Construction of Radial Flow Submersible Pump Double Stage Chamfered Impeller using a Simplified 3D Model Approach

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Abstract

This work entails construction of radial flow submersible pump double stage chamfered impeller using a simplified 3D model approach. Construction of 3D Model CAD Radial Flow Submersible Pump Double Stage Chamfered Impeller for Oil Drilling/Production is so necessary nowadays as crude oil production is in increase due to high demand of petroleum products globally. For such to be achieved something more meaningful have to be done in order to improve the working conditions of the Radial Flow Submersible Centrifugal Pumps manufactured by Gruppo Aturia, Italy used in Nigerian Oil Drilling for increase in the head, static pressure and total pressure of the modeled pump impeller but by increasing the blade number, the head of the pump increases slightly, it will be recompensed by decrease of efficiency as revealed by Esfahani et al (2014). This work is aimed at providing a Simplified 3D Model Radial Flow Type Submersible Centrifugal Pump of Double Multi-Stage Chamfered Impeller Forward Bent Blades Profile Design, Procedure and Development of CAD 3D Model Pump using AutoCAD R2010 software. The overall dimensions of the existing impeller made by Gruppo Aturia- Italy were not changed but only the variation of its blades number (from Six “6” blades to Ten “10” blades) which simultaneously reduced the blades exit angles (from 25° to 10°). This design proved the work of Pandey et al. (2012) to be possible which “investigated the numerical studies on effect of blade number variations on performance of centrifugal pumps at 2500rpm. He determined that with the increase of blade number, the head and static pressure of the model increases and total pressure too”.

Keywords:
3D Model, CAD, Radial Flow, Submersible Centrifugal Pumps, Gruppo-Aturia.

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Background to the Study
A submersible pump is a type of centrifugal pump that has its whole assembly submerged in the fluid to be pumped. Although their constructional and operational features have undergone a continuous evolution over the years, yet their basic operational principle remained the same. Produced liquids, after being subjected to great centrifugal forces caused by the high rotational speed of the impeller, lose their kinetic energy in the diffuser where a conversion of kinetic to pressure energy takes place (Kaewnaniet et al. 2008). This is the main operational mechanism of radial and mixed flow pumps. Submersible centrifugal pump find its application in sewage pumping, industrial and slurry pumping. They are also used for residential, commercial, municipal and industrial extraction, water wells and in oil wells (Klin, 2008).

Submersible pumps are used in oil production to provide a relatively efficient form of "artificial lift", able to operate across a broad range of flow rates and depths. By decreasing the pressure at the bottom of the well (by lowering bottom hole flowing pressure, or increasing drawdown), significantly more oil can be produced from the well when compared with natural production (Gurupranesh et al. 2012). The pumps are typically electrically powered and referred to as Electrical Submersible Pumps (ESP).

Generally, centrifugal pump is a very simple machine and its major components consist of a set of rotating vanes, called impellers which are enclosed within a stationary housing known as casing. Impellers help in moving the fluid to a higher level by rotation. The inlet of the pump draws fluid into the centre of the rotating impeller where the whirling motion of the impellers accelerates the fluid outward between the vanes until it is thrown from the periphery of the impeller into the casing. The casing collects the fluid, converts a portion of its velocity energy into pressure energy, and directs the fluid to the pump outlet (Ashutosh, 2008).

Again, various parameters affect the performance of a centrifugal pump such as the impeller inlet and outlet diameter, inlet and outlet blade angle, the numbers of blades, blade thickness, and blade sweep. These play important roles in adding energy to the fluid and one need to understand the effects of these parameters in order to arrive at optimum efficiency of the pump (Ashutosh, 2008).

Radial Flow Type Submersible Centrifugal Pump
Radial Flow Type Submersible Centrifugal Pump, Forward Bent impeller’s blades; are most widely used impellers in the Oil exploitation and production nowadays because of its numerous advantages. In recent years lots of efforts are being made by pump manufacturers to simplify the Radial Type Impeller blade “vane” profile design procedures with improvement in pump performance. There are limited numbers of published blades profile design procedures. There was lack of explanation as well as detailed step by step procedure available for the designer to systematically design and develop forward bent type blades profile.
The vane profile design procedure presented by Tuzson (2013) uses simple computer programs and their program listing has to be used to execute calculation steps. Implementation of different computer programs have to be executed to calculate variables required to design the whole impeller. The variables such as radius of curvature cannot be used directly to generate the vane profile because this procedure recommends for using the Computer Aided Design software to construct the vane profile because the methods to fix the center of curvature and starting point of arc is not explain. To design a vane profile it requires design data such as inlet and outlet blade angle, inlet and outlet diameter of impeller and radius of curvature also blades should produce uniform angular momentum distribution.

The vane profile design procedure presented by Lobanoft (2013) has similar kind of deficiency. The procedure explains about the selection criteria of vane angle and the impeller diameter but the problem arises while constructing the vane profile because basic relation to calculate the radius of curvature, method to fix the center of curvature and starting point of arc is not clearly specified in the procedure. In industries, sometimes a popular geometric method is also being used for many years.

**A Simple Geometric Method of Constructing Vane Profile**
The simple method published by Sahu (2013) of constructing vane profile is to draw a single radius circular are using the calculated angles $\beta_1$, $\beta_2$ and radii $R_1, R_2$ but this may have serious implications on the performance of the pump. In this first line AM is drawn which makes an angle $\beta$, to AO as will be shown later in (Fig. 2.20). Then angles of $\beta_1$, $\beta_2$ are drawn at O with the radius OB and a line is drawn from A to point B, the intersection point on the radius $R$, and is extended up to D. Then a perpendicular line is drawn in the middle of AD which intersects at MA will be the radius of arc and arc AD will be the vane profile.

![Figure 1.1: A simple Geometric method of Constructing Vane Profile](image)

Another method suggested by Stepanoff *et al.* (2013) is brief explained as follows: first the selection for inlet and outlet vane angles $\beta_1$, $\beta_2$ is to be done. The usual angles used for $\beta_1$, $\beta_2$ are 13° and 20° as recommended by Stepanoff *et al.* (2013). For smooth flow, the vane must be designed such that these angle increases smoothly from 13° to 20°. The graph of vane angle $\beta$ against vane radius $r$ for inlet and outlet station is to be plotted as shown in Figure 2.21 to obtained intermediate values of radii $r$ corresponding to intermediate values of the angle $\beta$. Then Plot the radii against position angle $\theta$ to give the shape of the trailing edge of the vane.
Use sufficiently close spacing of radii to obtain smooth shape of vane profile. Here \( \theta \) can be calculated by using the relation \( d\theta = dr/\tan \beta \). To calculate the value of \( \theta \) i.e., to solve the given equation software’s like MATLAB or other computer software should be used. Here in this procedure the method to locate the start point of curve, centre of curvature and the relation to calculate the radius of curvature is not clearly specified, so the designer will meet the problem while constructing the vane profile.

The Singularity Method used for Methodology in the First Stage-Step One of Impeller Development

There are some other methods like the singularity method as inverse problem. In this method a numerical approach is used. Wrapping angle of blade is found by applying trapezium rule. Here, a cubic Bezier curve has been drawn with the help of a control polygon. A mathematical expression has been used to express the density of bound vortex intensity. The blade angle and warping angle has to be calculated which will give a way to generate modified camber line

![Graph](image)

Figure 1.2: A Numerical Approach; A graph for Blade Angle “\( \beta \)” against Blade Radius “\( R \)”

Methodology

The Flow Chart for the Impeller Development and Modeling

The flow chart below (Fig. 2) was developed for the successful completion of the impeller Development and Modeling was conducted using a Simplified 3D Model approach.
Impeller Development and Procedures Using Singularity Method for Impeller Development

Let $\beta$ and $\beta'$ be the inlet and outlet angles of this impeller respectively. Selection of $\beta$ is made generally for an optimum efficiency. An average value of $22.5^\circ$ is called normal for all specific speeds. The limit of $\beta$ in a good design is from $9.5^\circ$ (minimum) to $39.5^\circ$ (max). The values for $\beta$ and $\beta'$ are selected as $10^\circ$ and $40^\circ$. For smooth flow, the vane must be designed such that this angle increases smoothly from $10^\circ$ to $40^\circ$. The next step is to construct the vane shape. There are several methods to construct the vane shapes. The one used in practice consists of tangent circular arcs. The radius of the circular arc contained between the rings $D_1$ and $D_2$ is given by:

$$ R = \frac{R_2^2 - R_1^2}{2(R_2 \cos \beta_2 - R_1 \cos \beta_1)} \quad (2) $$

Where, $R_1$ and $R_2$ are the radii of inner and outer diameters of the impeller respectively, while using this method, the diameter of the impeller is divided into a number of concentric rings not necessarily equally spaced. The value of $R$ for any two consecutive concentric rings is calculated using the equation 1.0 and vane shape is plotted which is actually an arc tangent to both the rings. An accurate vane shape can be obtained by joining the areas as shown below.
The radius of Inner diameter is given by \( \frac{165}{2} = 82.5 \) mm and the radius of Outer diameter is given by \( \frac{183.74}{2} = 91.9 \) mm. Now, the intermediate values of radius can be found out by \( \frac{(R_i - R_f)}{n} \) where, \( n \) = number of intermediate concentric rings required. For better resolution the value of \( n \) is taken as 11. The values thus obtained are mm. similarly, the corresponding values for the vane angle \( \beta \) can be found out graphically.

Table 2.1: Table for Calculated Values of Inner Radius “\( R_i \)” and Outer Radius “\( R_f \)” Of Radius of Curvature “\( r \)”.

<table>
<thead>
<tr>
<th>( R_i )</th>
<th>( R_f )</th>
<th>( \beta )</th>
<th>( B_1 )</th>
<th>( R )</th>
</tr>
</thead>
<tbody>
<tr>
<td>41.5</td>
<td>49.83</td>
<td>10</td>
<td>15</td>
<td>52.40</td>
</tr>
<tr>
<td>49.83</td>
<td>58.16</td>
<td>15</td>
<td>20</td>
<td>77.41</td>
</tr>
<tr>
<td>58.16</td>
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<td>74.82</td>
<td>83.15</td>
<td>30</td>
<td>35</td>
<td>198.77</td>
</tr>
<tr>
<td>83.15</td>
<td>91.5</td>
<td>35</td>
<td>40</td>
<td>350.56</td>
</tr>
</tbody>
</table>

Development of Outer/Inner Diameters and Shaft Housing of the Impeller

The step one is possible by using these values to generate the vane profile. Then AutoCAD R2010 software is used to develop the impeller geometry as follows:

The first step is to draw the circle of 183.74 mm diameter using radius 91.9 mm as a reference solid equal to outer radius of impeller by taking C as a centre point. Then following the inner circle diameter of 165mm to be drawn using 82.5 mm as the radius while a third circle of 82.50 mm to be drawn using 41.25 mm as the radius in order to develop the modified Blade Length. Others were circles diameters of 55 mm and 40 mm using 27.5 mm and 20 mm as radii for development of the Shaft’s housing as shown below in Fig. 2.1.

Figure 2.2: Development of Outer, Inner and Shaft Housing Diameters of the Impeller
Development of all the Concentric Circles Using a Simply Geometric Method
The second step is to develop all the concentric circles using radii \( R \) calculated as shown in Table 2 by taking C as a centre point. Line AB represents the value of R which makes an inclination of 10° with AC or X-axis. The line AB of length 52.40 mm is to be drawn by taking point A of line AC as start point. The angle between X-axis or line AC and line AB must be equal to \( \beta = 10° \) as shown in Figures 2.2.

Figure 2.3: Developing All the Concentric Circles

Development of First Radius of Curvature “AE”
The third step is to develop the first arc AE having inlet vane angle \( \beta = 10° \) and radius of curvature 52.40 mm. The arc AE of radius 52.40 mm is to be drawn by taking end point B of line AB as a centre point of arc and point A of line AC as a start point where this arc intersects the second concentric circle is taken as end point for arc that is, point E as shown below in Figure 2.3.

Figure 2.4: Development of First Radius of Curvature “AE”
**Development of the Second Radius of Curvature “EF”**

The fourth step is to construct the second arc EF of radius of curvature of 77.41mm. For that the line ED of length 77.41mm is to be drawn by taking point E of arc AE as a start point. The line should pass through the end point B of first line AB. Then the arc EF of radius 93.21mm is to be drawn by taking end point D of second line as a centre point of arc and intersection point E as a start point. The point where this arc intersects the third concentric circle is taken as end point for arc that is, point F as shown below in Figure 2.4.

The same procedure has to be repeated for the remaining values of R as shown in the Table 2. Finally, the Forward Bent Blades “Vanes” profile is achieved in the form of a continuous arc as shown below in Figure 2.5.

![Figure 2.5: Development of the Second Radius of Curvature “EF”](image1)

![Figure 2.6: Completed View of the Plain Vane Profile](image2)
Design and Modeling
Development of 2D And 3D Impeller Blades Structures
Since this impeller contains 10 numbers of blades “Vanes”, the fifth step is to use the circular array or pattern command to have 10 vanes around a Z-axis. To achieve the gradual increase in Blade Inlet Height (B1) to Blade Outlet Height (B2) that is from 25.1 mm to 27.8 mm a revolve cut operation is used. The 2D and 3D impeller blades “vanes” Structures were therefore obtained as shown below in Figure 3 to Figure 3.4 and finally the Modeling of the CAD 3D Model of the Pump Impeller as shown in Figure 3.5

Figure 2.7: Development of Ten Blade 2D Model Impeller

Figure 2.8: Development of Ten Blades 3D Model Impeller

Figure 2.9: Addition of Shaft Sleeve into the vane profile
Finally, the sixth step is the assembling of various parts to get the final 3D model of impeller as shown in Figure 3.5.

Figure 2.10: Modeled double-plain vane beveled impeller

Modeling of CAD 3D Model of the Pump Impeller

Figure 3.1: Rendered double-plain vane beveled impeller

Figure 3.2: Rotated plain chamfered vane straight impeller
Conclusion and Recommendation
In conclusion however, by increasing the blade number from six “6” to ten “10” blades as in the design, Esfahani et al (2014) suggested that by increasing the blade number, the head of the pump increases slightly, but it will be recompensed by decrease of efficiency and further proved that the moment needed to rotate impeller at constant rotational velocity of 2900 rpm increases by increase of blade number, this shows that impeller with more blade number exert more moment to fluid and consequently cause to produce more head. Reversely the moment for each blade decreases with increase of blade number. This shows that by increasing the blade number, the moment exerted on every blade decreases, so thinner blades can be used. By increasing the blade number, fluid passes more smoothly through the passage, but from the other hand, more surface friction and impeller volume occupation cause decrease in efficiency.

Pandey et al. (2012) also proved that as the blade number is increasing, the head and static pressure of the model increases and total pressure too, but the variable regulation of efficiency are complicated, but there are optimum values of blade number for each one. While Houlin et al. (2010) proved that the increase of blade number is helpful to reduce the mixture loss of “jet” and “wake” in centrifugal pump and the uniformity distributions of vapor in impeller channel became obvious with the increase of blade number. In other to improve the efficiency of the pump too, the thickness of the blades which is 6mm must be reduced or become thinner.

Reference


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