INVESTIGATION OF THE EFFECTS OF VARIOUS PARAMETERS ON A SUBSONIC AXIAL FLOW FAN PERFORMANCE AND TONAL NOISE

A Thesis in
Aerospace Engineering

by

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ABSTRACT

This work was aimed at improving the efficiency and decrease the noise generated by a cooling fan used by GE transportation to cool a row of resistor networks which dissipate excess energy generated by regenerative power in an inverter application.

The first part of study focused on determining the effect of various parameters on the performance of the fan quantitatively by using FLUENT, a general purpose CFD code. The effects of parameters such as hub to tip ratio, chord length/thickness and solidity/number of blades was studied. Once the effect of these parameters was found a new fan was designed which increased the efficiency at the operating point from 73% to 77%. This new fan had a longer blade (increased by 0.75 inches), less number of blades (24 reduced to 18) and a larger chord length (the airfoil scaled by 1.5 time). The hub shape was also modified to reduce separation at the hub.

Although a more efficient design, this new fan was found to make more noise with the peak dB levels increasing from 89dB to 110dB due to a higher rate of change of axial force on blades which is proportional to the sound pressure level. Thus a different approach was adopted, an inlet guide vane (IGV) with different lean angles were experimented with. It was seen that a 6 degree lean angle eliminated the peak at the first harmonic of the blade pass frequency (2400Hz) and a 15 degree lean angle eliminated the first and the second harmonic of the blade pass frequency (2400Hz and 3600Hz). The first peak at the blade passing frequency remained approximately the same (~89dB) with the IGV lean.
FOREWORD

Firstly I would like to thank my supervisor Professor Savas Yavuzkurt for his support and guidance without which this project would not have been possible. I would also like to thank Ryan Linker and Sean Cillesesen for their technical support for the project. I am also grateful to Professor Cengiz Camci for providing me with technical knowledge related to the project. Also, I like to thank my parents and friends for their motivation throughout the project.

July 2015

Racheet Matai
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NOMENCLATURE

C  Chord length
C_{1\varepsilon}, C_{2\varepsilon}  Model constants
c  Ambient speed of sound
E  Internal energy
F  Gravitational body force
F  Unsteady rotor force
G_k  TKE source term
H  Total enthalpy
K  Thermal conductivity
K  Turbulence kinetic energy
N  Number of cells
N  Rotational speed
p  Pressure

Q  Flow rate
R  Gas constant
R_\varepsilon  Positive/negative contribution to \varepsilon
t  Temperature
\nu_i  Velocity tensor notation
\u_r  Velocity of moving frame

\u_r  Friction velocity
\nu  Absolute velocity

\nu_r  Relative velocity
\nu_t  Translational frame velocity
x  Distance between the fan and point of pressure measurement
Y_M  TKE dissipation term

y^+ (= \rho y u_\tau / \mu)  Dimensionless wall distance

y^*  Dimensionless wall distance (Eq. 9)

Greek letters
\alpha  Sweep angle
\alpha_s  Swirl constant
\alpha_k  Inverse effective Prandtl number for k
\alpha_\varepsilon  Inverse effective Prandtl number for \varepsilon
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$\beta$</td>
<td>Lean Angle</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Rate of dissipation of TKE</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Angle between fan axis and point of pressure measurement</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Viscosity</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
</tr>
<tr>
<td>$\tau$</td>
<td>Total stress</td>
</tr>
<tr>
<td>$\tau_w$</td>
<td>Wall shear stress</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>Characteristic swirl number</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>Angular velocity</td>
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1. INTRODUCTION

One class of machines that are capable of transferring energy to and from a fluid are known as turbomachines. Work is either done by or on the fluid by a rotating row of blades called the rotor. If work is extracted by the rotor from the fluid the turbomachine is known as a turbine and if the work is done by the rotor on the fluid the turbomachine is labelled as a compressor, fan or a pump. A common way to classify them is by looking at the nature of the flow through the rotor. If the flow is parallel to the axis of rotation then the machine is axial. If it is perpendicular to the rotation axis then it is a centrifugal machine and if the flow is a mixture of both types then it is a mixed flow machine. Figure 1 (a), (b) and (c) show the three types of turbomachines.

![Diagram of turbomachines](image)

Figure 1: Forms of turbomachines (a) axial flow, (b) mixed flow, (c) radial. [1]

In this study the turbomachine being investigated is a ducted axial flow fan with inlet guide vanes (IGV) and outlet guide vanes (OGV). The OGVs in the present setup do not contribute to any static pressure rise and are only present to reduce the swirl component of the velocity exiting the rotor.
1.1 Objectives

The objectives of this study are to determine the effects of various parameters such as hub to tip ratio, chord length, solidity on the performance and noise generated by an axial flow fan. Also, the acoustic signature of inlet guide vanes with different lean angles (6 and 15 degrees) is studied. The effect of changing the design of the hub on the performance of the fan is also examined.

1.2 Setup

Dynamic breaking utilizes electric traction motors as generators when slowing down a vehicle. The power generated by the generator can be dissipated by a resistor grid or can be used elsewhere. The fan under investigation is currently used to cool a row of resistors which dissipate heat generated by the regenerative power in an inverter application. Figure 2 shows the box geometry which contains the rows of resistors and the fan. This box has two double ended fans (The two fans are mounted on the same axis and face each other.), i.e., a total of 4 fans with each fan cooling a row of 5 to 6 resistors. The 24 inch (0.6096m) diameter fans rotate at speeds ranging from 2500-3900 RPM. The assembly has 37 inlet guide vanes (IGVs), 24 rotor blades and 33 outlet guide vanes (OGVs). One of the objectives of this project is to reduce the sound pressure levels generated by the fan operation without reducing performance. The project focuses on changes made to the original rotor (such as RPM, blade length, chord length and number of blades) for reduction of sound pressure levels and their effect on the performance. The two major strategies to reduce SPL were (a) reducing tip speed, (b) Introducing IGV lean. The performance of the fan is plotted as total pressure rise across the fan (ΔP) versus mass flow rate (ṁ) since the flow is compressible. However this curve is named as the “PV curve” since it is better known that way. Figure 3 shows the fan assembly which contains the IGVs and the rotor blade. The mesh air intake can be seen near the inlet of the blower and the air flow outlet after the row of resistors which are contained inside a box. Figure 4 shows the photo of the original fan.
Figure 2: Box containing four axial fans shown in figure 3.

Figure 3: Fan assembly with rotor and IGVs
1.3 Background Fan Performance Studies

The following few paragraphs discuss the effect of various parameters on the performance and noise levels of compressors and fans as reported in existing literature.

Fukano [2] reported that sound pressure levels decrease with decrease in rotational frequency. It was also reported [3] that decreasing the number of blades would decrease the SPL. Also, decreasing the number of blades would decrease the blade passing frequency. Increasing the chord length would also increase the SPL as stated by Fukano [3]. All the above results were theoretical and they were verified by experiments.

Fahmi [4] experimentally investigated the effect of blade aspect ratio on the performance of axial flow compressors. It was found that a compressor with a lower aspect ratio produced a higher static pressure rise and had a wider range of operation. This adverse
effect of a larger aspect ratio was attributed to wall stall, i.e. wall stall affects the blade with a larger aspect ratio. Wennerstrom [5] gave an overview of the trend in the industry as it shifted from first using moderate aspect ratio blades to high aspect ratio blades and then finally of low aspect ratio blades. Sans [6] also illustrated, as in figure 5, the trend in the industry to switch to higher aspect ratio blades. Britsch [7] reported that for constant solidity, a lower aspect ratio generally resulted in a larger flow range and higher efficiency.

![Figure 5: Trend of increasing solidity in compressor design.](image)

Bell [8] reported that an increase in solidity increased the $C_{p_{\text{max}}}$ and also decreased the slope of the PV curve. Britsch [7] also reported that for constant blade aspect ratio, higher solidity resulted in lower mass flow rates due to increase in blade blockage. Also, higher maximum efficiencies were observed for low solidity rotors at design speed. Hayashibara [9] studied the effect of solidity on a compressor cascade at low Reynolds number. It was reported that an increase in solidity delayed the stall at off design operating conditions.

Sarraf [10] used two fans of different blade thickness in his study and came to the conclusion that a thicker blade resulted in a drop of pressure rise at conception flow rate and efficiency. Thicker blades also led to lower wall pressure fluctuations and velocity fluctuations.
Roux [11] used mass flow inlet boundary condition with a pressure outlet boundary condition for the simulation of an axial flow fan. Jain [12] used ANSYS-FLUENT CFD code to successfully predict the performance of an axial flow fan used to cool a radiator. This study also used a mass flow inlet and a pressure outlet boundary condition. Narejo [13] used the pressure inlet and outlet boundary conditions in ANSYS-FLUENT with a profile specified at the inlet to simulate flow through the NASA’s “Rotor 67” transonic rotor. It was observed that both mass flow inlet and pressure inlet boundary condition yielded the same result when on the right side of the operating curve.

The current rotor under consideration is casted and thus the surface finish of the rotor is not as smooth as would be desirable. The blade is shown in figure 6. Roelke [14] showed that the surface finish could have significant impact on the rotor performance, especially for small turbines. The turbine used by them had an 11.15cm tip diameter.

![Figure 6: Picture of the casted rotor blade showing the rough finish.](image)

1.4 Background for Fan Noise

Tonal noise at the blade pass frequency (BPF) and its higher harmonic is caused by the unsteady flow at the inlet of the rotor blades [15].

1.4 Background for Fan Noise

Tonal noise at the blade pass frequency (BPF) and its higher harmonic is caused by the unsteady flow at the inlet of the rotor blades [15].
Zhang [16] explored the effect of the number of blades on the performance and the noise of an axial flow fan. It was observed that increasing the number of blades first caused the total pressure and efficiency to increase and then decrease with the maximum being at 11. Also, the overall sound pressure level was greatest with 11 blades.

Metzger [17] conducted a parametric investigation of low pressure ratio fan noise. It was concluded that at the same tip speed a higher solidity reduces noise. Also, lower pressure ratio reduces noise. It was also recommended that for low pressure ratio fans (1.1 to 1.25 pressure ratio) the number of blades should be 11 to 13 and the number of stator blades should be around 7. Also for fans with even lower pressure ratios (<1.1) the fan should have less than 11 blades for lower noise.

Metzger [17] also reported that at lower pressure ratios, increasing the distance between the rotor blades and the stator produced significant noise reduction. However, at higher pressure ratios, rotor tonal and broadband noise dominates and increasing the rotor and stator gap doesn’t have a significant impact on noise reduction.

Liu [18] studied the effects of stator lean on noise reduction. It was recommended that a positive lean angle with an angle greater than 10 degrees should be used for an appreciable amount of noise reduction.
2. **COMPUTATIONAL APPROACH**

This study used air as the working fluid and was considered compressible as the local flow Mach number exceed 0.3 (maximum Mach number was 0.34). It was assumed that air follows the ideal gas law. A steady state approach was used to predict the fan performance whereas noise prediction required an unsteady approach since it requires pressure time data. The multiple reference frame model [19] was used to model the flow in a steady state approximation and the sliding mesh technique [19] was used to model the flow as being unsteady. Both the approaches mentioned above are explained in the following sections.

### 2.2 The Multiple Reference Frame Model

In this steady state approximation, the domain is split into multiple zones each of which can be assigned different rotational speeds. Moving reference frame equations are solved in each of these zones with the rotational speed being set to zero for stationary zones. An interface is used to separate the different zones where the reference frame transformation takes place. ANSYS-FLUENT [19] modifies the equations of motions in the moving cell zones by including acceleration terms which are added due to the transformation from stationary reference frame to the moving reference frame. The equations of motion for a steadily moving frame are given by Eq. (1-3):

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \vec{v} = 0
\]  \hspace{1cm} (1)

\[
\frac{\partial \rho \vec{v}}{\partial t} + \nabla \cdot (\rho \vec{v} \vec{v}) + \rho [\vec{\omega} \times (\vec{v} - \vec{v}_t)] = -\nabla p + \nabla \cdot \vec{t} + \vec{F} \hspace{1cm} (2)
\]

\[
\frac{\partial \rho E}{\partial t} + \nabla \cdot (\rho \vec{v} H + p \vec{u}_r) = \nabla \cdot (k \nabla T + \vec{\tau} \cdot \vec{v}) + S_h \hspace{1cm} (3)
\]

In this approach the mesh remains fixed i.e. it is equivalent to observing the instantaneous flow field with the motion of the moving part frozen.
2.3 The Sliding Mesh Model

In this approach, the mesh and the boundaries move, i.e. the nodes of the mesh move relative to a fixed coordinate system. Since the domain changes with time, this model can be used for transient simulations and is particularly recommended when the rotor stator interaction is strong. The equations for such a model are expressed in the general form in Eq. (4):

\[
\frac{d}{dt} \int_V \rho \phi dV + \int_V \rho \phi \left( \vec{u} - \vec{u}_g \right) \cdot d\overrightarrow{A} = \int_V \nabla \phi \cdot d\overrightarrow{A} + \int_S \phi dV
\]

(4)

Where,

- \( \rho \) is the fluid density
- \( \vec{u} \) is the flow velocity vector
- \( \vec{u}_g \) is the mesh velocity of the moving mesh
- \( \Gamma \) is the diffusion coefficient
- \( S_\phi \) is the source term of \( \phi \)

\( \phi \) is a general scalar:

The RNG k-\( \varepsilon \) model was used as it takes into account the effect of swirl on turbulence by modifying the turbulent viscosity [19]. The equations for this model are given by [19]:

\[
\frac{\partial \rho k}{\partial t} + \frac{\partial \rho \mu k}{\partial x_l} = \frac{\partial}{\partial x_j} \left( \alpha_k \mu_{\text{eff}} \frac{\partial k}{\partial x_j} \right) + G_k + \rho \varepsilon - Y_m
\]

(5)

\[
\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial \rho \varepsilon u_i}{\partial x_l} = \frac{\partial}{\partial x_j} \left( \alpha_\varepsilon \mu_{\text{eff}} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R_\varepsilon
\]

(6)

In Eq. (5) and (6), \( G_k \) is the generation of TKE due to mean velocity gradients and, \( Y_m \) is the dissipation caused by the fluctuating dilation in compressible turbulence. The RNG model takes into account the effects of swirl or rotation by modifying turbulent viscosity and making it a function of the following parameters:
\[ \mu_t = \mu_{t0} f(\alpha_s, \Omega, \frac{k}{\epsilon}) \] (7)

Here, \( \Omega \) is a characteristic swirl number, \( \alpha_s \) is a constant which has its value dependent on the intensity of the swirl and \( \mu_t \) is given by Eq. (8) in the high Reynolds number limit.

\[ \mu_t = \rho C_{\mu} \frac{k^2}{\epsilon} \] (8)

For the steady state solutions, second order upwind spatial discretization schemes were used for density, momentum, TKE, Turbulent dissipation rate and Energy. The coupled solver was used to improve the convergence speed [19]. For the transient solutions, the default solver settings were used.

The computational domain was created using Pointwise [20] and consists of a little over 5.44 million cells. The rotor zone had 1.15 million cells, the IGV zone had 1.77 million and the OGV zone had 1.6 million cells. The rest of the cells, 74000 were present in the inlet and outlet domains. Figure 7 shows a schematic of the entire domain containing an inlet and outlet duct (ensuring a smooth flow at inlet and outlet), the rotors, IGV, and OGV.

![Schematic of the entire domain with the inlet, outlet, IGV, rotor, and OGV.](image)

Figure 7: A schematic of the entire domain with the inlet, outlet, IGV, rotor, and OGV.

Although the original rotor consists of 37 IGVs, 24 rotor blades and 33 OGVs the computation simulated 36 IGVs, 24 rotor blades with 36 OGVs to maintain periodicity, i.e., so that only a slice of 30 degrees would need to be simulated. This 30 degree slice contained 3 IGVs, 2 rotor blades and 3 OGVs. Figure 8 shows the mesh of the domain.
containing the 3 IGVs, 2 rotor blades and 3 OGVs. The shroud and the periodic boundaries have been removed for clarity. Note that the domain was extended beyond the IGV inlet and OGV outlet.

A wall function approach was used in this study as the results between a simulation with enhanced wall treatment and one using wall functions only had a difference in pressure of about 1% for a given mass flow rate. A wall function mesh is also much easier to make and saves some computation time and storage (RAM) as well. Because of the complex nature of the domain it was very difficult to keep the average wall $y^+ \sim 30$ as required by standard wall functions [21]. The wall $y^+$ for the computational domain varied between 5 and 50. Thus to overcome the problem of over refinement, scalable wall functions were used.

Figure 8: Top view of the mesh of the computational mesh.
Scalable wall functions forces the usage of the log law by introducing a limiter, i.e. for $y^*<11.225$ the log law is used no matter what the mesh refinement is [19]. $y^*$ and $u^*$ are defined as:

$$y^* = \frac{(\rho u^* \Delta y)}{\mu}$$  \hspace{1cm} (9)

$$u^* = C_{\mu}^{1/4} k^{1/2}$$  \hspace{1cm} (10)

Figure 9 shows the computational grid for one rotor blade and boundary conditions. The right most boundary is a periodic boundary. The top being the inlet of the rotor and bottom being the outlet of the rotor.

Figure 10 shows the structured boundary layer mesh around the rotor. The first cell height was 0.0001m with 15 layers and growth rate equal to 1 (the consecutive cells had equal height). The mesh outside this boundary layer mesh was also clustered towards the rotor.
Figure 10: Structured boundary layer mesh surrounding the rotor blade.

Table 1 shows the boundary conditions used for various parts of the domain.

Table 1 Boundary Conditions used for the present study

<table>
<thead>
<tr>
<th>Inlet</th>
<th>Pressure Inlet (total pressure = 101325Pa, Temperature = 288.15K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet</td>
<td>Pressure Outlet (static pressure defined, ranging from 400Pa to 1400Pa gauge pressure)</td>
</tr>
<tr>
<td>Rotor blade, rotor hub</td>
<td>Rotating at zero velocity with respect to the rotor zone (rotor zone rotating at 3000RPM)</td>
</tr>
<tr>
<td>Rotor shroud</td>
<td>Stationary</td>
</tr>
<tr>
<td>All other walls (IGV blades, OGV blades, etc.)</td>
<td>Stationary</td>
</tr>
</tbody>
</table>
The PV curves for the rotor blade were obtained by increasing the outlet static pressure which reduces the mass flow rate. It was observed that at a certain back pressure (>1300 Pa gauge pressure), the mass flow rate dropped suddenly and when the velocity contours were plotted after the rotor blade, large regions of reversed flow were seen. Thus it was concluded that the rotor could not provide enough pressure head to counter the back pressure at the outlet and the rotor blades had stalled due to the high adverse pressure gradient. It was observed that near stall the solution was oscillatory and thus these solutions were not included in the PV curves. Therefore it is possible that the peak pressures reported here may not be exactly the same as those seen in experiments. This is primarily due to the unstable solutions near the stall. The next three examples (figure 11) a blade which is running without stalling, (figure 12) a blade which has reversed flow near the hub and the tip and is about to stall (increasing the outlet static pressure by about 5-10Pa will cause the rotor blade to stall in this example), and (figure 13) a stalled blade. Blue regions indicate forward flow and red/yellow regions show back flow.

![Velocity contour plot showing some separation at the hub. Positive values indicate back flow.](image-url)
Figure 12: Velocity contour plot showing regions of significant backflow. Positive values indicate back flow.

Figure 13: Velocity contour plot showing only small regions of forward flow. Positive values indicate back flow.
2.4 Grid Sensitivity Analysis

From this point onwards OGVs were not used in simulations to save time and storage (RAM). The OGVs in this case were not acting as diffusers, thus removing them does not have an impact on the stall characteristics of the fan. Also, they had a very small impact on the pressure rise (2.5% decrease in pressure rise for a given mass flow rate). Adhering to basic meshing guidelines of creating boundary layers, a mesh containing 3.6 million cells was obtained. To check grid independence the number of points in the grid were varied from 3.6 million to 5.5 million cells. (Note that the CFD simulation whose results were matched with experimental results contained OGVs since the experiments contained OGVs). A coarser grid with 1.5 million cells was also tested. Figures 14, 15 and 16 show the change in various quantities at the outlet/inlet by changing the number of cells.

It is seen that an increase in number of cells after 3.6 million results in no change in the total pressure, mass flow rate at the exit and the static pressure at the inlet, i.e. grid independence is reached.

![Figure 14: Variation of total pressure at the exit with cell count](image-url)
It was also seen that the grid with 1.5 million cells was not able to capture certain areas of the wake from the rotor which were captured by the finer grids. These regions are highlighted in figure 17 and 18. Thus, it was concluded that a 3.6 million cell grid would be used for simulations in order to capture the flow field accurately.
Figure 17: Velocity contours after the rotor for the 1.5 million cell grid. Note the highlighted area.

Figure 18: Velocity contours after the rotor for the 3.6 million cell grid. Note the difference between the highlighted area in 11a and b.
Based on the grid sensitivity study it was determined that 3.6 million cells were adequate for the simulations. However, the study of different parameters may require higher number of cells due to the change in size of the domain, for example, a longer blade would require more number of cells.

For the acoustic analysis it was observed that the time step and the number of time steps played an important part in the result. If the time step was greater than a particular value the peaks at the blade pass frequency were not captured. It was found that the required time step for these simulations was 1.25e-5 seconds. Also, it was seen that the acoustic runs had to be run for at least 0.5 of a full cycle, i.e. the time taken for one full fan revolution. Figures 19 to 21 show the change in the FFT as the flow time increases. The peaks in these figures lie at the blade pass frequency (1200Hz) and its multiples and the highest peak at the BPF is ~89dB.

Figure 19: Fan at 3000RPM; half a cycle (2.5e-5 seconds of flow time)
Figure 20: Fan at 3000RPM; 3/4th of a cycle (3.75e-5 seconds of flow time)

Figure 21: Fan at 3000RPM; one full cycle (5e-5 seconds of flow time)
It can be seen that after 0.5 revolutions the first three peaks (the highest peaks) do not change position or amplitude. The position remains at 1200Hz which is the blade pass frequency. Letting the simulation run for another 0.25 of the time required for one revolution only adds more detail to the FFT without changing the peaks quantitatively as shown in figure 22.

![Graph](image)

Figure 22: Fan at 3000RPM; one full cycle (6.25e-5 seconds of flow time)

As a side note it was observed that changing the time steps in between the calculations resulted in the shifting of the peaks as the CFD code is not able to adjust for such a change. The time steps were changed in-between the simulation to ensure convergence as the solutions sometimes had a tendency to diverge if a larger time step was used throughout the simulation. Figure 23 shows the result when the time steps were reduced while the acoustic module was active. Thus for a quick convergence, larger time steps were used in the beginning and smaller when the acoustic module was switched on.
Validation of CFD Results

2.5.1 Performance Results

Validation of CFD results was carried out at 3000RPM since experimental data were only available at this speed. The experiments were carried out by the sponsors. The uncertainty in the experimental data and the exact test procedure standard were not specified by the sponsors. At 3000RPM the CFD results were compared with the experimental PV curves. Figure 24 shows the comparison (green line showing the experimental data and orange line showing CFD results). The maximum deviation occurs at 7.85 kg/sec and is ~3.5% (this deviation is probably due to the uncertainty in the experiments). This fan has a pressure ratio of 1.023.

This gave the assurance that the present CFD simulations correctly predicting the fan performance and thus the study of the effect of variation of parameters could be started.
The CFD results were also found to obey the fan affinity laws given in equations (11) and (12). This although not a strict validation in itself however it is critical that CFD follows the laws

\[
\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \tag{11}
\]

\[
\frac{\Delta p_1}{\Delta p_2} = \left(\frac{N_1}{N_2}\right)^2 \tag{12}
\]

The fan operation was simulated at two speeds, 3000 and 2700 RPM. Using fan affinity laws, the results from the 3000RPM run were used to calculate the flow rate and pressure at 2700RPM and these results were compared with CFD results at 2700RPM. Figure 25 shows that the two results match excellently with about 0.2% difference in pressure rise at a given mass flow rate. This figure shows the PV curve obtained from CFD at 3000RPM and the PV curve from CFD at 2700. Yellow line shows the PV curve for 2700RPM obtained from the data for 3000RPM using fan affinity laws.
2.5.2 Noise Level Results

The results were validated against experimental data. Figure 26 shows that we get a very good match with less than 4% difference (the first peak at 89dB) between the experimental data and the CFD results. Also, it can be seen that the peaks lie at the BPF (1200Hz) and its multiples. The BPF can be calculated as:

\[ BPF = \frac{N \times t}{60} \]  \hspace{1cm} (13)

Where, \( N \) is the rotational velocity in RPM and \( t \) is the number of rotor blades.
The acoustic simulations were run for the original fan at two speeds (2700RPM and 3000RPM) to test if the CFD simulations predicted the decrease in sound pressure levels as predicted by the following expression [22].

\[
SPL \alpha 20 \log \left( \frac{\text{Size}2}{\text{Size}1} \right)^{3.5} + 20 \log \left( \frac{\text{Speed}2}{\text{Speed}1} \right)^{2.5}
\]  

(14)

The size of both the rotors is the same and speed 2 is 2700RPM and speed 1 is 3000RPM. Using the above formula the SPL reduction should be proportional to 2.3dB. Simulations showed about 4.5 dB reduction thus validating the runs. This was an essential part of the validation process as CFD results should approximately follow the above expression.
3. EFFECT OF DIFFERENT PARAMETERS ON PERFORMANCE

This chapter discusses the effect on 3 parameters on the performance of the fan. To save computational time, the MRF approach as discussed in chapter 2 was used. The parameters investigated were

1) Hub to tip ratio

2) Chord length and thickness

3) Chord length with constant solidity

These three parameters were chosen such that a combination of the three could be used to design a rotor with lower tip speed as discussed in section 3.4. It was believed that a lower tip speed would reduce the overall noise produced by the fan. The acoustic results of the fan is discussed in chapter 4. Figure 27 shows the hub to tip ratio (blade length) and the chord length investigated in this study.

Figure 27: Geometry of the blade showing two parameters, blade length and chord length
3.1 Hub to Tip Ratio

The hub to tip ratio was decreased by decreasing the diameter of the hub. The outer diameter was not changed due to design limitations. The extended blade had the same linear twist as the original blade. Figure 28 shows the effect of reducing the diameter of the hub, and thus lengthening the blade, on the PV curve. The orange line shows the performance of the original rotor, the blue line shows the performance of the rotor with the longer blade length (longer by 0.75 inches) and the grey line shows the performance predicted by fan affinity laws for the rotor with the longer blade length at 2800RPM. It was decided to check the RPM at which the rotor with the longer blade gave similar performance to the original rotor. Dimensional analysis was used to determine the RPM and it was found to be 2800.

The diameter for the new blade was decreased by 1.5 inches (0.0381m) as this was a constraint set by the diameter of the motor. The motor is mounted in front of the fan and if the diameter of the hub was made any smaller the motor would restrict airflow into the fan inlet. The blade curvature for the extended blade was the same as the original blade.

Figure 28: Performance comparison between original rotor and rotor with extended blade.
Lengthening the blade increased the mass flow rate by 17% at the same pressure at a given RPM (in this case, 3000). This is due to the larger frontal area, which allows for a higher mass flow rate. However it was seen that rotor blade stalled at a higher mass flow rate (new blade stalled at 9.92kg/sec whereas the original rotor stalled at 8.84kg/sec). Stalling of the rotor implies that the flow separates over the suction surface due to high adverse pressure gradient. Once the rotor stalls there is a sharp decrease in the pressure rise created by the rotor blade. This behavior can be explained using the fact that since the blade angles are the same at the tip and follow the same twist as the original blade (at the tip the blade angle remains the same) they stall at the same inlet relative velocity angle, however, due to the larger inlet area the mass flow rate increases. The peak efficiency of the fan also decreased with increase in blade length from 88% to 80%. This was probably due to increased separation seen at the hub with a longer blade.

3.2 Chord length and thickness

In order to investigate the effect of chord length and thickness on performance of the fan, two different chord lengths, 33% larger than the original length, and 25% smaller than the original length were investigated. Figure 29 shows the comparison with the orange line showing the rotor with the original chord length (c), blue line with a 33% larger chord length and grey line with a 25% smaller chord length. Note that the number of blades was 24 in each case. Also the shape of the blade was kept the same when the chord length was increased i.e. increasing the chord length was a simple scaling operation. This meant that as the chord length was increased the solidity also increased. The rotor with a 33% larger chord had a solidity of 1.064 and the rotor with a 25% smaller chord had a solidity of 0.6. The original rotor had a solidity of 0.8. The blade length was also the same in all the cases (0.093726m).

It was observed that as the blade chord length is increased the slope of the performance curve increases and the rotor also stalled earlier (stall occurs around 8.87 kg/sec in the rotor that has a 33% longer chord compared to 8.44 kg/sec in the rotor having the original chord length). The early stalling can be explained by the fact that an increase in solidity causes the flow to accelerate faster and thus creates a low pressure region and a longer/thicker blade the flow has to overcome a larger adverse pressure difference which
causes the flow to separate earlier (the flow separates for a lower back flow pressure as compared to the case of a shorter blade). The steeper slope of the PV curve could be because of the fact that for thicker airfoils the flow area through the rotor reduces i.e. the solidity increases and this causes more flow deflection which means that more work is done (Euler Equation) and thus a small change in velocity changes the pressure significantly. The rotor with 1.33 times the chord length had a peak efficiency of 74% and the one with 0.75 times the chord length had a peak efficiency of 77%.

Figure 29: Comparison between rotors having 3 different chord lengths.

### 3.3 Chord length with constant solidity and blade surface area

The effect of changing chord length while keeping the total blade surface area and solidity constant was investigated with the help of two different cases and the results are shown in Figure 30. The two different cases are (i) a scaled up rotor with a blade having 33% longer chord length and 18 blades (yellow line in Fig. 30), (ii) a scaled down rotor with the blade having 33% smaller chord length with 36 rotor blades (blue line in Fig. 30). All the blades had a solidity of 0.80. It was seen that by keeping the same surface area and
decreasing the number of blades and increasing the chord length caused the rotor to stall at a lower mass flow rate but it had ~5% less rise in pressure compared to the original rotor. The same trend continued with the scaled down rotor blade, that produced ~4% more pressure rise at the same mass flow rate but stalled at higher mass flow rates. The blade having 33% longer chord with 18 blades has a peak efficiency of 78% and the blade with 33% smaller chord has a peak efficiency of 75%.

The early stalling of rotors with larger number (36 blades) of shorter chord length (0.67 times the original chord length) blades could be because the velocity inside the blade passage area increases significantly, and thus creates large number of adverse pressure gradient regions on the suction surface of the airfoils causing the airfoils to stall once the back pressure is increased to a particular value (>700Pa in this case)

3.4 Design for a Lower Tip Speed.

Based on the above results it a design with a lower tip speed was designed. The sponsors set a limit of 1.5 inch reduction on the hub diameter. A new set of IGVs and Rotor blades
were designed with a smaller hub diameter. Both the IGV and the rotor blades had the same twist was before i.e. the twist in degrees per unit radius was the same. But as reported before, an increase in the hub to tip ratio would not allow the blade to operate at lower mass flow rates. To enable the rotor to operate at lower mass flow rates the chord length of the blade was increased (by 50%) and the number of blades were decreased (to 18 from 24) to maintain the same solidity (0.8), NOTE: this is explained in section 3.

It was seen that increasing the chord length and decreasing the number of blades was causing separation of flow at the root as shown in figure 31.

![Backflow region](image)

Figure 31: Positive values of Z velocity indicate reverse flow which can be seen at the hub.

To reduce the separation of flow at the hub the design was modified from what is show in figure 32 to 33. This design change helped reduce separation by reducing the adverse pressure gradient at the hub. A downward decline acts as a diffuser and thus increase the static pressure and therefore creates an adverse pressure gradient. Hence an upward incline was added to make sure that the hub did not create an adverse pressure gradient.
Figure 32: Original hub design.

Figure 33: Redesigned hub with an upward incline. This design greatly reduced separation as shown in figure 34.

Figure 34: The redesigned hub reduced the separation at the hub

Figure 35 shows the comparison between the new design and the original design. It can be seen that the pressure for the same mass flow rate is higher for the new rotor and that
the new rotor can operate at mass flow rates similar to that of the original rotor. Also note that the new design operates at 2700RPM compared to the original 3000RPM.

![Graph showing comparison of pressure head (ΔP) and mass flow rate (ṁ) for different rotor designs.](image)

Figure 35: Comparison of performance of original rotor (24 blades) with the new rotor (18 blades)

It was also observed that the new rotor had a higher efficiency at approximately the same mass flow rate as shown in table 2.

Table 2: Pressure head, flow rate, horse power and efficiency comparison of the two rotors.

<table>
<thead>
<tr>
<th></th>
<th>Pressure head</th>
<th>Flow rate</th>
<th>HP</th>
<th>efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>New rotor (2700RPM)</td>
<td>9.09</td>
<td>13612</td>
<td>25</td>
<td>77</td>
</tr>
<tr>
<td>Current rotor (2700RPM)</td>
<td>7.60</td>
<td>12189</td>
<td>19</td>
<td>73</td>
</tr>
<tr>
<td>Current rotor (3000RPM)</td>
<td>9.38</td>
<td>13544</td>
<td>27</td>
<td>73</td>
</tr>
</tbody>
</table>
3.5 Summary of Important Results of Performance Studies

The effect of three different parameters (hub to tip ratio, chord length and solidity) was examined on fan performance. It was seen that increasing the blade length increased the mass flow rate for the same pressure rise by about 17% due to the increase in the frontal area. The new rotor (with an increased blade length) gave similar performance to the original rotor at a lower angular velocity (2800 RPM). Changing the chord length while keeping the number of blades resulted in changing the slope of the PV curve and changed the stall point. The PV curve became steep due to the reduction in rotor passage area in the case of longer chord length blades. The reduction of passage area caused the velocity to change with very little change in pressure. The final parameter change was made by increasing/decreasing the chord length but keeping the same surface area by decreasing/increasing the number of blades. This study revealed that a blade with 1.33 times the chord and less number of blades (18 compared to the previous 24) allowed the fan to operate at lower mass flow rates. The hub of the original rotor was also modified to reduce the separation at the hub. Based on the study of the effect of the three parameters a new rotor was designed with a higher efficiency for a given mass flow rate (the efficiency increased by 4%).
4. EFFECT OF DIFFERENT PARAMETERS ON NOISE

The two primary noise components would be tonal noise and broadband noise. Tonal noise is caused by the ingestion of periodic unsteady flow by the rotor due to the flow wakes of the inlet guide vanes. Broadband noise is caused by the ingestion of turbulence [15]. In this study we focus on the tonal noise production of various designs since that is the primary source of noise in this fan.

The acoustic performance of different rotors was tested by using the acoustic module in CFD code. The aerodynamic noise produced was tested at the peak efficiency for each design. The RPM was kept constant so as to keep the tip speed constant. Figure 36 shows FFT of noise signal for the original fan.

![Figure 36: FFT for original rotor](image)

As can be seen from figure 36, the peaks lie at the blade pass frequency (1200Hz) and its multiples. The highest peak is at the BPF with an amplitude of ~89dB.
4.1 Chord length and Thickness

The effect of chord length was investigated on this fan by testing the two designs previously mentioned. One with a 33% larger chord and one with a 25% smaller chord. Figure 37 and 38 show the effect of these parameters on noise.

![Figure 37: FFT for a chord length of 0.75c with the highest peak at 110dB](image)

It was seen that increasing the chord length did not change the highest peak which remained at ~90dB. However, decreasing the chord length increased the peak to 110dB. This is further illustrated by figure 39. Note that changing the chord length changed the solidity. The solidity of the original rotor was 0.8. The rotor with a 33% larger chord had a solidity of 1.064 and the rotor with 25% smaller chord had a solidity of 0.6.

According to curl’s equation, the pressure is a direct function of the rate of change of axial force. The equation is given by[23],

$$p(x, t) \approx \frac{\cos \theta}{4\pi xc} \left[ \frac{\partial F}{\partial t} \right]$$

(15)
Figure 38: FFT for a chord length of 1.33c with highest peak at ~90dB

Where, \( c \) is the ambient speed of sound, \( F \) is the unsteady rotor force, \( x \) is the distance between the fan and point of pressure measurement and \( \theta \) is the angle between fan axis and point of pressure measurement. From figure 39 it can be inferred that a rotor with a shorter chord length and thus lower solidity produces a higher axial force fluctuation and since all fans are rotating at the same speed, we can say that the lower solidity fan produced a higher drag force thus producing more noise. With a higher solidity (1.064) the rate of change of axial force is the same as that for a rotor with 0.8 solidity and thus it produces peaks with the same amplitude.

Also it should be noted that the all the peaks remain at the same frequency as that of the original fan, i.e. at the blade pass frequency (BPF) and its multiples since the RPM and the number of blades remain constant.
4.2 Chord length with Constant Solidity and Blade Surface Area

The effect of changing the chord length while keeping the same solidity was studied using two different cases as explained in chapter 3. The two different cases were, a scaled up rotor with a blade having 33% longer chord length and 18 blades and a scaled down rotor with the blade having 33% smaller chord length with 36 rotor blades. All the blades had a solidity of 0.80. Figures 40 and 41 show the effect of these parameters on noise.
It was seen that while keeping the same solidity, increasing the chord length while decreasing the number of blades the peak increased (111 dB). The same was observed when the chord length was decreased and the number of blades were increased (94 dB). Figure 42 illustrates this trend.

Figure 42 suggests that for a rotor solidity of 0.8, a chord length of 2.17 inches, with 24 blades is the configuration that produces the least amount of noise. Both the other configurations with 18 and 36 blades produced more noise (highest peak increased by 1% and 19% respectively). From Figures 41 and 42 also show a change in blade pass frequency which is expected due to the change in number of blades. The configuration with 36 blades has a BPF of 1800Hz and that with 18 blades has a BPF of 900Hz. The original fan had a BPF of 1200Hz.

Figure 41 also suggests that for less number of blades (18 blades) with the same solidity as other rotors (0.8), the highest peak shifts from the BPF to the first harmonic of the BPF (2*BPF = 1600Hz). This suggests that such a rotor produces pressure fluctuations with the highest amplitude at a higher frequency. It can also be inferred from figure 42 that the
rotors with 18 and 36 blades produce more axial force pressure fluctuations and thus more drag and more noise.

Figure 42: Peak dB levels for three different chord lengths and number of blades with the same solidity of 0.8.

4.3 Acoustic results for the design with a lower tip speed

This section discusses the acoustic signature of the fan design discussed at the end of chapter 3. It was seen that even at a lower RPM (2700 compared to the original of 3000) the new rotor had a higher peak at 106dB (14% higher).

The BPF for this rotor was 810Hz and figure 43 shows that all the peaks lie at and on the multiples of the BPF. The peaks were higher than the original rotor due to two reasons. First being that the rate of change of axial force for the rotor with 18 blades was higher, i.e. the rotor produced more drag and more noise. Second was that when the rotor was scaled the distance between the inlet guide vanes was reduced from 0.8c to 0.6c. As the distance between the obstruction in front of the rotor and the rotor is decreased the noise produced by the rotor increases [24].
4.4 Effect of IGV lean on sound pressure levels

This chapter discusses the effect of IGV lean and the distance between the IGV and rotor on the noise produced. As mentioned earlier, an increase in axial spacing between the inlet guide vane and rotor blades reduces interaction noise [24]. Introducing a sweep increases the axial distance between the tips thus reducing noise. Similarly introducing a lean increases the circumferential distance between the tips and thus reduces noise [25]. Two different lean angles were experimented with, 6 degree of positive lean and 15 degree of positive lean. The definitions of lean and sweep angles are shown in figure 44. Sweep wasn’t used because of the space constraints imposed by the sponsors.

Figure 43: FFT of noise produced by the new rotor design with reduced tip speed.
It was seen that increasing the lean angles reduced the peaks at the 1st and the 2nd harmonic of the blade pass frequency but it was observed that it did not have an effect on the first peak at the BPF. This is shown in figures 45 and 46.

From figure 45 it can be seen that for a 6 degree IGV lean, the second peak at 2400Hz has reduced from 85dB to ~72dB (15% reduction). However, the peak at the BPF at 1200Hz went up by ~1dB. The peak at 3*BPF also increases from 87dB to 90dB. Thus the comparison suggests that there is a redistribution of energy from higher to lower frequencies (2400Hz to 1200Hz) and lower to higher frequencies (2400Hz to 3600Hz).

Figure 46 shows that for a 15 degree IGV lean, there is a reduction in the second peak from 85dB to 83dB and a reduction in the third peak from 87dB to 72dB. However, the

Figure 44: Definitions of (a) sweep angle $\alpha$ (b) lean angle $\beta$ [26]
peak at the BPF at 1200Hz again went up by ~1dB. Thus there is a redistribution of energy from higher to lower frequencies.

Figure 45: IGV with 6 degree lean with rotor at 3000RPM

Figure 46: IGV with 15 degree lean with rotor at 3000RPM
4.5 Summary of Important Results of Noise Studies

It was observed that decreasing the chord length while keeping the number of blades the same increased the peak SPL by 20dB whereas increasing the chord length while keeping the same number of blades kept the peak SPL approximately the same. Increasing or decreasing the chord length while keeping the same solidity (by changing the number of blades from 24 to 18 and 36) increased the peak SPL by 4dB and 21dB respectively. The increase in SPL peak in each case was due to an increase in rate of change of axial force, i.e. an increase in drag produced by the rotor and thus an increase in noise. Although the fan which was designed to operate at a lower tip speed was more efficient at a given mass flow rate it was found that it increased the peak SPL by 13dB. Other than in increase in rate of change in axial force, the increase in peak SPL was also attributed to a decrease in axial spacing between the IGV and the rotor.

The effect of IGV lean was explored on the acoustic signature of the fan. A lean increases the circumferential distance between the tips of the inlet guide vanes and the rotor which reduces noise. Introducing a lean on the IGV decreased the peaks at the harmonics of the blade pass frequency. A six degree lean eliminated the peak at the first harmonic of the blade pass frequency (2*BPF) but increased the peak at the BPF by 1dB and at 3*BPF by 3dB. A fifteen degree lean reduced the peak at 2*BPF by 2% and the peak at 3*BPF by 17%. However, a 15 degree lean again increased the peak at the BPF by ~1dB.
5. CONCLUSIONS AND FUTURE WORK

The effect of three parameters (hub to tip ratio, chord length and solidity) was examined on the performance and noise of a fan. A fan which could operate at lower RPM and had a new hub was designed. The effect of introducing a lean on the inlet guide vanes was explored. The important results are summarized below:

5.1 Performance

It was observed that increasing the blade length increased the mass flow rate by 17% for the same pressure rise. The blade length was increased by 0.75 inches Changing the chord length changed the slope of the PV curve and the longer chord rotor stalled at higher mass flow rates (8.87kg/sec compared to original 8.44kg/sec). The shorter chord rotor had a peak efficiency of 77% and the longer chord length had a peak efficiency of 77%. The longer chord was 1.33 times the original chord length and the shorter chord was 0.75 times the original chord length. Changing the chord length while keeping the same solidity of 0.8 changed the stall point. A shorter chord stalled at higher mass flow rate (9.22kg/sec) and the longer chord stalled at lower mass flow rates (8.25kg/sec) compared to the original stall point at 8.44kg/sec. A new rotor was designed with 18 blades and a longer blade with an operational RPM of 2700. The hub of this rotor was redesigned to reduce separation at the hub. This rotor had a higher efficiency (77% compared to 73%) for the same mass flow rates.

5.2 Noise

Increasing the chord length by 33% kept the peak SPL at 90dB whereas decreasing the chord by 35% increased the peak SPL by 20dB. The blade pass frequency remained the same (1200Hz) because the number of blades and the RPM of the fan was the same. From curl’s equation it was inferred that the increase in dB was due to a higher rate of change of axial force. Changing the chord length and the number of blades while keeping the solidity the same increased the peak SPL. A blade with a 25% shorter chord and 36 blades increased the peak SPL by 19% and a 33% longer blade with 18 blades increased the SPL by 1%. Due to the different number of blades for each of the rotors, the BPF are different for each rotor. The rotor designed to operate at a lower RPM produced a 14% higher peak SPL. The effect of introducing a lean in the IGV was explored. A lean increases the
circumferential distance between the tip of the blades and thus reduces noise. It was seen that a 6 degree lean reduced the peak at 2\*BPF by 15% but increased the peak at the BPF by 1dB and at 3\*BPF by 3dB. A 15 degree lean decreased the peak at 2\*BPF by 2dB and the peak at 3\*BPF by 15dB. However, it too increased the peak at the BPF by 1dB.

5.3 Future Work

It was observed that without the IGV the performance of the rotor dropped because of the misalignment of the velocity vector at the inlet of the rotor. However it was also seen that this configuration decreased the peak at the BPF by ~15dB. Noise is a strong function of the pressure ratio \[17\], thus for a quieter rotor, a reduced pressure ratio requirement with a fan designed to operate without inlet guide vanes would be ideal. As suggested by literature \[17\] we could also increase the axial spacing between the inlet guide vanes and the rotor to reduce the noise.
REFERENCES


