Design and Performance Analysis of Double-Suction Centrifugal Pump

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Abstract- This paper presents the detail design of impeller and volute casing for double-suction centrifugal pump. The designed pump can develop a head of 150 ft (46 m) and deliver 335 ft^3/min or 2500 gal/min (9.48 m^3/min) of clean and cold water at the speed of 1760 rpm. The designed impeller has 7.32 in (19 cm) inlet diameter, 13.5 in (34 cm) outlet diameter, 13˚ inlet vane angle and 20˚ outlet vane angle. The number of vanes is 7. The inlet width and outlet width are 1.75 in (4 cm) and 1.98 in (5 cm) respectively. The discharge diameter is 6 in (15 cm) to operate the designed head and capacity. Moreover, the performance analysis of designed pump is also presented by considering on the various losses. The predicted maximum efficiency takes place at nearly designed head and capacity. The maximum efficiency is nearly 80.14% and the expected designed efficiency of designed pump is 80%. So, predicted and designed efficiencies are not large difference and the designed efficiency has a satisfactory value. The designed double-suction centrifugal pump can fulfil the requirements of agricultural process.

Index Terms- Double-suction centrifugal pump, head, flow rate, speed, performance characteristics**.**

I. INTRODUCTION

Pumping may be defined as the addition of energy to a fluid to move it from one point to another or to raise it to the required height. The energy given to the pump case forces the fluid to do work flowing through the pipes rising to the higher level. The input power of the pump is mechanical energy of the drive shaft driven the prime mover such as electric motor or small engine and the output energy is the hydraulic. In industries, throughout the world, pumps play in a major role. 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 applications such as chemical plants, firefighting, hydraulic system, and so on.

II. COMPONENTS AND OPERATION PRINCIPLE OF CENTRIFUGAL PUMP

The two main components of centrifugal pump are impeller and casing. The other components are suction nozzle, discharge nozzle, shaft, bearing, wear rings, stuffing box and mechanical seal. The centrifugal pump moves liquid by rotating one or more impellers inside a volute casing. The liquid is introduced through the casing inlet to the eye of the impeller where it is picked up by the impeller vanes. The rotation of the impeller at high speeds creates the centrifugal force that throws the liquid along the vanes, causing it to be discharged from its outside diameter at a higher velocity. This velocity energy is converted to pressure energy by the volute casing prior to discharging the liquid to the system.

In double-suction centrifugal pumps, they are usually large and are used in water service. A double-suction impeller is the same in effect as two single-suction impellers placed back to back on a horizontal shaft, supported by beings on either side. This construction has the effect of increasing the capacity without increasing the diameter of impeller. This type allows liquid to enter the eye of impeller from both sides. This action can be symmetrical about the centerline of double-suction impeller. The symmetry of impeller is significantly improving its hydraulic balance or does not exist axial thrust force. A central scroll serves both impellers and leads, through a single diffuser, to an exit flange. Such an arrangement often results in better efficiency because it reduces friction on the back side of the impellers, the disk friction loss, and because by splitting the flower in two the specific speed of each impeller sometimes becomes more favorable. As a result, double-suction volute pumps can produce higher pressure than single-suction centrifugal pumps.

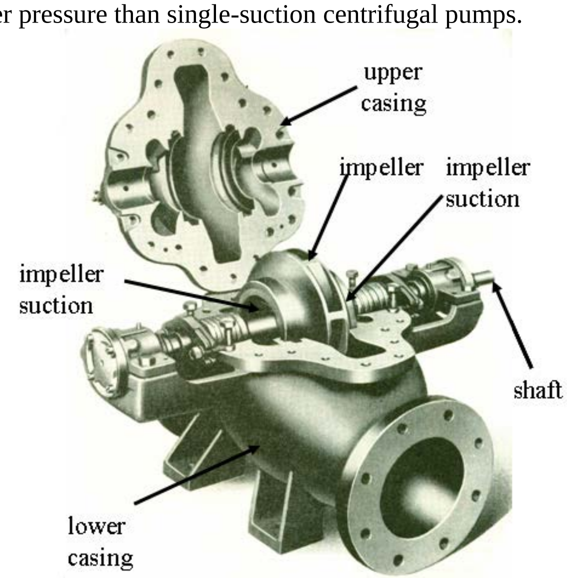

Fig. 1 Main Components of Double-suction Centrifugal Pump [1]

III. DESIGN OF CENTRIFUGAL PUMP'S IMPELLER

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 centrifugal pump will be designed to develop a head of 150 ft and deliver 335 ft^3/min or 2500 gpm of water. The pump type to be designed is double-suction type. It is to be direct-connected to a motor operating at 1760 rpm.

A. Selection of Specific Speed and Specification of Suitable Pump Type

Firstly, a specific speed (n_s) must be selected for the best efficiency point. Moreover, it is also used to classify the type of impellers on their performance, and proportion regardless of their actual size or the speed at which they operate. It is mathematically expressed as

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

When applied to double-suction impellers, it is necessary to consider such impellers as being equivalent to two single-suction impellers placed back to back or operating in parallel. This means that in applying to double-suction impellers the capacity used should be one-half of that handled by the pump. In this design, calculated value of specific speed (fps) based on required head and capacity is 1450 rpm and it is within the range of low specific speed pump that is greater than 500 and less than 1500 [2]. So, double-suction centrifugal pump typed is chosen in this study.

B. Determination of Shaft and Impeller Hub Diameters

Before the impeller dimensions can be fixed, the shaft size must first be approximated. It should be large enough to care for the torque and bending moment, to avoid excessive lateral deflection, and to keep the critical speed a safe distance from the operation speed. The shaft diameter is depending upon the torque alone and the torque is depending on the brake horsepower (*b.hp*) of designed pump. The brake horsepower is

$$b.hp = \frac{W.hp}{\eta} \tag{2}$$

Where, W.hp and η are fluid horse power of pump expected overall efficiency of designed pump respectively. This efficiency can be estimated by Fig. A and the estimated value is approximately and nearly 80%.

The shaft torque is

$$T = \frac{63000\ b.hp}{n} \tag{3}$$

The shaft diameter at hub section of impeller is

$$d_s = \sqrt[3]{\frac{16\ T}{\pi\ \tau}} \tag{4}$$

Allowable shear stress of material of shaft with key way, τ is 4500 psi because the main shaft is made of mild steel. The estimated shaft diameter will be increased 20% because it is difficult to predict the bending moment at this time.

The hub diameter, D_h is usually made 15%greater than shaft diameter. Moreover, the hub length, L_h is also from 1.0 times to 2.0 times of the shaft diameter [3].

C. Impeller Inlet and Outlet Velocities

The simplified inlet and outlet velocities diagrams for the impeller are shown in Fig. 2. In this figure, the effect of circulatory flow on the outlet diagram is shown in solid lines and the virtual diagram is dotted. 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.

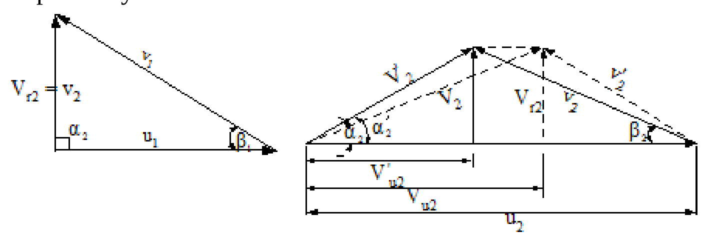

Fig. 2 Impeller Inlet and Outlet Velocity Diagrams

D. Determination of Impeller Inlet Dimensions and Vane Angle

A double-suction pump has a low percentage of leakage since the volume handled is relatively large compared to the leakage area. In this instance it will probably not exceed 2% [1]. The impeller is double-suction, so the total flow is divided by 2 and the approximate of impeller eye diameter will be.

$$D_0 = \sqrt{\frac{4}{\pi}\ \frac{1.02\ \times 144 \times Q}{2\ V_0} + D_H^2} \tag{5}$$

The inlet velocity through the eye of the impeller V_0 is usually slightly higher than the velocity in the suction flange, say 10 to 15 ft/sec average. So, the assumed value of V_0 is 11 ft/sec.

To insure smooth flow without excessive turbulence, the inlet vane edge diameter D_1 is usually made about the same as the eye diameter D_0. Thus, tangential velocity at impeller inlet, u_1 can be expressed as

$$u_1 = \frac{\pi D_1 n}{12 \times 60} \tag{6}$$

The impeller inlet passage width for per side is

$$b_1 = \frac{144\ Q}{\pi D_1 V_{r1} \varepsilon_1} \tag{7}$$

Since slowing up a fluid is always more inefficient than speeding it up, the radial inlet velocity at the vane inlet, V_{r1} is usually made 5% to 10% greater than V_0. So, assumed value of V_{r1} is 12 ft/sec. Moreover, the inlet area will be increased by the vane thickness. Hence, a contraction factor ε_1 which is generally between 0.8 and 0.9 is assumed to obtain the approximate inlet width b_1. So, assume the inlet contraction factor ε_1 as 0.85.

The water is usually assumed to enter the vanes radially, so that the absolute approach angle α_1 is 90°. And then, the vane inlet angle β_1 is found from

$$\tan\ \beta_1 = \frac{V_{r1}}{u_1} \tag{8}$$

The inlet relative velocity can also be expressed by

$$v_1 = \frac{u_1}{\cos\beta_1} \tag{9}$$

E. Determination of Impeller Inlet Dimensions and Vane Angle

The outlet diameter of impeller can be more easily obtained by means of the overall head coefficient Φ. The value of Φ varies between 0.9 and 1.2 with an average value very close to unity and this value is approximately taken as 1.05 in this study. Thus, the required outside impeller diameter is

$$D_2 = \frac{1480\ \Phi\sqrt{H}}{n} \tag{10}$$

And then, the peripheral velocity at impeller outlet is

$$u_2 = \frac{\pi D_2 n}{12 \times 60} \tag{11}$$

The vane outlet angle β_2 is usually made larger than the inlet angle β_1 to obtain a smooth, continuous passage. The amount of outlet angle β_2 usually has between 15° and 40°. So, the vane outlet angle is assumed that β_2 = 20° in this study.

The radial outlet velocity V_{r2} is made the same as, or slightly less (up to 15%) than the radial inlet velocity V_{r1} to avoid any sudden changes of velocity. So, the radial outlet velocity V_{r2} is assumed that is 11 ft/sec.

To care for the vane thickness, a contraction factor ε_2 which is generally between 0.90 and 0.95 must be assumed in determining the gross outlet area and width b_2. If this is tentatively taken to be 0.925, the approximate outlet width is

$$b_2 = \frac{144\ Q}{\pi D_2 V_{r2} \varepsilon_2} \tag{12}$$

The virtual tangential component V_{u2} of V_2 is

$$V_{u2} = u_2 - \frac{V_{r2}}{\tan\beta_2} \tag{13}$$

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

$$V'_{u2} = V_{u2}\eta_\infty \tag{14}$$

Thus, the actual outlet is found by

$$\tan\ \alpha'_2 = \frac{V_{r2}}{V'_{u2}} \tag{15}$$

To design correctly the volute or diffuser, the magnitude and direction of the absolute outlet velocity V_2 of the liquid leaving the impeller must be known. The absolute outlet velocity from outlet velocity diagram is

$$V'_2 = \sqrt{V_{r2}^2 + V'^2_{u2}} \tag{16}$$

And then, the water velocity at the impeller outlet is

$$v_2 = \frac{V_{r2}}{\sin\beta_2} \tag{17}$$

F. Impeller Blade Shape and Number of Blades

A method of drawing the impeller blade by three circular arcs is used for this present design. Each radius is given by

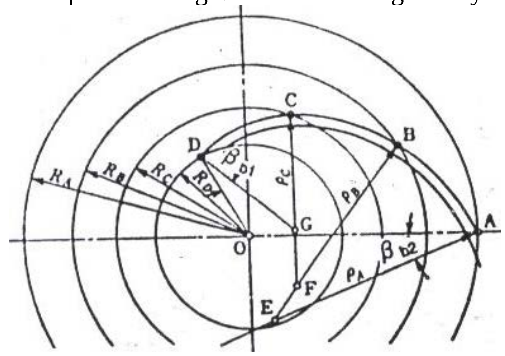

Fig. 3 Curvature of Impeller Blade [3]

$$\rho_A = \frac{\left(R_A^2 - R_B^2\right)}{2\left(R_A \cos\beta_2 - R_B \cos\beta_B\right)} \tag{18}$$

$$\rho_B = \frac{\left(R_B^2 - R_C^2\right)}{2\left(R_B \cos\beta_B - R_C \cos\beta_C\right)} \tag{19}$$

$$\rho_C = \frac{\left(R_C^2 - R_D^2\right)}{2\left(R_C \cos\beta_C - R_D \cos\beta_1\right)} \tag{20}$$

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

$$R_B = R_A - \frac{R_A - R_D}{3} \tag{21}$$

$$R_C = R_B - \frac{R_A - R_D}{3} \tag{22}$$

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

The vane thickness is scaled from the vane curvature drawing. In this design, the vane thickness at inlet width is 0.125 in (3 mm) and the thickness at outlet width is 0.275 in (7 mm). Moreover, the shroud thickness is taken as 0.19 in (5 mm).

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

$$Z \approx 6.5\frac{D_2 + D_1}{D_2 D_1}\sin\left[\frac{\beta_1 + \beta_2}{2}\right] \tag{23}$$

IV. DESIGN OF PUMP'S VOLUTE CASING

A volute to fit the impeller designed is used and the purpose of the volute is to convert the velocity head of water leaving the impeller as efficiently as possible. 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. The basic shaped of the cross section will be trapezoid with walls at a 30° angle with radial lines (θ = 60°).

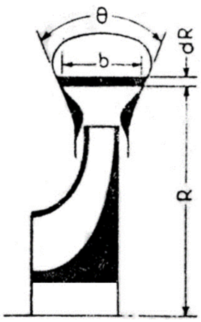

Fig. 4 Section through Volute Casing [1]

For pumps of medium specific speeds, $b_3 = 1.75\ b_2$. For small pumps of lower specific speed (b_2 is small), including multistage pumps, $b_3 = 2.0\ b_2$. For high specific speed pumps ($n_s > 3000$ double-suction) b_3 can be reduced to $b_3 = 1.6\ b_2$. Where b_2 is impeller width at discharge and b_3 is base width at D_2 [4]. Thus, the base width in this study is taken as

$$b_3 = 1.75 \times b_2 \tag{24}$$

The width of the volute at any point is calculated from

$$b = b_3 + 2x \times \tan\left(\frac{\theta}{2}\right) \tag{25}$$

Where, x is the distance between any radius R and impeller outside radius R_2. The volute is designed by determining the angle Φ^o measured from and assumed radial line by tabular integration of Equation (26).

$$\Phi^{o} = \frac{360\ R_2 V'_{u2}}{Q} \int_{R_2}^{R_\Phi} \frac{bdR}{R} = \frac{360 R_2 V'_{u2}}{Q} \sum_{R_2}^{R_\Phi} b \frac{\Delta R}{R} \tag{26}$$

The volute is considered to begin at the assumed base line, but it actually begins at the tongue radius R_t which 5% to 10% greater than the impeller radius R_2 to avoid turbulence and noisiness and to give the velocities of the water leaving the impeller a chance to equalize before coming into contact with the tongue [7].The tongue angle of volute casing is determined by

$$\Phi_t^{o} = \frac{132\ \log_{10}\ R_t/R_2}{\tan\ \alpha'_2} \tag{27}$$

To avoid shock losses the tongue should be made the same as the absolute outlet angle $\alpha_2´$ of the water leaving the impeller.

V. Results of Designed Turbine

The calculated results for both impeller and casing design of double-suction type centrifugal pump are clearly expressed in Table I. Moreover, detail drawings of impeller and volute casing designs are also shown in following figures.

Table I. Required Parameters for Runner Design

No	Descriptions	Symbols	Results
1	Input Power	P_{shaft}	175 hp
2	Shaft diameter	D_s	2.125 in
3	Hub diameter	D_H	2.5 in
4	Impeller eye diameter	D_o	7.3125 in
5	Impeller inlet diameter	D_1	7.3125 in
6	Impeller outlet diameter	D_2	13.5 in
7	Inlet angle of impeller blade	β_1	13°
8	Outlet angle of impeller blade	β_2	20°
9	Impeller passage width at inlet	b_1	1.75 in per side
10	Impeller passage width at outlet	b_2	1.98 in
11	Number of impeller blades	Z	7 blades
12	Base width of volute casing at D_2	b_3	3.465 in
13	Volute tongue angle	Φ°_t	17.5°
14	Largest volute diameter	D_v	21.3 in
15	Discharge nozzle diameter	D_d	6 in

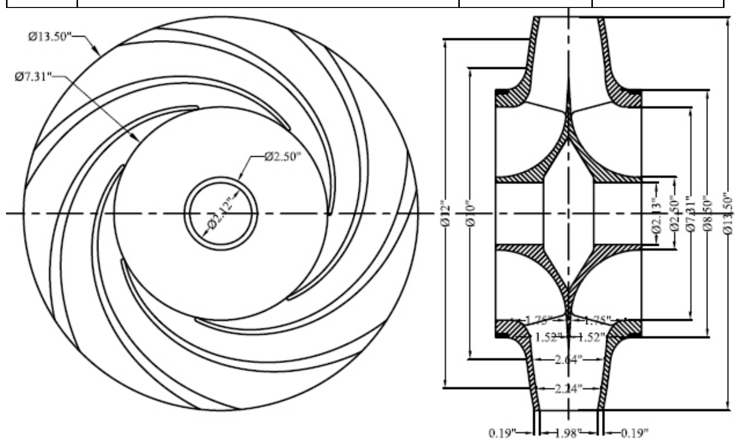

Fig. 5 Front View and Side View of Impeller

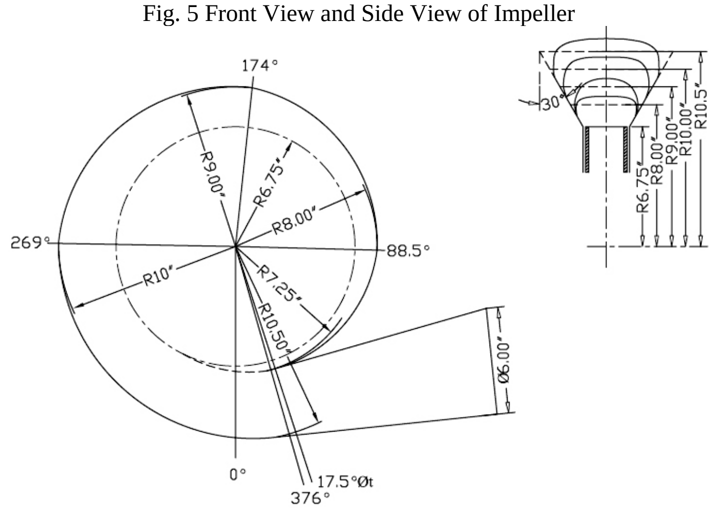

Fig. 6 Drawing of Volute Casing: Elevation of Volute Casing and Section through of Volute Casing

VI. Performance Analysis of Designed Pump

Performance needs to be known, not only at the rated, best efficiency point, but also off design. Pump specifications often impose special requirements, such as head at shut-off, maximum power demand, rate of head rise to assure stability, and so on. A good pump design process requires trail-and-error iteration, a check on anticipated performance with a trail geometry, and progressive approximation to the optimal design configuration.

A. Theoretical Head and Net Theoretical Head

Firstly, theoretical head is calculated by

$$H_{th} = \frac{1}{g}(u_2 V_{u2}) \tag{28}$$

Where,

Whirl velocity, $V_{u2} = u_2 - V_{r2} \cot \beta_2$

Flow velocity, $V_{r2} = \dfrac{Q_s}{\pi D_2 b_2 \varepsilon_2}$

The circulatory flow effect reduces the theoretical head developed in a practically constant ratio. Slip value for circulatory effect is

$$\sigma = 1 - \frac{(\sin \beta_2)^{1/2}}{Z^{0.70}} \tag{29}$$

By considering this effect, the whirl velocity and the net theoretical head are

$$V_{u2} = u_2 \sigma - V_{r2} \cot \beta_2 \tag{30}$$

$$H_{thn} = \frac{1}{g}(u_2 V_{u2}) \tag{31}$$

B. Shock Losses

Shock losses are considered as following expressing.

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

In the shut-off condition, $Q_s = 0$ and Q_N is design flow rate at maximum efficiency. Where, shut-off head is estimated by

$$H_{shut\text{-}off} = \frac{u_2^2 - u_1^2}{2g} \tag{33}$$

From the shut-off condition, the value of k can be calculated.

C. Friction Losses

The wall friction or skin friction losses, H_{f1} in the impeller follow the standard pipe friction model. Since the flow passage cross sections are irregular, a hydraulic radius and average flow velocities are used. The friction coefficient can be adjusted but has a default value of 0.05 in this study. The impeller friction losses are calculated by the following Equation (34).

$$H_{f1} = \frac{CF\,(D_2 - D_1)\,(v_2 + v_1)^2}{2\times(\sin \beta_2)\,H_{r1} \times 4g} \tag{34}$$

Where, H_{r1} refers to hydraulic radius and it is expressed by

$$H_{r1} = \frac{\pi b_2 D_2 \sin \beta_2}{Z\left(b_2 + \dfrac{\pi D_2 \sin \beta_2}{Z}\right)} \tag{35}$$

The volute friction losses can be found by

$$H_{f2} = \frac{CF\pi D_3 V_3^2}{2 \times 12\, g \sqrt{\dfrac{A_{th}}{\pi}}} \tag{36}$$

Where V_3 is the volute throat velocity and A_{th} is the volute throat area. Their relationship is as follow [5].

$$V_3 = \frac{Q_s}{A_{th}} \tag{37}$$

D. Diffusion Losses

A diffusion loss H_{df} needs to be taken into account, since separation invariably appears in the impeller at some point. When the ration of the relative velocity at the inlet v_1 and outlet v_2 exceeds a value of 1.4, it is assumed that a portion of the velocity head difference is lost. The diffusion loss is

$$H_{df} = \frac{0.25\, v_1^2}{2g} \tag{38}$$

E. Actual Head

Finally, the actual pump head is calculated by subtracting from the net theoretical head all the flow losses. Thus, the actual pump head is forecasted by the following relationship equation.

$$H = H_{thn} - (h_s + H_{f1} + H_{f2} + H_{df}) \tag{39}$$

F. Efficiency and Power

The overall efficiency can be predicted by the following relationship equation.

$$\eta = \eta_M \times \eta_{HY} \times \eta_V \tag{40}$$

The mechanical efficiency is

$$\eta_M = \frac{Output\ power}{Input\ Shaft\ Power} = \frac{\rho\, g\,(Q_s + q)\, H_{vir}}{P_{shaft}} \tag{41}$$

The hydraulic efficiency is

$$\eta_{HY} = \frac{actual\ measured\ head}{head\ imparted\ fluid\ by\ impeller} = \frac{H}{H_{thn}} \tag{42}$$

The volumetric efficiency is

$$\eta_V = \frac{delivered\ \ flow\ rate}{delivered\ flow + internal\ leakage\ flow} = \frac{Q_s}{Q_s + q} \tag{43}$$

By substituting these efficiencies into Equation (40), overall efficiency becomes

$$\eta = \frac{\rho g H Q_s}{P_{shaft}}$$

In this study, input shaft power is 175 hp.

VII. Performance Characteristics of Designed Centrifugal Pump

The performance characteristics of designed double-suction centrifugal pump are shown in following figures.

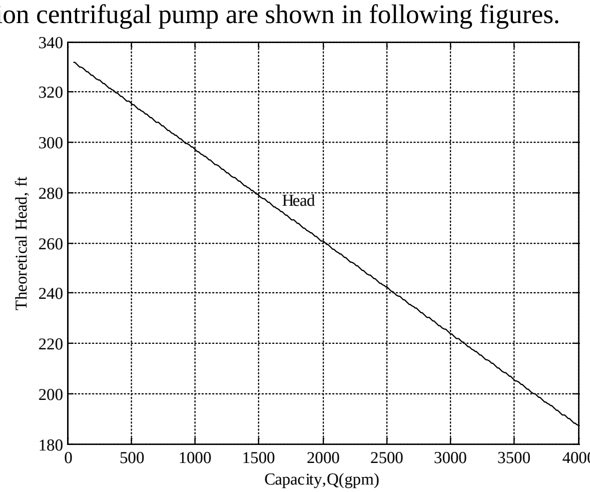

Fig. 7 Theoretical Head Curve

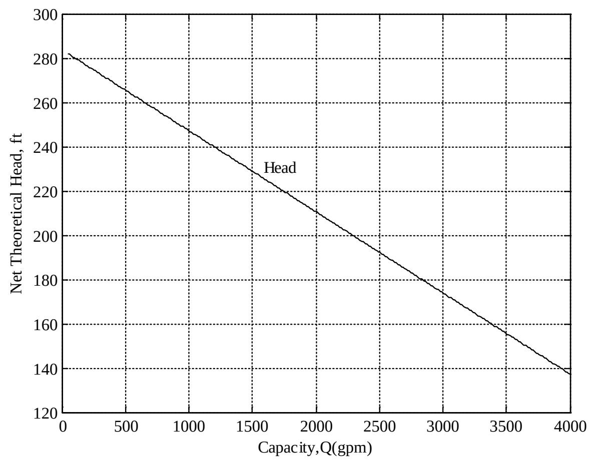

Fig. 8 Net Theoretical Head Curve

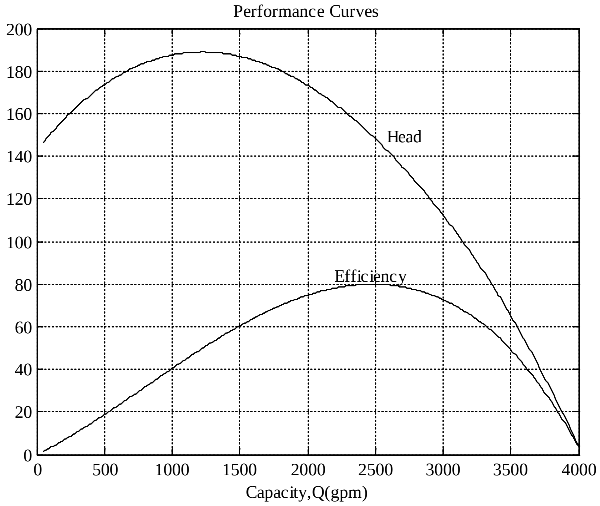

Fig.9 Performance Curves of Designed Centrifugal Pump

VIII. Conclusion

The designed pump is aimed to use in agriculture application for river pumping project which has about eight working hours per day and requires high head and capacity. So, double-suction centrifugal type is selected. The casing is horizontal split casing type. The designed pump can develop a head of 150 ft (46 m) and deliver 2500 gpm (9.48 m^3/min) of water at 1760 rpm. The designed impeller has 7.32 in (13 cm) inlet diameter, 13.5 in (24 cm.) outlet diameter, 13° inlet vane angle and 20° outlet vane angle. The number of vanes is 7. And then, the inlet width and outlet width are 1.75 in (3 cm) and 1.98 in (5 cm) respectively. The clearance between impeller and tongue of volute is 1/2 in (1.3 cm). This value is a reasonably safe value for the tongue. The diameter of discharge flange is 6 in (15 cm). The thickness of volute casing to withstand the discharge pressure, 1/4 in is selected depending upon the suction pipe diameter, 10 in. When the performance of the designed pump is predicted, the maximum efficiency has nearly 80%. At the maximum efficiency condition, we observed that it reaches at the head of about 150 ft and capacity of about 2500 gpm. 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 double-suction centrifugal pump can fulfil the requirements of agriculture application and industrial application, and then can improve pump efficiency.

Appendix

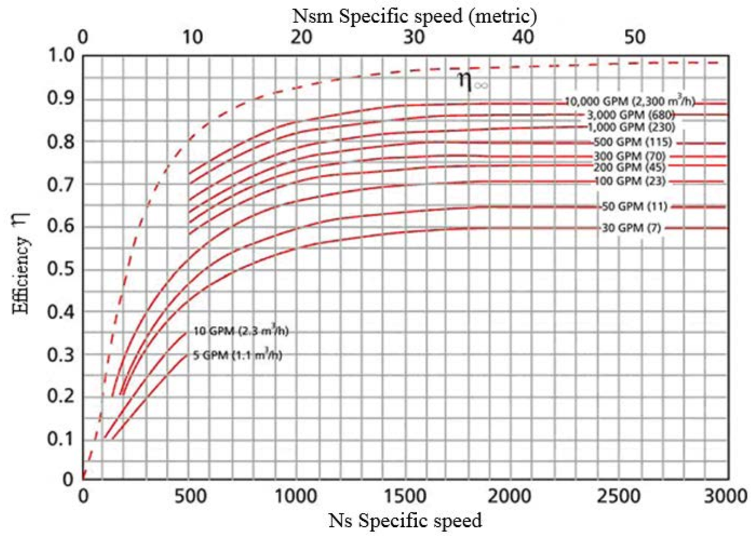

Fig. A Efficiencies as Related to Specific Speed and Flow Rate (source: The Pump Handbook published by McGraw Hill)

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