DESIGN AND OPTIMIZATION OF CENTRIFUGAL FAN RADIAL BLADE IMPELLER

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Abstract

Fans are one of the types of turbo machinery which are used to move air continuously with in slight increase in static pressure. Fans are widely used in industrial and commercial applications from shop ventilation to material handling, boiler applications to some of the vehicle cooling systems. A centrifugal fan is a mechanical device for moving air or other gases.

In this thesis the performance of centrifugal fan having backward curved blades is investigated. CFD analysis is performed on the fan to determine outlet pressures, velocities and mass flow rates by changing boundary condition inlet velocity and fan speed. Static Structural analysis is done on the fan by taking pressures from CFD analysis as boundary condition. Different materials Stainless Steel, Aluminum 7075, S Glass Epoxy and Aramid Fiber are considered for the analysis where deformations and stresses are determined. 3D model of the centrifugal fan is done in Creo 2.0. CFD and Static structural analyses are performed in Ansys.

Keywords: *centrifugal fan, CFD, Static structural analyses, Stainless Steel, Aluminum 7075, S Glass Epoxy and Aramid Fiber.*

1. Introduction

1.1 Centrifugal Fan

Centrifugal fans use the kinetic energy of the impellers to increase the volume of the air stream, which in turn moves against the resistance caused by ducts, dampers and other components. Centrifugal fans displace air radially, changing the direction (typically by 90°) of the airflow. They are sturdy, quiet, reliable, and capable of operating over a wide range of conditions.

1.2 Construction:

Main parts of a centrifugal fan are:

1. Fan housing
2. Impellers
3. Inlet and outlet ducts
4. Drive shaft
5. Drive mechanism

Other components like bearings couplings, impeller locking device, fan discharge casing, shaft seal plates etc.

1.3 Fan blades:

The fan wheel consists of a hub with a number of fan blades attached. The fan blades on the hub can be arranged in three different ways: forward-curved, backward-curved or radial

1.3.1 Forward-curved:

Forward-curved blades, as in Figure 3(a), curve in the direction of the fan wheel's rotation. These are especially sensitive to particulates and commonly are only specified for clean-air applications such as air conditioning. Forward-curved blades provide a low noise level and relatively small air flow with a high increase in static pressure.

1.3.2 Backward-curved:

Backward-curved blades, as in Figure 3(b), curve against the direction of the fan wheel's rotation. Smaller blowers may have **backward-inclined** blades, which are straight, not curved. Larger backward-inclined/-curved blowers have blades whose backward curvatures mimic that of an airfoil cross section, but both designs provide good operating efficiency with relatively economical construction techniques. These types of blowers are designed to handle gas streams with low to moderate particulate loadings. They can be easily fitted with wear protection but certain blade curvatures can be prone to solids build-up. Backward curved wheels are often heavier than corresponding forward-curved equivalents, as they run at higher speeds and require stronger construction.

Backward curved fans can have a high range of specific speeds but are most often used for medium specific speed applications—high pressure, medium flow applications.

Backward-curved fans are much more energy efficient than radial blade fans and so, for high power applications may be a suitable alternative to the lower cost radial bladed fan.

1.3.3 Straight radial:

Radial blowers, as in Figure 3(c), have wheels whose blades extend straight out from the center of the hub. Radial bladed wheels are often used on particulate-laden gas streams because they are the least sensitive to solid build-up on the blades, but they are often characterized by greater noise output. High speeds, low volumes, and high pressures are common with radial blowers, and are often used in vacuum cleaners, pneumatic material conveying systems, and similar processes.

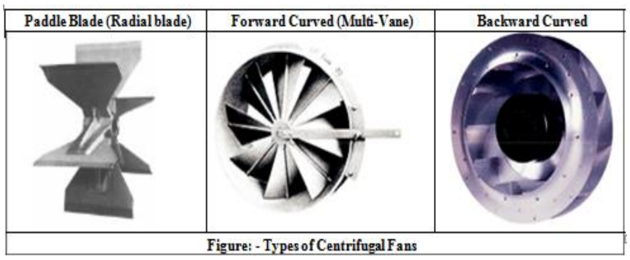

Figure: 1.3. Types of centrifugal compressor

1.4 Principles of operation:

The centrifugal fan uses the centrifugal power supplied from the rotation of impellers to increase the kinetic energy of air/gases. When the impellers rotate, the gas particles near the impellers are thrown off from the impellers, then move into the fan casing. As a result, the kinetic energy of gas is measured as pressure because of the system resistance offered by the casing and duct. The gas is then guided to the exit via outlet ducts. After the gas is thrown-off, the gas pressure in the middle region of the impellers decreases. The gas from the impeller eye rushes in to normalize this. This cycle repeats and therefore the gas can be continuously transferred.

Fans and blowers provide air for ventilation and industrial process requirements. Fans generate a pressure to move air (or gases) against a resistance caused by ducts, dampers, or other components in a fan system. The fan rotor receives energy from a rotating shaft and transmits it to the air.

1.5 Centrifugal Fan: Types

The major types of centrifugal fan are: radial, forward curved and backward curved . Radial fans are industrial workhorses because of their high static pressures (upto 1400 mm WC) and ability to handle heavily contaminated airstreams. Because of their simple design, radial fans are well suited for high temperatures and medium blade tip speeds.

Forward-curved fans are used in clean environments and operate at lower temperatures. They are well suited for low tip speed and high-airflow work - they are best suited for moving large volumes of air against relatively low pressures.

Backward-inclined fans are more efficient than forward-curved fans. Backward-inclined fans reach their peak power consumption and then power demand drops off well within their useable airflow range. Backward-inclined fans are known as "non-overloading" because changes in static pressure do not overload the motor.

1.6 Fan Performance Evaluation and Efficient System Operation

1.6.1 System Characteristics

The term "system resistance" is used when referring to the static pressure. The system resistance is the sum of static pressure losses in the system. The system resistance is a function of the con-figuration of ducts, pickups, elbows and the pressure drops across equipment-for example back-filter or cyclone. The system resistance varies with the square of the volume of air flowing through the system. For a given volume of air, the fan in a system with narrow ducts and multiple short radius elbows is going to have to work harder to overcome a greater system resistance than it would in a system with larger ducts and a minimum number of long radius turns. Long narrow ducts with many bends and twists will require more energy to pull the air through them. Consequently, for a given fan speed, the fan will be able to pull less air through this system than through a short system with no elbows. Thus, the system resistance increases substantially as the volume of air flowing through the system increases; square of air flow.

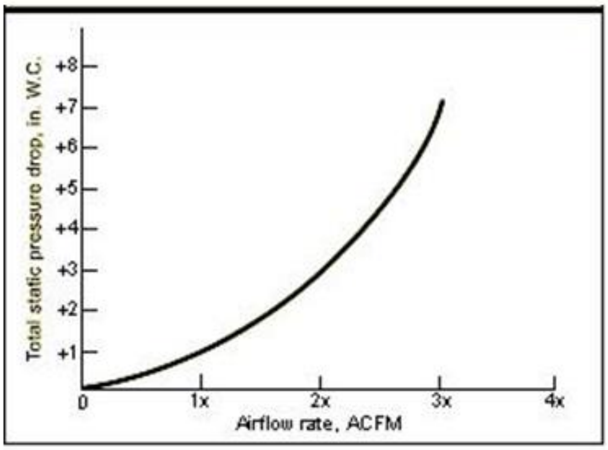

Figure: 1.6.1. Flowrate vs Static Pressure

Conversely, resistance decreases as flow decreases. To determine what volume the fan will produce, it is therefore necessary to know the system resistance characteristics.

In existing systems, the system resistance can be measured. In systems that have been designed, but not built, the system resistance must be calculated. Typically a system resistance curve is generated with for various flow rates on the x-axis and the associated resistance on the y-axis.

1.7 Fan Characteristics

Fan characteristics can be represented in form of fan curve(s). The fan curve is a performance curve for the particular fan under a specific set of conditions. The fan curve is a graphical representation of a number of inter-related parameters. Typically a curve will be developed for a given set of conditions usually including: fan volume, system static pressure, fan speed, and brake horsepower required to drive the fan under the stated conditions. Some fan curves will also include an efficiency curve so that a system designer will know where on that curve the fan will be operating under the chosen conditions (see Figure 5.6). In the many curves shown in the Figure, the curve static pressure (SP) vs. flow is especially important.

The intersection of the system curve and the static pressure curve defines the operating point. When the system resistance changes, the operating point also changes. Once the operating point is fixed, the power required could be found by following a vertical line that passes through the operating point to an intersection with the power (BHP) curve. A horizontal line drawn through the intersection with the power curve will lead to the required power on the right vertical axis. In the depicted curves, the fan efficiency curve is also presented.

1.8 System Characteristics and Fan Curves

In any fan system, the resistance to air flow (pressure) increases when the flow of air is increased. As mentioned before, it varies as the square of the flow. The pressure required by a system over a range of flows can be determined and a "system performance curve" can be developed (shown as SC)

This system curve can then be plotted on the fan curve to show the fan's actual operating point at "A" where the two curves (N_1 and SC_1) intersect. This operating point is at air flow Q_1 delivered against pressure P_1.

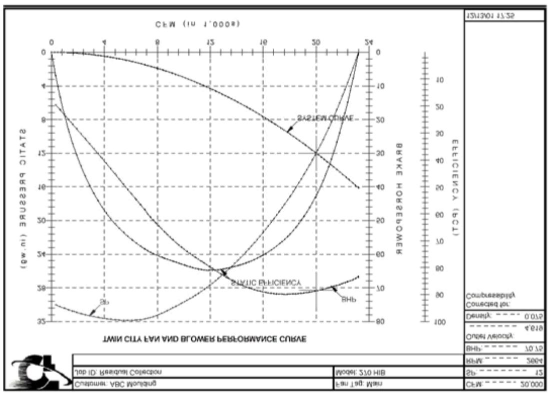

Figure 1.8.1. Fan Performance Curve

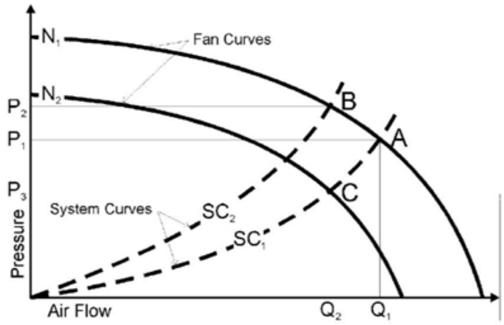

Figure 1.8.2. Fan pressure curves

A fan operates along a performance given by the manufacturer for a particular fan speed. (The fan performance chart shows performance curves for a series of fan speeds.) At fan speed N_1, the fan will operate along the N_1 performance curve as shown in Figure 5.7. The fan's actual operating point on this curve will depend on the system resistance; fan's operating point at "A" is flow (Q_1) against pressure (P_1).Two methods can be used to reduce air flow from Q_1 to Q_2:

First method is to restrict the air flow by partially closing a damper in the system. This action causes a new system performance curve (SC_2) where the required pressure is greater for any given air flow. The fan will now operate at "B" to provide the reduced air flow Q_2 against higher pressure P_2.

Second method to reduce air flow is by reducing the speed from N_1 to N_2, keeping the damper fully open. The fan would operate at "C" to provide the same Q_2 air flow, but at a lower pressure P_3.

Thus, reducing the fan speed is a much more efficient method to decrease airflow since less power is required and less energy is consumed.

1.9 Fan Laws

The fans operate under a predictable set of laws concerning speed, power and pressure. A change in speed (RPM) of any fan will predictably change the pressure rise and power necessary to operate it at the new RPM.

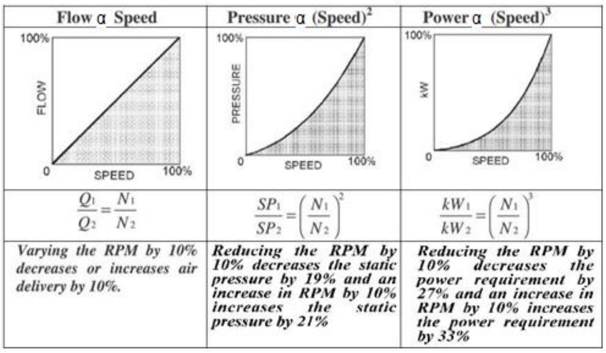

Figure 1.9.1 Fan Laws curve

Where Q – flow, SP – Static Pressure, kW – Power and N – speed (RPM)

1.10 Fan Design and Selection Criteria

Precise determination of air-flow and required outlet pressure are most important in proper selection of fan type and size. The air-flow required depends on the process requirements; normally determined from heat transfer rates, or combustion air or flue gas quantity to be handled. System pressure requirement is usually more difficult to compute or predict. Detailed analysis should be carried out to determine pressure drop across the length, bends, contractions and expansions in the ducting system, pressure drop across filters, drop in branch lines, etc. These pressure drops should be added to any fixed pressure required by the process (in the case of ventilation fans there is no fixed pressure requirement). Frequently, a very conservative approach is adopted allocating large safety margins, resulting in over-sized fans which operate at flow rates much below their design values and, consequently, at very poor efficiency.

Once the system flow and pressure requirements are determined, the fan and impeller type are then selected. For best results, values should be obtained from the manufacturer for specific fans and impellers.

The choice of fan type for a given application depends on the magnitudes of required flow and static pressure. For a given fan type, the selection of the appropriate impeller depends additionally on rotational speed. Speed of operation varies with the application. High speed small units are generally more economical because of their higher hydraulic efficiency and relatively low cost. However, at low pressure ratios, large, low-speed units are preferable.

1.11 Centrifugal Fan Applications

Because of the high pressure they create, centrifugal fans are ideal for high pressure applications such as drying and air conditioning systems. As all of their moving parts are enclosed and they also have particulate reduction properties that makes them ideal for use in air pollution and filtration systems. Centrifugal fans also offer distinct benefits:

a) **First-rate energy efficiency.** airflow allows centrifugal fans to generate energy that reaches up to 84% static efficiency. These higher efficiency levels are ideal for sustaining larger air systems.

b) **Enhanced durability.** These fans are durable enough to properly operate in the most corrosive and erosive environments.

c) **Ability to restrict over loading.** Certain centrifugal fans are fitted with non-overloading horsepower curves will ensure the motor will not overload if its capacity is exceeded.

d) **Easy to maintain.** Lighter material fans can be easily cleaned when you deem it necessary. Moreover, certain fans have self-cleaning characteristics, making daily maintenance that much easier.

e) **High versatility.** Centrifugal fans are useful for multiple airflow/pressure combinations, and they can process several airflow conditions, including clean, dry, and wet air

f) **Multiple sizes.** These fans are available in several sizes to accommodate diverse applications—such as those found in tight spaces or difficult to reach areas.

2 DESIGN OF IMPELLER

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

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

$V_1 = \frac{4Q}{\pi D1 x D1}$

$U_1 = \frac{\pi D1N}{60} = 1.1\, V_1$

$\therefore \frac{\pi D1N}{60} = 1.1 \frac{4Q}{\pi D1 x D1}$

Here Q=0.5 m3/s and

(i) speed of impeller rotation N=3200 rpm,

∴Impeller Inlet Diameter

D1=0.168 m = Deye

∴Peripheral speed at inlet

$U_1 = \frac{\pi D1N}{60}$ = 28.1344 m/sec

V_1=25.576 m/s = $V_{ml} = V_{eye}$

(ii) speed of impeller rotation N=3400 rpm,

∴Impeller Inlet Diameter

D1=0.168 m = Deye

∴Peripheral speed at inlet

$U_2 = \frac{\pi D1N}{60}$ = 28.8928 m/sec

V_2=27.175 m/s = $V_{ml} = V_{eye}$

(iii) speed of impeller rotation N=3600 rpm,

∴Impeller Inlet Diameter

D1=0.168 m = Deye

∴Peripheral speed at inlet

$U_3 = \frac{\pi D1N}{60}$ = 31.6512 m/sec

V_3=28.773 m/s = $V_{ml} = V_{eye}$

2.1.1 Impeller inlet blade angle

$\tan\beta_1 = \frac{v1}{u1} = \frac{22.45}{24.63} = 42.35^0$

2.1.2 Impeller width at inlet

Here Z=16 and assumed blade thickness t = 2 mm

$Q = [\pi\, D1 - Zt] X\, b_1 \times Vml$

$0.5 = [(\pi x0.168)-(16x2x10^{-3})]xb_1x22.45$

B_1=45mm

2.1.3 Impeller outlet parameters

The Fan Power =ΔPxQ=981.2x0.5=490.6W

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

So, 1.1 x the fan power =1.1x490.6=539.66W

Power, $P = m \times W_s$

∴specific work done ,

$W_s = \frac{539.66}{1.165x0.5}$ = 926.45 W/(kg/s)

Euler power = $mV_{U2}U_2$

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

$539.66 = 1.165 \times 0.5 \times 0.8\, U_2 \times U_2$

U_2 = 34.03 m/sec

V_{u2} = 0.8 x34.03 = 27.22 m/sec

And $U_2 = \frac{\pi D2N}{60}$ = 0.232 m = 232mm

Taking width of blade at inlet = outlet blade width

$\therefore b_1 = b_2$

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

$0.5 = [(\pi x0.232)-(16x2x10^{-3})] \times 0.045\, xV_{m2}$

V_{m2} = 16 m/sec

$W_{U2} = 0.2\, U2 = 6.81$ m/sec $(V^2_{U2} + Vm_2^2)$

$W^1_2 = \sqrt{(W^2_{U2} + Vm_2^2)} = \sqrt{(0.2U_2)^2 + Vm_2^2}$

$W^1_2 = \sqrt{6.81^2 + 16^2}$ = 17.39 m/s

$V^1_2 = \sqrt{(V^2_{U2} + Vm_2^2)}$

$V^1_2 = \sqrt{27.22^2 + 16^2}$ = 31.58 m/sec

$\tan\alpha_2 = \frac{Vm2}{Vu2}$ = 16/27.22 = 0.59

$\alpha_2 = 30.45^0$ (outlet velocity)

2.2 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,

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

$V_4^2 = \frac{-2(981.2)}{1.165} + 22.45^2 + 2(926.45)$

V_4^2 = 25.93 m/sec

$Q = A_v V_4$

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,

$r_3 = \frac{D2}{2} + 5 = \frac{232}{2} + 5$ = 121 mm

D_3 = 2x121 = 242mm

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

Let us take it 2.5 times. Hence
Bv=2.5b2=2.5x45=112.5m,m
$Q= A_v V_4$
$0.5= b_v(r_4-r_3)x25.93$
0.5 = (0.1125 (r4-0.121)x25.93
R4 = 292mm
D4 = 2x292 = 584mm

2.3 Hydraulic, leakage and power Losses

2.3.1 Leakage loss

$QL = Cd \times \pi \times D1 \times \delta \times \sqrt{\frac{2Ps}{\rho}}$

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 2mm as per fabrication requirement

$\delta = \pi \times 0.6 \times 0.169 \times 0.002 \times \sqrt{\frac{2X\frac{2}{3}X981.2}{1.165}}$

$\delta = 0.0213\ m^3/sec$

2.3.2 Suction pressure loss

$dp_{suc} = \frac{1}{2} \times k_i \times \rho \times v^2_{eye}$

where ki is a loss factor probably of the order of 0.5 to 0.8

$dp_{suc} = \frac{1}{2} \times 0.65 \times 1.165 \times 22.45^2$

$dp_{suc} = 190.76$ Pa

2.3.3 Impeller pressure loss

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

$dP_{imp} = \frac{1}{2} \times 0.25 \times 1.165\ (33.37\text{-}17.39)^2$

$dP_{imp} = 37.18$ Pa

2.3.4 Volute pressure loss

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

$dP_{vc} = \frac{1}{2} \times 0.4x\ 1.165\ (31.58\text{-}25.98)^2$

$dP_{vc} = 7.23$ Pa

2.3.5 Disc friction loss

$T_{df} = \pi f \rho \omega_2^2\ (r_2^5 / 5) = \pi f \rho (U_2/r_2)^2\ (r_2^5 / 5)$

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

$T_{df}= \pi \times 0.005 \times 1.165 \times \frac{(34.03x34.03)}{(0.116x0.116)} \times \frac{0.116x0.116}{5}$

$T_{df} = 0.0066$ N-m

Hence, Power loss due to Disc friction

$P_{df} = \frac{2\pi NT}{60} = \frac{2\pi \times 2800 \times 0.0066}{60} = 1.94$ W

2.4 Efficiencies

2.4.1 Hydraulic efficiency

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

$\eta_{hy} = (981.2) / (981.2+196.13+43.47+10.58)$
$\eta_{hy} = 0.8065$
$\eta_{hy} = 80.65\%$

2.4.2 Volumetric efficiency

$\boldsymbol{\eta_{vol} = (Q) / (Q+Q_L)}$
$\eta_{vol} = (0.5) / (0.5+0.0213) =0.959$
$\eta_{vol} =95.9\ \%$

2.2.3 Total efficiency

$\eta_{total}= \eta_{hy}+ \eta_{vol}= 0.8065 + 0.959 = 0.7736$
$\eta_{total}= 77.36\%$

** above calculations are considered from the NPTEL as a references with chapter -3 design of centrifugal fan and its methodologies.

3. MODELING AND ANALYSIS OF CENTRIFUGAL FAN RADIAL BLADE IMPELLER

Dimensions are taken from above calculations and the reference paper is "Design and Analysis of Impeller for Centrifugal Blower using Solid Works by KAY THI MYAING, HTAY HTAY WIN, specified as [7] in References.

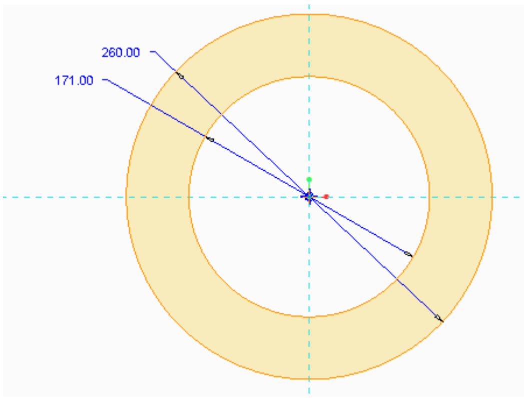

Figure: 3.0.1. 2-D sketch of centrifugal fan – impeller

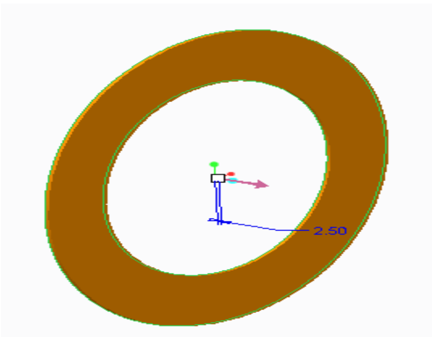

Figure: 3.0.2 Thickness of 2.5mm is added to centrifugal fan-impeller

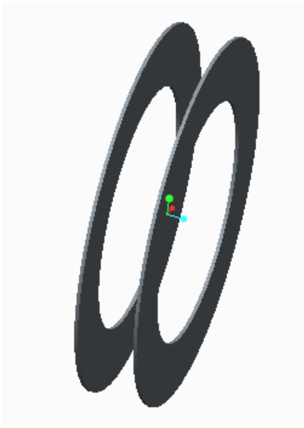

Figure: 3.0.3 Width of the fan- impeller

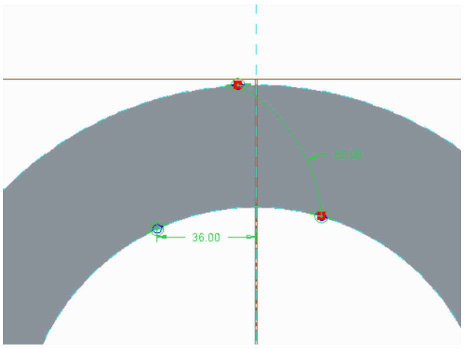

Figure: 3.0.4Blade profile as per calculation

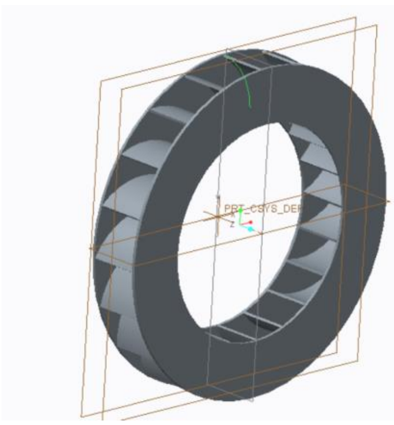

Figure: 3.0.5 Final view of centrifugal impeller with backward curved blade

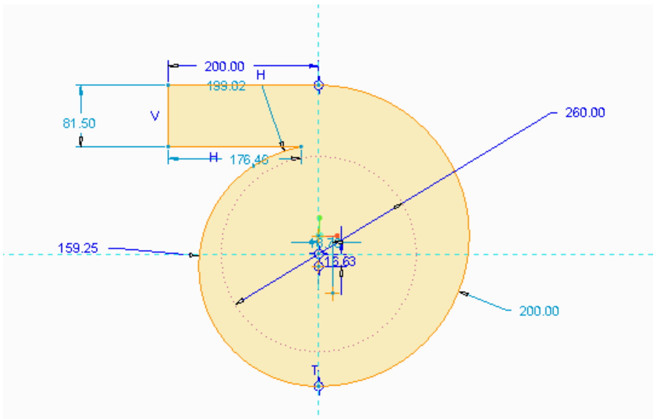

Figure: 3.0.6 2D sketch of hub

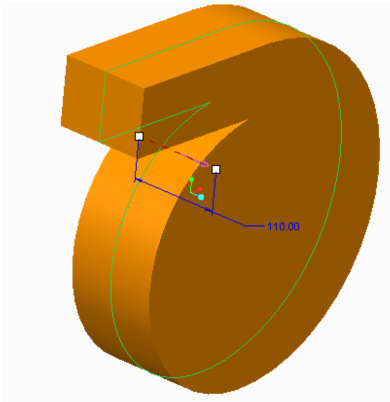

Figure: 3.0.7 Height is added to hub as pressure calculations

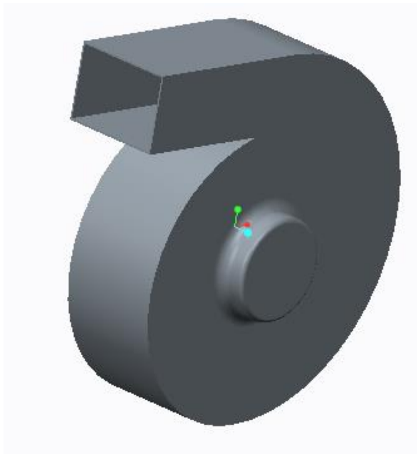

Figure: 3.0.8 Final centrifugal hub

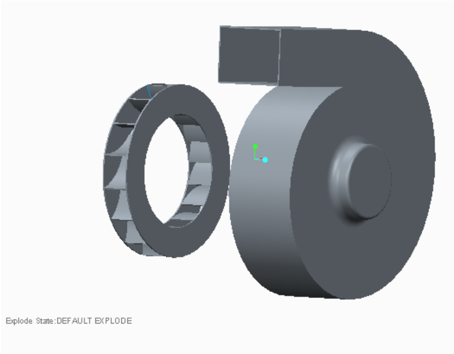

Figure: 3.0.9 Exploded view of assembly of impeller and hub

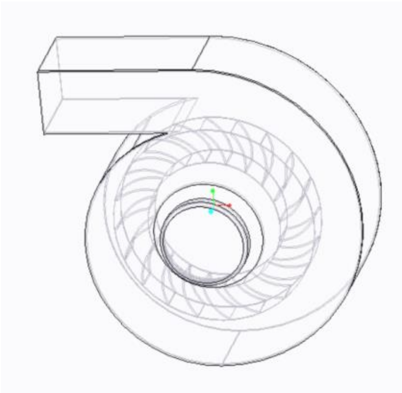

Figure: 3.0.10 Wireframe view of assembly

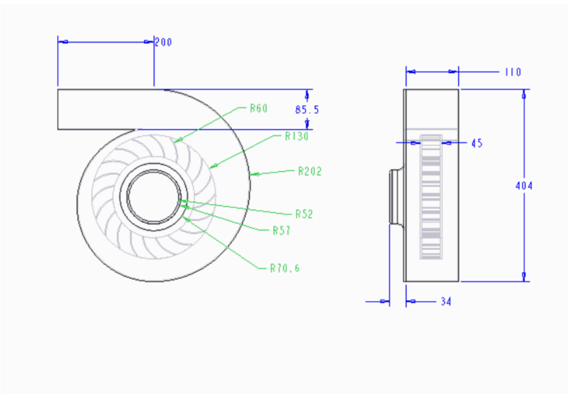

Figure: 3.0.11. 2D Drafting of assembly – centrifugal fan

3.1 Boundary conditions for analysis

* Input parameters (velocities) are taken from above calculations

Materials used for analysis

	Density (g/cc)	Young's modulus (MPa)	Poisson's ratio
Stainless Steel	9.01	310000	0.346
Aluminium 7075	2.81	71700	0.33
S-glass epoxy	2.495	93000	0.23
Aramid fiber	1.47	83000	0.3

Table: 3.1 Material used for analysis

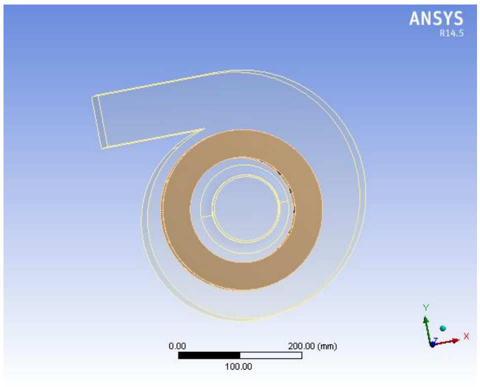

Figure: 3.1.1 Imported model of centrifugal fan

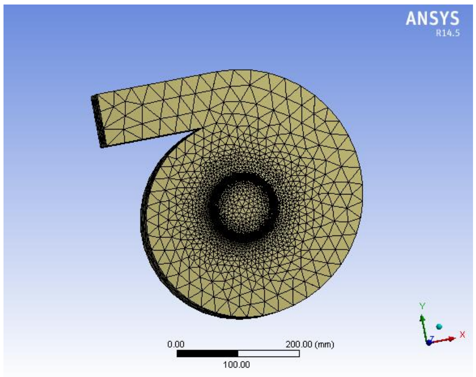

Figure: 3.1.2 Meshed model of centrifugal fan

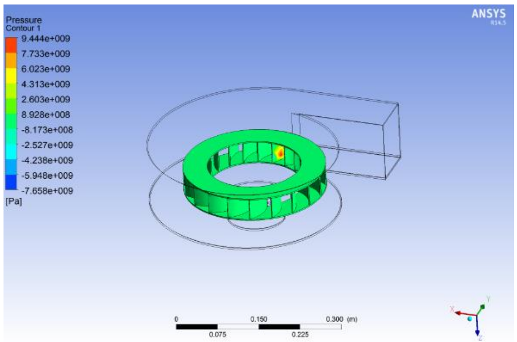

Figure: 3.1.3 Static Pressure on blades at velocity 22.45m/s and fan speed 3200rpm

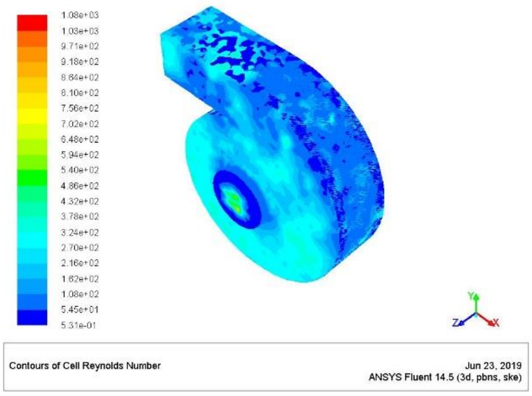

Figure: 3.1.4 Reynolds number at velocity 22.45m/s and fan speed 3200rpm

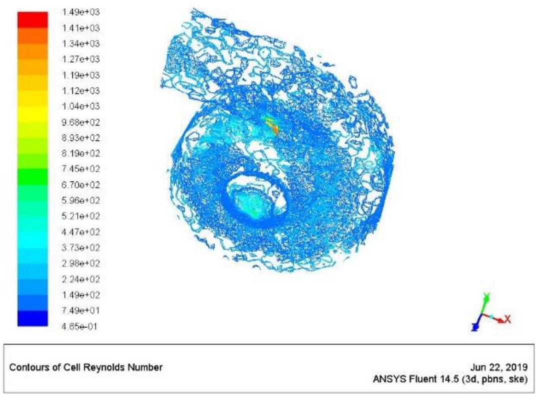

Figure: 3.1.5 Reynolds number at velocity 27.175m/s and 3400rpm

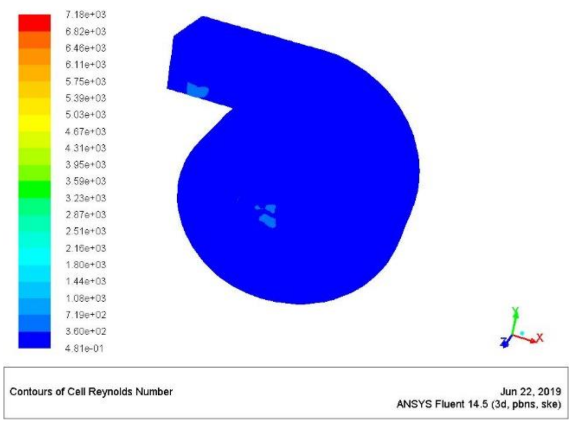

Figure: 3.1.6 Reynolds number at velocity 28.773m/s and fan speed 3600rpm

3.2 STATIC STRUCTURAL ANALYSIS

Static Structural analysis is done by taking pressures determined from CFD analysis as input.

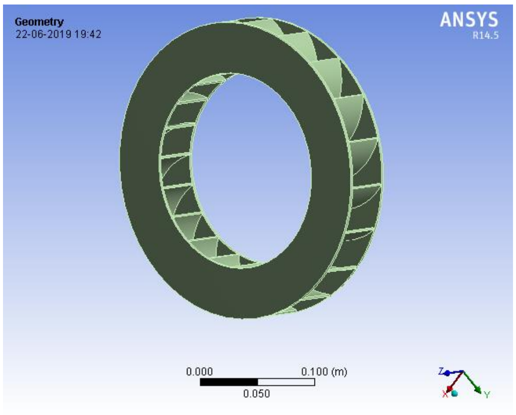

Figure:3.2.1 Imported model of centrifugal fan impeller

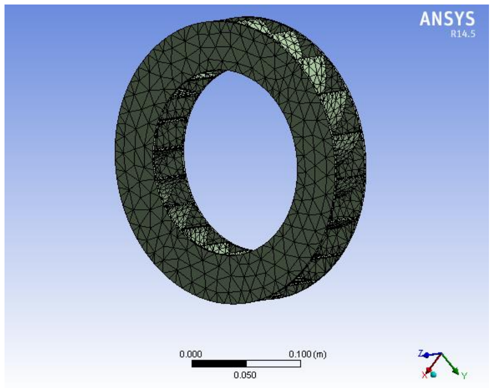

Figure: 3.2.2 Meshed model of centrifugal fan impeller

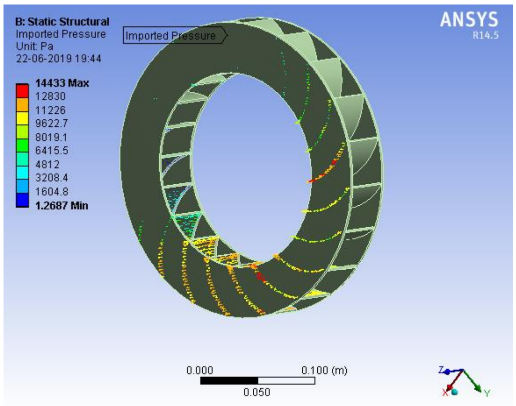

Figure: 3.2.3 Imported pressure from CFD analysis

4. Results

		Case 1	Case 2	Case 3
Input	v1 (m/s)	22.45	27.17	28.77
	N (rpm)	3200	3400	3600
Pressure (Pa)		9.44e+9	4.61e+11	7.80e+12
Velocity v2(m/s)		1.39e+4	3.91e+4	2.53e+5
Mass flow rate (kg/sec)		0. 0337	0.017872	0.012453
Reynolds Number		1.08e+3	1.49e+3	7.18e+3

Table 4.1. Results of CFD Analysis

* Inlet velocity of air (v1) in m/s, fan speed (N) in rpm.

	N (RPM)	Deformation (mm)	Strain	Stress (MPa)
Stainless Steel	3200	688.71	0.18525	1135.9
	3400	871.67	0.25595	3065.4
	3600	991.22	1.2461	5338.9
Aluminum 7075	3200	117.49	0.27244	1310.6
	3400	117.53	0.34403	1646.2
	3600	128.2	0.34483	1658.9
S-glass epoxy	3200	635.37	0.18525	848.33
	3400	804.15	0.23448	1135.9
	3600	914.77	1.1654	1437.8
Aramid fiber	3200	419.5	0.12285	500.06
	3400	530.92	0.15548	680.47
	3600	603.88	0.7636	861.25

Table 4.2. Results of Static Structural analysis

* Fan speed (N) in rpm. Deformation of blade in (mm)

5. Discussion

By observing Table.1, the pressure, velocity and Reynolds number are increasing by increasing the speed of fan but mass flow rate is decreasing. The pressure is increasing by about 80% when speed of fan is increased from 3200rpm to 3400rpm. The pressure is increasing by about 94% when speed of fan is increased from 3400rpm to 3600rpm.

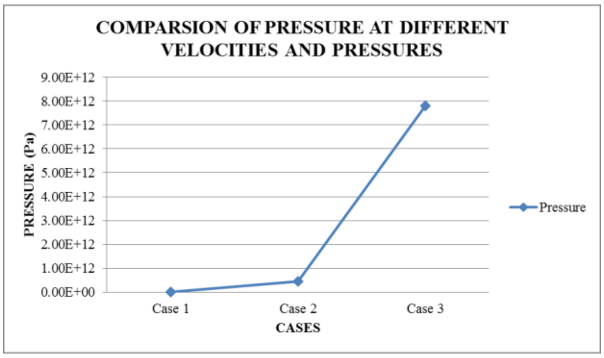

Figure: 5.1. Comparison of Pressure at Different Velocities and Pressures

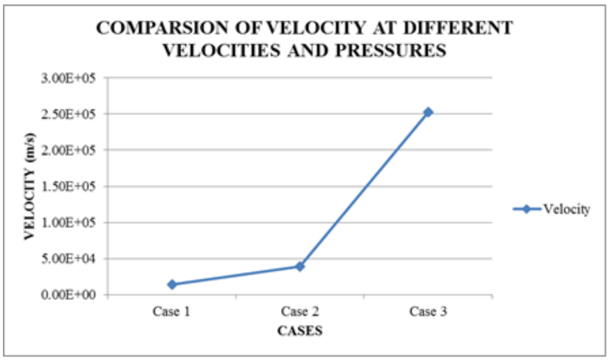

Figure: 5.2. Comparison of Velocity at Different Velocities and Pressures

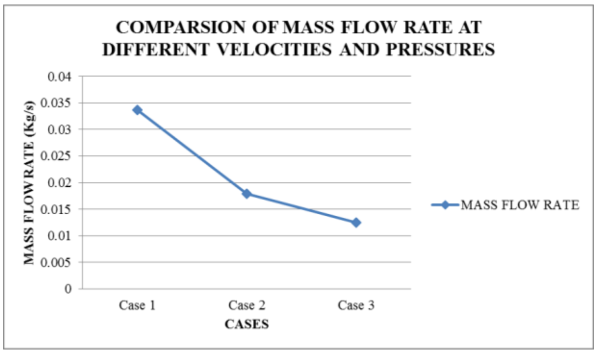

Figure:5.3. Comparison of Mass Flow Rate at Different Velocities and Pressures

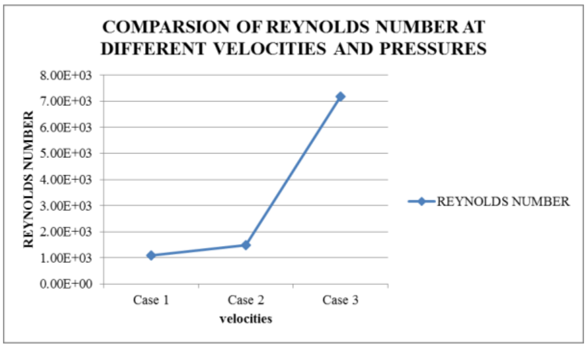

Figure 5.4. Comparison of Reynolds Number at Different Velocities and Pressures

Static structural analysis is done by applying the maximum pressure obtained from CFD analysis. The maximum pressure affects only a small portion of the blade. The maximum part of the blade has medium pressures acting on them. By observing above results, the stress values are increasing by increasing the fan speed. The stress values are less when Aramid Fiber is used when compared to other materials. The stress value is decreasing for Aramid Fiber by about 83% when compared with Stainless Steel, by about 48% when compared with Aluminum alloy, by about 40% when compared with S Glass Epoxy.

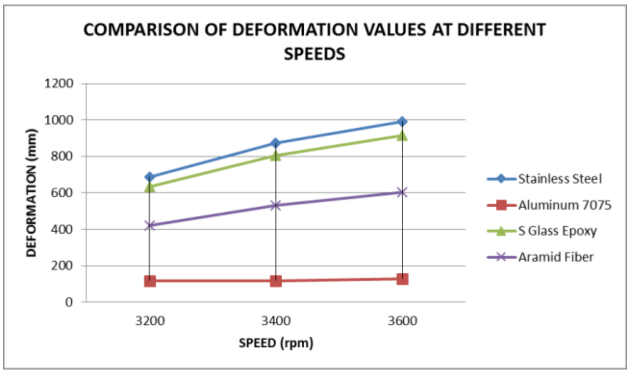

Figure: 5.5. Comparison of Deformation values at Different Speeds

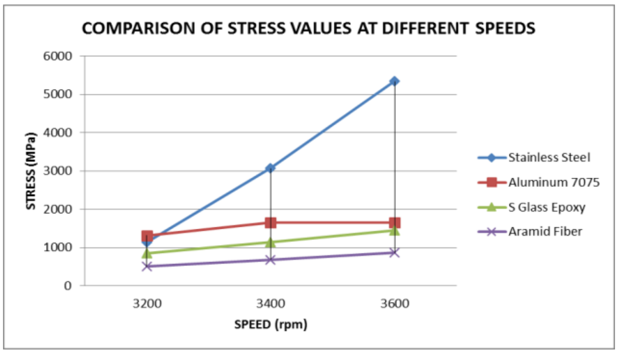

Figure:5.6. Comparison of Stress values at Different Speeds

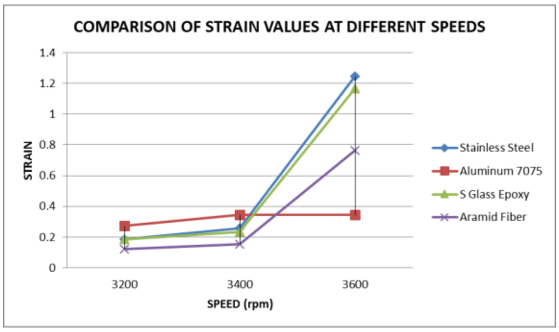

Figure 5.7. Comparison of Strain values at Different Speeds

6. Conclusion

CFD investigation is performed on the fan to decide outlet velocities, pressures, mass flow rates & Reynolds number at different speeds 3200rpm, 3400rpm and 3600rpm. By observing the results, the pressure, velocity and Reynolds number are increasing by increasing the speed of fan but mass flow rate is decreasing. The maximum pressure affects only a small portion of the blade. The maximum part of the blade has medium pressures acting on them. The pressure is increasing by about 80% when speed of fan is increased from 3200rpm to 3400rpm. The pressure is increasing by about 94% when speed of fan is increased from 3400rpm to 3600rpm.

Static investigation is done by applying pressures obtained from results of CFD analysis as boundary condition. Distinctive materials Stainless Steel, Aluminum 7075, S Glass Epoxy, Aramid Fiber are considered for the examination where displacements & stresses are established. By observing the results, By observing above results, the stress values are increasing by increasing the fan speed. The stress values are less when Aramid Fiber is used when compared to other materials. The stress value is decreasing for Aramid Fiber by about 83% when compared with Stainless Steel, by about 48% when compared with Aluminum alloy, by about 40% when compared with S Glass Epoxy.
It can be concluded that using Aramid Fiber is better.

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