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Design and Significant Effect of a Centrifugal Pump by using Various Blade Exist Angles

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Abstract: Some of the mechanical assisted devices include pneumatic pumps, axial flow pumps, and centrifugal pumps. Among the pumps available, the centrifugal pumps have attracted worldwide interest because centrifugal flow pumps have high efficiency. In this paper, centrifugal pump is analyzed by using a single-stage end suction centrifugal pump. Two main components of a centrifugal pump are the impeller and the casing. The impeller is a rotating component and the casing is a stationary component. In centrifugal pump, water enters axially through the impeller eyes and water exits radially. Centrifugal pumps are widely used in many applications such as water pumping project, domestic water raising, industrial waste water removal, raising water from tube wells to the fields. The application of designed pump is used the domestic water raising. Various parameters affect the pump performance and energy consumption. The impeller outlet diameter, the blade angle and the blade number are the most critical. The present paper describes significant effect of a centrifugal pump by varying outlet blade angle. In this study, the performance of impellers with the same outlet diameter having different outlet blade angles is thoroughly evaluated. In this paper, the pump is driven by one horse power electric motor and the design is based on Kyushu Method. The head and flow rate of this pump are 20 m and 0.004167m3/s and the motor speed is 2900 rpm. The number of impeller blade is 6 blades. The significant effect of centrifugal pump is carried out after designing the dimensions of centrifugal pump. So, shock losses, impeller friction losses, volute friction losses, disk friction losses and recirculation losses of centrifugal pump are also considered in performance analysis of centrifugal pump.

Keywords: Head, Flow rate, Losses, Impeller, Kyushu Method.

I. INTRODUCTION

A centrifugal pump is a rotodynamic pump that uses a rotating impeller to increase the pressure of a fluid. Centrifugal pumps are commonly used to move liquids through a piping system. The fluid enters the pump impeller along or near to the rotating axis and is accelerated by the impeller, flowing radially outward into a diffuser or volute chamber (casing), from where it exits into the downstream piping system. Like most pumps, a centrifugal pumps converts mechanical energy from a motor to energy of a moving fluid; some of the energy goes into kinetic energy of fluid motion, and some into potential energy, represented by a fluid pressure or by lifting the fluid against gravity to a higher level. The transfer of energy from the mechanical rotation of the impeller to the motion and pressure of the fluid is usually described in terms of centrifugal force, especially in older sources written before the modern concept of centrifugal force as a fictitious force in a rotating reference frame was well articulated.

The concept of centrifugal force is not actually required to describe the action of the centrifugal pump. In the modern centrifugal pump, most of the energy conversion is due to the outward force that curved impeller blades impart on the fluid. Invariably, some of the energy also pushes the fluid into a circular motion, and this circular motion can also convey some energy and increase the pressure at the outlet. The relationship between these mechanisms was described, with the typical mixed conception of centrifugal force as known as that time. Pumps are used in a wide range of industrial and residential applications. Pumping equipment is extremely diverse, varying in type, size, and materials of construction. There have been significant new developments in the area of pumping equipment. They are used as boiler feed pumps, hot well pumps, sewage and sump pumps, irrigation and drainage pumps, paper mills, deep well pumps and fire pumps. Centrifugal pumps leave a very small field for reciprocating pumps, a field where capacities are too low and pressures too high to permit a favorable type for a centrifugal pump.

However, this field is being gradually reduced further. Such progress in the development and application of centrifugal pumps is due to several factors.

- Their high adaptability for high speed electric motor and steam driver.
- Minimum of moving parts and,
- Small size and low cost for the amount of liquid moved;

The centrifugal pumps are available in sizes from 1/4 in to 100 in normal discharge. They may be either single or multi-stage. The single stage centrifugal pumps may be either single-suction or double-suction centrifugal pumps.Fig.1 shows component parts of a centrifugal pump [1].

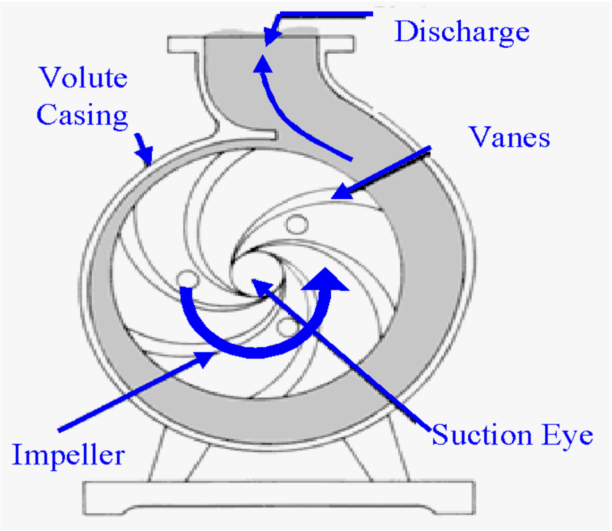

Fig.1. Liquid flow path inside a centrifugal pump.

II. DESIGN OF SIGNIFICANT EFFECT OF CENTRIFUGAL PUMP BY VARYING OUTLET BLADE ANGLE

A. Design of centrifugal pump

The design pump is a single stage centrifugal pump. Impeller is designed on the basic of design flow rate, pump head and pump specific speed. So, the design data are required to design the centrifugal pump .The design parameters are taken as follows:

Flow rate, $Q = 0.004167\ m^3/s$
Head, $H = 20$ m
Pump speed, $n = 2900$ rpm
Gravitational acceleration, $g = 9.81\ m/s^2$
Density of water, $\rho = 1000\ kg/m^3$

1. Specific Speed:

Selection of the specific speed value at the best efficiency point is the first step in the design of centrifugal pumps. This involves a selection of the relative speed and whether the required head should be produced in one or more stages. Another distinction in impellers is the liquid traverses and leaves the impeller blades. This is called the specific speed.

Specific speed;
$$n_s = \frac{n\sqrt{Q}}{H^{3/4}} \qquad \textbf{(1)}$$

The input power P is calculated by the following equation.

$$P = \frac{\rho g Q H}{\eta} \qquad \textbf{(2)}$$

From Figure pump efficiency versus discharge, the efficiency η is approximately taken 0.52 [5].

The volumetric efficiency is calculated by using the Equation.

$$\eta_v = \frac{1}{1 + 1.124 / n_s^{2/3}} \qquad \textbf{(3)}$$

The diameter shaft is obtained by using Equation.

$$d_s^3 = \frac{16T}{\pi \times S_s} \qquad \textbf{(4)}$$

For commercial steel shaft, permissible shear stress S_s is 24.5 MN/ m^2.

The torsion moment is estimated by:

$$T = \frac{P \times 60}{2 \times \pi \times N} \qquad \textbf{(5)}$$

The diameter of the impeller eye is

$$D_0 = K_0 \sqrt[3]{Q/n} \qquad \textbf{(6)}$$

The value such as K_{m1}, K_{m2}, K_u are obtained from the Stepanoff chart [5].

Vane inlet velocity,

$$V_{m1} = K_{m1}\sqrt{2gH} \qquad \textbf{(7)}$$

Vane outlet velocity,

$$V_{m2} = K_{m2}\sqrt{2gH} \qquad \textbf{(8)}$$

Impeller inlet diameter,

$$D_1 = (1.1 \sim 1.15) K_o \sqrt[3]{Q/n} \qquad \textbf{(9)}$$

The value of K0 is chosen as 4.5.

The outlet diameter of impeller is:

$$D_2 = 19.2 \left(\frac{n_{s\ opt;}}{100} \right)^{1/6} \frac{\sqrt{2gH}}{n} \qquad \textbf{(10)}$$

Inlet peripheral velocity, U_1 is calculated by using Equation.

$$U_1 = \frac{\pi \times D_1 \times n}{60} \qquad \textbf{(11)}$$

Outlet peripheral velocity, U_2 is calculated by using Equation.

$$U_2 = \frac{\pi \times D_2 \times n}{60} \qquad \textbf{(12)}$$

2. Vane Inlet and Outlet Angle:

The water is assumed to enter the vanes radially, so that the absolute velocity α_1 is 90˚. After V_{m1} and U_1 have been calculated, the vane inlet angle β_1 is obtained by the equation.

$$\beta_1 = \tan^{-1}\left(\frac{V_{m1}}{U_1} \right) + (0 \sim 6) \qquad \textbf{(13)}$$

In this design the vane outlet angle β_2, is assumed as 23˚ [5]. The number of blades is calculated by using the following Equation. The number of blades,

$$Z = 6.5 \times \frac{(D_2 + D_1)}{(D_2 - D_1)} \times \sin\left(\frac{\beta_1 + \beta_2}{2} \right) \qquad \textbf{(14)}$$

The inlet passage width b_1

$$b_1 = \left(\frac{Q_S'}{\pi D_1 V_{m1}} \right) \times \left(\frac{\pi D_1}{\pi D_1 - S_1 Z} \right) \qquad \textbf{(15)}$$

The outlet passage width b_2

$$b_1 = \left(\frac{Q_S'}{\pi D_2 V_{m2}} \right) \times \left(\frac{\pi D_2}{\pi D_2 - S_2 Z} \right) \qquad \textbf{(16)}$$

The minimum blades thickness is 2.0 mm and shroud thickness is 2.5 mm for an impeller having the diameter less than 200 mm. Flow through the impeller:

$$Q_s' = Q_s / \eta_v \qquad (17)$$

B. Required Parameters for Impeller Blade Shape

To draw the curvature of the blade curve equally spaced circles are drawn between impeller outside circle and impeller inside circle. Vane slope angles are also drawn. The angle between β_1 and β_2 are equally divided into three angles [5].

Impeller outside diameter, D_A= 140.83mm

Radius, $R_A = D_A / 2$ =70.415m

Impeller inside diameter, $D_D = D_{1h}$= 49.5 mm

$$\rho_A = \frac{(R_A^2 - R_B^2)}{2(R_A \cos\beta_2 - R_B \cos\beta_B)} \qquad (18)$$

$$\rho_A = \frac{(R_B^2 - R_C^2)}{2(R_B \cos\beta_B - R_C \cos\beta_C)} \qquad (19)$$

$$\rho_A = \frac{(R_C^2 - R_D^2)}{2(R_C \cos\beta_C - R_D \cos\beta_1)} \qquad (20)$$

Base circle radii and blade curved angles for impeller are as shown in Table1. Then the blade curve radii are calculated by using this data.Fig.2 illustrates drawing of impeller blade shape.

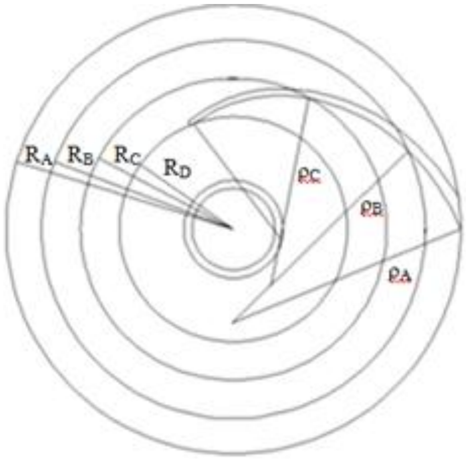

Fig.2. Drawing of Impeller Blade Shape.

TABLE I: Base Circle Radii and Blade Curved Angles

Base Circle Radii	mm	Angle	Degree
R_A	70.415	β_2	23
R_B	55.19	β_B	22
R_C	39.97	β_C	21
R_D	24.75	β_1	20

C. Design Calculation of Volute Casing

Flow rate, Q = 0.004167 m^3/s
Pump head, H = 20 m
Pump speed, n = 2900 rpm
Impeller diameter at outlet, D_2 = 140.83 mm
Impeller outlet width, b_2 = 6 mm
Shroud thickness = 2.5 mm

Average flow Velocity,

$$V_v = K_v \sqrt{2gH} \qquad (21)$$

The volute area at the throat,

$$A_v = Q / V_v \qquad (22)$$

Other volute sections,

$$A_{vi} = A_{vi} \times i/8 \qquad (23)$$

Where, the value of i is from 1 to 8 representing the volute sections. Volute base circle diameter D_3 is used in laying out the volute casing. It is first drawn and then other drawings of volute sections can be done. So, it is one requirement of volute casing design. The value of $\frac{D_3 - D_2}{D_2} \times 100$ is obtained from volute constant chart. Volute base circle diameter D_3 is calculated by the following relation.

$$\frac{D_3 - D_2}{D_2} \times 100 = 9 \qquad (24)$$

Volute width is estimated by Equation

$$b_v = b_2 + 2 \times \text{shroud thickness} + 2 \times \text{clearance} \qquad (25)$$

Requirements for Laying Out Volute Casing:

Other requirements are the values of ρ_{vi} and r_{vi} for laying out the cross sectional shapes of the volute sections.Table.3 shows values of volute areas and parameter for laying out cross section shape of volute sections.

$$\rho_{vi} = \sqrt{\frac{A_{vi} + 0.604 b_v^2}{0.367}} \qquad (26)$$

For i= 1, i.e., for drawing the cross sectional shape of the first volute cross section.

The relationship between ρ_{vi} and r_{vi}

$$r_{vi} = 0.206 \rho_{vi} \qquad (27)$$

The result data of the single-suction centrifugal pump are listed in Table 2.

TABLE II: Values of Voltage Areas and Parameter for Laying Out Cross Section Shape of Volute Sections

Section (i)	A_{vi} (mm^2)	P_{vi}(mm)	R_{vi}(mm)
1	60.00	21.01	4.33
2	119.47	24.57	5.06
3	179.20	27.68	5.70
4	238.94	30.48	6.28
5	298.67	33.04	6.81
6	358.40	35.42	7.30
7	418.14	37.65	7.76
8	477.87	39.75	8.19

III. SIGNIFICANT EFFECT OF CENTRIFUGAL PUMP BY VARYING OUTLET BLADE ANGLE

A calculation to estimate the theoretical performance of a pump is an indispensable tool in pump design. Performance needs to be known, not only at the rated, best efficiency point, but also off design. Pump specifications often impose special requirements, such as head at shutoff, maximum power demand, rate of head rise to assure stability, and so on. A good pump design process requires trail-and-error iteration, a check on anticipated performance with a trial geometry, and progressive approximation to the optimal design configuration. The best hydraulic design does not necessarily correspond to the best commercial pump product. Generally, the more accurate and detailed these calculations are, the greater the number of input variables needed: not only the desired head, flow rate, and rotational speed, but also the details of the geometrical description of the impeller and housing. When the characteristic cure is approached, pump's rotational speed, n is chosen 2900 rpm. The impeller inlet diameter, D_1 is 55.87mm, the outlet diameter, D_2 is 140.83mm, the impeller inlet width, b_1 is 11mm and the impeller outlet width, b_2 is 6mm. Inlet vane angle and out vane angle are 20° and 23° respectively. Number of vanes is 6 and the largest volute diameter, D_v is 153.50mm.

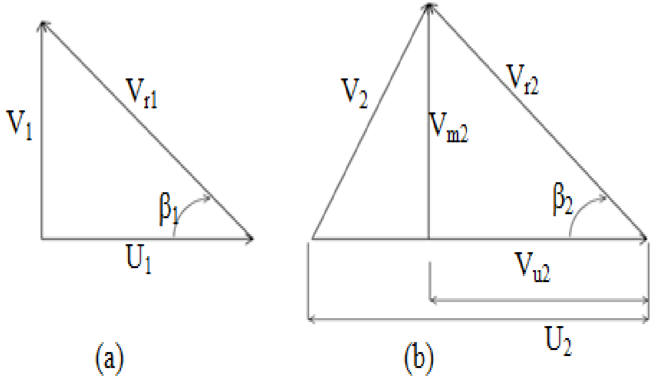

Fig.3. Velocity Diagrams (a) Inlet Velocity Diagram (b) Outlet Velocity Diagram.

A. Theoretical Head

The Euler head is determined from zero to maximum theoretically attainable flow using.

The theoretical head,

$$H_{th} = \frac{1}{g} U_2 V_{u2} \tag{28}$$

The whirl velocity,

$$V_{u2} = U_2 - V_{m2} \cot\beta_2 \tag{29}$$

Where V_{m2} and β_2 are outlet flow (fig.3) velocity and outlet blade angle.

B .Net Theoretical Head

If the slip factor is known, the net theoretical head may be obtained from Euler's head. It is possible to relate the theoretical characteristic obtained from Euler's equation to the actual characteristic for various losses responsible for the difference. The use of the slip factor which varies with flow rate enables the net theoretical head curve to obtain. This represents the net head developed by the impeller but does not account for losses. At flow rates below design flow rate, separation occurs on the suction side of the blade [1].

The net theoretical head is calculated by:

$$H_{thn} = \frac{U_2 V_{u2}}{g} \tag{30}$$

The whirl velocity at the outlet velocity is;

$$V_{u2} = U_2 \sigma - V_{m2} \cot\beta_2 \tag{31}$$

So, slip value is obtained by using the following equation.

$$\sigma = 1 - \frac{(\sin\beta_2)^{1/2}}{Z^{0.7}} \tag{32}$$

C. Shock Losses

The major loss considered is shock losses at the impeller inlet caused by the mismatch of fluid and metal angles. Shock losses can be found everywhere in the flow range of the pump. Shock Losses are given by following equation:

$$hs = k(Qs - QN)^2 \tag{33}$$

Maximum flow rate:

$$QN = \pi D_1 b_1 V_{m1} \tag{34}$$

The shut off head:

$$H_{shut\text{-}off} = \frac{U_2^2 - U_1^2}{2g} \tag{35}$$

In the shut –off condition, $Q_s=0$ and $hs = H_{shut\text{-}off}$. So, shock losses equation is formed by substituting in equation 26.

D. Impeller Friction Losses

The impeller was designed that the width of the impeller would become small and the friction loss at the flow passage would become large. Therefore to relive the increase in friction loss, radial flow passage on the plane of the impeller was adopted. The friction losses can be found for energy dissipation due to contact of the fluid with solid boundaries such as stationary vanes, impeller, casing, disk and diffuser, etc[1].

The impeller friction losses are:

$$h_1 = \frac{b_2 (D_2 - D_1)(V_{r1} + V_{r2})^2}{2\sin\beta_2 H_r 4g} \tag{36}$$

The hydraulic radius is calculated by using Equation 22.

$$H_r = \frac{b_2 \left(\frac{\pi D_2}{Z}\right) \sin\beta_2}{b_2 + \left(\frac{\pi D_2}{Z}\right) \sin\beta_2} \tag{37}$$

The inlet relative velocity,

$$V_{r1} = \frac{V_{m1}}{\sin\beta_1} \tag{38}$$

The outlet relative velocity,

$$V_{r2} = \frac{V_{m2}}{\sin\beta_2} \tag{39}$$

E. Volute Friction Losses

The volute friction losses: $h_2 = \frac{C_v V_3^2}{2g}$ **(40)**

The volute throat velocity: $V_3 = \frac{Q}{A_3}$ **(41)**

The volute throat area: $A_3 = V_{u2}(\frac{D_2}{D_3})$ **(42)**

The volute flow coefficient is obtained Equation (28).

$$C_v = 1 + (0.02 \times \frac{L_{vm}}{D_{vm}}) \quad \textbf{(43)}$$

The volute circumferential length:

$$L_{vm} = \frac{\pi D_i}{8} \quad \textbf{(44)}$$

The volute mean diameter:

$$D_{vm} = \frac{D_i}{8} \quad \textbf{(45)}$$

F. Disk Friction Losses

The disk friction power is divided by the flow rate and head to be added to the theoretical head when the shaft power demand is calculated [1].

The disk friction loss,

$$h_3 = \frac{f\,\rho\,\omega^3 (\frac{D}{2})^5}{10^9 Q_s} \quad \textbf{(46)}$$

The angular velocity ω, $\omega = \frac{2\,\pi\,n}{60}$ **(47)**

Loss coefficient of disk friction, f is assumed a 0.005.

G. Recirculation Losses

The recirculation loss coefficient depends on the piping configuration upstream of the pump in addition to the geometrical details of the inlet. The power of recirculation is also divided by the volume flow rate, like the disk friction power, in order to be converted into a parasitic head.

The recirculation loss,

$$h_4 = K\ \omega^3 D_1^{\ 2} (1 - \frac{Q_s}{Q_0})^{2.5} \quad \textbf{(48)}$$

The pump speed is carried out with the value of specific speed because impellers with relatively large inlet diameters (usually encountered in high-specific-speed pumps) are the most likely to recirculation. Coefficient of leakage loss K is assumed as 0.005 [1].

H. Actual Head

The output of a pump running at a given speed is the flow rate delivered by it and the head developed. Thus a plot of head against flow rate at constant speed forms the fundamental performance characteristic of a pump. In order to achieve this actual head, the flow rate is required which involves efficiency of energy transfer. The actual pump head is calculated by subtracting from the net theoretical head all the flow losses which gives the actual head/flow rate characteristic provided it is plotted against (fig.4). Therefore, the actual pump head,

$$H_{act} = H_{thn} - (h_s + h_1 + h_2 + h_3 + h_4) \quad \textbf{(49)}$$

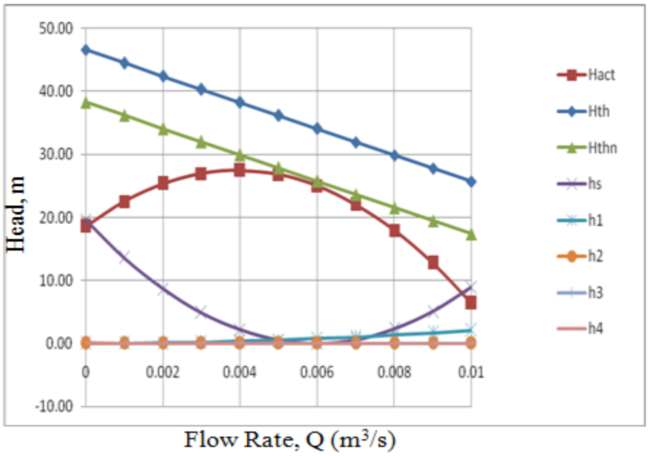

Fig.4. Prediction of Characteristic Curve of Centrifugal Pump.

I. Effect of Blade Exit Angle on Head and Efficiency

Changing some geometric characteristic of the impeller in centrifugal pumps improves their performance. It is known that blade exit angle plays very important role in the performance of a centrifugal pump. To investigate effect of blade exit angle on the performance of centrifugal pump by means of experiment is very expensive and lengthy process. In the present study three pumps of different specific speeds are taken for the investigation. In recent years, centrifugal pump have been increasingly utilized for various purposes, such as irrigation, water supply, steam power plants, oil refineries, air conditioning systems. Due to the vast application it is very important that centrifugal pump should work efficiently. There have been continuous efforts to improve the performance of centrifugal pumps. Therefore, the blade exit angle has significant effect on the head and efficiency of the centrifugal pump. It is found from this investigation, both head and efficiency of centrifugal pump increases with increasing in blade exit angle (fig.5).

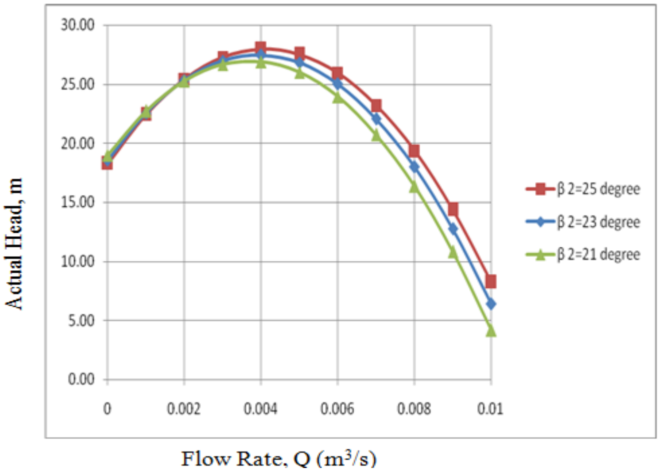

Fig.5. Head-flow characteristic graph at various blade exit angles.

J. Efficiencies

How small the losses or how good a machine in converting energy is indicated by its efficiency. The efficiency of a machine is always defined as the ratio of the power output of the machine to the power input into it. A centrifugal pump has four types of efficiencies. These are mechanical, volumetric, hydraulic and overall efficiencies.

K. Overall Efficiency

Overall efficiency is the ratio of the water power to the power supplied to the pump shaft at the coupling [5].

$$\eta_o = \frac{\rho g Q H}{P} \tag{50}$$

(or)

$$\eta_o = \eta_m \times \eta_v \times \eta_h \tag{51}$$

Where, η_o = Overall efficiency
η_h = Hydraulic efficiency
η_v = Volumetric efficiency
ρ = Density of the fluid being pumped
g = Gravitational acceleration (9.81 ms^{-2})
Q = Fluid flow rate through the pump
P = Shaft power

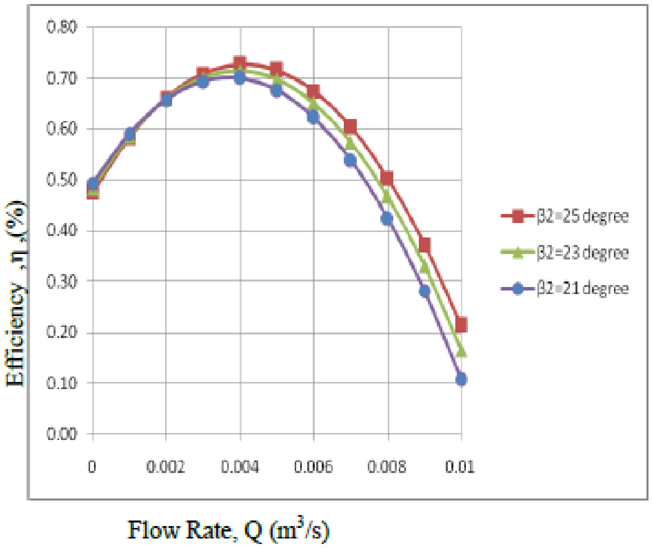

Figure6. Efficiency-flow characteristic graph at various blade exit.

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V. NOMENCLATURES

A_{vi} = The volute area at the throat (mm^2)
b_1 = Width of impeller at inlet (mm)
b_2 = Width of impeller at outlet (mm)
b_v = Volute width (mm)
D_1 = Impeller diameter at inlet (mm)
D_2 = Impeller diameter at outlet (mm)
D_{1s} = Shaft diameter at inlet (mm)
D_o = Eye diameter of impeller (mm)
D_{bt} = Hub diameter (mm)
d_s = Shaft diameter at hub section (mm)
g = Gravitational acceleration (m/s^2)
H = Head of the pump (mm)
n = Pump speed (rpm)
n_s = Specific speed
P = Input Power
Q = Flow rate of the pump (m^3/min)
Qs = Flow through impeller
T = Torsional moment (N-m)
U_1 = Inlet tangential velocity (m/s)
U_2 = Outlet tangential velocity (m/s)
Vm_1 = Vane Inlet Velocity (m/s)
Vm_2 = Vane Outlet Velocity (m/s)
Z = The number of blade
P = Density of water (kg/m^3)
β_1 = Impeller Inlet Vane Angle (degree)
β_2 = Impeller Outlet Vane Angle (degree)

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