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Design and Finite Element Analysis of Centrifugal Blower Fans

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ABSTRACT

In this study, the effects of design parameters on the fan noise level are investigated both theoretically and experimentally. For the theoretical study, a computational aero-acoustic method is used to predict the flow induced noise of a fan. This method involves the coupling of a flow solver and a wave equation solver. Unsteady flow analysis is performed with URANS using FLUENT. Then the time dependent data are processed with LMS Sysnoise to compute the acoustic radiation. Experimental studies are performed to verify the theoretical results and additionally to investigate the effects of different design alternatives on noise level of the fan. The sound pressure and intensity level measurements are performed in the full anechoic room of Arçelik A.Ş. Research and Development Laboratories. The validation experiments indicate that there is a good agreement between numerical and experimental results. The experimental study with different fan designs gives information about the noise reduction possibilities.

I. 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. 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 are industrial 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.

Figure 1: Types of Centrifugal Fan

II. DESIGN OF IMPELLER

Following calculations are considered from the NPTEL as a references 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 D1 Assuming no loss during 90º turning from eye inlet to impeller inlet, the eye inlet velocity vector will remain same as absolute velocity vector at the entry of impeller. Veye = V1 = 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_vV_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 lossQL = Cd x π x D1 x δ x $\sqrt{\frac{2Ps}{\rho}}$

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

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

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

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

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

III. 3D Modeling of Centrifugal Fan

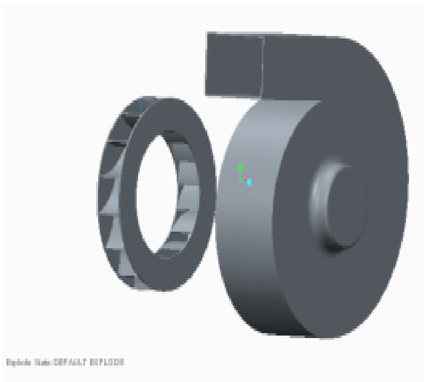

Figure 2: Exploded view of assembly – centrifugal fan

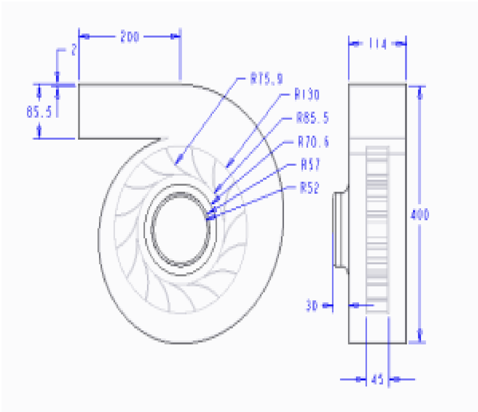

Figure 3: Drafting of assembly – centrifugal fan

3.1 Boundary conditions for ANSYS

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

3.2 CFD Analysis on Centrifugal Fan Velocity - 22.45m/S

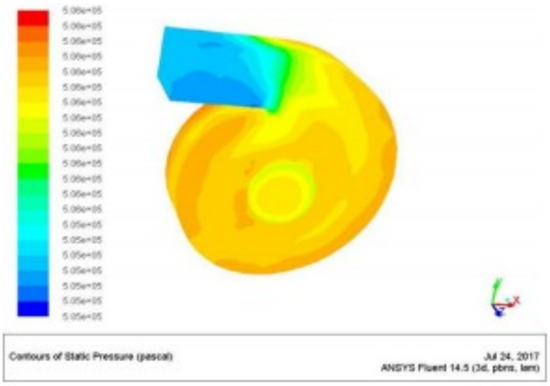

Figure 4: Pressure

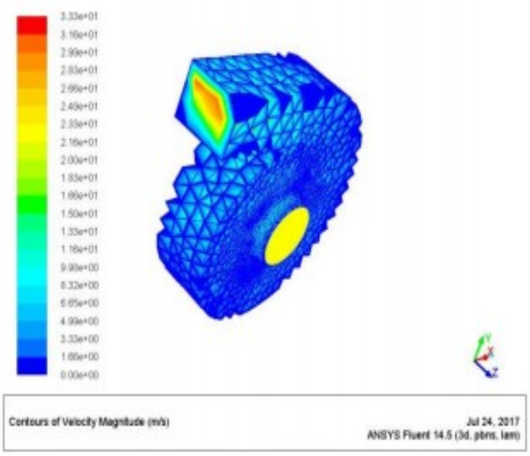

Figure 5: Velocity

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

3.3 Static Structural Analysis Velocity - 22.45m/S Steel material

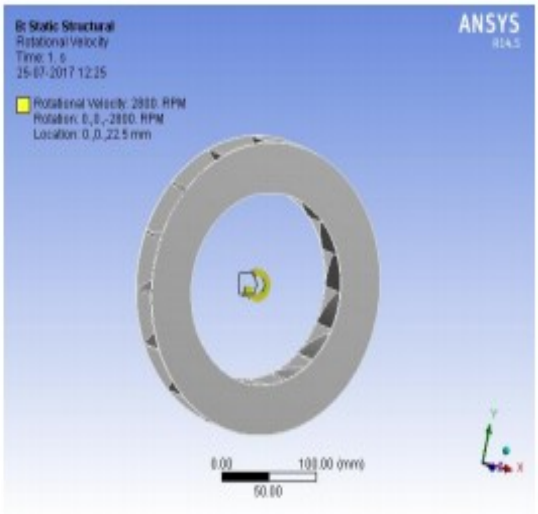

Figure 6: Rotational speed

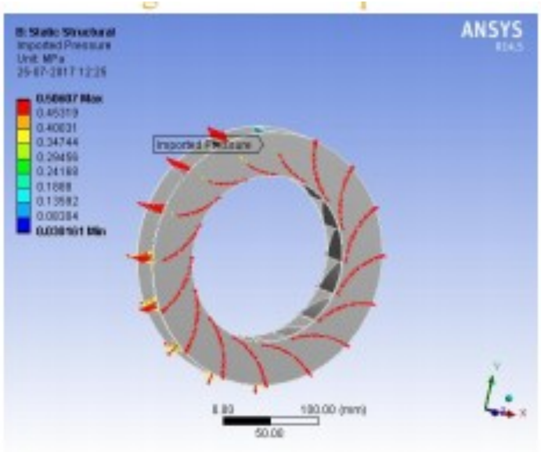

Figure 7: Imported pressure

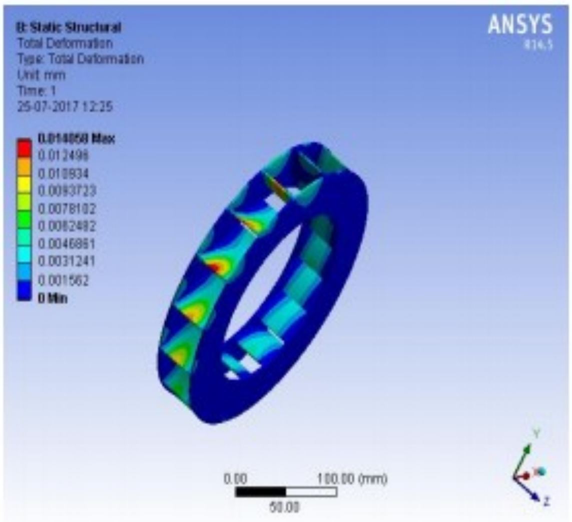

Figure 8: Total deformation

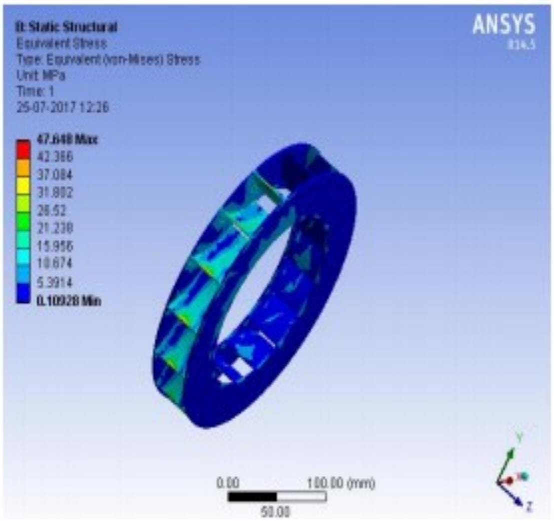

Figure 9: Stress

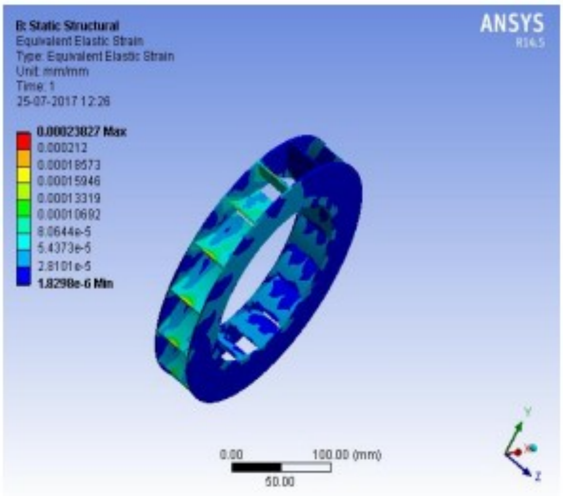

Figure 10: Strain

Table 1: 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

Table 2: Static Structural analysis

	Speed (RPM)	Defor-mation (mm)	Strain	Stress (MPa)
Steel	2800	0.014058	0.00023827	47.628
	3000	0.01625	0.00027542	55.076
	3200	0.017626	0.0002798	59.732
Alumi-num Alloy	2800	0.033424	0.00056546	40.142
	3000	0.038682	0.00065441	46.456
	3200	0.04159	0.00070358	49.947
E-glass tproxy	2800	0.032647	0.00056323	40.774
	3000	0.037784	0.00065186	47.191
	3200	0.40413	0.00070057	50.171
Aramid fiber	2800	0.20436	0.0034512	37.958
	3000	0.23661	0.0039957	42.946
	3200	0.25346	0.0042833	47.109

IV. 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% at2800rpm, by 14.3% at 3000rpm, by 15% at 3200rpmwhen 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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