Numerical Analysis of Centrifugal Air Blower

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Abstract— The turbo machine is an energy conversion device which converts mechanical energy to kinetic/pressure energy or vice versa. The conversion is done through the dynamic interaction between a continuously flowing fluid and rotating machine component. Turbo machines comprise various types of fans, blowers, compressors, pumps, turbines etc. More and more experimental research work is available in the field of turbo machine design and its evaluation. Literature review has revealed that a few literatures are available on three dimensional numerical analysis of a centrifugal fan/blower. Literature review in present work is highly focused on centrifugal blower and use of CFD techniques in turbo machines. In this course of work, input parameters and design parameters of centrifugal blower is obtained as per church and Osborne design methodology developed by Kinnari Shah, PROF. NitinVibhakar. Fluid model is made as per this design data in PRO-E SOFTWARE. And this fluid model is simulated using computational fluid dynamics (CFD) approach in ANSYS (CFX). Numerical analysis carried out in this work is to understand the flow characteristics at design and off-design conditions under varying mass flow rates, varying rotational speeds and number of blades in both design methodology. This numerical analysis is under consideration of steady flow and for rotational domain (frozen rotor interference) is used. Performance curves are obtained under different variable inlet parameters like volume flow rate, rotational speed and number of impeller blades. Here mass flow rate as a inlet boundary condition and static pressure as a outlet boundary condition. Volume flow rate is changed by changing the mass flow rate at inlet. Overall work carried out on flow behaviour and performance graphs for different cases are discussed in length in results and discussions chapter. Comparative evaluation of two design method indicates that error in static pressure gradient is higher in Osborne design rather than church design, and performance parameters are better for church design than the Osborne design.

Key words: Numerical Analysis, Centrifugal Air Blower

I. Introduction

Centrifugal Blower is one of the powers consuming turbo machine where large volumes of gas or air at low pressures are required. Pressure ratio or compressed air varies from 1.1 to 4.0

A blower, according to the Compressed Air Institute, is a machine to compress air or gas by centrifugal force to a final pressure not exceeding 240 KPa. It is not water cooled, as the added expense of the cooling system is not justified in view of the relatively slight gain at this pressure.

In these research work , two design methodology church and Osborne for centrifugal air blower was taken.

Numerical simulation under varying number of blades condition, and varying speed (rpm), and varying discharge at design as well as off-design conditions.

To understand the flow pattern inside centrifugal blower in detail.

To plot individual performance graphs for church design (FC radial tipped) and Osborne design (FC redial tipped) centrifugal blower.

Compare static pressure gradient of both design methodology with design condition.

Compare performance parameters of both design methodology with design condition.

Here taken optimized number of blade z =16. [6]

Flow Discharge Q	0.5 m^3/s
Static Suction Pressure	-196.4 N/m^2
Static Delivery Pressure	784.8 N/m^2
Static Pressure Gradient ΔPs	981.2 Pa
Speed of impeller rotation N	2800 rpm
Air Density ρ	1.165 kg/m^3
Optimized number of blade z	16
Outlet Blade Angle β_2	90°
Suction Temperature Ts	30 °C=303 K
Atmospheric Pressure P_{atm}	1.01325 x 10^5 Pa
Atmospheric Temperature T_{atm}	30° C = 303 K
Blade thickness t	2 mm

Table 1: Input Design Parameters [3]

Fan Design Optimum Parameters Comparison	Unit	Church Design	Osborne Design
At Impeller Inlet			
Inlet Duct Diameter D_{duct}	mm	200	215
Eye Diameter D_{eye}	mm	188	196
Eye Velocity V_{eye}	m/s	18.00	17.52
Peripheral Velocity U_1	m/s	28.00	29.57
Relative Velocity W_1	m/s	33.84	30.84
Meridian Velocity Vm_1	m/s	18.99	8.76
Absolute Velocity V_1	m/s	18.99	8.76
Impeller Diameter D_1	Mm	191	202
Width Of Blade b_1	Mm	49	95
Air Angle	Deg.	90	90
Blade Angle	Deg.	35.17	16.50
At Impeller Outlet			
Peripheral Velocity U_2	m/s	41.56	43.79
Relative Velocity W_2	m/s	18.09	11.65
Swirl Velocity V_{U2}	m/s	33.4	36.1
Meridian Velocity Vm_2	m/s	16.14	8.76
Absolute Velocity V_2’	m/s	37.10	37.15
Impeller Diameter D_2	mm	284	299
Width of Blade b_2	mm	37	64
Air Angle	Deg.	25.79	13.64
Blade Angle	Deg.	90º	90º
At Volute/Scroll Casing			
Width of Casing b_v	mm	111	128
Outlet Velocity of Casing	m/s	19.65	34.4

V_4			
Scroll Radius at Inlet r_3	mm	147	154
Scroll Radius at Outlet r_4	mm	376	274
Scroll Height Hs	mm	229	119
Radius of Tongue R_t	mm	152	161
Angle of volute Tongue θ_t	Deg.	9	17
Blade Profile Radius R_b	mm	70.3	62.7
Power Required To Run Fan P	watt	769.1	742.2
Total Efficiency η	%	81.67	82.72

Table 2: Optimum Design Parameters as per Church and Osborne Design [3]

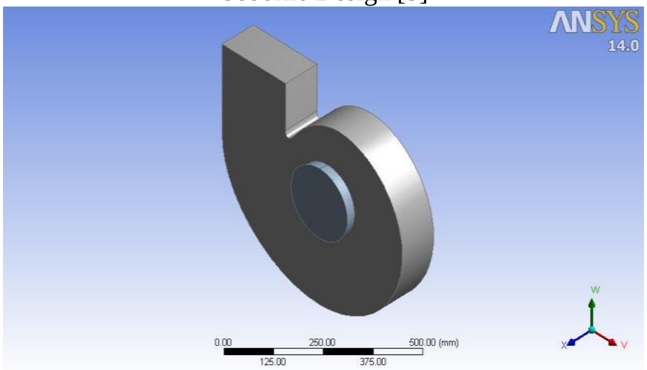


Fig. 1: Fluid model of centrifugal blower (church design methodology)

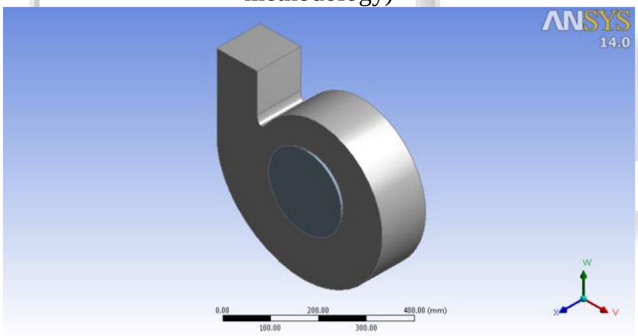


Fig. 2: Fluid model of centrifugal blower (Osborne design methodology)

II. SIMULATION PARAMETERS USED IN SOLVER

- Inlet boundary condition : mass flow rate at 5% turbulence intensity
- outlet boundary condition : static pressure at zero gradient turbulence intensity
- interface of domains nozzle and impeller interface, is given as frozen rotor impeller and casing interface, is given as frozen rotor

z	Nodes	Elements
8	38133	170312
12	38293	170360
16	38453	170408

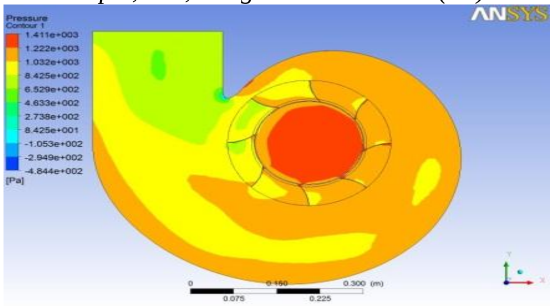
Table 3: Nodes and Elements at 8,12 and 16 number of blades for church design

z	Nodes	Elements
8	42313	152597
12	43209	153101
16	43977	153493

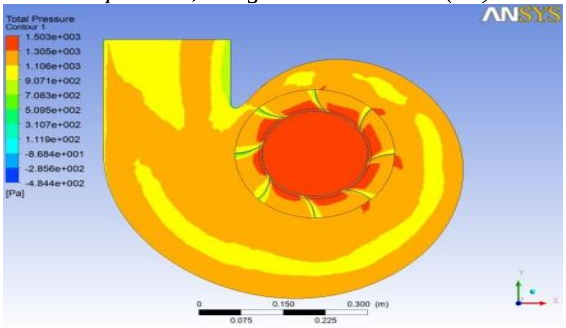
Table 4: Nodes and Elements at 8, 12 and 16 number of blades for Osborne design

A. From Church Design Methodology

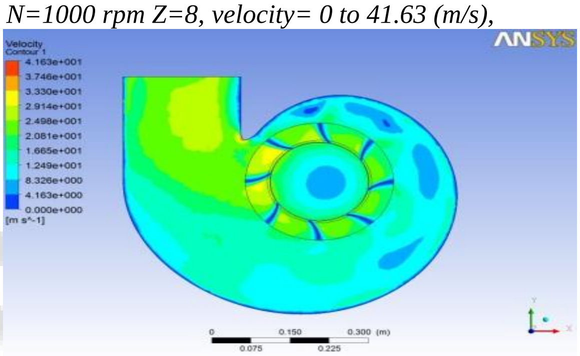
1) *N=1000 rpm,Z=8, Range: -484.4 to 1411(Pa)*



2) *N=1000 rpm Z=8, Range: -484.4 to 1503(Pa)*

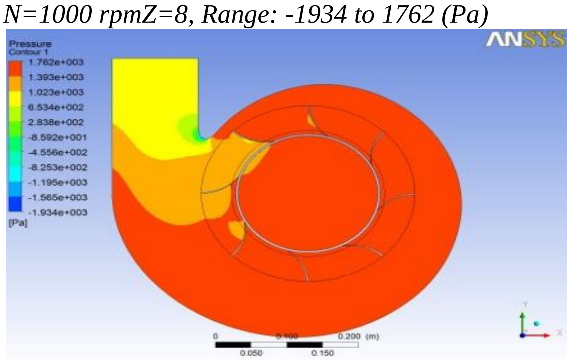


3) *N=1000 rpm Z=8, velocity= 0 to 41.63 (m/s),*

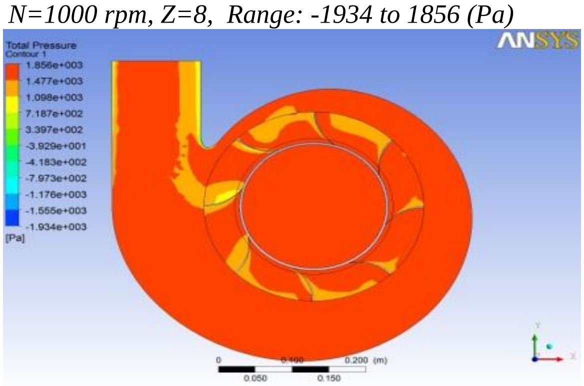


B. From Osborne Design Methodology

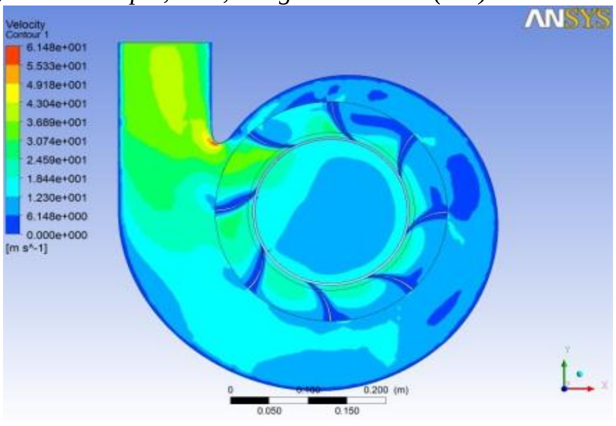
1) *N=1000 rpmZ=8, Range: -1934 to 1762 (Pa)*



2) *N=1000 rpm, Z=8, Range: -1934 to 1856 (Pa)*



3) N=1000 rpm, Z=8, Range: 0 to 61.48 (m/s)



III. PERFORMANCE GRAPHS

Overall performance of any turbo machine is generally shown by using graphs of dimensionless coefficients. Dimensionless coefficients normally used are:

1) Flow coefficient $\emptyset = \frac{Q}{ND^3}$
2) Pressure rise coefficient $\Psi = \frac{Pt_2 - Pt_1}{\rho N^2 D^2}$
3) Power coefficient $\wedge = \frac{P}{\rho N^3 D^2 D^3}$
4) $\eta = \frac{\emptyset \times \Psi}{\wedge}$

Where, Q = Volume flow rate in m^3/s
N = Rotational speed of impeller in rps
D = Outer diameter of impeller in m
ρ = Air density in kg/m^3
P = Power drawn by fan in W = Torque * Angular Speed = T*ω
P_{t2}-P_{t1} = Total Pressure rise across fan in Pa

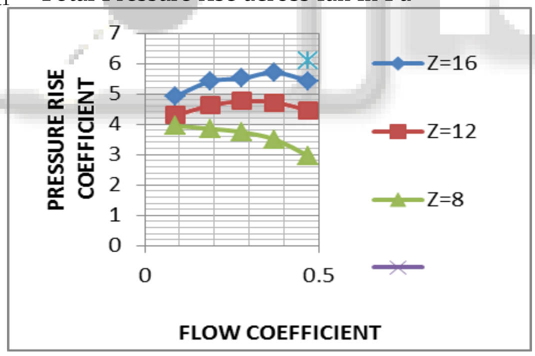


Fig. 3: Flow coefficient v/s pressure rise coefficient for church design methodology for Q=0.1 to 0.5 m3/s, and N=2800 rpm and z=8, 12, 16 and design point

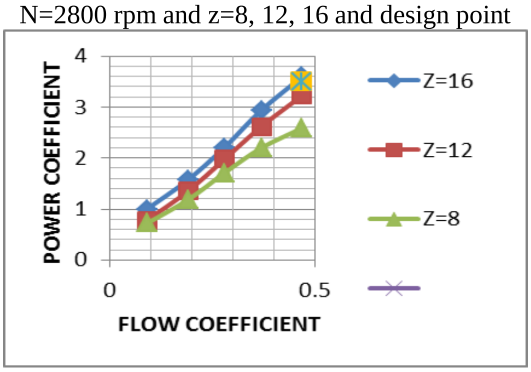


Fig. 4: Flow coefficient v/s power coefficient for church design methodology for Q=0.1 to 0.5 m3/s,and N=2800 rpm and z=8, 12, 16 and design point

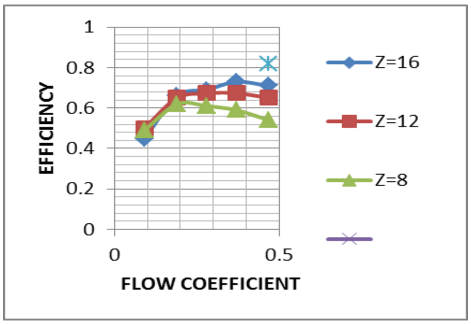


Fig. 5: Flow coefficient v/s efficiency for church design methodology for Q=0.1 to 0.5 m3/s, and N=2800 rpm and z=8, 12, 16 and design point

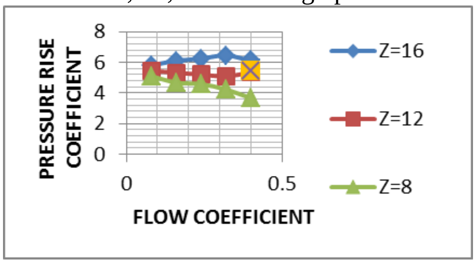


Fig. 6: Flow coefficient v/s pressure rise coefficient for osborne design methodology for Q=0.1 to 0.5 m3/s, and N=2800 rpm and z=8, 12, 16 and design point

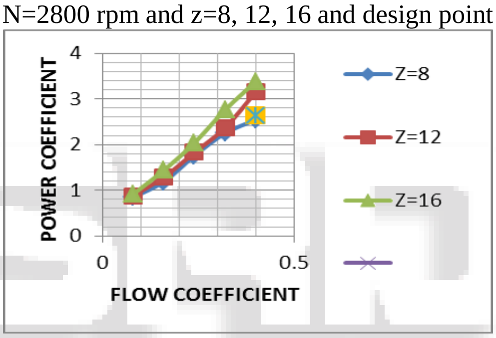


Fig. 7: Flow coefficient v/s power coefficient for osborne design methodology for Q=0.1 to 0.5 m3/s,and N=2800 rpm and z=8, 12, 16 and design point

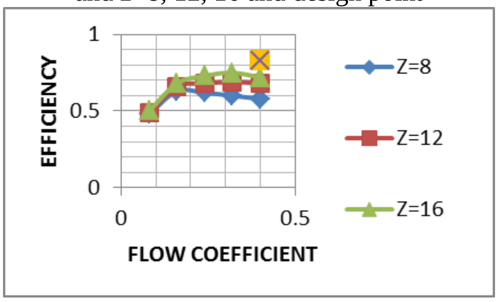


Fig. 8: Flow coefficient v/s efficiency for osborne design methodology for Q=0.1 to 0.5 m3/s, and N=2800 rpm and z=8, 12, 16 and design point

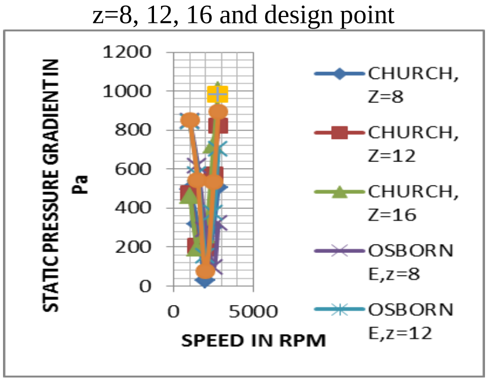


Fig. 9: Comparison of static pressure gradient v/s speed for church and osborne design methodology for Q=0.5 m^3/s, and N=1000,1500,2000,2500,2800 rpm and z=8, 12, 16 and design point

IV. CHURCH DESIGN

A. *Static Pressure Gradient*

From simulated data static pressure gradient = 785.251-(-222.951) =1008.2 Pa

At, design condition, static pressure gradient =981.2 Pa

So, % error in static pressure gradient = 2.68%

V. OSBORNE DESIGN

From simulated data static pressure gradient=785.078 – (-109.323) = 894.4 pa

At, design condition, static pressure gradient = 981.2 Pa.So, % error in static pressure gradient = 8.8 %

VI. CONCLUSION

Numerical results have given enormous flow visualization results within blower geometry under study. Many flow parameters are obtained numerically and graphically.

After critical evaluation of information obtained from all simulated cases and comparing it with designed and experimental data available for backward and forward curved radial tipped centrifugal fans with varying N, Q and Z, following conclusions are derived.

1) Comparative evaluation of church and Osborne design methodology (forward curved radial tipped blade) indicates that error in static pressure gradient is higher in Osborne design methodology rather than church design methodology.
2) Non dimensional parameters also shows better performance values with design values for church design method than the Osborne design method.
3) The theoretical and numerical analysis (CFD) is closer to design point conditions in centrifugal fan under study
4) Better performance parameters are achieved in church design methodology than the Osborne design methodology.
5) Efficiency for both design methodology are closer to the design point efficiency .
6) Design of tongue is very important in design to reduce back flow and recirculation.

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