Design of Single Suction Centrifugal Pump and Performance Analysis by Varying the Speed of Impeller

Cho Cho Khaing*1, Nyi Nyi2**

*1 Department of Mechanical Engineering, Technological University (Mandalay),Myanmar
**2 Mechanical Precision Engineering Department, Technological University (Cyber City),Myanmar

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Abstract- This paper presents the design of impeller and casing of single-suction centrifugal pump for clean and cold water. Type of pump is single stage centrifugal pump with closed impeller and it can develop a head of 20 m and deliver 0.015 m^3/s of water. The designed impeller has 99 mm inlet diameter, 250 mm outlet diameter, 20° inlet vane angle and 23° outlet vane angle. The number of vanes is 6 and input shaft power is 6 hp. The inlet width and outlet width are 20 mm and 12 mm respectively. The discharge diameter is 80 mm to operate the designed head and capacity. The performance analysis of the designed pump is also presented on various speed. The predicted maximum efficiency is nearly 65% and the expressed actual efficiency of designed pump is 61%. Therefore, the designed efficiency has a satisfactory value. The designed single-suction centrifugal pump can fulfill the requirements of water pumping system for irrigation, and domestic usage in multistory building.

Index Terms- head, flow rate, speed, performance characteristics**.**

I. Introduction

A pump is a device which lifts water from a lower level to a higher level at the expense of mechanical energy. It consists an impeller rotating within a volute casing. In rotodynamics pump, the energy is transferred by rotary motion and by dynamic action. The input power of the pump is mechanical energy of the drive shaft and the output power is hydraulic energy. The rotating blade system imparts a force on the fluid, thereby making the fluid to move. Pumps are widely used for irrigation and are most common where pumping from surface water supplies such as river, lakes and streams and rising water to a higher level. Moreover, they are widely used in many other industries such as chemical plants, steam power plants, food processing factories, hydraulic system, and so on.

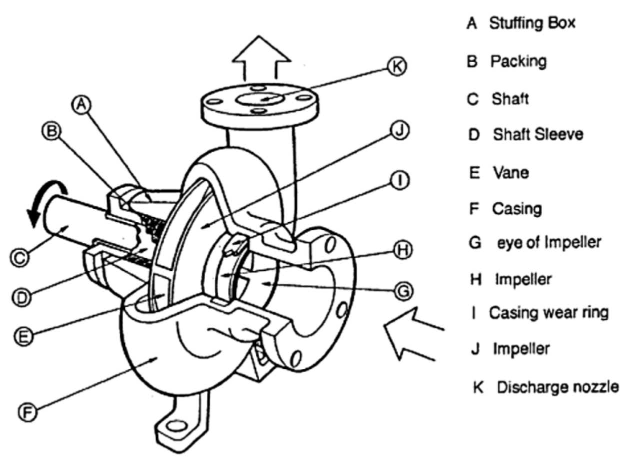

Fig 1.Single Suction Centrifugal Pump

II. Design Of Centrifugal Pump

The two main components of centrifugal pump are impeller and casing. The impeller is enclosed in a water tight casing that the kinetic energy of water is converted into pressure energy before the water leaves the casing. The other components are suction pipe, discharge pipe, shaft, bearing, wear rings, stuffing box, mechanical seal and various types of valves and gauges.

When the overall design of pump is considered, the shape of an impeller is the most important for optimum efficiency. Impeller design should be in such a way that, losses must be as low as possible. The design of a pump's impeller can be divided into two parts. The first is the selection of proper velocities and vane angles needed to obtain the desired performance with the best possible efficiency. The second is the layout of the impeller for the selected angles and areas.

The specifications of pump that will be designed are:

Pump head, $H = 20$ m
Discharge, $Q = 0.9$ m^3/min
$Q_s = (Q/60)$ m^3/s $= 0.015$m^3/s
Rotational Speed, $n = 1800$ rpm
Density of water, $\rho = 1000$ kg/m^3

A. Design of impeller

Specific speed is an essential criterion to determine the impeller shapes. It is mathematically expressed as

$$n_s = \frac{n \times \sqrt{Q}}{H^{3/4}} \tag{1}$$

In this design, calculated value of specific speed based on required head and capacity is 180 rpm and it is within the range of low specific speed pump that is greater than 80 and less than 600. So, end-suction type single stage centrifugal pump with closed impeller is chosen.

Pump efficiency, η is assumed by using Fig. A1. and also the diameter of suction pipe D_s can be estimated from this chart. The discharge pipe diameter D_d is usually selected equal to or one size smaller than that of the suction pipe. Thus, velocities in these pipes are given by

$$V_s = \frac{Q_s}{\pi \frac{D_s^2}{4}}, V_d = \frac{Q_s}{\pi \frac{D_d^2}{4}} \tag{2}$$

Input power of centrifugal pump can be determined by following equation.

$$L = \frac{\rho Q_s g H}{\eta} \tag{3}$$

For charge condition of the pump work, maximum shaft power or rated output of an electric motor L_r (kW) is decided by using Equation (4).

$$L_r = \frac{(1+F_a) \times L}{\eta_{tr} \times 1000} \tag{4}$$

Where, F_a is the allowance factor, and 0.1~ 0.4 for an electric motor and larger than 0.2 for engines And then, η_{tr} is the transmission efficiency, and 1.0 for direct coupling and 0.9 ~ 0.95 for belt drive.

The shaft diameter at hub section of impeller is

$$d_s = \sqrt[3]{\frac{16\,T}{\pi\,\tau}} \tag{5}$$

Where, T is the torsional moment and it can be estimated by

$$T = \frac{60\,L_r}{2\,\pi\,n} \tag{6}$$

Allowable shear stress of material of shaft, τ is 24.5 MPa because the main shaft is made of S30C. The estimated shaft diameter will be increased because it is difficult to predict the bending moment at this time.

The hub diameter, D_h is usually taken from 1.5 to 2.0 times of the shaft diameter and the hub length, L_h is from 1.0 times to 2.0 times of the shaft diameter.

The diameter of impeller eye, D_o is calculated by

$$D_o = \sqrt{\frac{4Q_s'}{\pi V_{mo}} + D_h^2} \tag{7}$$

Where, the flow rate through the impeller, Q_s' is Q/η_v and volumetric efficiency η_v is estimated by

$$\eta_v = \frac{1}{1+\dfrac{1.124}{n_s^{2/3}}} \tag{8}$$

For Equation (7), the velocity at the eye section is given by

$$V_{mo} = K_{mo}\sqrt{2gH} = (1.5 \sim 3.0) \leq V_{m1} \tag{9}$$

$$K_{mo} = (0.07 \sim 0.11) + 0.00023\, n_s \tag{10}$$

For a fluid flowing through the rotating impeller, u is the tangential velocity, V is the absolute velocity and v is the relative velocity of a fluid particle to impeller rotation. The angle between V and u is α and the angle between v and u is β and it is the angle made by tangent to the impeller vane and a line in the direction of motion of the vane. The tangential component and radial component of absolute velocity V are V_u and V_r respectively. The outlet velocities triangle with solid lines represents the actual diagram.

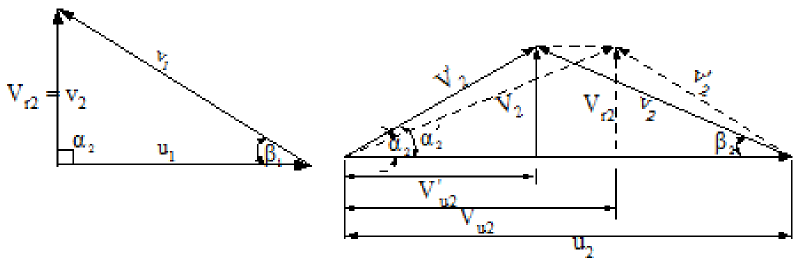

Fig. 2 Impeller Inlet and Outlet Velocity Diagrams

The parameters K_u (speed constant), K_{m1}, K_{m2}, and D_1/D_2 are obtained on the value of specific speed in Fig A2.
The outlet diameter D_2,

$$D_2 = \frac{u_2 \times 60}{\pi \times n} \tag{11}$$

Where, the peripheral velocity at impeller outlet is

$$u_2 = K_u\sqrt{2gH} \tag{12}$$

The peripheral velocity at the inlet is also expressed by

$$u_1 = \frac{\pi D_1 n}{60} \tag{13}$$

O And then, flow velocities at the inlet and outlet are

$$V_{r1} = K_{m1}\sqrt{2gH} \quad \text{and} \quad V_{r2} = K_{m2}\sqrt{2gH} \tag{14}$$

If the incoming flow has no pre-rotation, the blade angle β_1 (deg) is given by

$$\beta_1 = \tan^{-1}\left[\frac{K_{b1}V_{r1}}{u_1}\right] \approx \tan^{-1}\left[\frac{V_{r1}}{u_1}\right] + (0 \sim 6) \tag{15}$$

Where, K_{b1} = 1.1~1.25

The amount of outlet angle β_2 usually has between 15˚ and 35˚. So, the vane outlet angle is assumed that β_2 = 23˚ in this design. From the velocity triangles, inlet and outlet relative velocities are

$$v_1 = \frac{u_1}{\cos\beta_1} \quad \text{and} \quad v_2 = \frac{V_{r2}}{\sin\beta_2} \tag{16}$$

The virtual tangential component V_{u2} of V_2 is

$$V_{u2} = u_2 - \frac{V_{r2}}{\tan\beta_2} \tag{17}$$

For radial-type impellers, the slip factor, η_∞ varies between 0.65 and 0.75and it is assumed that η_∞ = 0.7 average. Thus, the actual tangential component V'_{u2} of V'_2 is

$$V'_{u2}=V_{u2}\eta_{\infty} \tag{18}$$

Thus, the actual outlet is found by

$$\tan\ \alpha'_2 = \frac{V_{r2}}{V'_{u2}} \tag{19}$$

The absolute outlet velocity from outlet velocity diagram is

$$V'_2=\sqrt{V_{r2}^2+V'^2_{u2}} \tag{20}$$

The number of blades, *Z* is decided by using the Plfeiderer formula.

$$Z \approx 6.5\frac{D_2+D_1}{D_2-D_1}\sin[\frac{\beta_1+\beta_2}{2}] \tag{21}$$

In this design, blade thickness and shroud thickness are taken as 2.5 mm and 3.0 mm respectively for $D_2$is greater than 200 mm.

The inlet passage width b_1 and outlet passage width b_2 are calculated by

$$b_1 = [\frac{Q'_s}{\pi D_1 V_{r1}}][\frac{\pi D_1}{\pi D_1\text{-}S_1 Z}] \quad \text{and} \qquad b_2 = [\frac{Q'_s}{\pi D_2 V_{r2}}][\frac{\pi D_2}{\pi D_2\text{-}S_2 Z}] \tag{22}$$

Where, S_1 is (δ_1/sin β_1), S_2 is (δ_2/sin β_2), and δ_1 and δ_2 are blade thicknesses near the leading edge and trailing edge respectively. Moreover, S_2 can also be determined by the following relationship equation.

$$\frac{\pi\ D_1}{(\pi\ D_1\text{-}\ S_1 Z)}=\frac{\pi\ D_2}{(\pi\ D_2\text{-}\ S_2 Z)} \tag{23}$$

The impeller blade is drawn by three circular arcs method with solid work software.

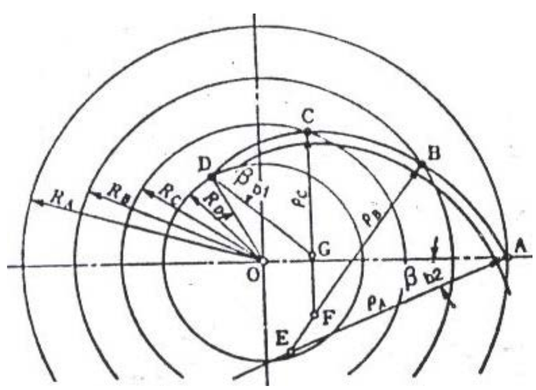

Fig. 3 Curvature of Impeller Blade

$$\rho_A = \frac{\left(R_A^2\text{-}R_B^2\right)}{2\left(R_A\cos\beta_2\text{-}R_B\cos\beta_B\right)}, \qquad \rho_B = \frac{\left(R_B^2 - R_C^2\right)}{2\left(R_B\cos\beta_B - R_C\cos\beta_C\right)} \quad \text{and} \qquad \rho_C=\frac{\left(R_C^2 - R_D^2\right)}{2\left(R_C\cos\beta_C - R_D\cos\beta_1\right)} \tag{24}$$

Where, R_A, R_B, R_C and R_D are base circle radii, R_A=D_2/2 and R_D = D_{1h}/2.

$$R_B = R_A - \frac{R_A - R_D}{3} \quad \text{and} \quad R_C = R_B - \frac{R_A - R_D}{3} \tag{25}$$

The angles between β_1 and β_2 are divided into three angles.

B. *Design of volute casing*

Design of volute casing is calculated depending on the D_2 and the basis of constant average flow velocity in volute casing. The volute casing increases proportionally in size from cut water to the discharge nozzle. In rear velocities distribution, across volute section is not uniform. Volute angle is read from volute constant chart shown in Fig. A3 and in this design, the volute angle, α_v is 8° based on n_s value.

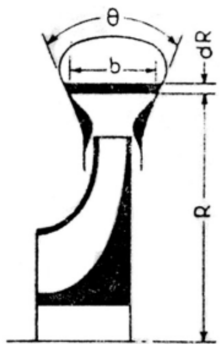

Fig. 4 Section through Volute Casing [4]

The width of the volute at any point may be calculated from

$$b = b_3 + 2x \times \tan(\theta/2) \tag{26}$$

Where, x is the distance between any radius R and impeller outside radius R_2. The volute is designed by determining the angle Φ° measured from and assumed radial line by tabular integration of Equation (27).

$$\Phi^\circ = \frac{360\, R_2 V'_{u2}}{Q} \int_{R_2}^{R_\Phi} \frac{bdR}{R} = \frac{360 R_2 V'_{u2}}{Q} \sum_{R_2}^{R_\Phi} b \frac{\Delta R}{R} \tag{27}$$

The tongue angle of volute casing is determined by

$$\Phi_t^\circ = \frac{132 \ \log_{10} \ R_t / R_2}{\tan \ \alpha'_2} \tag{28}$$

Volute wall thickness is chosen according to suction pipe diameter and it is taken as 6 mm since the suction pipe diameter is within 100 and 150 mm in this design.

III. Designed Results Of Centrifugal Pump

A. *Calculated Results*

The calculated results for both impeller and casing design of centrifugal pump are clearly expressed in Table I.

Table I

Calculated Results of single-suction Centrifugal Pump Design

No	Descriptions	Symbols	Results
1	Input Power	L	6 hp
2	Shaft diameter	d_s	34 mm
3	Hub diameter	D_h	51 mm
4	Hub length	L_h	68 mm
5	Impeller eye diameter	D_o	97 mm
6	Impeller inlet diameter	D_1	99 mm
7	Impeller outlet diameter	D_2	250 mm
8	Inlet angle of impeller blade	β_1	20°

9	Outlet angle of impeller blade	β_2	23°
10	Impeller passage width at inlet	b_1	20 mm
11	Impeller passage width at outlet	b_2	12 mm
12	Number of impeller blades	Z	6 blades
13	Base width of volute casing at D_2	b_3	24 mm
14	Volute tongue angle	$\Phi°_t$	15.51°
15	Discharge nozzle diameter	D_d	330 mm

B. *Modelling of Centrifugal Pump*

The three dimensional centrifugal pump is created by using Solidworks Software.

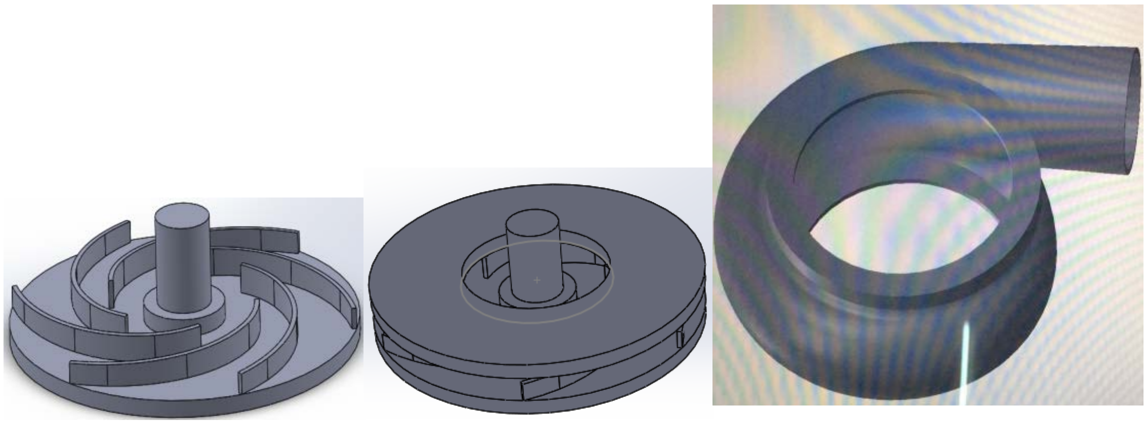

Fig.5 Three Dimensional View of Impeller and Casing for Designed Pump

IV. Performance Analysis By Varying Speed Of Impeller

Performance Characteristic curves of actual head and efficiency on capacity are presented by varying the speed of impeller in Fig 6. The actual head is achieved by subtracting of shock losses, diffusion losses, friction losses, circulatory flow effect, leakage losses and mechanical losses from the theoretical head.

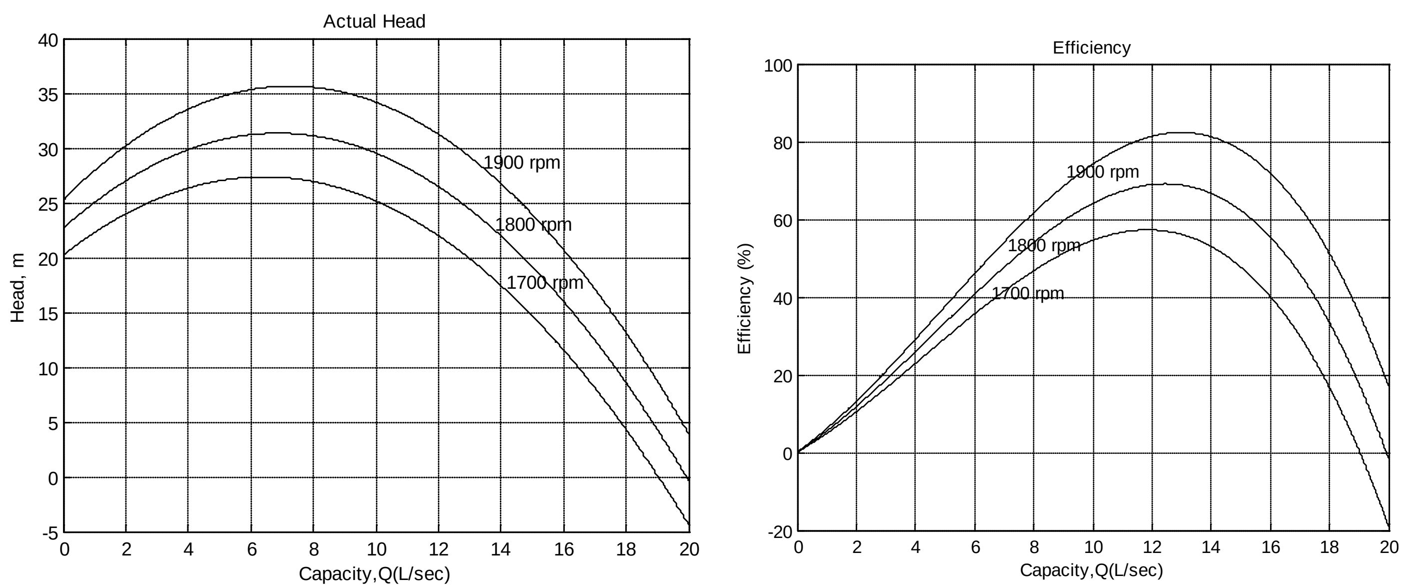

Fig. 6 Performance Characteristic Curves of Actual Head and Efficiency on Capacity for Various Speeds

V. Conclusion

The designed pump is aimed to use in agricultural application especially for river pumping project.. The clearance between impeller and tongue of volute is 3 mm. This value is a reasonably safe value for the tongue. The diameter of discharge flange is 80 mm. The thickness of volute casing to withstand the discharge pressure, 6 mm is selected depending upon the suction pipe diameter. When the performance of the designed pump is predicted, the maximum efficiency has nearly 65%. According to Fig 6, the designed centrifugal pump satisfies for head of 20 m and capacities of 15 L/sec at speed of 1800 rpm. At the maximum efficiency condition, we observed that the head is 25 m although the capacity is 13.5 L/s. The materials to be used should be selected depending upon the type of water. The impeller is made of bronze to protect corrosion. To reduce the leakage from discharge to suction between the casing and impeller, the clearance must be made very small. It is used only to pump water at 70˚ F and if very hot water is used this pump will be damaged. The designed single-suction centrifugal pump can fulfill the requirements of domestic application and industrial application, and then can improve pump efficiency.

APPENDIX

A. *FIGURES*

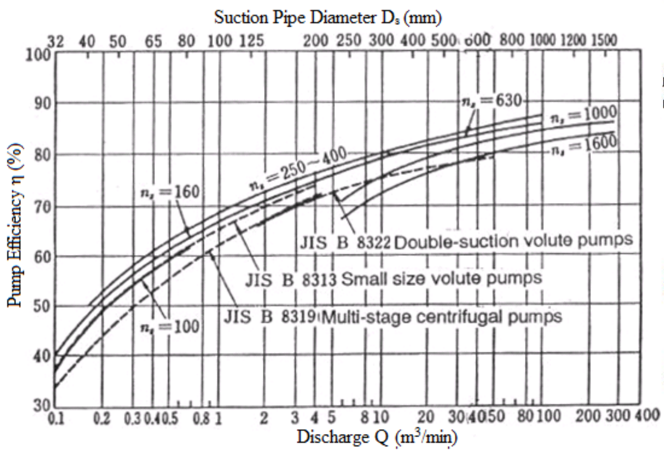

Fig A1 Overall Efficiency Curve [2]

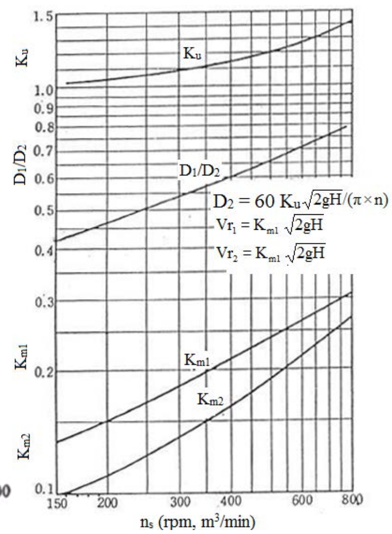

Fig A2 Stepanoff Chart

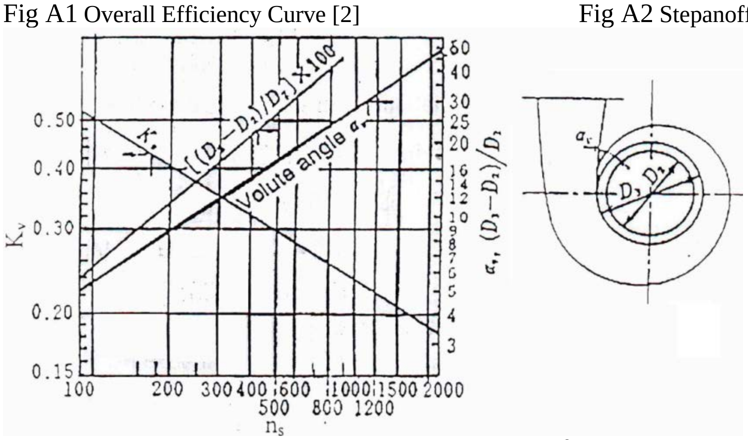

Fig. A3 Design Parameters for Volute Casing [3]

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AUTHORS

First Author – Cho Cho Khaing, M.E (Mech),Technological University(Mandalay), khaingcho999@gmail.com
Second Author – Dr. Nyi Nyi, Ph.D (Mech), Technological University(Cyber City), yaytakon@gmail.com