Design of Mixed Flow Pump Impeller Blade using Mean Stream Line Theory and its Analysis

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\textbf{Abstract}

Given the importance of blade design in effective performance of the mixed flow pump, the present work demonstrates the designing of the mixed flow pump impeller blade using almost unexplored Mean stream line theory. The Mean stream line theory, though been used sparingly but has found to give comparable results to that of other templates of design. The design process has been carried out in AUTOCAD 2013 and Solid Works Premium 2014 software. The analysis for equivalent stress, equivalent elastic strain, Total deformation and the directional deformation have been carried out in ANSYS 2014 for different construction material of the blade i.e., stainless steel, titanium alloy, bronze, and copper alloy. Total deformation was found to be maximum for impeller blade made from titanium alloy whereas the equivalent stress and strain was least for titanium alloyed impeller blade. Further, a comparison analysis has been carried out for the equivalent stresses in blade designed using mean stream line theory and free vortex theory. It was observed that the equivalent stress in impeller blade designed using free vortex theory was lesser than that designed using mean stream line theory.

\textbf{Keywords}: Mixed flow pump impeller blade, AUTOCAD, Solid Works Premium, ANSYS, Mean stream line theory, free vortex theory,

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1. Introduction

A pump functions to convert mechanical form of energy to hydraulic energy. Pumps can be classified into radial, axial and mixed depending on the configuration of the impeller and specific speed the pump. Rotary mechanism forms the basis of working principle of a centrifugal pump. Combination of centrifugal action and lifting action of the liquid of the vane produces the desired head in case of a mixed flow pump. In case of the mixed flow pump liquid enters the pump in axial direction whereas it exits the pump both in radial and axial
direction. Thus mixed flow pump encompasses the advantages of both the radial and axial flow pump. The advantages lend unique operating characteristics to the mixed flow pump and versatility in their operation. The versatile characteristics makes mixed flow pump suitable for a wide range of applications such as flood control, power station cooling systems etc. Scientific community faces acute challenge to extend the operating range of mixed flow pumps owing to their versatile and reliable characteristics. A large number of geometric parameters are involved in designing of mixed flow pumps. Further, performance prediction is relatively cumbersome task.

“Nomenclature [36]

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>( N_s )</td>
<td>Specific speed of the pump, rpm</td>
</tr>
<tr>
<td>( H )</td>
<td>Head of the pump, m</td>
</tr>
<tr>
<td>( D_{2i} )</td>
<td>Inlet blade diameter at tip, mm</td>
</tr>
<tr>
<td>( D_{10} )</td>
<td>Outlet blade diameter at hub, mm</td>
</tr>
<tr>
<td>( D_{m0} )</td>
<td>Mean blade diameter outlet, mm</td>
</tr>
<tr>
<td>( U_0 )</td>
<td>Outlet flow velocity, m/s</td>
</tr>
<tr>
<td>( D_0 )</td>
<td>Diameter on the outlet side of the blade, mm</td>
</tr>
<tr>
<td>( \Phi )</td>
<td>Outlet fluid angle, degree</td>
</tr>
<tr>
<td>( V_{20} )</td>
<td>Velocity of flow at outlet, m/s</td>
</tr>
<tr>
<td>( V_{10} )</td>
<td>Absolute velocity at outlet, m/s</td>
</tr>
<tr>
<td>( V_1 )</td>
<td>Velocity of slip, m/s</td>
</tr>
<tr>
<td>( \Psi )</td>
<td>Cone angle of the impeller, degree</td>
</tr>
<tr>
<td>( \alpha_m )</td>
<td>Mean blade angle, degree</td>
</tr>
<tr>
<td>( S )</td>
<td>Pitch, mm</td>
</tr>
<tr>
<td>( H )</td>
<td>Manometric efficiency</td>
</tr>
<tr>
<td>( C )</td>
<td>Chord length, mm</td>
</tr>
<tr>
<td>( L )</td>
<td>Blade height from hub to tip, mm</td>
</tr>
<tr>
<td>( \Delta )</td>
<td>Difference between old and new stagger angle, degree</td>
</tr>
<tr>
<td>( C_m )</td>
<td>Mean stream line chord length, mm</td>
</tr>
<tr>
<td>( T )</td>
<td>Torque applied, N-m</td>
</tr>
<tr>
<td>( B )</td>
<td>Width of the plate, mm</td>
</tr>
<tr>
<td>( L )</td>
<td>Length of the plate, mm</td>
</tr>
<tr>
<td>( I )</td>
<td>Moment of inertia, ( \text{mm}^4 )</td>
</tr>
<tr>
<td>( U_d )</td>
<td>Distortion energy density, W m(^3)/kg</td>
</tr>
<tr>
<td>( E )</td>
<td>Modulus of elasticity, N/m(^2)</td>
</tr>
<tr>
<td>( x_{mn} )</td>
<td>( x ) coordinate on the mean stream line</td>
</tr>
<tr>
<td>( y_{mn} )</td>
<td>( y ) coordinate on the upper surface of the aerofoil</td>
</tr>
<tr>
<td>( y_1 )</td>
<td>( y ) coordinate on the lower surface of the aerofoil</td>
</tr>
</tbody>
</table>

Various design templates have been used over the years in designing of the pump impeller blade. The first design template for the mixed flow pump was put forth by Wislicensus [1] which was latter on modified by Myles [2], Stepanoff [3], Neumann [4], Gahlot and Nyiri [5] have developed and suggested their own design templates for mixed flow pumps. Experimental results were used by Varchola and Hlbocan [6] in development of design template for a mixed flow pump.

The inverse time marching method [7], the pseudo-stream function method [8], and the Fourier expansion singularity method [9] were some of the real 3D inverse design methods. However, the proposed 3D inverse
design methods such as inverse time marching method and pseudo-stream function methods takes significant amount of time in development of design and also difficulties are encountered while correlating the various design parameters with the geometry of the blade. On the other hand the Fourier expansion singularity method is confronted with the convergence problem. To address the major challenge of time, a quasi-3D method had been proposed [10] that performs a blade-to-blade solution. Further the method uses one representative surface from hub-to-shroud of the impeller blade. The approach resulted in significant computational time saving. In view of saving computational time, Anagnostopoulos [11] had developed a fast numerical method for analysis of fluid flow and hence in designing of the pump impeller blades. Assumptions of inviscid fluid is considered while developing the all the above models for designing of pump impeller.

Computational fluid dynamics was employed by Li et al. [12] for carrying out the optimum design of the pump impeller blade. Spalart-Allmaras turbulent model was the basis used for simulating the three-dimensional flow field in the whole flow passage of a mixed-flow pump. The simulation results aided in establishing the reason as to why the flow rate of the pump was not able to match the design requirements. Similar simulations were carried out for a low-specific-speed high-speed centrifugal pump by Jafarzadehet al. [13] using CFD code. The governing equations related to flow filed were solved using commercial CFD code and then the results were used for creating optimum design of the pump. Similar simulations results were obtained using CFD code and used for optimum designing of pump impeller blade by Chaudhari et al. [14] and Desai and Naik [15].

There have been numerous attempts by the researchers in studying the effect of various forces on the blade. Numerical investigations were carried out for the hydrodynamic radial forces for several flow rates in a mixed-flow pump by Mingxiong et al. [16]. The analysis was carried out using CFD simulations and validations. Significant influence on hydrodynamic radial forces and pressure fluctuations of the recirculation flow pattern under part load conditions were reported by the study. Dynamic and static stability of rotating impeller blade was analysed for axial periodic force by Sakar and Sabuncu [17]. Blade was found to be more stable for increasing rotational speed and disk radius. Li et al. [18] investigated the erosion effect of the water drop on a turbine blade. The damage to the piping system because of the pressure fluctuations arising due to excitations of the centrifugal pump at the blade passing frequency was studied by Kaneko and Hayashi [19]. Kikuyama et al. [20] investigated the effect of the inlet swirl on static pressure acting on impeller blades of centrifugal pump.

Performance enhancement is another crucial factor that needs to be considered while carrying out the optimum design of the pump. An analysis to enhance the performance behaviour of single flow pump for the unsteady flow behaviour was carried out by Pei et al. [21]. The oscillations induced for an impeller with
unsteady flow were numerically and experimentally investigated by Pei et al. [22]. Structural and thermal analysis are also crucial in deciding the performance of a pump. In this regard, a number of researchers [23-25] as for instance Ramamurti and Balasubramaniam [26], Jonker and Essen [27], Lemeš and Nermina [28], Bhope and Padole [29], Arewar and Bhope [30] and Das et al. [31] have contributed their efforts and studied the stresses for highly complex blade profiles of a wide range of turbo machinery. Kan et al. [32] carried out their investigation for dynamic stress analysis of impeller blade based on bidirectional fluid-structure interaction. CFD analysis of centrifugal pump impeller for the performance enhancement was carried out by Kumar et al. [33]. Kocaaslan et al. [34] numerically investigated the effect of number of blades on the pump performance. Design and stress analysis for impeller blade designed using free vortex theory was carried out by Srivastava et al. [35]. The von misses stresses was analysis and compared for impeller blade of mixed flow pump at different positions in the meridional annulus.

Owing to the importance of efficient design and performance analysis of the pumps, in the present work, the design of the impeller blade of the mixed flow pump using Mean Stream line method has been carried out. The 3D modelling has been done in Solid Works Premium Software 2014. The analysis for equivalent stress, maximum principal stress, equivalent elastic strain, maximum principal elastic strain, minimum principal elastic strain, total deformation and directional deformation has been carried out for the blade designed using Mean Stream line theory. A comparison analysis has been carried out for the equivalent stress in blade designed using free vortex method. Comparison analysis has been carried out for different construction material of the blade i.e., Titanium alloy, Stainless steel, Copper alloy and Bronze. Specification of the mixed flow pump under consideration was kept identical for both the methods of design templates. Analysis of stress was carried out using commercially available finite element software i.e. ANSYS 14.0.

2. Specifications of Pump

The design and its analysis had been carried out for a pump with a specific speed ($N_s$) of 105.74 rpm.

Assuming,

$$D_2 = 250 \text{mm}$$

$$\eta = 0.76$$

$$P = 15 \text{ kW}$$

Following assumptions have also been made for designing the blade:

- One dimensional approach,
- Constant meridional velocities at inlet and outlet impeller sections,
• No meridional curvature,
• Straight hub to shroud contours
• Un-shrouded impeller

3. Specific Speed

It is defined as the speed in revolutions per minute at which a geometrically similar impeller would operate if it were of such a size as to deliver one cubic meter per second against one meter of hydraulic head. Specific speed characterizes the performance for a pump handling a particular fluid and is given by the Eq. (1):

\[ N_s = \frac{N\sqrt[4/3]{Q}}{(H)^{1/4}} \]  

In the present work the blade of the mixed flow pump has been to produce a discharge (Q) of 0.125 m³/sec, working under the hydraulic head (H) of 5 m and rotational speed of 1000 rpm. The operating conditions are inserted in Eq. 1, yielding a specific speed (N_s) of 105.74 rpm. The impeller blade had been designed with respect to the linear distribution of pressure from inlet to outlet and from hub to tip.

4. Initial Design Layout of Mixed Flow Pump Impeller Blade

Figure 1 delineates the step for 3D modelling of the impeller blade using mean stream line theory. Stepanhoff [3] had plotted the variation of diameter ratio (e) with the specific speed (N_s) of the pump. The variation has been depicted in [36]. The calculations for the initial design parameters of the blade i.e., D_2i, D_10, D_20 and D_m0 have been tabulated and shown in [36].

Taking into account the initial design parameters of the blade, the layout of the blade was obtained in AUTOCAD 2013 by choosing a suitable semi-cone angle (ψ/2) of 30°. The layout has been delineated in [36] and was used for the calculation of the remaining initial parameters of the blade. The values obtained have been tabulated in [36].

5. Final Design Layout of the Mixed Flow Pump Impeller Blade with Consideration to Flow Separation

When the boundary layer travels to far off distance in the backdrop of adverse pressure gradient and the speed of boundary layer falls to zero relative to the object then the conditions becomes favourable for flow separation to take place. Under such situations the fluid flow detaches itself from the surface of the object and takes the form of vortices and eddies. Vibrations in structure may result in the aftermath of the shedding vortices and serious failures results when the frequency of the shedding vortices matches the resonance frequency of the structure. Therefore it becomes necessary to avoid flow separation in the impeller blade of the pump. To remedy this effect, Myles [2] suggested that the ratio D_m0/D_mi should be 1.20. Final Geometrical parameters of the impeller
blade were then obtained avoiding the phenomenon of flow separation. The calculations have been shown in [36]. The final layout of the impeller blade was then obtained using the calculated values and has been delineated in Figure 2.

6. Design Methodology

The 3D model of the impeller blade was obtained following the below mentioned design steps:

6.1 Calculation of Camber Line Chord Length, Stagger Angle, Radius of curvature of camber line

The blade (A-A’-J-J’), Figure 2, was divided into ten sections. For each one of the sections Camber line chord length(C’) and its stagger angle(λ) was obtained and depicted in [36]. Calculations for the inlet and outlet circumferential velocities, inlet velocity of flow, the fluid inlet angle ‘θi’, the tangential component of the velocity ‘Vω’, the fluid outlet angle ‘ϕ’, the slip velocity (V S) arising due to the generation of eddies, the modified outlet blade angle, the mean blade angle ‘α m’, Pitch ‘S’ and the inlet Di (mm) and outlet diameters Do (mm) have been shown in [36]. These calculations were done for different sections of the blade. The results obtained have been tabulated in [36]. Using the above calculations, the actual chord length of the camber line and its stagger angle were obtained for different blade sections using an iterative procedure as delineated in [36]. The iterations were carried out till the difference between old stagger angle and the modified stagger angle is 0.01°. The radius of curvature of the camber line ,R was also obtained for different sections of the blade. These calculations have also been tabulated in [36]. Using AUTOCAD 2013, the chord and the camber lines were plotted for different sections of the blade and the plot has been shown in [36]. Deviation rules were followed while making the calculations [37].

6.2 Chord length and Radius of Curvature of Mean Stream Curve

The flow velocity can be geometrically represented through streamlines. The velocity vector can be defined in terms of time and space coordinates in accordance with the Eulerian method. A streamline is obtained when a space curve is drawn in such a way that it is tangent to velocity vector at a fixed instant of time. Therefore, the Eulerian method gives a series of instantaneous streamlines of the state of motion.

Streamlines were obtained for different sections of the impeller blade in the present work. The procedural steps followed to obtain the streamline for a particular section of the blade have been delineated in [36]. Mean stream line for section J-J’ of the impeller blade is depicted in Figure 3 [38]. The above procedural steps, once followed will result in different geometric parameters for different sections of the impeller blade i.e., length of chord for mean stream line(C’), stagger angle (λ) for the mean stream line chord and the radius of curvature of the mean stream line curve (R). The measured results have been tabulated [36].
Curve fitting was done for the mean stream line chord length and also for its radius of curvature to obtain a smooth blade profile and the final results have been retabulated and depicted in [36].

6.3 2D Profile of the Blade Section in Solidworks Premium 2014

To generate the complete blade profile, the coordinates of the upper and lower surface of the mean stream line are required to be calculated. The calculations have been shown in [36]. Figure 4 depicts the 2D profile for a particular blade section of the impeller blade.

6.4 3D modeling of the pump impeller blade

Procedural steps of section 6.1-6.3 were followed and two dimensional blade profiles of different sections of the blade were obtained. The different blade profiles of the impeller blade were stacked together at their respective stagger angles and the 3D model of the impeller blade was obtained, depicted in Figure 5.

7. Analysis of the blade in ANSYS 14.0

Having the importance of blade stresses acknowledged, the analysis of the stresses needs to be of high accuracy. The calculation of stresses and strains in a mixed flow pump impeller blade is extremely complicated due to the complex loading characteristics and the geometry of the blade. To get an accurate estimation of the stresses in the blades, validation of stress values are required to be compared with the calculated values. To accomplish the above, a simplified method of stress validation among the calculated and numerically predicted values was carried out by replacing the twisted blades with an equivalent plate having rectangular crosssection, which acts like a cantilever. The material properties and the volume of both plate and blade were kept identical. The theoretical calculation of Von Misses stress was carried out using the Eqs. (2-10). It is assumed that each blade of the mixed flow pump impeller acts as a cantilever.

The dimensions of the equivalent plate are: 58×162.72×10.41.

Torque applied on the plate:

\[ T = \frac{wt^2}{2} \]  

Also,

\[ T = \frac{60P}{2\pi NZ} \]  

From Eqs (32-33), Eq.4 was obtained,

\[ w = \frac{60P}{\pi NZ lb^2} \]  

Where,
Therefore,

\[ w = 10588 \text{ N/m} \]

For a rectangular beam in pure bending, the shear stress is given by the Eq. 5

\[ \tau = F \cdot \frac{Ay}{Ib} \]  \hspace{1cm} (5)

Substituting the respective values, the shear stress calculated is given by Eq. 6

\[ \tau = 2750 \text{ N/m}^2 \]  \hspace{1cm} (6)

The bending stress \( \sigma_b \) was calculated using Eq. 7.

\[ \sigma_b = \frac{My}{I} = \frac{3wl^2}{bh^2} \]  \hspace{1cm} (7)

The values obtained are substituted in Eq. (6) to obtain the value of \( \sigma_b \) i.e:

\[ \sigma_b = 12.08 \times 10^6 \text{ N/m}^2 \]

Distortion energy theory was then used to obtain the equivalent stress, according to which, a ductile solid will yield when the distortion energy density reaches a critical value for that material. Eq. 8 was used to calculate the distortion energy density associated with yielding:

\[ U_d = \frac{1 + \nu}{3E} \sigma_y^2 \]  \hspace{1cm} (8)

Thus, the energy density given in Eq. 8 is the critical value of the distortional energy density for the material. Then according to von Mises’s failure criterion, the material under multi-axial loading will yield when the distortional energy is equal to or greater than the critical value for the material:

\[ \frac{1 + \nu}{3E} \sigma_{VM}^2 = \frac{1 + \nu}{3E} \sigma_y^2 \]

\[ \therefore \sigma_{VM} \geq \sigma_y \]  \hspace{1cm} (9)

in terms of general stress components,

\[ \sigma_{VM} = \sqrt{\sigma_y^2 + 3\tau^2} \]  \hspace{1cm} (10)

\[ \therefore \sigma_{VM} = 12.8 \text{ MPa} \]

Once the value of the Von Mises stress was calculated for the plate, FEM convergence test was carried on the plate with different size of the element to find out the optimum size of the element using ANSYS 14.0. Using
the optimum element size, numerical stress analysis for the pump impeller blades was carried out. The Von Misses stresses were calculated considering the material properties of Stainless steel. The optimal size of the element was obtained to be 0.52 mm. Here, tetrahedral element was used for the analysis. Span wise variation of Von Misses stress for the plate for an element size of 0.52 mm is shown in Figure 6. Since the impeller blade is required to be subjected to surface force density, calculations were done for the surface force density. This calculation was done owing to the complex nature of the blade geometrical profile. This was then applied onto the blade surface for carrying out the stress analysis. The surface area of the entire blade was found to be 0.018 m² and therefore the calculated surface force density was 69977 N/m². Figures (7-12), shows the analysis results for stainless steel blade obtained using ANSYS 14.0. Similar procedure was adopted to obtain the values of different physical quantities of interest i.e., equivalent stress, equivalent elastic strain, Total deformation and the directional deformation for bronze alloy, copper alloy and titanium alloy.

8. Results and Discussion

The results for equivalent stress, equivalent elastic strain and total deformation obtained were compared for different construction material of the modeled blade. The comparison results are shown in Figures 13-15. As can be seen from Figure 13 the equivalent stress is least in copper for the modeled blade while it is maximum for stainless steel. The comparison of the equivalent strain result shows that the value is least for stainless steel whereas it is maximum for bronze. The comparison has been shown in the Figure 14. A quick glance at Figure 15 shows that the total deformation is least for bronze and maximum for stainless steel. The variations in the equivalent stress, equivalent strain and total deformation value may be attributed to the different mechanical properties for different construction material of the blade.

Table 1 below shows the results obtained for directional deformation in X, Y and Z axes for different construction material of the blade. The results clearly show that the directional deformation is least in the X-axis for different construction material of the blade. The deflection in Z and Y axes are almost similar for different construction material of the blade. The fact may be attributed to the geometric profile of the modeled blade.

9. Conclusion

- The design of the impeller blade using mean stream line theory is comparatively more tedious and cumbersome than designed using other design templates yet the results are at par with other templates of design.
According to the equivalent stress results, copper can be selected as the construction material for the modeled blade for the construction material considered in the present work. Table 2 shows the equivalent stress for different materials of the blade which has been designed using free vortex theory (Srivastava et al. 2014).

Since the stress in the Titanium alloy blade is the least, therefore according to the free vortex theory one can select titanium as the material for the blade for the construction material considered in the present work.

Figure 16 shows the comparison of variation of equivalent stresses for different materials of the blade, which has been designed using mean stream line theory and the free vortex theory. The comparison of variation of stresses shows that the stresses for blade designed using mean stream line theory are always on the higher side as compared to the blade designed using free vortex theory. The difference in stress area accounted for the difference in the stress values.

The total deformation results show that if it all it is considered as the criteria of selection of construction material for the modeled blade, then titanium could be choosen over the other material of construction under study.

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References


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Edited Book Volume, 15 Book Chapters, 110 international Journal publications, 18 International and 8 National Conference publications to his credit. He is on the editorial board and review panel of 7 International and 1 National Journals of repute. He has been felicitated with many awards and honors.

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<table>
<thead>
<tr>
<th>Material</th>
<th>Stainless steel</th>
<th>Bronze</th>
<th>Titanium</th>
<th>Copper</th>
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<td>Directional deformation X-axis (mm)</td>
<td>1.590 e^-5</td>
<td>2.935 e^-5</td>
<td>3.540 e^-5</td>
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<td>Directional deformation Y-axis (mm)</td>
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<td>Directional deformation Z-axis (mm)</td>
<td>0.0007</td>
<td>0.0013</td>
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<td>24.322</td>
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