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OPTIMIZATION OF DESIGN OF RADIAL FLOW CENTRIFUGAL PUMP IMPELLER USING CFD

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Abstract: This work aims to analyze the hydraulic performance and characteristics of a radial flow centrifugal pump using ANSYS CFX (ver.13.0). A centrifugal radial flow pump has been designed to deliver 0.0074 m3/s of water with a head of 30 m running at a speed of 2870 rpm. The centrifugal radial flow pump has been modeled using PTC Creo (ver. 2.0). Computation fluid dynamics (CFD) has been used to analyze the flow characteristics. The performance of the pump was first determined using the existing thickness of blade as well as inlet blade angle and then, the thickness of blades as well as inlet blade angles have been varied to analyze the pump's performance. The results show that for an initial 10mm blade thickness, the efficiency of the pump was 83.00%. However, the efficiency of pump increased by 3.22% for the optimized 5mm blade thickness & 22° inlet blade angle.

Index Terms **-Computational Fluid Dynamics (CFD) Analysis, Radial Flow Pump, Blade thickness, inlet blade angle, Overall Efficiency**

I. INTRODUCTION

Centrifugal pumps have a snail shaped casing which can be easily identified and are also used in our homes and industries. Radial flow centrifugal pumps are used when moderate head and discharge are required. The fluid enters the axially through the eye of Impeller, acquires tangential and radial velocity as a result of momentum transfer, and leaves the Impeller radially outward into the volute after acquiring momentum and pressure. Depending on the impeller geometry, centrifugal pumps are classified; 1) backward inclined blades, 2) straight or radial blades, and 3) forward inclined blades. Backward inclined blades are the most common and offer the highest efficiency compared to others. The radial blade type (straight blade) has the simplest geometry and produces the largest pressure rise over a wide range of volume flow rates.

To increase pump performance, designers are constantly working to provide more efficient, affordable and reliable machines. Gehlot and Nyari [1] observed that in doing so, it is extremely important to define the shape and size of the impeller vanes. Short passage lengths can result in flow separation and eddy formation. However, longer path results in greater frictional losses. Many researchers have performed numerical modeling, simulation, and analysis of 3-D turbulent flow in centrifugal pump impellers using RANS equations. It has been observed that optimizing the impeller geometry increases hydraulic efficiency within typical errors of less than 10%. [2, 3]. Jude and Homentcovscki [4], used conformal mapping and boundary element techniques to study flow through a 2-D centrifugal propeller using equiangular blades of arbitrary geometry [4]. With only slight differences, the results proved to be a reasonable match with previously obtained results for logarithmic spiral blades.Jain et al. [5] investigated the methods to optimize the geometric and operational parameters of a centrifugal pump running in turbine mode. The rotational speed was varied 900 to 1500 rpm with no, 10% trimming, and 20% trimming of impellers. The results revealed that the performance increases at lower speeds rather than rated speed. Blade rounding led to 3–4% rise in efficiency at rated speed with the original impeller. Oyelami et al. [6] observed that the amount of energy imparted to the fluid is proportional to the velocity at the edge or tip of the blade of the propeller. Various vane profiles were used to evaluate the performance of the designed blower. Furthermore, it was concluded that the performance of the pump was also affected by the fact whether the impeller is open type or closed type. Closed Impeller with curved vanes at the rear showed the best performance or efficiency with respect to output speed and flow rate Jain et al. [7] Methods for optimizing the geometric and operational parameters of centrifugal pumps operating in turbine mode were investigated. With varying rotational speeds between 900 rpm to 1500 rpm, and with 10% trimming and 20% trimming of the propeller, performance increased at lower speeds rather than at rated speed. Blade rounding also increased efficiency by 3-4%. Pandit et al. [8] studied the effect of trimming the impellor diameter in a radial-submersible pump. With a 10% reduction, the efficiency of the original impellor changed. Thus, this method can be selected as a useful improvement technique for large size impellors, without the need for new construction.As observed by Somashekar and Purushothama [9], cavitation phenomenon has a strong relations with the design, operation and renovation of centrifugal pumps. Cavitation occurs near the suction surface, and propagates toward the trailing edge. It is observed that, blade width (b) greatly affects the performance of the pump. As b increases, leakage increases but, efficiency and head of the pump decrease [10].CFD is an effective tool widely used by many researchers to conduct various studies and investigations on centrifugal pumps. CFD analysis is better than trial and error methods [11]. Steady state 3-D Navier–Stokes equations combined with k-epsilon turbulence models have been widely used. Ajit and Isaac [12] investigated the flow through a centrifugal pump impeller with forward and backward curved vanes using ANSYS. The performance of backward curved vanes was found to be better than its counterpart. CFD has also been used to study, analyze, and predict various parameters affecting the impeller performance of radial centrifugal pumps. Studies such as Kaewnai et al. [13], showed that surface roughness has a high impact on pump losses, that is, as the surface roughness increases, the loss coefficient also increases.

Due to lack of research papers that efficiently explain the radial flow type vane profile design processes, designers these days have to reverse engineer the vanes that are popularly available in the market. Thus this paper is an attempt to provide step by step guidance for designing radial type vane profile using double arch method. The objective of this research is to analyze the effect of blade thickness (T) as well as inlet blade angle on the performance of a radial flow pump and find out the optimum value that can increase the overall efficiency of the pump.

II. PUMP SPECIFICATIONS

2.1. Impeller geometry

The performance of a radial flow pump is highly dependent on its impeller geometry. In the present work, a pump has been designed with the specifications shown in Table-1, and the effect of blade thickness (t) as well as inlet blade angle on its performance has been analyzed. Detailed studies have been done on the geometrical characteristics of the impellor. The model of the designed pump and impellor is shown in Fig 1 and Fig 2.

Table-1: Design specification of pump

Design	Specifications
Flow rate (m^3/s)	0.0074
Head (m)	30
Rotating speed (rpm)	2870

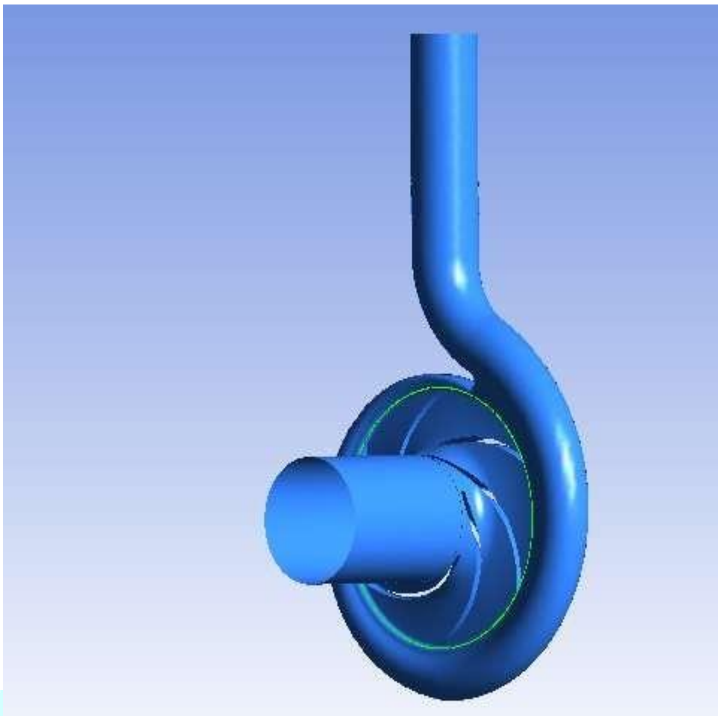

Fig-1: Model of Pump

Fig-2: Model of Pump Impeller

To investigate the effects of these geometrical characteristics on pump flow and impeller performance, parameterization has been performed by reducing the number of controlled geometrical variables as shown in Table-2.

Table-2: Geometrical features of the impeller

Parameter	Size
Inlet diameter (d_1)	66 mm
Outlet diameter (d_2)	172 mm
Vane inlet angle (β_1)	23°
Vane outlet angle (β_2)	29°
Number of blades (z)	6
Blade thickness (t)	10 mm
Shaft diameter (d_{sh})	25 mm
Blade inlet height (b1)	15 mm
Blade outlet height (b2)	6 mm

III. METHOD FOR CONSTRUCTING THE VANE SHAPE

3.1 Circular Arc Method

An impeller is usually divided into a number of concentric rings between radii R1 and R2 in a random manner. Thus the vane profile can be defined using one arc or two arcs of a circle. Double arch method gives better results than single arch method.

Vane profile in double arc method is constructed by joining arcs of the two circles, drawn through points A and B. The circle is divided into „z" equal parts which pass through the inlet edge of the blades of diameter. C_1P_1, C_2P_2 are the tangents drawn from the division points C_1, C_2, C_3, etc touching the radius of circle, $\delta = d_1 \sin\beta_1$. From the points of intersection (P_1, P_2 etc.), arcs having radius $\rho_1 = P_1C_1 = P_2C_2$ and so on, are drawn. Each arc here forms the inlet partof vane and an approximate portion of the involute curve.Line OE is further extended to meet the circle of radius R_2 atpoint B. Another line BG is drawn at an angle of β_2 to line OB. Where, the remaining part of the vane is formed by a smooth curve or another circular arc with centre G.

The radius of an arc of the circle is determined by;

$$\rho_2 = \frac{R_2^2 - R_1^2}{2(R_2 \cos\beta_2 - R_1 \cos\beta_1)}$$

Where; radius R_F is equal to OF and, angle β_1 is angle EFO. If the blade thickness is kept constant, the drawn profile actsas the centre line for the vane.

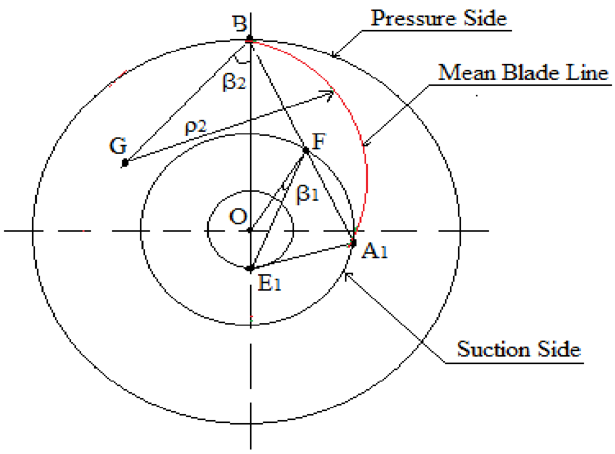

Fig-3: Double Arc method

IV. METHODS TO CALCULATE VOLUTE CASING

4.1.Principle of Constant Moment of Momentum

According to this principle, the moment of momentum remains constant at different sections, which can becalculated by;

$$M = C_u\, r = C_{u_1}\, r = \text{constant}$$

4.2 Principle of Constant Mean Velocity

Discharge at any section of the volute is given by

$$Q_{\emptyset} = \frac{Q \times \emptyset^{\circ}}{360}$$

And, the Area of volute at any section, by;

$$A_{\emptyset} = \frac{Q_{\emptyset}}{C_3} = \frac{Q \times \emptyset^{\circ}}{360 \times C^3}$$

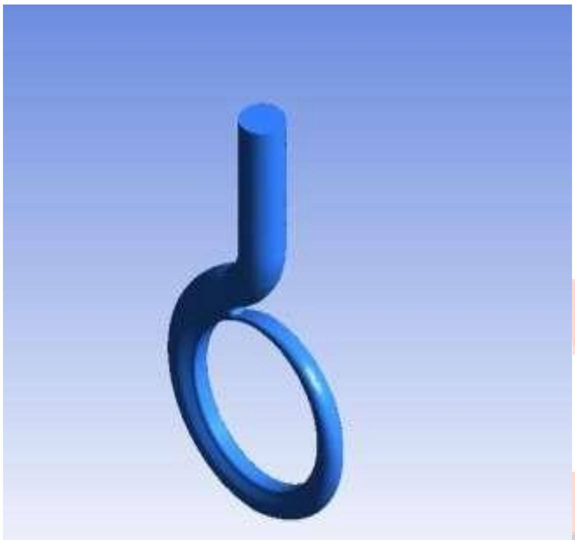

Fig-4: Model of Casing

V. MESHING OF PUMP ASSEMBLY

ANSYS was used to develop the final mesh of the pump assembly, as shown in Fig.5. Total number of elements and nodes are given in Table-3.

Table-3: Mesh information of pump assembly

Total elements	**Total nodes**	**TRI_3**	**TETRA_4**	**LINE_2**
5900980	1180196	245102	5711460	8340

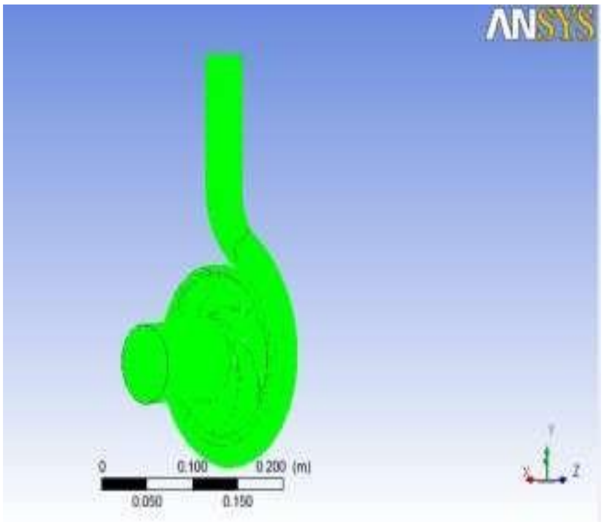

Fig-5: Model of Casing

VI. BOUNDARY CONDITIONS

The numerical computation assumes a steady-state with the following following boundary conditions. Radial flow pump impeller domain is a rotating frame of reference with working fluid as water at 25˚ C, rotational speed of 2870 rpm, 1 atm. pressure at inlet, and 0.0074 m3/s of discharge at the outlet. Turbulence intensity of 5% is considered in k-ε Model assumes a disturbance intensity of 5%.

6.1 Velocity Stream Contour

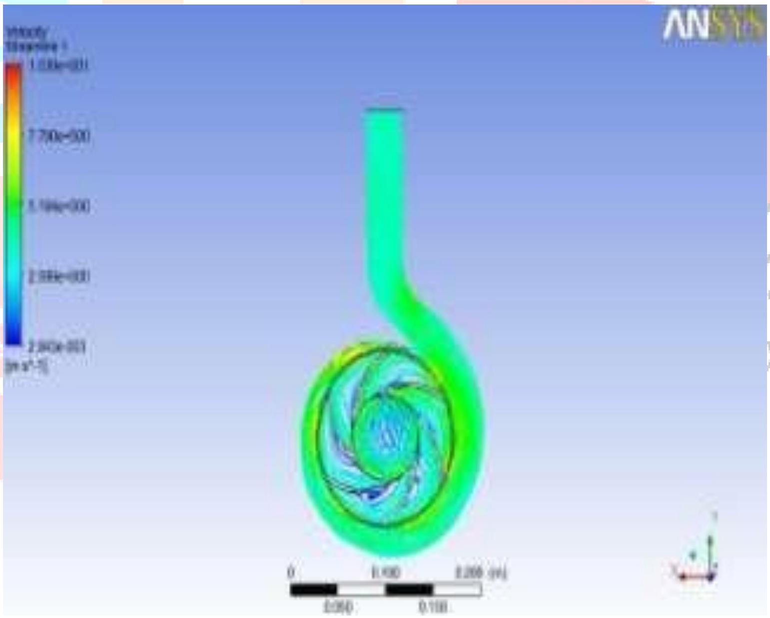

Fig-6: Velocity Stream Contour

6.2 Pressure Contour

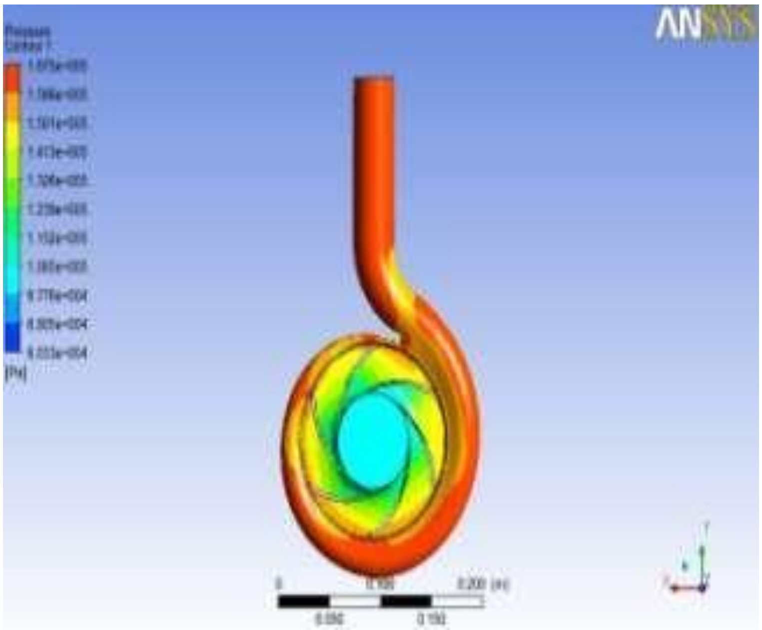

Fig-7: Pressure Contour

VII. RESULTS

Inlet power (IP) can be calculated as;

⇒ 2πNT /60*1000

⇒ (2π*2870*16.054)/60*1000

⇒ 4.825 KW

And, Outlet power (OP) can be calculated as;

⇒ (P0-Pi)*Q/1000

⇒ (560137.11-18920.89)* 0.0074 /1000

⇒ 4.005 KW

VIII. OPTIMIZATION OF RESULTS

8.1 Optimization of Inlet Blade Angle & Blade thickness

Table-4: Vane angles (β1) and Blade thickness (t) for modified impeller

Impeller	Inlet angle, β1 (deg)	Blade Thickness, t (mm)	Efficiency (%)
Impeller 1	22	5	85.77
Impeller 2	22	10	84.48
Impeller 3	22	15	83.31
Impeller 4	23	5	83.08
Impeller 5	23	10	83.00
Impeller 6	23	15	82.74
Impeller 7	24	5	82.54
Impeller 8	24	10	81.30
Impeller 9	24	15	81.02

8.2. Results from Optimization

Table-5: Impeller efficiency

Impeller	Inlet power (KW)	Outlet power (KW)	Efficiency (%)
Impeller 1	6.031	5.173	85.77
Impeller 2	6.002	5.071	84.48
Impeller 3	5.592	4.659	83.31
Impeller 4	5.001	4.155	83.08
Impeller 5	4.825	4.005	83.00
Impeller 6	4.66	3.856	82.74
Impeller 7	4.21	3.475	82.54
Impeller 8	4.012	3.262	81.30
Impeller 9	3.921	3.177	81.02

8.3 Efficiency Vs Mass Flow Rate

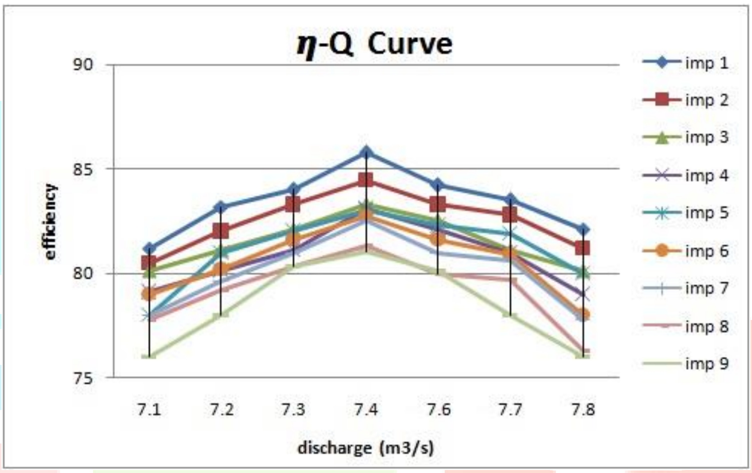

Fig-8: Efficiency Vs Mass Flow Rate

IX. CONCLUSION

Based on the detailed design and extensive CFD analysis of the radial flow impellor, it can be concluded that the thickness of the propeller blade as well as the blade inlet angle have a significant impact on the performance of the radial flow pump. Optimum efficiency was obtained for the Impeller 1 with a blade thickness of 5 mm (opt.) and a blade inlet angle of 22°. It is observed that at the optimum value the overall efficiency of the pump increases by 3.22%.

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