Research on Noise of Tractor Cab Based on FE-SEA Method

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Abstract. The mid-frequency noise in a tractor cab is predicted, the FE-SEA hybrid model of tractor cab is established, the cab structure modal density, internal loss factor and coupling loss factor are obtained by theoretical calculation and test method, and the finite element-statistical energy analysis is simulated after loading vibration and noise excitation. Comparing the cab acoustic pressure level obtained by the simulation with the measured data, the analysis and comparison show that the result of the model using FE-SEA hybrid method in the mid-band is highly fitting with the test value, and the prediction absolute error is less than 2dB, which can be used for further analysis.

1. Introduction
The dynamic behavior of the frequency band in the structure of the tractor cab combines the characteristics of high frequency and low frequency.

If the dynamics of the high frequency and low frequency are separately modeled and then coupled, the finite element method and the statistical energy analysis method can be compensated. Limitations of band analysis. The finite element-statistical energy analysis hybrid method was proposed and improved by Langley [1-4] and has a wide range of applications in various fields [5-6], but the research on the tractor cab has not been published yet. Methods the tractor cab beam structure subsystem was established. The cab wall structure subsystem was established by statistical energy analysis method, and then connected correctly. The hybrid finite element statistical energy analysis model of the tractor cab was obtained and then combined simulation.

2. Tractor cab FE-SEA hybrid model establishment
Firstly, the geometric model of the tractor cab is established. In order to facilitate the subsequent processing, all the beam structures are simplified into hollow square tube beams of a certain thickness during the process of building the cab model, and the small ribs and curved structures are removed, and all will be removed. The connection between the beams is defined as a common node connection. Simplify the smaller ribs and excessive rounding, and seal all the smaller holes to facilitate the establishment of the later sound cavity, simplifying the glass and cover plates to a solid structure of a certain thickness. Since the door is closed, the door glass is the connection between the beams and between the plates and the beams is also defined as a common node connection. The model is mainly divided into three parts: beam structure, glass plate surface, and metal plate surface.

The built-in tractor cab geometry model is imported into Hypermesh for extraction mid-surface processing, and the extracted mid-surface is geometrically cleaned, meshed, and finite element mesh is
divided into quadrilateral meshes with a size of 10mm and localized by triangular meshes. The total number of nodes is 168,585, and the total number of units is 171,082, as shown in Figure 2.

![Figure 1. Geometric model of tractor cab.](image1)

![Figure 2. Finite element model of tractor cab.](image2)

After the establishment of the finite element model of the tractor cab, the FE-SEA hybrid model is established, and the finite element model is introduced into the VA ONE to divide the subsystems. The thin plate parts with higher modal density, such as the front windshield and the left and right fenders, divided into the SEA system, the beam structure with lower modal density is divided into FE system, then the cab SEA cavity is established, and then the frame beam structure, the SEA plate structure and the SEA cavity structure are coupled to ensure that the energy can be correctly followed. The subsystem is passed.
3. Parameter calculation of mixed model

3.1. Modal density

The SEA subsystem in the hybrid FE-SEA model is simplified to a two-dimensional plate to calculate its modal density. Assuming the density $\rho$, elastic modulus $E$, Poisson's ratio $\mu$, and thickness of the two-dimensional flat material, the thickness of the plate is $h$. The modal density calculation formula of the two-dimensional plate vibration system in the principle of statistical energy analysis is [7].

$$n(f) = \frac{A}{2RC_1}$$  \hspace{1cm} (1)

Where $f$ is the 1/3 octave center frequency; $A$ is the area of the plate; $R$ is the radius of curvature of the bend, and $R=1/(2\sqrt{3})$, which is the thickness of the plate; $C_1$ is the two-dimensional longitudinal wave velocity of the plate.
\[ C_1 = \sqrt{E / (\rho(1-\mu_2))} \]  

(2)

For the convenience of calculation, the SEA subsystems such as the front windshield, the left and right door glass, the roof panel, the left and right fenders and the bottom plate are directly calculated by the VA ONE software, and the results are shown in Table 1.

The modal density is the number of modalities within the unit band width, and the modal density is expressed as

\[ n(f) = m / (f_u - f_l) \]  

(3)

Where \( m \) is the number of modalities in the band; \( f_u \) and \( f_l \) the upper and lower frequencies of the 1/3 octave center frequency, respectively.

### Table 1. Subsystem modal density.

| Subsystem                  | Original structure | Simplified structure | Modal density |
|----------------------------|-------------------|---------------------|--------------|
| Front windshield           | Curved sheet      | Two-dimensional tablet | 0.335129     |
| Right glass door           | Curved sheet      | Two-dimensional tablet | 0.219712     |
| Right rear window glass    | Curved sheet      | Two-dimensional tablet | 0.133876     |
| Rear window glass          | Curved sheet      | Two-dimensional tablet | 0.133553     |
| Right side panel           | Curved sheet      | Two-dimensional tablet | 0.209192     |
| Bottom plate               | Irregular sheet   | Two-dimensional tablet | 0.481672     |
| Bottom plate               | Irregular sheet   | Two-dimensional tablet | 0.541242     |

#### 3.2. Internal loss factor

The internal loss factor \( \eta_i \) of the structural subsystem \( i \) is calculated by the theoretical formula [8].

\[ \eta_i = \eta_{i_0} + \eta_v + \eta_{ib} \]  

(4)

Where \( \eta_{i_0} \) is the structural loss factor formed by the internal friction of the structural subsystem itself; \( \eta_v \) is the loss factor formed by the vibration attenuation of the structural subsystem vibration; \( \eta_{ib} \) is the loss factor formed by the boundary connection. The structural loss factor is partially calculated in VA ONE as shown in Figure 5.

The loss factor in the acoustic cavity can be obtained by calculating the average sound absorption coefficient of the sound field. The theoretical formula is

\[ \eta = \frac{2.2}{T_{60}f} = \frac{13.82}{T_{60}\omega} \]  

(5)

\( T_{60} \) is the reverberation time, which refers to the time required for the attenuation of the acoustic energy level in the acoustic cavity by 60dB(A). There is an empirical relationship between the average sound absorption coefficient and the reverberation time \( T_{60} \).

Where \( V \) is the volume of the acoustic cavity; \( S \) is the total sound absorption surface area of the acoustic cavity; \( C_0 \) is the propagation velocity of the sound.

The more complex acoustic cavity system can be determined by the test method. The data measured by the test in [10] is used in this paper. The acoustic cavity loss factor of the cab is shown in Fig. 6.
3.3. Coupling loss factor

There are three types of connections between subsystems: point connections, line connections, and area connections. General calculation of line connections and face links when calculating coupled connection loss calculations.

The theoretical calculation formula for the line connection coupling loss factor is:

\[
T_{60} = \frac{60V}{1.086C_o S_a} = \frac{0.161V}{S_a}
\]  

(6)

There is only one sound cavity subsystem in this paper, so only the surface connection between the structure and the sound cavity is calculated. The theoretical formula is

\[
\eta_y = \frac{l_{AB} c_g}{\pi \omega A_i} \tau_y
\]

(7)

Where \(l_{AB}\) is the length of the connection curve, \(c_g\) is the group velocity of subsystem \(i\), \(\omega\) is the circular frequency, and \(A_i\) is the area of subsystem \(i\). \(\tau_y\) is the direction of the incident wave.

There is only one sound cavity subsystem in this paper, so only the surface connection between the structure and the sound cavity is calculated. The theoretical formula is
Where $\eta_{sc}$ is the coupling loss factor of the structure to the acoustic cavity, $\eta_{cs}$ is the coupling loss factor of the acoustic cavity to the structure, $\rho_0$ is the acoustic cavity density, $\rho_s$ is the structural density, $\eta_s$, $\eta_c$ are the modal density of the structure and the acoustic cavity, respectively, and $\sigma$ is the acoustic emission coefficient. Figure 7 shows the coupling loss factor between some subsystems.

### 3.4. Radiation efficiency

The vibration noise excitation radiates energy to the cab cavity through the body structure, and the radiation capability is described by the radiation efficiency. The radiation efficiency calculation formula [9] is

$$\sigma = \frac{P_w}{\rho c A v^2} \quad (10)$$

Where $P_w$ is the power radiated by the plate to the acoustic cavity; $\rho$ is the air mass density; $c$ is the sound velocity; $A$ is the coupling area of the structure and the acoustic cavity; and $v$ is the root mean
square value of the vibration velocity of the structural plate. Figure 8 shows the calculated radiation efficiency of the partial mechanism plate.

4. Vibration noise excitation determination

4.1. Vibration excitation
The engine and road vibration excitation are transmitted through the four suspension points of the cab (Fig. 9). The vibration excitation can be measured by a three-way vibration accelerometer placed on four suspension points. This paper uses the test [10]. Data, part of the data is shown in Figure 10. The x direction is the front and rear of the cab, the y direction is the left and right direction of the tractor, and the z is vertical.

![Figure 9. Vibration excitation loading position.](image)

![Figure 10. Right front suspension point vibration acceleration spectrum.](image)

4.2. Cab wall acoustic excitation
By measuring multiple points on each side of each wall subsystem, and then taking the average value as the acoustic excitation load of the surface, the data obtained from the literature [10] is used in this paper. Figure 11 shows the noise sound pressure level at 15cm from the front windshield.
5. Model simulation verification analysis

5.1. Tractor cab noise prediction and simulation verification

The modal density, internal loss factor, coupling loss factor, vibration excitation spectrum and noise excitation spectrum obtained by theoretical calculation and experiment are loaded into the tractor cab FE-SEA hybrid model, and the school packaging process is performed to obtain the rated speed of Fig. 12. The sound pressure level of the acoustic cavity noise of the tractor cab is compared with the test results.

![Figure 11. Noise level of windscreen.](image)

![Figure 12. Bridge noise simulation calculation and experimental comparison.](image)

It can be concluded from Fig. 12 that the FE-SEA hybrid model of the tractor cab has a large difference between the simulation results of the frequency band below 200hz and the frequency band above 1000hz, and the simulation results in the frequency range of 200-1000hz are in good agreement with the experimental results. The main reason is that the modal density is low in the low frequency band, and the modal modes do not overlap each other. The finite element method can better describe the response between subsystems, and the modal overlap phenomenon in the high frequency band is serious. The response is uncertain, and the average energy of space and frequency is used to describe it more accurately, so statistical energy analysis can more accurately describe the subsystem response at this time. The mid-band has the characteristics of global determinism and local uncertainty formed by the superposition of low frequency and high frequency. The FE-SEA hybrid method combines the characteristics of low-frequency and high-frequency research methods to better analyze and study in the middle frequency band. The above results show that the sound pressure level of the hybrid tractor
cab model constructed by FE-SEA hybrid method is consistent with the test results, which can correctly predict the noise pressure level of the tractor driving.

5.2. Cab cavity sound energy input analysis
Figure 13-14 shows the four subsystems with the largest total energy output. The sum of the contributions of the left and right side plates accounts for about 41.3% of the total energy input, followed by the left and right side plates and the windshield.

![Energy input](image1)

**Figure 13.** Energy input.

![Energy input ratio of each subsystem](image2)

**Figure 14.** Energy input ratio of each subsystem.

6. Conclusion
Using the finite element method to establish the tractor driving frame structure subsystem, the statistical energy analysis method to establish the cab panel subsystem, and correctly coupling the FE-SEA hybrid cab model established by each subsystem to compare the single finite element method. The established model has the advantages of fast modeling speed and short analysis time. Compared with statistical energy analysis method, FE-SEA hybrid method broadens its application frequency band and improves the accuracy of noise prediction. The hybrid modeling method is compared with the traditional modeling method. The use of the mid-band has a high analytical accuracy and is an effective mid-band noise research and analysis method.

Through the analysis of the noise input energy of the cab, it can be concluded that the left and right side panels have the greatest influence on the cab sound cavity, and the acoustic package improvement can be started from here.

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