Nomenclature	
d_1- inlet diameter of impeller, mm	β_2-blade outlet angle, °
d_2- outlet diameter of impeller, mm	H- head developed, m
B_1-blade width at inlet, mm	Q- discharge, Lit/sec
B_2- blade width at outlet, mm	η_s- speed of the impeller, rpm
β_1-blade inlet angle, °	η-efficiency, %

2. Design of Impeller

The pump unit selected for redesign has the following specifications

1. Inlet Diameter (d_1) = 60 mm
2. Outlet Diameter (d_2) = 148 mm
3. Head (H) = 22 m
4. Discharge (Q) = 4 lit/sec
5. Blade Inlet Angle (β_1) = 25 °
6. Blade Outlet Angle (β_2) = 27°
7. Power = 3 hp
8. Speed (η_s) = 2800 rpm

Blade angles β_1 and β_2 have been selected as variable parameters to optimize head developed H and discharge Q.
Head H is calculated [8] as

$$H = 2.79 \times 10^{-4} \times \eta_s^2 \left[d_2^2 - d_1 d_2 \frac{\tan\beta_1}{\tan\beta_2}\right] \quad \ldots\ldots\ldots\ldots \text{I}$$

Discharge Q is calculated [8] as

$$Q = 0.1644 \times d_1^2 \times B_1 \times \eta_s \times \tan\beta_1 \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{II}$$

The selected range for β_1 and β_2 are 40° to 55° in both cases to determine the head and discharge developed by the pump using analytical treatment. Table 1 provides the data generated for head H and discharge Q using equation I and II for different values of β_1 and β_2

Table 1: Values of head H and discharge Q for different values of β_1 and β_2

β_1	β_2	*H*	*Q*
40°	40°	28.53	6.55
	50°	34.29	6.55
	55°	36.56	6.55
45°	40°	23.98	8.58
	50°	31.08	8.58
	55°	33.88	8.58
50°	40°	20.35	9.87
	50°	28.53	9.87
	55°	31.75	9.87

Figure 1 shows the plot of head H against β_2 and Figure 2 shows the plot of discharge Q against β_2 for different values of β_1. It is seen from Figure 1 and Figure 2 that increasing value of β_2 provides increase in value of head H a combination of β_1=40° and β_2= 55° provides maximum head of 36.56 m. The value of discharge Q is depend on the blade inlet angle β_1 hence maximum value of discharge Q is 9.87 lit/sec at β_1=50°.

But as per the power consumption limitation we cannot allow the value of β_2=55°. The combination of β_1=50° and β_2= 55° provides the head H of 31.75m and discharge Q of 9.87 lit/sec. Similarly the combination of β_1=45° and β_2= 55° provides the head H of 33.88m and discharge Q of 8.58 lit/sec. These two combinations seem good for improve performance but value of β_2 is restricted below 55° to reduce the power consumptions.

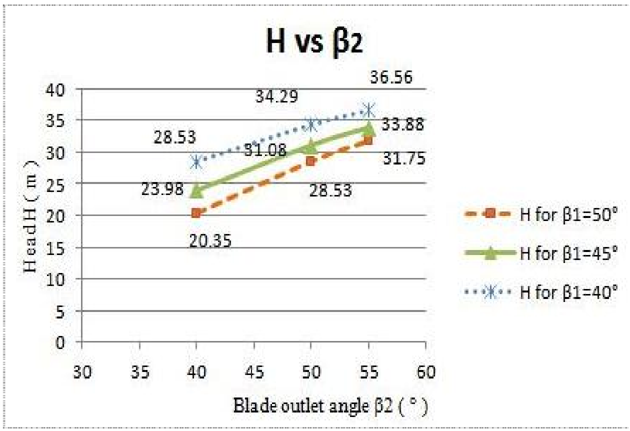

Figure 1: H vs. β_2 for different values of β_1

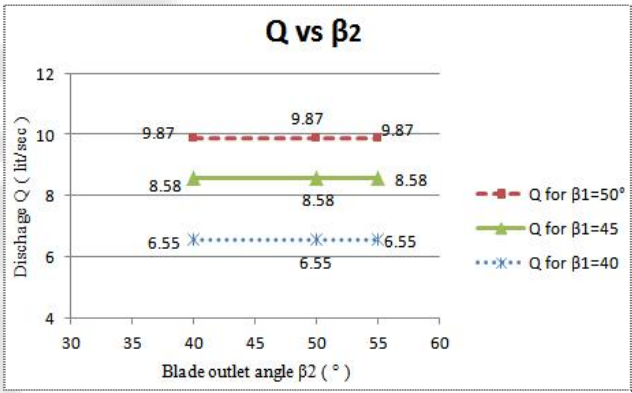

Figure 2: Q vs. β_2 for different values of β_1

In the present design a combination of β_1= 45° and β_2= 50° is selected for head H of 31.08 m and discharge Q of 8.58 lit/sec. this value of head H is calculated without considering the losses in casing and pipe.

3. Modeling of Impeller using CATIA

Figure 3 and Figure 4 show the model for closed type impeller and its casing respectively as per original dimensions. The impeller has four vanes enclosed with two end plates.

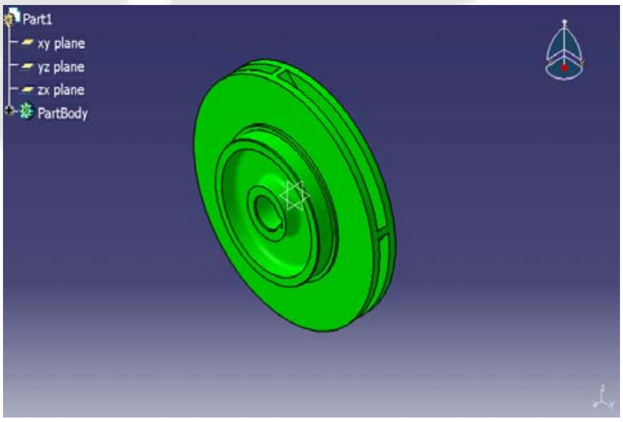

Figure 3: Model of impeller using CATIA

Figure 4: Model of impeller using CATIA

Figure 5 shows the assembly of the impeller and casing using CATIA V5R20.

Figure5:3D CAD model of impeller and casing assembly

4. Computational Fluid Dynamics

Table 2 shows the meshing details used in analyzing the CAD domain impeller model using ANSYS-Fluent software. Considering geometry complexities, unstructured tetrahedral meshing is preferred instead of structured hexahedral as shown in Figure 6 (a) and (b).

Table 2: meshing details

Model No.	*Mesh Type*	*Maximum Element Size Global(mm)*	*Minimum Mesh Quality*	*Number of Elements*
01	Tetrahedral	2	0.30	1099413

On applying boundary conditions and solving the following results have generated.

(a)Total view of impeller Mesh (b) Inside view of impeller mesh

Figure 6: Meshing of details

Table 3 shows the solver set up for analysis where the boundry conditions are provided.

Table 3: solver setup

<table>
<tr><th>Solver</th><th colspan="2">Selection</th></tr>
<tr><td>Type</td><td colspan="2">Pressure based</td></tr>
<tr><td>Time</td><td colspan="2">Steady</td></tr>
<tr><td>Velocity formulation</td><td colspan="2">Absolute</td></tr>
<tr><td>Cell Zones</td><td>V_CASING</td><td>V_IMPELLER</td></tr>
<tr><td>Operating Conditions</td><td colspan="2">Operating Pressure 101325 Pa</td></tr>
<tr><td>Turbulence Model</td><td colspan="2">RNG K-E</td></tr>
<tr><td>Pressure Velocity coupling</td><td colspan="2">SIMPLE Scheme</td></tr>
<tr><td rowspan="5">Spatial discretization</td><td colspan="2">Gradient- Green gauss Cell Based</td></tr>
<tr><td colspan="2">Pressure- Second Order</td></tr>
<tr><td colspan="2">Momentum- Second Order Upwind</td></tr>
<tr><td colspan="2">Turbulent K.E.- Second Order Upwind</td></tr>
<tr><td colspan="2">Turbulent dissipation Rate- Second Order Upwind</td></tr>
<tr><td>Convergence Monitor</td><td colspan="2">Equations Residual Checking Criterion (1e-5)</td></tr>
<tr><td>No of Iterations</td><td colspan="2">800</td></tr>
<tr><td>Reference Value</td><td colspan="2">Compute From- Water Inlet</td></tr>
</table>

Figure 7, 8 show the static pressure contours distribution on casing wall and impeller wall respectively.
Figure 9 shows velocity magnitude contours on impeller wall.
Figure 10, 11 show velocity vector Contours on casing wall and impeller wall respectively.
Figure 12 shows the flow path line of flowing fluid.

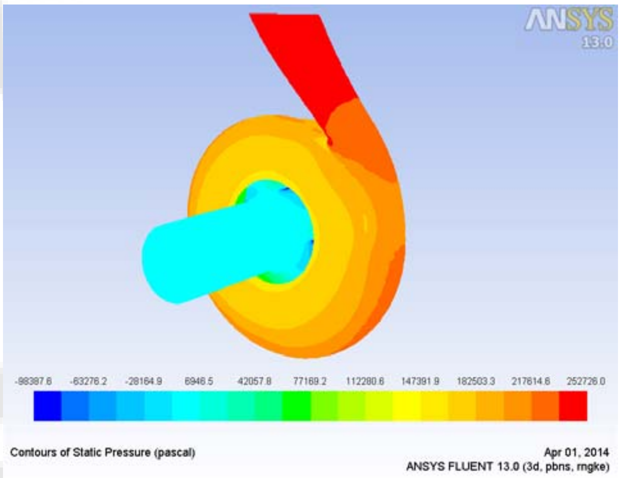

Figure 7: Contours of static presure on wall_casing

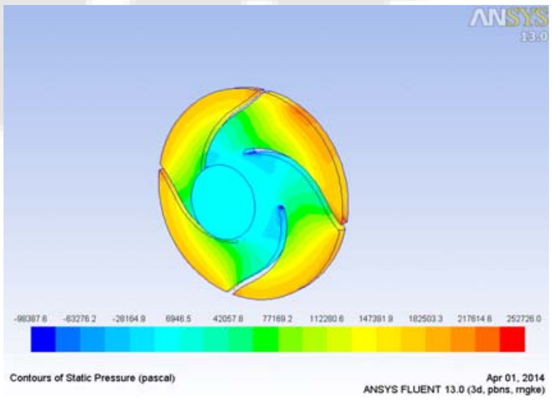

Figure 8: Contours of static presure on wall_impeler

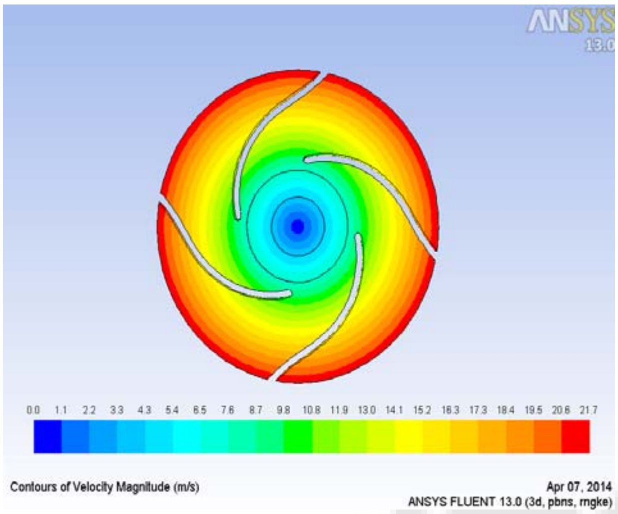

Figure 9: Contours of velocity magnitude on wall_impeler

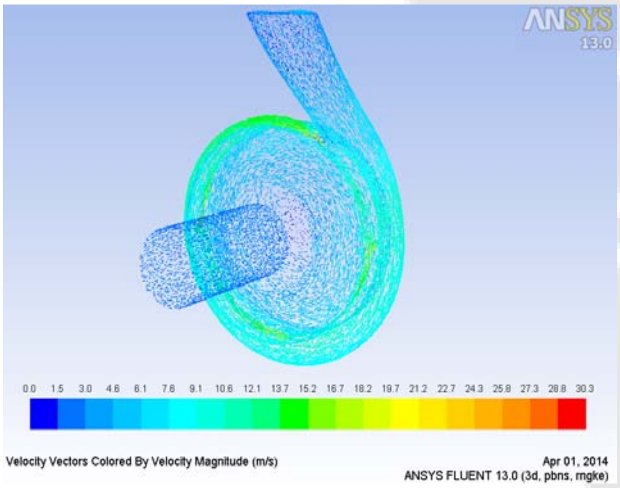

Figure 10: Contours of velocity vectores on wall_casing

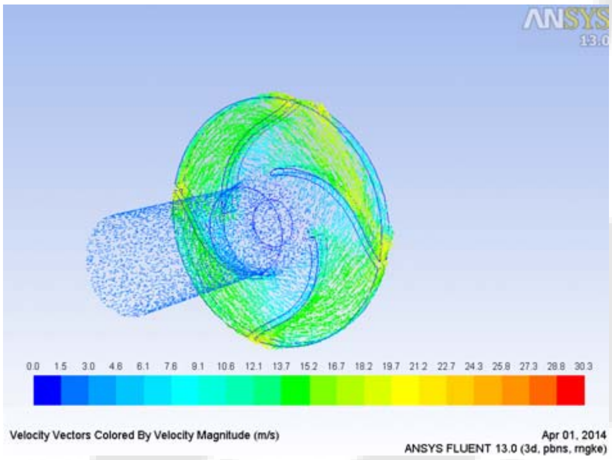

Figure 11: Contours of velocity vectores on wall_impeler

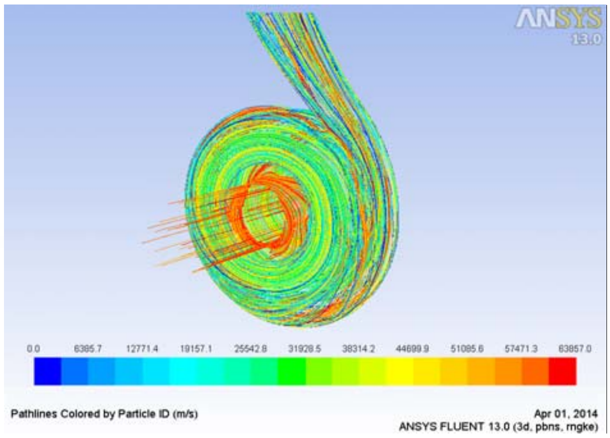

Figure 12: Flow pathline traced on wall_impeler

The result summery for this analysis is shown in Table 4. The analytical results show the redesigned impeller which can develop the head H of 26.65 m and discharge Q of 5 lit/sec at 2800 rpm speed.

Table 4: Result Summery

RESULT SUMMARY	
Area-Weighted Average Absolute Pressure	(pascal)
water_inlet	97179.008
water_outlet	358634.56
Net	261455.55
HEAD = 26.65 meter = 87.43 Feet	
Area-Weighted Average Static Pressure	(pascal)
water_inlet	-3145.9927
water_outlet	250009.55
Net	123636.86
Area-Weighted Average Velocity Magnitude	(m/s)
water_inlet	2.5042601
water_outlet	2.962451
Net	2.7337267
Mass Flow Rate	(kg/s)
water_inlet	5.0000029
water_outlet	-5.0000029
Net	2.2553741e-07
Discharge = 5 lit/sec	

5. Experimental Set-up

Figure 13 shows the experimental set up used for testing the pump performance with redesigned impeller (β_1= 45° and β_2= 50°)

Figure 13: Experimental Set up

In Figure 13,
1. Pump set up using designed impeller
2. Underground Water tank
3. Flow sensor for head measurement
4. Discharge pipe.

Figure 14 shows the control pannel for the experimental setup.

Figure 14: Digital Display board for experimental set up
1. Voltage, Current measurement meter
2. Slip measurement meter
3. Flow measurement meter

6. Experimental Results and Discussion

Table 5 and Figure 15 show the comparison of the results obtained using theoretical results, CFD results and experimental results.

Table 5: Comparison of results

Particulars	*Theoretical Results*	*Analytical Results*	*Experimental Results*
Discharge Q (lit/sec)	8.58 lit/sec	5 lit/sec	4.06 lit/sec
Head H (m)	31.08 m	26.65 m	24.44 m

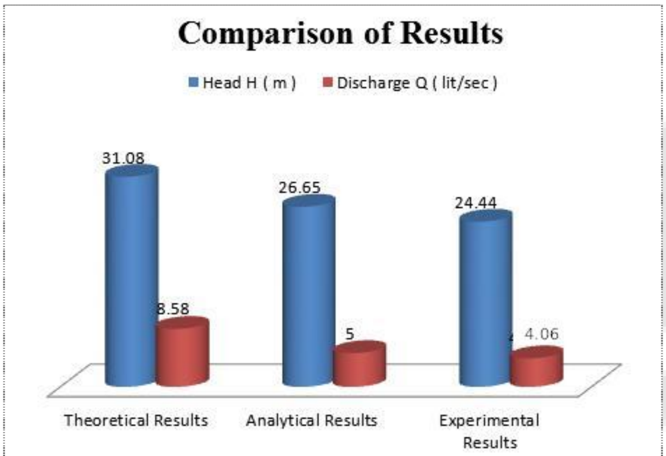

Figure 15: Graphical presentation of results

It is seen that the comparative results shows decreasing trends for head H and discharge Q developed by the redesigned impeller. It is the effect of fluid friction between casing and impeller and pipe.

7. Conclusion

A centrifugal pump impeller is modeled and analyzed for optimizing the performance for the rating of 3 hp at 2800 rpm using theoretical analysis and CFD analysis.
The redesigned pump is tested using standard test setup. The results obtained were compared for the respective trends. It is seen that the theoretical results show higher values when compared with experimental values. This is the effect of neglecting friction in theoretical analysis. There can be 10 % reduction in theoretical result of head due to the friction in casing and bents in pipe.

Using experimental results it is seen that for the same input rating by changing the impeller inlet blade angles from 25° to 45° and outlet angle from 27° to 50° the head (H) developed is increased for 22 meters to 24.44 meters and discharge (Q) is increased for 4 lit/sec to 4.06 lit/sec.

This is concluded that the CFD analysis can well predict the pump performance in design state. The authors are thank full to M/S. Torna Pumps, Kirloskarwadi, Sangli, Maharashtra for providing the problem for redesign of impeller and the cooperation extended for manufacturing, testing of redesigned impeller.

8. Future Scope

The design of an open well submersible pump is improved by varying inlet and outlet blade angle to improve the pump performance. The results obtained clearly demonstrate the achievement of the objectives as indicated above. This work can be extended in future with following aspects for further research.
1. Changing Material: There is major scope to change the material of pump, so that the losses of the pump will be reduces.
2. Skew cut: Skew cut one of the modern technique used for improving head. In this technique the backward vane profile cuts or trims at the periphery.
3. Coating to components: To reduce erosion–corrosion effects the coating technology is widely implement in the pump industries.

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