PERFORMANCE OPTIMIZATION 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 centrifugal pump using ANSYS CFX (ver.14.0). A centrifugal radial flow pump has been designed to deliver 0.0074 m^3/s of water with a head of 30 m running at a speed of 2870 rpm. The pump unit 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 and then, the thickness of blades has been varied to analyze the pump's performance. The results show that for an initial 10mm blade thickness, the efficiency of the pump was 82.98%. However, the efficiency of pump increased by 1.87% for the optimized 5mm blade.

Keywords: *Computational Fluid Dynamics (CFD) Analysis, Radial Flow Pump, Blade thickness, Overall Efficiency.*

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1. INTRODUCTION

Centrifugal pumps have a snail shaped casing which can be identified easily and have its application all around in our houses and as well as industries. Radial flow centrifugal pumps are used when moderate head and discharge is required. The fluid enters axially through impeller at the eye, acquires tangential and radial velocity as a result of the momentum transfer, and leaves the impeller radially outwards into the volute, after gaining speed and pressure. Based on impeller geometry, centrifugal pumps are classified as; 1) Backward inclined blades, 2) Straight or Radial blades, and 3) Forward inclined blades. Backward inclined blades are the most common ones and yield highest efficiency of others. Radial blade type (straight blade) has simple geometry and produces greatest pressure rise over a wide range of volume flow rates.

In order to enhance the pump performance, designers are consistently working on to provide the machines that are more efficient, cheap, and reliable. Gahlot & Nyiri [1] observed that in doing so, it is of utmost importance to define the shape and size of impeller vanes. A shorter passage length might result in flow separation and eddy formation. However, a longer passage results in higher frictional losses. Several 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 the hydraulic efficiency increases by optimizing the impeller geometry within typical errors ranging below 10% [2, 3]. Jude & Homentcovschi [4], used conformal mapping and boundary element technique in order to study the flow through a 2-D centrifugal impeller using equiangular blades of arbitrary geometry [4]. With only minute differences, the result proved to be a fair match with the results previously obtained for logarithmic spiral blades.

Oyelami et al. [5] observed that the amount of energy imparted to the liquid is proportional to the velocity at the edge or tip of the vane of impeller. Different vane profiles were used to evaluate the performance of a designed blower. In addition, it was concluded that the pump performance was also influenced by the fact that the impeller is open-type or closed-type. Closed impeller having backward curved vanes showed the best performance or efficiency with respect to output speed and flow rate. Jain et al. [6] investigated the methods to optimize the geometric and operational parameters of a centrifugal pump which runs in turbine mode. With varying rotational speed between 900 rpm to 1500 rpm, and with no, 10% trimming, and 20% trimming of impellers, the performance increased at lower speeds rather than at rated speed. Blade rounding also led to 3–4% efficiency rise. Pandit et al. [7] studied the effect of trimming of impeller diameter in a radial-submersible pump. With 10% reduction, the capacity of the original impeller changed. Thus, this method could be opted as useful correction technique for oversized impellers, without the need of building new ones.

As observed by Somashekar & Purushothama [8], a cavitation phenomenon has a strong relation with the Design, operation and refurbishment of centrifugal pumps. Cavitation takes place near the suction surface, and expands towards the trailing edge. It has been seen that, the blade width (b) greatly affects the pump performance. As b increases, leakage increases but, efficiency and head of the pump decrease [9].

CFD is a widely used effective tool by many researchers to carry out different studies and investigations on the centrifugal pumps. CFD analysis is better when compared with trial and error methods [10]. Steady state 3-D Navier-Stokes equations combined with the k-epsilon turbulence model have been used widely. Ajith & Isaac [11] investigated the flow through centrifugal pump impeller with forward and backward curved vanes using the ANSYS. The backward curved vanes were found to have better performance than its counterpart. CFD has also been used to study, analyze and predict various parameters affecting impeller performance of a radial flow-type centrifugal pump. Studies like of Kaewnai et al. [12], revealed that the surface roughness has high impact on the pump losses, i.e. as surface roughness increases, loss coefficient too increases.

A dearth in research papers explaining efficiently the radial flow type vane profile design procedures, Designers these days have to reverse engineer the vanes which are popularly available in the market. This paper is thus an effort to provide a step by step guidance to design a radial type vane profile using Double Arc method. This research aims to analyze the effect of blade thickness (t) on the performance of radial flow pump, and to find out the optimal value that could enhance the overall efficiency of the pump.

2. 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 its specifications shown in Table-1., and effect of blade thickness (t) on its performance has been analyzed. Detailed study has been performed on the geometric features of the impeller. Model of the designed pump and impeller 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

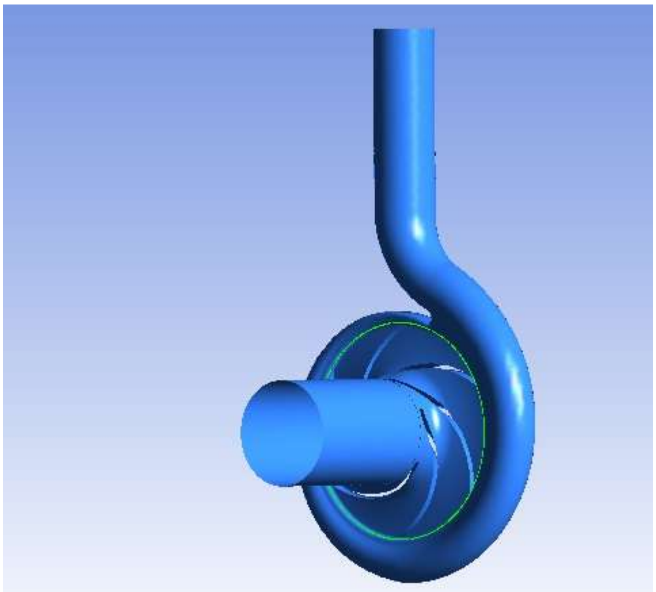

Fig-1: Model of Pump

In order to investigate the effects of these geometric features on the pump flow and impeller performance, parameterization has been done by reducing the number of controlling geometric 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

Fig-2: Model of Pump Impeller

3. METHOD FOR CONSTRUCTING THE VANE SHAPE

3.1 Circular Arc Method

An impeller is usually divided into number of concentric rings between radius R_1 and R_2 in a random manner. Vane profile thus can be defined by using either one arc or two arcs of the circle. The double arc method gives better results as compared to the single arc 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 part of vane and an approximate portion of the involute curve. Line OE is further extended to meet the circle of radius R_2 at point 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;

$$\rho_2 = \frac{R2^2 - R1^2}{2(R_2 Cos\beta_2 - R_1 Cos\ \beta_1)}$$

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 acts as the centre line for the vane.

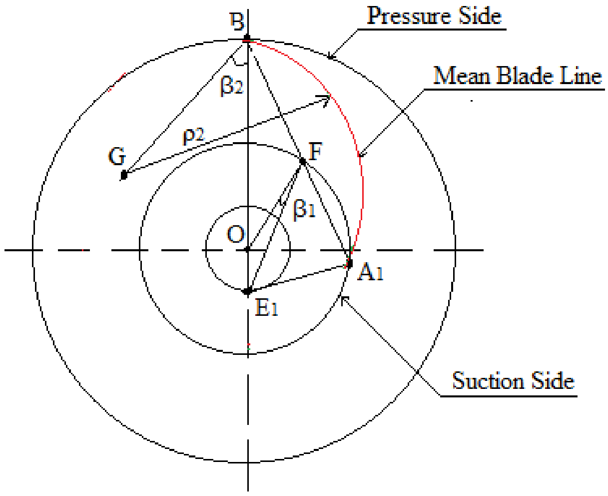

Fig-3: Double Arc method

4. 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 be calculated by;

$$M = Cu\ r = Cu_1\ r = \text{constant}$$

4.2 Principle of Constant Mean Velocity

Discharge at any section of the volute is given by;

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

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

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

Fig-4: Model of Casing

5. 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
5851460	1170596	235102	5611060	8180

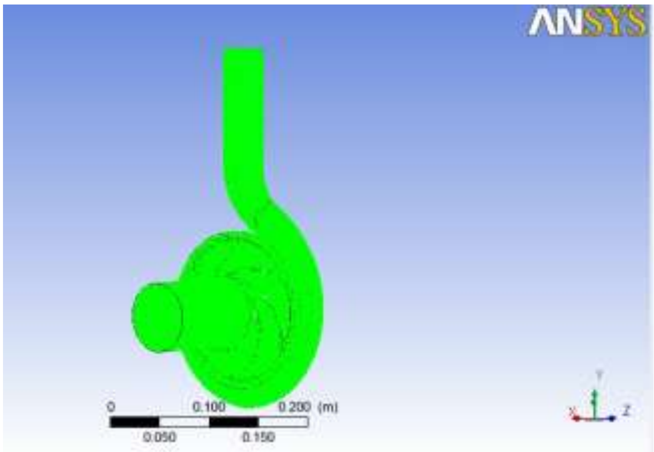

Fig-5: Meshing of designed pump.

6. BOUNDARY CONDITIONS

The numerical computation assumes a steady-state condition with the 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 m^3/s of discharge at the outlet. Turbulence intensity of 5% is considered in k-ε model.

6.1 Velocity Stream Contour

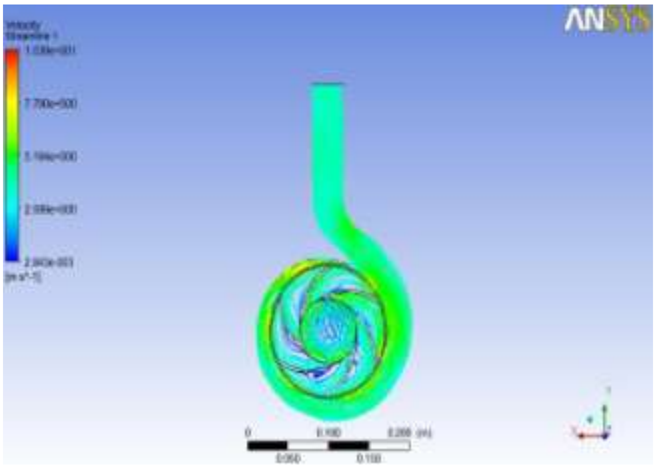

Fig-6: Velocity Stream Contour

6.2 Pressure Contour

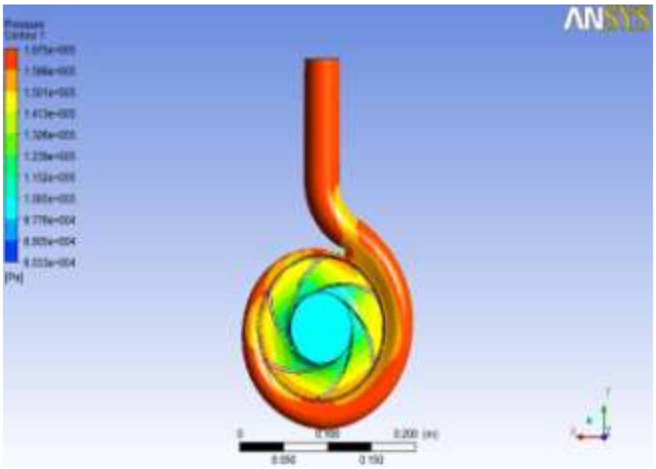

Fig-7: Pressure Contour

7. RESULTS

Inlet power (IP) can be calculated as;

$$\Rightarrow 2\pi NT / 60*1000$$

$$\Rightarrow (2\pi*2870*19.63)/60*1000$$

$$\Rightarrow 5.899 \text{ KW}$$

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

$$\Rightarrow (P0-Pi)*Q/1000$$

$$\Rightarrow (680522.42-18900.8)*\ 0.0074 /1000$$

$$\Rightarrow 4.896 \text{ KW}$$

So, Overall efficiency = (Outlet power / Input power)

$$\Rightarrow 4.895/5.899$$

$$\Rightarrow 0.8299$$

8. OPTIMIZATION OF RESULTS

8.1 Optimization of Number of Blades

Table-4: Variation in blade thickness

Impeller	Blade Thickness, t (mm)	Efficiency (%)
Impeller 1	10	82.99
Impeller 2	9	83.22
Impeller 3	8	83.52
Impeller 4	7	83.75
Impeller 5	6	84.13
Impeller 6	5	84.57

8.2 Results from Optimization

Table-5: Impeller efficiency

Impeller	Inlet power (KW)	Outlet power (KW)	Efficiency (%)
Impeller 1	5.899	4.896	82.99
Impeller 2	5.657	4.708	83.22
Impeller 3	5.746	4.799	83.52
Impeller 4	5.921	4.959	83.75
Impeller 5	6.043	5.084	84.13
Impeller 6	6.035	5.104	84.57

8.3 Efficiency Vs Mass Flow Rate

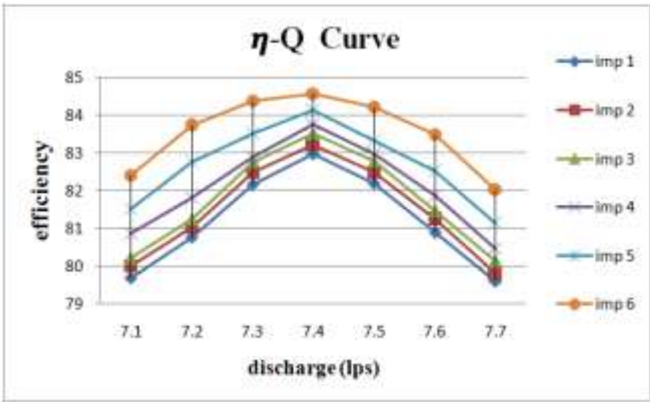

Fig-8: Efficiency Vs Mass Flow Rate

9. CONCLUSION

Based on the detailed design and extensive CFD analysis of radial flow impeller, it can be concluded that thickness of impeller blade has a significant effect on the performance of radial flow pump. The Optimum efficiency was obtained for Impeller 6 having blade thickness of 5mm (opt.). It has been observed that the overall efficiency of pump at optimum value increases by 1.87%.

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BIOGRAPHIES

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