Simulation and Analysis of Noise from Acoustic Hood of Micro Electric Vehicle Range Extender*

Chun Yuan¹, Haowen Yuan², Hongchun Su³,*, and Long Li³

¹School of Intelligent Manufacturing and Automobile, Chongqing Vocational College of Transportation Chongqing, China
²College of Electrical Engineering, Sichuan University Chengdu, China
³Correspondence Sergeant School Army Engineering University Chongqing, China

*Corresponding author e-mail: shc6500@126.com

Abstract. Aiming at the noise control requirements of micro electric vehicle(EV) range extender, the author designed an acoustic hood and base it to build a 3D model and make a modal analysis and acoustic simulation to analyze the effect of noise control. Verifying the design of acoustic hood is reasonable. At the same time, it's one of the important reasons the natural frequency of the hood results in the difficulty of noise control which can supply reference for improving design of acoustic hood of range extend.

1. Introduction
Electric vehicles have the characteristics of high energy utilization, low-speed acceleration performance and low working noise. However, the battery capacity is the major constraint on extending the mileage of a micro electric vehicles [1]. Increasing the battery capacity can increase the mileage, but this will increase the weight and volume, reduce the driving efficiency and increase the difficulty of the vehicle design. To resolve this contradiction, range extenders came into being [2]. A range extender is actually a generator set that works when the energy of the battery drops quickly or when the amount of electricity falls below a certain value, supplements the battery or directly supplies power to the drive motor of the EV. However, compared with the pure electric working, the range extender generates more noise, which affects the comfort of the occupants [3]. To this end, this article describes the acoustic hood and related modeling and simulation experiments for a micro EV range extender [4].

2. Modelling
2.1. Geometric models
Extender acoustic hood simplified model shown in Fig. 1. The range extender includes a gasoline generator set with a variable frequency controller the power part is a four-stroke water-cooled motorcycle engine. The carburetor, most of the cooling water tank and the exhaust muffler exhaust pipe is placed outside the acoustic hood, in which there are the gasoline engine, generator and controller. Acoustic hood uses thin-walled steel material to reduce weight as much as possible. The connecting parts of air inlet folding plate, air outlet folding plate, inclined deflectors, the upper and lower plates and thin-walled acoustic hood are made of elastic elements to support and seal.
2.2. Mathematical models

According to the Law of quality, we can use formula (1) to calculate the sound insulation of a single-layer board when the sound waves are incident vertically.

\[ TL_0 = 10 \log[1 + \left(\frac{\omega m}{2pc}\right)^2] = 20 \log mf - 42.5 \]  

In which, \(\omega\) is the circular frequency, \(m\) is the area density (kg/m\(^2\)), \(\rho\) is the air density (1.225 kg/m\(^3\)), \(c\) is the speed of sound (340 m/s), \(f\) is the frequency of the sound source (Hz).

When the sound wave is incident on the wall obliquely at an angle, the formula for calculating the sound insulation is:

\[ TL_\theta = 10 \log[1 + \left(\frac{\omega m \cos \theta}{2pc}\right)^2] \]  

In practical engineering applications, the actual sound insulation is smaller than the theoretical value affected by many factors, and here is an empirical formula [5]:

\[ TL = 14.5 \log m + 14.5 \log f - 26 \]  

Before the calculation of sound insulation using acoustic boundary method, the structural modal frequencies of each order of the sound insulation hood must be obtained by the following formula:

\[ f_{mn} = \frac{\pi}{2} \sqrt{\frac{D}{\rho h}} \left[\left(\frac{m}{a}\right)^2 + \left(\frac{n}{b}\right)^2\right] \]  

Where \(a\) is the length of the simply supported plate \((m)\), \(b\) is the width of the simply supported plate \((m)\), \(m\) and \(n\) are integers greater than zero, \(\rho\) is the density of the plate \((kg/m^3)\), \(h\) is the thickness of the plate \((m)\), \(D\) is the bending stiffness of the plate, which can be calculated by formula (5):

\[ D = \frac{E h^3}{12(1-\mu^2)} \]
In which $E$ is Young's modulus and $\mu$ is Poisson's ratio.

Structural modes respond differently to the sound field. Whether the structure and sound should be coupled according to the coupling coefficient $\lambda$, if $\lambda$ is greater than 1, coupling is considered, instead, there is no need to consider coupling.

$$
\lambda = \frac{\rho c}{\rho T \omega}
$$

Where $\rho_0$ is the density of air (kg/m$^3$), $c$ is the speed of sound (340 m/s), $\rho_s$ is the structural density (kg/m$^3$), $T$ is the equivalent thickness of the structure (m), $\omega$ is the sound frequency (rad/s).

3. Calculation and discussion

3.1. Assumptions

For purposes of noise calculation and analysis of the acoustic hood, we made following assumptions:

- Each baffle of the acoustic hood and the deflector connected rigidly;
- There is no sound absorbing material on the surface of the acoustic hood, and the acoustic hood is only a thin-walled steel plate.
- The air density in the acoustic hood is constant, and the air velocity is far less than the wind speed, which is regarded as the situation where the air is stationary [6].
- The noise of the gasoline engine is much higher than that of the generator in the acoustic hood [7].

To this end, we set a sound source near the cylinder block of the gasoline engine to simulate the noise in the acoustic hood, and the sound pressure is 96dB (A), as shown in Fig. 2.

![Fig 2. Sound source location.](a) Main section view  (b) Left section view)

3.2. Analysis conditions

Use the Hyper mesh software to mesh the acoustic hood to obtain the volume mesh and the surface mesh. Select four sets of fulcrum points at the bottom of the acoustic hood to connect to the frame, as shown in Fig. 3.
Fig 3. Restraint at the bottom of acoustic hood.

The material of the acoustic hood is steel and isotropic, and the elastic modulus is $2 \times 10^5$, Poisson's ratio is 0.3, the density is $7.9 \times 10^3 \text{ t/mm}^3$, the modal frequency band of the structure is generally selected to be twice the acoustic response frequency band. Here we choose 0-4000Hz, which means the acoustic frequency band for 0-2000Hz.

As shown in Fig. 4, we set the ground reflection surface (green) and field point grid (yellow), and 0.5m away from the air outlet side of the acoustic hood is the distance from the extender to the cockpit of the micro electric vehicle.

Fig 4. Field grid and reflecting surface.

3.3. Modal analysis and acoustic simulation results
After completing the preprocessing of the mesh and parameters, import the model into Nastran for constrained modal analysis. TABLE 1 shows the first 20 natural frequencies.
### Table 1. The first 20 natural frequencies of acoustic hood and range extender.

| Order | Frequency (Hz) | Order | Frequency (Hz) |
|-------|----------------|-------|----------------|
| 1     | 343.5002       | 11    | 957.6468       |
| 2     | 376.2696       | 12    | 1038.193       |
| 3     | 468.1217       | 13    | 1136.883       |
| 4     | 650.6962       | 14    | 1152.962       |
| 5     | 707.9203       | 15    | 1430.745       |
| 6     | 726.7023       | 16    | 1594.551       |
| 7     | 773.5751       | 17    | 1636.495       |
| 8     | 856.7505       | 18    | 1797.336       |
| 9     | 887.5551       | 19    | 1854.227       |
| 10    | 900.4168       | 20    | 1901.508       |

![Air outlet](image1)

![Air inlet](image2)
In Fig. 5 we find that the first three orders generator set itself has obvious mode shapes, and from fourth-to-sixth order exhaust air flap modes have obvious mode shapes.

The modal analysis data, surface mesh modeling and field point mesh were imported into Virtual Lab, and acoustic boundary element method was used for noise simulation analysis [8]. The acoustic hood scheme mainly examines the influence of the operating noise of the extender on one side in the cockpit. Therefore, we focus on observing the sound pressure cloud map of field points grid near the cockpit side [9]. The results are shown in Fig. 6.
(b) 800Hz sound pressure cloud chart

(c) 1200Hz sound pressure cloud chart

(d) 1600Hz sound pressure cloud chart
In order to visually observe the sound pressure spectrum near the cockpit side, we select three measurement points on the grid of field points near the cockpit side, and the positions of the points are shown in Fig. 7, obtaining the noise spectrum of three measuring points, as shown in Fig. 8.

Fig 6. Acoustic pressure nephogram of field point grid.

Fig 7. Noise measuring point location.
(a) Sound pressure spectrum of noise measuring point

(b) Peak value of sound pressure at point 1
From the analysis of the noise at the three measuring points in Fig. 8, we can see that when the maximum sound pressure values of measuring point 1 and 2 are 78.6dB(A) and 76.8dB(A) respectively, the noise spectrum of measuring point 1 and 2 demonstrate distinct peaks. The peak values of sound pressure at point 2 and 3 are shown in Fig. 8(c) and 8(d) respectively.
when they appear at 656.8 Hz; the maximum sound pressure measuring point 3 is 74.4 dB(A) at 521 Hz; the root mean square values of the sound pressures at points 1, 2, and 3 at 0-2000 Hz are 67.5 dB(A), 67.44 dB(A), and 65.61 dB(A), respectively. Therefore, it can be considered that the designed acoustic hood has a better noise control effect.

4. Conclusions
In this paper, through structural modal analysis and acoustic simulation of the acoustic hood, it is verified theoretically that the acoustic hood structure has a good noise control effect. At the same time, we found the shortcomings. The following conclusions are obtained:

1. The acoustic simulation was successfully implemented by using Virtual Lab, and the intuitive results of noise control were obtained from the sound pressure cloud diagram and the spectrum diagram, which provided a useful reference for the optimization and improvement of the acoustic enclosure structure.

2. The 4th order natural frequency in Tab. 1 is 650.69 Hz, and the highest sound pressure values of measurement point 1 and 2 are both 656.8 Hz, and the 4th order in Fig. 5 is at the exhaust flap obvious. It can be considered that 650.69 Hz is the focus of the improved design of the sound insulation structure.

3. The root-mean-square values of the sound pressures at measurement points 1, 2, and 3 at 0-2000 Hz are 67.5 dB(A), 67.44 dB(A), and 65.61 dB(A), respectively, and the range during the transmission of vehicle noise to the cockpit, it is necessary to penetrate the cockpit's baffle, which will reduce the noise. Therefore, the acoustic hood has better noise control effect.

Acknowledgment
We gratefully acknowledge the kind cooperation of Dr. Zhebin Zhang in the preparation of this Application note. This work is supported by the Science and Technology Research Program of Chongqing Municipal Education Commission (Grant No. KJZD-K201805701)

References
[1] Wang Xiaole. The Study on Vibration and Noise Analysis and Control of Range-Extender [D]. Shanghai: Shanghai University of Engineering Science, 2014.
[2] Milio F, Piu A. Analysis of a HT-PEMFC range extender for a light duty full electric vehicle(LD-FEV)[J].International journal of hydrogen energy, 2016, 41: 37-50.
[3] Zhang Shufeng. Noise Analysis of Medium-Power Silent Type Diesel Generator Set and Optimization of Its Acoustic Enclosure [D]. Chengdu: University of Electronic Science and Technology of China, 2015.
[4] Morandin M, Bolognani S, Pevere A. Active Torque Ripple Damping in Direct Drive Range Extender Applications: a Comparison and an Original Proposal[C]. IEEE Vehicle Power and Propulsion Conference, 2015.
[5] Song W, Zhang X, Tian Y. A charging management-based intelligent control strategy for extended-range electric vehicles[J]. Journal of Zhejiang University-SCIENCE A, 2016, 17(11): 903-910.
[6] Wang D, Song CX, Song SX. Parameter Matching and Performance Simulation for a Distributed Power Extended-Range Electric Vehicle[J]. Applied Mechanics and Materials, 2014, 49(10): 436-447.
[7] C. Q. Ren, D. H. Cai, J. P. Liu and M. M. Fan, “Experimental and numerical study of the cooling performance of automobile engine cabin”, Journal of HuNan University (natural science), Vol. 39, pp. 37-41, Apr. 2012.
[8] Li Na. Study on Sound Insulation Characteristic and Structure Optimization of Portable Generator Sets Enclosure [D]. Tianjin: Tianjin University, 2013.
[9] Hogblom O, Andersson H. A simulation framework for prediction of thermoelectric generator system performance[J]. APPLIED ENERGY, 2016, 180: 472-482.