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Fluid Structural Analysis of Centrifugal FAN Using FEA

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ABSTRACT
In this thesis the performance of centrifugal fans having blades of Airfoil section, forward blades is investigated and compared. Theoretical calculations are done to design the impeller blade at different speeds 2800rpm, 3000rpm, 3200rpm and also the efficiencies at 2800rpm are calculated.CFD analysis is performed on the fan to determine outlet pressures, velocities and mass flow rates by changing boundary condition inlet velocity. Static analysis is done on the fan by taking pressures from CFD analysis as boundary condition. Different materials Steel, Aluminium alloy, E – Glass Epoxy and Aramid Fiberare considered for the analysis where deformations and stresses are determined.3D model of the centrifugal fan is done in Creo 2.0 and CFD analysis is performed in Ansys.

INTRODUCTION
Fans and blowers give air for ventilation and requirements of industrial processes. Fans generate a pressure to maneuver air (or gases) against a resistance caused by ducts, dampers, or different parts in a very fan system. The fan rotor receives energy from a shaft and transmits it to the air.

Centrifugal Fan: Types
The major kinds of centrifugal fan are: backward curved, radial and forward curved. Due to their high static pressures (up to 1400 mm WC) and skill to handle heavily contaminated airstreams radial fans areindustrial workhorses. Due their simple design, radial fans are suited well for medium seeds at blade tip and high temperatures. Forward-curved fans are utilized in clean environments and operate at lower temperatures. Backward-inclined fans have high efficiency than forward-curved fans. Backward-inclined fans reach their peak power consumption so power demand drops off well within their useable range of air flow. They are called "non-overloading" because static pressure changes don't overload the motor.

Fig – Types of Centrifugal Fan

LITERATURE SURVEY
[1]. A Syam Prasad, BVVV LakshmipathiRao, A Babji, Dr P Kumar Babu,presented study of static and modal analysis of a pump blade that is created of 3 totally different alloy materials. (viz., Inconel alloy 740, Incoloy alloy 803, Warpaloy).The most effective material for blade design is Inconel alloy 740. Specific modulus of Inconel alloy 740 obtained in static analysis 10% more thanother materials. The natural frequency in modal analysis is 6% more than other material. The deformation of Inconel alloy 740 in static analysis is decreased by 12%.

[2] KarthikMatta, KodeSrividya, InturiPrakash,The impeller modelling was done utilizing solid modelling package, CATIA V5 R18. It's planned to design a blower with composite material, analyze its strength and deformation utilizing FEM software. So as to gauge the effectiveness of composites and metal impeller and blower utilizing FEA packaged (ANSYS). Modal analysis is performed on both composite material and Aluminium centrifugal blower impeller to seek out initial five natural frequencies. If no. of blades and outer diameter will increase stresses and deformation conjointly will increase and all are within the allowable

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limit. Total analysis result compares and located that composite materials are having less deformation and stresses

DESIGN OF IMPELLER

Note :-Below calculations are considered from the NPTEL as a references : – chapter -3 design of centrifugal fan and its methodologies.

Impeller eye and inlet duct size

Let inlet duct size be 10% higher than impeller eye size or impeller inlet diameter. This will make conical insertion of inlet duct and flow acceleration at impeller eye or inlet.

$D_{duct} = 1.1\ D_{eye} = 1.1\ D_1$

Assuming no loss during 90° turning from eye inlet to impeller inlet, the eye inlet velocity vector will remain same as absolute velocity vector at the entry of impeller.

$V_{eye} = V_1 = V_{ml}$

Further let tangential velocity component be 10% higher than axial velocity component for better induction of flow.

So, Inlet Tip velocity

$U_1 = 1.1V_1 = 1.1V_{ml}$

Discharge $Q = \frac{\pi}{4} D^2_{eye} \times V_1$

$Q = \frac{\pi}{4} D^2_1 \times V_1$

Speed of impeller rotation N=2800 rpm,

Impeller Inlet Diameter$D1$=Deye

Peripheral speed at inlet$U_1 = \frac{\pi D1N}{60} V_1$

Impeller inlet blade angle$\tan\beta_1 = \frac{v1}{u1}$

Impeller width at inlet $Q=[\pi\ D1 - Zt] X\ b_1 \times Vml$

Impeller outlet parametersThe Fan Power =ΔPxQ

Considering 10% extra to accommodate flow recirculation and impeller exit hydraulic losses.

So, 1.1 x the fan power Power, $P = m \times W_s$

Euler power = $mV_{U2}U_2$

Taking $V_{u2} = 0.8\ U_2$ (assuming slip factor = 0.8 for radial blades)

$b_1=b_2$

$Q=[\pi\ D2 - Zt] X\ b_2 \times V_{m2}$

Design of Volute Casing

Analyzing steady flow energy equation at inlet and exit:

$\frac{P1}{\rho 1} + \frac{1}{2} V_1^2 + gz_1 + W_s = \frac{P2}{\rho 2} + \frac{1}{2} V_4^2 + gz_2$

Neglecting potential difference,$V_4^2 = \frac{-2[p2-p1]}{\rho f} + V_1^2 + 2 W_s$

$Q = A_v V_4$, Where A_v is exit area of the volute casing = $A_v = b_v(r_4 - r_3)$

Allowing for 5 mm radial clearance between impeller and volute tongue,

$r_3 = \frac{D2}{2} + 5$

Width of volute casing (bv) is normally 2 to 3 times $b1$

Let us take it 2.5 times. HenceBv=2.5b2

Leakage loss$Q_L = Cd \times \pi \times D1 \times \delta \times \sqrt{\frac{2Ps}{\rho}}$

Here, $Ps = \frac{2}{3} \Delta\ Ps$, And coefficient of discharge Cd is 0.6 to 0.7,δ = clearance between impeller eye inlet and casing

Suction pressure loss$dp_{suc} = \frac{1}{2} \times k_i \times \rho \times v^2_{eye}$ wherek_i is a loss factor probably of the order of 0.5 to 0.8

Impeller pressure loss$dP_{imp} = \frac{1}{2} \times k_{ii} \times \rho\ (W_1 - W'_2)^2$

Volute pressure loss$dP_{vc} = \frac{1}{2} \times k_{iii} \times \rho\ (V'_2 - V_4)^2$

Disc friction loss$T_{df} = \pi f \rho \omega_2^2\ (r_2^5 / 5)$

Where f is material friction factor in order of 0.005 for mild steel sheet metal

Hence, Power loss due to Disc friction$P_{df} = \frac{2\pi NT}{60}$

Hydraulic efficiency $\eta_{hy} = (\Delta P) / (\Delta P + dp_{suc} + dp_{imp} + dp_{vc})$

Volumetric efficiency $\eta_{vol} = (Q) / (Q+Q_L)$

Total efficiency$\eta_{total} = \eta_{hy} + \eta_{vol}$

3D MODELING OF CENTRIFUGAL FAN

Note:-Dimensions are taken from above calculations

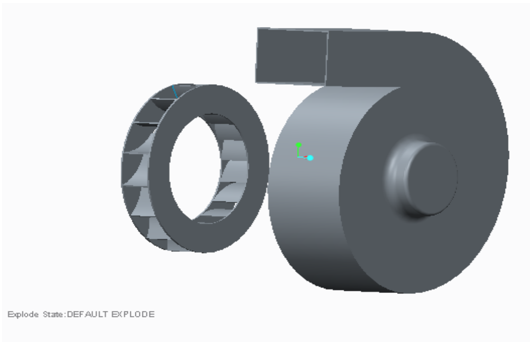

Fig:- exploded view of assembly – centrifugal fan

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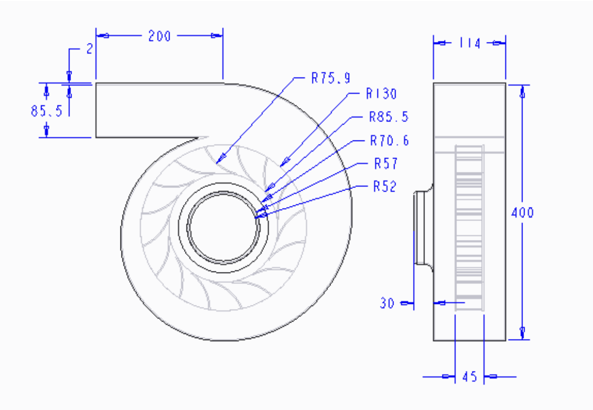

Fig:-Drafting of assembly – centrifugal fan

Boundary conditions for ANSYS

Note:-Input parameters (velocities) are taken from above calculations

CFD ANALYSIS ON CENTRIFUGAL FAN

VELOCITY - 22.45m/s

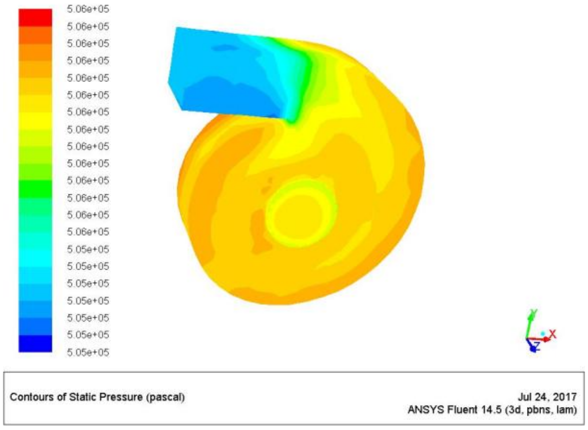

Fig: - Pressure

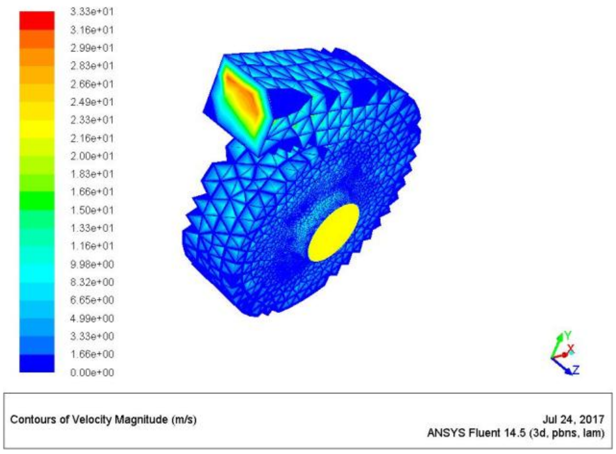

Fig: - Velocity

Mass Flow Rate	(kg/s)
inlet	0.23360464
interior-casing	0
interior-fan	0
interior-fluid	-0.39950582
outlet	-0.23298062
wall-casing	0
wall-fan	0
wall-fluid	0
Net	0.00062401593

STATIC STRUCTURAL ANALYSIS

VELOCITY - 22.45m/s

MATERIAL - STEEL

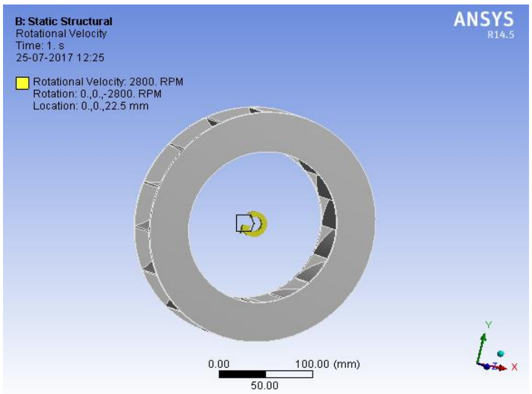

Fig: - Rotational speed

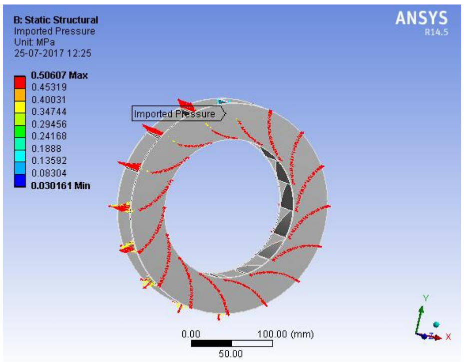

Fig: - Imported pressure

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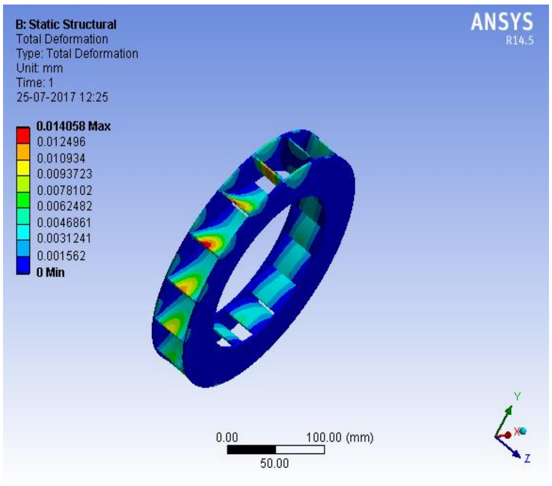

Fig: - Total deformation

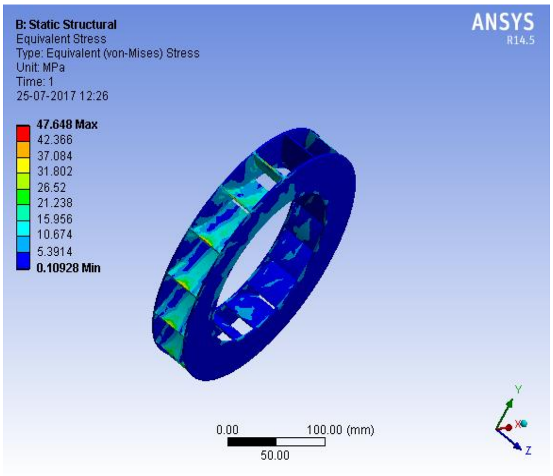

Fig: - Stress

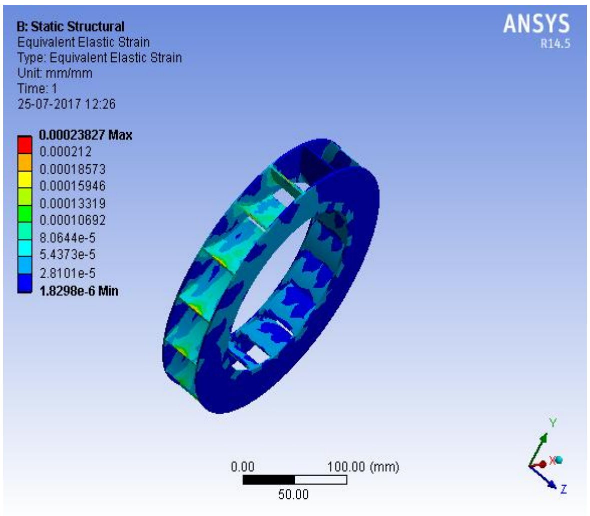

Fig: - Strain

RESULTS TABLE

CFD Analysis

Velocity (m/s)	Pressure (Pa)	Velocity (m/sec)	Mass flow rate (kg/sec)	Reynolds number
V1	5.06e+5	33.3	0.000124	3.23e+4
V2	5.86e+5	35.56	0.000313	3.45e+4
V3	6.27e+5	38.16312	0.000482	3.7e+4

Static Structural analysis

	Speed (RPM)	Deformation (mm)	Strain	Stress (MPa)
Steel	2800	0.014058	0.00023827	47.628
	3000	0.01625	0.00027542	55.076
	3200	0.017626	0.0002987	59.732
Aluminium alloy	2800	0.033424	0.00056546	40.142
	3000	0.038682	0.00065441	46.456
	3200	0.04159	0.00070358	49.947
E-glass epoxy	2800	0.032647	0.00056323	40.774
	3000	0.037784	0.00065186	47.191
	3200	0.040613	0.00070057	50.717
Aramid fiber	2800	0.20436	0.0034512	37.958
	3000	0.23661	0.0039957	43.946
	3200	0.25364	0.0042833	47.109

CONCLUSION

By observing the CFD analysis results, the static pressure is increasing by 13.6% & by 19.2% by increasing the speed to 3000rpm, 3200rpm respectively. The velocity is increasing by 6.3% & by 12.7% by increasing the speed to 3000rpm, 3200rpm respectively. The mass flow rate is increasing by 60.38% & by 74.2% by increasing the speed to 3000rpm, 3200rpm respectively. The stresses are reducing for all materials Aluminium, E – Glass Epoxy and Aramid Fiber when compared with that of Steel. The stresses are decreasing for E – Glass Epoxy material by about 14.3% at 2800rpm, by 14.3% at 3000rpm, by 15% at 3200rpm when compared with Steel. The stresses are decreasing for Aramid Fiber by about 20.3% at 2800rpm, by 23.3% at 3000rpm, by 21.1% at 3200rpm when compared with Steel.

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