DESIGN AND CFD ANALYSIS OF SINGLE STAGE, END SUCTION, RADIAL FLOW CENTRIFUGAL PUMP FOR MINE DEWATERING APPLICATION

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ABSTRACT

Heavy centrifugal pumps are used in slurry and mine dewatering. While working these pumps are more subjected to the wear and consumes more power. Hence single stage end suction radial flow pumps are used for these applications. The pump performance is mainly depends on the vane shape, impeller and volute and as well as the supporting mechanical parts. Hence, design has been concentrated on the flow path of the fluid through the vanes and volute section. Since energy is transmitted to the fluid through vanes, so vane shape plays a very important role in effective energy transfer. The impeller and volute is been designed by Walter K Jekat method and Error triangle method given by Stepanoff A J., which is modified during this work by taking equal divisions and varying vane inlet angle form hub to shroud is used to generate the vane which has given an improved efficiency with reduced power consumption with compact design. The model prepared is been analyzed in CFD tool CF Design 2010 and its performance is analyzed at different flow rates. Major work is concentrated on vane shape.

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1. INTRODUCTION

A **pump** is a device used to move fluids, such as liquids, slurries etc., and displaces the fluid by physical or mechanical action. In simple words fluid enters a pump at certain velocity and pressure which may be even zero, and leaves it with increased energy, that is, velocity and pressure. For this, a pump consumes a certain amount of energy from any external source. There are rotating components inside which move the fluid either by confining it in definite volumes and then displacing it or by imparting energy to the fluid by dynamic action of the moving parts and increasing velocity and pressure of the fluid. Centrifugal pumps are the most common type of pump used to move liquids through a piping system. The fluid enters the pump impeller along or near to the rotating axis and is accelerated by the impeller, flowing radially outward or axially into a diffuser or volute chamber, from where it exits into the downstream piping system. Centrifugal pumps are typically used for large discharge through smaller heads.

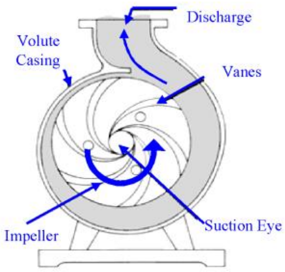


Fig 1: Liquid flow path inside a centrifugal pump

2. DESIGN OF CENTRIFUGAL PUMP

Centrifugal pump design is being done from last two centuries, for different application the design is modified to suit the application so as to obtain the highest efficiency. Here we are discussing the design which is used to design the centrifugal pump for mine dewatering application. The specialty of this design is it can be used other applications too without much changes and has flexibility in varying the parameters independently so as to achieve the maximum output for given input. The design of centrifugal pump is divided in two categories: Impeller Design and Volute Design

Following are the required Duty Parameters:

a) Head, H b) Flow Rate, Q c) RPM, N d) Density of the Fluid, ρ and e) Shear strength of shaft material, τ, f) Type of application, g) Particle size

*1. **Specific Speed of the pump, Ns:***

$$Ns = \frac{N\sqrt{Q}}{H^{\wedge}0.75} \quad (1)$$

2. ***Power Required, P:***

$$P = \frac{\rho Q g H}{\eta} \tag{2}$$

Where η is deduced from graph below based on specific speed calculated in step -1.

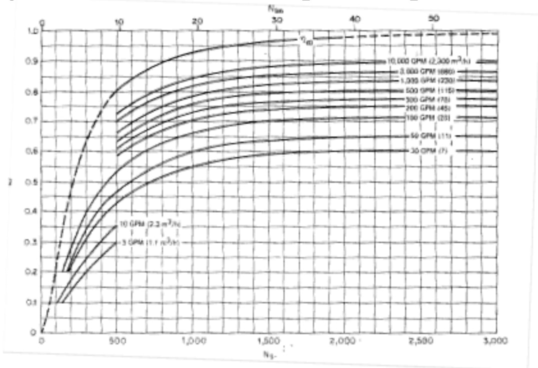


Fig 2: Efficiency versus Specific speed

3.***Blade outlet angle, β_2:*** Select the outlet blade angle at calculated specific speed.

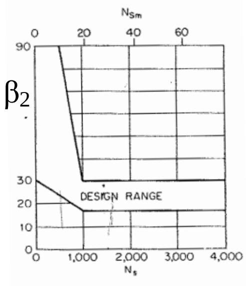
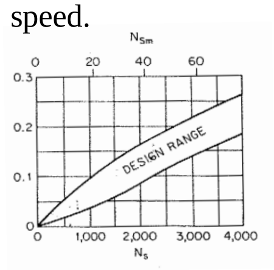


Fig 3: Blade outlet angle versus Specific speed

4.***Ratio of Outlet Meridional Velocity***, *C_{m2} to the Outlet Peripheral Velocity, U_2:* Select the appropriate ratio at calculated specific speed.



$$\frac{C_{m2}}{U_2}$$

Fig 4: Ratio of Outlet Meridional Velocity to the Outlet Peripheral Velocity versus specific speed

5.***Comparison of Slip Factor:*** Compare the slip factor calculated by Stodola's formula with the slip factor calculated by Pfleiderer's formula.

Stodola's formula: $\mu = 1 - \frac{\pi \sin \beta 2}{z}$ (3)

Pfleiderer's formula: $\mu = \dfrac{1}{1 + \frac{a}{z}\left(1 + \frac{\beta 2}{60}\right) \frac{2}{1 - \frac{r1^2}{r2^2}}}$ (4)

a = 0.65 i.e. Taking 65% of the flow in Volute, $z = \beta_2/3$

$r_1/r_2 = 0.5$ (Assumed value for radius ratio of inlet radius to the outlet radius) whichever value of μ is less consider that value

Note: Take low value of slip factor for further calculation.

6. ***Hydraulic Efficiency, η_H:***

$$\eta_H = 1 - \frac{0.071}{Q^{0.25}} \quad (5)$$

7. ***Head Coefficient***, *Ψ:* It is calculated to ascertain the effect of ratio of outlet meridional velocity to the outlet peripheral velocity and outlet blade angle on the total head.

$$\Psi = \frac{H}{U2^2/2g} = 2\,\mu\,\eta_H\left(1 - \frac{Cm2}{U2}\cot\beta_2\right) \quad (6)$$

8. ***Outlet Peripheral Velocity,*** *U_2:*

$$U_2 = \sqrt{(2gH/\Psi)} \quad (7)$$

9. ***Absolute impeller discharge velocity from velocity triangle:***

$$C_{u2} = U_2 - C_{m3}\cot\beta_2 \quad (8)$$

Considering Slip $C_{u2'} = \mu\, C_{u2}$

10. ***Impeller Diameter***, *D_2:*

$$D_2 = \frac{60 \times U2 \times 1000}{\pi N} \quad (9)$$

11. ***Outlet width of Impeller***, *b_2:*

$$b_2 = \frac{Q \times 10^6}{2\pi r_2 cm_2} \quad (10)$$

12. ***Inlet Meridional flow angle***, *β_o:*

This angle is always taken as 17 deg where NPSH is not critical i.e. NPSH > 3 meter. If NPSHr is below 3 m then the value is taken from 10 – 15 deg.

13. ***Suction Diameter***, *D_1:*

$$D_1 = 2897(Q/k\,N\tan\beta_o) \quad (11)$$

$k = \frac{\text{Hub diameter}}{\text{Suction Diameter}}$ Standard value = 1

14. ***Eye diameter,*** *De:*

$$De = (1 \text{ to } 1.5)\, D_1 \quad (12)$$

15. ***Inlet Impeller Width***, *b_1:*

$$b_1 = \frac{k}{4}\, Ds\, \frac{Ds}{D_1} \quad (13)$$

16. ***Mean Inlet Radius***, *r_{1m}:*

$$r_{1m} = \left\{\frac{r1^2 + rH^2}{2}\right\}^{0.5} \quad (14)$$

*17. **Check radius ratio**:*

$$\frac{r1m}{r2} < 0.5 \text{ (assumed in step 5)} \tag{15}$$

*18. **Inlet Vane angle**, β_1:*

$$\tan \beta_1 = \frac{\tan \beta o}{1 - \frac{zS_1}{2\pi r_1 \times \sin \beta_1}} \tag{16}$$

*19. **Mean Inlet Vane angle**, β_{1m}:*

$$\tan \beta_{1m} = \frac{\tan \beta o}{1 - \frac{zS_1}{2\pi r_1 \times \sin \beta_{1m}}} \tag{17}$$

*20. **Meridional velocity ahead of vanes**, C_{mo}:*

$$C_{mo} = \frac{Q \times 10^6}{\pi r_1^{\ 2}} \tag{18}$$

*21**. Inlet Peripheral Velocity**, U_1*

$$U_1 = \frac{\pi \times r_1 \times N}{30 * 1000} \tag{19}$$

*22**. Inlet area between vanes,** A_1:*

$$A_1 = \pi\, r_1^2 \sin \beta_{1m} \tag{20}$$

*23. **Discharge area between Vanes**, A_2:*

$$A_2 = b_2\,(2\,\pi\, r_2 \sin \beta_2 - z\, s_2) \tag{21}$$

*24. **Velocity Triangles**:*

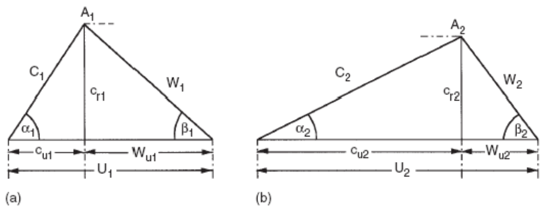


Fig 5: Velocity Triangle

Volute Design

The detailed procedure of single volute casing is as follows:

*1. **Ratio of Throat velocity to impeller outlet velocity**:*

Taken from the following graph

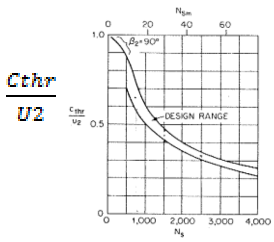


Fig 6: Ratio of volute throat velocity to the outlet impeller peripheral velocity

*2. **Volute Tongue Diameter**, D_t:*

$$D_t = D_2 + (0.07xR_2) \quad (22)$$

The increasing factor is taken 7% of impeller radius.

*3. **Volute Throat Area:***

$$A_{thr} = \frac{Q\times10^6}{Cthr} \quad (23)$$

*4. **Section Area at different angles**:*

$$A_v = A_{thr}\frac{\emptyset v}{360} \quad (24)$$

*5. **Volute Width**, b_3:*

Volute width is taken as 1.6 to 2 times the impeller outlet width, b_2.

*6. **Volute Casing Factor**, C:*

$$C = \frac{Cthr\times r_4}{C'u3\times r_2} \quad (25)$$

Note: The Volute Casing factor should be in the range from 0.9 to 1 because this factor assures the gradual change in sectional volute areas at required angles.

*7. **Sectional Volute velocity**, Cv:*

$$Cv = \frac{r2}{rv}\times C\ c'_{u3} \quad (26)$$

*8. **Sectional Area Check**:*

$$Av = \frac{Q\times10^6}{Cv}\ x\ \frac{\emptyset v}{360} \quad (27)$$

*9. **Sectional Area Pitch Circle radial distance from base circle**, a:*

$$a = \frac{b3}{2}\ \cot(\sigma/2) \quad (28)$$

σ for maximum efficiency it is taken as 40 or 45 degrees depending upon the impeller width. If Impeller width is less than 20mm then angle 40 is preferred and for more than 20mm width angle 45 is preferred.

*10. **Sectional Area Arc Radius**, r:*

$$Av = \frac{\pi r^2\sigma}{360} - \frac{b3^2\cot(\frac{\sigma}{2})}{4} \quad (29)$$

*11. **Delivery branch Diameter**, D_b:*

$$D_b = (\sqrt{(GPM/20)})\ x\ 25.4 \quad (30)$$

GPM = Volume flow rate in gallons per minute.

*12. **Angle of divergence**, δ_o:*

$$\tan\delta_o/2 = \frac{Db-Dthr}{2Ds} \quad (31)$$

*13. **Overall Pump Efficiency**, η:*

$$\eta = \frac{1}{\eta_V \eta_H} + \frac{P_{DF}}{P_W} + \frac{P_M}{P_W} \tag{32}$$

*14. **Power Required**, P_s:*

$$P_s = \frac{1}{\eta} \times 9.81 \times Q \times H \times 100 \tag{33}$$

Volute Layout

Area Division Method is used to construct the volute, trapezoidal section is used to form the volute profile. In this method the throat volute area (considering 5% blockage due to corner radii) is divided into no of desired sections. The radius of arc for each section is given by equation 40, and 41.

3. CASE STUDY (WITH MODIFIED ERROR TRIANGLE METHOD)

The centrifugal pump design methodology as discussed above is used to design a pump of following duty parameters. The equations are used to calculate the dimensions of the pump. The duty parameters are:

1. Head = 200 m, 2. Flow rate = 500 m^3/hr, 3. RPM = 1800, 4. Particle Size = 20 mm
5. Density = 1000 Kg/ m^3, 6. Given Shaft size = 76 mm

The calculated parameters are as follows:

1. Specific Speed, N_s = 12.6
2. Overall efficiency = η = 73% (from fig 2)
3. Power Required = P = 373.3 KW
4. Blade outlet angle, β_2 = 27 deg (from fig 3)
5. Ratio of Outlet Meridional Velocity, C_{m2} to the Outlet Peripheral Velocity, U_2= 0.025 (from graph 3)
6. Comparison of Slip Factor
   a. Stodola's Formula = μ =0.714
   b. Pfleiderer's formula = μ = 0.665

   Taking lower value = 0.665
7. Hydraulic Efficiency, η_H = 88.4 %
8. Head Coefficient ψ = 1.117
9. Outlet Peripheral Velocity, U_2 = 59.25 m/s
10. Meridional Outlet velocity = Cm_2 = 1.48 m/s
11. Absolute impeller discharge velocity = Cu_2 = 56.34 m/s
12. Impeller Diameter, D_2 = 630 mm

13. Outlet width of Impeller, b_2 = 47 mm
14. Suction Diameter, D_1 = 200 mm
15. Inlet Impeller Width, b_1 = 60 mm
16. Mean Inlet Radius, r_{1m} = 82.8 mm
17. Radius ratio = 0.26
18. Inlet Vane angle, β_1 = 20.5 deg
19. Mean Inlet Vane angle, β_{1m} = 21.5 deg
20. Meridional velocity ahead of vanes, C_{mo} = 4.42 m/s
21. Inlet Peripheral Velocity, U_1 = 18.84 m/s
22. Inlet area between vanes, A_1 = 11514 mm^2
23. Discharge area between Vanes, A_2 = 38471 mm^2
24. NPSHr = 6.4 m
25. Overall pump efficiency = 73.3%
26. Power required = 372 KW

Volute Casing:

1. Ratio of Throat velocity to impeller outlet velocity = 0.75
2. Throat Velocity = c_{thr} = 44.4 m/s
3. Volute Tongue Diameter, D_t = 675 mm
4. Volute Throat Area = 3123.5 mm^2
5. Volute Width, b_3 = 70 mm
6. Delivery branch size = 152 mm
7. Angle of divergence = 25.07 deg

Table 1: Error triangle values for case study

A-A'	R	H	Phi	B-B'					C-C'		
	287.75	53.972	10.75		287.501	53.972	10.75		287	54.953	
	260.26	56.206	12.37		260.002	56.383	12.42		259	57.59	10.9
	232.76	580592	14.42		232.503	58.974	14.53		231	60.44	12.7
	205.27	61.18	17.1		205.004	61.766	17.26		203	63.527	15
	2E+06	63.897	20.59		177.505	64.786	20.9		175	66.89	17.9
	150.27	66.862	25.5		150.006	68.065	26		147	70.565	21.9
	122.78	70.067	32.69		122.507	75.556	34.5		119	74.667	27.5
	100.17	73.552	42.1		95.061	79.866	46.1		90.08	79.07	35.1

In this error triangle method is used to generate the vane. First modification is taking equal divisions for all the three profiles i.e., for hub shroud and Meridional, secondly by varying the vane inlet angle from hub to shroud, thirdly by varying the plane angles between the planes. This gives the increased vane area which in turn transfers more energy to the fluid.

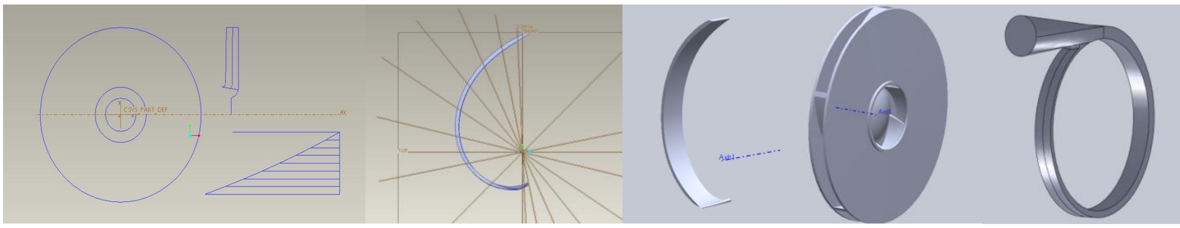


Fig 7: Vane, Impeller, Volute

	Throat Area=	3000 mm^2		
	Base circle dia=	650 mm		
	theta	C S Area	r	R
1	15	125	1.7	327
2	45	375	5	330
3	90	750	10	335
4	135	1125	15	340
5	180	1500	20	345
6	225	1875	25	350
7	270	2250	30	355
8	315	2625	35	360
9	360	3000	40	365

Table 2: Volute Area division values

4. CFD ANALYSIS

Material is assigned to the parts of the pump as

1. Casing and Impeller: Aluminium alloy
2. Hydraulic Region: Water
3. Rotating part: Rotating region

Boundary Conditions:

Boundary conditions are applied to the inlet and outlet of the pump i.e. 0 pa at inlet, 500 m^3/hr at outlet, and 1800 RPM.

CFD Results

After analysis is been carried out the following results are obtained. The results are taken only when the convergence is obtained for the solution.

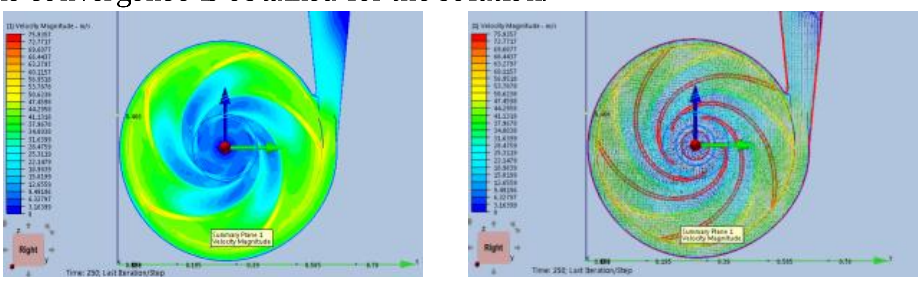


Fig 8: Velocity Profile

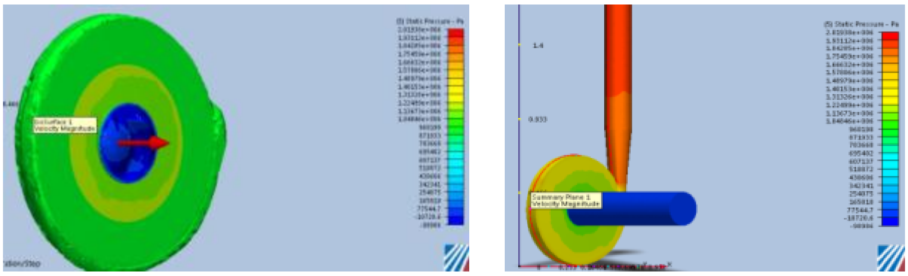
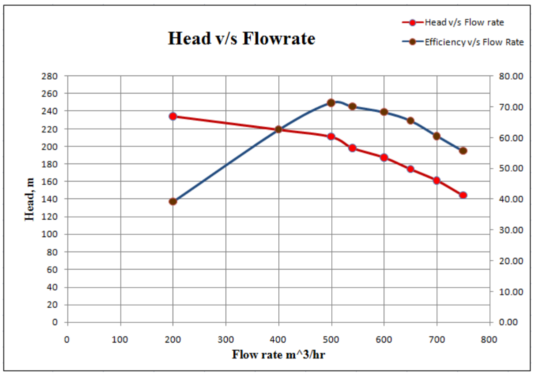
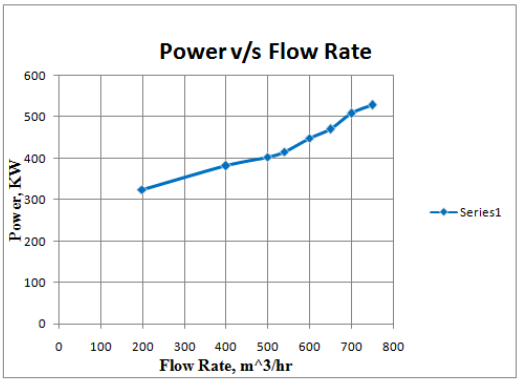


Fig 9: Static Pressure distribution

Flow Rate, Q m^3/hr	Head, m	Torque, N-m	Power, KW	Hydraulic Power, KW	Efficiency, %	
200	234	1725	325.197	127.53	39.22	
400	218	2025	381.753	237.62	62.24	
500	207	2140	403.4328	282.0375	71.26	BEP
540	198	2206	415.87512	291.357	70.06	
600	187	2378	448.30056	305.745	68.20	
650	174	2495	470.3574	308.1975	65.52	
700	161	2695	508.0614	307.1075	60.45	
750	144	2805	528.7986	294.3	55.65	

Table 3: Performance Data





Graph 1: Head and Efficiency v/s Flow Rate, Graph 2: Power v/s Flow Rate

COMPARISON

The comparison of the velocity profile and static pressure profile is shown in the below figures. It shows that modified design results are close to non modified results. Although some recirculation if found in the delivery branch but still the performance is high in modified design. Because vane area along the length is increased due to which more transfer of energy takes place and hence more efficiency.

Without modification Design CFD Result *With modification Design CFD Result*

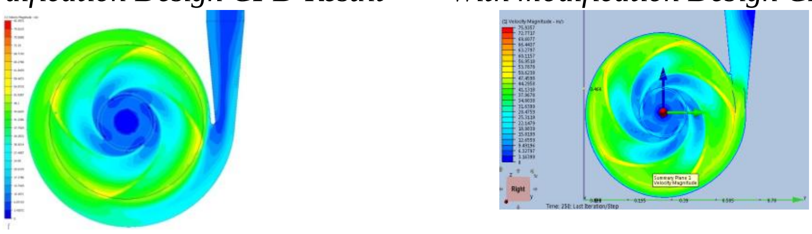
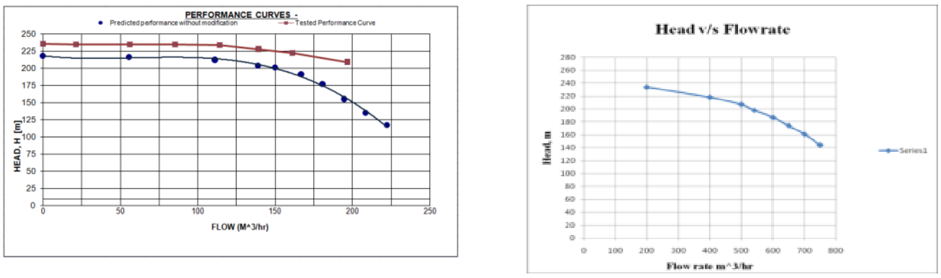


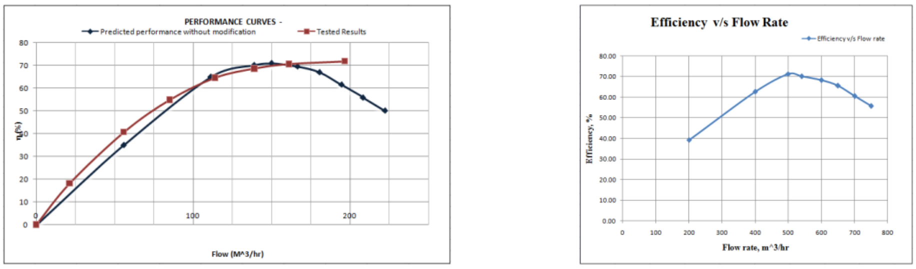
Fig 10: Velocity Profile comparison

Fig 11: Static pressure Profile comparison

The tested result graph and modified design analysis results graph for both head v/s flow rate and efficiency v/s flow rate has given the same pattern.



Graph 3: Performance curves Head v/s Flow rate comparison with Modified method



Graph 4: Performance Curve Efficiency v/s Flow Rate Comparison

Below table gives the complete comparison between the modified design and non modified design with the tested results. **The tested results are of the pump which is manufactured earlier without vane modification technique.** Here we see if we use modified vane method the pump performance can be increased and can reduce power consumption.

	CFD Results without modified method	Tested Results	CFD Results for Modified method
Specific Speed	12.6	12.6	12.6
Head, m	212	228	210
Flow Rate, m^3/hr	501.23	500.65	500.8
Power, KW	426	454	403
Efficiency, %	68	67.5	71.2

Table 4: Table of Comparison

5. CONCLUSION AND FUTURE WORK

The mine dewatering pump has designed by Walter K Jekat method and analyzed with CFD. It is observed that there is no much deviation with Walter K Jekat method, CFD and the experimental tested results.

In the comparison with tested results both the design procedure and CFD results are validated, the design and analysis has given the results with minimum error. This shows that design is good in increasing overall efficiency of mine dewatering pumps. From this work it can be concluded that, design of mine dewatering pumps can be done economically, efficiently and precisely for all three ranges of pumps *i.e.* low, medium and high specific speed pumps. This design procedure is flexible and helps the new designer to understand the effect of each parameter in mine dewatering pumps and gives more choices and flexibility to the experienced designer.

The method used for vane generation has given more flexibility to design the vane according the required fluid flow, this result in increase in efficiency due to a little twist provided at the leading edge of the vane by varying the leading edge angle from the hub to shroud. The comparison shows that there increase in efficiency achieved with reduction in power required. This comparison concludes that small modification in the vane can give very good results.

The future work can be carried out on the parameters observed during the analysis and still further performance of the pump can be increased. The work can be carried out on slurry pump used in *cement industry, sewage transport* and also where low specific speed pumps are used. The method can also be applied on *double suction* and *double volute* pumps (used in oil refineries) and their performance can be enhanced.

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