Design of a mixed flow pump impeller and its validation using FEM analysis

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Abstract

The design of mixed flow pump impellers of high specific speed is a much tougher than the other types of flow impellers and it becomes more complicated while deciding the blade positioning in the meridional annulus. In this work, natural frequency and deformation of mixed flow pump impeller were evaluated considering two different blade positions in the meridional annulus. ANSYS was used for the investigation of natural frequency and deformation. It was observed that the mixed flow pump impeller with inlet inclined blade position in the meridional annulus was more suitable than the trapezoidal one.

Keywords: Mixed flow pump impeller, meridional annulus, ANSYS, natural frequency, deformation

1. Introduction

The mixed flow pumps are generally being used in large thermal power plants for cooling water duties. The mixed flow pump is a unification of radial & axial characteristics. The design of mixed flow impellers of high specific speed is a direct extension of the well established empirical methods of the designing of radial flow impellers. The extension of similar methods serves for the design of mixed flow impellers, but the introduction of near diagonal flow layout at a still larger specific speed stimulated the incorporation of axial pump impeller design techniques in

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mixed flow pump technology.

The usual industrial design practice of mixed flow pump impeller design starts with the estimation of approximate meridional streamlines by dividing the annulus by the equal area method. Empirical co-efficient depending on the specific speed used to fix the inlet and outlet blade angles. Similar co-efficients are used to determine overall impeller layout before the meridional streamlines are estimated. The inlet and outlet angles being fixed in this manner, the blade sections are laid out on the developed stream surfaces.

The above industrial method is basically based on some empirical co-relations and design constants [Hao et al. (2013)]. The industrial design method often ignores the actual happening within the pump flow passage and is consequently a poor guide when the question of a new design & development of pumps comes to picture. In the above design process the designer has less control than desirable over events. The lack of clear cut rational basis also inhibits the correction of manufacturing of shortfalls in expected performance. To overcome the above difficulties, one has to formulate a rational basis for the designing of impellers starting from basic principles, such that the use of empirical co-relations is minimized. Such a design from the basic principles has advantages that the designer will have more control over the outcome of his design, while keeping the physical principle constantly in view and enables him to rectify any faults in the performance of the pump.

**Nomenclature**

| Symbol | Description |
|--------|-------------|
| \(C_m\) | meridional velocity, m/sec. |
| \(D\) | blade diameter, mm |
| \(g\) | acceleration due to gravity, m/sec\(^2\) |
| \(H\) | pressure head, m |
| \(I\) | blade span, mm |
| \(N\) | rotational speed, rev./min. |
| \(P\) | power, kW |
| \(Q\) | volumetric discharge,m\(^3\)/min. |
| \(r\) | radius,mm |
| \(S\) | blade spacing, mm |
| \(u\) | tangential velocity, m/sec. |
| \(V_S\) | slip velocity,m/sec |
| \(Z\) | number of blades |
| \(\lambda\) | blade stagger angle,degree |
| \(\rho\) | mass density of water,kg/m\(^3\) |
| \(\sigma\) | Von Mises stress, MPa |
| \(\sigma_B\) | bending stress,MPa |
| \(\tau\) | shear stress, MPa |
| \(\phi\) | semi cone angle of the impeller, degree |
| \(\Omega\) | dimensionless specific speed |
| \(\omega\) | angular velocity, rad./sec. |

Over a period of time research on mixed flow pump has been carried out by various researchers. Wislicenus (1965) initiated the design of a mixed flow pump impeller. The modification of mixed flow pump impeller was carried out by Myles (1965). Busemann (1928) developed a formula for slip velocity for a mixed flow pump. Senoo and Nakase (1972), Inoue et al. (1980) developed a design method by calculating the meridional stream line. A.J. Stepanoff (1967) gave a design procedure for mixed flow pump impeller. Neumann’s (1991), Gahlot and Nyiri (1993) have suggested the step-by-step design procedure for designing Mixed flow pumps. Yumiko Takayama and Hiroyoshi Watanabe (2009) presented a multi-objective optimization strategy of mixed-flow pump design by means of Three Dimensional Inverse Design Approach. Jim- Hyuk Kim & Kwang-Yong Kim (2011) developed an optimization procedure for high efficiency design of mixed-flow pumps. Hao et al. (2013), Mehta and Patel(2013) studied the effects of meridional flow passage shape on hydraulic performance of mixed-flow pump impellers.

K. Sham Sunder (1981) presented a three-dimensional method of stress analysis using finite element techniques for determining the stress distribution in centrifugal impellers. Ramamurti and Balasubramanian (1987), Jonker and Van Essen (1997), Jonker and Van Essen (1997), Samir Lemeš and Nermina Zaimović-Uzunović (2002), Bhope and Padole (2003), Arewar and Bhope(2013) contributed much on the stress analysis for highly complex blades of various turbomachines.

In this present work, design of a mixed flow pump impeller was carried out for two different blade positioning in the meridional annulus based on basic principles of fluid mechanics and turbo machinery. The above models were compared on the basis of natural frequency and impeller blade deformation using ANSYS 11 for design acceptability.
2. Design methodology

Here the design of the mixed flow pump impeller was based on free vortex theory. The stream surfaces through the meridional annulus were kept parallel to hub and casing, whereas, the hub and casing are parallel to one another. It was also assumes that the meridional velocity distribution remains uniform across the annulus [Myles (1965)]. In the present case the mixed flow pump impeller having the following specifications: discharge (Q) = 0.125 m³/sec., head developed (H) = 5 m, speed of rotation (N) = 1000 rev/min.

2.1 Design procedure

Calculation of non-dimensional specific speed (Ω) using the relation,

\[ \Omega = \frac{60\sqrt{Q}}{(gH)^{1/4}} = 1.998 \text{ rad./sec} \]  

(1)

The inlet diameter (D₁) of the impeller has been calculated using the relation

\[ H = \frac{1}{2gK_u} \left[ \frac{\pi D_1 N}{e} \right]^2 \]  

(2)

Where, \( e = \frac{D_1}{D_2} \) \hspace{1cm} (3)

\( K_u = \frac{u^2}{\sqrt{2gH}} \) \hspace{1cm} (4)

The values of \( e \) and \( K_u \) for a given specific speed were taken as suggested by Stepanoff (1957).

The ideal power requirement is calculated using the relation

\[ P = \frac{Q H}{1000} \text{ kW} = 15 \text{ kW} \]  

(5)

The choice of cone angle of the mixed flow pump impeller was chosen as 60⁰ as suggested in [Myles (1965)]. The preliminary layout of the blade profile of the impeller is shown in Fig. 1. From the Fig. 1, the inlet diameters (D₁₁, D₁₂) and outlet diameters (D₂₁, D₂₂) of the impeller were measured. Then the meridional velocity at inlet (C₁₁) and outlet (C₁₂) were calculated using the relation:

\[ C_m = \frac{Q}{2\pi l} \]  

(6)

For a proper design, \( C_{m2}/C_{m1} \) should be in between 1.2 to 1.22 [Myles (1965)]. So, to keep the value of \( C_{m2}/C_{m1} \) within the specified limit, the rectangular position of the blade in the meridional annulus has been modified, resulting in two different blade positioning in the meridional annulus as shown in Fig.2. The blade span was divided into ten different sections parallel to hub and casing.
Calculation of blade angles was carried out using the following relations:

\[ u_1 = \frac{\pi D_1 N}{60} \text{ m/sec} \]  

(7)

\[ u_2 = \frac{\pi D_2 N}{60} \text{ m/sec} \]  

(8)

\[ \tan \alpha_1 = \left( \frac{u_1}{C_{m1}} \right) \]  

(9)

\[ \tan \alpha_2 = \left( \frac{u_2 - C_{\alpha_2} - V_S}{C_{m2}} \right) \]  

(10)

where, \( V_S = \frac{-m \cos \alpha_2 \sin \phi}{Z} \)  

(11)

mean blade angle,

\[ \tan \alpha_m = \left( \frac{\tan \alpha_1 + \tan \alpha_2}{2} \right) \]  

(12)

pitch,

\[ S = \frac{\pi D_m}{Z} \]  

(13)

and

\[ D_m = \left( \frac{D_1 + D_2}{2} \right) \]  

(14)

actual chord,

\[ C = \frac{\lambda}{\cos \lambda} \]  

(15)

The blade stagger angle \( \gamma \) has been calculated using Carter’s correlation. The blade angles calculated for each section along the blade span using above relations have to be verified to satisfy Leiblein blade diffusion factor \( D_L \) to avoid separation of boundary layer from the blade surfaces. The Leiblein blade diffusion factor

\[ D_L = \left[ 1 - \frac{\cos \alpha_1}{\cos \alpha_2} + \left( \frac{S}{C} \right) \frac{\cos \alpha_1}{\cos \alpha_2} (\tan \alpha_1 - \tan \alpha_2) \right] < 0.6 \]  

(16)

2.2 Construction of 3D model of the impeller and FEM analysis

Once the blade angles were calculated, the straight two-dimensional cascade was transformed into the corresponding conical cascade using conformal transformation method. The blade sections were staged one over another by maintaining proper stagger angle. Finally, using surface revolution method, 3D model of the mixed flow pump impeller has been prepared. In the present work the optimal number of blade was found to be eight. Here, NACA 10C4 blade profile was used for the impeller blades as suggested in [Inoue et al. (1980)]. To find out the natural frequency and pump impeller blade deformation in the above impellers finite element analysis (FEM) was carried out using ANSYS software.
3. Results and discussion

The design parameters such as blade angles, camber angles, stagger angles, chord dimension were calculated at different sections along the blade span for two different blade positioning in the meridional annulus. Span wise variation of blade angles for inlet inclined blade position (case-I) and for trapezoidal blade positioning in meridional annulus are shown in Fig.3 and Fig.4 respectively.

Once the design of the impeller blades was completed, 3-D model of the impeller was prepared for the case-I and case-II and are shown in Fig. 5 and Fig. 6 respectively. Before proceeding to FEM analysis on the pump impeller, estimation of optimum size of element for FEM analysis was carried out by replacing the twisted blades with an equivalent plate having rectangular cross-section, which acts like a cantilever. The material properties and the volume of both plate and blade were kept identical. The theoretical calculation of Von Mises stress was carried out using the following relation:

\[ \sigma = \sqrt{\sigma_{1}^2 + 3\sigma_{2}^2} \]  

(17)

While calculating the Von Mises stress, material properties of Bronze were taken into consideration. Fig. 7 shows the analysis to find out the optimal element size for FEM analysis. From Fig. 8 it is observed that the optimal size of the element is 4.25 mm. Here, tetrahedral element was used for the present analysis. Span wise variation of Von Mises stress for the plate for an element size of 4.25 mm is shown in Fig. 8. Using the optimum element size,
numerical stress analysis for the pump impeller was carried out for natural frequency and deformation of impeller blades.

The variation of natural frequency at six different modes for pump impeller having inlet inclined blade positioning (case-I) and trapezoidal blade positioning (case-II) in the meridional annulus are shown in Table 1.

| Modes of natural frequency | Inlet inclined (case-I) | Trapezoidal (case-II) |
|----------------------------|------------------------|-----------------------|
| 1                          | 1948.5 Hz              | 1849 Hz               |
| 2                          | 1948.9 Hz              | 1849.7 Hz             |
| 3                          | 2080.6 Hz              | 1860.8 Hz             |
| 4                          | 2081 Hz                | 1861 Hz               |
| 5                          | 2146.2 Hz              | 1926.1 Hz             |
| 6                          | 2148.3 Hz              | 1927.6 Hz             |

The variation of displacement with natural frequency for pump impeller with inlet inclined blade positioning is shown in Fig. 9(a-f). The variation of displacement with natural frequency for pump impeller with trapezoidal blade positioning is also shown in Fig. 10(a-f).
4. Conclusion

It can be concluded from the above results that the mixed flow pump impeller having inlet inclined blade positioning (case-I) in the meridional annulus, is getting lesser deformed than the trapezoidal blade positioning (case-II) which itself signifies that the impeller with inlet inclined blades positioning is a better choice than the other one. Moreover the higher values of natural frequency at six different modes for case I than the case-II also confirm that the mixed flow impeller of case I is more stable at higher rpm than the impeller of case II.

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