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A Review of Different Blade Design Methods for Radial Flow Centrifugal Pump

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Abstract: ***Centrifugal pumps are widely used for pumping water over short to medium distance through pipeline where moderate head and discharge are required. For optimum performance of pump the vanes should be properly designed. As there are very few papers that explain the radial type vane profile design procedure, therefore it becomes very difficult for designers to design a vane and they are forced to reverse engineer the vane profiles popularly available in the market. This paper is an effort to give a step by step guidance to design a radial type vane profile based on the fundamental understanding of published procedures.***

Keyword: CFD, Design, Impeller, Pump, Radial Flow, Vane.

1. Introduction

Radial flow centrifugal pumps are widely used where head and discharge required are moderate. In radial type vanes, the vane profile is a curve that connects the inlet and outlet diameter of the impeller. Infinite number of curves can be drawn between two points, the length of the vane and hence the passage length can be different for same diameter D_1 and D_2 and same blade angle β_1 and β_2. Hence, it is necessary to define the shape of the vanes. If the length of the passage is short, the divergence angle may increase gradually which will result in separation of flow and formation of eddies. In a longer passage frictional loss will be more. Only an appropriate passage length will give minimum losses. For designing an efficient blade profile an appropriate blade design method should be selected [1].

Different researchers have studied the effect of blade design method on the efficiency of pump like Anagnostopoulos et. al.(2006) performed CFD analysis and design effects in a radial pump impeller. CFD software was used for computations of the steady flow field in the impeller and the characteristic performance curves were constructed. The results showed that hydraulic efficiency of pump can be increased by modifying the impeller geometry [3]. Kyparissis et. al.(2009) conducted parametric study on performance of a centrifugal pump based on simple and double-arc blade design methods-1,2 and 3. They found that when pump was operated at nominal flow rate simple arc method and double arc method-3, models gave best efficiency but when the pump was operated below nominal flow rate double arc method-3, caused a significant improvement of the hydraulic efficiency. On the other hand for flow rates greater than nominal simple arc method analysis gave higher hydraulic efficiency [5]. A simplified 3d model approach in constructing the plain vane profile of a radial type submersible pump impeller using 3D CAD software was developed by Gundale et.al.(2013). The impeller of a radial flow centrifugal pump was developed with different blade generation method and it was concluded that concentric circular arc method is the most simple method to generate a blade profile [6]. Singh and Natraj (2014) investigated the performance of impeller by developing the vane profile with circular arc method and point by point method. The investigation was carried out in Computational Fluid Dynamic commercial package solid works flow simulation (SWFS). The vane profile with forward and backward curves were analysed and the results showed that maximum efficiency was obtained with backward curved circular arc method [7].

2. Methodology

General methods available to design radial flow impeller vanes are simple arc method, double arc method, concentric circular arc method and point by point method. Each of these methods are discussed here in detail along with the calculations of blade design with concentric circular arc method and point by point method. The details of pump which are used for calculation are shown in table-1.

Table 1: Design parameters of the impeller used to construct the blade profile

S.No	*Description*	*Values*
1	Impeller inlet diameter (D_1)	66 mm
2	Impeller outlet diameter (D_2)	173 mm
3	Vane inlet angle (β_1)	23°
4	Vane outlet angle (β_2)	29°
5	Number of blades (Z)	7
6	Vane or blade thickness	5 mm
7	Shaft diameter (D_{sh})	25 mm
8	Blade inlet height (B_1)	15 mm
9	Blade outlet height (B_2)	6 mm
10	Mass flow rate (Q)	7.4 kg/s
11	Head (H)	30 m
12	Rotation (N)	2870 RPM

2.1 Simple Arc Method

According to Pfleiderer's analytical method as discussed by Kyparissis et. al.(2009), in simple arc method the blade mean line is drawn with a single curve. The blade mean line AC is drawn from centre of curvature E with radius of arc R. To draw the blade mean line first an auxiliary circle C_a is drawn concentric with suction and pressure side of the impeller with diameter d_1 given by

$$d_1=D_1\sin\beta_1$$

The centre of curvature E is defined at the tangent of the auxiliary circle C_a which starts from A and point E is at a distance equal to the radius R of the blade mean curve from point A.

$$R = \frac{1}{2}\frac{\left(\frac{D_2}{2}\right)^2 - \left(\frac{D_1}{2}\right)^2}{\left(\frac{D_2}{2}\right)\cos\beta_2 - \left(\frac{D_1}{2}\right)\cos\beta_1}$$

Where D_1 and D_2 are impeller diameters at suction and pressure side respectively and β_1 and β_2 are vane angles at leading and trailing edge respectively [5]. Fiigure 1 shows blade mean line generated using simple arc method

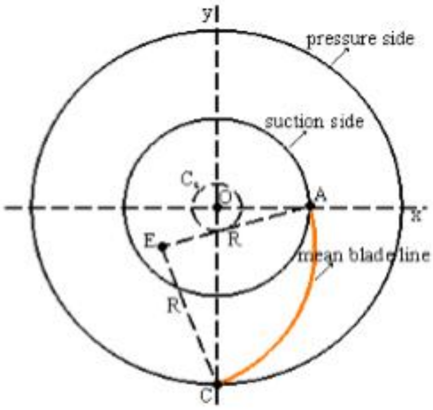

Figure 1: Blade mean line drawn using SAM (Simple Arc Method)

2.2 Double Arc Method

In double arc method the blade mean line is determined by construction of two curves. Pfleiderer has given three types of double arc method namely DAM1, DAM2 and DAM3 explained in Kyparissis et. al.(2009). DAM 1 (Double Arc Method-1) is considered as the poorest continuity of two curves to the connection point B as shown in Figure 2.

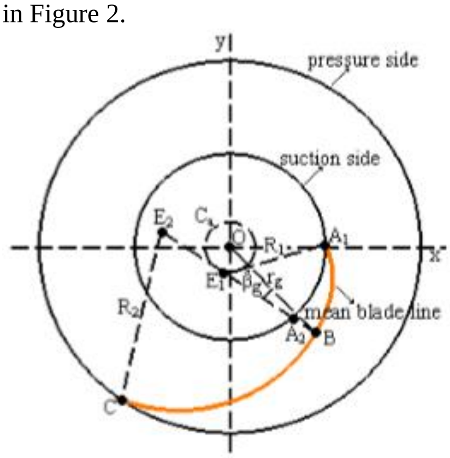

Figure 2: Blade mean line drawn using DAM1

The blade mean line consists of two arcs A_1B and BC with E_1 and R_1 as centre of curvature and the radius of arc A_1B respectively. In the same manner E_2 and R_2 are centre of curvature and radius of arc BC respectively. The auxiliary circle C_a is drawn in the same manner as drawn in Single arc method. The periphery of suction side is divided into equal parts, just as the no of blades. The tangent of the auxiliary circle from points A_1 and A_2 intersects at point E_1. The point B is end of first arc and it is defined at the extension of the line that connects the point E_1 and A_2 at a distance R_1, equal to the distance between points E_1 and A_1. The radius of second arc BC i.e. R_2 is defined as

$$R_2 = \frac{1}{2}\frac{\left\{\frac{D_2}{2}\right\}^2 - {r_g}^2}{\left\{\frac{D_2}{2}\right\}cos\beta_2 - r_g cos\beta_g}$$

Where r_g is equal to OB and β_g is the angle between E_1B and OB as shown in Figure 2.

In DAM 2 (Double Arc Method-2) keeping the initial point A_1 fixed the arc length A_1B is changed and the formula used for new arc length $A_1B$is given as

$$A_1B_{DAM2}=0.75A_1B_{DAM1}$$

As a result of this centre of curvature E_2 for second arc BC shifts downwards and the arc length of the blade decreases while the magnitude of E_1, R_1 and R_2 remains unchanged. In DAM3 (Double Arc Method-3) the radius R_1 is kept 20% bigger than the radius of DAM1 i.e.

$$R_{1DAM3}=1.2\ R_{1DAM1}$$

As a result of this the centre of curvature E_1 shifts to left and E_2 moves downward while radius R_2 remains constant. It is obvious that the shift of E_2 in DAM3 is greater when compared to DAM2. Thus, the arc length of blade DAM3 becomes shorter than the corresponding of DAM2 [5].

2.3 Circular Arc Method

In circular arc method the diameter of the impeller is divided into a number of concentric circular rings not necessarily equally spaced. The value of radius of circular arc R for any two consecutive concentric circular rings is calculated using the equation and vane shape is plotted which is actually an arc tangent to both the rings.

$$R = \frac{1}{2}\frac{\left(\frac{D_2}{2}\right)^2 - \left(\frac{D_1}{2}\right)^2}{\left(\frac{D_2}{2}\right)\cos\beta_2 - \left(\frac{D_1}{2}\right)\cos\beta_1}$$

Where D_1 and D_2 are impeller diameters at suction and pressure side respectively and β_1 and β_2 are vane angles at leading and trailing edge respectively.

The radius of inner diameter and outer diameter are R_1 and R_2 respectively. The radius of intermediate rings can be obtained by adding the term $(R_2\text{-}R_1)/n$ to the radius of preceding ring. Similarly, the corresponding value of β can be obtained by establishing a straight line relationship between β and R [7]. The blade mean line drawn using concentric circular arc method is shown in figure 3.

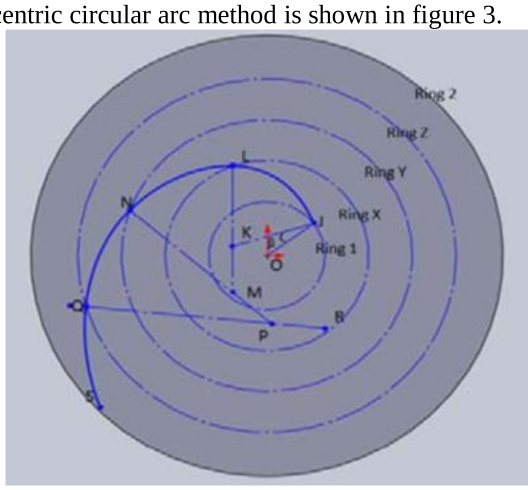

Figure 3: Blade mean line drawn using concentric circular arc method

2.4 Point By Point Method

In point by point method the blade mean line is drawn by determining number of intermediate points between the inner and outer diameter of the impeller and the final blade profile is obtained by drawing a smooth curve through these points joining the inner and outer diameter of the impeller. We know that, to specify a point here we need two parameters radius R and angle θ [1]. Figure 4 show the representation of blade mean line using point by point method.

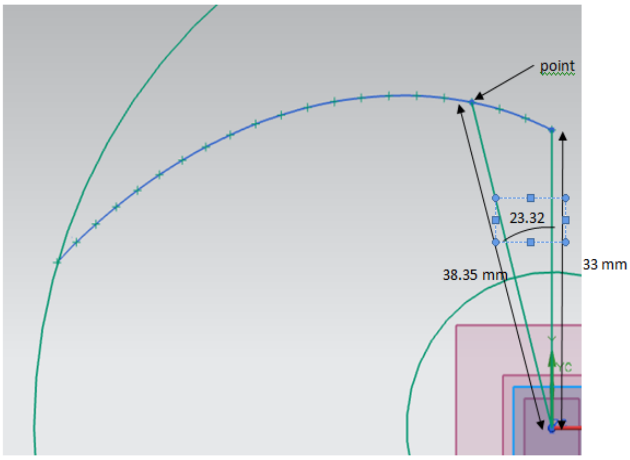

Figure 4: Blade mean line drawn using point by point method

3. Results

On the basis of blade design methods discussed, calculations for concentric circular arc method and point by point method for the pump parameters are given where table-2 and table-3 shows the calculation for concentric circular arc method and point by point method respectively based on the procedure explained in this paper.

Table 2: Calculated values for R1, R2, β1, β2 and R

R_1	R_2	β_1	β_2	R
33.00	37.86	23.00	23.55	39.77
37.86	42.72	23.55	24.09	45.62
42.72	47.58	24.09	24.64	51.65
47.58	52.44	24.64	25.18	57.74
52.44	57.30	25.18	25.73	64.08
57.30	62.16	25.73	26.27	70.44
62.16	67.02	26.27	26.82	77.12
67.02	71.88	26.82	27.36	83.78
71.88	76.74	27.36	27.91	90.86
76.74	81.60	27.91	28.45	97.87
81.60	86.50	28.45	29.00	105.35

Table 3: R and θ values calculated using point by point method

$Pt.$	R	dR	C_m	W	$\beta = sin^{-1}\frac{C_m}{W}$	$B = \frac{1}{R * tan\beta}$	$dA = dR * \frac{B_n + B_{n+1}}{2}$	ΣdA	$\theta = \frac{180}{\pi} * \Sigma dA$
1	33.00	5.35	3.40	8.70	23.00	0.071	0.353	0.353	20.23
2	38.35	5.35	3.31	8.36	23.32	0.061	0.302	0.655	37.55
3	43.70	5.35	3.23	8.01	23.78	0.052	0.260	0.915	52.42
4	49.05	5.35	3.15	7.67	24.25	0.045	0.227	1.142	65.45
5	54.40	5.35	3.06	7.32	24.71	0.040	0.203	1.346	77.10
6	59.75	5.35	2.98	6.98	25.27	0.036	0.182	1.528	87.52
7	65.10	5.35	2.89	6.64	25.80	0.032	0.161	1.688	96.72
8	70.45	5.35	2.81	6.29	26.54	0.028	0.145	1.833	105.00
9	75.80	5.35	2.72	5.95	27.20	0.026	0.131	1.964	112.51
10	81.15	5.35	2.64	5.60	28.13	0.023	0.118	2.081	119.25
11	86.50	5.35	2.55	5.26	29.00	0.021	0.112	2.194	125.69

4. Conclusions

This paper thus presents description of various blade design methods and the calculations required, on the basis of which designers can design a blade mean line joining the inlet and outlet diameters of the pump impeller. Further modelling of pump impeller with these methods and CFD analysis can be carried out to obtain the performance curve for comparing the efficiency and head obtained with different blade design method at various discharge conditions.

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