

# Design and optimization of mixed flow pump impeller blades by varying semi-cone angle

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**Abstract.** The mixed flow pump is a cross between the axial and radial flow pump. These pumps are used in a large number of applications in modern fields. For the designing of these mixed flow pump impeller blades, a lot number of design parameters are needed to be considered which makes this a tedious task for which fundamentals of turbo-machinery and fluid mechanics are always prerequisites. The semi-cone angle of mixed flow pump impeller blade has a specified range of variations generally between 45o to 60o. From the literature review done related to this topic researchers have considered only a particular semi-cone angle and all the calculations are based on this very same semi-cone angle. By varying this semi-cone angle in the specified range, it can be verified if that affects the designing of the impeller blades for a mixed flow pump. Although a lot of methods are available for designing of mixed flow pump impeller blades like inverse time marching method, the pseudo-stream function method, Fourier expansion singularity method, free vortex method, mean stream line theory method etc. still the optimized design of the mixed flow pump impeller blade has been a cumbersome work. As stated above since all the available research works suggest or propose the blade designs with constant semi-cone angle, here the authors have designed the impeller blades by varying the semi-cone angle in a particular range with regular intervals for a Mixed-Flow pump. Henceforth several relevant impeller blade designs are obtained and optimization is carried out to obtain the optimized design (blade with optimal geometry) of impeller blade.

## 1. Introduction

Mixed flow pump is a combination of radial and axial flow pumps. It can be considered as a cross between radial and axial flow pumps. Impellers are most important components of the mixed flow pumps. It is always a needful to understand and study the flow passage of the impeller blades in order to understand the occurrence of low hydraulic performance of pump.

Empirical calculations are used for incorporating the design methodology where the various occurrences that takes place in the flow passage are ignored. It always acts as a complicated task if any problem in the designing methodology occurs. Since the designer always has less idea about the happenings inside the impeller blade it is very difficult to identify the stages where the problems are found. Rotational-based designing methodology has to be implemented considering fundamentals of fluid mechanics and turbo machinery as prerequisites to resolve the issues related to the performance of pump. In the recent past, many research works have been carried out in this field. Wislicensus [1] was first to initiate this design methodology in 1965, where he focused on the design methodology considering impeller blades as helical surface. Since this designing methodology lacked logical approach and sufficient theoretical support to this concept, it was partially acknowledged. Later Myles [2] in 1965 related the Wislicensus [1] design methodology by superimposing the aerofoil in cascade in a centrifugal field/region assuming that shape of the blades could only be imposed to provide with actual flow field boundary conditions. Considering the effect of slip velocity Busemann



[3] formulated the expression for slip velocity for radial flow pumps. However, its incorporation into a mixed flow pump was not a successful step, which led to the introduction of correction factor into the expression of slip velocity. Senoo, Nakase [4] and Inoue [5] developed a methodology for calculating the meridional stream functions for the swirl distribution. Analysis was carried out for the blade-to-blade flow rate, which later was used for the determination of surface and meridional velocity distribution of the blade from hub to tip. In all the above calculations, they assumed the flow to be axisymmetric in nature. It was also concluded that with considering a large number of blades the approximations were found to be even better. For particular angle of blade angles variations Stepanhoff [6] provided the blade design procedure. Neumann [7] came out with a detailed design optimization with all the required steps in design methodology of a mixed flow pump. Gahlot and Nyiri [8] provided a detailed procedure for designing Radial, Francis and Mixed flow Pumps. Yumiko and Watanabe [9] used the 3D inverse design methodology, CFD, DOE, RSM for designing of mixed flow pump impeller blades and used Multi Objective Genetic Algorithm for optimization. For the efficient design of mixed flow pumps Kim [10] came out with a procedural methodology. He considered Navier Stokes Equation with shear stress turbulent model and determined the efficiency. Sham Sunder [11] in the year 1981 visualized the distribution of stress in impeller blades using FEM. Ramamurthi and Balasubramanian [12] performed the steady state analysis of centrifugal pumps considering cyclic loading. Janker and Van Essen [13], Samir Lemeš and NerminaZaimović- Uzunović [14], Bhope and Padole [15], Hao et al. [16] and Mehta and Patel [17] summarized the results for analysis of highly complex blades in various pumps.

### Nomenclature

$N_s$	Specific Speed of Pump	$\beta_1$	Blade angle at inlet section
$H$	Head of Pump (m)	$\beta_2$	Blade angle at outlet section
$Q$	Volumetric flow rate of Pump ( $m^3/s$ )	$u_1$	Inlet Circumferential Velocity (m/s)
$N$	Rotational Speed of Pump (rpm)	$C_{\theta 2}$	Tangential component of absolute velocity at outlet (m/s)
$D_1$	Diameter of blade at inlet (mm)	$C_m$	Mean absolute velocity in meridional plane (m/s)
$D_2$	Diameter of blade at outlet (mm)	$w$	Relative velocity of flow(m/s)
$D_{m2}$	Mean blade diameter at the outlet (mm)	$\alpha_1$	Fluid inlet angle ( $^\circ$ )
$D_{2o}$	Outlet blade diameter at tip (mm)	$\alpha_2$	Fluid outlet angle ( $^\circ$ )
$D_{2i}$	Outlet blade diameter at hub (mm)	$\alpha_{1'}$	Modified absolute flow angle at inlet ( $^\circ$ )
$P$	Power (kW)	$\alpha_{2'}$	Modified absolute flow angle at outlet ( $^\circ$ )
$S$	Blade spacing (mm)		

### 2. Specifications of Pump

In the present work the pre assumed parameters for undergoing the pump test are  $Q = 0.125 m^3/s$  and total head developed ( $H$ ) = 5m. From the Pump model test code (I.E.C -1976) specify a minimum impeller size (Impeller tip diameter) of 200mm whereas the Hydraulic Institute Standards (1961) recommends a minimum impeller tip diameter at outlet of 300mm. Calculating range of inlet diameter is considered to be 200 mm to 300 mm. The specific speed of a pump (dimensional) is given by,

$$N_s = \frac{N \sqrt{Q}}{(H)^4} \quad (1)$$

### 3. Design Calculations

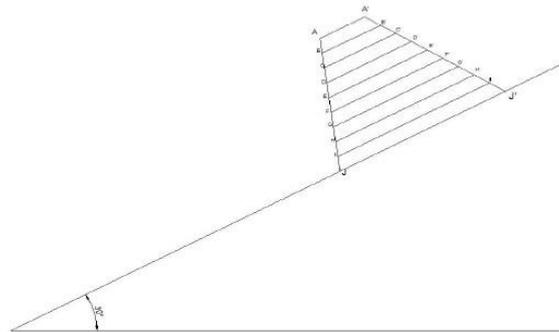
The adopted values for performing the calculations using the basic equations of fluid mechanics are  $N_s = 105.74 \text{ rpm}$ ,  $D_1 = 250 \text{ mm}$ ,  $D_2 = D_{2o} = 324 \text{ mm}$ ,  $P = 20 \text{ kW}$ .

$$r_{m1} = \sqrt{\frac{r_{1o}^2 + r_{1i}^2}{2}} \quad (2)$$

$$\frac{D_1}{D_{m2}} = 0.9 \quad (3)$$

$$D_{m2} = \sqrt{\frac{D_{2o}^2 + D_{2i}^2}{2}} \quad (4)$$

From the above relations  $D_{m2} = 278$  mm and  $D_{2i} = 224$  mm. With a semi cone angle of  $30^\circ$  specified layout of the blade was drawn. The blade height is found to be 58mm from the layout for semi cone angle of  $30^\circ$ . In the same way from the above equations  $r_{m1}$  and  $r_{m2}$  are found to be 102mm and 139mm. In order to satisfy the relations given by Myles [2] these values are modified and found to be 109mm and 132mm respectively. By performing the calculations the ratio of  $C_{m1}$  and  $C_{m2}$  is determined as 1.21 and the designed meridional profile gets changed to trapezoidal profile, which further is divided into ten equal sections from hub to tip names as A-A, B-B, C-C, D-D, E-E, F-F, G-G, H-H, I-I and J-J. All these ten sections as A-A, B-B, C-C, D-D, E-E, F-F, G-G, H-H, I-I and J-J are parallel to each other as depicted in Figure 1.



**Figure 1.** Blade Sections from Hub to Tip.

### 3.1 Sample Design Calculations for A-A Section of Blade

For section A-A of blade,

$$u_1 = \frac{\pi D_{2i} N}{60} = \frac{\pi \times 0.274 \times 1000}{60} = 14.35 \text{ m/s} \quad (5)$$

$$\tan \alpha_1 = \frac{u_1}{C_{m1}} = \frac{u_1}{C_1} = \frac{14.35}{3.15} \quad (6)$$

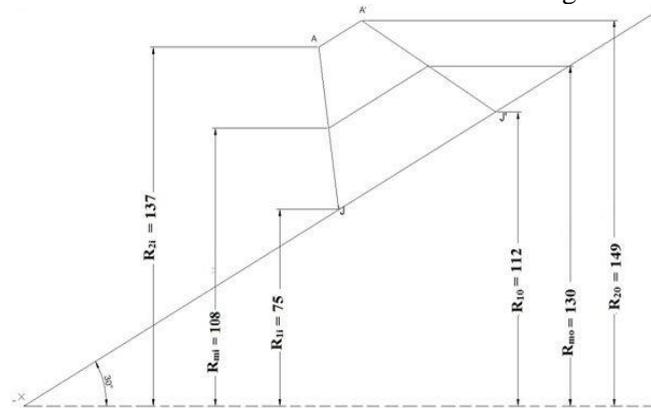
$$\alpha_1 = \tan^{-1} 4.55 = 77.62^\circ$$

Also, 
$$H = \frac{\eta u_2 C_{\theta 2}}{g} \quad (7)$$

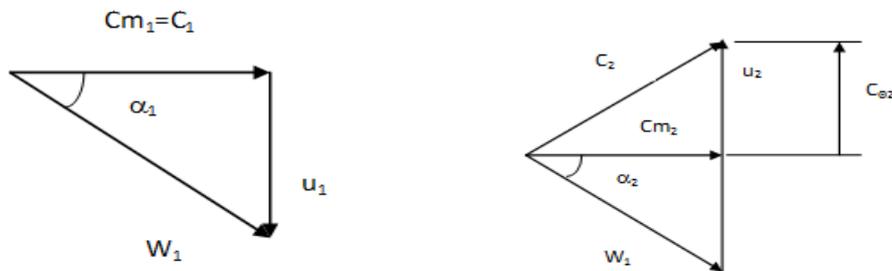
From the relation 
$$\tan \alpha_2 = \frac{u_2 - C_{\theta 2}}{C_{m2}} = \frac{15.6 - 4.14}{2.6} = 4.41 \quad (8)$$

The semi-cone angle of mixed flow pump impeller blade has a specified range of variations generally between  $45^\circ$  to  $60^\circ$  (source: Pump model test codes and data book) [2]. From the above sample equations the layout of the meridional plane is depicted in Fig. 2 for semi cone angle of  $30^\circ$ . The inlet

and the outlet velocity triangles are shown in Fig. 3. In the same fashion the velocity triangles for all the sections can be drawn. It is found that even of the dimensions of the pump layout gets varied the outline of pump layout still remains the same for various semi-cone angles.



**Figure 2.** Blade Layout



**Figure 3.** Inlet and Outlet Velocity Triangles

It was a significant conclusion by researcher Stodola [18] that for any given tip speed and flow rate for the impeller, if the number of blades gets increased then it results in the reduction in eddy. Hence taking 8 blades into consideration and also the slip velocity in this particular case from the Carter's correlations; pitch, modified blade angle and actual chord lengths for each of the sections of the blade were also determined. In Table 1 below the variation of diameter of blade, angles of blade and spacing of blade for all sections of blade with 30° semi cone angle are shown.

**Table 1.** Variation of diameter of blade, angles of blade and spacing of blade for all the sections (30° semi cone angle)

Position	D <sub>1</sub> (mm)	D <sub>2</sub> (mm)	$\alpha_1(^{\circ})$	$\alpha_2(^{\circ})$	$\alpha_m(^{\circ})$	S(mm)
A-A(tip)	274	298	77.62	76.45	77.06	112.31
B-B	260	290	76.97	75.74	76.39	108.00
C-C	244	280	76.15	74.77	75.49	102.89
D-D	230	272	75.34	73.84	74.62	98.57
E-E	218	264	74.57	72.82	73.73	94.64
F-F	204	256	73.57	71.60	72.65	90.32
G-G	190	248	72.43	70.23	71.39	86.00
H-H	176	240	71.14	68.58	69.95	81.68
I-I	162	232	69.62	66.55	68.18	77.36
J-J(hub)	150	224	68.14	64.09	66.32	73.43

### 3.2 Optimal Design of Impeller blades for Mixed Flow Pump

Stepanoff [6] and Stodola [18] performed the calculations considering a constant semi-cone angle of 30° and varied the other factors those affect the design methodology like specific speed, inlet diameter and outlet diameter for several blade profiles. A comparative study of variation of blade parameters with varying semi-cone angle is tabulated in Table 2. Considering the boundary conditions and results, calculations were performed for obtaining the optimized blade profile. The adopted techniques signify the fact that deflection at the inlet section of blade shows a reduction in the thickness by 0.6mm.

**Table 2.** Blade parameters at Various Semi-Cone Angles

Semi-cone Angle (°)	Blade Height (mm)	Width of Blade (mm)
22.5	54.32	44.04
25	55.17	42.57
30	58	24
35	61.04	21.92

### 4. Conclusion

In this present work authors have considered semi-cone angle of 22.5°, 30°, 25° and 35° the design calculations were performed. For a mixed flow pump comparative study of the blade specifications were performed by varying the semi-cone angle. From the meridional profile of the blades it was observed that mixed flow pumps have identical inlet diameter at tip but the outlet diameters for hub and tip and hub inlet diameter is more in case of 22.5°, 25° and 35° semi-cone angles. As semi-cone angle reduces; the axial component of the flow increases to accommodate the free flow of fluid without flow separation. It was concluded that a semi-cone angle of 30° results as the most optimal design of impeller blades for mixed flow pumps.

### Acknowledgement

The authors sincerely acknowledge the comments and suggestions of the reviewers that have been instrumental for improving and upgrading the paper in its final form.

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