Design Calculation of Impeller for Axial Flow Pump

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Abstract- Pump technology is a proven technology in the world. The application and use of pumps today are universal. Modern public utilities, chemical plants, municipal water and sewage works, and other fields too numerous to mention would be seriously handicapped if these machines did not exit. An axial flow pump is widely used in water supply process and agricultural process and is designed for the application of high flow rate and low pressure. The axial flow pump includes the impeller rotating inside a casing of sufficient length and that ensures uniform incoming and outgoing flow. Thus, the impeller forces the liquid into a rotary motion by impeller action. This study relates to the impeller design of axial flow pump that can develop a head of 3 m and deliver 0.3 m^3/s of water at the speed of 1000 rpm. In the design impeller, outlet diameter is 350 mm, entrance vane angle is 12.78° and discharge vane angle 14.19° at outlet diameter, the hub diameter is 175mm, entrance vane angle is 24.4° and discharge vane angle is 37.62° at hub diameter respectively. The number of blades is four. The clearance between impeller and pipe or casing is 0.35 mm. The designed axial flow pump can fulfill the requirements of agricultural process.

Index Terms- axial flow pump, head, flow rate, speed, specific speed, impeller

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 Operational Principle of Axial Flow Pump

Axial flow pumps have a very large discharge and are best suited for irrigation purposes. Water enters the pump through the intake bell. It is discharged into the distributor section and then out the discharge elbow. Flowing in essentially a straight line along the pump axis keeps friction and turbulence to a minimum. The components of axial flow pump are rotating impeller, guide vane, casing, suction pipe, discharge pipe and shaft.

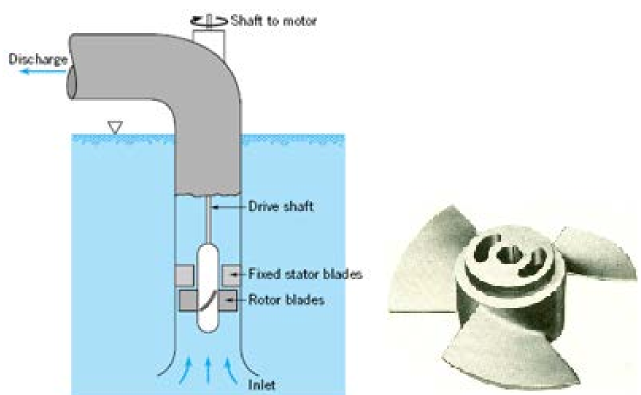

Fig. 1 Schematic Diagram of Axial Flow Pump and Impeller

The water from the sump well is sucked into the suction pipe in an axial direction, and passed through the runner vanes (impeller) and finally discharged through another set of guide blades into the delivery pipe. Suction pipe is essential to provide appropriate suction piping for securing proper pump operation. Head losses along the suction lines should be made minimal. The size of discharge piping must be selected so as to provide appropriate head losses. The shaft is to transmit power to the impeller while withstanding hydraulic thrust and weight of the rotating parts. The guide vane is a device which guides the flowing water. It sets up outlet from the impeller or enters from the suction pipe to be correct passage. By using vane in pump, it is obtained more efficiently that the pump without guide vane.

III. Design of Axial Flow Pump's Impeller

Now it is the time to articulate the research work with ideas gathered in above steps by adopting any of below suitable approaches: 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 specifications of pump that will be designed are:

Pump head, $H = 3$ m
Discharge, $Q = 18$ m^3/min $= 0.3$ m^3/s
Rotational Speed, $N = 1000$ rpm
Density of water, $\rho = 1000$ kg/m^3

A. Selection of Specific Speed and Suitable Pump Type

Specific speed is defined as the speed in revolutions per minute at which an impeller would operate if reduced proportionately in size so as to deliver one unit of capacity against one unit of total head. It is 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

$$N_s = \frac{3.65 \times N \times \sqrt{Q}}{H^{3/4}} \tag{1}$$

In this design, calculated value of specific speed based on required head and capacity is 877 rpm and it is within the range of high specific speed pump that is 400 and 1200. So, axial flow pump type that is high specific speed pump is chosen in this study.

B. Determination of Impeller Hub-Ratio and Blade Number

The ratio of impeller hub diameter and outside diameter, D_h/D_o can be determined based on specific speed. Thus, the impeller hub ratio is

$$D_d = 26.8\,(N_s)^{-0.603} \tag{2}$$

For the calculate value of specific speed 887 rpm, the hub-ratio, D_d or D_h/D_o is 0.5.

Table I. Specific Speed versus Impeller Hub-ratio and Number of Blades [5]

Specific speed (rpm)	400	600	800	1000	1200
Impeller Hub- ratio	0.6	0.55	0.5	0.45	0.4
Number of blades	6	5	4	3	2

From Table I, the number of blades can be selected based on N_s and the hub ratio. Thus, the number of blades, Z is 4.

C. Determination of Input Power

The water horsepower can be determined by the following equation.

$$WHP = \rho g Q H \tag{3}$$

Then, the brake horsepower can also be determined by using the following equation.

$$BHP = \frac{WHP}{\eta_o} \tag{4}$$

Where, η_o is overall efficiency of axial flow pump and it is taken as 0.85 in this study.

D. Prediction of Pump's Shaft Diameter

The shaft diameter at hub section of impeller is

$$d_s = \sqrt[3]{\frac{16\,T}{\pi\,\tau}} \tag{5}$$

Where, T is the torsional moment and it can be estimated by

$$T = \frac{60\,BHP}{2\,\pi\,n} \tag{6}$$

Allowable shear stress of material of shaft, $\tau$is 24.5 MPa because the main shaft is made of S30C. The estimated shaft diameter will be increased about 20% because it is difficult to predict the bending moment at this time [7]. Thus, the estimated shaft diameter is 33 mm.

E. Determination of Impeller Diameters and Clearance

The impeller diameter can be determined by

$$D = (0.1 \sim 0.08)\sqrt{\mathrm{Q} \times 60} \tag{7}$$

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Thus, $$D_{\max} = 0.1 \times \sqrt{\mathrm{Q} \times 60} \qquad (8)$$

$$D_{\min} = 0.08 \times \sqrt{\mathrm{Q} \times 60} \qquad (9)$$

For the average value of D_{max} and D_{min}, the impeller diameter is D is 0.35 m.

The calculated impeller outside diameter can be checked up by using the following optimum diameter equations.

$$D_{opt} = (4 \sim 4.6) \times \sqrt{\frac{1}{(1-D_d^2)}} \times \sqrt[3]{\frac{Q}{N}} \qquad (10)$$

Thus, $$D_{opt1} = 4 \times \times \sqrt{\frac{1}{(1-D_d^2)}} \times \sqrt[3]{\frac{Q}{N}} \qquad (11)$$

$$D_{opt2} = 4.6 \times \times \sqrt{\frac{1}{(1-D_d^2)}} \times \sqrt[3]{\frac{Q}{N}} \qquad (12)$$

In this design, the value of impeller diameter D = 0.35 m is within the range D_{opt1} = 0.309 m and D_{opt2} = 0.356 m.

Moreover, the impeller diameter, D can also be checked up the flow velocity equations.

$$v_z = (0.06 \sim 0.08) \times \sqrt[3]{Q \times N^2} \qquad (13)$$

Thus, $$v_{z1} = 0.06 \times \sqrt[3]{Q \times N^2} \qquad (14)$$

$$v_{z2} = 0.08 \times \sqrt[3]{Q \times N^2} \qquad (15)$$

And then, the impeller diameter can be compared with the following diameter equations.

$$D_{\max} = 2 \times \sqrt{\frac{Q}{\pi \times v_{z1} \times (1-D_d{}^2)}} \qquad (16)$$

$$D_{\min} = 2 \times \sqrt{\frac{Q}{\pi \times v_{z2} \times (1-D_d{}^2)}} \qquad (17)$$

The maximum and minimum diameters are 0.308 m and 0.356 m respectively. Therefore, the impeller outside diameter for axial flow pump D_o is taken as 0.35 m.

After selecting the suitable impeller outside diameter, inner diameter or hub diameter of impeller can be determined by the following equation.

$$D_h = D_d \times D_o \qquad (18)$$

If the hub-ratio is 0.5, the hub diameter is 0.175 m.

The clearance between casing pipe and outside diameter of impeller blades is

$$\delta = 0.001 \times D_o \qquad (19)$$

F. Impeller Radii, Chord Length and Blade Spacing

The radii of runner can divided into five cylindrical sections and these sections can be expressed by the following equations.

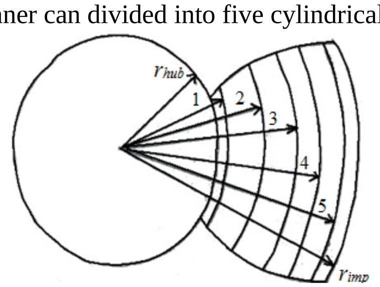

Fig. 2 Five Cylindrical Sections of Runner Blade

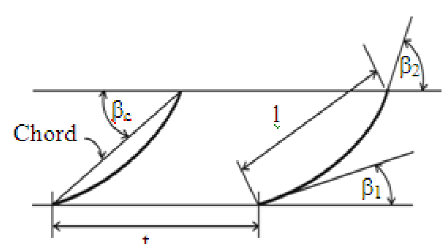

Fig. 3 Chord Length and Blade Spacing

Section I, $$r_1 = \frac{d}{2} + (0.015 \text{ to } 0.025)d \qquad (20)$$

Section III, $$r_3 = \frac{D}{2}\sqrt{\frac{(1+d)^2}{2}} \qquad (21)$$

Section V, $$r_5 = \frac{D}{2} \quad (0.015 \text{ to } 0.025)D \qquad (22)$$

Section II, $$r_2 = r_1 + \frac{r_3 - r_1}{2} \qquad (23)$$

Section IV, $$r_4 = r_3 + \frac{r_5 - r_3}{2} \tag{24}$$

Vane length called chord length (l) is one of the important factors affecting the head and the specific speed of axial flow turbine. The ratio of the vane area to the free area between the casing outer wall and the hub is referred to as solidity. However, the ratio of chord length and vane spacing (l/t_s) is used more frequently for the same purpose. The ratio l/t_s varies along the radius, increasing toward the hub. The increase in l/t at the hub is desirable for mechanical reasons.

The allowable ratio l/t at the outside diameter of runner can be determined by the following relationship.

$$\frac{l}{t} = 5.95\, K_H \tag{25}$$

Where, K_H is head coefficient and it can be expressed by the following equation.

$$K_H = \frac{H}{D^2\,(N/60)^2} \tag{26}$$

The relation between the values of l/t at the hub and at the outside diameter of the runner can be shown by the following equation.

$$\left(\frac{l}{t}\right)_{hub} = (1.25\ to\ 1.30)\left(\frac{l}{t}\right)_{periphery} \tag{27}$$

Where, t is the spacing between the adjacent vanes and it can be determined by the following.

$$t = \frac{2\,r\,\pi}{z} \tag{28}$$

G. Circulation Speed of Fluid Element (Γ)

The total circulation around the blade of pump can be calculated by using Equation

$$\Gamma = \frac{2 \times \pi \times g \times H}{\omega \times \eta_{hyd}} \tag{29}$$

Where, ω is angular velocity and it is expressed by

$$\omega = \frac{2\pi \times N}{60} \tag{30}$$

In this design, hydraulic efficiency of pump η_{hyd} is taken as 0.85. The circulation speed per blade is

$$\Gamma_n = \frac{\Gamma}{z} \tag{31}$$

IV. Results of Impeller Design

The simplified inlet and outlet velocities diagrams for the impeller are shown in Fig. 4 and Fig. 5. 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 $\alpha$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_z respectively.

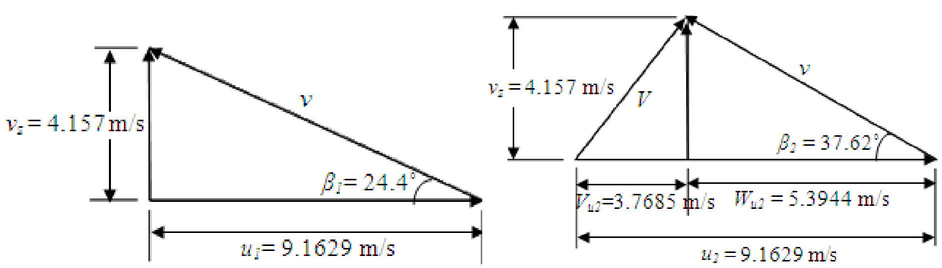

Fig. 4 Inlet and Outlet Velocity Diagrams at Hub Diameter

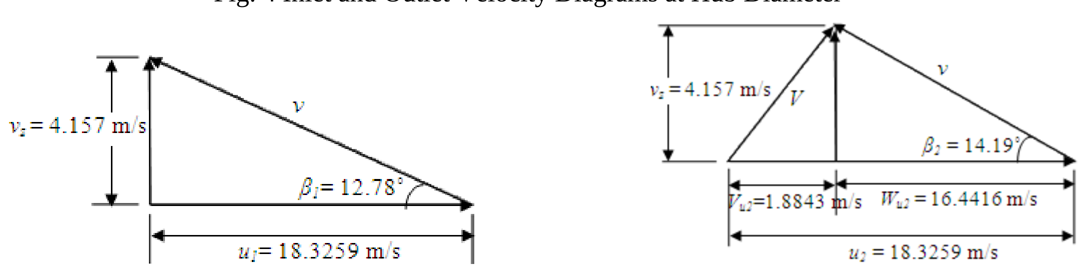

Fig. 5 Inlet and Outlet Velocity Diagrams at Outside Diameter

The calculated results in Table II are parameters of impeller design for the required head 3 m and 0.3 m^3/s.

Table II. Calculated Results of Minimum Allowable Solidity of Impeller Blades

	Units	I	II	III	IV	V
r	m	0.095	0.12	0.14	0.155	0.17
$v_Z = \dfrac{4Q}{\pi(D_o^2 - D_h^2)}$	m/s	4.157	4.157	4.157	4.157	4.157
$u = \dfrac{\pi \times D \times N}{60}$	m/s	9.948	12.566	14.661	16.232	17.802
v_{u1}	m/s	0.000	0.000	0.000	0.000	0.000
v_{u2}	m/s	3.471	2.748	2.355	2.128	1.940
$\tan\beta_1 = \dfrac{v_z}{u - v_{u1}}$	–	0.418	0.331	0.284	0.256	0.234
$\tan\beta_2 = \dfrac{v_z}{u - v_{u2}}$	–	–	0.642	0.423	0.338	0.295
β_1	rad	0.396	0.319	0.276	0.251	0.229
β_1	deg	22.70	18.300	15.800	14.400	13.100
β_2	rad	0.571	0.401	0.326	0.287	0.256
β_2	deg	32.70	23.000	18.700	16.400	14.700
$\Delta\beta$	deg	10.00	4.700	2.900	2.000	1.600
$\left(\dfrac{l}{t}\right)_{allowable}$	–	0.800	1.000	0.660	0.500	0.500
$\left(\dfrac{l}{t}\right)$	–	–	–	–	–	0.500

Table III. Calculated Results of Thin and Equivalent Camber Line of Impeller Blade Profile

	I	II	III	IV	V
R	0.095	0.120	0.140	0.155	0.170
T	0.149	0.188	0.220	0.243	0.267
l/t	0.800	0.700	0.630	0.550	0.500
L	0.119	0.132	0.139	0.134	0.133
$T_o = t/l$	1.250	1.428	1.587	1.818	2.000
Γ	2.072	2.072	2.072	2.072	2.072
Γ_n	0.518	0.518	0.518	0.518	0.518
$w_{u\ ave} = u - \left(\dfrac{v_{u1} + v_{u2}}{2}\right)$	8.213	11.192	13.484	15.168	16.832
$\tan\beta_{ave} = \dfrac{v_z}{w_{u\ ave}}$	0.506	0.371	0.308	0.274	0.247
$\beta_{ave} = \alpha'$	26.800	20.400	17.100	15.300	13.900
$w_{ave} = \dfrac{w_{u\ ave}}{\cos\beta_{ave}}$	9.205	11.939	14.110	15.727	17.338
$L = f(To, \alpha')$	1.800	1.700	1.800	1.750	1.800
$\beta_p = \dfrac{57.3 \times \Gamma_n}{w_{ave} \times l} \times \dfrac{1}{L}$	15.059	11.078	8.409	8.286	7.151
$\Delta\alpha = f(T_o, \beta_p)$	0.600	0.250	0.125	0.125	0.120

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A	27.4	20.650	17.225	15.430	14.020

Table IV. Calculated Results of Blade dimension at Various Diameter

	I	II	III	IV	V
r	0.095	0.120	0.140	0.155	0.170
D	0.190	0.240	0.280	0.310	0.340
β_2	32.700	23.00	18.70	16.40	14.70
$\theta_2 = 90 - \beta_2$	57.300	67.00	71.30	73.60	75.30
$(f/l) = f(l/t, \theta_2)$	0.120	0.10	0.080	0.030	0.005
$dm/l = f(l/t, \theta_2)$	0.100	0.075	0.055	0.035	0.020
$\left(\frac{\Delta\beta}{2}\right) = \tan^{-1}\left[2\times\left(\frac{f}{l}\right)\times\left(\frac{d_m}{l}\right)\right]$	1.375	0.859	0.504	0.120	0.011
$\beta_d = \beta_p + 2\left(\frac{\Delta\beta}{2}\right)$	17.809	12.797	9.417	8.526	7.174
$R_d = \frac{l}{2\sin\beta_d}$	0.195	0..298	0.425	0.452	0.532
$l_d = \frac{0.0175\times l\times\beta_d}{\sin\beta_d}$	0.121	0.133	0.140	0.135	0.134

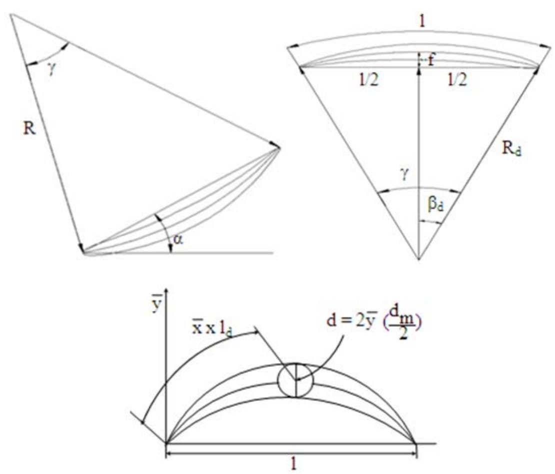

Fig. 6 Sketch for Geometric Characteristic of Blade Curvature [5]

Table V. Results of x_d or $\bar{x} \times l_d$ for blade profile

	I	II	III	IV	V
$\bar{x}$	$x_d = \bar{x} \times l_d$	$x_d = \bar{x} \times l_d$	$x_d = \bar{x} \times l_d$	$x_d = \bar{x} \times l_d$	$x_d = \bar{x} \times l_d$
0.0000	0.000	0.000	0.000	0.000	0.000
0.0025	0.303	0.333	0.350	0.338	0.335
0.0050	0.605	0.665	0.700	0.675	0.670
0.0100	1.210	1.330	1.400	1.350	1.340
0.0250	3.025	3.325	3.500	3.375	3.350
0.0500	6.050	6.650	7.000	6.750	6.700
0.1000	12.100	13.300	14.000	13.500	13.400
0.1500	18.150	19.950	21.000	20.250	20.100
0.2000	24.200	26.600	28.000	27.000	26.800
0.3000	36.300	39.900	42.000	40.500	40.200
0.4000	48.400	53.200	56.000	54.000	53.600
0.4500	54.450	59.850	63.000	60.750	60.300
0.5000	60.500	66.500	70.000	67.500	67.000
0.6000	72.600	79.800	84.000	81.000	80.400
0.7000	84.700	93.100	98.000	94.500	93.800
0.8000	96.800	106.400	112.000	108.000	107.200
0.9000	108.900	119.700	126.000	121.500	120.600
0.9500	114.950	126.350	133.000	128.250	127.300
0.9700	117.370	129.010	135.800	130.950	129.980

Table VI. Results of $r_d = \bar{y} \times d_m / 2$ for blade profile

	I	II	III	IV	V
$\bar{y}$	$\bar{y}\times\frac{d_m}{2}$	$\bar{y}\times\frac{d_m}{2}$	$\bar{y}\times\frac{d_m}{2}$	$\bar{y}\times\frac{d_m}{2}$	$\bar{y}\times\frac{d_m}{2}$
0.000	0.000	0.000	0.000	0.000	0.000
0.147	0.875	0.728	0.562	0.345	0.196
0.196	1.166	0.970	0.749	0.461	0.261
0.294	1.749	1.455	1.123	0.691	0.391
0.405	2.409	2.005	1.547	0.952	0.538
0.516	3.070	2.554	1.971	1.213	0.686
0.662	3.939	3.277	2.529	1.556	0.880
0.763	4.539	3.777	2.915	1.793	1.015
0.840	4.998	4.158	3.209	1.974	1.117
0.949	5.647	4.698	3.625	2.230	1.262
0.998	5.938	4.940	3.812	2.345	1.327
1.000	5.950	4.950	3.820	2.350	1.330
0.982	5.843	4.861	3.751	2.308	1.306
0.895	5.325	4.430	3.419	2.103	1.190
0.756	4.498	3.742	2.888	1.777	1.005
0.560	3.332	2.772	2.139	1.316	0.745
0.342	2.041	1.698	1.310	0.806	0.456
0.222	1.321	1.099	0.848	0.522	0.295
0.168	0.999	0.832	0.642	0.395	0.223

0.9900	119.790	131.670	138.600	133.650	132.660
1.0000	121.000	133.000	140.000	135.000	134.000

0.092	0.547	0.455	0.351	0.216	0.122
0.000	0.000	0.000	0.000	0.000	0.000

Results data of x and y coordinates for blade profiles of five sections are shown in Table V and VI. By using these results, detail drawings of blade profiles are shown in following figures.

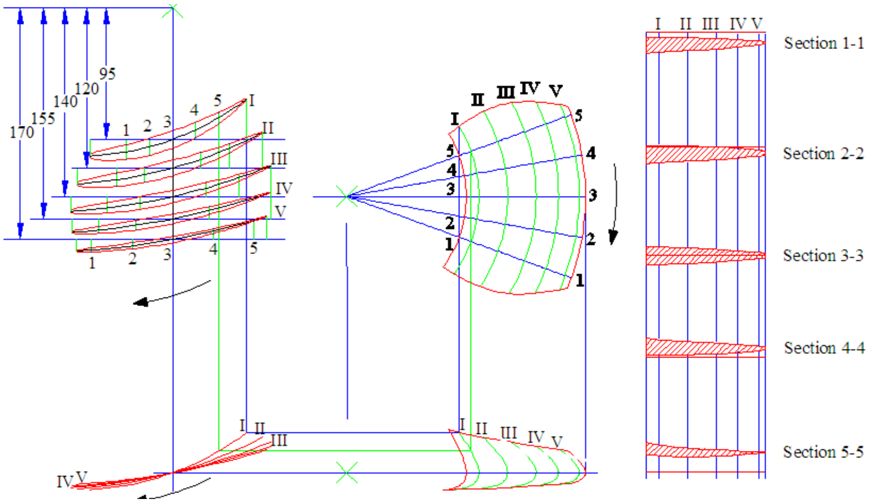

Fig. 7 Two Dimensional View of Blade Profile for Impeller

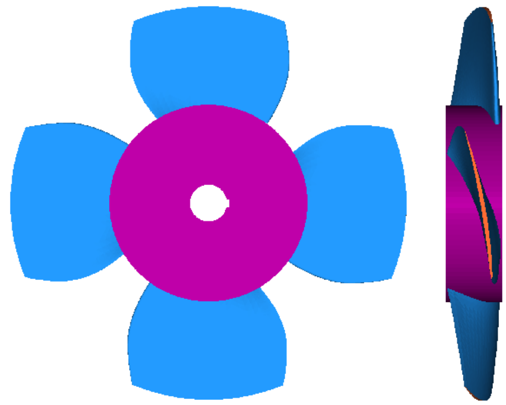

Fig. 8 Three-Dimensional View of Impeller

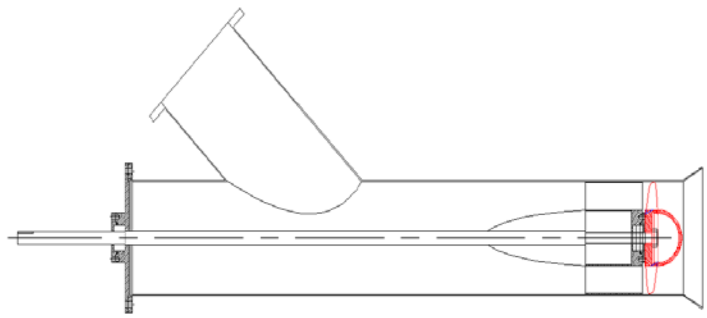

Fig. 9 Assembly Drawing of Axial Flow Pump

V. Conclusion

The designed pump is aimed to use in the application of agriculture process and water supply process which have about eight working hours per day and requires low head and high capacity. So, axial flow pump type is selected. The designed pump can develop a head of 3m and deliver 0.3m^3/sec of water at 1000rpm. According to the design result, impeller has 14in (350mm) outlet diameter, 12.78˚entrance vane angle and 14.19˚discharge vane angle at outlet diameter and 7in (175mm) inlet diameter, 24.4˚entrance vane angle and 37.62˚discharge vane angle at inlet diameter respectively. The number of vanes is four. The clearance between impeller and pipe or casing is 0.01in (0.35mm). The design pump is used only to pump water at 70˚F and if very hot water is used this pump will be damaged. The designed axial flow pump can fulfill the requirements of agricultural process and then can improve pump efficiency.

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