OPTIMIZATION AND ANALYSIS OF INDUSTRIAL FLOW FAN BY USING CFD

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ABSTRACT

A newly developed computational approach different from the conventional methods which are popularly adopted by the fan designers is employed to study the flow field and performance characteristics of an Industrial flow fan by using the commercial code (Fluent) of computational fluid dynamics. A Fluid Structural Interaction (FSI) methodology is developed to predict the flow performance of an industrial flow fan and to quantify the source strength and sound pressure levels at any location in the system. The acoustic and flow performances of the fan are predicted simultaneously using a computational acoustic technique combining transient flow analysis and noise propagation. The calculated sound pressure levels compare favorably with the measured sound pressure data per ISO 3744 Rectangular parallelepiped

Keyword: *Mancooler ,Modal Analysis, harmonic analysis , transient analysis, Optimization*.

1. INTRODUCTION

Due to stringent government regulations and competitive pressures, meeting noise goals are as important design criteria as meeting the flow rate and pressure head requirements. Although experimental sound measurement is quite straightforward, analyzing the origin of noise remains elusive. The challenge is to predict the flow induced noise for major noise contributors of agricultural and construction machinery such as cooling fans, exhaust muffler and air handling system. There is several commercially available software in the market that addresses the noise generated due to fluid structure interaction. However, the choice is very limited for predicting flow induced noise and subsequent fluid structure interaction.

In this paper, CFD is developed to predict the compressible flow performance of industrial flow fan and to quantify the source strength and sound pressure levels at any location in the system. The acoustic and flow performances of the subject fan are predicted simultaneously using technique combining transient flow analysis and noise propagation. Corresponding flow induced noise solution is directly available at the boundary of the domain. The sound pressure level of the subject fan is then calculated by integrating over the rectangular (1m radius).

1.1 Industrial Flow Fan

An industrial fan type “ALMONARD E300S”of 3 straight blades which is used in many industries generally.This type of industrial fan is called “Industrial Mancooler”. ALMONARD has included Mancooler to facilitate the shop floor in the industries to provide a better environment for their employees which in turn results by the increases in production. ALMONARD Mancoolers are designed for large Air Volume discharge with high velocity and for long distance sweep. ALMONARD Mancoolers can be used in various applications such as cooling the electrical equipment for general comfort and proper ventilation in the shop floors.

1.2 Acoustic Analysis

Acoustic deals with the study of generation, propagation, absorption and refelection of sound pressure wave in the fluid medium.Pressure distribution in the fluid at different frequenices, pressure gradient, particle velocity and sound pressure level can be analysed.Coupled acoustic analysis takes the fluid structure interaction into account. In acoustic Fluid Structural Interaction (FSI) problems, the structural dynamics equation must be considered along with the Navier-Stokes equations of fluid momentum and the flow continuity equation.

The acoustic wave equation is given by:

$$\nabla\cdot\left(\frac{1}{\rho_0}\nabla p\right)-\frac{1}{\rho_0 c^2}\frac{\partial^2 p}{\partial t^2}+\nabla\cdot\left[\frac{4\mu}{3\rho_0}\nabla\left(\frac{1}{\rho_0 c^2}\frac{\partial p}{\partial t}\right)\right]=-\frac{\partial}{\partial t}\left(\frac{Q}{\rho_0}\right)+\nabla\cdot\left[\frac{4\mu}{3\rho_0}\nabla\left(\frac{Q}{\rho_0}\right)\right]$$

2. POST PROCESSING IN CFD

The ANSYS CFD post processor is flexible, scriptable and customizable. It can post process from a number of CFD results formats and it supports structured, unstructured and hybrid grids.

2.1 Modal Analysis

The mode shapes and natural frequencies of a structure are its basic dynamic properties. These derived matrices are based on the mass, stiffness and damping properties which involves in actual boundary conditions. Different mode shapes are found to get the fundamental frequencies of industrial fan. Modal Analysis is done for 6 mode shapes. The figure shown below is deformed model for an industrial fan at which resonant frequency is noted. The red colour represents the high sound pressure level and the blue area is the low sound pressure level.

- Mode Shape 1 Mode Shape 2

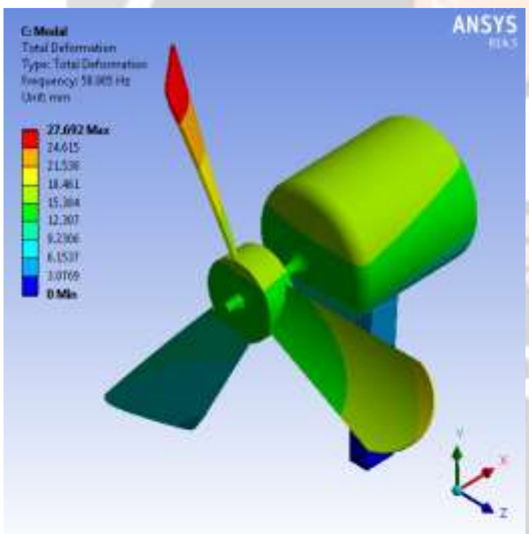

Figure 1 Mode Shape 1

Natural frequency - 50.905 Hz

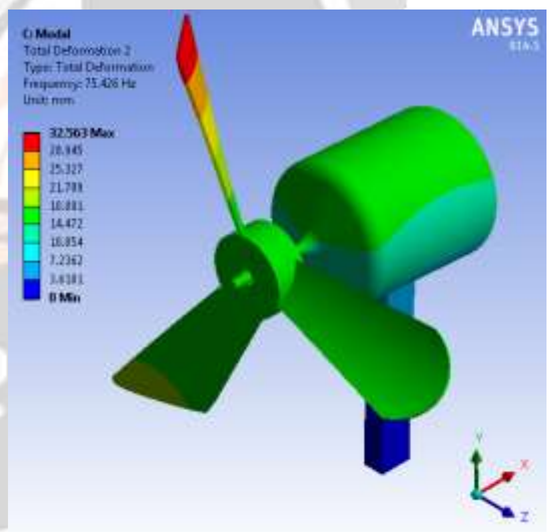

Figure 2 Mode Shape 2

Natural frequency: - 75.423 Hz

- Mode Shape 3

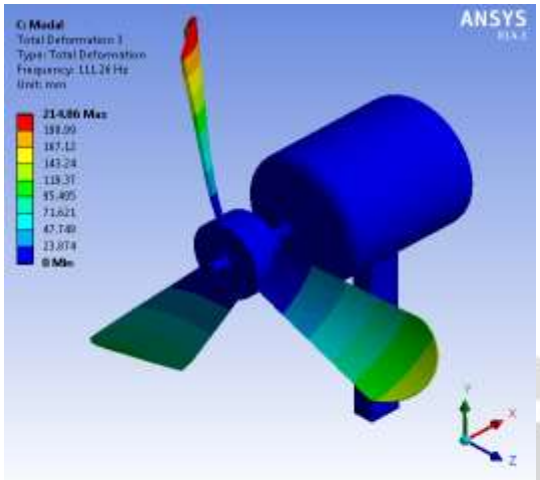

Figure 3 Mode Shape 3

Natural frequency: - 111.26 Hz

Mode Shape 4

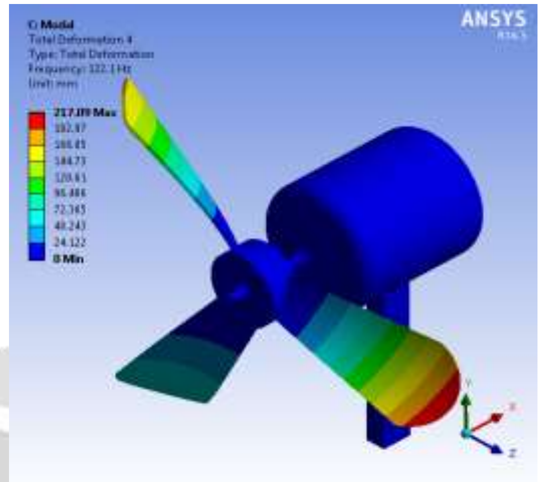

Figure 4 Mode Shape 4

Natural frequency:- 122.1 Hz

- Mode Shape 5

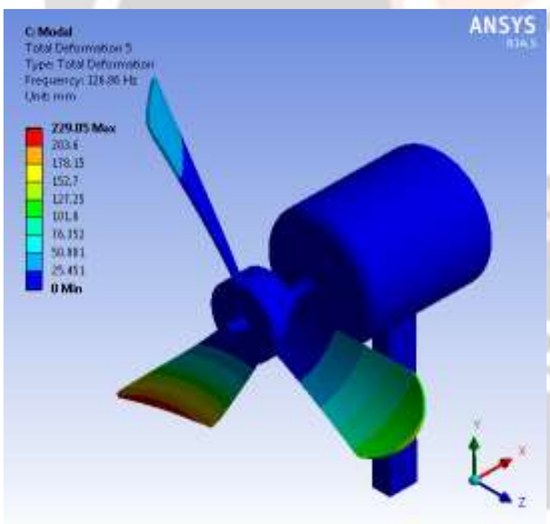

Figure 5 Mode Shape 5

Natural Frequency:- 126.86 Hz

Mode Shape 6

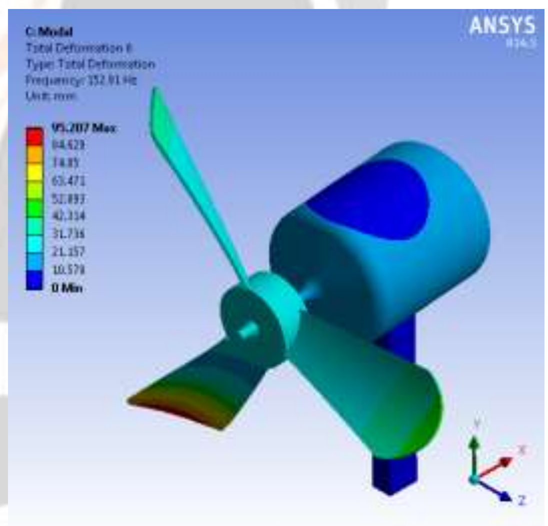

Figure 6 Mode Shape 6

Natural frequency:- 152.91 Hz

The table 1 shown below highlighted natural frequency for first six modes.

Table 1 Mode Shape of Industrial fan

Mode shape	Natural frequency (Hz)	SPL (dB)
1	50.905	27.692
2	75.423	32.563
3	111.26	214.86
4	122.1	217.09
5	126.86	229.05
6	152.91	95.207

From above results it can be observed that frequency of each industrial fan is very high and finding out the frequency (126 Hz) at which the sound pressure value is maximum as compare with experimental results.The harmonic and transient analysis can be obtained using this modal analysis.

2.2 Harmonic Analysis

The harmonic analysis solves the time-dependent equations of motion for linear structures undergoing steady-state vibration. It used to determine the response of a linear structure to loads those vary harmonically with respect to time. For this type of analysis, the entire structure should have constant or frequency-dependent stiffness, damping and mass effects. The structure responses at several frequencies are plotted in figure 7– 9the frequency which was in form of change in rotational velocity was varied with vibration amplitude (mm/s2). The frequency response of industrial fan was captured on all three axes (Z, Y and X).

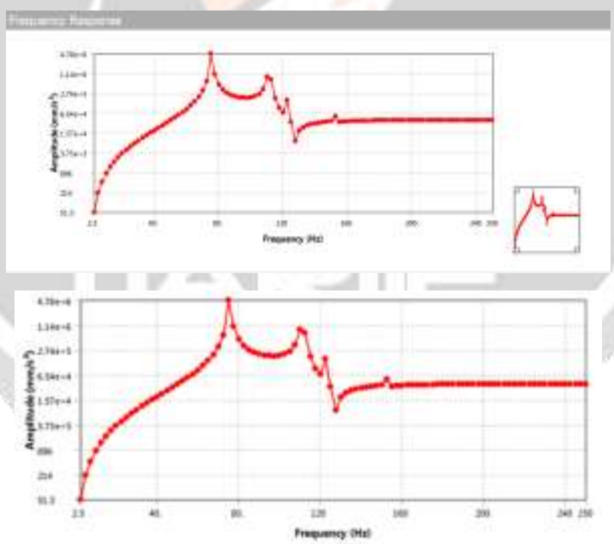

Figure 7 shows vibration amplitude in Z axis of Industrial fan

It is seen that the amplitude were increase with respect to frequency. At around the frequency 80 Hz amplitude were maximum but in the range of frequency 80-120 Hz it were slightly decrease.

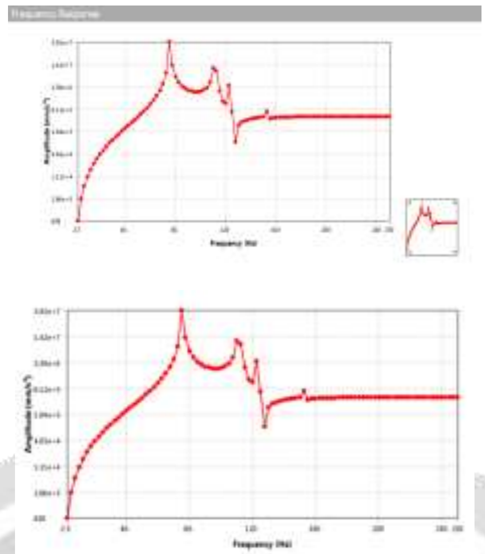

Figure 8 shows vibration amplitude in Y axis of Industrial fan

It is seen that amplitude was decrease with increase in frequency. At the frequency 80Hz, amplitude is slightly decrease but in the frequency range 120 -125 Hz amplitude is decrease in large amount.

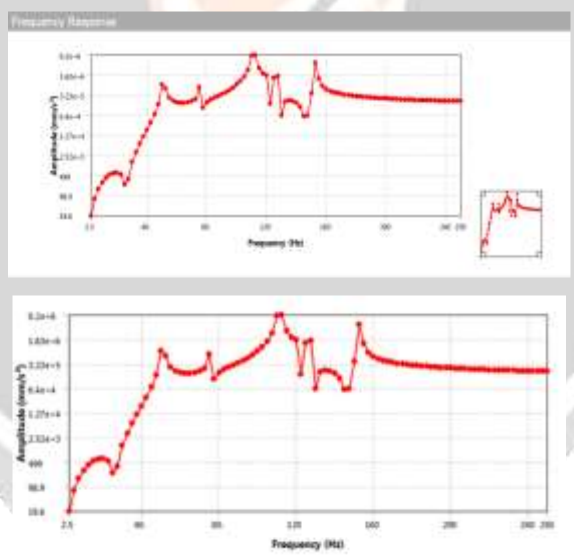

Figure 9 shows vibration amplitude in X axis of Industrial fan:

From figure 9, at the frequency range 120 to 126 Hz (i.e. blade frequency) the highest peak of amplitude was found as same in experimental value.

From figure 7-9 , it shows that as the frequency increases there were abrupt change in vibration amplitude of industrial fan’s geometry. It is indicated that vibration is maximum at this blade frequency. So these frequency components were responsible to produce noise.

2.3 Transient Dynamic Analysis

This type of analysis was used time-dependent problem in flow field. These types of analysis were used to determine the dynamic response of a structure under the action of any general time-dependent loads. In these analyses, time step sizes of 0.01 sec were selected. To obtain an acceptable solution in an acoustic transient analysis, the time increment Δt is determined by Δt =1/ (2fmax).The maximum operating frequency was estimated to determine the mesh size in the model. The spatial distributions of the pressure field were obtained by fine meshing near the vicinity of industrial fan blade to resolve the spatial variation of the pressure. For a strong coupled solution, both structure and acoustic fluid interact with each other via coupling boundary conditions in the Fluid Structure Interaction (FSI) model.

2.3.1 Distribution of pressure -

The distribution of pressure near industrial fan is shown in figure 12.

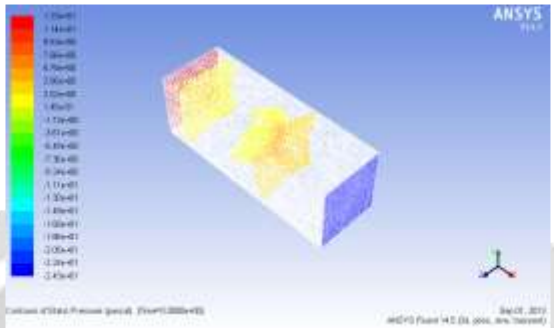

Figure 12 Static Pressure Contours

Figure 12 shows the static pressure contours. The red colour represents the high sound pressure level distribution and the blue area is the low sound pressure level. Industrial fan is applied with rotational velocity results in pressure difference. The pressure value from 13.3 Pa to 11.4 Pa was maximum which occurs at front side of industrial fan. The pressure value from -24.3 to -22.4 Pa was minimum which occurs at back side of industrial fan.

The sound is the result of pressure variation or oscillations in elastic air medium generated by vibrating surfaces.

2.3.2 Distribution of velocity

The distribution of velocity near industrial fan is shown in figure

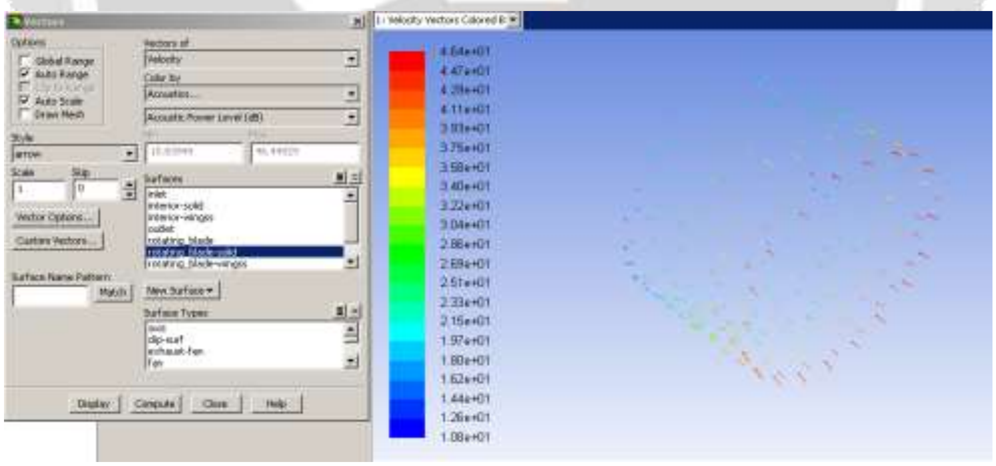

Figure 13 Velocity contours

Figure 13 shows the velocity value from 46.4 dB to 42.9 dB was maximum which occurs at front side of industrial fan. The velocity contours colored by acoustic power level which were used for further acoustic analyis. It was observed that sound power level is maximum at rotating blade. The sound pressure level were plotted in polar coordinates to understand the various positon of sound as shown in figure 14. The sound dispersion were measured by placing the microphone around the circumference of industrial fan. Every circle diameter indicate SPL value of industrial fan.

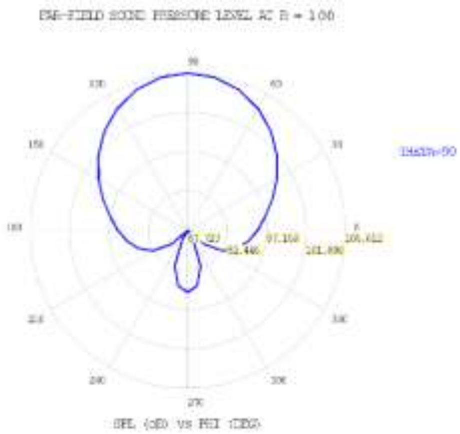

Figure 14 Far Field SPL

The polar plot shown in figure 14 was used to determine focused frequency range.It was used to evaluate industrial fan noise in horizontal plane with phase cross over frequency of 125 Hz. The maximum SPL value 106 dB was recorded over the blade frequency.

3. Optimization of Industrial Fan parameters

Parametric studies were carried out to know the effect of various factors affecting the noise level in industrial fan. The noise level was unaffected by variation in fan width and tip angle. The variation in blade thickness enhances noise level. But it cannot be chosen as optimization parameter due to its structural rigidity. It was observed that the industrial fan noise was dependent on the rake angle which was chosen as optimization parameter in the range of 250-450.

3.1 Analysis of Industrial Fan without optimization

The figure14 shows the SPL value for the industrial fan without optimization of fan parameters. Harmonic responses of industrial fan were carried out with frequency in range of 1000 Hz.

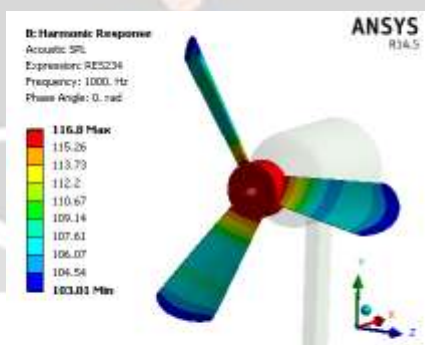

Figure 15 SPL value for Industrial fan without optimization

Figure 15 shows the red colour represents the high sound pressure level distribution and the blue area was the low sound pressure level. The maximum SPL 116.8 dB were concentrated on the hub of shaft and minimum at the tip of the blade around 103.01 dB.

The value of SPL is plotted with frequency as shown figure 16.

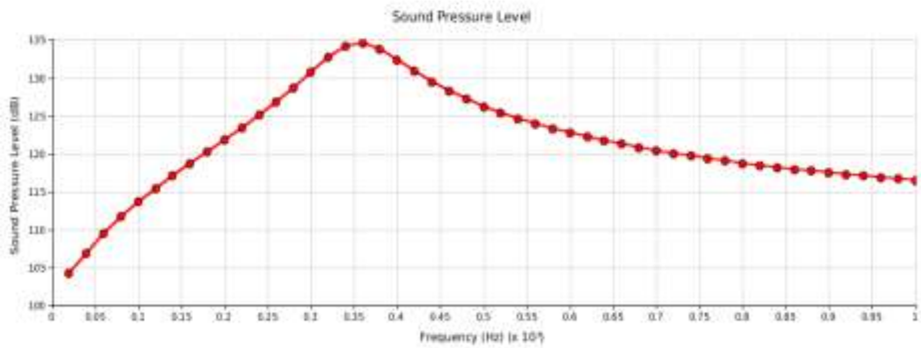

Figure 16 show that there were abrupt changes in noise level of un-modified industrial fan with increases in frequency at various locations as recorded in polar chart.

The maximum value 110 dB of SPL was located at the blade frequency as same locating in polar chart.

3.2 Analysis of Industrial fan with optimization

The optimizations were carried out by design of experiments. Harmonic responses of fan with optimization were carried out and it is observed that the maximum value occurs at blade side.

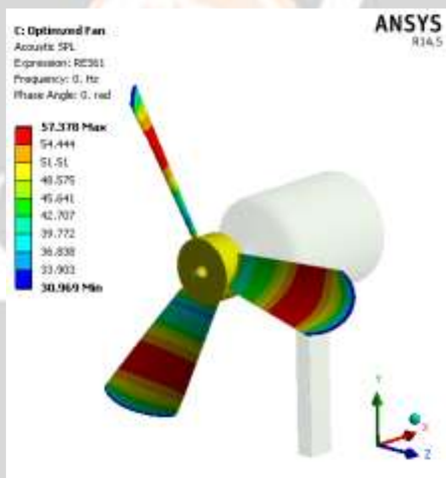

Figure 17 SPL for the Industrial fan with optimization

The figure 17 shows the maximum 57.378 dB SPL value was occurring at blade and minimum at tip of blade around 30.969 dB.

The value of SPL is plotted with frequency as shown figure 18

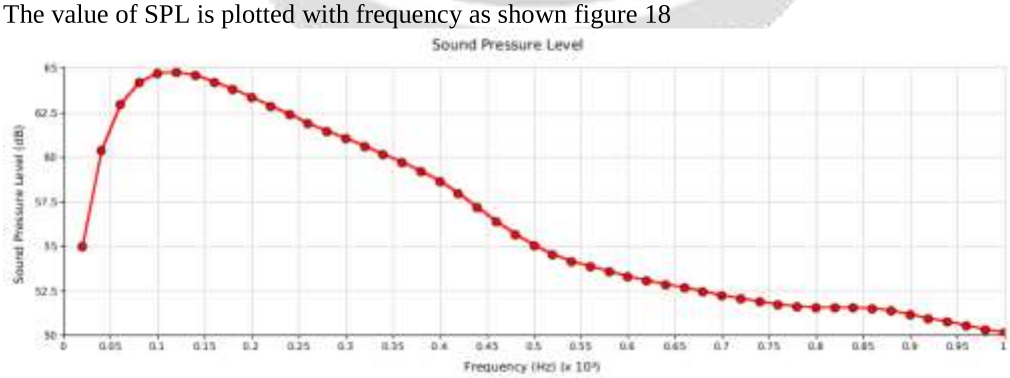

Figure 18 Relation of SPL with. Frequency rise (with optimization)

The figure 18 shows there were abrupt change in noise level of optimized industrial fans with the increases in frequency at various locations as recorded on polar chart. At the blade frequency 125 Hz the maximum 62dB value of SPL were noted.

After optimization, sound pressure level was varying with respect to rake angle as shown in following figure

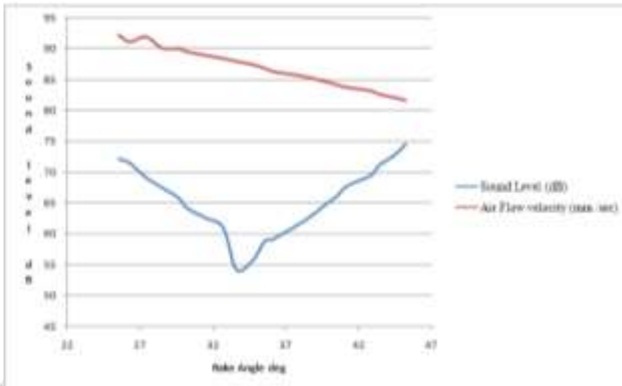

Figure 19 Relation of SPL with rake angle

It was observed that SPL is decrease with increase in rake angle. When the rake angle is in range of 32^0-34^0 sound pressure level is minimum in some extent.

4.CONCLUSION

An efficient analysis procedure is developed for the analysis of anindustrial fanusing computational fluid dynamics. The Fluid Structure Interaction technique showed significant promise for flow induced noise analysis. Predicted sound pressure level (dB) for the industrial flow fan at constant speed correlated favorably with the test data. Further design sensitivity study is being conducted to optimize the fan for flow and noise.

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