

USING WAVES TO SIMULATE DISTORTION IN INTERNAL FLOWS IN AN AXIAL FLOW FAN

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Abstract

When the compressor experiences off design operation, it is likely to be due to an inlet distortion, which determines its operational sustainability. Every compressor in each flight segment undergoes non-uniform inlet flow fields and responds differently to that of a clean inlet flow, for which it is designed. To meet specific performance requirements, it is often required for a compressor to pass through off-design environments. Distortion, simulated as waves or combination of waves in propagation, have been studied in this work. It has been observed that the wave first elongates circumferentially in opposite direction to that of the rotor rotation direction till the rotor exit plane, and then in the rotor rotation direction in the exit domain. For a given velocity in distorted passages spike wave shows more detrimental effect compared to normal / modal distortion waves.

1 Introduction

Usually any axial flow compressor is designed for uniform flow at the inlet sector; however in practice this may not be always realized. The installation of compressor normally involves some form of inlet duct to direct the flow. Due to flight maneuvers, intake design, bends, boundary layer build up and even obstacle in the flow path the flow entering the compressor may be far from clean or uniform.

Axial flow compressor operation range depends on the availability of stall margin during operation. Before commencing use of a compressor the user must be aware of its stall margin. Basically there are two types of stall

mechanisms in an axial flow compressor that have been identified. The first type of stall mechanism is associated with “modal wave” or “long length scale disturbance” and second type of stall mechanism is associated with “spike” or “short length scale” disturbance.

Modal wave is a small amplitude and low frequency propagating disturbance that affect entire flow field with a characteristic circumferential wavelength, which is comparable to the circumference of the compressor; that’s why it is also called “long length scale disturbance”. These disturbances rotate around the compressor annulus in the direction of the rotor rotation. Modal wave propagates at approximately 20-45% of rotor speed, and increases to 40-60% they grow to fully develop rotating stall. Modal waves grow smoothly into fully developed stall cells without the presence of “short length scale disturbance” and the compressor is said to have stalled modally.

Spike wave types of disturbances have a circumferential length of a few blades pitches, about 30% of circumference, originate near the tip region, rotate at close to the shaft speed (70 -80% initially), and grow fully developed stall cells within a few rotor revolutions.

Spike and modal stall inceptions may appear distinct but are not really exclusive; both types of inception may be seen simultaneously in the same axial flow compressor. Whether a compressor stalls by (i) the modal wave, (ii) the spiky wave, or (iii) some combination of the two seems to be determined by the blade loading, particularly that at the rotor tip, when the compressor is operating near the system surge line. From figure.1 it is clear that stall inception by modal distortion is gradual or

progressive where as stall inception by spike distortion is abrupt or sudden.

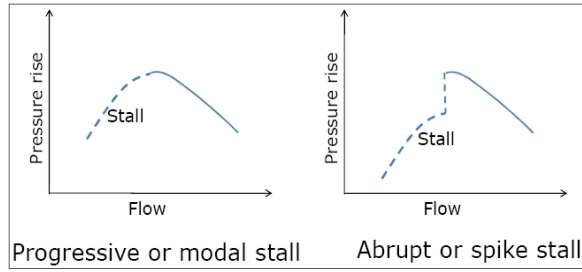


Figure 1 Modal and Spike Stalls

The aim of the present work may be stated as:

- Design of various distortion waves to simulate smooth, sharp & constant variation of total pressure at the Inlet.
- Using these distortion waves as the inlet boundary condition for a single stage axial flow fan model in commercial CFD software CFX[®] to obtain the flow fields.
- Study the effect of shape, size & number of distortion waves on the performance of axial flow fan model respectively.
- Calculate the distortion coefficient for each type of distortion waves generated and study its variation from inlet to outlet of the flow domain.
- Study distortion wave propagation from the inlet through the outlet of the fan.

1.1 Distortion: Inlet distortions are usually classified as (i) total pressure distortion or temperature distortion, based on which parameter is of major concern, and (ii) as radial distortion or circumferential distortion, based on how the spoiled flow sector is distributed shown in figure: 1, and as (iii) steady distortion and instantaneous distortion based on the time duration of distortion.

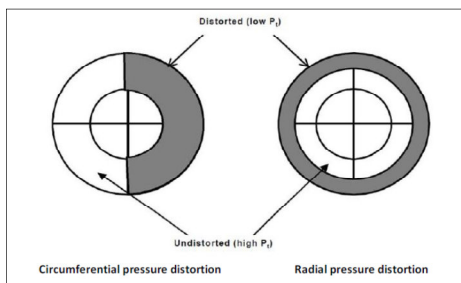


Fig.2 Circumferential and Radial Distortions
Distortion is basically defined as inlet non-uniformities driven, flow field response through

rotor and its implications on the performance and its characterisation. These non uniformities could be velocity driven or pressure variation also with local mass flow fluctuations.

ARP 1420 [22] report provides a set of guidelines by which gas turbine engine aerodynamic stability and performance, as affected by the quality of the airflow delivered to the engine, can be evaluated consistently. The subjects addressed in ARP are Distortion Descriptor, Stability Assessment, and Testing.

2.0 Literature survey

There have been numerous experimental and theoretical studies on inlet distortion and the compressor response over the past several decades. Extensive research in understanding the nature of distorted flow and its effect on the axial flow compressor performance has been done by researchers. There is considerable literature available which presents the experimental as well as numerical simulations on inlet distorted flow behaviour, its propagation and its effect on the compressor.

AR Wadia [1,2] et al evaluated the effects of inlet distortion on the stall margin of a two stage fan with (i) GE Swept Forward Aero Research fan test vehicle (GESFAR): forward swept rotor-I configuration and (ii) for High Tip Speed Compressor (HTSC): radial rotor-I configuration - where the second stage is common to both. A total of five distortion screen were used of which three of the five had either radial or circumferential inlet distortion patterns. The other two screens were a combination of the radial and circumferential distortions.

Huang Jian and Wu Ju [3] investigated the flow field to enhance the physical understanding of the relationship between inlet distortion and the Circumferentially Groove Casing Treatment (CGCT) on axial flow compressor with the help of NUMECA[®] software. They found that CGCT can expand the operating range of axial flow compressor with circumferential pressure distortion at inlet at the expense of a slight drop in the isentropic efficiency. The CGCT is capable of effectively suppressing the enhancement and forward

movement of tip leakage vortex in axial compressor. CGCT also eliminates the low energy zone in the outer wall region very effectively compared to a clean untreated inlet.

Ng et al [6] studied the inlet distortion propagation in compressor using integral approach with Taguchi method. They also found out that the major parameters which affects inlet distortion propagation are the ratio of drag to lift coefficient of the blade, the inlet distorted velocity coefficient and the blade inlet incidence angle. These parameters are ranked according to their influence on the distortion using a Taguchi table: the drag-to-lift coefficients of the blade has the most influence on the distortion, followed by the x -axis distorted velocity coefficient in inlet. The inlet incidence angle has the least influence on distortion.

F Lin and J chen [7] performed a theoretical study to investigate the stall triggering mechanism of inlet distortion induced instability of axial compressor system. Main focus was on the transient stall inception processes when the compressor is throttled from stable operating conditions to stall. A model for steady circumferential distortion screens, a new system model captures the non-linear interaction between inlet distortion and the axial compressor. It is found that the unstable long length scale disturbances always grow out of the spoiled sectors, but under certain conditions they may evolve into short length scale disturbance before rapidly triggering full size stall. It is pointed out that under appropriate conditions, modes and spikes may be independent from each other. They found that this is due to non-linearity of the compressor characteristics and of the distortion screen characteristics.

Chaoqun Nie [9] has experimentally studied the response of single stage low speed axial flow compressor to rotating inlet distortion. A rotating distortion generator equipped with different distortion sectors was designed to produced different rotating inlet distortion clockwise or anti clockwise with up to 100% of the compressor design speed. Three types of distortion sector are used in the research work, which are single 30 degree sector, four 30 degree sectors and single 120

degree sector. It is found that the total pressure loss caused by rotating sectors increase when the distortion speed rises. From Fig.5 it is seen that the distortion coefficient increases with the inlet distortion speed. With $4 \times 30^\circ$ distorted sectors DC_T obtained are much higher than $1 \times 30^\circ$ sector.

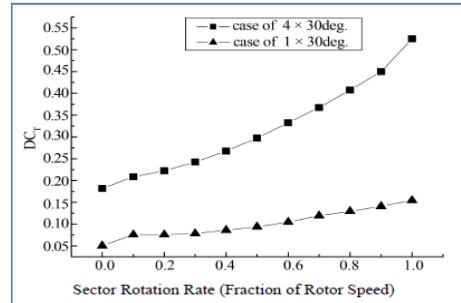


Fig.5 DC vs Distortion Speed [9]

Reid C [20] found that for the stationary circumferential total pressure distortion, while keeping the total distorted sector fixed but when split it into smaller segments, the loss of stability margin decreases as the value of individual sector angle increases. For rotating distortion, it is shown in Fig.6 that at the same distortion speed ratio the character line of $4 \times 30^\circ$ and $1 \times 120^\circ$ distorted sectors seems to have the same pressure rise coefficient at the stall point, but the latter has higher flow coefficient.

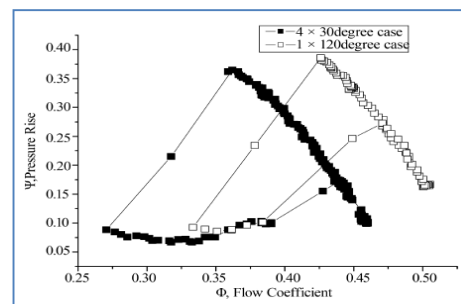


Fig: 6 Effect of divided distorted sectors [20]

Doulgeris [18] proposed a method for assessing the effect of inlet distortion on the fan of an aero engine through the coupling of a streamline curvature model with parallel compressor theory. Fan annulus is split in to circumferential sectors according to parallel compressor theory, while each sector is

modelled using streamline curvature method. In this way, the effect of highly non uniform flow patterns on the fan performance was assessed under inlet distortion.

3.1 Distortion Formulae: There are many formulae that have been given in literature to assess the circumferential distortion and radial distortion respectively. Most of the formulae are given using the terms average total pressure, maximum total pressure, minimum total pressure etc. Distortion index or distortion coefficient is the term used for indicating distortion value. Researchers have been developing formulae as per their application or availability of experimental test data and validated test results. Some of the distortion formula which are frequently used is given below.

The standard practice [3, 22] for specifying inlet distortion patterns for turbo fan engine manufacturers is to compute the circumferential distortion index (CDI), which is defined as

$$CDI = (P_{tmax} - P_{tmin})/P_{tmax}$$

Eisemann [17] defined a general distortion coefficient in a plane as:

$$\text{Distortion Coefficient} = (\theta_{max} - \theta_{min})/\theta_{avg}$$

Based on above concept an inlet distortion coefficient can be defined as below, which includes radial and circumferential distortion:

$$\text{Inlet DC} = (P_{tmax} - P_{tmin})/P_{tavg}$$

In a case of experimental study of distortion [1] followings formulae are used (Table 1).

The index for circumferential inlet distortion, IDC, is calculated using the inlet distortion rakes as:

$$IDC = 1.0 - (PM/PFA)$$

where, PM is the minimum total pressure and PFA is the face averaged total pressure (area averaged)

Similarly, the index for radial distortion, IDR, is computed using the inlet probe rakes as:

$$IDR = 1.0 - (PMR/PFA)$$

where, PMR is the minimum ring averaged total pressure (area averaged) and PFA is the face averaged total pressure (area averaged)

3.2 The Rotating Distortion Coefficient: It is well known that the total pressure is reduced when air passes through the distortion sectors. In 1969 C Reid [20] used a parameter, the distortion coefficient DC_ϕ , to quantify the total pressure loss caused by the rotating distortion sector(s). The parameter is defined with the inlet total pressure P^*_{Inlet} , the average total pressure behind the distortion sector(s) $\langle P^*_{Dist} \rangle$, and the dynamic head, $0.5\rho U^2$.

$$DC_T = \frac{P^*_{Inlet} - \overline{P^*_{Dist}}}{0.5\rho U^2}$$

A list of useful distortion formulae (radial and circumferential distortions) and of rotating distortion coefficient are given in Table.1.

Table: 1 List of Distortion Formulae

	Various Distortion Formulae
1	Circumferential Distortion Coefficient; $CDI = (P_{tmax} - P_{tmin})/P_{tmax}$
2	Distortion Coefficient; $DC = (\theta_{max} - \theta_{min})/\theta_{avg}$; Where, θ can be any flow variable
3	Most Usable Distortion Formula: Inlet $DC = (P_{tmax} - P_{tmin})/P_{tavg}$
4	In case of Experimental index : Circumferential Distortion Coefficient; $IDC = 1.0 - (PM/PFA)$
5	In case of Experiment : Radial Distortion Coefficient; $IDR = 1.0 - (PMR/PFA)$
6	Rotating Distortion Coefficient ; DCT $= [P^*_{inlet} - \langle P^*_{Dist} \rangle] / \frac{1}{2} \rho \cdot U^2$

4.0 Details of Axial Flow Fan Model: The axial fan available in the Propulsion laboratory, IIT Bombay is a single stage axial flow fan, rotating with a designed speed of 2400 rpm and a design pressure rise of 1000 Pa. The fan has been modeled in CATIA® & meshing has been done in ANSYS® Turbo grid. The blade profile

was generated using C4 profile equation in MATLAB[®]. The leading particulars of low speed single stage axial flow fan are given in Table:2. The same model has been used in this study in ANSYS-CFX[®] for analysis of effect of inflow distortion. It consists of mainly three module inlet domain, rotor domain and outlet domain which are shown in Fig. 8.

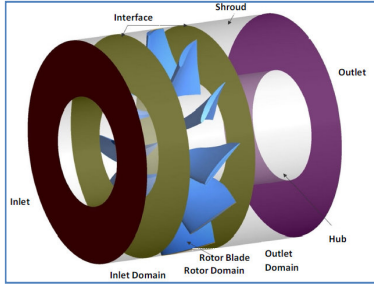


Fig: 8 Rotor and Annulus Flow Domains

Table: 2 Leading Parameters of Axial Fan

Number of rotor blades	11
Aspect Ratio	1
Shroud Diameter	40.6 cm
Hub Diameter	20 cm
Tip Clearance	0.02% of span
Stagger Angle	31 ⁰ at Mid span
Camber Angle	32.2 ⁰ Mid span
Diffusion Factor	0.44
Solidity	1.16
Design Mass Flow Rate	3.5 Kg/s
Design Axial Velocity	29.14 m/s
Design Total P Rise	1000 Pa
Total Length of Domain	40 cm

4.1 Grid Generation

Grid generation of the axial flow fan model has been done and is explained here. For single passage and complete annulus rotor analysis, the grid has been generated using ANSYS[®] Turbo-grid[®]. Structured grid with multiple blocks is used. The entire flow within a single passage is captured by approximately 0.7mn nodes to get grid independent solution. The O-grid around the blade surface is created with 48 elements and width of the O-grid from hub to tip is created with an expansion ratio of 1.213. Maximum Y-plus value for the uniform flow case for the grid independent mesh is found to be around 1.9 which is a reasonable value for an SST turbulence model. J-type topology grid is

used both at leading and trailing edge of the blade. Flow in the region above the blade tip is captured with H-grid non-matching.

An element density of 80 is used between blade hub and tip while the passage has 50 elements of constant height. These values are taken in order to capture the span-wise flow in a better way. The grid between the blade tip and shroud has 20 elements for a tip clearance of 0.02 of span. For the inlet and outlet domain, H-type topology grid with one to one interface with rotor is used. Once single passage meshing is done, it has been transformed into a full annulus rotor domain by rotating it by 360 degrees in CFX[®], assuming axi-symmetry.

4.2 Solver and Boundary Conditions

The problem chosen involves a steady state simulation of flow through the full annulus of the rotor where the flow is turbulent from the inlet to outlet. Therefore, the numerical simulation involves solving the mass, momentum and energy equations with SST turbulence modelling. For advection, a specified blend factor of 0.95 has been used.

The boundary conditions for the simulation are based upon inlet total pressure, inlet total temperature and mass flow rate at the outlet. For a flow in a rotor problem, inlet and outlet are stationary domains while rotor is a rotating domain. It is necessary to specify the shroud as a counter rotating wall.

Table.3 Boundary Conditions for Clean Flow

Parameter	Value
Inlet Total Pressure	101325 Pa
Inlet Total Temperature	298 K
Mass flow rates at Outlet (kg/s)	2.8, 2.9, 3.0, 3.1, 3.2, 3.3, 3.4, 3.5, 3.6
turbulent intensity	5% (Inlet)
Rotational Speed	2400 rpm
Flow Region	Subsonic
Flow Direction	Normal to Boundary

Table.3 gives the design point boundary condition used for clean uniform inlet flow analysis, result of which will be used as reference value to compare results of various types of distorted flow conditions imposed.

Boundary conditions used in the cases of distorted inflow are: (i) inlet total pressure, (ii) total temperature - as inlet boundary conditions, and (iii) flow rate as exit boundary condition.

4.3 Turbulence Model , Convergence Criteria

The standard k- ϵ model is used for most turbulent flow calculations because of its reasonable accuracy and robustness. However this model performs poorly when there is stall in the flow. It will predict the stall too late and also will under predict the amount of separation. To solve this type of problems SST turbulence model is often used. The model works by solving turbulence based model (k- ω) at the wall and k- ϵ in the bulk flow. Blending functions ensures the transition between these two models. Through experimentation it has been found that a blending factor of 0.8-1.0 gives better performance for the full annulus geometry with and without inlet distortion. A convergence criterion of 1×10^{-5} RMS is used for full annulus simulations. The domain details of inlet, rotor & outlet are given in Table: 4.

Table: 4 Domain Details for Uniform Flow

Domain	Frame Type	Region	Boundary Conditions
Inlet Domain	Stationary Frame	Inlet	Total Pressure & Temperature
Outlet Domain	Stationary Frame	Outlet	Mass Flow Rate, kg/s
Rotor Domain	Rotating Frame	Blade, Hub, Shroud	No Slip Wall
Inlet and Outlet Domain	Stationary Frame	Hub, Shroud	Free Slip Wall

5.0 Sector & Passage Definition: The Single stage axial flow fan model is having three domain named as inlet, rotor & outlet domain respectively. This single stage axial flow fan is having 11 rotor blades & hence same number of passages generated in inlet domain & in outlet domain while transforming from single to full annulus in CFX-pre®. Each passage occupies about 32.72 degree. Further for sake of applying distortion circumferentially each passage is divided into two sectors. In this report 1 sector

means 16.36° & 1 (blade) passage means 32.72° of circumferential spread at the rotor inlet.

5.1 Method of Applying Distortion Waves:

There are mainly two methods of applying distortion waves circumferentially. The first method is to keep the number of distortion waves constant in a sector from hub to tip. In this method distortion wave shape is the same from hub to tip but not the size. For example if we apply a sine curve from the hub to the tip the sine curve size changes from hub to tip because the circumference changes. Second method of applying distortion wave is to keep the shape and the sizes same from the hub to the tip. In such a case the number of distortion waves will vary from the hub to the tip. In the present study the first method has been used.

5.2 Circumferential spread of Axial Velocity:

As we know that fan or compressor are designed to give maximum performance for uniform inlet flow. There are many reasons which can lead to non-uniform flow at the inlet. If axial velocity is constant at all inlet circumferential location then it is a case of uniform flow. But during maneuvering of aircraft this may not be the case. The inlet axial velocity may vary in the circumferential direction means out of 360° , in some sectors velocity is different from the rest. In this study out of 11 passages, only three passages in circumferential direction are applied with distortion waves. Most of the experimental distortion results are available for 90° sector. In this study the 3 passages equals 96° . The inlet total pressure values are varied in the 3 passages in such a manner that the variations resemble different waves like sine wave, square wave, triangle wave, Gaussian wave - within the three passages. There is no variation of pressure or velocity field in the radial direction from the hub to the tip in the distorted three passages. Inlet axial velocity is held uniform and constant in the remaining 8 passages in all directions.

To generate fan map for various kinds of distortion waves, one has to use different mass flow rate conditions and total pressure as the inlet boundary conditions in CFX-pre®. In CFX-pre® analysis one boundary condition has to be pressure based - only then it produces solution.

The velocity profile is then used to obtain total pressure profile at the inlet using standard relations keeping total inlet pressure atmospheric. Since this is a low speed fan the operating Mach number is low and flow can be assumed to be incompressible. Hence the stagnation pressure is derived from the Bernoulli's equation for incompressible flow.

5.3 Types of Distortion Waves Used: There are in total 10 inflow distortion wave patterns generated in MATLAB® & the same have been embedded in the inlet boundary conditions in CFX-pre®. Wave patterns are chosen such that it simulates non uniformity of inlet axial velocity & hence the total pressure distortion at the inlet plane. Mainly three types of distortion wave patterns are used for simulation smooth wave, sharp wave & constant. The shape & description of waves are described below.

A Matlab code has been generated to create the velocity and pressure profile to simulate the distorted flow condition at the inlet of compressor. A passage is the area of flow between the two adjacent blades of the compressor. Since present compressor model consist of 11 rotor blade and hence each passage will be of about 32.72 degree. Further each passage is divided into two sector of angle 16.72 degree each. It is assumed that for each sector velocity is varying sinusoidally along the circumferential direction whereas it remains constant along the radial direction. Hence in one passage there is twice sinusoidal variation of velocity take place.

This is achieved by developing two concatenated 'for' loop. The outer loop changes the radius from the hub to the shroud of the compressor, whereas, the inner loop spans over the complete 360 deg angle of the compressor. A sector is defined in the inner loop by providing the start and end angle depending upon the compressor inlet geometry and velocity is made to vary sinusoidal with its argument varying from 20 deg to 160 deg to obtain a symmetric profile. The remaining area has given constant velocity parameter. Hence, for a particular radius set by the outer loop, complete 360 deg profile has been generated.

Overall performance analysis and circumferential survey of total pressure, static pressure and velocity are obtained for a single stage axial flow fan with inlet disturbances applied to three passages of inlet of a fan or about 27% of the inlet annulus area. To simulate inflow distortion in the above mentioned region total pressure profile wave equations e.g. sine, square, triangular & Gaussian of different strength have been applied as inlet boundary conditions (Fig.9). These are circumferential type of distortions where total pressure variation is along the circumferential direction only and in the radial direction it is held constant.

For applying the above wave conditions at inlet domain, the total pressure profiles are generated in MATLAB to simulate distortion waves. Air mass flow rate is used as the outlet boundary condition. It has been found that the fan performance is directly related to the type and size of distortion wave applied at the inlet plane. For all distortion waves applied the distortion coefficient (DC) values have been calculated to characterize the waves (Table.8).

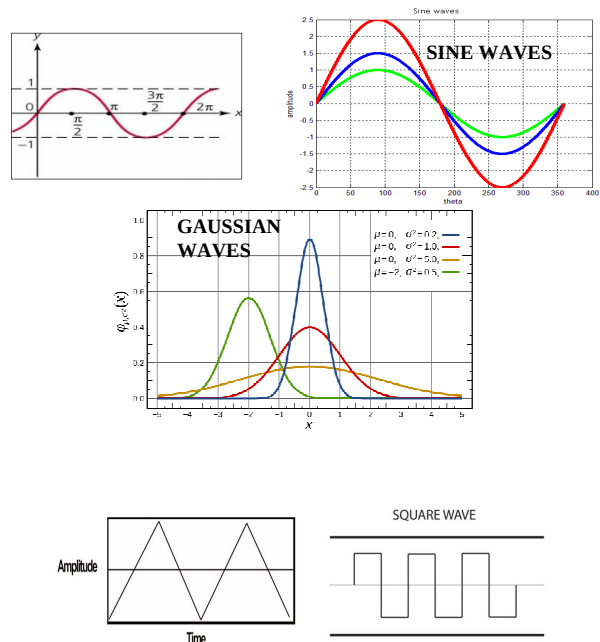


Fig.9. Wave forms applied as distortion waves

The velocity profile is then used to obtain distorted inlet total pressure profile using the Bernoulli's equation, keeping the average total inlet pressure equal to the atmosphere.

The code also incorporates a factor 'k' in order to amplify or attenuate the amplitude of the velocity profile. The obtained result is then converted from Polar to Cartesian axis system and exported in '.csv' format as required by CFX. The results has been plotted using Matlab[®] inbuilt contour plot function for the different cases.

6. Results and Discussions

6.1 Distortion Coefficient Variation: After calculating distortion coefficient at all the planes it has been plotted with respect to axial distance as shown Fig.11. From the plot it can be seen that the value of DC is almost constant in inlet domain (stn.s 1-3) and just before the rotor entry face (stn.4) DC starts increasing due to the rotor. Therefore DC increases in the rotor domain & then start decreasing as the flow goes past the rotor. Also, from the results available it is clear that for distortion wave which has the highest inlet velocity DC is the highest at inlet, since it has highest dynamic pressure. DC from the inlet to the exit of the fan model is lowest for the clean inlet flow (Fig.11 & 12, and Table 8). DC curve for clean inlet flow represents the inbuilt distortion present in the system (Fig.11 & 12). Flow through the distorted quadrant remains moderately distorted in the exit. In the full annulus, wherein one quadrant is distorted, (Fig.12) the flow exits the flow domain with low DC, with peaks are clearly within the rotor.

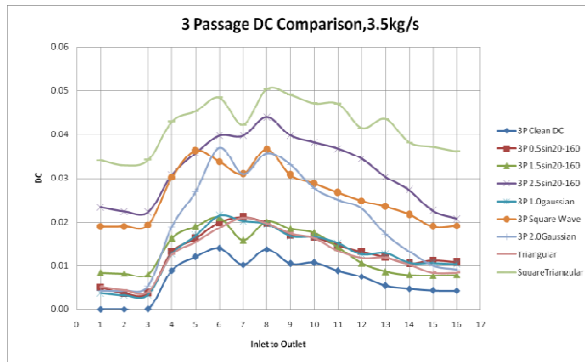


Fig: 11 Three Passage Distortion Coefficients

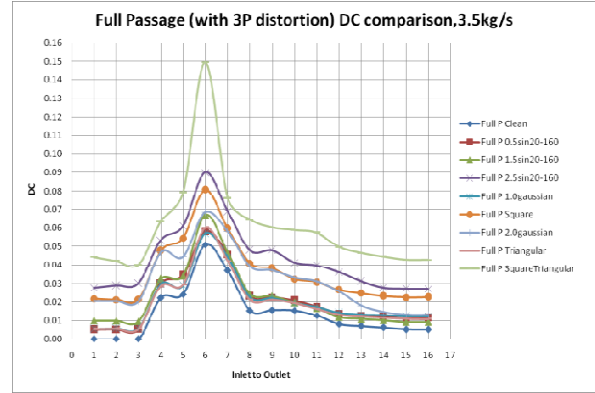


Fig: 12 Full Passage Distortion Coefficients

Table: 8 Performance Results at Design Point

Types of Wave	Stall Margin %	P_{02}/P_{01}	D C ($\times 10^{-2}$)
Clean	22.20	1.0096	5.09
0.5sin20-160	2.80	1.0093	5.81
1.5sin20-160	13.80	1.0094	6.70
2.5sin20-160	2.80	1.0051	8.99
Square Wave	8.30	1.0077	8.07
1.0Gaussian	4.10	1.0090	5.71
2.0Gaussian	13.80	1.0083	6.85
Triangular Wave	6.90	1.0091	6.01

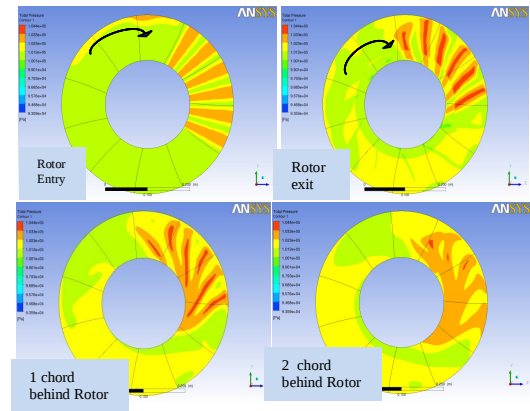
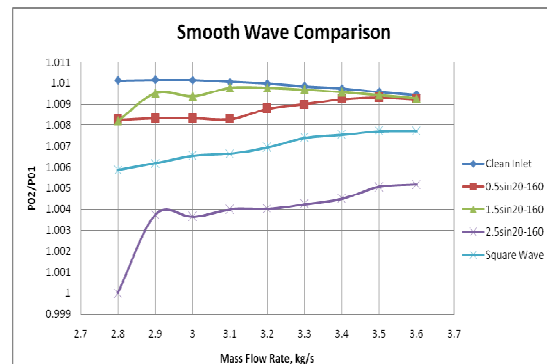


Fig.13 3-Passage 6-Wave-2.5sin20-160 Distortion effect through the rotor



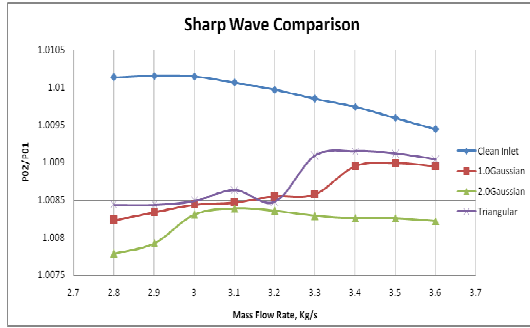


Fig.14. Effects of modal (smooth) and Spike (sharp) waves on axial fan performance

Spread of the distortion in the circumferential direction is captured in Fig.13. Radially uniform modal distortion, after passing through the axial fan rotor immediately spreads itself counter to the rotor rotation. Downstream the distortion loses strength and it wraps itself around the hub in the direction of rotation of the rotor.

Performance of the axial fan, when the modal waves or the spiky waves are applied at the inlet, have been captured in Table.8 and in Fig.14. Two inferences can be drawn from the figure : (i) more the frequency of waves in a sector worse is the performance, (ii) spiky waves (Gaussian) give worse performance and much lower stall margin, compared to the modal waves.

7.0 Conclusion

- There is a direct relationship between the number of distortion waves per passage & the performance of axial fan; usually more the number of distortion waves poorer the performance
- If the distortion wave is applied on a few passages then pressure rise behaviour of distorted passage, undistorted passages & full passage are different for smooth waves and spike waves – the reason being difference in average total pressure in both the cases.
- There are a number of ways one can define the strength of distortion wave numerically.
- For the given maximum velocity in the distorted passages spike wave shows more detrimental effect compared to the normal distortion waves.
- Distortion coefficient remains almost constant in inlet domain & increase in rotor

domain reaches highest at the rotor tip then starts decreasing till outlet

- Distortion wave propagation through an axial fan has been studied and it has been observed that it first spreads circumferentially in the opposite direction to the rotor rotation till the rotor exit plane, and thereafter in the rotation direction in the flow outlet domain (Fig.13).

8 Nomenclature

D_t	Shroud Diameter
D_h	Hub Diameter
$\dot{m}$	Air Mass Flow Rate
A	Annulus Area
C_a	Axial Velocity at Inlet
P_{01}	Average Total Pressure at Inlet Plane
P_{02}	Average Total Pressure at Outlet Plane
P_{tmax}	Maximum Total Pressure in a Plane
P_{tmin}	Minimum Total Pressure in a Plane
P_{tavg}	Average Total Pressure in a Plane
DC	Distortion Coefficient
DC_T	Rotating Distortion Coefficient
IDC	Circumferential Distortion Index
IDR	Radial Distortion Index
$CGCT$	Circumferential Groove Casing Treatment
C_p	Coefficient of Pressure
U	wheel speed at mean diameter
ρ	Air Density
θ_{max}	maximum of flow variable in a plane
θ_{min}	minimum value of the flow variable
θ_{avg}	mass-weighted average flow variable
μ	mean or expectation of distribution
σ	Standard Deviation
σ^2	Variance

Subscripts:

Dist	Parameter inside distorted flow
Inlet	Inlet Parameter
Static	Static Pressure/ Temperature
Stn.....	Parameter in Stationary Frame
Rel.....	Parameter in Relative Frame

Abbreviations:(refer Fig-9)

***sin20-160 ...	Wave extent from 20° to 160°
0.5sin20-160....	Half wave in a sector passage
1.5sin20-160 -	One & half waves in a sector
2.5sin20-160 -	Two & half waves in a sector
1.0Gaussian -	One Gaussian wave in a sector
2.0Gaussian -	Two Gaussian waves in a sector

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