Study of Stresses Developed on the Impeller of Centrifugal Pump at Different Speed using Ansys

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***Abstract*— Pump is a mechanical device generally used for raising liquids from a lower level to a higher one. Centrifugal Pumps are the machines, which employ centrifugal force to lift water from a lower level to a higher level by developing pressure. Computational fluid dynamics (CFD) analysis is being increasingly applied in the design of Centrifugal Pumps. CFD is an important tool for pump designers. This paper is mainly focused on design development of pump from customer requirement at initial design phase and FEA analysis of Centrifugal Pump using ANSYS for checking stresses and deformation from point of safe design. The rotating device i.e. Impeller has basic issues of failure at different speed and forces. As a case study, pilot project with research methodology in the preliminary stage of design of Centrifugal Pump is considered. The Pump dimensions are calculated according to the customer requirements of input parameters like head, discharge, and speed. Stress analysis of Impeller of Centrifugal Pump at different speed is done randomly to test design is safe or not.**

Keywords*— Stress analysis, CFD, FEA, ANSYS, Centrifugal Pump.

I. INTRODUCTION

Computational fluid dynamics (CFD) is the analysis of the system involving fluid flow, heat transfer and associated phenomena such as chemical reactions by means of computer based simulation. With the aid of the CFD approach, the complex internal flows in the impellers, which are not fully understood yet, can be well predicted; to speed up the pump design procedure.The prediction of behavior in a given physical situation consists of the values of the relevant variables governing the processes of interest. To obtain an approximate solution numerically, we have to use a discretization method which approximates the differential equations by a system of algebraic equations, which can then be solved on a computer. The approximations are applied to small domains in space and/or time so the numerical solution provides results at discrete locations in space and time. Much as the accuracy of experimental data depends on the quality of the tools used, the accuracy of numerical solutions is dependent on the quality of discretization used. CFD is used in pump design industry to analyze the effect of various forces, vibrations, fluid pressure and its effect. Optimization in the existing design of pump can produce large savings in material costing of pump with increase of head & efficiency.

II. PROBLEM DESCRIPTION

Impeller is very important part of Centrifugal Pump and it is experiencing various loads like fluid pressure, inertia, unbalanced forces, and vibrations during rotation. Excessive stresses may lead to development of cracks in the impeller and ultimately failure of impeller. Hence it is essential to analyze the stresses developed in the impeller.

III. SCOPE OF WORK

ANSYS Workbench which is a Finite Element Analysis (FEA) software used in the numerical simulation of the centrifugal impeller with the finite element method. The equivalent stress distribution of the impeller, which caused by the centrifugal load, the coupling effect of centrifugal load and aerodynamic load, is analysed. Based on the fluid components transport theory, the distribution of the flow field inside the impeller is analysed, and discussed the influence of the stress distribution. Analytical design &CFD analysis of Pump at 1800 rpm and validation of an impeller design using stress analysis at various speed such as 1050 rpm, 1250 rpm, 1450 rpm is analysed. Post processing shows the stress and deformation results at each node. This is analysed to check whether the results are in safe limit or not.

IV. DESIGN OF CENTRIFUGAL PUMP

A. Customers requirement for Pump specification

Head (H) = 13.8 m, Flow rate (Q) = 16 lit /s ,Speed (n) = 1800 rpm, Fluid=water, Pump type =Radial

B. General dimensions of impeller are calculated as below

1.Specific speed $n_s = n x \frac{\sqrt{Q}}{H^{\frac{3}{4}}}$

$=1800 x \frac{\sqrt{0.016}}{13.8^{3/4}}$

= 31.79 rpm

2.Nominal diameter $D_1 = 4.5 \text{ x } 10^3 \text{x} \sqrt[3]{\frac{Q}{n}}$

$= 4.5 \text{ x } 10^3 \text{ x} \sqrt[3]{\frac{Q.016}{1800}}$

= 93.1 mm

3.Hydraulic efficiency, $\eta_h = 1 - \frac{0.42}{(\log D1 - 0.172)^2}$

= 87%

4.Volumetric efficiency ,$\eta_v = \frac{1}{1+0.68(ns)^{-2}/3}$

=93.72%

5.Assuming mechanical efficiency ,η_m= 0.96

6.Overall efficiency, $\eta = \eta_h \times \eta_v \times \eta_m$

= 78.27%

7.Output Power, No = γQH / const.

= (9.81 x 1000 x0.016 x 13.8) / 1000

= 2.17 KW(2.91hp)

8.Input power, Ni =$\frac{NO}{\eta}$

= 2.7468 KW(3.68hp)

9.Assuming an overload of 15% ,

input power (Ni) = 2.74 x 1.15

= 3.15 KW(4.22 hp)

10.Torque, T =$\frac{Ni}{\omega} = \frac{3.15 \times 60}{2 \times 3.14 \times 1800}$

= 0.0167 KN-m

11.Taking the shaft material as EN8, Ultimate stress (f_m) as 35 N/mm^2,and taking factor safety (FS) as 2 for uniform speed of rotation.

12.Working stress(f_s)=$\frac{fm}{FS}$

= 17.5 N/mm^2

13.Shaft diameter, $d_s = \sqrt[3]{\frac{16T}{3.14fs}}$

= 0.01687 m= 17 mm

Taking fatigue stress (bending and shear) into account, minimum shaft diameter (d_s)is taken as 25 mm, d_s = 25 mm

14.Hub diameter, $d_h = 1.25\, d_s$ = 1.25 x 25 = 30 mm.

C. Inlet dimensions are calculated as below

1. Theoretical discharge, $Q_{th} = Q / \eta_v$

   = 0.016 / 0.9372 = 0.0170 m^3/s

2.Eye diameter of impeller (D_o)is taken as 76.4mm ,

the axial velocity at an impeller eye (C_o) is

$C_o = \frac{4Qth}{3.14\ ds^2}$

= 3.7 m/s

3.The diameter of the inlet edge of impeller blade (D_1)is taken as 90 mm

Inlet breadth (B_1) =$\frac{Qth}{3.14D1Cm1}$

= 25.8 mm (Considered as 26 mm)

4.Taking K_1 = 1.4 ,

$C_{m1} = K_1 \times C_{mo}$

= 1.4 x 2.26

= 3.164 m/s

5.$u_1 = \frac{3.14D1n}{60}$

= 8.5 m/s

6.Inlet blade angle (β_1)will be calculated as,

$\beta_1 = \tan^{-1}(Cm1 / u1) = \tan^{-1}(3.164/8.5)$

= 20.42^0(taken as 25^0)

D. Outlet dimensions are calculated as below

1.Manometric head(H_m) ,

$H_m = \frac{H}{\eta h}$

= 17.23 m

2. First approximation of u_2 taking, C_{u2} = 0.5

First approximation of u_2,

$u_2 = \sqrt{\frac{gHm}{Cu2}}$

= $\sqrt{9.81\ x\ 17.23/0.5}$

= 18.45 m/s

3. D_2 1st approximation,

Outside diameter(D_2),

$D_2 = \frac{60u_2}{3.14n}$

= 196 mm

4.Taking, $C_{m3} = 0.8 \times C_{mo}$= 0.8 x 2.26 = 1.81 m/s

5.Taking K_2 = 1.2 and w_1 / w_2 = 1.18

$\sin\beta_2 = \sin\beta_1 \frac{K2w1Cm3}{K1w2Cm0}$ = 0.3419

Outlet blade angle, $\beta_2 = 20^0$

6.Outlet flow velocity(C_{m2}),

$C_{m2} = 0.687 \times C_{m1}$

= 0.687 x 3.164

= 2.18 m/s

7.No of blades , Z = 6.5 x $\frac{D2+D1}{D2-D1}$ x sin $\frac{\beta1+\beta2}{\beta2}$

= 6.01,Z is taken as 6

8.Assuming p as 0.2915

$H_o = (1 + p) H_m$ = (1+0.2915) 17.23 = 22.26 mm

9.Second approximation of u_2,

Outlet blade velocity, $u_2 = \frac{Cm2}{2\tan\beta2} + \sqrt{\frac{Cm2}{2\tan\beta2} + gH_m}$

= 18.07 mm/s

10. D_2 2nd approximation,

Outside diameter (D_2) $= \frac{60u_2}{3.14n}$

= 0.192m

D_2 1st approximation (D_21^{st} = 196 mm) and D_2 2nd approximation (D_2 2nd = 192mm) closely agrees. Final value of outer diameter D_2 is taken as 200 mm

11.$C_{m3} = \frac{Cm3}{K2}$ = 1.82 m/s

12.Outlet breadth, $B_2 = \frac{Qth}{3.14D2Cm3}$ = 15mm

E. Verification of flow coefficients are calculated as below

1.$K_1 = 1/(1-\frac{Z\delta1}{3.14D1Sin\beta1})$ = 1.414

2.$K_2 = 1/(1-\frac{Z\delta2}{3.14D2Sin\beta2})$ = 1.195

3. $W1 = \frac{Cm1}{\sin\beta1}$ = 7.49 m/s

4.$W_2 = \frac{Cm3}{\sin\beta2}$ = 6.37 m/s

5.$\frac{W1}{W2}$ =1.18

F. Volute Casing design steps are as below

1.Inlet width of volute, $B_3 = 1.8\ B_2 = 1.8 \times 17 = 30.6$ mm

2.Outlet width of volute, $D_3 = 1.15\ D_2$

$= 1.15 \times 0.192 = 220.8$ mm

3.$R_v = \frac{D2+D3}{4}$ = 0.10

4.Width at X distance $B_x = B_3 + \frac{2Rv-D2}{1.73}$ = 30.60 mm

G. Dimensions of pump according to customer requirements are calculated and listed below

Specific Speed (Ns) =31.79 rpm, Pump type=Radial

Torque 0.0167 KN-m, Hydraulic efficiency =87 % ,

Volumetric efficiency =93.72 % ,Overall efficiency =78.27 % ,

Input power 3.15 KW(4.22hp) ,Output power 2.17 KW

Shaft diameter (ds)=17 mm, Hub diameter= 30 mm
Eye diameter (D_o) =76.4 mm, Inlet diameter (D_1) =90 mm
Outlet diameter (D_2)= 192 mm ,Number of blades (z)= 6
Inlet breadth (B_1)= 26 mm, Outlet Breadth (B_2) =15 mm
Blade velocity (u_1) =8.5 m/s, Blade velocity (u_2) =18.45 m/s
Blade angle (β1) =25^0 ,Blade angle (β2) =20^0

V DETAILS OF FEA

The FEA analysis is carried out randomly at different speed of 1050 rpm,1250 rpm and 1450 rpm for testing whether theoretical design calculation of pump dimension are safe or not. The assumptions made in FEA are given in below Table 1.

Table 1. assumptions made in FEA

Parameters	Values
Fluid-	Water
Material	Mild Steel
Young's Modulus	1.1e+005MPa
Poisson's Ratio	0.28
Structural Density	7.2e-006 Kg/mm^3
Thermal Expansion	1.1e-005 1/ °C
Tensile Yield Strength	240 MPa
Ultimate Tensile Strength	840 MPa
Specific Heat	875J/Kg °C
Thermal conductivity	5.2e-002 W/mm °C

A. Geometry of Impeller Model–

The model of impeller and casing is made in CATIA and imported in ANSYS for deformation and stress analysis.

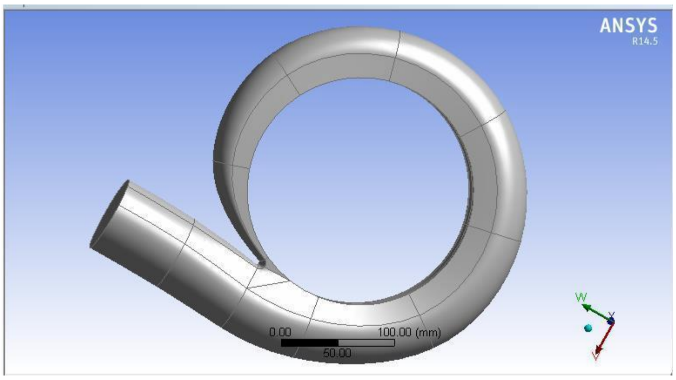

Fig. 1 Model of Casing

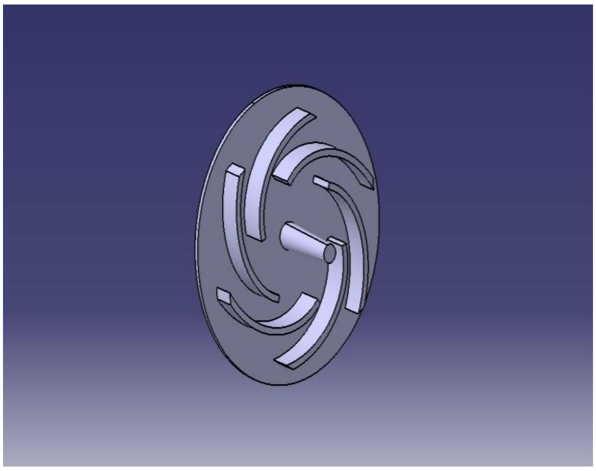

Fig. 2 Model of Impeller

B. Mesh Model

The impeller blade and casing model is meshed and hexa mesh is obtained by giving body sizing and edge sizing.

ISSN: 2278-0181
ICIATE - 2017 Conference Proceedings

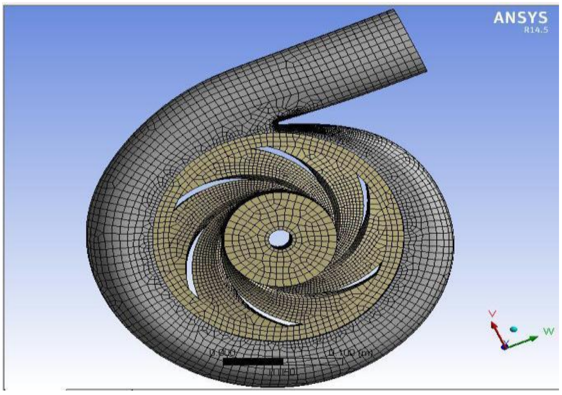

Fig. 3 Meshing of Pump

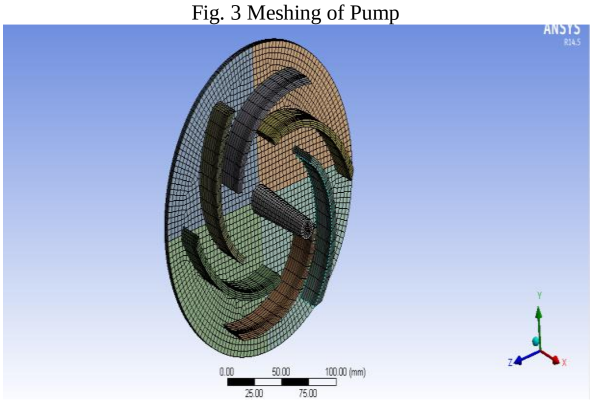

Fig. 4 Meshing of Impeller

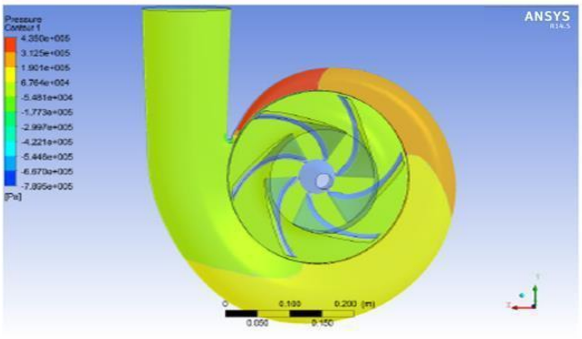

Fig. 5 Pressure distribution

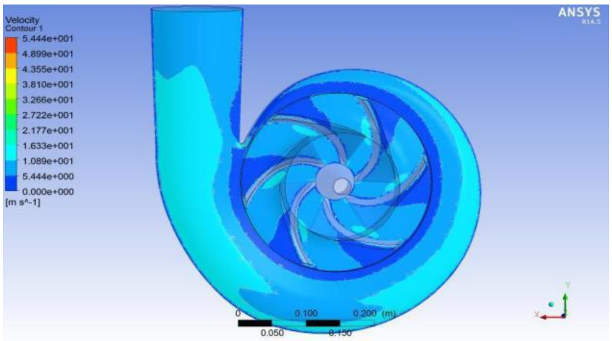

Fig. 6 Velocity distribution

C. Boundary Conditions –

Using the following boundary conditions static structural analysis is performed.

Case 1- Boundary Condition With 1050 rpm-

As per static structural analysis the Rotational velocity is applied to all bodies is 1050 RPM , Fixing condition is applied as Remote displacement that is Rotation in X direction is free and other Degree of freedom is fixed. The other end of the shaft is fixed. This remote displacement support is applied on the circumference of the one end of the shaft.

- Rotational Velocity-

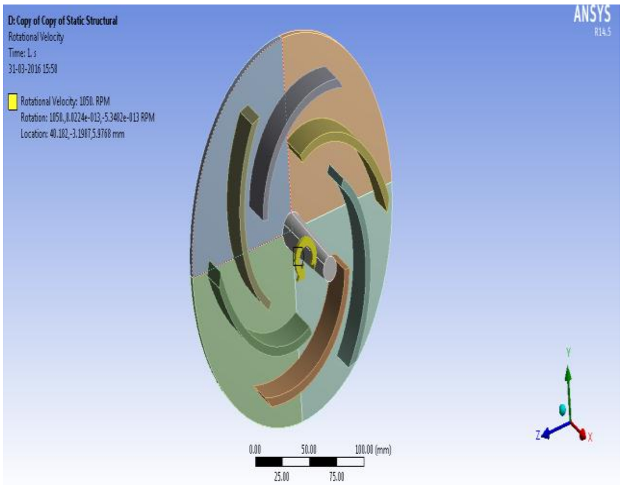

Fig. 7 Rotational velocity at 1050 rpm

- Pressure at suction side – 9.45 x 10^{-2} MPa

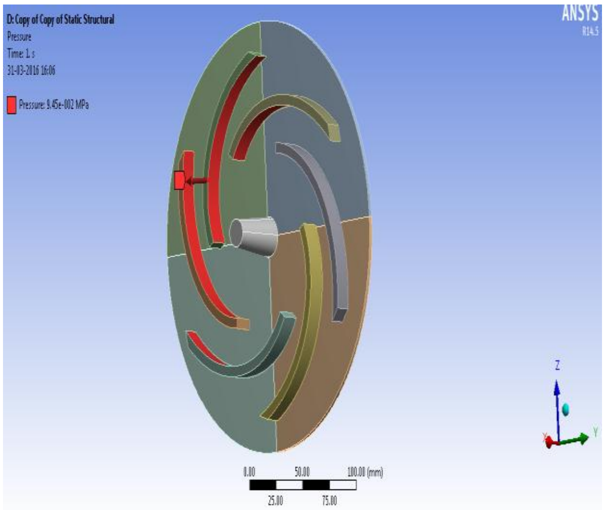

Fig. 8 Pressure at suction side at 1050 rpm

- Pressure outer side – 6.66 x 10^{-2} MPa

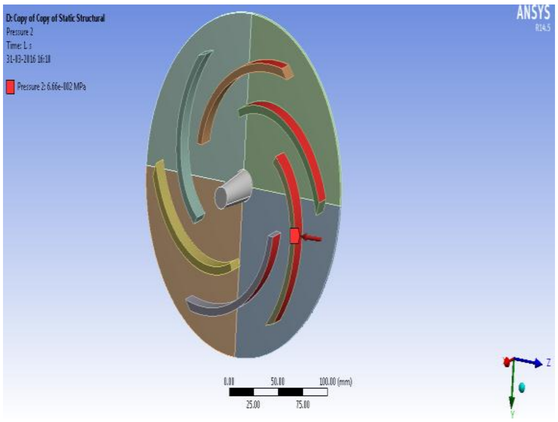

Fig. 9 Pressure at discharge side at 1050 rpm

- Pressure on top side – 0.15 MPa

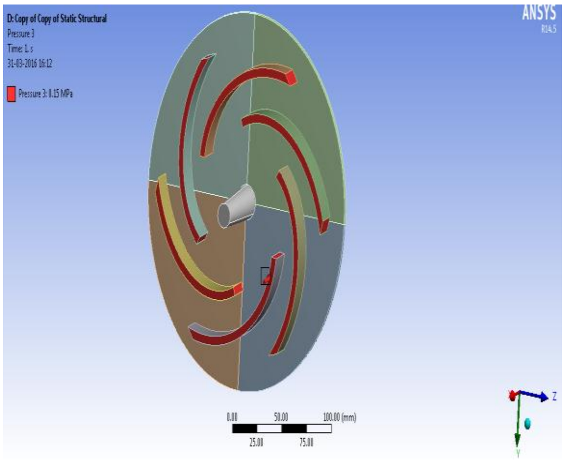

Fig. 10 Pressure at top side at 1050 rpm

- Remote Displacement –

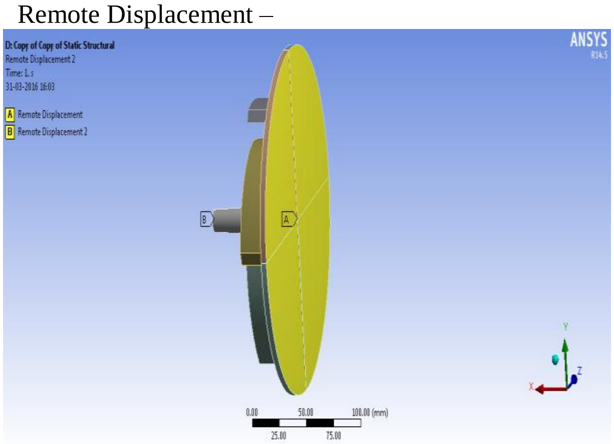

Fig. 11 Remote displacement at 1050 rpm

- Standard Earth Gravity for self-weight –

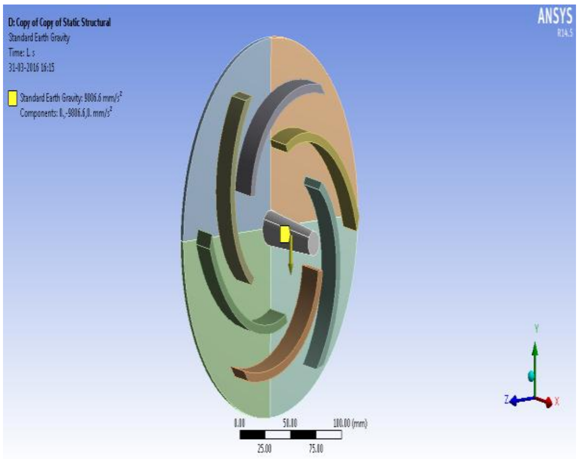

Fig. 12 Standard Earth Gravity at 1050 rpm

Results –

- Total Deformation-

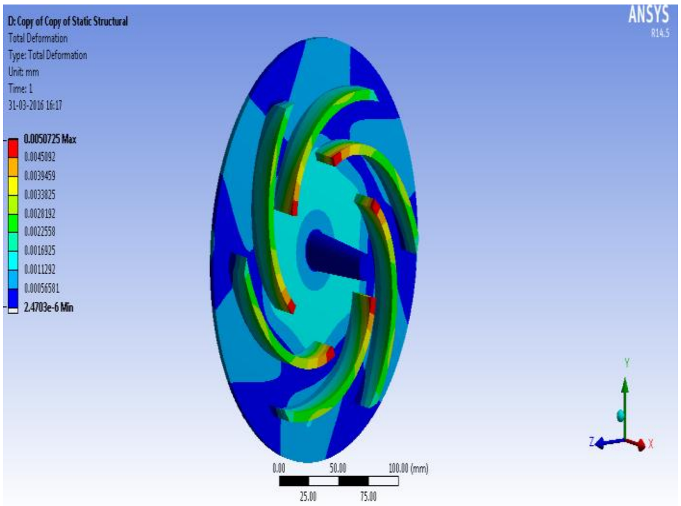

Fig. 13Total deformation at 1050 rpm

- Equivalent Von-Misses Stress –

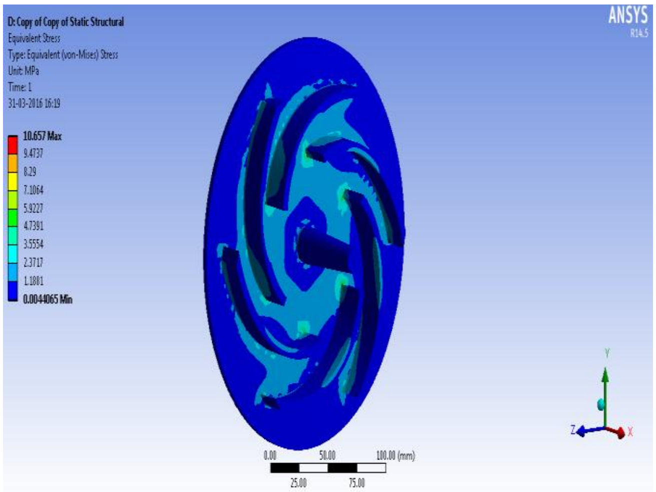

Fig. 14Von-Mises stress at 1050 rpm

Observations-

- The maximum total deformation of 0.005 mm is within safe limit and can be considered safe.
- The maximum Von Misses stress value of 10.65 MPa is less than the allowable limit and hence the impeller can be considered as safe.

Case 2- Boundary Condition With 1250 rpm-

For case 2 all the boundary conditions are similar as case 1 except the rotational velocity which is 1250 rpm.

- Rotational Velocity- 1250 rpm

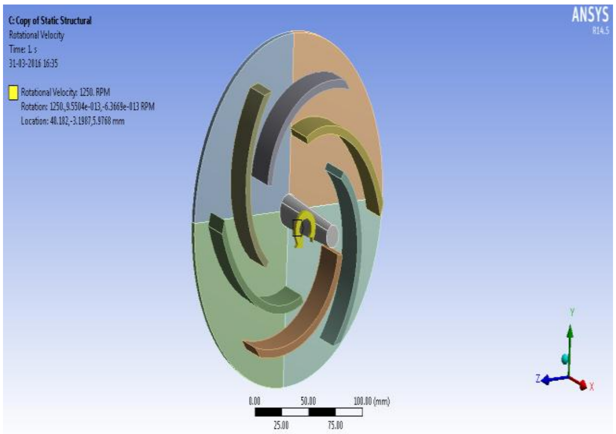

Fig. 15 Rotational Velocity at 1250 rpm

- Results –

Total Deformation –

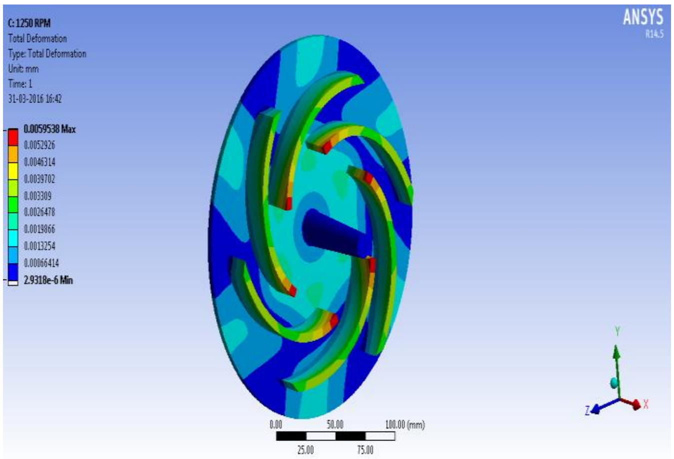

Fig. 16 Total deformation at 1250 rpm

- Equivalent Von-Misses Stress –

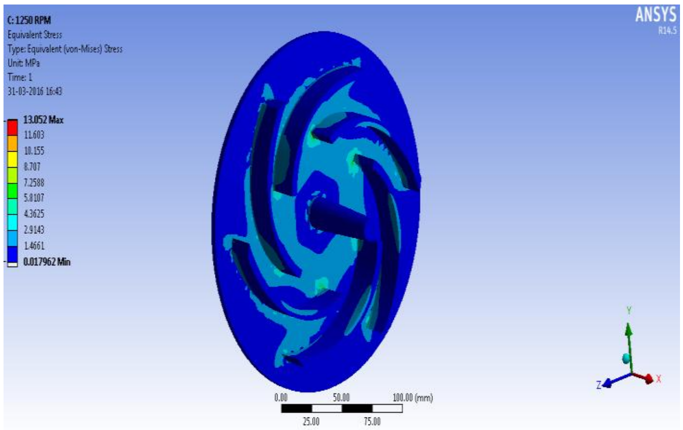

Fig. 17Von-Misesstress at 1250 rpm

Observations-

- The maximum total deformation of 0.0059 mm is within safe limit and can be considered safe.
- The maximum Von Misses stress value of 13.052 MPa is less than the allowable limit and hence the impeller can be considered as safe.
-

Case 3. Boundary Condition With 1450 rpm-

For case 3 all the boundary conditions are similar as case 1 except the rotational velocity which is 1450 rpm.

- Rotational Velocity at 1450 rpm-

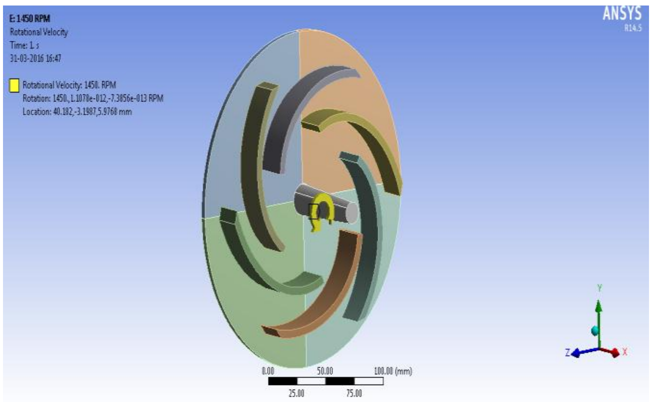

Fig. 18 Rotational Velocity at 1450 rpm

Results-

- Total Deformation-

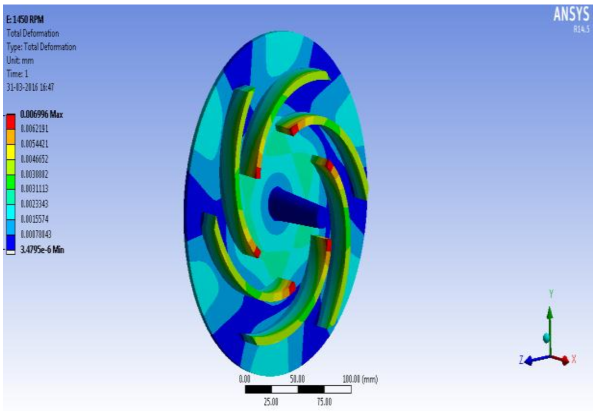

Fig. 19 Total deformation at 1450 rpm

- Equivalent Von-Misses Stress –

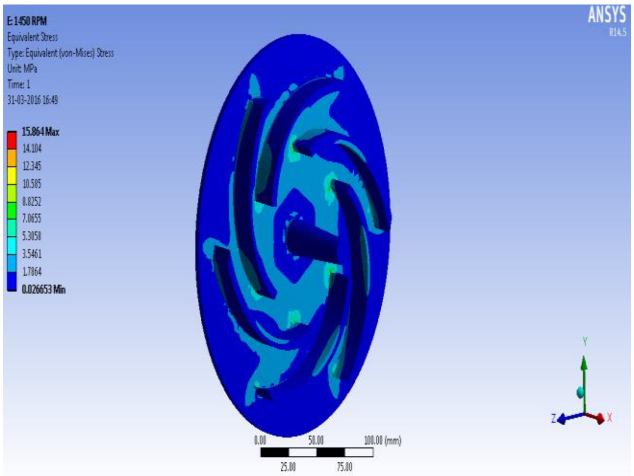

Fig. 20 Von-Misesstress at 1450 rpm

Observations-

- The maximum total deformation of 0.0069 mm is within safe limit and can be considered safe.
- The maximum Von Misses stress value of 15.864 MPa is less than the allowable limit and hence the impeller can be considered as safe.

VI CONCLUSIONS

Stress analysis of impeller is done at different speeds. It is found that the impeller blade is safe under rotational velocity of 1050 rpm, 1250 rpm and 1450 rpm .The results show that stresses increases with speed, but are still under safe limit.

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