Design, Modeling & Analysis of a Submersible Pump and to improve the Pump Efficiency

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Abstract- There are two ways of optimizing the pump performance, first by optimizing the cost and the second by increasing the overall efficiency of the pump. The second way, i.e. increasing the overall efficiency of the pump, involves improving the quality, analysis and design; hence this method of optimizing is opted. The main aim of the project is to increase the pump efficiency of the model G554T and thus increase its overall efficiency. The overall efficiency constitutes of Motor efficiency and Pump efficiency. The first step is to analyze the existing model G554T & determine its existing overall efficiency. The second step is to re-design the pump such that for the same power input, greater power output is obtained. The new pump design is modeled in SOLIDWORKS. The modeled pump is analyzed using ANSYS and finally the newly designed pump is evaluated and confirmed for improvement of overall efficiency.

Keywords – Submersible Pump, Optimization of Pump Performance, Overall Efficiency and Pump Efficiency.

I. INTRODUCTION

In recent days, there is a huge competition among all the pump industries. To survive in this competitive market, an industry has to produce high quality pumps at a reasonable price. Supplying higher quality pumps at competitive price to people is a big challenge for all the pump industries. So the main aim of the industry will be either to reduce the cost of the pump by reducing the material cost or to improve the quality of the pump by improving its performance. In order to reduce the cost, the small scale industries go for reducing the material cost. Whereas in industries with R&D section try for improving the performance of the pump.

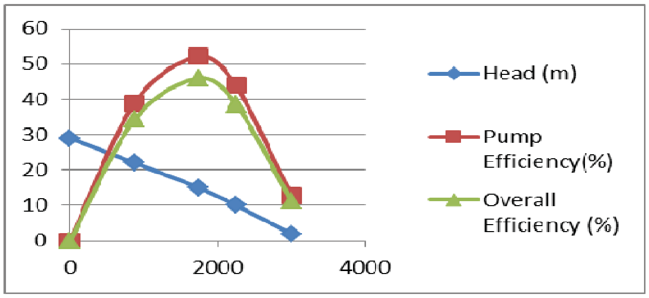
There are two ways of optimizing the pump performance, first by optimizing the cost and the second by increasing the overall efficiency of the pump.

The second way, i.e. increasing the overall efficiency of the pump, involves improving quality, analysis and design; hence this method of optimizing is opted. The pump model G554T, which is a de-watering, semi-open and mixed flow type of submersible pump, is the fastest moving submersible pump manufactured by Mody Pumps (India) Pvt. Ltd. So this project work is conducted on the model G554T. Even a small increase, say by 1%, in the overall efficiency of the pump yields a lot of profit for the firm.

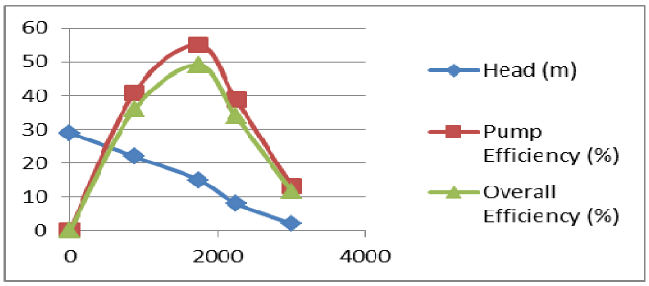
The main aim of the project is to study and analyze the existing pump model G554T and to re-design the model G554T so that its pump efficiency can be increased, thus increasing its overall efficiency.

The overall efficiency constitutes of Motor efficiency and Pump efficiency. The motor efficiency is related to the stator winding and the number of poles it is having. The pump efficiency is given by the product of Mechanical, Disc friction, Volumetric and Hydraulic efficiency [1]. The existing motor of G554T is providing an efficiency of 88.2% at 100% load [2]. Thus there is hardly any necessity of improving its efficiency. Hence the concentration is done on improving the pump efficiency, which thereby increases the overall efficiency.

II. EXPERIMENTAL STUDY OF EXISTING PUMP

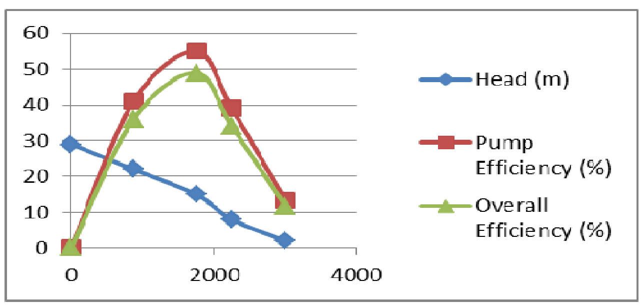


X-axis Discharge in LPM
Figure 1. Performance of G554T – 1308554



X-axis Discharge in LPM

Figure 2. Performance of G554T 1308549



X-axis Discharge in LPM

Figure 3. Performance of G554T 1307222

From the above graphs it can be seen that duty point is at discharge of 1750 LPM and head of 15m. The overall efficiency, at duty point, is in the range of 45% to 50.6%. So the overall efficiency of the existing pump model is close to the value provided by the firm, which is 44%. Hence the new pump model should be designed such that the overall efficiency of the pump should be 52% or greater.

III. DESIGN OF IMPELLER

The impeller is the main part of the submersible pump which determines the pump performance. The impeller is designed according to the standards set by SiTarc, which is a well-known educational institute giving information regarding pumps. The data from of pump's motor performance is obtained prior proceeding with design. The data required are Motor Efficiency at 100% load, Input voltage, power factor and RPM and the overall efficiency which is need to be achieved is assumed. The discharge and head for which the impeller is to be designed is determined as follows [4]

$$\text{Overall efficiency } \eta_o = \frac{\text{Power Output}}{\text{Power Input}}$$

$$\eta_o = \frac{\rho Q g H}{\text{Power Input}} \quad (1)$$

A. Impeller inlet design [1]

1] Specific Speed,

$$N_s = \frac{N\sqrt{Q}}{H^{0.75}} \quad (2)$$

Depending on the specific speed the type of the impeller is selected. The impeller selected may be radial, mixed or axial impeller.

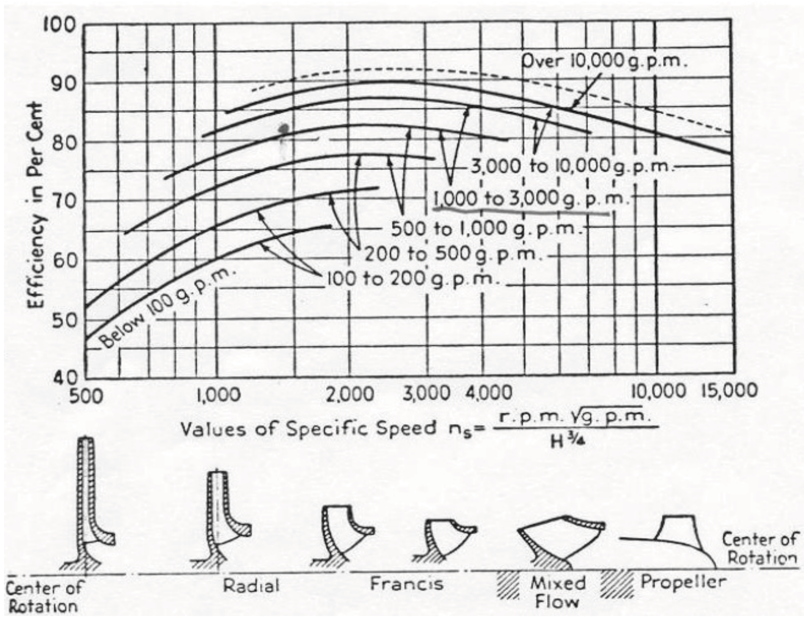


Figure 4. Approximate Impeller shapes and efficiencies as related to specific speed

2] Impeller Inlet design

a) Eye diameter,

$$D_e = K\,(Q/N)^{0.33} \quad (3)$$

where K is a constant and = 4.66 for a well-designed pump

b) Calculation of shaft diameter and hub diameter at impeller,

Shaft Diameter,

$$D_s = 127^*\,(kW/N)^{0.33} \quad (4)$$

Assume 2 to 3 mm greater than obtained so as to take care of critical speed and strength.

Hub Diameter,

$$D_h = (1.5 \text{ to } 1.6)\, D_s \quad (5)$$

c) From ratio of maximum head to maximum output head, obtain Twisted inlet angle

d) Vane Thickness,

$$t = \text{Impeller diameter} / 100 \quad (6)$$

But the minimum vane thickness should be 4 mm

e) Mean stream diameter

$$D_{1m} = \sqrt{\frac{D_e^2 + D_h^2}{2}} \quad (7)$$

f) Vane Inlet Angles

Inlet Area,

$$A_{inlet} = (\pi/4)(D_e^2 - D_h^2) \quad (8)$$

Velocity in the eye,

$$V_{eye} = \left(\frac{Q}{A_{inlet}}\right) \quad (9)$$

Meridional velocity,

$$V_{m1} \approx 1.1\ V_{eye} \tag{10}$$

Tangential velocity can be estimated as
At tip streamline,

$$V_{tip}=\frac{\pi * D_e * N}{60} \tag{11}$$

$$\tan \beta_{1t}=\frac{V_{m1}}{V_{tip}} \tag{12}$$

At mean streamline,

$$V_m=\frac{\pi * D_m * N}{60} \tag{13}$$

$$\tan \beta_{1m}=\frac{V_{m1}}{V_m} \tag{14}$$

At hub streamline,

$$V_h=\frac{\pi * D_h * N}{60} \tag{15}$$

$$\tan \beta_{1h}=\frac{V_{m1}}{V_h} \tag{16}$$

Add incidence angle to all three streamline by 2^0 to 3^0, such as to make it a whole number
Mean relative velocity at inlet,

$$W_{1m}=\sqrt{V_{m1}^2+V_m^2} \tag{17}$$

g) Inlet breadth
Flow Rate,

$$Q= A_{inlet} * V_{m1} \tag{18}$$

Inlet flow including leakage,

$$Q^1=\frac{Q}{\eta_v} \tag{19}$$

Initially assume an volumetric efficiency of 95%
The diffuser is having four vanes and the vanes of the impeller should be one less than diffuser [5]. Hence the number of vanes on Impeller is three. Z=3

Inlet Breadth,

$$b_1=\frac{A_{inlet}}{\pi D_{1m}-\frac{Z*t}{\sin\beta_{1m}}} \tag{20}$$

B. Impeller outlet design [1]

1] Slip method

a) Type number

$$K_s=\frac{2\pi N\sqrt{Q}}{60(gH)^{0.75}} \tag{21}$$

b) Choosing head co-efficient H_c value of type number

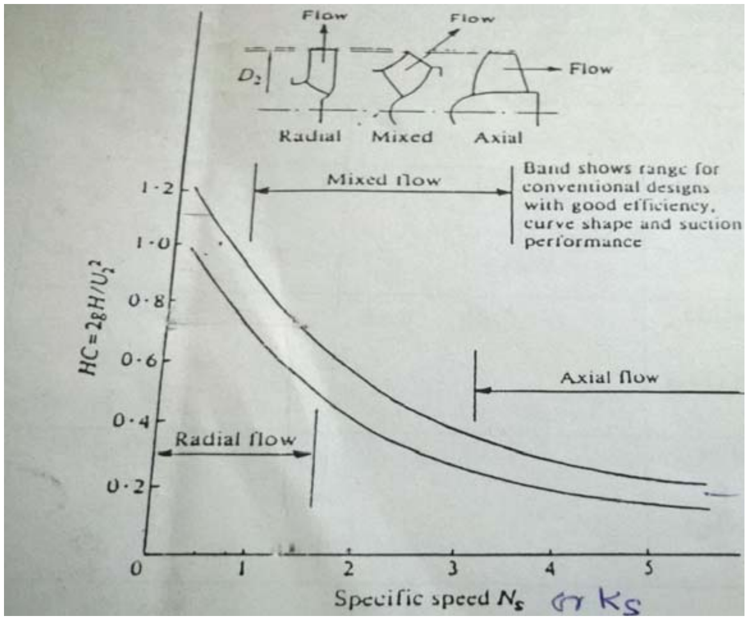


Figure 5. Values of head co-efficient for a range of specific speeds

$$H_c = \frac{2gH}{U_2^2} \tag{22}$$

$$U_2 = \sqrt{\frac{2gH}{H_c}}$$

c) Impeller outside diameter

$$D_2 = \frac{60*U2}{\pi*N} \tag{23}$$

d) Hydraulic efficiency

$$\eta_h = 1 - \frac{0.071}{Q^{0.25}} \tag{24}$$

e) Mean outlet Whirl Velocity

$$V_{u2} = \frac{gH}{U_2\,\eta_h} \tag{25}$$

As an initial estimate, elect $\beta_{2m} = 25^0$

f) Calculation of slip,

According to Wiesner,

$$h_0 = 1 - \frac{\sqrt{\sin\beta_{2m}}}{Z^{0.7}} \tag{26}$$

$$\text{Slip} = (1-h_0)\,U_2 \tag{27}$$

According to Myles,

$$h_0 = 1 - \frac{\pi}{Z}\sin^2\left(\frac{\beta_1+\beta_2}{2}\right)\left[2 - \frac{D_{1m}}{D_{2m}} - \left(\frac{D_{1m}}{D_{2m}}\right)^2\right] \tag{28}$$

Calculate slip using equation (27)

Consider the slip with lower value

g) Meridional Velocity,

$$V_{m2} = \tan\beta_{2m}\,(U_2 - V_{u2} - \text{slip}) \tag{29}$$

h) Outlet Breadth

$$b_2 = \frac{Q^1}{V_{m2}\left(\pi D_{2m} - \frac{Zt}{\sin\beta_{2m}}\right)} \tag{30}$$

i) Absolute Velocity

$$V_{2m} = \sqrt{V_{u2}^2 + V_{m2}^2} \quad (31)$$

j) Relative velocity

$$W_{2m} = \frac{V_{m2}}{\sin \beta_{2m}} \quad (32)$$

k) Absolute flow angle

$$\alpha_{2m} = \tan^{-1} \frac{V_{m2}}{V_{u2}} \quad (33)$$

2] Area Ratio Method

a) Type number

$$K_s = \frac{2\pi N\sqrt{Q}}{60(gH)^{0.75}} \quad (34)$$

b) To find out the values of Head Co-efficient and Flow Co-efficient

$$K_s{*}Y = 0.8 \text{ to } 1.2 \quad (35)$$

For most of the cases $K_s*Y = 1$ is used.

Obtain the value of Y and find out the values of Head Co-efficient and Flow Co-efficient

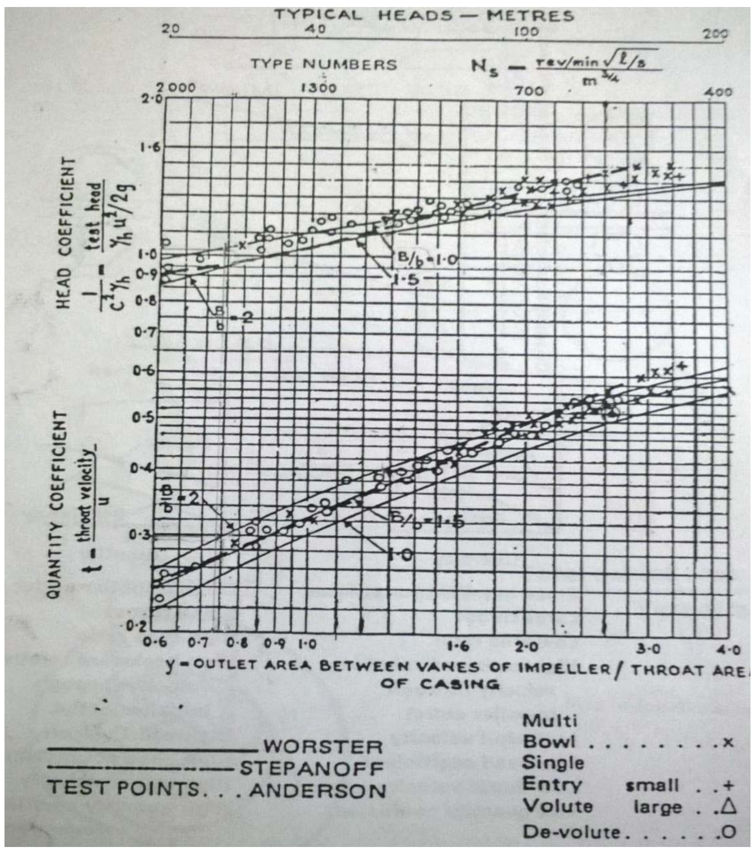


Figure 6. Area ratio plot

c) To find outer diameter

$$U_2 = \sqrt{\frac{2gH}{H_c}} \quad (36)$$

$$D_2 = \frac{60*U_2}{\pi*N} \quad (37)$$

d) Throat velocity

$$V_3 = Q_c * U_2 \quad (38)$$

$$\text{Diffuser flow area } A_3 = \frac{Q}{V_3} \tag{39}$$

Outer Area between vanes of impeller,

$$OABV = A_3 * Y \tag{40}$$

$$\text{Also } OABV = 0.95\pi\, D_2 b_2 \sin\beta_2 \tag{41}$$

$$b_2 = \frac{OABV}{0.95\pi D_2 \sin\beta_2}$$

3] Stepanoff Design Method

a) For values of N_s, the values of K_u and K_{m2} is obtained

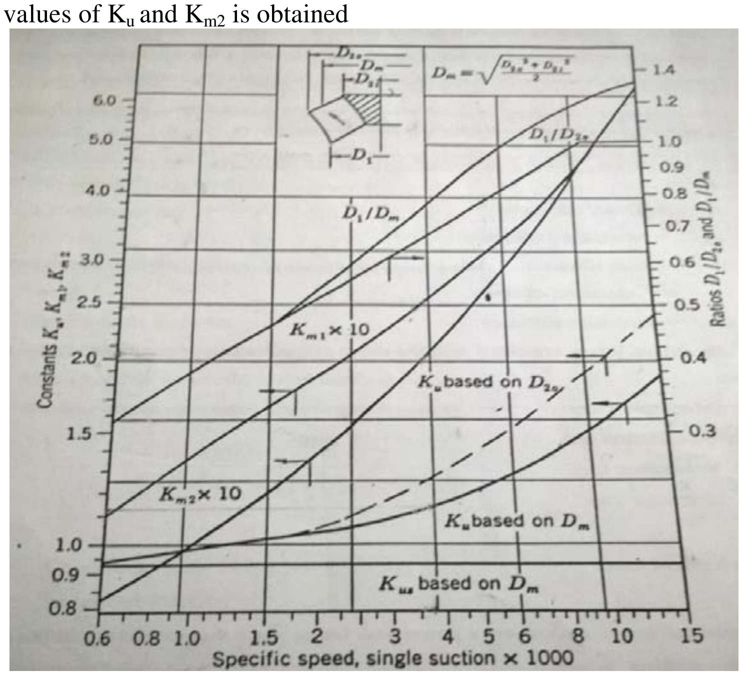


Figure 7. Specific speeds v/s various impeller constants

b) To find the outer diameter

$$U_2 = K_u \sqrt{2gH} \tag{42}$$

$$D_2 = \frac{60 * U_2}{\pi * N} \tag{43}$$

c) To find outlet breadth

$$V_{m2} = K_{m2} \sqrt{2gH} \tag{44}$$

Outlet Breadth,

$$b_2 = \frac{Q^1}{V_{m2}\left(\pi D_2 - \frac{zt}{\sin\beta_{2m}}\right)} \tag{45}$$

Select the outlet design which yields the largest outside diameter and lesser outlet breadth.

C. Loss Estimation [1]

$$\text{Overall pump efficiency } \eta_0 = \eta_m * \eta_d * \eta_v * \eta_h \tag{46}$$

a) Mechanical Losses- Assuming higher margin of 4%

$$\eta_m = \frac{\text{Power Input} - \text{Mechanical losses}}{\text{Power Input}} \tag{47}$$

$$\eta_m = 96\ \%$$

b) Disc Friction loss

$$\text{1] Pfleiderer's equation} = 10^{-6} \gamma U_2^3 D_2^2 \tag{48}$$

2] A.H.Gibson = $\frac{D_2^{4.83} N^{2.83}}{2.78*10^6}$ (49)

3] Stodola Equation = $10^{-6}\beta\gamma U_2^3 D_2^2$ (50)

The above equation 47, 48 and 49 gives the disc power. The average of the three is considered.

Disk friction efficiency $\eta_d = \frac{\rho Q g H}{\rho Q g H + \text{Disc power}}$ (51)

c) Volumetric efficiency - While designing the pump, the volumetric efficiency was assumed to be 95% and hence the pump is designed to have volumetric efficiency η_v= 95%

d) Hydraulic efficiency

1] According to Jekat, $\eta_h = 1 - \frac{0.071}{Q^{0.25}}$ (52)

2] According to Peck, $\eta_h = 100 - \frac{K_h}{(\frac{Q}{\sqrt{H}})^{0.25}}$ (53)

where K_h= 35 for small pump with good surface finish, Q is in GPM and H in feet

3] According to G.F. Wislecenus, $\eta_h = \sqrt{\eta_o - a}$ (54)

where a= 0.02 to 0.04 and η_o is obtained from figure 1

4] According to Lomakin, $\eta_h = 1 - \frac{A*C}{(\log D_e - B)^2}$ (55)

where A is co-efficient reflecting quality of surfaces of water ways and usually A= 0.42
B is empirical constant and usually B= 0.172
C is co-efficient reflecting efficiency deterioration due to departure and usually C= 1 for normal design
D_{eq} is equivalent diameter at inlet to the impeller

$D_{eq} = D_e(1-r^2)^{0.5}$ (56)

r is ratio of hub diameter to eye diameter

Take the average of all the four hydraulic efficiency

Hence overall efficiency $\eta_o = \eta_m * \eta_p$ (57)

$H_{shut} = 0.585(\frac{U_2^2}{g})$ (58)

The current consumed by the impeller

$I = \frac{\rho Q g h}{1.732 * V * \eta o * \cos\varphi}$ (59)

The motor is providing a current of 14.2 Amps. Considering 1 Amps from safety point of view, the impeller design which consumes 13 Amps is selected. After several iteration, for a Discharge of 0.034 m^3/s and head of 15 m, the current consumed is 13 Amps. The dimension of the selected impeller

D_e = 110 mm, D_s = 20 mm, D_h = 35 mm, t = 4 mm, D_{1m} = 81 mm, $\beta_{1t} = 17^0$, $\beta_{1m} = 22^0$, $\beta_{1h} = 42^0$, b_1 = 37 mm,

D_2 = 135 mm, b_2 = 25 mm, $\beta_{2m} = 25^0$, Shut valve head = 24.36 m, η_p = 76 % and η_o = 67 %

D. Layout of Meridional Section and Vane Shape [1]

Step 1- Draw the mean stream line. Mean stream diameter is 81 mm, hence mean stream radius is 40.5 mm. Outer diameter is 135 mm, hence outer radius is 67.5 mm. For submersible pumps, semi cone angle is 60^0. Hub diameter is 35 mm, hence hub radius is 17.5 mm. Mean stream line radius is obtained by

$R_m = (D_2*0.1 + b_2/2) = 25$ mm (60)

Step 2 – Divide mean streamline into eight equal intervals

Step 3- Using volumetric method, the areas should be equal on both sides of mean streamline at each section

$$2\pi r_1 l_1 = 2\pi r_2 l_2 \quad (61)$$

Step 4 - Drawing smooth curves, one extending tip streamline and the other extending hub streamline, such that the curve connects the tangent of the circles

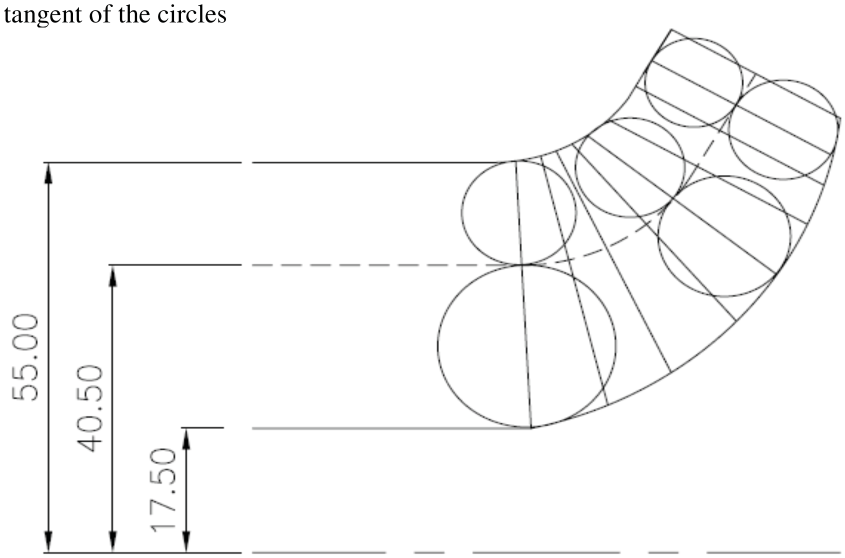


Figure 8. Meridional section of impeller

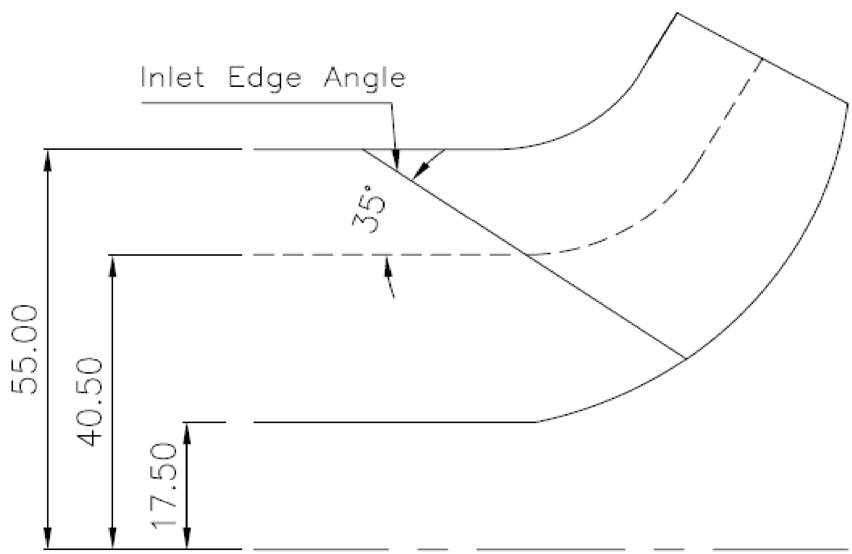


Figure 9. Final Meridional section of impeller

Table-1 Determination of vane shape along mean streamline

Straight Line	Δx	β	tanβ	Radius (R)	1/R tanβ	Δx(1/R tanβ)	Δφ	totalφ
0		25	0.47	67.5	0.032			0
	5.365					0.172	9.85	
1		27.1	0.51	63.5	0.031			9.85
	5.365					0.166	9.51	
2		29.2	0.56	59.5	0.03			19.36

	5.365					0.161	9.22	
3		31.4	0.61	54.5	0.03			28.58
	5.365					0.161	9.22	
4		33.5	0.66	50	0.03			37.8
	5.365					0.161	9.22	
5		35.6	0.72	46.5	0.03			47.02
	5.365					0.161	9.22	
6		37.7	0.77	43	0.03			56.24
	5.365					0.161	9.22	
7		39.9	0.84	41.5	0.029			65.46
	5.365					0.156	8.94	
8		42	0.9	40.5	0.027			74.68

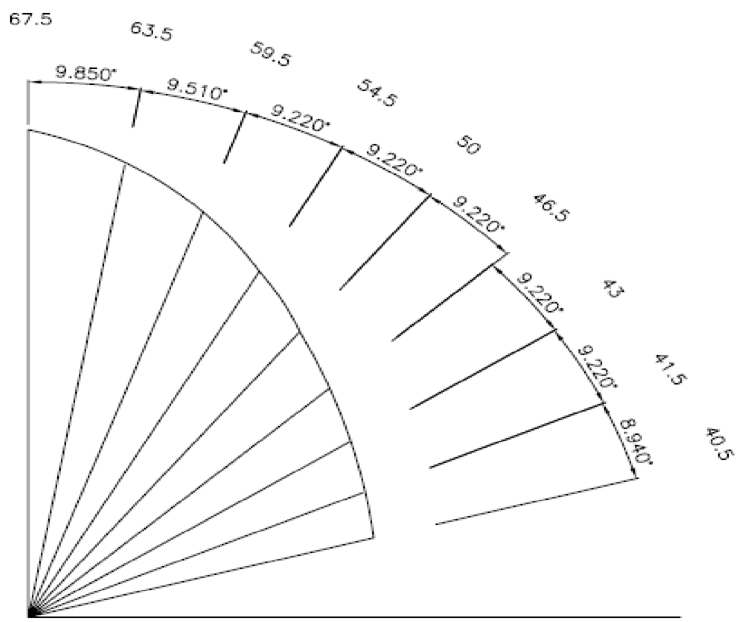


Figure 10. Vane layout of the impeller

Also the angle of overlap, i.e. angle between the inlet of one vane and outlet of the consecutive vane, should be between 35^0 and 50^0 for optimum performance of vane. In the SOLIDWORKS model the angle of overlap is maintained at 35^0.

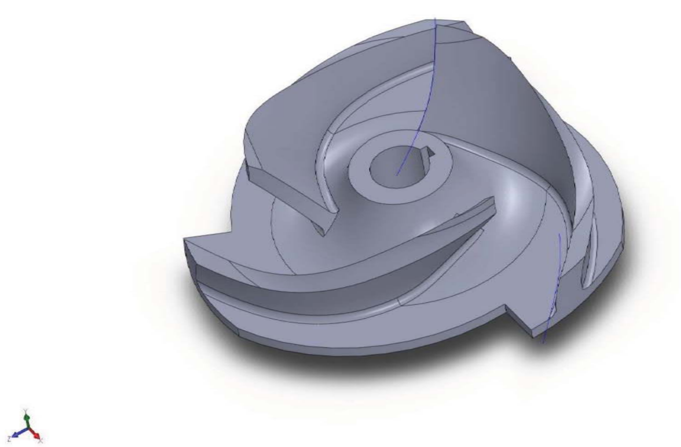


Figure 11. SOLIDWORKS model of the impeller

IV. ANALYSIS AND RESULT

The SOLIDWORKS 2012 models of all the three proposed design is constructed and analyzed for stress, deformation and factor of safety by utilizing ANSYS 14.0.

Auto meshing is done which yielded 33428 nodes and 18660 elements. Keyway is fixed. Force applied 1] 360 N by water and 2] Thrust force is 10% of weight = 0.1*65 kg = 6.5 kg = 65N.

The equivalent stress is as showed in figure 9. It can be seen that the maximum stress acting on the impeller is 2.5375 MPa which is within the limits. The yield stress of stainless steel is 241 MPa [7]. Hence stress is safe stress. The maximum deformation is 0.0014715 mm as shown in figure 10, which is minimum within the limits and hence the deformation is safe. The figure 11 shows that the impeller is safe as the factor of safety is 15. Hence safety factor is satisfactory.

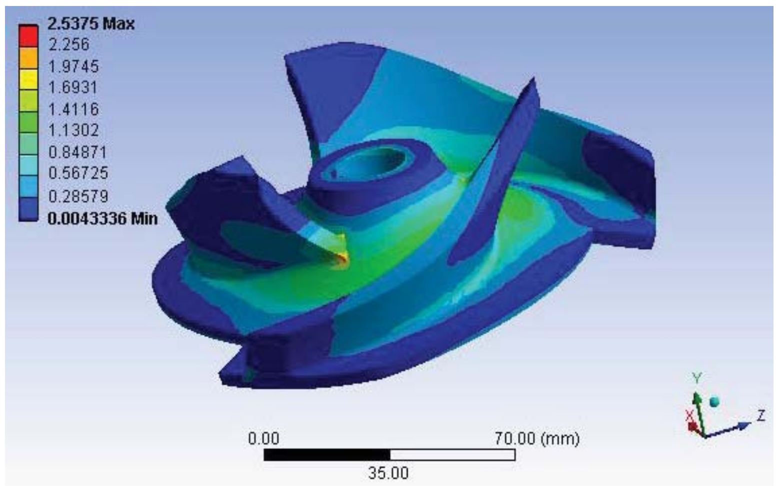


Figure 12. Equivalent Von Mises Stress

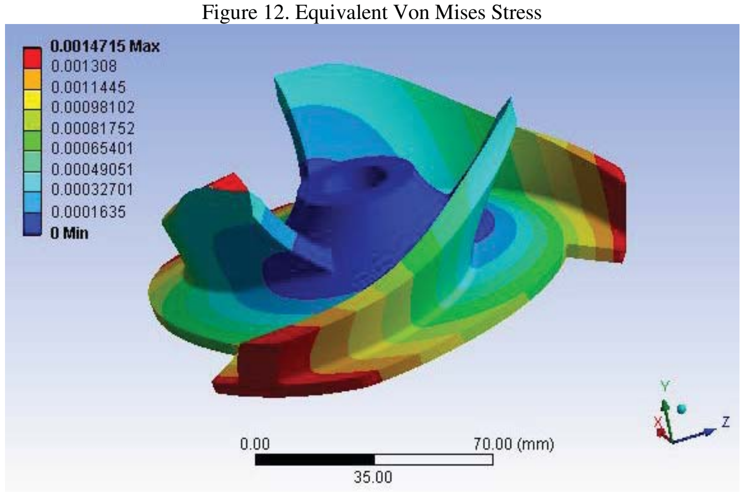


Figure 13. Total Deformation

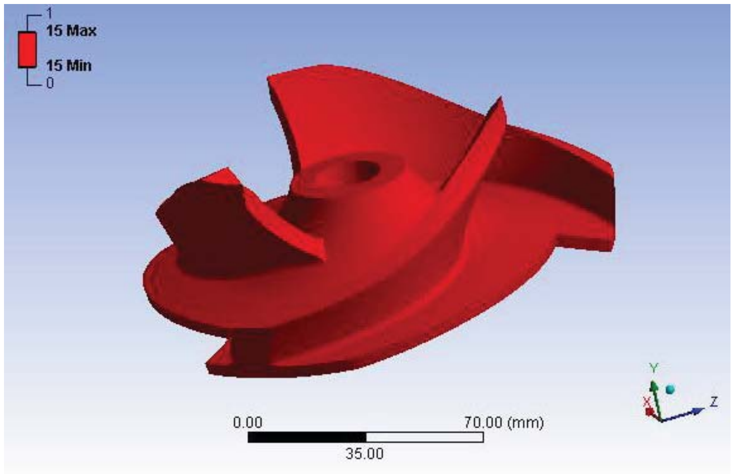


Figure 14. Safety Factor

V. CONCLUSION

- The designed pump is in prototype stage.
- The pump is designed according to the standards setup by SiTarc Pump institute, Coimbatore, which is the only pump institute in India which gives information and education regarding submersible pumps.
- The designed pump is yielding a theoretical pump efficiency of 76% at the duty point, i.e. the best efficiency point. The figure 6.4 indicates that the theoretical pump efficiency of the existing pump to be 68%. Hence the prototype pump is giving a pump efficiency which is greater than 8% produced by the existing pump.
- The prototype pump is using the complete load produced by the motor of the pump which produces a greater discharge and hence increase in pump efficiency and thus increase in overall efficiency.

- From the ANSYS result, all the three designs are safe from mechanical point of view as the stresses acting are very less. Stresses are very less as there are a hardly few regions with maximum stress acting spots on the result plot. Also the stress acting is very lower, in the entire three models, than the yielding stress of stainless steel.
- The total deformation is also very less in all the three models in order of μm.
- It can also be seen that the factor of safety is satisfactory for all the three models. But the third design is selected as it is consuming the entire load supplied by the motor.

VII. FUTURE SCOPE

- Further development of this designed impeller can be done by CFD analysis of the impeller. This will provide a closer view of dynamics present between the impeller and water.
- Another way of development is by increasing the number of vanes. In this design as the numbers of vanes on diffuser were 4, impeller with three vanes could be used. The number of vanes on impeller can have a maximum value of 7 [1]. By increasing the vanes, there might be improvement in discharge or head or both. Hence increasing of impeller vanes play an important role in overall efficiency, since overall efficiency is directly proportional to discharge and head.
- Here the angle of overlap is maintained at 35^0. Variation of the angle can be made upto 50^0 and its effect on the impeller performance can be studied.
- Even trial and error methods can also have an impact on the performance of the submersible pump. Some research papers [6] have shown that increasing the inlet angle may improve the overall efficiency to some extent. Also there has been improvement in the overall efficiency to some extent by increasing the inlet angle and decreasing the outlet angle.

- Here in this project, focus was not laid on motor efficiency. Even increasing the motor efficiency can increase the overall efficiency of the pump. By providing a cooling jacket around the stator casing, the motor will be kept cooler and there would be an increase in motor efficiency [3]. Although the flowing water will keep the motor cool, while water moves towards the outlet, but providing cooling jacket is more effective than that cooled by water.

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