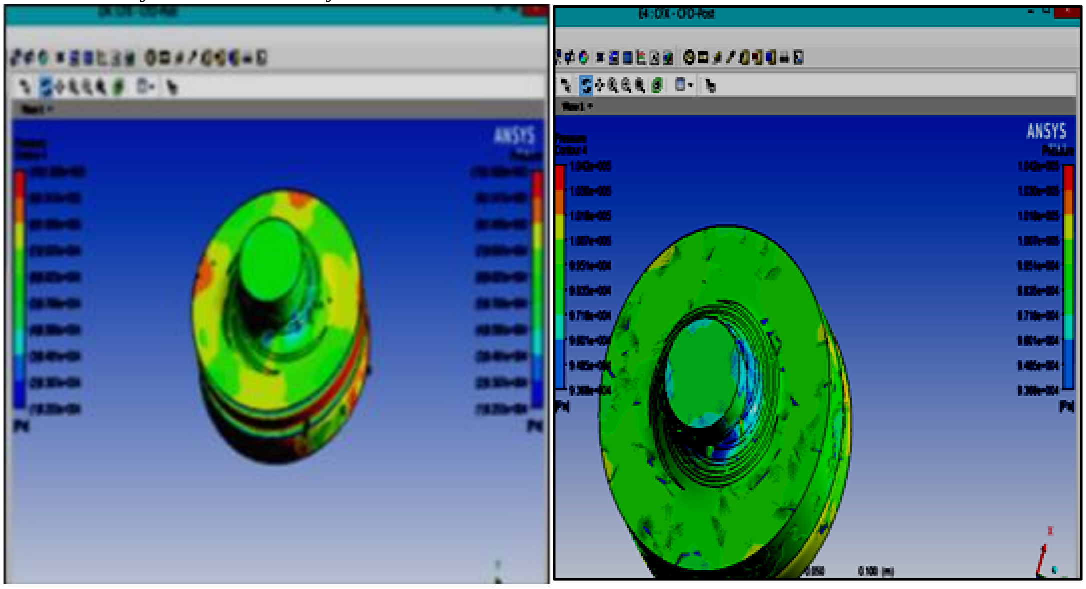


Figure 9 Boundary condition definition in CFX PRE

IV RESULT AND DISCUSSION

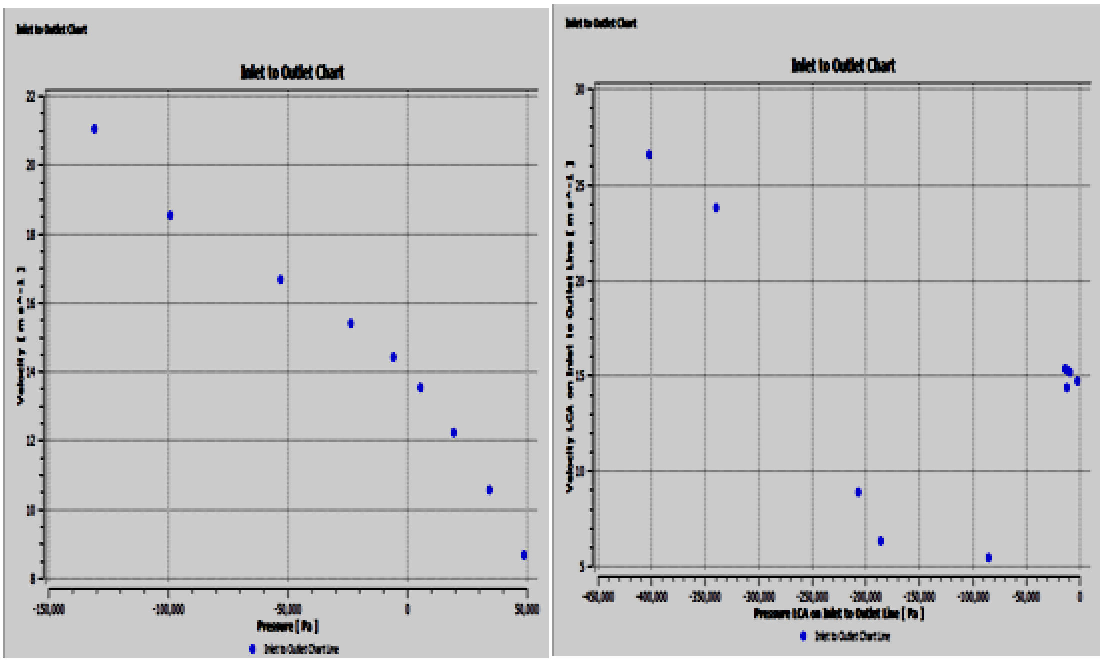
Results of the CFD analysis have been achieved in CFD post. Value of pressure and velocity at inlet and outlet is achieved velocity and pressure distribution across hub to shroud is found. And also the value of torque is found that will be utilized to find the hydraulic efficiency.



(a)New designed *(b) Existing design*

Figure 10 Pressure distribution of new deigned and existing impeller

Above results can be expressed in the form of charts.



(a) New designed *(b)Existing design*

Figure 11 New designed and existing designs Velocity vs. pressure chart for inlet to outlet

Functional calculator will provide all the values of pressure and velocity at inlet and outlet. Values of pressure and velocity can be applied in the Bernoulli's equation and developed head can be found from it.Total head developed H $= \frac{P2-P1}{\rho g} + \frac{v2^2}{2g}$

And hydraulic efficiency is given by

$\eta_h = \frac{\rho g Q H}{T\omega} m$, Where ρ=density of water,g = gravitational acceleration=9.81 m/sec^2,Q=discharge=0.0078m^3/sec.,T=torque,ω= angular velocity=301.59 rad/sec

Applying the values of pressure and velocity at inlet and outlet and value of torque from CFD post analysis will give following results.

4.1 Comparison of CFD result of designed and existing impeller

	DESIGNED IMPELLER	EXISTING IMPELLER
INLET PRESSURE	30367.9 Pa	30592.1 Pa
OUTLET PRESSURE	90081.1 pa	59029.4 Pa
OUTLET VELOCITY	7.74 m/sec	7.94 m/sec
HEAD DEVLOPED	9.13 m	8.12 m
TORQUE	15.28 Nm	13.82 Nm
HYDRAULIC EFFICIENCY	90 %	67.1 %

4.2 COMPARISON OF ANALYTICAL AND CFD RESULTS

Parameter	Analytical calculation		CFD Results	
	Existing design	New design	Existing design	New design
Head	7.45 m/stage	9m/stage	8.12 m/stage	9.13m/stage
Hydraulic efficiency	65%	82%	67.1%	90%

V CONCLUSION

- Impeller having lesser outlet structure angle and less number of blade have improved head of almost 12% with constant discharge rate of 0.0078m^3/sec. Also improved head give better hydraulic efficiency.
- Analytical design improves the 20% hydraulic efficiency and CFD analysis gives much better result of 25% efficiency improvement.
- CFD analysis gives much accurate result than analytical or experimental.

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