Performance Prediction of Centrifugal Pump

DR. HTAY HTAY WIN [1], DR. THAN THAN HTIKE [2], MA AYE AYE MYO [3]
[1, 2, 3] *Department of Mechanical Engineering, Mandalay Technological University*

Abstract- This paper presents performance analysis of double-suction centrifugal pump. It is commercial and the most useful mechanical rotodynamics machine in fluid works which is used in domestic, irrigation, industry, large plants and river water pumping system in Myanmar. Moreover, centrifugal pumps are produced by manufacturing processes in Myanmar. In this research, the pump is driven by a 334 kW electric motor and its speed is 1450 rpm. The design data are taken from the Twingyi Pump Irrigation project, Myaung Township, Sagaing Region with the nominal flow of 25 m3/min, 45m of head and backward curved blades impeller. Moreover the losses which can appear in it are also found out to predict the operating point. Moreover the losses which can appear in it are also found out to predict the operating point. So, shock losses, impeller friction losses, volute friction losses, disk friction losses and recirculation losses of centrifugal pump are also considered in performance

I. INTRODUCTION

Centrifugal pumps are widely used in many places such as irrigation, water supply plants, oil refineries, chemical plants, steel mills, food processing factories and various industries. Double-suction centrifugal pump is widely used in river pumping project in Myanmar. In this thesis, impeller and casing are calculated. Before impeller design and casing design have been calculated, flow rate and head must be considered. Required data is attained in Myuinkinn river pumping project, Magway.

Pumping system is developed from the primitive pumping devices operated either by man or animal to the positive displacement and dynamic pumps that are fashioned today. Typical progress in the development of pumps is found in municipal water works, power plants, agriculture, transport and many other utility service and industries.

A pump has been defined differently by different investigators; the different definitions are however, all similar and nearly equivalent.(i)A device which raises or transfers liquids at the expense of power input. (ii)A machine designed to elevate, deliver and move various liquid.(iii)A unit that transfers the mechanical energy of a motor or engine into potential and kinetic energy of liquid. Figure 1 illustrate a centrifugal pump.

The performance of centrifugal pump is described by a graph plotting the pressure 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, a power input 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.

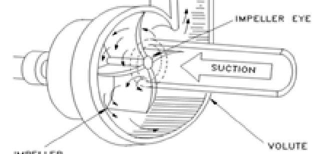

Figure 1. Centrifugal Pump

II. ENERGY LOSSES IN CENTRIFUGAL PUMP

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 predicted by the Euler Equation. The energy transfer quantities the losses between fluid power and mechanical power of the impeller or runner. The fluid passes through the blade passages receives energy from the moving blades or imparts energy to them. The pump increases the energy of the liquid which may be used to lift the liquid and to overcome the hydraulic resistance of the delivery pipe. Thus, centrifugal pumps may be taken losses of energy. The hydraulic losses can be occurred within the pump consist of the following factors (a)Shock or eddy losses at the entrance and the exit from the impeller.(b)Friction losses in the impeller(c) Friction losses in the casing or guide vane.

a) Shock Losses or eddy losses

Shock losses occur at the entry to the impeller. Fluid flow in a pump tends to avoid shock by acquiring pre-rotation at the impeller inlet and a velocity gradient in the volute casing at the impeller discharge. The nature of the hydraulic loss at the impeller entrance, when liquid approaches at a high angle of attack, is that due to a sudden expansion or diffusion after separation. At the impeller discharge, the loss is mostly caused by a high rate of shear due to a low average velocity in the volute and high velocity at the impeller discharge. Shock loss is proportional to the tangential velocity.

The shock loss is;

$$h_s = k\left(Q_s^2 - Q_N^2\right) \tag{1}$$

Where Q_N is the volumetric flow rate corresponding to the maximum efficiency.

b) Impeller Friction Losses

Hydraulic pump design must consider all hydraulic forces on the impeller. Fluid friction force on the impeller, volute, disk friction, absorbs shaft power and affects the overall efficiency of the pump. Friction losses account for energy dissipation due to contact of the fluid with solid boundaries such as stationary vanes, impeller, casing etc. the wall friction or skin friction losses in the impeller and diffuser or volute follow the standard model. Since the flow passage cross sections are irregular, a hydraulic radius and average flow velocities are used.

The geometrical measurements are taken at the impeller exit, blade height, the passage width, and the circumference. They are divided by the number of blades andmultiplied by the sine of the blade angle at the exit.

The hydraulic radius is;

$$H_r = \frac{b_2\left(\frac{\pi D_2}{Z}\right)\sin\beta_2}{b_2 + \left(\frac{\pi D_2}{Z}\right)\sin\beta_2} \tag{2}$$

Where H_r is the hydraulic radius of the impeller blade passage. It is the ratio passage cross section area is divided by the half of the circumference.

The impeller head loss is;

$$h_1 = \frac{b_2\left(D_2 - D_1\right)\left(V_{r1} + V_{r2}\right)^2}{2\sin\beta_2 H_r 4g} \tag{3}$$

c) 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 lost. 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 velocity approaching the volute throat is;

$$V_3 = \frac{Q}{A_3} \tag{4}$$

The volute cross-sectional area is;

$$A_3 = V_{u2}\left(\frac{D_3}{D_2}\right) \tag{5}$$

Volute Flow Coefficient;

$$C_v = 1 + \left(0.02 \times \frac{L_{vm}}{D_{vm}}\right) \tag{6}$$

Where L_{vm} ~ volute mean circumferential length

D_{vm} ~ volute mean diameter

Volute mean circumferential length is calculated by

$$L_{vm} = \frac{\pi R_i^2}{8} \quad (7)$$

Volute mean diameter;

$$D_{vm} = \frac{R_i^2}{8} \quad (8)$$

Volute friction loss of head,

$$h_2 = \frac{C_v V_3^2}{2g} \quad (9)$$

d) Disk Friction Losses

The impellers were designed to investigate the effect of disk friction on total power. The disk friction increases proportionally to the fifth power of disk diameter and through the relation with head must also be taken into account as small as 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 [07Khi].

The disk friction loss,

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

e) 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 recirculationloss,

$$h_4 = K\omega^3 D_1^2 \left(1 - \frac{Q_s}{Q_0}\right)^{2.5} \quad (11)$$

III. PERFORMANCE OF CENTRIFUGAL PUMP

The performance of centrifugal pump is described by a graph plotting the pressure 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, a power input 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.

- Theoretical Head

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

$$H_{th} = \frac{1}{g} U_2 V_{u2} \quad (12)$$

- 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 [07Khi].

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

$$\sigma = 1 - \frac{(\sin \beta_2)^{1/2}}{Z^{0.7}} \quad (13)$$

The whirl velocity at the outlet,

$$V_{u2} = U_2 \sigma - V_{\omega 2} \cos \beta_2 \quad (14)$$

The net theoretical head,

$$H_{thn} = \frac{U_2 V_{u2}}{g} \tag{15}$$

- 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,

$$H_{act} = H_{thn} - (h_s + h_1 + h_2 + h_3 + h_4) \tag{16}$$

- Leakage Losses

Leakage loss is a loss of capacity through the running clearances between the rotating element and the stationary casing parts. Leakage can take place according to the type of pump. The leakage losses decline with decreased Reynolds number i.e. with increased viscosity. They are;

i. between the casing and the impeller at the impeller eye
ii. between the two adjacent stages in multi-stage pumps
iii. through the stuffing box
iv. through axial balancing devices
v. past vanes in open impeller pumps
vi. through bleed-off bushings when used to reduce the pressure on the stuffing boxat any bleed-off used for bearing and stuffing box cooling.

- Efficiencies

How small the losses or how good a machine is 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.

- Mechanical Efficiency

It is the ratio of the energy transferred to water to the mechanical energy delivered at the shaft coupling.

$$\eta_m = \frac{P_{motor}}{P} \tag{17}$$

- Volumetric Efficiency

The volumetric efficiency is due to the leakage loss which is a loss of capacity through the running clearances between the rotation elements and the stationary casing parts. The volumetric efficiency is defined as

$$\eta_v = \frac{Q}{(Q+q)} = \frac{1}{1+\frac{1.124}{n_s^{2/3}}} \tag{18}$$

- Hydraulic Efficiency

It is the ratio of the actual output power of the pump to the power supplied to the pump shaft minus power required to overcome disc friction and bearing friction.

$$\eta_h = \frac{H}{H_{th}} \tag{19}$$

- Overall Efficiency

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

$$\eta_0 = \frac{\rho g Q H}{P} \text{ (or) } \eta_0 = \eta_m \times \eta_v \times \eta_h \tag{20}$$

IV. CALCULATION OF PERFORMANCE OF CENTRIFUGAL PUMP

Table 1 shows specification data of single stage double-suction centrifugal pump.

Table 1. Specification data

Parameter	Value	Unit
Head, H	45	m
Discharge, Q	25	m^3/min
Rotational speed, n	1450	rpm
Density of water, ρ	1000	Kg/m^3
Acceleration due to gravity	9.81	m/s^2

The result data are shown in Table 2.

Table 2 Result data

Input Power	238.89	kW
Motor power	334.46	kW
Volumetric efficiency	97.5	%

Theoretical Head

Outlet tangential velocity

$$U_2 = \frac{\pi D_2 n}{60} = 35.77 \text{ m/s}$$

Assume the outlet contraction factor, $\varepsilon_2 = 0.95$

Outlet flow velocity

$$V_{m2} = \frac{Q_s}{\pi D_2 b_2 \varepsilon_2} = 8.804 Q_s$$

The whirl velocity

$$V_{u2} = U_2 - V_{m2} \cot \beta_2 = 35.77 - 8.804 Q_s \cot 20$$

$= 35.77\text{-}24.18Q_s$ m/s

TheoreticalHead,

$$H_{th} = \frac{1}{9.81}(35.77)(35.77 - 24.18 Q_s)$$

$= 130.427\text{-}88.2Q_s$

Table 3 Relation between Theoretical Head and Flow Rate

Sr.No	Q_s (m^3/s)	H_{th}(m)
1	0.1	121.61
2	0.125	119.41
3	0.15	117.20
4	0.175	114.99
5	0.2	112.79
6	0.225	110.59
7	0.25	108.38
8	0.275	106.18
9	0.3	103.98
10	0.325	101.77
11	0.35	99.57
12	0.375	97.36
13	0.4	95.16
14	0.425	92.95

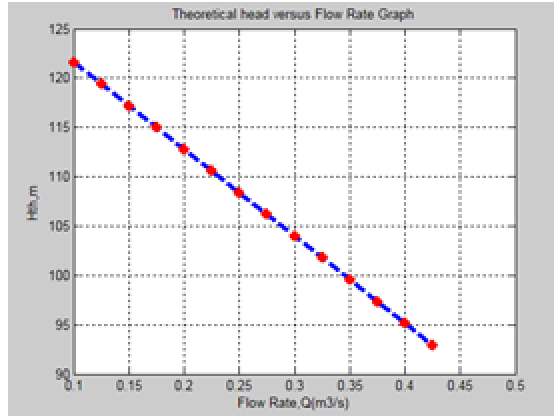

Figure 2 Theoretical head versus Flow rate Graph

Figure 2 illustrates increasing the flow rate causes the decreasing of the theoretical head.

Net Theoretical Head

The slip factor value is obtained by using equation 13.

$$\sigma = 1 - \frac{(\sin \beta_2)^{1/2}}{Z^{0.7}} = 0.8$$

The whirl velocity at outlet,

$$V_{u2} = U_2 - V_{m2} \cot \beta_2 = 35.77 - 8.804 Q_s \cot 20$$

$= 30.76\text{-}24.18Q_s$

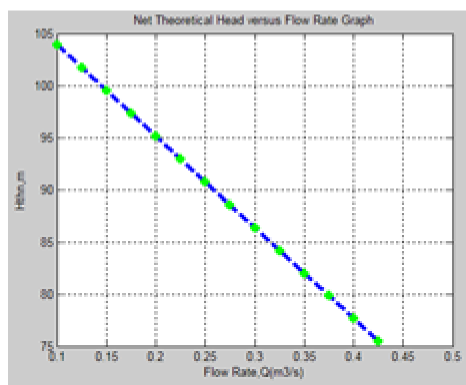

Figure 3 .Net Theoretical Head versus Flow rate Graph

Figure 3 illustrates the greater the value of flow rate the smaller the value of net head.

Shock Losses

$$h_s = k\left(Q_s^2 - Q_N^2\right)$$

Where $Q_s = 0$, $h_s = H_{shut\text{-}off}$, $k = 301.74$

$$h_s = 301.74(Q_s - 0.385)^2$$

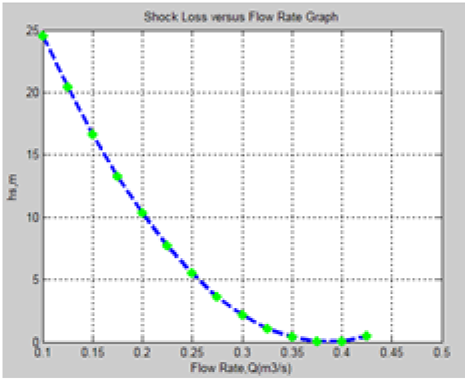

Figure 4. Shock Loss versus Flow Rate

Figure 4 illustrates shock loss versus flow rates. The shock loss of head increases when the flow rate decreases. If this condition is over, the shock loss of head is high.

(a) Impeller Friction Losses

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

Substitute the necessary values in the above equation

$$h_1 = 204.182 Q_s^2$$

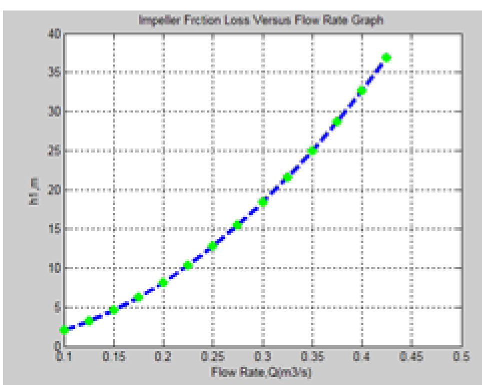

Figure 5. Impeller Friction Loss versus Flow Rate

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.

Volute Friction Losses

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

Substitute the necessary values in the above equation

$$h_2 = \frac{1.063 \left(\frac{Q_s}{30.86 - 20.866 Q_s} \right)^2}{2 \times 9.81}$$

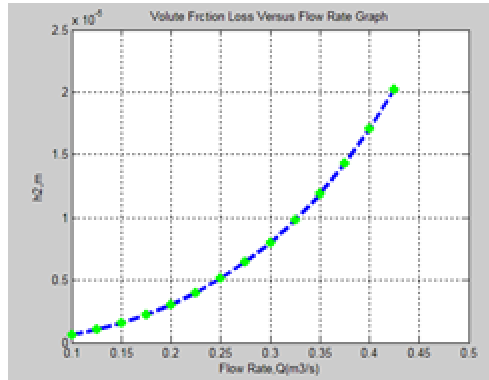

Figure 6. Volute Friction Losses Relative to Flow Rate

The volute losses versus flow rate graph in Figure 6. The volute friction loss of head increases when the flow rate is increases. The volute friction coefficient decreases for small values of the volute flow coefficient.

Disk Friction Losses

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

Substitute the necessary values in the above equation

$$h_3 = \frac{1.271 \times 10^{-5}}{Q_s}$$

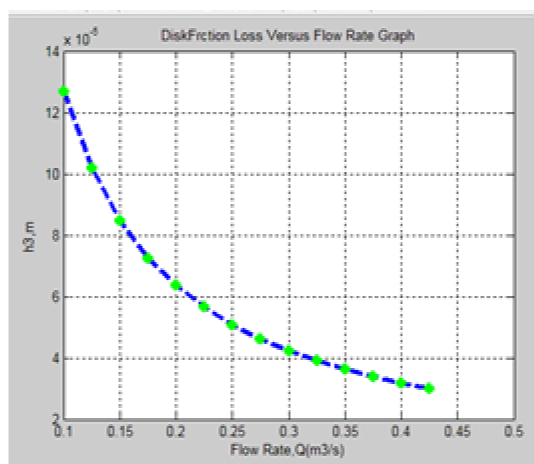

Figure 7 Disk Friction Loss versus Flow Rate

When the low flow condition changes the other condition, the disk friction loss of head is immediately high which is shown in Figure 7. If the flow accelerates, the loss of head changes about the normal rate.

Recirculation Losses

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

Substitute the necessary values in the above equation

$$h_4 = 13.011\left(1-\frac{Q_s}{0.4167}\right)^{2.5}$$

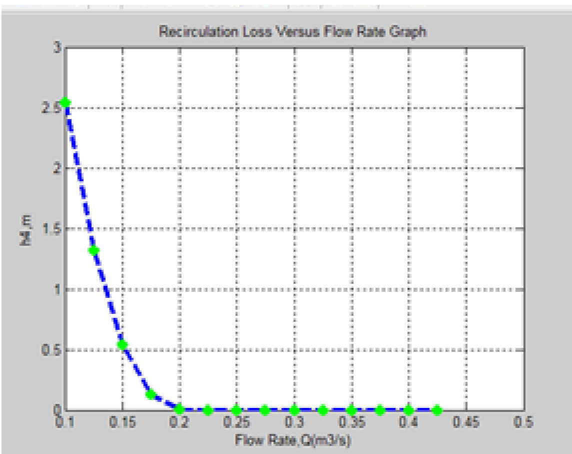

Figure 8. Recirculation Losses versus Flow Rate

Actual Head

The actual pump head,

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

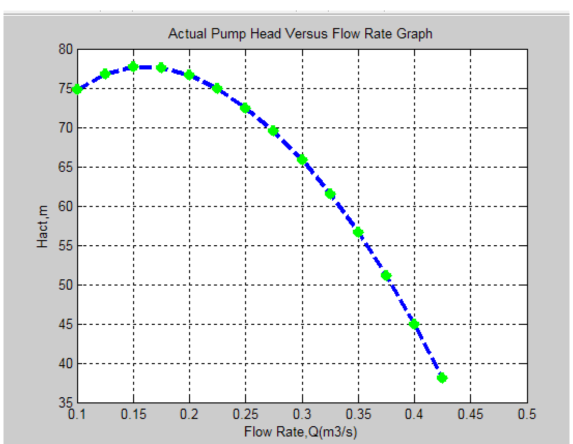

Figure 9. Actual Pump Head versus Flow Rate Graph

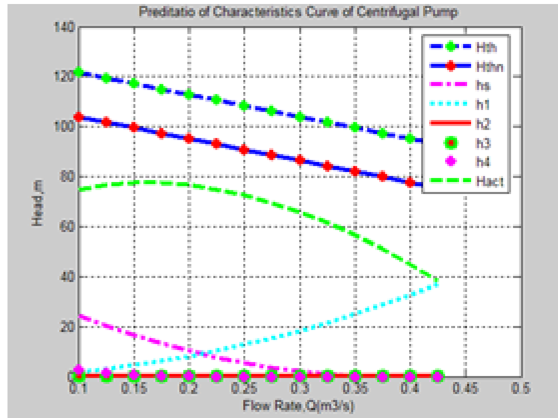

Figure 10. Head versus Flow Rate Graph

V. DISCUSSION AND DISCUSSION

The aim of this research is to use in river pumping project which has about twelve working hours per day and require high head and capacity. So double - suction centrifugal pump is selected. The casing is horizontal split casing type. The design head is 45 m and the discharge is 0.4167 m3/s. The pump can runat 1450 rpm to attain higher head.

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 transfer quantities are losses between fluid power and mechanical power of the impeller or runner. This paper focus on losses of centrifugal pump. There are many losses such as (a) shock Losses or eddy losses (b) impeller friction losses (c) volute friction losses (d) disk friction losses (e) recirculation losses.

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. The impeller friction losses, volute friction losses, disk friction losses are less than shock losses.

Recirculation losses are also considered. 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. The performance of centrifugal pump is more effected by shock losses.

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