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Research Paper Engineering

CFD Analysis of Mixed Flow Submersible pump Impeller

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ABSTRACT

The submersible pumps are used highly at the domestic and commercial level, so to increase the efficiency study of detail flow field in pump is necessary nowadays. With this regard an experimental was conducted on the mixed flow pump in pump manufacturing company. The head achieved by experimental result was 8.08 m. By using design parameter of existing impeller as used in experiment the geometry and cavity model of impeller was modeled in pro-e. After that CFX mesh, inlet, outlet boundary conditions, hub, shroud and blade profile defined in turbo mode for CFD analysis in ANSYS 12.1. Results from the CFD code showed good agreement with experimental result and it was 7.45 m. So, the efficiency of pump calculated by experimental result was 53.27 % and by CFD analysis 49.6 %.

Keywords : Computational Fluid Dynamics, Pump Impeller, Hub, Shroud, Blade, Electric

INTRODUCTION

ESP systems are effective for pumping produced fluids to surface. The submersible pumps used in ESP installations are multistage centrifugal pumps operating in a vertical position. Although their constructional and operational features underwent a continuous evolution over the years, their basic operational principle remained the same. 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. Mixed flow pumps, as the name suggests, function as a compromise between radial and axial flow pumps, the fluid experiences both radial acceleration and lift and exits the impeller somewhere between 0–90 degrees from the axial direction. It is concluded that dynamic characteristic of impeller is an important task in the design and development of submersible pump. The main applications of the submersible pump are Domestic & Community water supply, Industries, High rise buildings, Agriculture, Dairies, Fire fighting systems, Cooling water circulating systems.

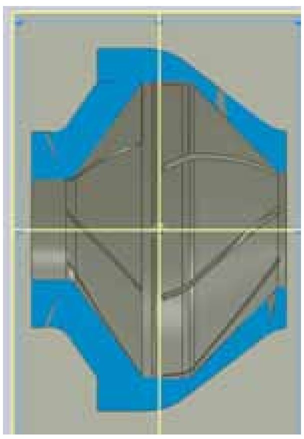


Fig. 1: Mixed Flow Pump.

Mixed flow pump develops head partly by centrifugal force and partly by the lift of the vanes on the liquid. This type of pump has a single inlet impeller with the flow entering axially and discharging in both axial and radial directions (Fig 1).

Review of previous investigations

The CFD analysis to predict the performance of mixed flow pump was carried by Manivannan (2010) . The CFD analysis was conducted to know the flow pattern inside the impeller and velocity, pressure in the outlet of the impeller is predicted. The efficiency of the existing impeller is calculated by the empirical relations. The calculation of head , power rating & efficiency of existing impeller was found to be 19.24 m, 9.46 KW, & 55% respectively. The existing impeller was modified to get the better results by varying the inlet and outlet angle of the impeller vane. Inlet angle of an impeller is calculated based on the inlet flow condition. The inlet diameter and the velocities at the inlet C1 and U1 determine the inlet blade angle β1. The actual blade angle is given by the following relationship.

$$\tan\beta_1 = \frac{U_1}{C_1} \qquad \text{Eq. (1)}$$

The circumferential velocity (U1) and axial velocity (C1) depend on the inlet diameter, rotational speed and discharge. The circumferential velocity and axial velocity can be calculated from the following equations,

$$U_1 = \frac{\omega * D_1}{2} \qquad \text{Eq. (2)}$$

$$C_1 = \frac{4 * Q}{\pi * D^2} \qquad \text{Eq. (3)}$$

Based on the above equation impeller with optimum vane angles have to be modeled and analyzed to obtain the effect of vane angles on impeller performance. The CFD techniques to optimize the hydrodynamic performance analysis of a mixed flow water-jet pump was studied by HWOh et al. (2008). A water jet pump system has been designed to satisfy the performance of the head coefficient (ψ) of 0.557 at the flow coefficient (φ) of 0.747 as a mixed-flow pump, with the specific speed (Ns) 2.44 and the specific diameter 1.79. To simulate the performance characteristics of the water jet pump and performed experiments on the hydraulic performance for the head rise, input power, pump efficiency versus flow rate, and the cavitation characteristics. The numerical analysis of a three-dimensional steady-state turbulent flow based on the Reynolds averaged Navier–Stokes equations has been performed using the k–ω-based shear stress transport model to give accurate predictions of the flow separation phenomena. To define the boundary conditions, the total pressure was described at the inflow boundary, whereas the mass flow rate was specified at the outlet section of the water jet pump. The blade loading results for impeller and diffuser was obtained using CFD. Results shows that the blade loading for the static pressure was ideally distributed along the streamline of the impeller and diffuser. The comparison was made between the computed & measured performance characteristic curves of a mixed flow water jet pump. The results agree fairly well with the measured performance curves over the entire operating conditions. The effect of positive & negative curved blade stacking shape on the performance of a mixed-flow pump impeller was investigated by H W Oh (2010). The result data obtained in study with the help of Ansys CFX was relative flow angle distributions, tangential velocity distributions, relative velocity distributions, surface pressure blade loading at the mid-span, streamline distribution on the pressure & suction surface, streamlines through the tip-clearance region. The effect of Non-cavitating and cavitating for impeller a & b was presented with the help of performance characteristic curves. After analyzing all those results the paper reports that the effect of the positive curved blade stacking unstable the suction performance & negative curved blading improve the stability of suction performance over a wider flow range. Furthermore the simulation of the stationary flow in the impeller & stator of a mixed centrifugal pump using CFD techniques was developed by Maitelli et al. (2010). The three conditions C1, C2, C3 were simulated to obtain the pressure field in the impeller and stator in a mixed flow pump. The first condition was the impeller simulation with the blade length equivalent to the real model. The second condition was the complete connection of the pump, impeller & diffuser with the real blades length. In third condition the impeller & diffuser's external radii increased by 4 mm. The head characteristic curves for C1, C2, C3 compared with the manufacturer's data & meridional profiles of the impeller and the diffuser channels was investigated. The results show that the C1 condition have a high value of head than other condition. In C2 & C3 conditions, the pressure in the impeller channels increases from the entrance to the discharge. This occurs due to the friction losses, recirculation of fluids. The CFD analysis to compute the flow field within the first-stage rotor and stator of a two-stage mixed flow pump was carried out by Miner (2004) .The 3D Reynolds-averaged Navier-Stokes equations in rotating and stationary cylindrical coordinate systems for the rotor and stator, respectively. Turbulence effects are modeled using a standard k − ε turbulence model. Stage design parameters are rotational speed 890 rpm, flow coefficient φ = 0.116, head coefficient ψ = 0.094, specific speed 2.01, flow rate 0.38 m3/s, and head rise 13.1 m. The velocities, and static & total pressures for both the rotor and stator studied in this paper. After that Non dimensional impeller results & Non dimensional stator results was compared to get the final result.

The scope of this work covers the analysis of an impeller used in a mixed flow multi-stage pump, and to investigate the applicability of using CFD code as an aid in designing effective mixed-flow pump. The design of hydraulic machinery involves a lot of parameters which makes the problem more complex for the human brain to manipulate. The innovation in computational fluid dynamic field has made it possible to simulate and visualize some of the features of flow through rotating machines that would be difficult to measure experimentally.

MODELING and ANALYSIS

The modeling and cavity model was generated using pro-e. Geometric parameter used for the modeling of mixed flow pump impeller was,

Table 1. Geometric parameter of mixed flow impeller.

Parameter	Size
Impeller inlet diameter(D1)	59.46 mm
Impeller outlet diameter(D2)	89.41 mm
Number of blades(Nb)	6
Number of stages	6
Shaft Diameter(Ds)	20.25
Blade Inlet Height(B1)	14.72 mm
Blade Outlet Height(B2)	13.45 mm
Mass Flow Rate(Q)	415 LPM
Head(H)	8.08 m
Rotation(N)	2850 rpm

CFX meshing was used for the CFD analysis purpose. The number of nodes and elements generated in ansys mesh was 11621 and 49372 respectively. There are many different types of inlet boundary condition combinations for the mass and momentum equations. For all other transport equations, the value is specified directly at the inlet, or specified in terms of a simple relationship that constrains the dependent variable. Due to the constant mass flow rate we have applied the mass flow rate at the inlet and outlet boundary condition.

Table 2. The inlet boundary condition applied to the impeller.

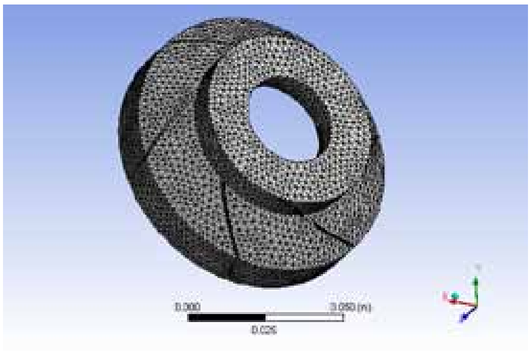
Flow Direction	Normal to Boundary Condition
Mass And Momentum	Mass Flow Rate
Mass Flow Rate	6.9170e+00 [kg s^-1]

Table 3.The outlet boundary condition applied to the impeller.

Mass And Momentum	Mass Flow Rate
Mass Flow Rate	6.9170e+00 [kg s^-1]

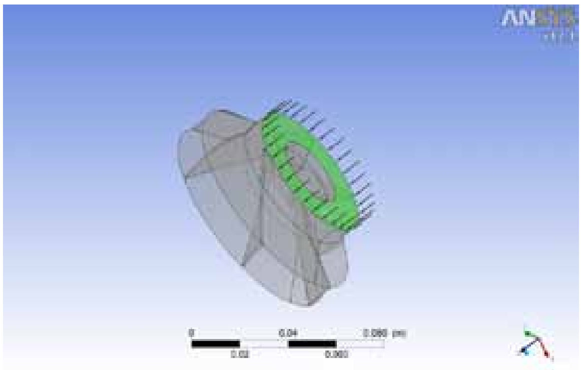
Walls are solid (impermeable) boundaries to fluid flow. Walls allow the permeation of heat and Additional Variables into and out of the domain through the setting of flux and fixed value conditions at wall boundaries. The hub and shroud was defined as a wall as shown in fig (c) and (d).

(a)



(b)

Fig. 2: (a) Impeller model generated in PRO-E. (b) Impeller with mesh.



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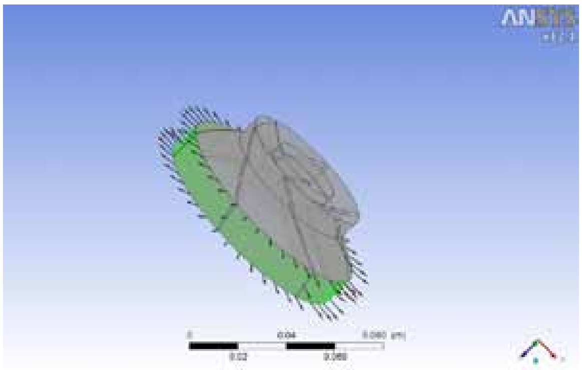

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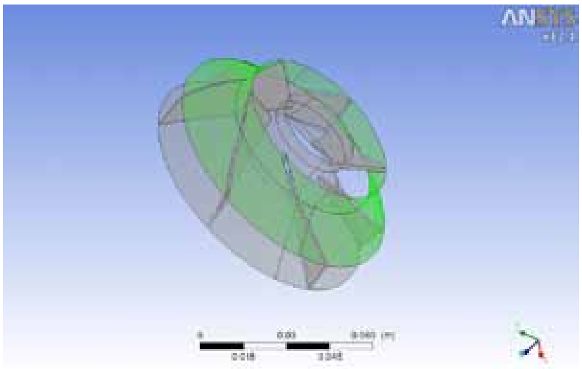

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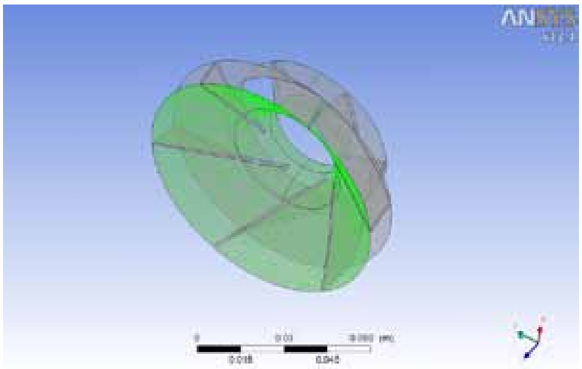


(d)

Fig. 3:The boundary condition applied to the pump impeller was (a) Inlet boundary condition. (b) Outlet boundary condition. (c) Hub as wall. (d) Shroud as wall.

RESULTS and DISCUSSION

The meridional view of impeller is showed in below fig 4.

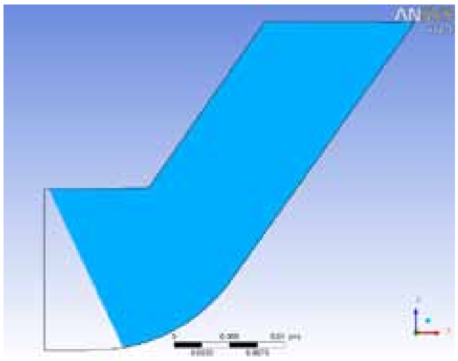
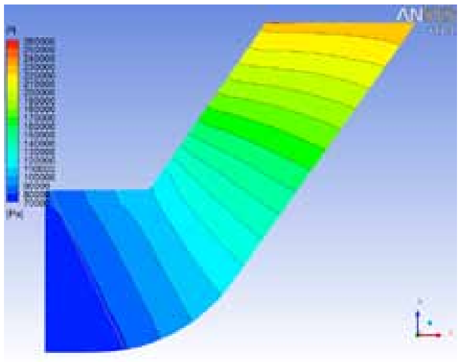


Fig. 4: Meridional view of impeller



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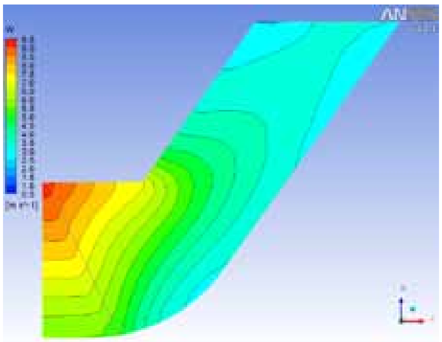
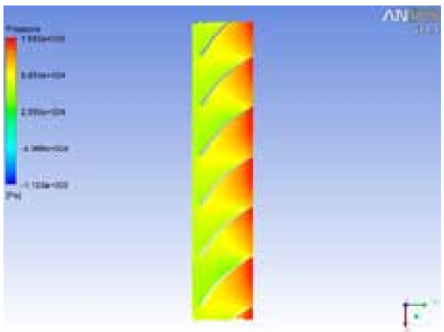
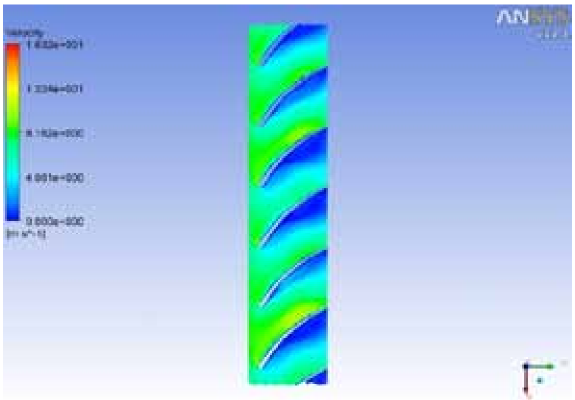


(b)

Figure 5: (a)Pressure distribution (b)Velocity distribution in meridional view of impeller.



(a)



(b)

Fig. 6: (a)Pressure distribution (b)Velocity distribution in blade to blade view of impeller

As the flow entering the impeller eye, it is diverted into the blade-to-blade passage (fig. 6). Due to the unsteady effect developed at upstream, the flow entering the passage is no longer tangential to the leading edge of impeller blade .Fig 5 shows the velocity vector and pressure field with separation and inflow. Increased flow velocity can be observed at the blade inlet due to the blockage of the flow, whereas on the contrary the pressure is reduced (Fig.5). Further downstream the contours become smooth between the blades and the pressure increases continuously towards the exit of the computational domain. The distribution of pressure and velocity were identical. Pressure was increases from inlet to outlet at an steady rate and Velocity was decreases from inlet to outlet flow(Fig 6).

Head variation

Results show that the head achieved by the experimental analysis is agree fairly with the head achieved by the pump analysis report.

Table 4. Comparison of Head

	Results obtained by Ansys CFX	Experimental Results
Head (m)	7.45	8.08

Efficiency variation

Efficiency of the pump can find by,

$$\eta = \frac{Head * Q}{6120 * KW} \qquad \text{Eq. (4)}$$

where,

- Head in mm.
- Discharge in liter per minute.
- Power in KW.

Table 5. Comparison of Efficiency

	Results obtained by Ansys CFX	Experimental Results
Efficiency (%)	49..6	53.27

CONCLUSION

In this paper a computational simulation of the mixed flow pump impeller was implemented. A CFD code, the ANSYS® CFX® 12.1, was used to obtain the head and pressure, velocity streamlines. The analysis results show the head of 7.45m and the head achieved by the experimental work in industries was 8.08 m.The efficiency find by experimental result was 53.27 % and by CFD analysis 49.6 %.In the CFD analysis high values were obtained for the head, comparing to the manufacturer experimental head. Because in CFD analysis there is no influence from the diffuser, so the friction losses are smaller, affecting the pressure fields and increasing the head values. This fact represents the necessity to introduce the friction losses due to coupling between the diffuser and impeller. Result shows pressure in the impeller channels increases from the entrance to the discharge in successive ranges. This fact shows the energy transfer in the impeller. At the same time velocity in the impeller decreases from the inlet to the outlet in successive range. This actually means that fluid displacement occurs due to the rotational movement of the fluid. In order to shorten the design periods and lowering the manufacturing, prototyping and test costs of the pump, a commercial Computational Fluid Dynamics (CFD) can be used for millions of calculations required to simulate the interaction of fluids with the complex surfaces used in engineering.

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