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Design and analysis of a car radiator fan assembly to mitigate the effect of aeroacoustic dipole noise

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ABSTRACT

Engine cooling fans contribute a major portion of the total noise generated by the different mechanisms in a car. Therefore the design aspect of the fan blades and housing need impeccable research and careful machining to limit the noise level to a bare minimum. The noise generated from the design aspect of the fan can be grouped into two categories, rotational and irrotational. The rotational domain takes into consideration the effect of turbulence and inflow distortion while the irrotational domain considers the effects of laminar boundary layer vortex shedding, blade interaction with tip clearance and the phenomena of blade stall. The objective is to analyze the acoustic behavior and response from the fan blade and housing when exposed to incoming airflow in a car. The process due to which the aeroacoustic dipole noise is generated is investigated and the parameters affecting the noise level are assessed. The flow and acoustic analysis are carried out on SOLID WORKS flow simulation and the generated data for our particular case is represented on a graphical scale against frequency. The causes behind noise generation in a fan assembly is studied and major emphasis is laid on the aerodynamic factors, which affect the noise generation in the radiator cooling fans. After validating the theoretical procedure, an attempt is made to redesign the existing fan assembly structure by adding grills and MPP dampeners and source modification are done to reduce noise levels.

Keywords— *Acoustic behavior, Aeroacoustic dipole noise, Solidworks, Turbulence, Vortex shedding*

1. INTRODUCTION

A conventional automotive engine cooling system consists of a radiator, cooling fan, water pump, coolant, reservoir, thermostat, heater core. Radiator cooling fan is a device that works to pull the air through a radiator, to regulate engine temperature. Nowadays, there is a significant reduction of noise in the interior and exterior of a car. The demand for reduction in the automobile radiator fan noise is a point of interest for every carmaker. The motive is to drive for a reduction in noise level, which could enhance the comfort in the journey for passengers. Noise level increases due to the non-uniform flow of air over the engine components. Radiator cooling fans are found to be the major contributors for vehicle noise other than the engine. The Blade passing frequency noise is studied to be primarily repulsive to the ears. Considering the regulations and growing importance of acoustic comfort in many markets, there is high value in addressing cooling fan flow-induced noise problems as early as possible during product development. A cooling fan, in particular, creates a pressure difference by exchanging momentum from the blades to the surrounding gaseous fluid. The primary purpose of the fan assembly is to maneuver a required volume flow rate of air. The type of fan-based noises in a radiator fan assembly can be distinguished into

Four categories mainly consisting of tonal noise, broadband aerodynamic noise, motor, and mechanical noise. The fans incorporated into the housing in our particular drawing can be further characterized into two types, axial and centrifugal. Radial, backward-curved and forward-curved types of fans are mainly used in industrial applications. The radial fan is considered the noisiest and least efficient but works really well in the case of dirty and corrosive flows. We are using a low-tip speed axial fan in this experiment with a steel housing and blades.

2. LITERATURE REVIEW

Design of cooling fan for noise reduction using CFD. G.V.R. Seshagiri Rao, Dr. V. V. Subba Rao, International Journal of Scientific and Engineering Research Volume 2, Issue 9, September-2011 ISSN 2229-5518. Noise reduction is one of the prime factors of consideration in the design of cooling fans. In this paper, a cooling fan of a seawater pump is kept as a test subject and the noise source is first identified. Once it is done then they can be eliminated or reduced by modifying various design parameters and operating conditions. The fan blades, in this case, exchange momentum to the surrounding medium thereby creating a pressure difference. The extensive study of parameters affecting the noise generation in this paper reveals that fan noises mainly consist of the following components, namely, tonal noise, aerodynamic noise, and mechanical noise. This process is carried out in two main phases, the first being a CAD modeling and designing of the fan followed by the analysis, the second phase involves actual experimental setup and testing which is then compared to software test results. We observe that a 10-blade axial flow fan is taken into consideration. It is assumed to run at a constant single speed for simpler analysis. The main objective is to reduce fan blade noise without interfering with its operating speed, and propose a design criterion for the same. A baseline fan design with 10 blades is designed using CATIA, and a meshed model in exported to Fluent, where necessary boundary conditions are set. A control volume technique is used, which is governed by Naiver-Stoke equations. Then, a detailed CFD analysis is carried out, with a set of parameters. Using the mean velocities at different points and fluid flow through the fan at different operating speeds, the noise radiated from the fan is predicted. In an attempt to reduce the Blade Pass Frequency (BPF) noise of this fan, a mixed flow fan with a seven-blade airfoil design is tested under similar operating conditions.

Study of Fan Noise Reduction for Automotive Radiator Cooling Fans, Atsushi Suzuki Tetsuo Tominaga, Tsuyoshi Eguchi, Toshifumi Kudo, Tomoshige Takata, Mitsubishi Heavy Industries, Ltd. Technical Review Vol. 43 No. 3 (Sep. 2006). According to this paper, in order to reduce radiator fan noise, we must reduce Blade Passing Frequency (BPF). They are able to predict using Computational Fluid Dynamics (CFD). The prediction was accurately done using element test. They are able to compare the experimental and predicted results of BPF noise . They investigated the degree of interaction of the upstream and downstream, flow inter-action devices, straight type, and cross type, were prepared and used. The parameters were fixed for fan diameter, blades, BPF (Primary), RPM, the elemental test was conducted to find the pressure distribution by flow, and the analysis was noted. The flow interaction for both the bottom and top device position experimented at each point for time step and static pressure. They found out that a phenomenon similar to the potential interaction occurs near the fan rotational center at 108-time steps, causing the static pressure to increase. The region with more fluctuation is on suction surface side and on the pressure side. Finally, they compared BPF noise level for all the five models they verified the accuracy of BPF noise prediction, the BPF noise level obtained in the experiment was over-lapped onto the BPF noise prediction result and plotted to compare the predicted value with the experimental value.

Micro-Perforated materials for the reduction of flow-induced noise. 22nd International Congress on Acoustics. Engineering Acoustics: Paper ICA2016 – 188. Teresa Bravo, Cédric Maury, and Cédric Pinhède, Carlos de la Colina. The paper showcases the results obtained when an insulating partition composed of micro-perforated plates is exposed to a TBL (turbulent boundary layer excitation). The results are based on simulations which are established to have a comparison between the performance of the MPP control devices when the physical configurations of the partition and the nature of the primary noise excitation are varied. Noises generated due to Turbulent boundary layer excitation is considered a key factor and a major source of air induced sound power level generated in a fan assembly. The authors have performed experiments in the low-speed wind tunnel and have determined the acoustic and aerodynamic TL performance of a number of MPP multilayer partitions. With numerous statistical data, it is found that if we insert a micro-perforated panel within the cavity at unequal distances from the front and back panels then we can dampen more efficiently the Mass-Air-Mass controlled resonances of the Panel- Cavity-Panel system with respect to those of the MPP-Cavity-Panel system, which is already damped by the front MPP. This results in a higher TL difference between the triple and double partitions with a plain front panel.

Shape optimization for aerodynamic noise control, Alison L. Marsden, Meng Wang, and Bijan Mohammadi. Center for Turbulence Research Annual Research Briefs 2001 University of Montpellier, France. Noise generated by turbulent boundary layers near the trailing edge of the lifting surface is a serious issue for many applications. In this study, a shape optimization technique based on control theory in conjunction with a gradient-based minimization algorithm known as a method of incomplete sensitivities has been formulated and implemented for reducing the aerodynamic noise from a lifting surface. A model problem consisting of two-dimensional unsteady laminar flow over an airfoil is considered with initial trailing edge tip angle of 45 degrees and whose upper surface of the right half is allowed to deform. In this method of incomplete sensitivities, first, a general optimization problem is stated consisting of a cost function with given partial differential equation and its boundary conditions. Then the gradient of the cost function is computed and finally, we outline the steps in the algorithm used to optimize the airfoil shape. Here we wish to minimize the given cost function which is related to an acoustic source in our problem and for that, a flow simulation is performed using the incompressible Navier-stokes solver. At low Mach number, the noise generated by large-scale vortex shedding, the airfoil is acoustically compact, prevails dipole radiation.

Noise Reduction for Automotive Radiator Cooling Fans, Sabry ALLAM, Mats ÅBOM Automotive Technology Department, Faculty of Industrial Education, Helwan University. 11232, Cairo-Egypt. KTH - Competence Centre for Gas Exchange (CCGEx) the Marcus Wallenberg Laboratory (MWL). SE-100 44 Stockholm, Sweden. (Conference Paper · April 2015). Radiator cooling fans in automobiles are major contributors of noise other than the engine. It is necessary to take into consideration the components of noise reduction in the earlier stages of design to reduce the costs. Noise in fans can be reduced by two methods that are, noise reduction by source modification and noise reduction by transmission path modification. This research contributes to noise reduction by transmission path and the paper focuses on the blade passing frequency of the fans and is a detailed study on flow generated noise, characterize the heat exchanger damping properties and investigate the use of near-field noise control by Micro-Perforated (MPP) shrouds and tuned MPP dampers. The results of the experiment showed that the total sound power radiated from radiator cooling fan when a Micro-Perforated (MPP) was incorporated was reduced by 1.5 to 4.5 dB(A), depending on the covered area and fan speed. The absorption on the backside is significantly increased which in turn reduced the noise levels further. The application of tuned MPP damper was significant and gave a reduction of 3-4 dB (A). Incorporating both MPP shroud and tuned MPP can reduce the total sound power around 6 dB (A).

3. FAN NOISE MECHANISM

The fan considered for sound level testing is of a low- tip speed axial flow type. An axial fan utilizes the swirling tangential motion of the rotating impeller blades to propel the blades in the axial direction i.e. parallel to the axial shaft. The sound level depends on the airflow and structural aspect of the fan assembly. The fluid flow can be characterized into a rotational and irrotational flow. The rotational part deals with the laminar boundary layer vortex shedding, blade interaction and with tip clearance vortex and blade stall, while the irrotational aspect involves parameters such as inflow distortion and turbulence. These factors are responsible for aero-acoustic dipole noise generation. In this analysis, the monopole and quadruple noise

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are neglected owing to its negligible impact on subsonic fans. As observed from the recent experiments in the field of aero-acoustics we discover that the aerodynamic factors are a major source of dipole noises in fans. In the case of a non-uniform inlet flow as the angular position of the blades changes a non-uniform aerodynamic force acts on the blades resulting in an increase in noise level at blade passing frequency and at its harmonics. Similarly, if the fan blades are not properly spaced from the rotor casing or if the structure enclosing it is asymmetric in shape than also a dipole noise is induced. Another important parameter to consider is the noise from the stall. It occurs when the flow is locally disturbed and takes place at low flow rates. Stall induces a series of imbalanced forces on the fan assembly causing noise and vibrations. In addition to steady aerodynamic forces, all the unsteady factors also contribute towards the noise generation. Due to vortex shedding occurring at the laminar boundary layer of a low tip speed fan, flow separation turning the noise into broadband and due to a vortex generated between the fan blade tip and the casing a decent level of sound power level is observed.

4. NOISE REDUCTION METHODOLOGY

The approach is to design a CAD model of a cooling fan inside a radiator assembly using solid works followed by a flow-simulation. The objective behind doing a flow-simulation of the CAD model is to trace the flow pattern and establish the sound power levels at each point where the air flows. The dimensions of the fan and shroud assembly are maintained as per standards used in the market nowadays. In this case, study an FEA analysis test is performed with varying parameters. The parameters directly affect the sound level. The prime focus has been to identify factors, which affect the sound and vibration level during the fan operation. Some of the steps taken are:

- Varying space between the fan and casing to prevent tip induced vortex generation and hence study its effect of sound power level.
- Inserting anti-stall devices and check the sound power level.
- Reduction in noise and vibration by altering the fan RPM.
- Reducing vortex shedding behind the fan.
- Altering the number of fan blades to detect the changes in sound power level.

Since this is just a computational analysis, no physical models were created to prove the experiment.

5. PRELIMINARY DESIGN APPROACH

The preliminary design approach had numerous components that were assembled to provide a cooling assembly unit. The components are as follows:

5.1 Axial Fan Blade

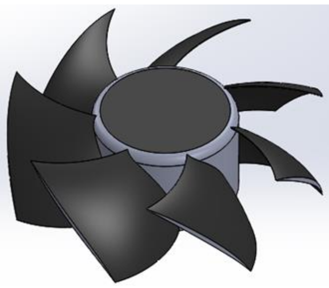

Fig. 1: Axial fan blade -7 blade setup

Material: Hard plastic and nylon Specifications:

- Number of blades: 7
- Diameter: 250 mm
- Type of fan: Axial flow fan.
- Bearing Type: sleeve / Dual ball bearing.
- Operating speed: 900 - 1800 RPM.

5.2 Fan Casing

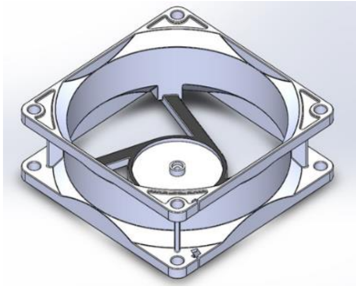

Fig. 2: Stainless steel alloy casing

Material: Stainless steel Specifications:

- Dimensions: 350 mm * 350 mm * 250 mm
- Diameter: 300 mm
- Stator and rotor setup

5.3 Laminated disc rotor setup

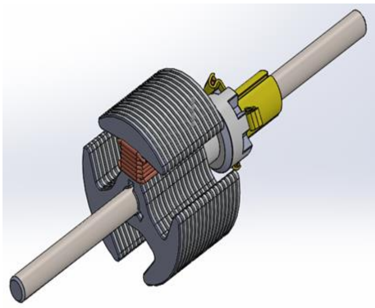

Fig. 3: Laminated disc rotor setup

Material: Carbide steel Specifications

- Shaft: iron/ 52 mm
- Rotor diameter: 100 mm
- Core disc: 16 plates/ stacked/ steel alloy

5.4 Flow simulation Test (SOLIDWORKS)

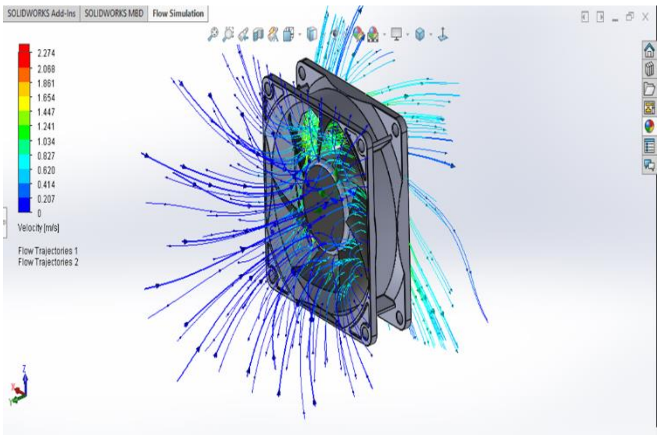

Fig. 4: Flow simulation model

The designed CAD model of the fan was subjected to velocity flow simulation. The parameters were as follows:

- Fan speed: 900 RPM
- Three dimensional flow
- Wind Speed:10 m\s
- Medium air considering no humidity

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- Reference axis: Y-axis
- External rotational flow: Local averaging
- Flow type: Laminar and Turbulent
- Pressure: 101325 Pa
- Temperature ambient: 303 K
- Fan Temperature: 350 K
- Computational domain: 50 mm * 100mm* 50 mm (deceased and made lighter to enable faster processing).
- The rotating region defined diameter: 75 cm
- Real wall Inner part of the housing is defined as the stator (Boundary defined).
- Temperature (wall): 303 K
- Pressure (wall): 101325 Pa

Flow trajectories
- Offset: 20 mm
- Instances: 160
- Defining the maximum acceptable wind speed: 2.2 m\s at 900 RPM at the blade edges, as per the simulation the maximum value of the speed obtained = 1.4m\s

5.5 Sound Power Level Simulation (SOLIDWORKS)

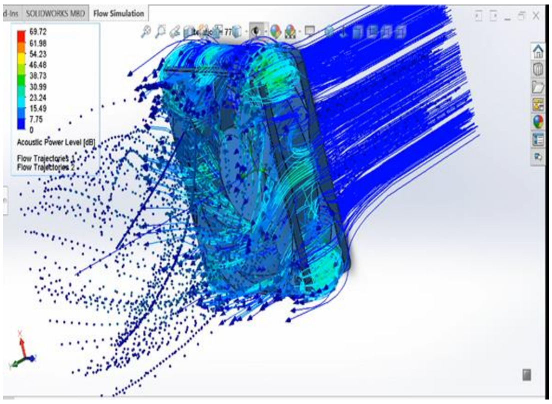

Fig. 5: Sound Power level simulation

The designated CAD model is subjected to Acoustic power Flow simulation
The parameters were as follows:
- Fan speed: 600 RPM
- Wind Speed: 15 m\s
- Medium air considering no humidity
- Reference axis: Y-axis
- External rotational flow: Local averaging
- Flow type: Laminar and Turbulent
- Pressure: 101325 Pa
- Temperature ambient: 303 K
- Fan Temperature: 350 K
- Computational domain: 100 mm * 100mm* 100 mm (deceased and made lighter to enable faster processing).
- Rotating region defined diameter: 75 cm

Real wall Inner part of the housing is defined as the stator (Boundary defined)
- Temperature (wall): 303 K
- Pressure (wall): 101325 Pa

6. OBSERVATION

It is observed that the static pressure distribution is maximum on the blade leading edges and it reduces as we move towards the trailing edges. The increment is in a monotonic manner. The maximum value of the sound power level obtained is 67 dB and the observed value is close to 30 dB at the edges. Now after changing the parameters such as the number of fan blades and increasing the distance between the fan blade tip and the casing another flow simulation is conducted.

6.1 Sound Power Level Simulation – Test 2

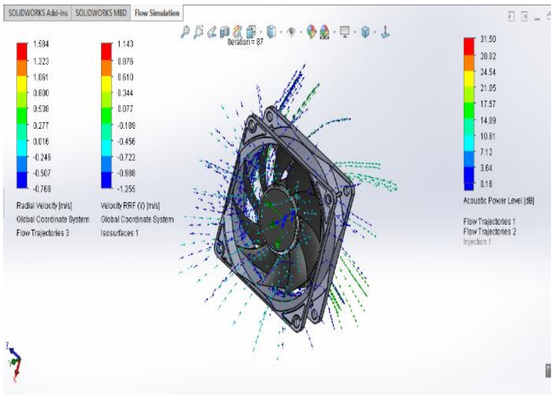

Fig. 6: Sound Power level simulation with varied parameters

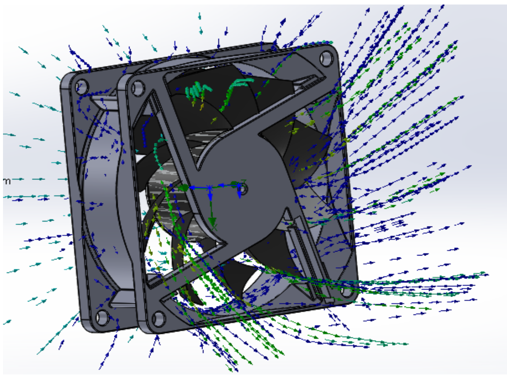

Fig. 7: Flow trajectories showing high and low-pressure concentration

7. OBSERVATION

In this process, the number of fan blades was increased from seven to 10. The spacing between the blade tip and casing was increased significantly and the size of the blades was reduced. The collective result of this modification resulted in an increase in sound power level at the edges. The occurrence of vortex shedding, boundary layer turbulence, and blade pressure field interaction contributes towards broadband of “white noise”.

We discover tonal noise, also referred to as Blade Passing Tone (BPT) occurring at the blade passing frequency. It is a result of rotor – stator interaction. Increment in the number of blades results in a decrease of aerodynamic noise near the tip of the fan blade. However, a proportion of the broadband noise will first increase and then decrease. Typically, the air pressure increases with the increase in the number of blades from seven to ten. Thus causing an increase in the blade tip induced noise.

8. CALCULATION

The sound power produced by fans can be approximated by using a formula:

$$LW = KW + 10 \log10 Q + 20 \log10 P + BFI + CN$$

Where,
LW= sound power level (dB)
KW = specific sound power level depending on the type of fan.
Q = volume flow rate (cfm)
P = total pressure
BFI = Blade Frequency Increment = correction for pure tone produced by the blade.

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BPF = blade passing frequency = number of blades × RPM/60 (Hz)

CN = efficiency correction (because fans that are operated off their optimum flow conditions get noisier)

$$CN = 10 + 10 \log 10 (1-\eta)/\eta$$

η	Cn
90%	0
75%	5.2
40%	12.2

η = Hydraulic efficiency of the fan = Q×P/ (6350×HP)
HP = nominal horsepower of the fan drive motor.
Blade passing frequency = number of blades × RPM/60 (Hz)
= 70 Hz
BPI = 4
Center frequency = 65 Hz
Kw = 32 dB

We have considered a tubular centrifugal fan setup in the design and the value of specific sound power level based on the center frequency of 2000 Hz is 32 dB (information acquired by the empirical table provided by fan manufacturers). The value of efficiency considering non-ideal condition is taken as 75% and simultaneously the value of CN obtained is 5.234. Two conditions are taken into account.

Condition 1

- RPM: 600
- Number of blades: 7
- Fluid considered: air
- Humidity level: 30%
- Wind speed: 15 m/s
- Diameter of the blade setup: 250 mm
- Volume flow rate: 2500 CFM
- Pressure: 407 inches of water.

$$LW = 32 + 10 \log 10 (2500) + 20 \log 10 (407) + 4 + 5$$
$$= 127 \text{ dB}$$

Condition 2

- RPM: 1200
- Number of blades: 10
- Fluid considered: air
- Humidity level: 50%
- Wind speed: 20 m/s
- Diameter of the blade setup: 250 mm
- Volume flow rate: 3000 CFM
- Pressure at exit: 450 inches of water

Since,

Blade passing frequency = number of blades × RPM/60 (Hz)
= 200 Hz

BPI = 5.5
Center frequency = 250 Hz
Kw = 43 dB

$$LW = 43 + 10 \log 10 (3000) + 20 \log 10 (450) + 5.5 + 5$$
$$= 141.3 \text{ dB}$$

9. OBSERVATION

(a) Sound power level increases with the increase in the number of blades.
(b) The airflow at the rear of the fan is increased due to more blade air interaction.
(c) With the increase in distance between the fan tip and the shroud, eddy starts to form at the edges. This results in the formation of unsteady forces at the end corner, which induces structural vibrations.
(d) This results in an increase in the noise level.
(e) Due to an increase in static pressure at the exit a phenomenon known as "wake" is observed. The wake region is formed due to less flow velocity and high pressure of air. It causes lag to the motion of the fan thereby inducing vibration and noise.

10. CONCLUSION

The topic revolves around the basic approach taken for measuring the sound power level in radiator fans. However, a further detailed study involving physical models and proper CFD codes need to be done. The flow simulation and the pressure fluctuation tables conducted in this research paper could provide the acoustic power and noise level data. To further research into this topic a wind tunnel test is highly recommended as it displays all the parameters involved in noise generation. In addition, the paper also covers conditional experiments carried out in a design and modelling software, which can be, improved by testing the model in a CFD system so that static pressure fluctuation and its effect on parts of the blade can be obtained.

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