Prediction of Aerodynamic Performance of Axial Fan with Leading Edge Serration using Computation Fluid Dynamics

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Abstract:
Axial fans are found to be one of the major sources of the noise in modern devices such as PCs, HVAC systems, turbo engines and etc. Lot of researches were going on for decades to reduce this unwanted noise. Researchers have found that application of leading-edge serrations will lead to reduction of noise from the bio-inspired study on the wings of barn-owl. This project involves in the study of effect of Aerodynamic performance of axial fan in terms of pressure drop and aerodynamic efficiency using commercial CFD tool ANSYS FLUENT. Analysis have been carried out for a base model without serration and same model with serrations for various flow rates. The results are validated with the experimental results obtained by Krömer et al [1] and found to be in reasonably good agreement.

Keywords: Axial Fan, Aerodynamics, CFD, and Leading-edge serrations.

1. Introduction:
In 1997, Beiler et al. [2] have proposed that use of blade skew effects the aerodynamic performance of the axial fan but also resulting in the improved aeroacoustics performance. Further in 1999 he concluded that forward skewed blade has good noise reduction capability compared to the other types. It was found that blade skew influences the 3-D flow field in axial fans. Ana F. Tinetti and Jeffrey J. Kelly [3] proposed that using of passive porosity technology in stators will reduce the noise produced due to wake stator interaction. Vad et al. [4] proposed an idea of using soft coat to reduce the noise. Their study concluded that usage of soft velvet coat in leading edge reduces the inlet turbulence effect. Further they also state that coating increases friction which causes adverse aerodynamic effects. Yang et al. [5] have made optimization study on axial fans with forward skewed blades. They compared their optimized design with archetype radial impeller and found that at 6.1 degree forward skew angle the total pressure efficiency is increased by 1.27%. Luo et al. [6] performed a computational study on axial fan performance with microplates. As a conclusion they found that use of microplate was not able to improve the total fan efficiency but whereas it increases the stall characteristics of blade as the microplates diminishes the radial migration of boundary layer flow on blade surface. Wang et al. [7] have made a bio-inspired study on axial fans with leading edge serrations, trailing edge fringes and soft downy coating. This idea was inspired from owl. They conclude that the bio-inspired noise control techniques have the potential to achieve low-noise radiation. Although there are several challenging issues that have to be considered and solved to satisfy the requirements of more stringent airworthiness. Czwielong et
al. [8] made an experimental study on effect of using leading edge serration in different spanwise location. Further they found that the fan design which is having leading edge serration throughout the span has the least pressure drop. They also concluded that serration reduces the aerodynamic efficiency compared to normal model. Krömer et al. [1] made a combined study of axial fan blades with blade skew and also leading-edge serrations. Their results showed that application of serration leads to reduction of greater noise. From the literature survey we understand that applying serration have a greater impact on the performance of the fan. It is also clear that most of the studies have been performed experimentally. So, we decided to perform computational study of axial fan with rotating motion using steady state multiple reference frame model with non-conformal meshing for rotor stator interaction. We choose the fan model designed by Krömer et al. [1] for validation and to study the effect of serration we applied sinusoidal profile with amplitude of 13% of mean aerodynamic chord and with frequency of 2mm⁻¹.

2. Model and Domain:

The process of simulation can be categorized as stages accordingly

1. Mathematic modelling
2. Meshing
3. Setting up Solver and Physics
4. Result monitoring
5. Post Processing

2.1 Modelling of Unskewed Fan:

The first process of the project is to model the fan. The model was designed with the help of CAD package, Solidworks and shown in Figure 1. In order to validate the results, the base fan model was chosen from the experimental work done by Krömer et al [1]. The blade profile was made of NACA 4510. The diameter of the hub, dhub = 247.5mm and the duct diameter d duct = 500mm. The diameter of fan d fan = 497mm. The fan has 9 blades and clearance from the tip to duct is 1.5mm with constant stagger angle of 20 degrees. The fan blade zero sweep angle and skew angle.
2.2. Modelling of Unskewed Fan with Leading edge serration:

The same unskewed fan was applied with leading edge serration and shown in Figure 3. Sinusoidal pattern serration as shown in Figure 2 was applied throughout the leading edge. The sinusoidal equation applied is given below.

$$y = a_{LE} \sin(y_{LE}x)$$ (1)

Where

$a_{LE} = 13.3\%$ of mean aerodynamic chord which is equal to 69.6mm.

$y_{LE} = 2\text{mm}^{-1}$. 

Figure 1: CAD Model of Unskewed Fan

Figure 2: Leading Edge parameter
2.3. Flow domain:
For analysis of external flow over the Axial fan a flow domain similar to the Krömer et al [1] experimental setup has been modelled as shown in Figure 4.
2.4. Meshing:

A good mesh can lead to a proper solution. Fully tetrahedral mesh has been modelled. The elements at the rotating domain is controlled to be less than 4mm as shown in Figure 6 and the remaining outer region is set to global sizing of 1000mm as shown in Figure 7. By doing this it allows us to get a more refined solution at the region of interest. The average skewness of the mesh was about 0.223. Also, in order to reduce the discretization errors grid independency test have been performed to maximum extend of machine and shown in Table 1.

Figure 5: Rotating Domain

Figure 6: Closer view of mesh at rotating domain
3. Numerical Analysis:

3.1. Inflow Conditions:
As a first step of solving the analysis have been carried out to find the inflow characteristics. During this process simulation have been carried out on fans without blades and the hub is set to be stationary with an inlet flow rate of 1.4 m\(^3\)/s.

The following physics have been enabled in solver.

- Steady state
- Incompressible flow
- \( k- \varepsilon \) turbulence model with standard wall function
- Fluid – Air at \( \rho = 1.225 \text{ kg/m}^3 \)
- Inlet – Defined with mass flow rate
- Outlet – Defined with Atmospheric pressure condition at Sea level

Pressure-Velocity coupling is done using coupled algorithm with second order discretization solution method is used with convergence criteria of \( 1 \times 10^{-3} \).

The solver solves the flow governing equations including continuity, momentum, turbulent kinetic energy and dissipation through the finite volume method as given below:

Continuity Equation

\[ \nabla \cdot (\rho \vec{v}) = 0 \]  \hspace{1cm} (2)

Momentum Equation

\[ \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla P + \nabla \cdot (\vec{T}) \]  \hspace{1cm} (3)
Turbulent kinetic Energy Equation

\[ \frac{\partial (\rho k)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_i} \right) + \frac{G_k}{\sigma_k} + \frac{G_b}{\sigma_b} - \rho \varepsilon - Y_M \]  

Dissipation Equation

\[ \frac{\partial (\rho \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \]  

3.2. Fan Analysis:

After analyzing the inflow conditions, the blades are added and same mesh setting is used with addition of inflation for blades. Now the rotating domain have been set as moving reference frame (MRF). First the blades without serrations are analyzed then followed by analysis of blade with serration.

Along with the physical setups used in previous case, the duct and fan are set to rotational moving reference frame. The stationary frame can be converted to moving frame by using the velocity as follow

\[ \vec{v}_r = \vec{v} - \vec{u}_r \]  
\[ \vec{u}_r = \vec{u}_t + \vec{\omega} \times \vec{r} \]  

Hence at the moving reference frame Eqns. 2 and 3 become

\[ \nabla \cdot \vec{p}_{\vec{v}_r} = 0 \]  
\[ \nabla \cdot (\rho \vec{v}_r \vec{v}_r) + \rho (2\vec{\omega} \times \vec{v}_r + \vec{\omega} \times \vec{\omega} \times \vec{r} + \vec{\omega} \times \vec{r} + \vec{a}) = -\nabla P + \nabla \cdot \vec{t}_r \]  

The equations have been solved for different flow rates ranging from 0.2 m³/s to 1.2 m³/s with an increment of 0.2 m³/s each time. The pressure drop has been measured by calculating the difference between suction side total pressure and exhaust side static pressure. The pressure at exit is measured by taking area weighted average of outlet plane of duct.

3.3. Measuring technique and Validation:

Along the radial direction at measuring point 85 mm from the inlet of duct the velocity c₁ along the x₁ direction have been measured at different points. In order to compare and validate the results the velocity and location parameters have been converted into non dimensional parameters known as normalized c₁ = c₁/cref, where cref is global mean velocity of all measured point and r/r_duct.
From Figure 7, it is clear that the variation of velocity is very small in most of the regions except the region near hub as the flow gets disturbed by hub. Also, from the obtained we can also find that the percentage of deviation between simulation and experiment is 12.5%.

Figure 8 shows the changes in pressure drop with respect to the flow rate for the baseline fan without serration. It is found that as expected, the pressure drop decreases monotonically as the flow rate increases. The numerical results of the same have reasonably good agreement with those of experiments.
4. Results and Discussion:

After the analysis of different fan designs separately the final results are compared and a quantitative study is performed.

4.1. Pressure Drop Characteristics:

Pressure drop obtained for various designs are combined together and shown in Figure 9. From Figure 9 we can observe that for flow rate up to 0.7 m$^3$/s the pressure rise of serrated fan is found to lower than the normal blades. At higher flow rates it is also clear that the pressure drop is almost same for both the designs. From this obtained result we can also say serrated fan have good aerodynamic properties along with the acoustic properties.

![Figure 9: Pressure Drop Characteristics for various designs](image)

4.2. Efficiency:

Based on the pressure drop using Eq.10, the efficiency of the fan is calculated and shown in Figure 10. From Figure 10, it is observed that for the baseline fan without serration, the efficiency increases with flow rate up to 0.6 m$^3$/s, then it decreases continuously whereas for the serrated fan, this drop in efficiency occurs at the flow rate of 0.8 m$^3$/s. However, at the lower flow rates the efficiency of the baseline case is higher than that of the serrated fan whereas at the higher flow rates the efficiency of the baseline is lower as compared to that of the serrated fan.

$$\eta = \frac{\dot{V} \Delta P}{M \cdot 2\pi n}$$  \hspace{1cm} (10)

where, $\dot{V}$ is the flow rate, $\Delta P$ is the pressure drop, $M$ is the torque and $n$ is the speed in rpm.
5. Conclusion:
Aerodynamic performance of an axial fan with and without serration has been studied numerically for various flow rates. Based on the obtained results, it is observed clearly that application of serration gives a greater noise reduction also along with the improvement of aerodynamic characteristics at higher flow rates. It can also be replaced with normal conventional blades as the pressure drop is almost close at higher flow rates and even lesser at lower flow rates. Further study can perform to predict aeroacoustics performance of the same with various serration parameters.

Nomenclatures

| Symbols | Name |
|---------|------|
| $d_{hub}$ | Diameter of fan hub (mm) |
| $d_{duct}$ | Diameter of duct (mm) |
| $d_{fan}$ | Diameter of fan (mm) |
| $\alpha$ | Angle of attack in ° |
| $l_c$ | Chord length of blade (mm) |
| $a_{LE}$ | Amplitude of leading-edge serration (mm) |
| $f_{SE}$ | Frequency of serration (mm⁻¹) |
| $\lambda_{SE}$ | Wavelength of serration (mm) |
| $d_{sec}$ | Section diameter of blade (mm) |
| $c_1$ | Velocity in $x_1$ direction (m/s) |
| $x_1$ | Global z direction |
| $x_2$ | Global x direction |
| $x_3$ | Global y direction |
| $r$ | Radius |
| $k$ | Turbulence kinetic energy |
| $\varepsilon$ | Rate of dissipation |
| $\rho$ | Density of fluid (kg/m³) |
| $P$ | Pressure (Pa) |
| $\rightarrow v$ | Velocity vector (m/s) |
\( \tau \)  
Shear tensor (N/m^2)  

\( \mu \)  
Eddy Viscosity  

\( \sigma \)  
Turbulent Prandtl number  

\( G_b \)  
Generation of turbulent kinetic energy due to velocity gradient  

\( G_b \)  
Generation of turbulent kinetic energy due to buoyancy  

\( \gamma_m \)  
Fluctuating dilation  

\( \dot{v}_r \)  
Radial Velocity (m/s)  

\( \dot{\omega} \)  
Angular velocity (RPM)  

\( \ddot{a} \)  
Angular acceleration (rad/s^2)  

\( \dddot{a} \)  
Acceleration  

\( c_{ref} \)  
Reference velocity Global mean (m/s)  

\( \eta \)  
Efficiency  

\( \dot{V} \)  
Flow rate (m^3/s)  

\( \Delta P \)  
Pressure Drop (Pa)  

\( M \)  
Torque  

\( n \)  
RPM  

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