Education & Science Journal of
Policy Review and Curriculum Development
Hard Print: 2315-8425 Online: 2354-1660

November, 2016

Vol. 6, No. 1

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

[1]**S. I. Iwenofu,**
[2]**C. C. Nwobi-Okoye &**
[3]**I. U. Onyenanu**
[1&2]Department of Mechanical Engineering, C.O.O.U. Uli
[3]Innoson Kiara Academy, Enugu

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

Corresponding Author:
S. I. Iwenofu

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 effectsof 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".

http://internationalpolicybrief.org/journals/international-scientific-research-consortium-journals/edu-and-sci-jrnl-of-policy-review-and-curriculum-dev-vol6-no1-november-2016

Bakground 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 (Kaewnani*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 other 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 β_1, β_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 β_2 to AO as will be shown later in (Fig. 2.20). Then angles of β_1, β_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_1 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.

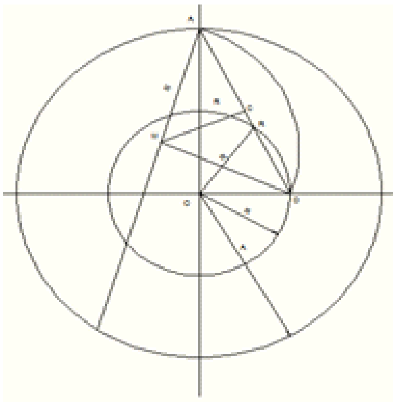

Figure 1.1: A simple Geometric method of Constructing Vane Profile

Another method suggested by Stepanoff *etal*. (2013) is brief explained as follows: first the selection for inlet and outlet vane angles β_1, β_2 is to be done. The usual angles used for β_1, β_2 are 13° and 20° as recommended by Stepanoff *etal*. (2013). For smooth flow, the vane must be designed such that these angle increases smoothly from 13° to 20°. The graph of vane angle β 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 β. Then Plot the radii against position angle θ 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 θ can be calculated by using the relation $d\theta = dr/r\tan\beta$. To calculate the value of θ 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

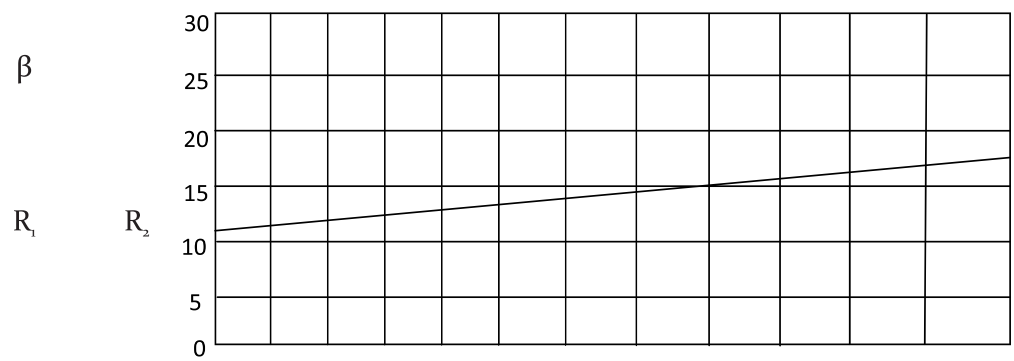

Figure 1.2: A Numerical Approach; A graph for Blade Angle "β"against Blade Radius "R"

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.

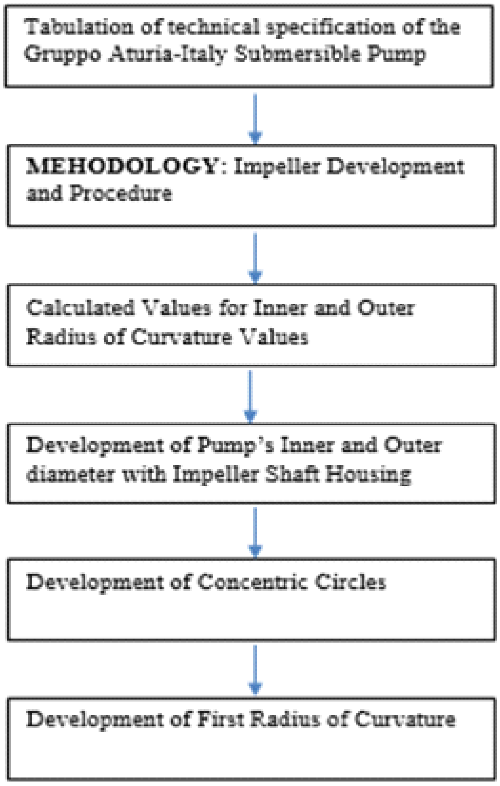

Figure 2.1: Flow chart for Impeller Development and Modeling

Impeller Development and Procedures Using Singularity Method for Impeller Development

Let β_1 and β_2 be the inlet and outlet angles of this impeller respectively. Selection of β_2 is made generally for an optimum efficiency. An average value of 22.5° is called normal for all specific speeds. The limit of β_2 in a good design is from 9.5° (minimum) to 39.5° (max). The values for β_1 and β^2 are selected as 10° and 40°. For smooth flow, the vane must be designed such that this angle increases smoothly from 10° to 40°. The next step is to construct the vaneshape. There are several methods to construct the vane shapes. The one used in practice consist of tangent circular arc. The radius of the circular arc contained between the rings D_1 and D_2 is given by.

$$R = \frac{R^2_2 - R^2_1}{2(R_2 COSB_2 - R_1 COSB_1)} \qquad (2)$$

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 165/2 = 82.5 mm and the radius of Outer diameter is given by 183.74/2 = 91.9 mm. Now, the intermediate values of radius can be found out by (R_2–R_1)/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 β can be found out graphically.

Table 2.1: Table for Calculated Values of Inner Radius "R_1" and Outer Radius "R_2" Of Radius of Curvature "r".

R_1	R_2	β_1	B_2	R
41.5	49.83	10	15	52.40
49.83	58.16	15	20	77.41
58.16	66.49	20	25	93.21
66.49	74.82	25	30	129.35
74.82	83.15	30	35	198.77
83.15	91.5	35	40	350.56

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.

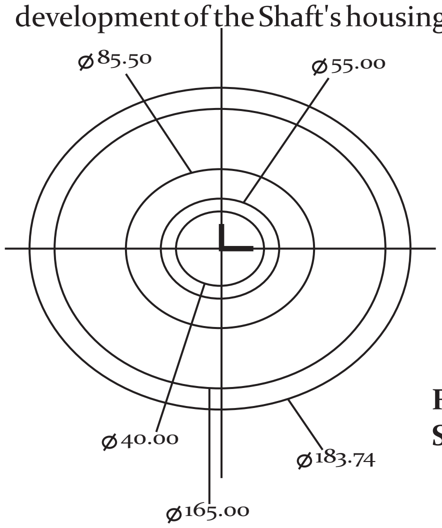

Figure 2.2: Development of Outer, Inner and Shaft Housing Diameters of the Impeller

Development of all the Concentric Circles Using a Simply Geometric Method

The second step is to develop all the concentric circles using radii R_1 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^o 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 $\beta_1=10^o$ as shown in Figures; 2.2.

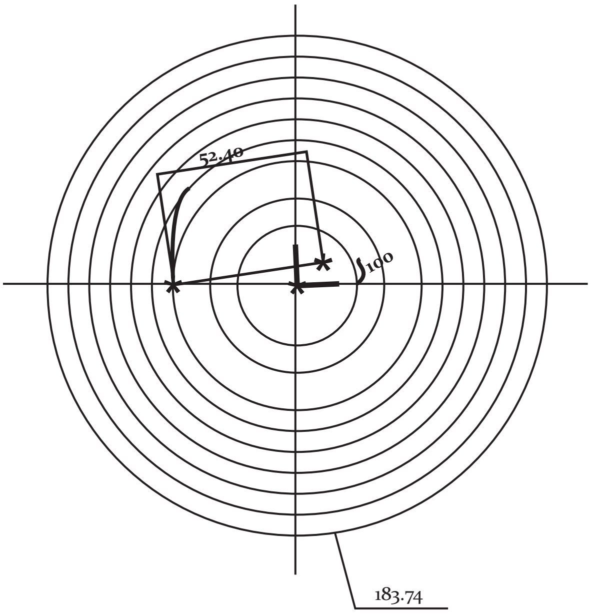

Figure 2.3: Developing All the Concentric Circles

Development of First Radius of Curvature "AE"

The third step is to develop the first arc AE having inlet vane angle $\beta_1=10^o$ 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.

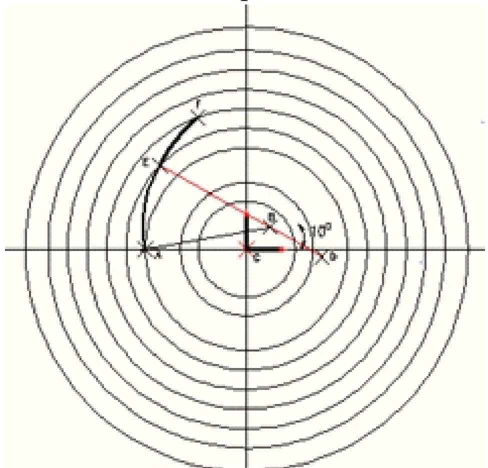

Figure 2.4: Development of First Radius of Curvature "AE"

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.

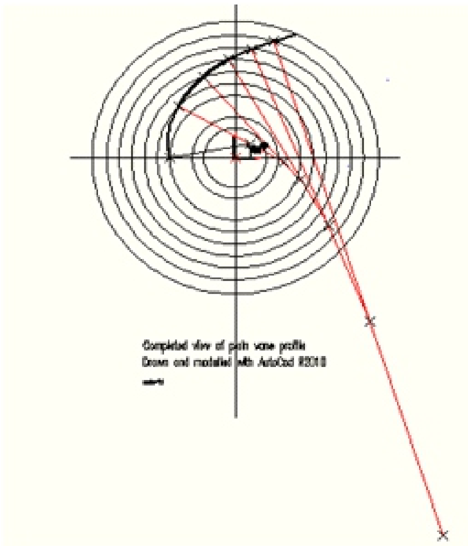

Figure 2.5: Development of the Second Radius of Curvature "EF"

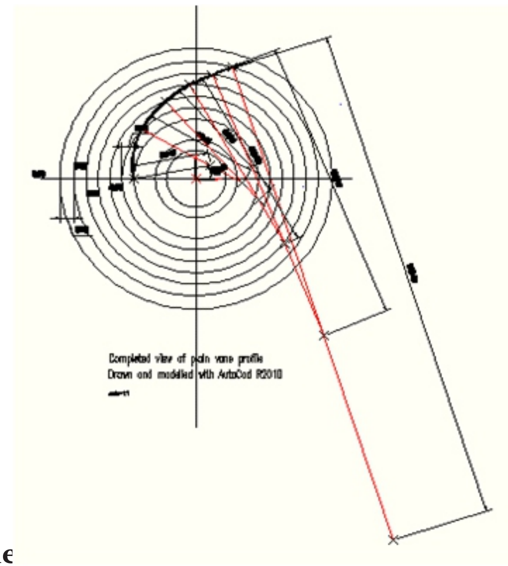

Figure 2.6: Comple

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

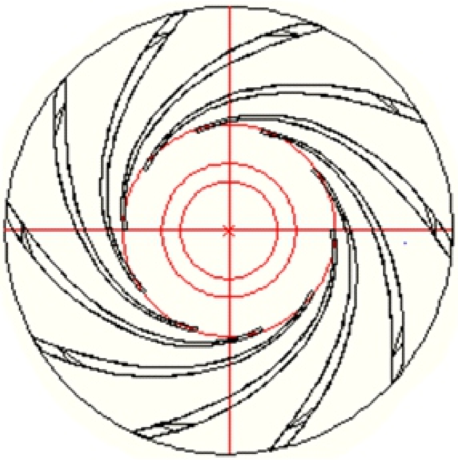

Figure 2.7: Development of Ten Blade 2D Model Impeller

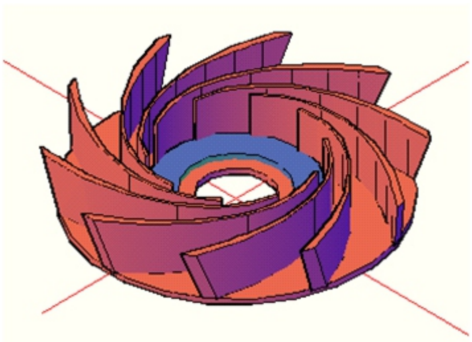

Figure 2.8: Development of Ten Blades 3D Model Impeller

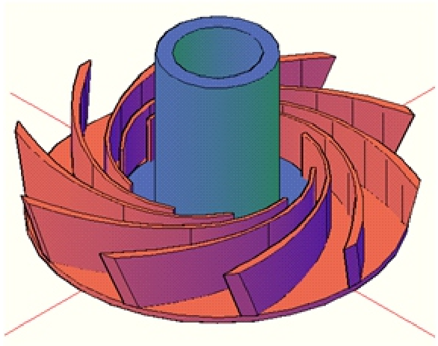

Figure 2.9: Addition of Shaft Sleeve into the vane profile

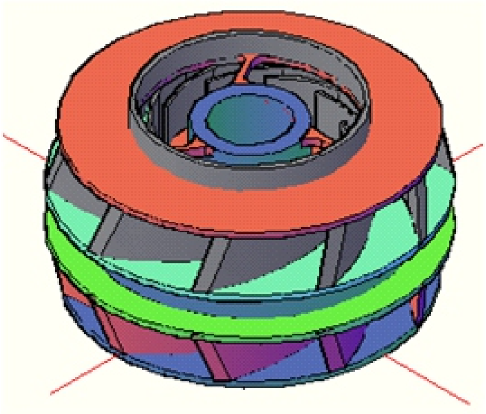

Figure 2.10: Modeled double-plain vane beveled impeller

Modeling of CAD 3D Model of the Pump Impeller
Finally, the sixth step is the assembling of various parts to get the final 3D model of impeller as shown in Figure 3.5.

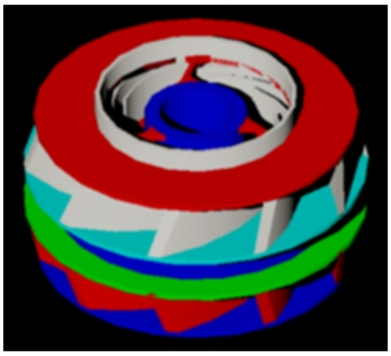

Figure 3.1: Rendered double-plain vane beveled impeller

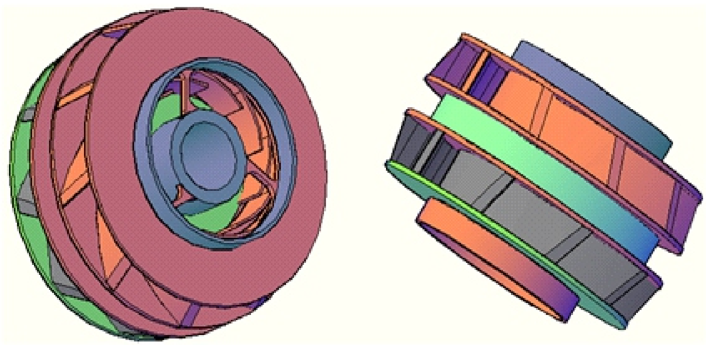

Figure 3.2: Rotated plain chamfered vane straight impeller

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
Ashutosh, K. (2008). *Numerical study of performance of centrifugal pump for handling ash slurry*. Dissertation submitted in partial fulfillment for M. Eng. Thapar University, Patiala, India.

Centrifugal pump impellers. Retrieved from http://en.wikipedia.org/wiki/centrifugal pump impellers on 16th June, 2014.

Diagrammatic representation of type of flows in an impeller (2014). Retrieved from Grundfos industry hand book on 19th June

Esfahani, J.A., Moghadam, B.J., Nouri, M. & Mahmoudi, A. (2014). *Numerical and parametric study of a centrifugal pump, effect of blade number and volute shape*. International conference on advances in mechanical and robotics engineering, ISBN: 978-1-63248-002-6.

Gonzale, J., Fernandez, J., Blanco, E. & Santolaria, C. (2006). Numerical simulation of the dynamic effects due to impeller volute interaction in a centrifugal pump, ASME *Journal of Fluids Engineering*, vol. 124.

Gundale, V.A & Joshi, G.R. (2013). A simplified 3d model approach in constructing the plain vane profile of a radial type submersible pump impeller" www.isca.in.

Gurupranesh, P., Radha, R.C. & Karthikeyan, N.(2012). CFD analysis of centrifugal pump impeller for performance enhancement. IOSR. *Journal of mechanical and civil engineering* (IOSR-JMCE), www.iosrjournals.org.

Houlin, L., Wang, Y., Shouqi, Y., Minggao, T., & Kai, W. (2010). Effects of blade number on characteristics of centrifugal pumps.*Chinese Journal of Mechanical Engineering. 23*.

Introduction to centrifugal pump. Retrieved from http://www.solvedrilling.com/news/brief-introduction-of-centrifugal-pump.shtml on 7th May, 2014.

Kaewnai, S., Chamaoot, M. & Wongwises, S. (2008). Predicting performance of radial flow type impeller of centrifugal pump using CFD. Journal of Mechanical Science and Technology.

Kergourlay, G., Younsi, M., Bakir, F. & Rey, R. (2007). Influence of splitter blades on the flow field of a centrifugal pump: Test-analysis comparison. International Journal of Rotating Machinery, vol. 2007.

Li, W.G, (2011). Effect of exit blade angle, viscosity and roughness in centrifugal pumps investigated by cfd computation. Department of Fluid Machinery, Lanzhou University of Technology, http://www.bop.com.pl.

Li, W.G. (2011). Inverse design of impeller blade of centrifugal pump with a singularity method. Jordan Journal of Mechanical and Industrial Engineering (JJMIE), 5(2).

Lobanoft, (2013). A simplifier 3d model approach in constructing the plain vane of a radial type submersible pump impeller, www.isca.in.

Pandey, K. M, Singh, A. P. & Chakraborty, S. (2012). Numerical studies on effects of blade number variations on performance of centrifugal pumps at 2500 rpm. Journal of environmental research and development, Vol. 6, No. 3A.

Patel, M.G. & Doshi, A.V. (2013). Effect of impeller blade exit angle on the performance of centrifugal pump. International Journal of Emerging Technology and Advanced Engineering, Website: www.ijetae.com Volume 3, Issue 1.

Shahu, S.R. (2013). CFD for centrifugal pump: A review for the state of art, Procedia Engineering, Vol. 51, pp 715 – 720.

Shojaeefard, M. H., Tahani, M., Ehghaghi, M. B., Fallahian, M. A. & Beglari, M, (2012). Numerical study of the effect of some geometric characteristic of a centrifugal pump impeller that pumps viscous fluid. In Proc. C & F' 60, p.61-70.

Stepanoff et al. (2013). A simplifier 3d model approach in constructing the plain vane of a radial type submersible pump impeller, www.isca.in

Tuzson, (2013). A simplifier 3d model approach in constructing the plain vane of a radial type submersible pump impeller, www.isca.in.

Velocity triangles at suction and discharge of the impeller (2014). Retrieved from Grundfos industry hand book on 19th June