Performance prediction of a centrifugal pump

NWE NI WIN
Ph.D. Researcher, Department of Mechanical Engineering
Yangon Technological University, Yangon, Myanmar.
Email - shwekyikhin@gmail.com

Abstract: This paper deals with the design and performance prediction of a centrifugal pump. In this paper, centrifugal pump is predicted by using a single-stage double-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. The pump casing is to guide the liquid to the impeller, converts into pressure the high velocity kinetic energy of the flow from the impeller discharge and leads liquid away of the energy having imparted to the liquid comes from the volute casing. A design of centrifugal pump is carried out and analyzed to get the best performance point. The design and performance analysis of centrifugal pump are chosen because it is the most useful mechanical rotodynamic machine in fluid works which widely used in domestic, irrigation, industry, large plants and river pumping projects. The design data of the model pump are the flow rate (0.13 m^3/sec), head (35 m), operating speed (1470 rpm) and pump efficiency is 70% respectively. The performance analysis 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.

Key Words: Centrifugal pump, Double-suction impeller, **Kyushu Method, Euler Equation**, SolidWorks Software.

1. INTRODUTION:

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 two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving parts, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, cast iron, stainless steel as well as other materials. The diffuser (also called as volute) houses the impeller and directs the water off the impeller. 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.Centrifugal pumps are widely used for irrigation, water supply plants, stream power plants, sewage, oil refineries, chemical plants, hydraulic power service, food processing factories and mines. Moreover, they are also used extensively in the chemical industry because of their suitability in practically any service and are mostly used in many applications such as water pumping project, domestic water raising, industrial waste water removal, raising water from tube wells to the fields. In a centrifugal pump, the liquid is forced by atmospheric or other pressure into a set of rotating vanes. A centrifugal pump consists of a set of rotation 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. Thus, centrifugal pump may be taken losses of energy. The kinds of loss of centrifugal pumps can be differentiated in internal losses and external or mechanical losses. The internal loss is hydraulic losses or blade losses by friction, variations of the effective area or changes of direction losses of quantity at the sealing places between the impeller and housing at the rotary shaft seals. The external or mechanical loss is sliding surface losses by bearing friction or seal friction.

2. DESIGN PROCEDURE OF CENTRIFUGAL PUMP:

The centrifugal pump is analysed single stage double-suction centrifugal pump. The general method of design developed by "Kyushu Method" is used in this paper. The usual design is based upon a certain desired head (H), and capacity (Q) at which the pump will operate most of the time, and the type of driver (motor, turbine) and its speed may be specifiedThe design procedure of the double-suction impeller involves the following steps:

For radial type impeller, the specific speed is calculated by the following equation. The specific speed,

$$n_s = \frac{n\sqrt{Q}}{H^{\wedge\frac{3}{4}}} \quad (1)$$

The input power,

$$L = \frac{\rho g Q H}{\eta} \quad (2)$$

2.1. Rated Output Power of Electric Motor

The rated output power of electric motor, L_r is obtained by

$$L_r \geq \frac{(1+f_a)L}{\eta_{tr}} \quad (3)$$

The allowable factor, $f_a = 0.4$ (for electric motor)

Transmission efficiency, $\eta_{tr} = 1$ (for direct coupling)

The volumetric efficiency is calculated by using the following equation.

$$\eta_v = \frac{1}{1+\frac{1.124}{n_s^{\frac{2}{3}}}} \quad (4)$$

The diameter of end of the main shaft is obtained by

$$d_c \geq K_{sh}\sqrt[3]{\frac{L_r}{n}} \quad (5)$$

The shaft diameter D_{sh} is obtained twenty percent greater than the diameter of the end of the main shaft. Therefore,

$$D_h = (1.5 \sim 2.0)\, D_{sh} \quad (6)$$

The length of the hub, L_h at the impeller is calculated by the Equation:

$$L_h = (1.0 \sim 2.0)\, D_{sh} \quad (7)$$

The diameter of impeller eye D_0 is calculated from the following equation:

$$D_0 = \sqrt{\frac{4Q_s'}{2\pi V_{m0}} + D_h^2} \quad (8)$$

Where the velocity at the eye suction is

$$V_{m0} = K_{m0}\sqrt{2gH} \quad (9)$$

$$K_{m0} = 0.07 + 0.00023 n_s \quad (10)$$

Vane inlet velocity, V_{m1} is $V_{m1} = K_{m1}\sqrt{2gH}$ (11)

Vane outlet velocity, V_{m2} is

$$V_{m2} = K_{m2}\sqrt{2gH} \tag{12}$$

Vane outlet tangential velocity, U_2 is

$$U_2 = K_u\sqrt{2gH} \tag{13}$$

2.2. *Dimensions of Impeller Inlet and Outlet Diameter*

The impeller inlet diameter is obtained by

$$D_{1m} \sim D_1 = \frac{D_{1h} + D_{1s}}{2} \tag{14}$$

The impeller outlet diameter is obtained from

$$D_2 = \frac{60\,U_2}{\pi\,n} \tag{15}$$

Inlet tangential velocity, U_1 is

$$U_1 = \frac{\pi\,D_1 n}{60} \tag{16}$$

2.3. *Dimensions of 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.

$$\beta_1 = \tan^{-1}\left(\frac{V_{m1}}{U_1}\right) \tag{17}$$

In this design the vane outlet angle β_2, is assumed as 22.5˚ which is the average value for all specific speed.

The number of blades, Z is $Z = 6.5 \times \frac{D_2 + D_1}{D_2 - D_1}\sin\left(\frac{\beta_1 + \beta_2}{2}\right)$ (18)

The inlet passage width b_1 and b_2 are calculated by

$$b_1 = \left(\frac{Q_s'}{2\pi D_1 V_{m1}}\right) \times \left(\frac{\pi D_1}{\pi D_1 - S_1 Z}\right) \tag{19}$$

$$b_2 = \left(\frac{Q_s'}{\pi D_2 V_{m2}}\right) \times \left(\frac{\pi D_2}{\pi D_2 - S_2 Z}\right) \tag{20}$$

2.4. *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.

Impeller outside diameter,

$D_2 = D_A$= 360 mm

Impeller inside diameter,

$D_D = D_{1h} = 171$ mm

The required parameter to layout he impeller blade,

$$\rho = \frac{R_b^{\,2} - R_a^{\,2}}{2(R_b \cos\beta_b - R_a \cos\beta_a)} \tag{21}$$

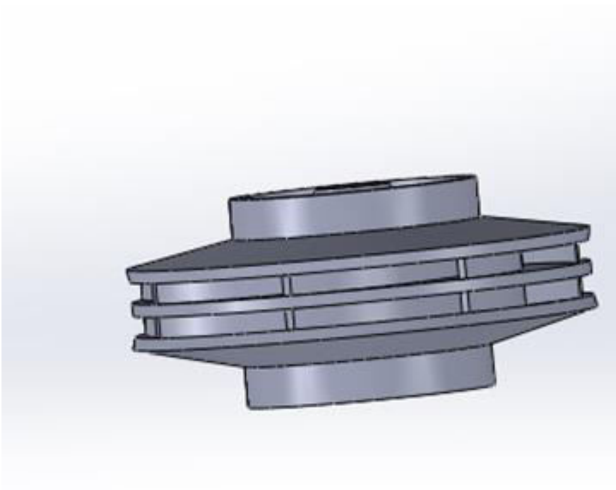

Figure1. Modelling of Double-suction impeller by Solidworks

Impeller blade curve is drawn by using radii of base circle, diameter of the shaft, diameter of the hub, blade height, and shroud thickness. The three dimensional view of the centrifugal pump is shown in the following figure 2.

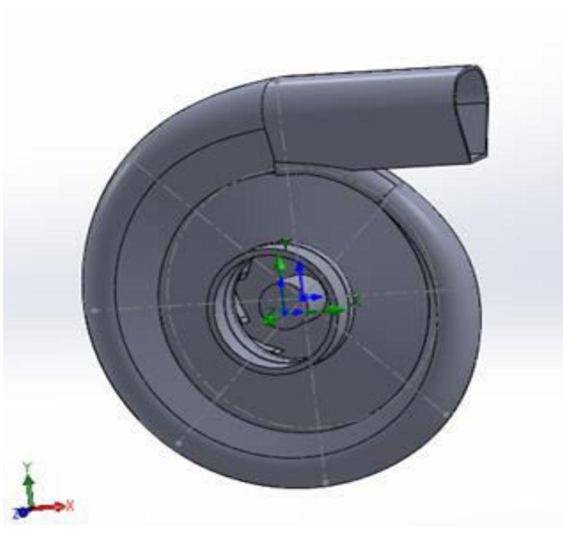

Figure2.Assembly of impeller and volute casing by SolidWorks Software

3. PERFOPMANCE PREDICTION OF CENTRIFUGAL PUMP:

The performance of centrifugal pump is described by a graph plotting the head generated by the pump over a range of flow rates. A typical pump performance curve are included its efficiency and brake horsepower, both of which are plotted with respect to flow rate. The output of a pump running at a given speed is the flow rate delivery by it and the head developed. Thus, head is against flow rate at constant speed forms fundamental performance characteristic of a pump.

In order to achieve this performance, an input power is required which involves efficiency of energy transfer. The efficiency of a pump is the ratio of the pump's fluid power to the pump shaft horsepower. An important characteristic of the head/flow curve is the best efficiency point. At the best efficiency point, the pump operates most cost-effectively both in terms of energy efficiency and maintenance considerations. The efficiency of a centrifugal pump depends upon the hydraulic losses, disk friction mechanical losses and leakage losses.

3.1. Theoretical Head

The Euler head is determined from zero to maximum theoretically attainable flow using. The theoretical head: $H_{th} = \frac{1}{g} U_2 V_{u2}$

where, U_2 and V_{u2} are outlet tangential velocity and whirl velocity.

Whirl velocity: $V_{u2} = U_2 - V_{m2} \cot\beta_2$

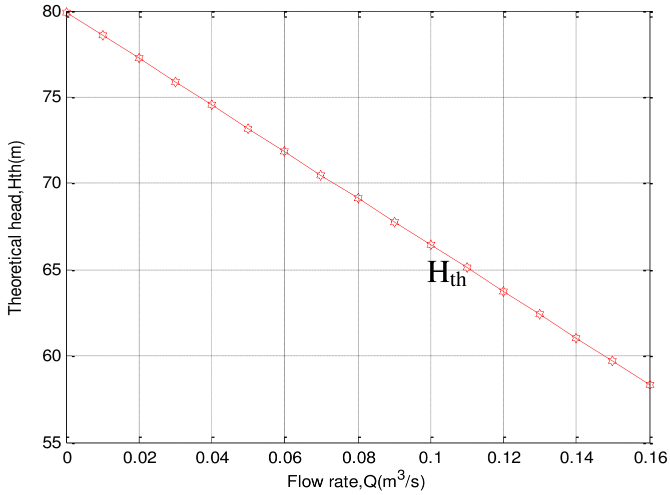

Figure 3. Theoretical Head versus Flow Rate Graph

3.2. 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 obtained. At flow rates below design flow rate, separation occurs on the suction side of the blade. The net theoretical head is calculated by:

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

The whirl velocity at the outlet is; $V_{u2} = U_2\sigma - V_{m2} \cot\beta_2$

where, σ is the slip value.

Slip value is obtained by using the following equation: $\sigma = 1 - \frac{(\sin\beta_2)^{1/2}}{Z^{0.7}}$

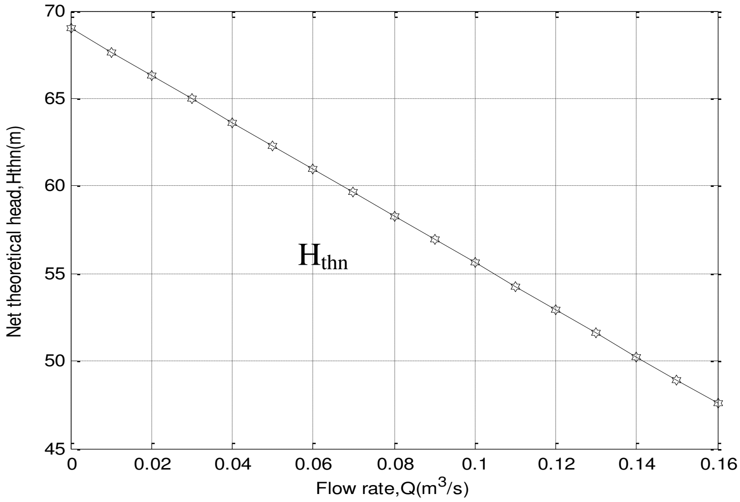

Figure 4. Net Theoretical Head versus Flow Rate Graph

3.3. 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: $h_s = k(Q_s - Q_N)^2$

Maximum flow rate: $Q_N = \pi D_1 b_1 V_{m1}$

The shut off head: $H_{shut\text{-}off} = \dfrac{U_2^2 - U_1^2}{2g}$

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

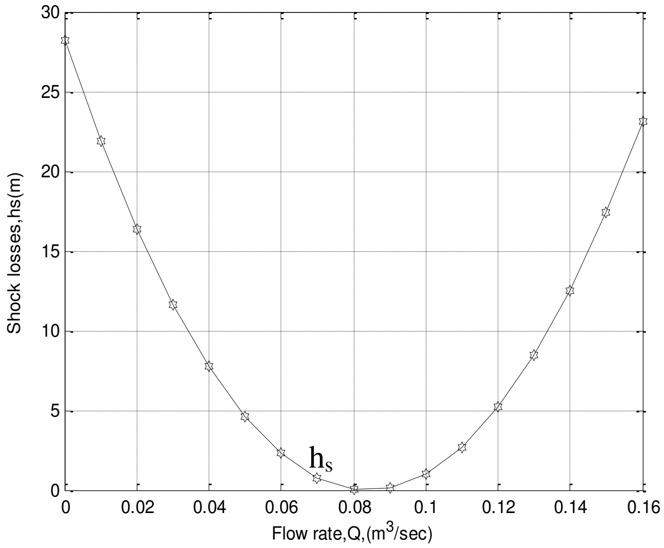

Figure 5. Shock losses versus Flow Rate Graph

3.4. Impeller Friction Losses

The impeller were 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.

The impeller friction loss,

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

The hydraulic radius is calculated by,

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

The inlet relative velocity, $V_{r1} = \dfrac{V_{m1}}{\sin\beta_1}$

The outlet relative velocity, $V_{r2} = \dfrac{V_{m2}}{\sin\beta_2}$

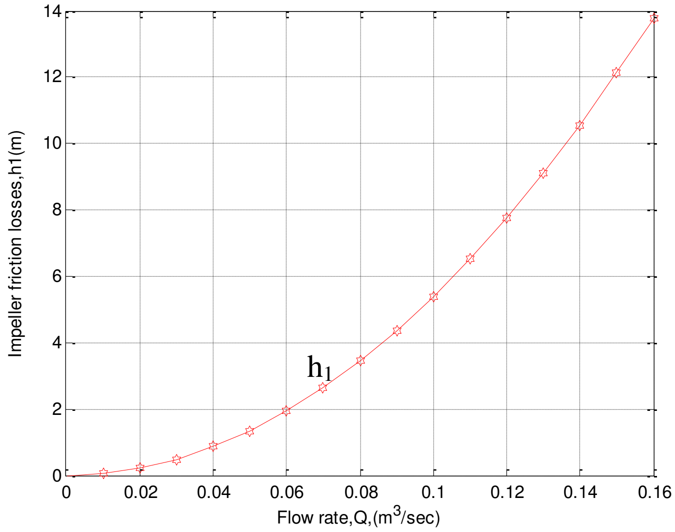

Figure6. Impeller Friction Losses versus Flow Rate Graph

3.5. Volute Friction Losses

This loss results from a mismatch of the velocity leaving the impeller and the velocity in the volute throat. If the velocity approaching the volute throat is larger than the velocity at the throat, the velocity head difference is less. The volute friction losses are calculated by

$$h_2 = \frac{C_v V_3^{\,2}}{2g}$$

The volute throat velocity: $V_3 = \dfrac{Q}{A_3}$

The volute flow coefficient is $C_v = 1 + (0.02 \times \dfrac{L_{vm}}{D_{vm}})$

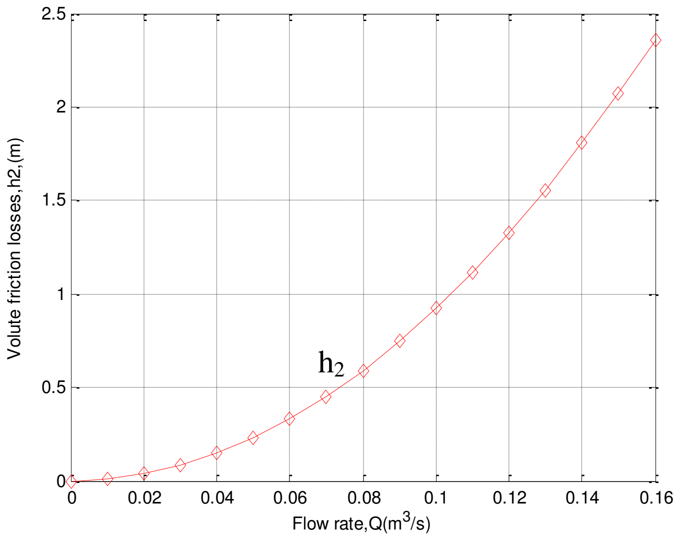

Figure 7.Volute Friction Losses versus Flow Rate Graph

3.6. 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. The disk friction loss is;

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

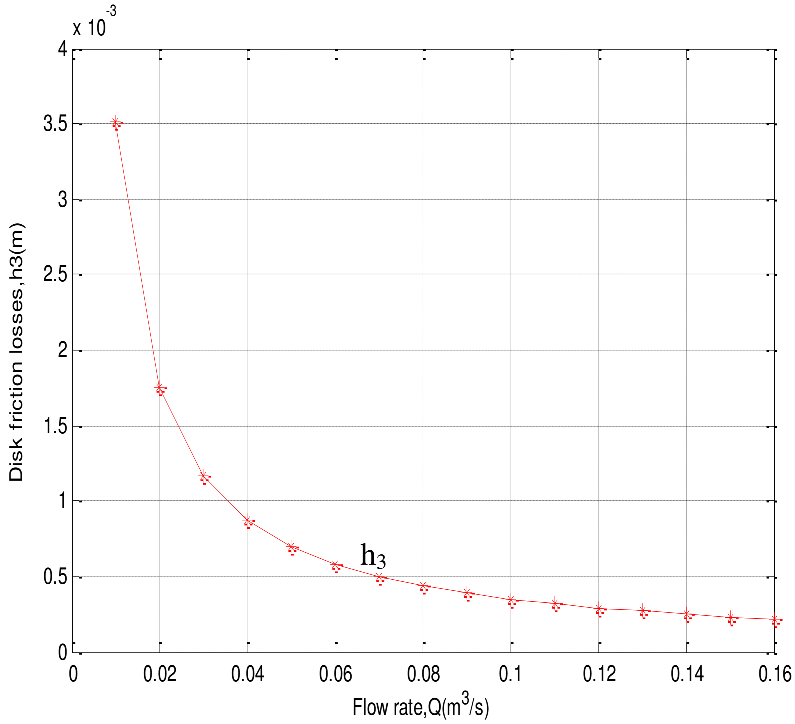

Figure 8.Disk Friction Losses versus Flow Rate Graph

3.7. 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 is calculated by using the following eqation.The recirculation loss,

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

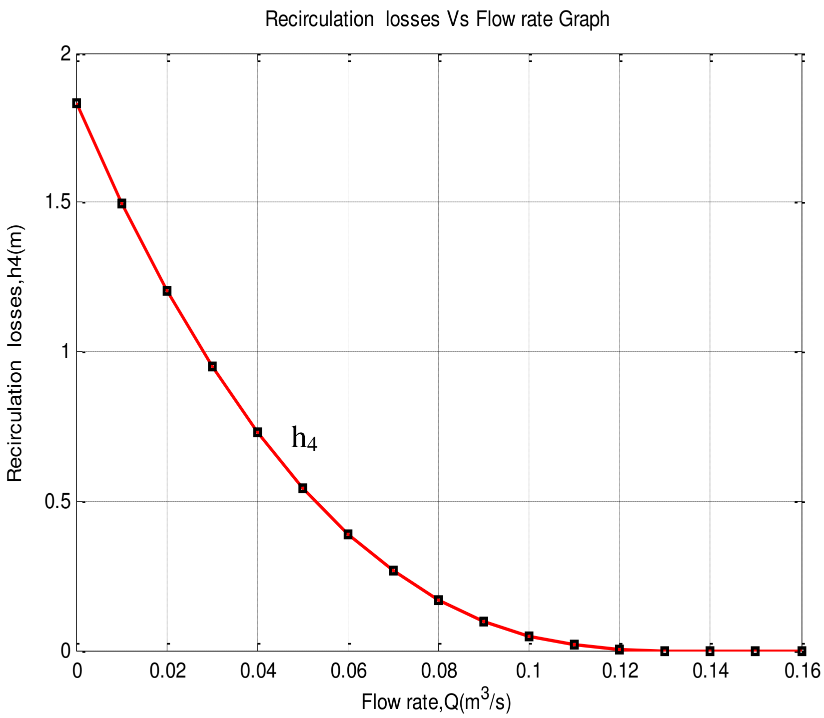

Figure 9. Recirculation Losses versus Flow Rate Graph

3.8. Actual Head

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. So, the actual pump head is calculated by using;

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

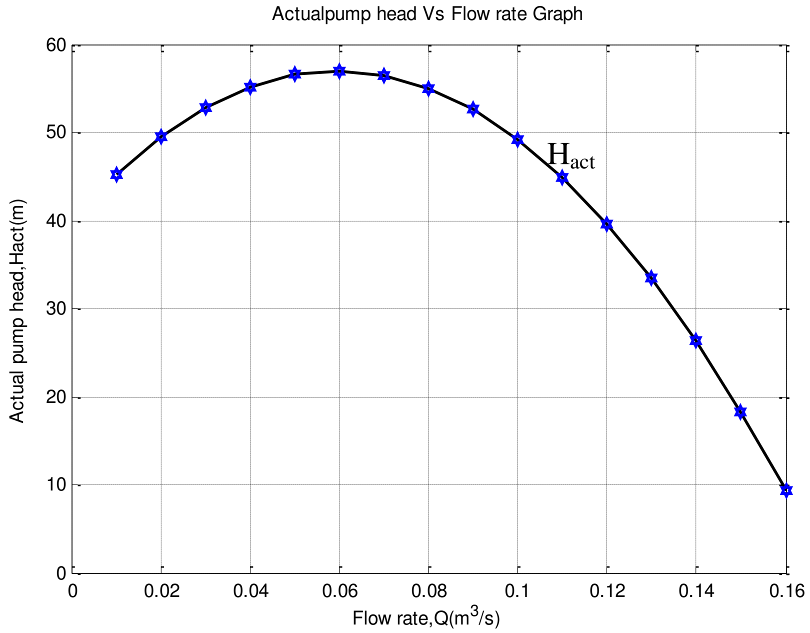

Figure 10. Actual Pump Head versus Flow Rate Graph

The actual pump head (H_{act}) is 34.03 m by using "Matlab Software". The design head of the centrifugal pump is 35 m. The design flow rate (Q) is 0.13 m^3/sec.

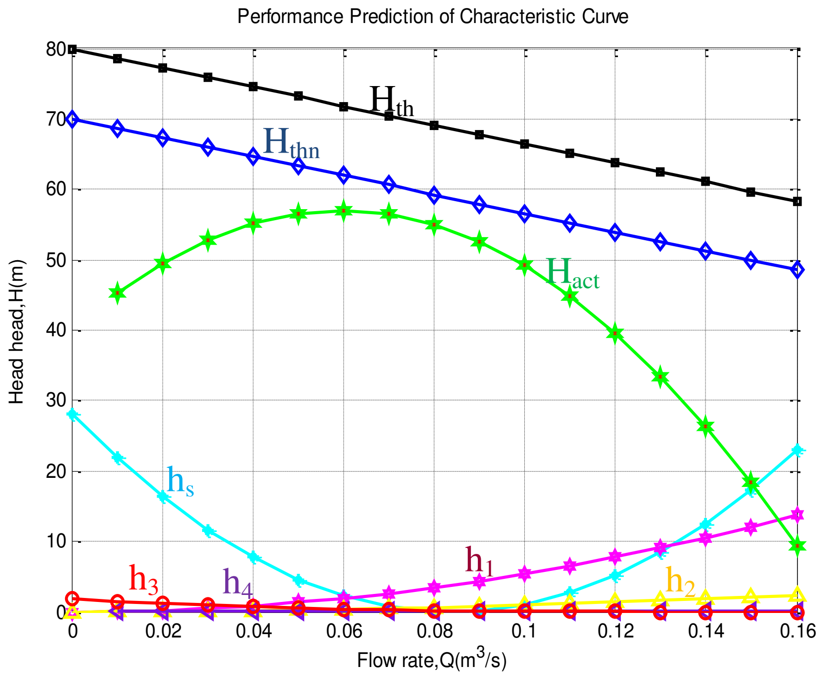

Figure 11. Prediction of Characteristic Curve of Centrifugal Pump

The figure 11 is illustrated prediction of characteristic cure of centrifugal pump by using "Matlab Software".

3.9. Efficiency

The efficiency of a pump is the ratio of the pump's fluid power to the pump shaft horsepower. The overall efficiency of the centrifugal pump is investigated by applying the following equation. The overall efficiency is

$$\eta_{overall} = \frac{\rho g Q H_{act}}{L}$$

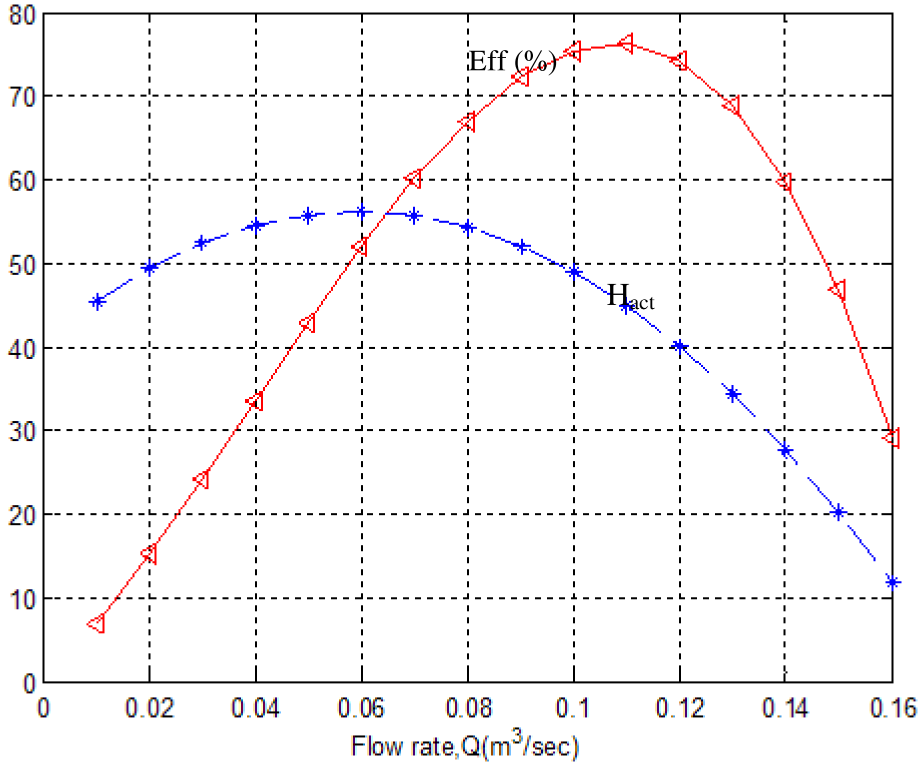

Figure12. Actual Head and Efficiency Curve of Centrifugal Pump

It is found that the overall efficiency for design flow rate 0.13 m^3/sec of centrifugal pump is 68.88%. The maximum efficiency is 76.02% at the flow rate 0.11 m^3/sec.

4. CONCLUSION:

The 3D model of impeller and volute casing was generated with SolidWorks Software.The performance of centrifugal pump is described by a graph plotting the pressure head generated by the pump over a range of flow rates. A typical pump performance curve are included its efficiency and brake horsepower, both of which are plotted with respect to flow rate.

Some looses of centrifugal pump with the values Q and H are determined for the various operating points are shown. Centrifugal pumps are fluid-kinetic machines designed for power increase within a rotating impeller. In centrifugal pumps, the delivery head depends on the flow rate. This relationship, also called pump performance, is illustrated by curves. To get characteristic curve of a centrifugal pump, values of theoretical head, slip, shock losses, recirculation losses and other friction losses are calculated by varying volume flow rate. The performance of centrifugal pump is predicted in this paper. And then, the actual performance curve of centrifugal pump is also predicted.

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REFERENCES:

1. SolidWorks Corporation: *Driving Better Product Design with SolidWorks Simulation,* (2012), http://www.solidworks.com/sw/wp-driving- product-design-with-simulation.htm.
2. Larry Bachus and Angel Custodio: *Known and Understand of Centrifugal Pump,* Japan: Bachus Company, Tokyo 113, (2003).
3. Igor, J. Joseph, P. and Charles, C.: *Pump Hand Book,* McGraw-Hill Company, Third Edition, USA, (2001).
4. Tuzson, J.2000. "*Centrifugal Pump Design".* USA: John Wiley and Sons.Inc.
5. Frank M. White: *Fluid Mechanics,* McGraw-Hill Book Co, (1999).
6. Nelik, Lev: *Centrifugal and Rotary Pump, Fundamental with Application,* New York, (1999).
7. S.L.Dixon, B.Eng, Ph.D.: *Fluid Mechanics and Thermodynamic of Turbomachinery,* Fourth Edition, (1998).
8. Kyushu Institute of Technology, *Fluid Mechanics of Turbo Machinery Training Course,* (1996).
9. Val S. Lobanoff, Robert R. Rosset al,: *Centrifugal Pump: Design and Application,* Second Edition, (1992).
10. Kumar, Dr. Ds, *Fluid Mechanical and Fluid Power Engineering,* 1987.
11. Frank, A. 1953. *Pumps.* *2nd Edition.* Mc Graw-Hill.
12. Daugherty, L. 1951. *Centrifugal Pumps.* McGraw-Hill Book Company.
13. Sam Yedidiah, No Date. *Centrifugal Pump User's Guide Book,* an International Thomson Publishing.