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Reduce the Cooling FAN Blade Pass Frequency Noise in Electrical Vehicle

Chuanxue Duan, Wei Qian

Geely Automobile Research Institute (Ningbo) Co., Ltd, Hangzhou Bay, Ningbo, China

Email address:
Chuanxue.duan@geely.com (Chuanxue Duan)

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Abstract: Air duct in the inlet of the cooling fan is often used in EV (electrical vehicle) due to the usage of AGS (active grille system) and profile. It often leads to the unexpected noise of the cooling fan, the overall level and the BPF (blade pass frequency) noise becomes much worse than before significantly. The experimental results shown that the boundary conditions of the cooling fan affect the noise sensitively. So, the whole engine room CAE model is set up to apply the Computational Fluid Dynamics (CFD) and the Computational Aero Acoustics (CAA). Thus, the CFD and CAA synthesis methods are used to predict the fan noise. After theoretical analysis, it is found that the fan shroud is the most sensitive area which significantly affect the BPF noise. Then different shroud proposals are applied to find the best solutions for the BPF noise, The CAA results shown that the BPF frequency peak of the interior noise reduces 5dB (A) for the best shroud case. Then the best proposal one is manufactured and assembled in EV to validate the effects. The Noise in the EV also cut down in 5-8 dB (A). The Experimental and CAE synthesis method for the whole engine room analysis is confirmed to be useful to solve the cooling FAN BPF noise issue. According to the above researches the cooling FAN NVH design rules are drawn and used in EV NVH design.

Keywords: EV, Cooling FAN, Blade Pass Frequency (BPF) Noise, CFD, CAA

1. Introduction

Axial flow fans are widely used in civil electronic equipment such as air conditioning, vehicle cooling system, computer rooms, relatively high flow rate and low cost. Near all the cooling fans in the electrical vehicle are axial fans. Except the flow rate performance, the aerodynamic performance becomes more and more important that affects the quality of the people's living and working environment. So it is very meaningful for the improvement to study aerodynamic performance of axial fans and find ways to reduce the noise [1-6].

The main factor which influences the aerodynamic performance is the fan internal flow fields, such as adverse flow and vortex flow. Most of the aerodynamic noise is blade pass frequency noise, ant it can be studied by CFD simulation and experiments.

According to experimental results, a fan speed reduction can decreases the noise level to about 3-7 dB. [8]. Increase the diameter directly affects the overall noise generated. The increase in number of blades from 6 to 7 however was observed to change the noise spectrum in addition to a decrease in overall noise due to change in the phase angle with which the individual signals from each sector of the entire fan interact [11].

Another method to study the FAN noise is using CFD (Computational Fluid Dynamics) and CAA (Computational Aero Acoustics) techniques [7, 10]. There are two methods are used to compute the cooling fan noise source. In the first method the source is computed from the flow field obtained using the unsteady Reynolds-averaged Navier-Stokes equations (unsteady RANS, or URANS) model. In the second method, the azimuthal modes of the flow field obtained using the steady RANS with the moving reference frame (MRF) model are treated as the "sound source" [12-14].

Also the dual fans and more fans are assembled together are also studied [9]. It affects the total noise greatly, but it is not studied in this paper.

There are two typical noise in the cooling fan: One is the rotational noise and the other one is the vortex noise. The rotational noise is from the rotation and output power of the fan, which the order characteristic is prominence. The vortex noise is related to the fan blade, inlet and outlet of the fan, and it is very hard to be estimated.

To improve the production competitive, the air and heat management becomes more and more critical, also the comfortable requirements asked for lower and lower noise in vehicle which is mainly come from the cooling fan operation. The two factors, air flow rate and noise are conflictive in design normally after boundary conditions are limited in EV. Then the quiet fan with the required air flow rates becomes more and more important. Normally the subjective assessment is used to estimate the fan noise, but there is no directive method about how to improve the fan noise and keep the air flow rate or improve it.

With the development of the EV, more and more inlet grills are used to control the air flow rate, and the engine room is often packaged to prevent the heat leakage. The boundary conditions of the cooling fan changed greatly comparing with the classical vehicle. The resistance of the fan is improved at the same air flow rate requirements, then the aerodynamic performance of the fan is totally difference than before.

In this paper, the CFD method is used to obtain the air flow distribution in the engine room of the EV, thus the fan model is set up, which the engine room of the vehicle is involved to provide the fan inlet an outlet boundary condition exactly. Then the CAA method using URANS is established and calculates the time serious data for the noise. After FFT (Fast Fourier Transform) processing, the frequency spectrum is achieved. With the CAE analysis and experiment in EV, the proposal with the FAN shroud is conformed to be serviceable. The results show that the BPF noise in EV is cut down in 7-10 dB (A).

2. The Cooling Fan BPF Noise

When the cooling FAN is assembled in the new EV, all grill and deflector are installed, the BPF noise of the cooling FAN is terrible while the air conditioner is running. Comparing with the former condition, which there is no inlet deflect assembled in the vehicle, the same cooling fan has different noise performances. The comparisons between the interior noise with and without the deflector in EV are shown in Figure 1.

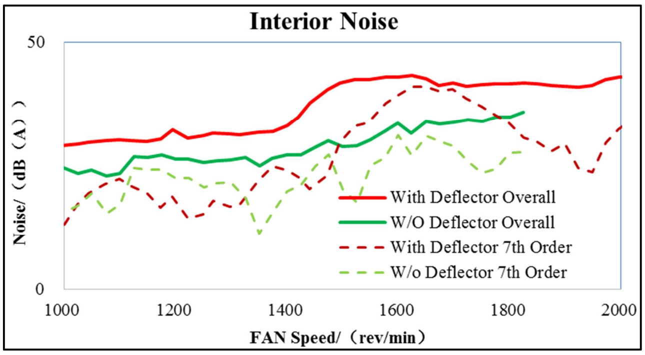

Figure 1. Comparison between with and without deflect.

There is grill, AGS and deflector in front of the fan, refer to Figure 2, which is different with the normal vehicle.

The noise of the cooling FAN becomes worse significantly and the 7th order is the BPF (blade pass flow) order. It has above 10 dB (A) gaps than before. There is significantly peak of 1500-1800rpm in the BPF noise order line. It sounds terrible inner the vehicle and should be the serious issue.

The first action is disconnecting the cooling FAN, which identifies the structural path influences. The order noise result is shown in the Figure 3. It proved that the contribution of the structural path can be ignored in the cooling FAN noise transform path.

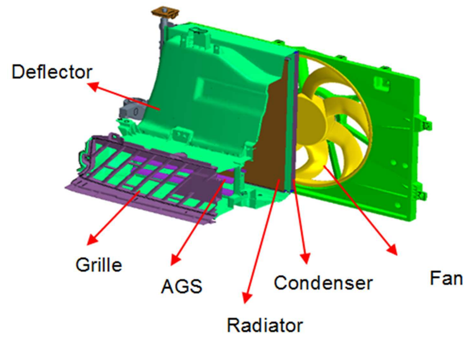

Figure 2. The inlet components.

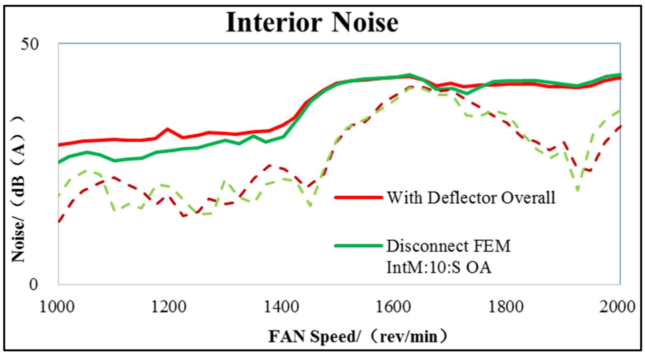

Figure 3. The structural influence results.

3. Theory Analysis

3.1. CFD Theory

The flow calculations for the conservation of mass and momentum are performed through steady-state Reynolds Averaged Navier-Stokes equations (RANS) whose governing equations for the turbulent incompressible flow are

Continuity

$$\frac{\partial}{\partial x_i}(\rho \bar{u}_i) = 0 \quad (1)$$

Momentum

$$\frac{\partial}{\partial x_j}\left(\rho \overline{u_i u_j}\right) = -\frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j}\sigma_{ij} + \frac{\partial}{\partial x_j}\left(-\rho \overline{u_i' u_j'}\right) \quad (2)$$

In the two equation, the subscripts i and j indicates the ith and jth components of the Cartesian coordinate respectively, $\bar{u}$ is the mean velocity, m/s and $\bar{P}$ is the mean pressure, Pa, ρ is the density of the fluid, kg/m3, σ_{ij} is the stress tensor, Pa, $-\rho\overline{u_i' u_j'}$ is the Reynold stress R_{ij}, Pa.

The RAN k-ε model is used to set up the CFD model.

Based on the assumed condition that the air is incompressible, the equation is as follows,

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\alpha_k \mu_{\text{eff}} \frac{\partial k}{\partial x_j}\right)+G_k+G_{\text{b}}-\rho\varepsilon-Y_{\text{M}}+S_k \tag{3}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\alpha_\omega \mu_{\text{eff}} \frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_{\text{b}})-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \tag{4}$$

G_k is the turbulence momentum by the velocity gradient of the static flow, G_b is the floatage turbulence momentum, Y_{M} is the wavelets of the diffusion item, C_1, C_2, C_3 are the static number, α_k is the Turbulent Prandtl number for k equation and α_ε is the Turbulent Prandtl number for ε equation. S_k and S_ε are defined by different case.

$$R_\varepsilon=\frac{C_\mu \rho\eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}\frac{\varepsilon^2}{k} \tag{5}$$

There, $\eta=Sk/\varepsilon$, $\eta_0=4.38$, $\beta=0.012$.

3.2. CAA Method

The Ffowcs Williams and Hawkings (FWH) equation represents an exact rearrangement of the continuity equation (1) and the momentum equation (2) into the form of an inhomogeneous wave equation (6) [7, 15].

$$\frac{\partial^2\rho'}{\partial t^2}-c_0^2\frac{\partial^2\rho'}{\partial x_i^2}=\frac{\partial}{\partial t}\left(\rho_0 v_{\text{si}}\delta(f)\frac{\partial f}{\partial x_i}\right)+\frac{\partial}{\partial x_i}\left(\sigma'_{ij}\delta(f)\frac{\partial f}{\partial x_i}\right)+\frac{\partial^2}{\partial x_i\partial x_j}\left(T_{ij}H(f)\right) \tag{6}$$

There, ρ' is density fluctuation, ρ_0 is density at rest, t is reception time, c_0 is the speed of sound in the flow., T_{ij} is the Lighthill's Tensor, δ is the dirac delta function, f is a function of the geometry and kinematics of the moving surface, H is Heaviside step function, v_{si} is the velocity at the moving surface and σ' is the compressive stress.

In equation (6), $\frac{\partial^2}{\partial x_i\partial x_j}\left(T_{ij}H(f)\right)$ is quadrupole source

$\frac{\partial}{\partial x_i}\left\{\sigma'_{ij}\delta(f)\frac{\partial f}{\partial x_i}\right\}$ is Dipole source (or) loading source

$\frac{\partial}{\partial t}\left\{\rho_0 v_{\text{si}}\delta(f)\frac{\partial f}{\partial x_i}\right\}$ is Monopole source (or) thickness source

Since the air flow is in the very low velocity, only the dipole source is considered.

$$\frac{\partial^2\rho'}{\partial t^2}-c_0^2\frac{\partial^2\rho'}{\partial x_i^2}=\frac{\partial}{\partial x_i}\left(\sigma'_{ij}\delta(f)\frac{\partial f}{\partial x_i}\right) \tag{7}$$

3.3. FAN Noise Simulation Methodology

The Hybrid CAA which involves two stages: CFD and CAA, and the cooling FAN noise spectrum is from FFT of the time series.

The first stage is the solution of the flow data, The CAE model includes the inlets and outlets of the FAN instead of the cooling FAN single model. Refer to Figure 4.

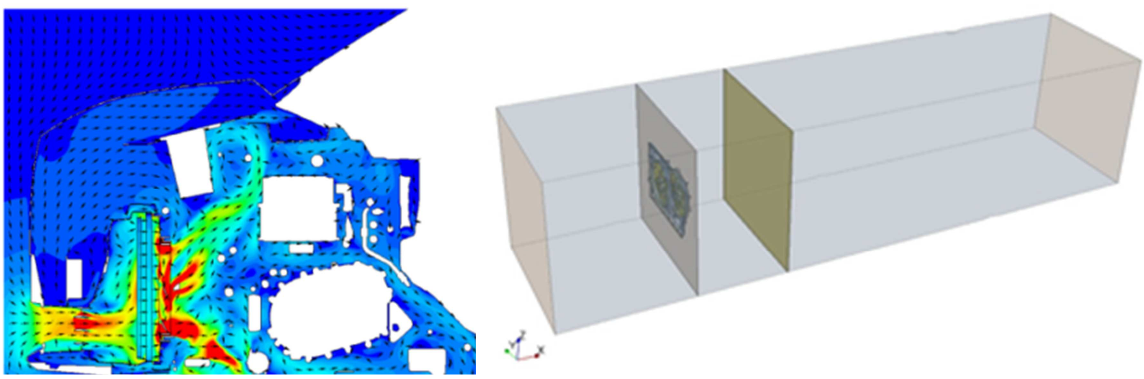

Figure 4. *Whole CFD model.*

The second stage is the computation of the acoustic data from the flow field. Steady-state flow data of the fan which are extracted from the CFD simulation acts as the acoustic source input for the acoustic propagation. The equation (6) is used to predict the sound power level at the defined receiver point. The receiver point is the driver's ear.

$$\text{SPL}=20\log_{10}\left(\frac{p}{p_{\text{ref}}}\right) \tag{8}$$

Where, $p_{\text{ref}}=2\times10^{-5}\text{Pa}$

One second length data in 10^6 steps is calculated and the frequency spectrum is obtained by FFT processing.

4. CAE Model Results and Solutions

The air flow rate is depending on the difference of the inlet and out let the fan unit. There the fan rotation speed is 1600rpm. The CFD results is shown in Figure 5. And the air flow velocity map near the FAN and the FAN plane section are shown in Figure 6.

Since the FAN have 7 blades, the BPF frequency is

$$1600\div60\times7=187\ \text{Hz}$$

The acoustic map distribution with band pass filter (160-210Hz) are shown in Figure 7.

It can be seen that the noise is from left side and from the gap between the shroud and the wheel. Thus different cases about the shroud wings are carried out to find the best one. Finally the best proposal which have lowest BPF noise is shown Figure 8.

The CAE results comparisons in 160-210 Hz frequency section are shown in Figures 9&10. It be seen that the noise distribution near the FAN rings changes significantly. The noise in the Right sight inner the engine room is reduced significantly.

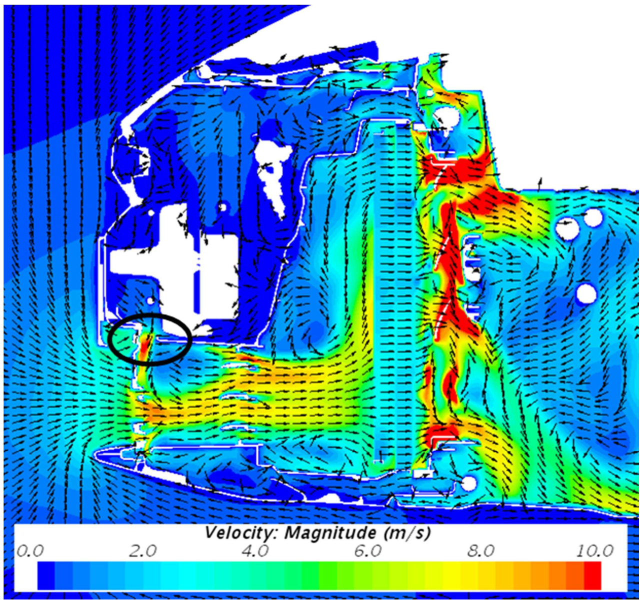

Figure 5. *CFD model.*

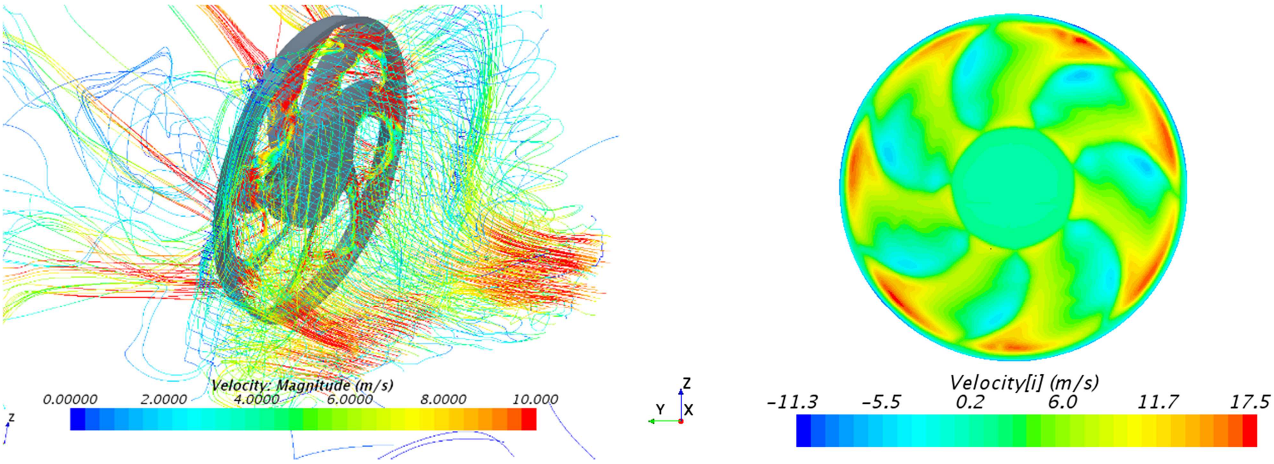

Figure 6. *Air flow near the FAN.*

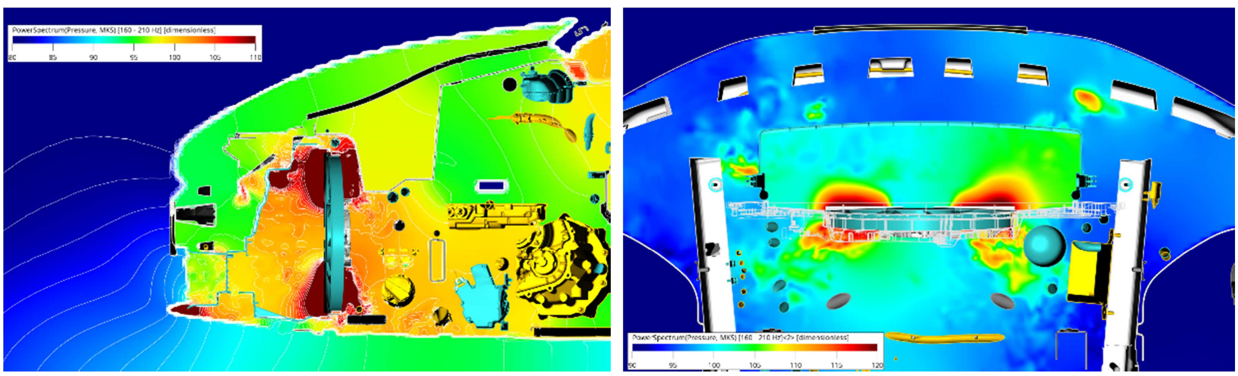

Figure 7. *Acoustic Map in Y and Z section.*

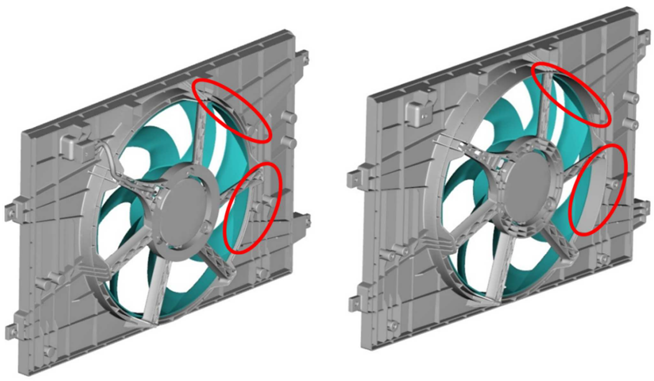

Figure 8. *Modification of the FAN.*

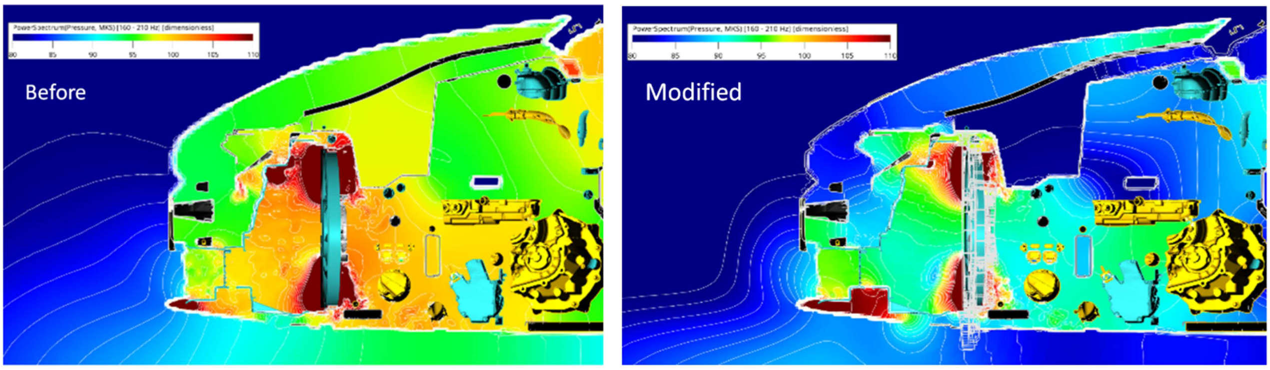

Figure 9. *Noise results comparison –Y section (160-210Hz).*

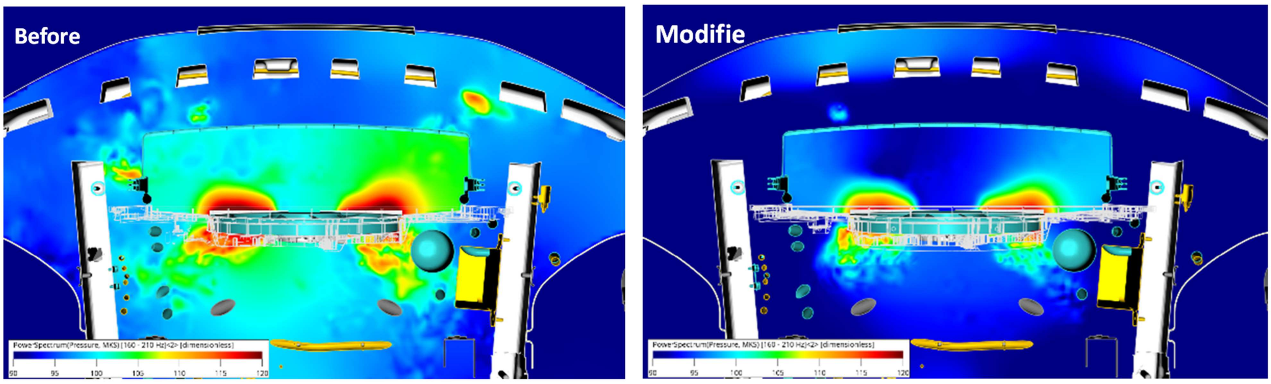

Figure 10. *Noise results comparison –Z section (160-210Hz).*

The spectrum of the CAA analysis results of the point which inner the engine room are shown in Figure 11. The peak of the spectrum in 190 Hz is reduce in 5dB (A). The CAE shows the solution is useful.

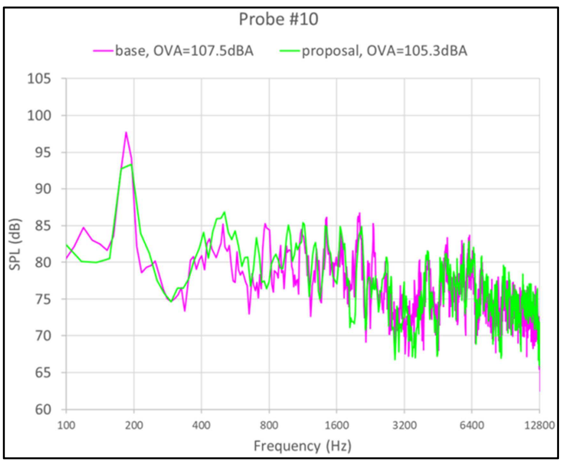

Figure 11. *Spectrum of CAA Results.*

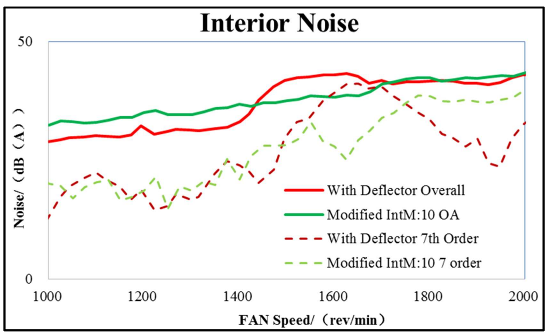

Figure 12. *Modified Noise results in Vehicle.*

Then the FAN is assembled in the vehicle, and the noise test is carried out again. The inner noise results are shown in Figure 12 and compared with the baseline. The overall noise is modified and becomes linear. The BPF noise is getting more linear, the terrible BPF noise in 1500-1800rpm is eliminated, the peak is cut down in 5-8 dB (A).

The modifications about the FAN are the correct proposals for the noise issue.

5. Conclusions

The Experimental and CAE synthesis methods are useful for the cooling FAN noise analysis in vehicle. The whole engine room CAE model is necessary and the Computational Fluid Dynamics (CFD) and Computational Aero Acoustics (CAA) synthesis method is the proper way.

The fan shroud is the most sensitive area which significantly affects the BPF noise. Based on the CAA analysis, the solutions are drawn, which the shroud wall is modified in some of the circles.

The CAA analysis results shows that the noise is reduced 5dB (A).

The experiments results show that the BPF noise in the EV is cut down in 5-8 dB (A) too. Experimental and CAA synthesis method is useful to solve the cooling FAN BPF noise issue.

The vertex of the flow will cause some special noise while the fan running in different loadings. It will be validated and researched in vehicle in followings.

And there are many other factors which can influences the r the cooling FAN noise and its distribution. Such as the uneven noise in left and right side in passenger cabin. The deformation of the fan while it is running will effect the distance between the fan and shroud. This additional case will be carried out in future.

References

[1] Quinlan D. A., Bent P. H.: High Frequency Noise Generation in Small Axial Flow Fans, Journal of Sound and Vibration, No. 02, 1998, 177-204.

[2] Bianchi S., Corsini A., and Sheard A. G.: Experimental Characterisation of the Far-Field Noise in Axial Fans Fitted with Shaped Tip End-Plates, ISRN Mechanical Engineering, Vol. 102, 2012, 01-09.

[3] Ouyang. H., Li Y., Du Z. H., et al.: Experimental Study on Aerodynamic and Aero Acoustic Performance of Low Blades, Journal of Aerospace Power, vol. 21, No. 04, 2006, 668-674.

[4] Jin G. Y., Ouyang. H., et al.: Research of Aerodynamic Noise Source in Tip Region of Axial Fans with Circumferential Skewed Blades at Off-Design Conditions, Journal of Shanghai Jiaotong University, vol. 45, No. 06, 2011, 345-349.

[5] Sutliff D. L., Tweedt D. L., and Fite B. E.: Low Speed Fan Noise Reduction with Trailing Edge Blowing, Glenn Research Center, Cleverland, Ohio, 2002.

[6] Woodward R. P., Fite E. B., and Podboy G. G.: Noise Benefits of Rotor Trailing Edge Blowing for a Model Turbofan, Glenn Research Center, Cleverland, Ohio, 2007.

[7] Avinash, D., Shankar, M., Maller, R., and Ravindran, V., “Reduction of Aero-Acoustics Tonal Noise for a Tractor Cooling Fan,” SAE Technical Paper 2021-26-0299, 2021, doi: 10.4271/2021-26-0299.

[8] Teymourpour, Sh., Mahdavi-Vala, A., Yadegari, M., Kia, S. et al., “Engineering Approach for Noise Reduction for Automotive Radiator Cooling Fan: A Case Study,” SAE Technical Paper 2020-01-5085, 2020, doi: 10.4271/2020-01-5085.

[9] Karlsson, M. and Etemad, S., “Installation Effects on the Flow Generated Noise from Automotive Electrical Cooling Fans,” SAE Technical Paper 2020-01-1516, 2020, doi: 10.4271/2020-01-1516.

[10] Fares, O., Weng, C., Zackrisson, L., Yao, H. et al., “Numerical Investigation of Narrow-Band Noise Generation by Automotive Cooling Fans,” SAE Technical Paper 2020-01-1513, 2020, doi: 10.4271/2020-01-1513.

[11] Tare, K., Mukherjee, U., and Vaidya, R., "Design Optimization of Automotive Radiator Cooling Module Fan of Passenger Vehicle for Effective Noise Management Using CFD Technique," SAE Technical Paper 2017-26-0183, 2017, doi: 10.4271/2017-26- 0183.

[12] Kuthada, T., Wittmeier, F., Bock, B., Schoenleber, C. et al., "The Effects of Cooling Air on the Flow Field around a Vehicle," SAE Int. J. Passeng. Cars - Mech. Syst. 9 (2): 2016, doi: 10.4271/2016-01-1603.

[13] Karim, A., Mehravaran, M., Lizotte, B., Miazgowicz, K. et al., "Computational Aero-Acoustics Simulation of Automotive Radiator Fan Noise," SAE Int. J. Engines 8 (4): 2015, doi: 10.4271/2015-01-1657.

[14] Yoshida, K., Semura, J., Kohri, I., and Kato, Y., "Reduction of the BPF Noise Radiated from an Engine Cooling Fan," SAE Technical Paper 2014-01-0631, 2014, doi: 10.4271/2014-01-0631.

[15] Ffowcs Williams, J. E. and Hawkings, D. L., “Sound Generation by Turbulence and Surfaces in Arbitrary Motion,” Philosophical Transactions of the Royal Society (A) 264 (1969): 321-342.