Design and Performance Analysis of Centrifugal Pump

Khin Cho Thin, Mya Mya Khaing, and Khin Maung Aye

Abstract**—**This paper deals with the design and performance analysis of centrifugal pump. 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. 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 water pumping system. Moreover, centrifugal pumps are produced by manufacturing processes in Myanmar.

In this paper, the pump is driven by one horse power electric motor and the design is based on Berman Method. The head and flow rate of this pump are 10 m and $0.179m^3/s$ and the motor speed is 2900 rpm. The low specific speed is chosen because the value of specific speed is 100. The number of impeller blade is 9 blades. 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.

Keywords**—**Berman Method, Euler Equation.

I. Introduction

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 to transfer liquids from low-pressure to high pressure in this system, the liquid would move in the opposite direction because of the pressure difference.

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.

A centrifugal pump delivers useful energy to the fluid on pumpage largely through velocity changes that occur as this fluid flows through the impeller and the associated fixed passage ways of the pump. It is converting of mechanical energy to hydraulic energy of the handling fluid to get it to a required place or height by the centrifugal force of the impeller blade. The input power of centrifugal pump is the mechanical energy and such as electrical motor of the drive shaft driven by the prime mover or small engine. The output energy is hydraulic energy of the fluid being raised or carried.

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.

II. Design and Performance Analysis of Centrifugal Pump

A. Design of Centrifugal Pump

The design pump is one horse power motor drive 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, $Q = 0.00293\ m^3/s$
Head, $H = 10$ m
Pump speed, $n = 2900$ rpm

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gravitational acceleration, $g = 9.81\ m/s^2$
density of water, $\rho = 1000\ kg/m^3$

The design of centrifugal pump involves 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 horsepower given to the fluid passing through the impeller. Before pump design or selection can be got, specifications is needed to be established which express several requirements.

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

$$\text{Specific Speed: } n_s = 3.65n\frac{\sqrt{Q}}{H^{3/4}} \quad (1)$$

Capacity, volute 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 and V are area of pipe and volume flow rate respectively.

Water Power and Shaft Power

The power imparted to the water by the pump is called water power. To calculate 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 larger amount of power must be provided to the pump shaft. This 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

$$N = \rho g H Q \quad (3)$$

The shaft power is:

$$\textit{Shaft power} = \textit{water power}/\ \eta_o \quad (4)$$

Pump efficiency is

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

Maximum shaft power is:

$$M_{max:} = \alpha_1 \rho g H Q/\ \eta_o \quad (6)$$

α_1 is the safety factor in charge condition of the work of pump

Inlet diameter of impeller is:

$$D_1 = (1.1\sim1.15)K_o\sqrt[3]{\frac{Q}{n}} \quad (7)$$

The value of K_0 is chosen as 4.5.

The outlet diameter of impeller is:

$$D_2 = 19.2\left(\frac{n_{sopt}}{100}\right)^{1/6}\frac{\sqrt{2gH}}{n} \quad (8)$$

D_0 is the eye diameter of impeller.

$$D_o = K_o\sqrt[3]{\frac{Q}{n}} \quad (9)$$

where K_0 is the constant parameter which value is chosen as 4.5.

The shaft diameter at the hub section is:

$$d_s = \sqrt[3]{\frac{T}{0.2\tau}} \quad (10)$$

The torsional moment is estimated by:

$$T = 9.65\frac{N_{max:}}{n} \quad (11)$$

The hub diameter is usually 1.5 to 2 times of the shaft.
The hub diameter is:

$$D_{bt} = (1.5 \sim 2)\ d_{sh} \quad (12)$$

The hub length is two times of the shaft.

$$L_{bt} = 2d_{sh} \quad (13)$$

Inlet width of the impeller is: $b_1 = R_0/2$ (14)

R_0 is the radius of the impeller eye.

Outlet width of the impeller is:

$$b_2 = 0.78(n_{s\ opt:}/100)^{1/2}(Q/n)^{1/3} \quad (15)$$

The hydraulic efficiency is;

$$\eta_r = 1 - \frac{0.42}{(\log D_0 - 0.172)^2} \quad (16)$$

Leakage head is:

$$H_y = H_T - \frac{V_2^2}{2g} - \frac{U_2^2}{8g}\left(1 - \frac{D_y^2}{D_2^2}\right) \quad (17)$$

where H_T and D_y are the pressure head and seal diameter.

The pressure head is: $H_T = H/\eta_r$ (18)

The seal diameter is: $D_y = D_0 + 10$ (19)

The minimum clearance between the war ring and casing is:

$$\delta = 10^{-3}\ D_y \quad (20)$$

Inlet blade angle of impeller is:

$$\tan\beta_1 = \frac{U_1}{V_{m_1}} \quad (21)$$

The outlet angle of impeller is assumed as 40°.

Blade numer:

$$Z = \frac{6.5\frac{D_2 + D_1}{D_2 - D_1}\sin((\beta_1 + \beta_2))}{2} \tag{22}$$

The require parameters to draw the impeller blade are calculated by the following equations:

The radius of the impeller at outlet is: $R_A = D_2/2$ (23)

The value of the radius $R_B$is assumed as 27.5 mm which is taken slightly than the radius R_A.

The value of the radius R_C is assumed as 27.5 mm which is taken slightly than the radius R_B.

Require parameter to layout the impeller blade is:

$$\rho_A = (R_A^2 - R_B^2)^2/2\times 1/(R_A\cos\beta_2 - R_B\cos\beta_1) \tag{24}$$

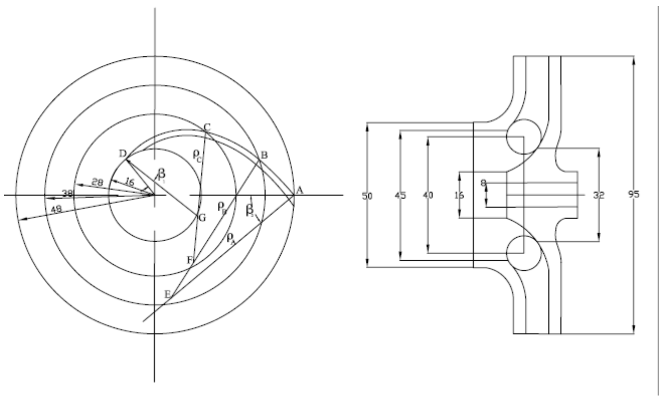

Fig. 1 Drawing of Impeller blade

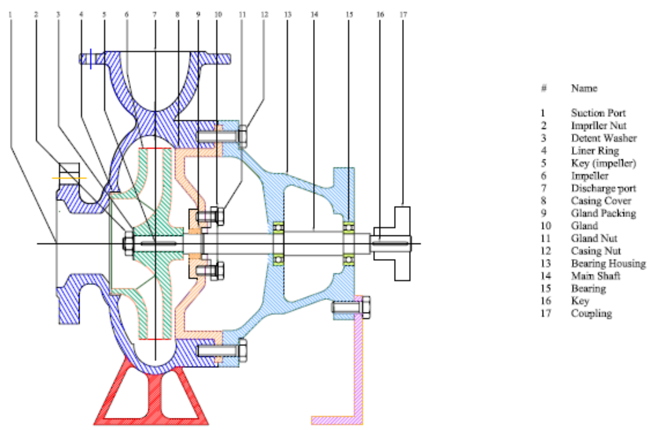

Fig. 2 Detail Assembly of Centrifugal Pump

B. Calculation of Variable Parameters in Cascade

The known parameters are:
Outlet diameter of impeller, D_2= 95 mm
Eye diameter of impeller, D_o= 45 mm
Width of impeller at outlet, b_2 = 8 mm
Hub diameter of impeller, D_{bt}= 10 mm
Hub length of impeller, L_h= 16 mm

The inlet area of the impeller is;

$$A_1 = \frac{\pi}{4} D_0^{\ 2} \tag{25}$$

The outlet area of the impeller is;

$$A_2 = \pi D_2 b_2 \tag{26}$$

The radius in cascade is divided into nine sections between inlet area A_1 and outlet area A_2. So, variable parameters in cascade are calculated as nine sections.

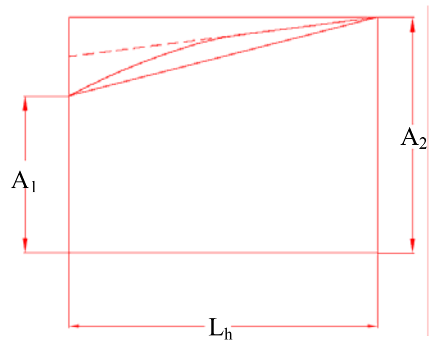

Fig. 3 Variable Areas in Cascade

TABLE I
VARIABLE PARAMETERS IN CASCADE

No.	R	A	N	S	t	θ	σ	σ_1
1	25	1590	10.123	14	1.184	47	5	6.837
2	32	1720	8.555	12.5	0.332	44	4.8	6.911
3	37	1780	7.657	11	0.485	40	4.65	7.234
4	42.5	1830	6.853	9.5	0.779	35	4.6	8.02
5	48	1890	6.267	8.2	1.201	31	4.5	8.737
6	55	1980	5.729	7	1.594	30	4.3	8.6
7	65	2100	5.142	5.3	2.413	28	4.15	8.839
8	75.7	2200	4.638	4	2.957	26	4	9.125
9	88.4	2300	4.141	3	3.211	25	3.8	8.992

TABLE II
VARIABLE PARAMETERS IN CASCADE

No.	t_1	ψ	β	Q_k	V_m	V	V_r
1	26.179	0.738	40	3112	1.957	12.092	3.045
2	33.509	0.793	40	3112	1.809	12.269	2.815
3	38.745	0.813	40	3112	1.748	12.341	2.719
4	44.505	0.819	37.5	3112	1.701	12.209	2.794
5	50.264	0.826	35	3112	1.646	12.073	2.871
6	57.594	0.851	32.5	3112	1.571	11.958	2.925
7	68.066	0.87	30	3112	1.482	11.858	2.964
8	79.061	0.885	30	3112	1.415	11.975	2.829
9	92.569	0.903	30	3112	1.353	12.081	2.706

In this result, the outlet area is greater than the inlet are of impeller. If the radius of impeller is changed, other parameters are also variable which parameters are power, length, thickness of blades, constant flow coefficient and flow velocity, etc.

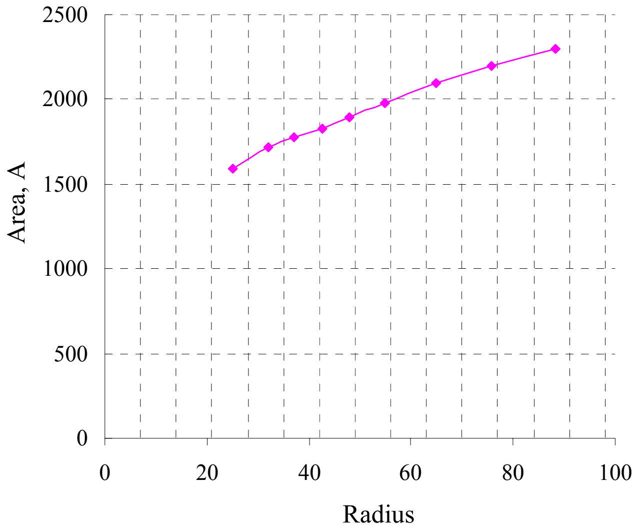

Fig. 4 Area versus Radius in Cascade

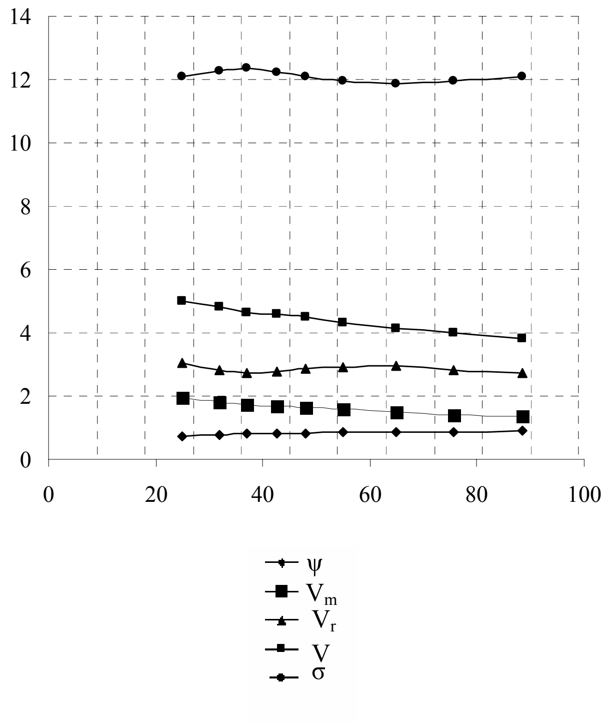

Fig. 5 Variable Parameters in Cascade

III. Performance Analysis 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.

A. Theoretical Head

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

The theoretical head: $H_{th} = 1/gU_2V_{u2}$ (27)

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

Whirl velocity: $V_{u2} = U_2 - V_{m2} \cot\beta_2$ (28)

where V_{m2} and β_2 are outlet flow velocity and outlet blade angle.

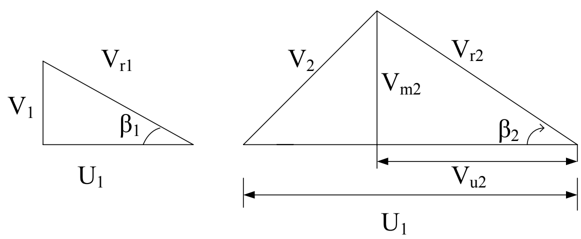

Fig. 6 Inlet and Outlet Velocity Diagram

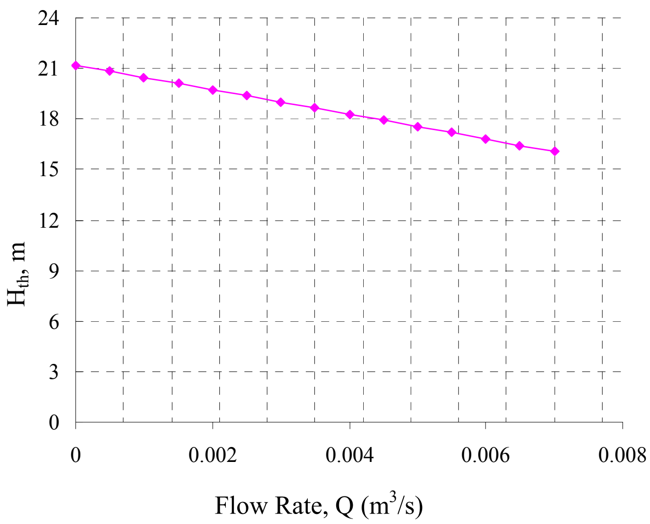

Fig. 7 Theoretical Head versus Flow Rate Graph

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 obtained. At flow rates below design flow rate, separation occurs on the suction side of the blade.

The net theoretical head is calculated by:

$$H_{thn} = \frac{U_2 V_{u2}}{g} \tag{29}$$

The whirl velocity at the outlet is;

$$V_{u2} = U_2\sigma - V_{m2} \cot\beta_2 \tag{30}$$

where, σ is the slip value.

Slip value is obtained by using the following equation:

$$\sigma = 1 - \frac{(\sin\beta_2)^{½}}{\beta_1 Z^{0.7}} \tag{31}$$

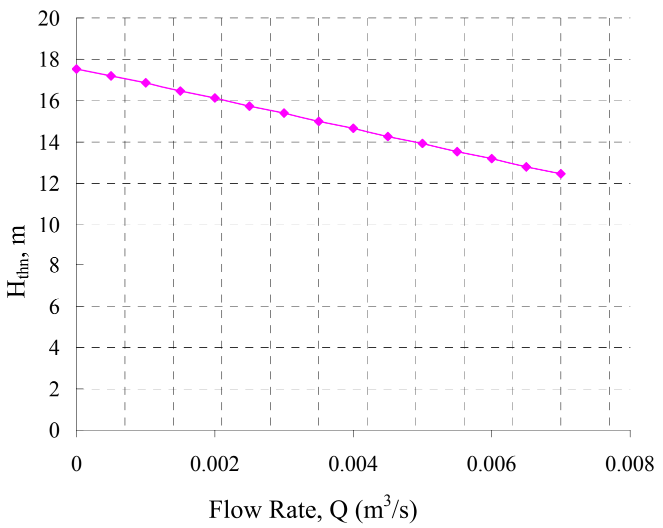

Fig. 8 Net Theoretical Head Relative to Flow Rate

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:

$$h_s = k(Q_s - Q_N)^2 \tag{32}$$

Maximum flow rate: $Q_N = \pi D_1 b_1 V_{m1}$ (33)

The shut off head: $H_{shut\text{-}off} = \dfrac{U_2^{\,2} - U_1^{\,2}}{2g}$ (34)

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

So, shock losses equation is formed by substituting in equation 16.

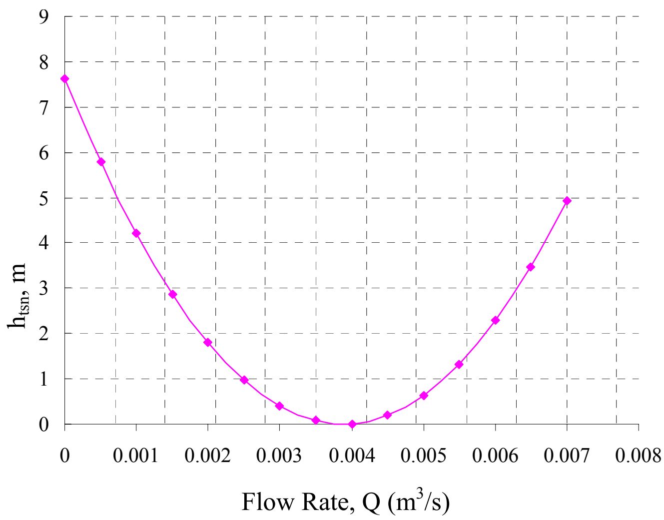

Fig. 9 Shock Losses versus Flow Rate Graph

Fig. 7 is the flow rate versus the shock loss of head. The shock loss of head increases when the flow rate decreases. Shock loss does not have in the design point condition. If this condition is over, the shock loss of head is high.

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.

The impeller friction losses are:

$$h_1 = \frac{b_2(D_2 - D_1)(V_{r1} + V_{r2})^2}{2\sin\beta_2 H_r 4g} \tag{35}$$

The hydraulic radius: $H_r = \dfrac{b_2\left(\dfrac{\pi D_2}{Z}\right)\sin\beta_2}{b_2 + \left(\dfrac{\pi D_2}{Z}\right)\sin\beta_2}$ (36)

The inlet relative velocity is;

$$V_{r1} = \frac{V_{m1}}{\sin\beta_1} \tag{37}$$

The outlet relative velocity: $V_{r2} = \dfrac{V_{m2}}{\sin\beta_2}$ (38)

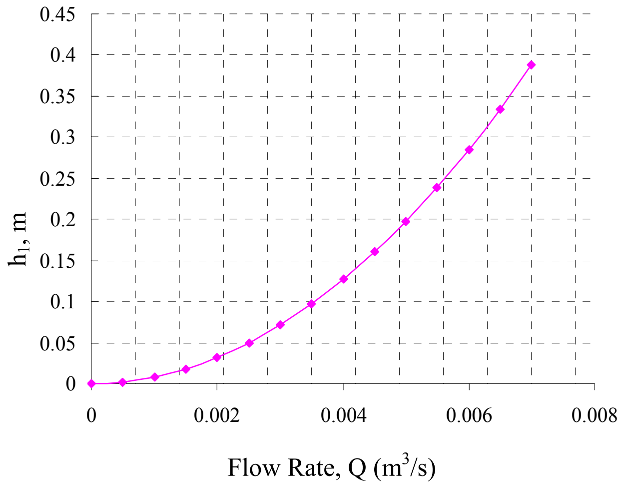

Fig. 10 Impeller Friction Losses versus Flow Rater Graph

The influence of the geometry of the impeller friction loss is obtained in Fig. 8. The analysis of the curves shows that small differences between the points for the flow rate versus the impeller friction loss of head. The impeller loss of head increases when the flow rate is increase.

E. 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 velocity approaching 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 losses:

$$h_2 = \frac{C_v V_3^{\,2}}{2g} \tag{39}$$

The volute throat velocity:

$$V_3 = \frac{Q}{A_3} \tag{40}$$

The volute throat area: $A_3 = V_{u2}(\frac{D_2}{D_3})$ (41)

The volute flow coefficient is,

$$C_v = 1 + (0.02 \times \frac{L_{vm}}{D_{vm}}) \tag{42}$$

The volute circumferential length: $L_{vm} = \frac{\pi D_i}{8}$ (43)

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

The volute mean diameter: $D_{vm} = \frac{D_i}{8}$ (44)

So, the results of these calculations are shown in Fig. 11 by substituting in equation.

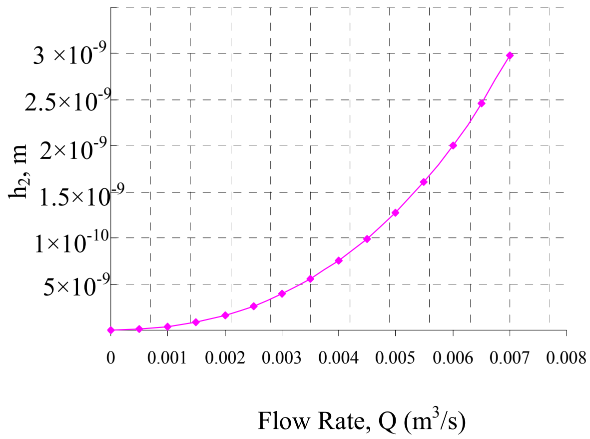

Fig. 11 Volute Friction Losses versus Flow Rate Graph

The volute losses versus flow rate graph are Fig. 11. The volute friction loss of head increases when the flow rate is increased. The volute friction coefficient decreases for small values of the volute flow coefficient

F. Disk Friction Losses

The impeller was designed to investigate the effect of disk friction on total power. The disk friction increases proportionally to the fifth power of disk diameter In order to examine the relation between the height of disk friction losses and the geometry of disks in real centrifugal pump housing disks without and with modified outlet sections with various numbers, angles and widths are investigated. Disks with modified outlet sections were examined to approach a real impeller in real centrifugal pump housing.

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;

$$h_3 = \frac{f\,\rho\,\omega^3 (\frac{D}{2})^5}{10^9 Q_s} \tag{45}$$

Loss coefficient of disk friction, f is assumed as 0.005. Substituting in equation, the result of the disk friction loss of head is shown by Table VIII and Fig. 12.

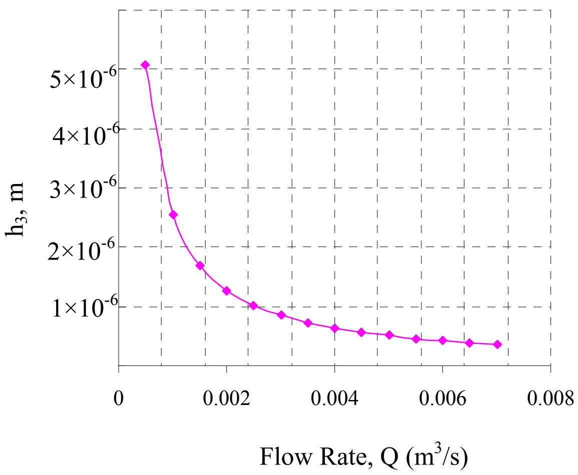

Fig. 12 Disk Friction Losses versus Flow Rate Graph

The disk friction coefficient increases with the increase the angle of the modified outlet sections of disks. The influence of geometrical parameters of disks means that affect the angular velocity of the developed. When the low flow condition changes the other condition, the disk friction loss of head is immediately high which is shown in Fig. If the flow accelerates, the loss of head changes about the normal rate.

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 head of recirculation is;

$$h_4 = K\ \omega^3 D_1{}^2 (1 - \frac{Q_s}{Q_0})^{2.5} \tag{46}$$

Impellers with relatively large inlet diameters usually encountered in high-specific speed pumps are the most likely to recirculate. The loss contains a default value of 0.005 for the loss coefficient.

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

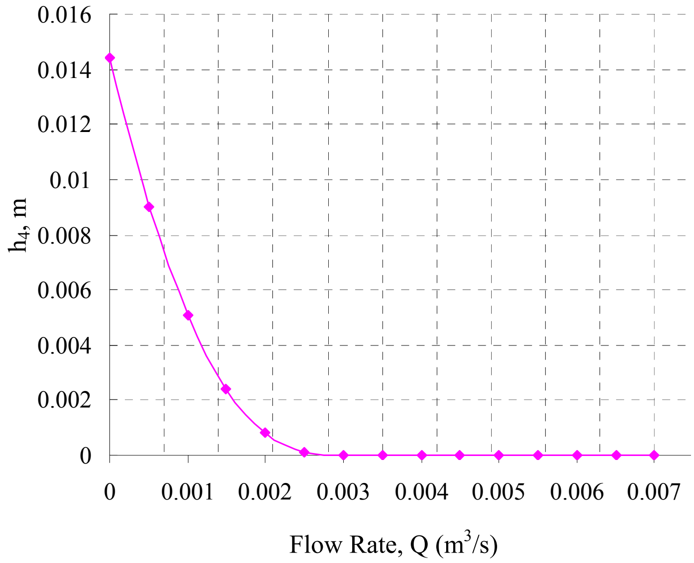

Fig. 13 Recirculation Losses versus Flow Rate Graph

The recirculation loss of head is high when the flow rate decreases. This graph is shown in Fig. 13. If the flow rate is high, the recirculation loss of head is nearly zero.

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. Therefore, the actual pump head is;

$$H_{act} = H_{thn} - (h_s + h_1 + h_2 + h_3 + h_4) \tag{47}$$

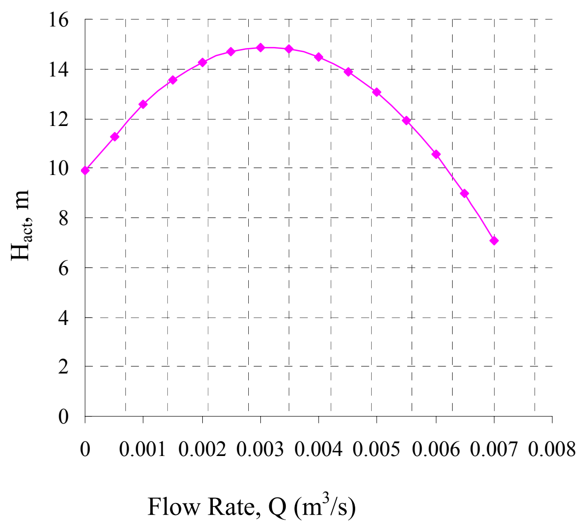

Fig. 14 Actual Pump Head Losses versus Flow Rate Graph

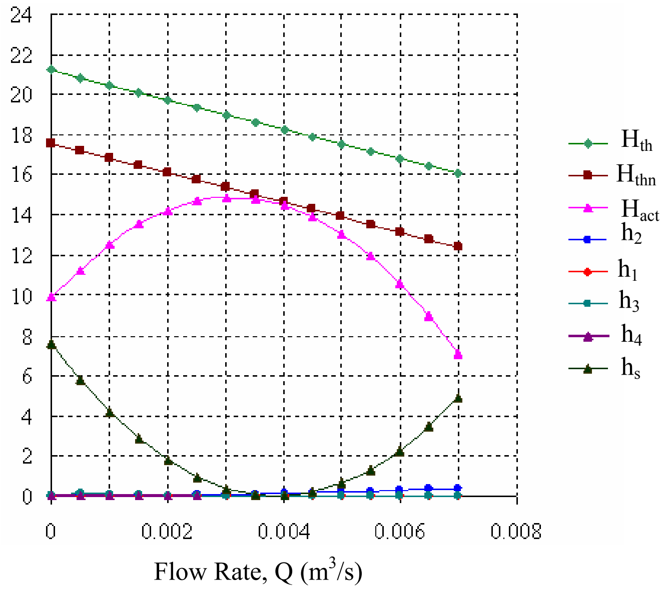

Fig. 15 Prediction of Characteristic Curve of Centrifugal Pump

IV. CONCLUSION

We show some looses of centrifugal pump with the values Q and H are determined for the various operating points. 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.

In a today competitive and sophisticated technology, centrifugal pump is more widely used than any other applications because the advantages of following factors are effect on the centrifugal pump.

1. Its initial cost is low
2. Efficiency is high
3. Discharge is uniform and continuous flow
4. Installation and maintenance is easy.
5. It can run at high speeds without the risk of separation of flow.

The performance analysis of centrifugal pump is also predicted in this paper. The impeller friction losses, volute friction losses and disk friction losses are considered to less the friction effect on centrifugal pump. Moreover, recirculation losses are also considered. And then, the actual performance curve of centrifugal pump is predicted obtained.

REFERENCES

[1] Larry Bachus., and Angel Custodio. 2003. Known and Understand Centrifugal Pump.Japan: Bachus Company. Tokyo 113.

[2] Igor, J., Joseph, P., and Charles, C. 2001. Pump Hand Book.USA: McGraw-Hill Company.

[3] Khin Maung Aye, U. December. 2000. Fluid Machinery for Mechanical Engineering.

[4] Kyshu Institute of Technology. 1996. Fluid Mechanics of Turbomachinery. Training Course.Japan: Kyshu Institute of Technology.

[5] Ing.F-K.Berna, Dr. No Date. Measurement of the Characteristic of Centrifugal Pump. Faculty of Engineering Sciences, University Duisburg-Essen.

[6] Tuzson, J. 2000. Centrifugal Pump Design. USA: John Wiley and Sons. Inc.

[7] Stepanoff, A.J, 1957, Centrifugal and Axial Flow Pumps.

[8] Austin. H. Church. 1972. Centrifugal Pump and Blowers. New York: John Wiley and Sons, Inc.

[9] William. L. Amminger., and Howard. M. Bernbaum.1973. “Centrifugal Pump Performance Prediction”.

[10] Herbert Addison. 1995. Centrifugal and Other Rotodynamic Pumps. Chapan and Hall. Ltd.

[11] Nemdili, A., and Hellmann, H. 2007. “Investigations on Fluid Friction of Rotational Disk in Real Centrifugal Pump”.

[12] Sam Yedidiah, No Date. Centrifugal Pump User’s Guide Book. an International Thomson Publishing.

[13] Hiroshi Satoh, No Date. Designing an Ultra- Low Specific Speed Pump. Sanso Electric co. Ltd.

[14] Daugherty, L. 1915. Centrifugal Pumps. McGraw-Hill Book Company. Fnc.

[15] Frank, A. 1953. Pumps. 2nd Edition. Mc Graw-Hill.

[16] Karassik., and Roy Carter, J. 1960. Centrifugal Pump Selection, Operation and Maintenance. McGraw-Hill Book Company.