

Design Calculation and Performance Analysis of Single Suction Centrifugal Pump

Aye Thida San*, Min Min Oo**, Chaw Wint Yee Zaw**

*Department of Mechanical Engineering, Technological University (Patheingyi), Myanmar

**Department of Mechanical Engineering, Technological University (Patheingyi), Myanmar

**Department of Mechanical Engineering, Technological University (Thanyin), Myanmar

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Abstract- Centrifugal pumps are used widely for hydraulic transportation of liquids through pipelines where the requirement of head and discharge are moderate. This paper presents the design of impeller and casing of single suction centrifugal pump and performance analysis of losses. The impeller and the casing are a rotating component and a stationary component. Water enters axial flow through the impeller eyes and exits radial flow in centrifugal pump. The pump casing is to control the liquid to the impeller, transfers into pressure the high velocity kinetic energy of the flow from the impeller discharge and leaves liquid away of the energy having imparted to the liquid comes from the volute casing. The design calculation and performance analysis of single suction centrifugal pump are describe because it is the most essential useful mechanical dynamic machine in fluid works which used for water supply plants, irrigation, industry, steam power plants, hydraulic power service, mine and river water pumping system,

In this paper, pump is single stage single suction centrifugal pump and it has a head of 50 m, the flow rate of 200 m³ /hr and the motor speed is 1480 rpm. The value of specific speed is 18.46. The number of blades is 6 blades. The pump power is 29.98KW electric motor and the design is Berman Method is used. The performance analysis of centrifugal pump is carried out the dimensions of centrifugal pump. So shock losses, impeller friction losses, volute friction losses of centrifugal pump are determine in performance analysis of centrifugal pump.

Index Terms- impeller, centrifugal pump design, performance, Berman Method, Euler's Equation

I. INTRODUCTION

The centrifugal pump is a mechanical device which transports water from a lower level to higher level or from a lower pressure area to a higher pressure area. Mechanical energy is given to the pump and it converts into hydraulic energy of fluid. It consist an impeller rotating within a pump casing. In dynamics pump, the energy is transferred by rotary motion is 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. The fluid coming into the pump is pushed towards the outlet mechanically where positive pressure is generated. Pumps are classified in number of the ways based on their purpose, specifications, design and environment.

In a centrifugal pump, the liquid forces by atmospheric and other pressure into a set of rotating vanes. A centrifugal pump consists of a set of rotating vanes enclosed within a housing or casing that is used to impart energy to a fluid through centrifugal force.

A pump transfer mechanical energy from some external source to the liquid flowing through it and losses occur in any energy conversion process. The energy transferred is predicted by the Euler equation. The energy transfer quantities are losses between fluid power and mechanical power of the impeller or runner.

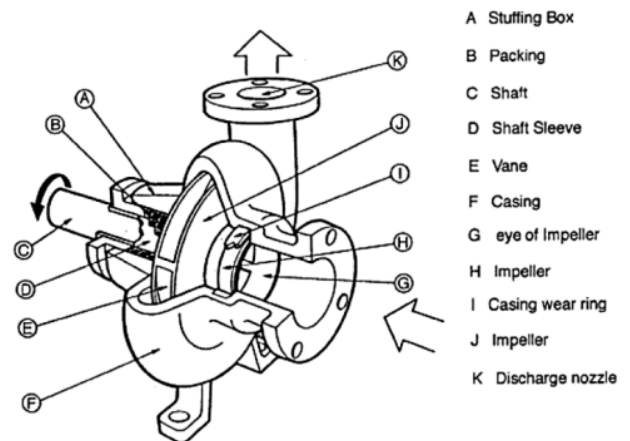


Figure 1: Single Suction centrifugal Pump

II. DESIGN CALCULATION AND PERFORMANCE ANALYSIS OF SINGLE SUCTION CENTRIFUGAL PUMP

A. Design of Centrifugal pump

The design pump is 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. For design calculation, the design parameters are taken as follows:

Flow rate, Q	= 200 m ³ /hr
Head, H	= 10 m
Rotational speed, N	= 1480 rpm
Density of water, ρ	= 1000 kg/m ³
Inlet impeller diameter, D ₁	= 110 mm

Outlet impeller diameter, $D_2 = 310$ mm
 Impeller width at inlet, $b_1 = 40$ mm
 Impeller width at outlet, $b_2 = 25$ mm
 Overall efficiency, $\eta_0 = 90\%$
 Suction pipe diameter = 6 inch
 Discharge pipe diameter = 3 inch

The design of centrifugal pump includes a large number of interdependent variables so there are several possible designs for the same duty. One of the most difficult design problems is to predict the impeller head slip. The difference between the theoretical head for a number of impeller vanes and the theoretical head deduced from the net horse power given to the fluid passing through the impeller. Before pump design or selection can be got, specification is needed to be established which express several requirements.

Specific speed is used to classify impeller on the basis of their performance, and proportions regardless of their actual size or the speed at which they operate.

$$\text{Specific speed; } n_s = 3.65N \frac{\sqrt{Q}}{H^{3/4}} \quad (1)$$

Where, n_s = specific speed
 Q = flow rate
 N = rotational speed
 H = head

Capacity, volume flow rate of a pump is the amount of water pumped per unit time and it is also known traditionally as volume flow rate. The capacity is directly related with the velocity of flow in the suction pipe.

$$\text{Capacity; } Q = AV \quad (2)$$

Where, A = area of pipe
 V = volume flow rate

Water power and shaft power

The water power is the power imported to the water by the pump. To calculate the water power, the flow rate and the pump head must be known. As a result, to provide a certain amount of power to the water a large amount of power must be provided to the pump shaft. The power is called brake power. The efficiency of the pump determines how much more power is required at the shaft.

The water power is determined from the relationship

$$N = \rho gHQ \quad (3)$$

Where, ρ = density of water
 g = gravitational acceleration

The shaft power is;

$$\text{shaft power} = \frac{\text{waterpower}}{\eta_0} \quad (4)$$

Where, η_0 = overall efficiency

Pump efficiency is

$$\eta_0 = \eta_m \times \eta_v \times \eta_r \quad (5)$$

Where, η_m = mechanical efficiency
 η_r = hydraulic efficiency
 η_v = volumetric efficiency

The inlet area of an impeller is;

$$A_1 = \frac{\pi D_1^2}{4} \quad (6)$$

Where, D_1 = inlet impeller diameter
 The outlet area of an impeller is;

$$A_2 = \pi D_2 b_2 \quad (7)$$

Where, D_2 = outlet impeller diameter
 b_2 = impeller width at outlet

Velocity diagram

A study of the several component velocities of flow through an impeller is best carried out graphically by means of velocity vectors. The shape of such vector diagram is triangular and they are called velocity triangles. It can be drawn for any point of the flow path through the impeller, but usually attention is focused on the inlet and outlet triangles.

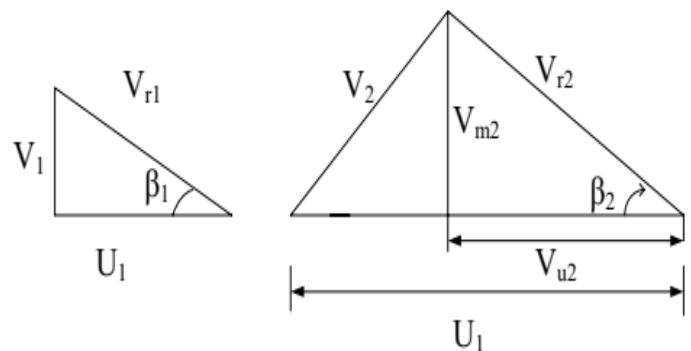


Figure 2: Inlet Flow Velocity and Outlet Flow Velocity Diagram

The blade velocity at inlet is;

$$U_1 = \frac{\pi D_1 N}{60} \quad (8)$$

Where, U_1 = blade velocity
 N = rotational speed

The relative velocity at inlet is;

$$V_{f1} = \frac{Q}{\pi D_1 b_1} \quad (9)$$

Where, V_{f1} = radial velocity at inlet
 b_1 = impeller width at inlet

The blade angle at inlet is;

$$\tan\beta_1 = \frac{V_{f1}}{U_1 - V_{\omega 1}} \quad (10)$$

Where, $V_{\omega 1}$ = tangential velocity
 β_1 = blade angle at inlet

The blade velocity at outlet is;

$$U_2 = \frac{\pi D_2 N}{60} \quad (11)$$

Where; U_2 = tangential blade velocity

The relative velocity at outlet is;

$$V_{f2} = \frac{Q}{\pi D_2 b_2} \quad (12)$$

Where, V_{f2} = radial velocity at outlet
 b_2 = impeller width at outlet

the blade angle at outlet is;

$$\tan\alpha_2 = \frac{V_{f2}}{V_{\omega 2}} \quad (13)$$

Where; α_2 = guide vane angle at outlet
 $V_{\omega 2}$ = tangential velocity at outlet

$$\tan\beta_2 = \frac{V_{f2}}{U_2 - V_{\omega 2}} \quad (14)$$

Where; $V_{\omega 2}$ = tangential velocity at outlet
 β_2 = blade angle at outlet

Theoretical head is;

$$H_{th} = \frac{1}{g} U_2 V_{\omega 2} \quad (15)$$

Where; H_{th} = theoretical head

Blade number

Higher friction losses are related to many blade casing low blade loading. The choice of fewer blades leading to a higher blade loading hydraulic losses may rise due to increasing secondary flow and stronger deviating between blade and flow direction.

$$Z \approx k \frac{D_2 + D_1}{D_2 - D_1} \sin \frac{\beta_1 + \beta_2}{2} \quad (16)$$

Where, σ = slip factor

$$\sigma = 1 - \frac{(\sin\beta_2)^{0.5}}{\beta_1 Z^{0.7}} \quad (19)$$

Shock Losses

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

$$h_s = k(Q_s - Q_N)^2 \quad (20)$$

Maximum flow rate;

$$Q_N = \pi D_1 b_1 V_{\omega 1} \quad (21)$$

The shut off head;

$$H_{shut-off} = \frac{U_2^2 - U_1^2}{2g} \quad (22)$$

In the shut off condition is, $Q = 0$ and $h_s = H_{shut-off}$

III. PERFORMANCE ANALYSIS OF SINGLE SUCTION CENTRIFUGAL PUMP

Net Theoretical Head

If the slip factor is known, the net theoretical head may be obtained from Euler's head. By using Euler's equation, It is possible to relate the theoretical characteristic obtained to the actual characteristic for various losses responsible for the difference. The slip factor is used which varies with flow rate, enables the net theoretical head curve to be obtained. At flow rate below design flow rate, separation occurs on the suction side of the blade.

The net theoretical head is calculated by;

$$H_{thn} = \frac{U_2 V_{\omega 2}}{g} \quad (17)$$

At the outlet, the whirl velocity is ;

$$V_{\omega 2} = U_2 \sigma - V_{\omega 2} \cot\beta_2 \quad (18)$$

The hydraulic efficiency is;

$$\eta_r = 1 - \frac{0.42}{(\log D_1 - 0.172)^2} \quad (23)$$

The flow rate and the shock loss of head is showed in Figure 3. The shock loss of head is increasing when the flow rate is decreasing. Shock loss is not in the design point condition. If this condition is over, the value of the shock loss of head is high.

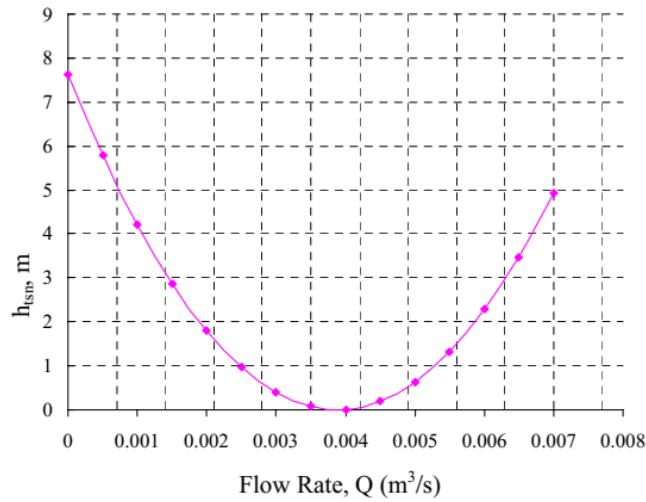


Figure 3 Shock Losses and Flow Rate Graph

Impeller Friction losses

The impeller was designed that the width of the impeller would become small and friction loss at the flow passage would become large. Therefore to relieve the increase in friction losses, 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.

The impeller friction losses are;

$$h_1 = \frac{b_2(D_2 - D_1)(V_{r1} - V_{r2})^2}{2\sin\beta_2 H_r 4g} \quad (24)$$

The hydraulic radius;

$$H_r = \frac{b_2 \left(\frac{\pi D_2}{Z} \right) \sin\beta_2}{b_2 + \left(\frac{\pi D_2}{Z} \right) \sin\beta_2} \quad (25)$$

The inlet relative velocity is;

$$V_{r1} = \frac{V_{\omega 1}}{\sin\beta_1} \quad (26)$$

The outlet relative velocity;

$$V_{r2} = \frac{V_{\omega 2}}{\sin\beta_2} \quad (27)$$

In figure 4 is showed the influence of the geometry of the impeller friction losses. The analysis of the curves describes that small difference between the points for the flow rate and the impeller friction loss of head. The impeller loss of head is increasing when the flow rate is increasing.

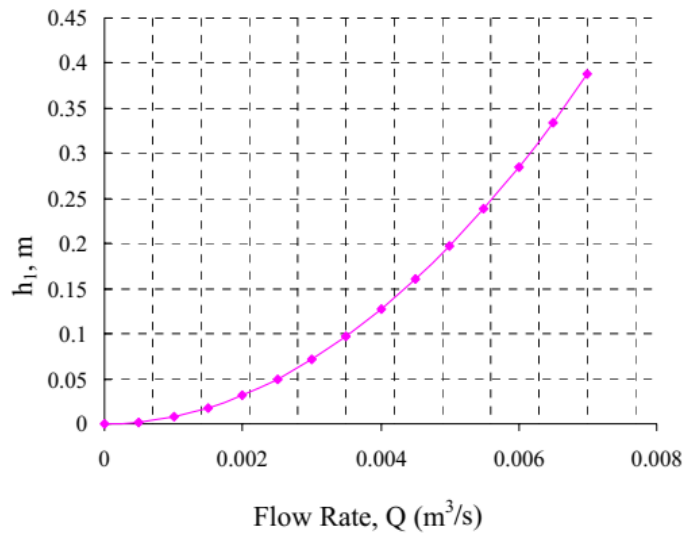


Figure 4 Impeller Friction Losses and Flow Rate Graph

The losses of Volute Friction

The losses of volute friction results from a mismatch of the velocity is outer the impeller and the velocity in the volute throat. If the velocity reaching the volute throat is larger than the velocity at the throat, the velocity head difference is less. The velocity reaching the volute throat by assuming that the velocity is leaving the impeller decreases in proportional to the radius because of the conservation of angular momentum.

The volute friction loss are;

$$h_2 = \frac{C_v V_3^2}{2g} \quad (28)$$

The volute throat velocity is;

$$V_3 = \frac{Q}{A_3} \quad (29)$$

The volute throat area;

$$A_3 = V_{u2} \left(\frac{D_2}{D_3} \right) \quad (30)$$

Whirl velocity;

$$V_{u2} = U_2 - V_{\omega 2} \cot \beta_2 \quad (31)$$

The volute flow coefficient is;

$$C_v = 1 + \left(0.02 \times \frac{L_{vm}}{D_{vm}} \right) \quad (32)$$

The volute circumferential length;

$$L_{vm} = \frac{\pi D_1}{8} \quad (33)$$

The diameter of volute is get tangent circle from the geometry of volute casing.

The volute mean diameter is;

$$D_{vm} = \frac{D_1}{8} \quad (34)$$

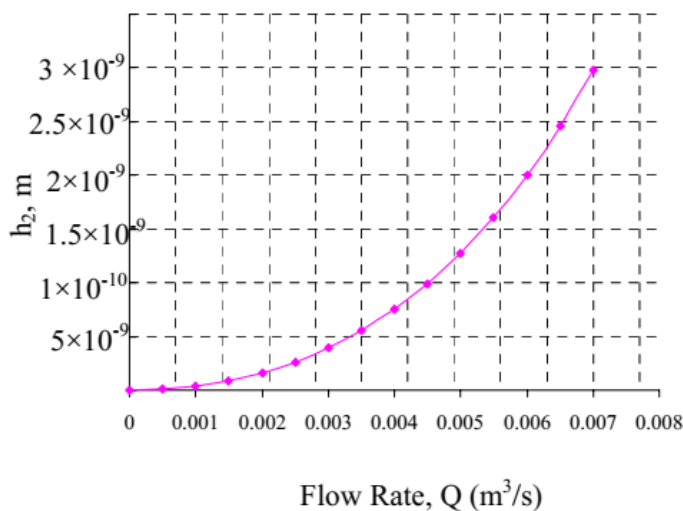


Figure 5 Volute Friction Losses and Flow Rate Graph

The volute losses and flow rate graph are figure 5. When the flow rate is increased, the volute friction loss of head increases. The volute friction coefficient is decrease for small values of the volute flow coefficient.

IV. DESIGN RESULTS OF CENTRIFUGAL PUMP

Table I: calculation Result of single suction centrifugal pump

No	Descriptions	Symbol	Results
1	Guide Vane angle at inlet	α_1	0
2	Guide Vane angle at outlet	α_2	6.23°
3	Blade angle at inlet	β_1	24.98°
4	Blade angle at outlet	β_2	32°
5	Number of blades	Z	6
6	Inlet area of impeller	A_1	0.0863 m²
7	Outlet area of impeller	A_2	0.0243 m²
8	Specific Speed	n_s	18.46
9	Water power		20.12Hp
10	Shaft power		29.98 KW

V. CONCLUSION

Centrifugal pump are fluid kinetic machines designed for increasing power within a rotating impeller. In centrifugal pumps, the delivery head is depended on the flow rate. This relationship, are called pump performance, is illustrated by curves. Characteristic curve of a centrifugal pump value is getting of theoretical head, slip, shock losses, impeller losses and volute losses are calculated by varying volume flow rate. The performance analysis of centrifugal pump is predicted in this paper.

A centrifugal pump is using a rotating impeller to increase the pressure and flow rate of a fluid. The fluid is entering the pump impeller along to the rotating axis and is accelerating by the impeller, flowing radial outwards into a diffuser or volute chamber, from where it exits into the downstream piping system. The angle value of inlet and outlet blade angel degree are large the pump pressure is fall down. The impeller is very important in pump because that can be change many performance of a fluid work.

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First Author – Aye Thida San, M.E (Mech), Technological University (Pathein) and ayethidasan1980@gmail.com

Second Author – Min Min Oo, M.E (Mech), Technological University (Pathein) and minminoo97.mm@gmail.com

Third Author – Chaw Wint Yee Zaw, M.E (Mech), Technological University (Thanlyin) and chawzaw7@gmail.com