Study on FE-SEA Modeling and Acoustic Performance of Heavy Duty Commercial Vehicle Based on Experimental Statistical Energy Parameters

Jintao Su 1,2 and Zhaoxiang Deng 1,*

1 Institute of Machinery and Transport, Chongqing University, Chongqing 401122, China; nvh2012@163.com
2 Institute of Automotive Studies, Jilin University, Changchun 130012, China
* Correspondence: zxd@163.com

Abstract: Due to the difficulty of obtaining statistical energy parameters of complex structures and the complexity of modeling connection and model verification, the hybrid FE-SEA model has many problems in modeling complex structures. Therefore, in order to solve the above problems, this paper provides a reference for the application of the hybrid FE-SEA model in complex structures. In this paper, the hybrid FE-SEA commercial vehicle model is established by an experimental statistical energy parameter modeling method and a modification method. The model division and subsystem connection modeling of a complex substructure of a heavy vehicle cab are studied. In the hybrid model, the hybrid line connection and the hybrid point connection are established. On this basis, the parameters of the cab model were studied, and the statistical energy parameters such as modal density, internal loss factor, and coupling loss factor were obtained by the experimental method. The statistical energy parameters of the cab acoustic model are modified. Finally, the accuracy of the model is verified by vehicle test. In addition, the acoustic performance of the cab was optimized, and airtightness and acoustic packaging were verified. The full parameter modeling and correction method is adopted in this paper, which is an effective supplement to the traditional statistical energy parameter modeling method.

Keywords: FE-SEA; statistical energy parameters; test statistical energy

1. Introduction

In the late 1990s, Langley and Bremner proposed a hybrid statistical energy model (FE-SEA) combined with the finite element method [1]. In this method, the local finite element model is used as a grid model instead of some statistical energy plates. It is suitable for a plate with a small area and complex and changeable surface shape, which improves the calculation accuracy of low and middle frequency. The boundary region connection form of finite element subsystem and statistical energy subsystem is the key to the successful application of this hybrid method in the engineering field. This is because it involves the conversion of two noise calculation methods in the whole calculation process [2]. Important parameters such as transfer loss, vibration energy, and sound energy are obtained by the traditional finite element method, and the traditional statistical energy analysis method and hybrid method are compared and analyzed. It is found that the accuracy of the hybrid method is higher than that of the other two methods in calculating the intermediate frequency noise. This method can be embedded in the Monte Carlo algorithm to realize acoustic calculation with uncertain structural parameters [3]. A lot of work has been done on the structure and parameters of hybrid models. Langley and Cicirello [2,4] derived the hybrid connection equation of the FE plate subsystem and cavity system, which solved the coupling transfer problem between structure and cavity. FE-SEA hybrid method can be used to solve the response trend of the system in the case of subsystem uncertainty. In this kind of problem, there are two aspects of uncertainty, one is
non-parametric uncertainty (SEA subsystem) and the other is parametric uncertainty (FE subsystem). However, the algorithm of the proposed method requires a large amount of calculation. Some literature [5] uses the first-order reliability method and Laplace method to solve the response trend of the system. The above research has made a preliminary exploration of the FE-SEA principle, but the processing method of the calculation results is still relatively simple, only considering the energy transfer relationship between adjacent subsystems. In the application of hybrid models, Chadwyck T. Musser et al. [3] established an FE-SEA hybrid model including the whole interior of the car, and premeasured the noise inside the car with high precision. Jerome E.Manning [4] established a vehicle hybrid model through experiments and simulations. Steven G Mattson [5] used FE-SEA to predict the insertion loss of the muffler. Phil Shorter et al. [6] predicted and diagnosed the acoustic transmission loss of the front bulkhead by using the hybrid FE-SEA method. Literature [7] studied the contribution of heavy commercial vehicle cab noise based on the method of the hybrid model. A hybrid finite element model of heavy commercial vehicle cabs was established. The hybrid models of commercial vehicles include structural and acoustic models as shown in Figure 1. The deterministic method is the core of the hybrid model. The deterministic method in the hybrid FE-SEA adopts the traditional finite element method. The hybrid FE-SEA method also has some disadvantages of the traditional finite element method. Based on the basic idea of the hybrid FE-SEA model, scholars have proposed the hybrid WB-SEA method, the hybrid spatial waveform unwrapping -SEA method, and the hybrid FE-EFE method. This is to improve the computational efficiency of specific problems. Cicirello and Langley [8] extended the research method of the hybrid FE-SEA model. In this extended method, two kinds of parametric stochastic models, the probabilistic stochastic model, and the interval distributed nonprobabilistic stochastic model, are introduced to the stochastic subsystem, and the hybrid solution mode of parametric stochastic subsystem and nonparametric stochastic subsystem is established, and the hybrid FE-SEA method is extended. Monte Carlo simulation is used to deal with the stochastic subsystem of parameters, and the computational cost increases sharply with the increase of uncertain parameters. Some literature [7,9] used the first-order reliability method to study the hybrid model based on average response theory, but this method ignored the probability distribution of the hybrid FE-SEA method itself. The Laplace method was used to improve the stability of the model, and the lognormal distribution was obtained by fitting the probability distribution of the hybrid FE-SEA by means of the mean and variance. Finally, the integral of the probability density was solved by the numerical method. Yin et al. [10,11] introduced interval parameter uncertainty into the hybrid FE-SEA method. A hybrid nonparametric-interval parameter uncertainty model is established for the acoustic vibration system. In order to improve the computational efficiency, a hybrid second-order interval perturbed finite element method (IFEA) and statistical energy analysis method were proposed. As the higher-order terms of the Taylor series are ignored, the above method is only applicable to narrow parameter ranges. Although the hybrid model theory has made some progress, there are still uncertainties in solving the complex structure model. For example, the calculation parameters of the hybrid model are difficult to obtain, and the parameters are uncertain. Therefore, the deviation and uncertainty of calculation parameters limit the accuracy of the hybrid model. The coupling loss factor, modal density, internal loss factor, and other parameters in complex structures need to be obtained and modified by means of an experimental statistical energy method. Therefore, it is very necessary to establish a hybrid model with the aid of the experimental parameters of complex structures. Based on this, the experimental statistical energy parameter method of the hybrid FE-SEA model is studied in this paper.
In this paper, the statistical energy experimental modification method is used to model the hybrid model, and statistical energy parameters such as modal density, internal loss factor, and coupling loss factor are obtained. At the same time, the acoustic material parameters of the cab are determined on the basis of experimental modification. The hybrid model of heavy cargo commercial vehicles based on experimental statistical energy parameters was studied, and the heavy cargo commercial vehicle model based on experimental statistical energy parameters was established. The accuracy of the model is verified by experiments. Finally, the acoustic performance of the cab was optimized based on the acoustic strain energy superposition method, acoustic package optimization method, and airtightness optimization method. The effectiveness of the optimization method is verified by experiments. In this paper, the full parameter modeling method is adopted to study on the basis of experimental correction. It is an effective supplement and extension to the existing statistical parameter modeling method. Compared with the existing references,
this paper adopts a new parameter evaluation system, namely the experimental mixed statistical energy method. The structural sound hybrid model of the cab is modeled by numerical simulation and experiment. The accuracy of the mixed model is guaranteed by updating the numerical model with experimental modification. A new parameter evaluation system was used to modify and evaluate the model. In the experimental modification method, modal density, coupling loss factor, and damping loss factor are evaluated and updated. Compared with the existing theoretical parameter estimation methods, the calculation accuracy is higher.

2. Basic Theoretical Parameters of FE-SEA Method

2.1. Modal Density

In the hybrid FE-SEA model, the three basic parameters required by the model are modal density, internal loss factor, and coupling loss factor between different subsystems. Within the research frequency range, the modal density is one of the important parameters of the subsystem, which represents the energy storage capacity of the subsystem.

2.1.1. Lateral Vibration Modal Density of One-Dimensional Beam

Modeling of one-dimensional elements is often encountered in hybrid models. In the process of modeling, the transverse vibration of a one-dimensional beam is applied more often, and its free state vibration equation is [8–12]:

\[
\frac{\partial^2 u(x, t)}{\partial t^2} + R^2 C_l^2 \frac{\partial^4 u(x, t)}{\partial t^4} = 0
\]

(1)

where, \( u(x, t) \) is the displacement of point \( x \), \( R \) is the radius of rotation at point \( x \), \( C_l \) is the longitudinal wave velocity, and:

\[
C_l = \sqrt{\frac{E}{\rho}}
\]

(2)

In the above formula, \( E \) is the elastic modulus of the material and \( \rho \) is the material density. In Equation (1), \( u(x, t) = \varphi(x)e^{i\omega t} \), then the equation can be rewritten as [9–13]:

\[
\frac{\partial^4 \varphi}{\partial x^4} - K_B^4 \varphi = 0
\]

(3)

In the above formula, \( K_B \) is the number of bending waves, and:

\[
K_B = \sqrt{\frac{\omega}{RC_l}} = \frac{\omega}{\sqrt{\omega RC_l}} = \frac{\omega}{C_B}
\]

(4)

In the above equation, \( C_B = \sqrt{\omega RC_l} \) is the bending wave velocity, then the modal density of the transverse vibrating beam is expressed as:

\[
n(\omega) = \frac{l}{2\pi C_B}
\]

(5)

\[
n(f) = 2\pi n(\omega) = \frac{l}{C_B}
\]

(6)

2.1.2. Modal Density of Two Dimensional Plates

Thin plate modeling is often used in hybrid model modeling. For a two-dimensional flat plate, the modal density is expressed as [10–14]:

\[
n(f) = \frac{A}{2RC_l}
\]

(7)
\[ C_l = \sqrt{\frac{E}{\rho(1 - \mu^2)}} \]  

In the above formula, \( A \) is the total area of the thin plate, \( R \) is the radius of bending rotation, \( C_l \) is the two-dimensional longitudinal wave number of the flat plate, \( E \) is the elastic modulus of the corresponding flat plate, \( \rho \) is the density of the corresponding material, and \( \mu \) is the Poisson’s ratio of the corresponding material. For curved plate, the empirical formula of modal density is as follows [10–14]:

1. When the system frequency ratio meets the condition \( f / f_r \leq 0.48 \):
   \[ n(f) = \frac{5A}{\pi h C_l} \left( \frac{f}{f_r} \right)^{1/2} \]  

2. When the system frequency ratio meets the condition \( 0.48 < f / f_r \leq 0.83 \):
   \[ n(f) = \frac{7.2A}{\pi h C_l} \left( \frac{f}{f_r} \right) \]  

3. When the system frequency ratio meets the condition \( f / f_r > 0.83 \):
   \[ n(f) = \frac{2A}{\pi h C_l} \left\{ 2 + 0.596 \frac{F}{F - 1} \left[ F \cos \left( \frac{1.745 f_r^2}{f^2} \right) - 1 \right] \cos \left( \frac{1.745 f_r^2 f^2}{f^2} \right) \right\} \]  

In Equations (9)–(11), \( F \) is the center frequency of 1/3 octave, \( f_r \) is the frequency, and \( F \) is the frequency factor. In 1/3 octave, \( F = 1.122 \), and:

\[ f_r = \frac{C_l}{2\pi r} \]  

\[ F = \left( \frac{f_{\text{upper}}}{f_{\text{lower}}} \right)^{1/2} \]  

In the above formula, \( f_{\text{upper}} \) represents the upper limit frequency of each frequency band of 1/3 octave, and \( f_{\text{lower}} \) is the lower limit frequency of each frequency band corresponding to it. According to the above equation, a common feature of the modal density of flat and curved plates can be obtained, that is, the modal density does not change with frequency.

2.1.3. Sound Modal Density

In the process of hybrid modeling, the sound cavity is an essential subsystem. The significance of the existence of the sound cavity lies in the surface coupling with a deterministic subsystem or random subsystem to form the necessary connection of the hybrid model. The mode number of the sound cavity system can be expressed by the wave number \( K_B \) of the bending wave in the sound field:

\[ N(K_B) = \frac{K_B^3 V_0}{6\pi^2} \]  

Then the modal density of the sound cavity can be obtained by the following formula:

\[ n(\omega) = n(K_B) \frac{dK_B}{d\omega} = \frac{\omega^2 V_0}{2\pi^2 c^3} \]
If the surface area of the acoustic cavity and the edge correction term is taken into consideration, the above equation can be rewritten as:

\[ n(\omega) = \frac{\omega^2 V_0}{2\pi^2 c^3} + \frac{\omega^2 A_s}{16\pi c^2} + \frac{\omega l_1}{16\pi c} \]  

(16)

In the above formula, \( V_0 \) is the volume of the sound field, \( c \) is the speed of sound, \( A_s \) is the surface area of the sound field, and \( l_1 \) is the total length of all edges of the sound field.

2.2. Internal Loss Factor

2.2.1. Internal Loss Factor of Structure

The internal loss factor of the structure is used to represent the ratio of the energy loss per unit time of each subsystem to the average energy within the unit frequency interval. In general, the internal loss factor of the structure consists of three basic components, as follows [12,13]:

\[ \eta_i = \eta_{is} + \eta_{ir} + \eta_{ib} \]  

(17)

where, \( \eta_{is} \) is the loss factor of the structure, which is generated by the internal friction of the structure, \( \eta_{ir} \) is the loss factor generated by the “acoustic-vibration radiation” of the structure, and \( \eta_{ib} \) is the loss factor generated by the damping at the boundary junction. \( \eta_{is} \) follows the change function of the material characteristics of the structure. \( \eta_{ir} \) is generated by the outward radiation of the vibration of the structural subsystem. The acoustic radiation loss factor in the lightweight plate contributes more to the internal loss factor of the plate. The acoustic radiation loss factor \( \eta_{ir} \) is a function of frequency \( f \), and the relationship is as follows:

\[ \eta_{ir}(f) = \frac{\rho_0 c c_{sa}}{2\pi f \rho_s} \]  

(18)

where \( \rho_0 \) is the air density, \( c \) is sound velocity, \( c_{sa} \) is the radiation ratio of the subsystem, \( \rho_s \) is the area mass density of the subsystem. The calculation formula is as follows:

\[ \frac{\partial^2 u(x, t)}{\partial t^2} + R^2 C_l \frac{\partial^4 u(x, t)}{\partial t^4} = 0 \]  

(19)

2.2.2. Internal Loss Factor of the Acoustic Cavity

The internal loss factor of the sound cavity can generally be obtained by measuring the average sound absorption coefficient of the sound field. For a closed cavity, the internal sound field is a reverberation sound field formed by wave reflection. The average absorption coefficient of the whole sound cavity can be calculated by the sound absorption coefficient on the surface of each sound cavity, and the expression is [10–13]:

\[ a_{avg} = \frac{S_1 a_1 + S_2 a_2 + \cdots + S_n a_n}{S_1 + S_2 + \cdots + S_n} \]  

(20)

where, \( S_n \) is the area of the surface \( N \) of the sound cavity, and \( a_n \) is the sound absorption coefficient of the surface \( N \). The sound absorption coefficient is expressed by reverberation time, and its empirical formula can be expressed as:

\[ T_{60} = \frac{60V}{1.086C_a S_{aT}} = \frac{0.161 V}{S_{aT}} \]  

(21)

The time required for attenuation of 60 dB (A) in the sound cavity can be obtained. By measuring the reverberation time, the sound absorption coefficient of the sound cavity can be obtained by the following formula:

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

(22)
2.3. Coupling Loss Factor

2.3.1. Coupling Loss Factor between Structures

The coupling loss factor between structures represents the strength of coupling between different subsystems and represents the energy loss generated when energy is transferred in each subsystem. All subsystems transfer energy through the coupling effect of the system. When one subsystem is excited by the external system, the energy of the subsystem is transferred to the other subsystem through the coupling effect, so as to achieve energy transfer and balance. The coupling effect of different subsystems mainly has the following methods: surface connection, point connection, and line connection. There is a line connection mode between different thin plates, and the coupling loss factor can be expressed as [10–14]:

$$\eta_{jk} = \frac{l c_w}{\pi \omega A_j} \tau_{jk}$$

(23)

where the $l$ is the plate connection between the curve length. $c_w$ represents the velocity of the bending wave of the excitation plate, $A_j$ represents the area of subsystem $j$, $\omega$ represents the center frequency of the research frequency band, and $\tau_{jk}$ represents the propagation coefficient of sound waves when they are transmitted from subsystem $j$ to subsystem $k$.

2.3.2. Coupling Loss Factor between Sound Cavity and Structure

When the system is coupled with the structure for the sound cavity, the coupling loss factor of the system can be expressed as:

$$\eta_{jk} = \frac{\rho_k C_k}{\omega \rho_j} \sigma_{jk}$$

(24)

where $\rho_k$ represents the mass density of the sound cavity subsystem, $C_k$ represents the sound velocity in the sound cavity, $\rho_j$ represents the mass density of the structural subsystem, and $\sigma_{jk}$ represents the sound radiation coefficient of the structural subsystem.

2.3.3. Coupling Loss Factor between Cavity and Cavity

If the coupling connection of the system exists between two cavities, there are two ways of general connection: one is the direct connection between the cavity and the cavity, and the other is the indirect connection between the cavity and the cavity. A direct connection means that there is no other partition between subsystem $j$ and subsystem $k$, and an indirect connection means that there is partition subsystem $p$ between subsystem $j$ and subsystem $k$. If there is a partition subsystem $P$ between the sound cavity subsystem $j$ and the sound cavity subsystem $K$, then the calculation formula of the coupling loss factor between them is as follows [11–14]:

$$\eta_{jk} = \frac{C_0 A_p}{4 \omega V_j} \tau_{jk}$$

(25)

where, $C_0$ is the speed of sound wave propagation in the sound cavity. $A_p$ is the coupling area at the junction, $V_j$ is the volume of the cavity subsystem $j$, $\omega$ is the circular frequency, and $\tau_{jk}$ is the transfer coefficient of the cavity subsystem $j$ and the cavity subsystem $k$. When the two subsystems are directly connected, then $\tau_{jk} = 1$ in the above formula.

2.4. Power Flow Balance Equation

The transfer relationship between the input power and output power of the hybrid FE-SEA model system is defined as follows [12–15]:

$$P_{in,dir}^{(m)} = P_{out,rev}^{(m)} + P_{diss,m}$$

(26)
where \( P_{\text{in,dir}}^{(m)} \) is defined as the power of the direct field received by the subsystem, \( P_{\text{out,rev}}^{(m)} \) is defined as the power lost by the boundary constraint of the subsystem in the direct field, and \( P_{\text{diss,}\text{m}} \) is the power lost inside the subsystem. The direct field power \( P_{\text{in,dir}}^{(m)} \), received by the subsystem is expressed as \([13–15]\):

\[
P_{\text{in,dir}}^{(m)} = P_{\text{in,0}}^{(m)} + \sum_{n} h_{nm} \frac{E_{n}}{n_{m}}
\]

(27)

where, \( P_{\text{in,0}}^{(m)} \) is defined as the power obtained by the subsystem under external excitation. \( \sum_{n} h_{nm} \frac{E_{n}}{n_{m}} \) represents the sum of the power output by the other subsystems to the current subsystem in the reverberation field and the power of the subsystem in its reverberation field. \( h_{nm} \) is the power transfer coefficient, namely the coupling loss factor between systems. \( \frac{E_{n}}{n_{m}} \) is the ratio of statistical subsystem energy to statistical subsystem modal density, which is called modal energy density. Therefore, the output power \( P_{\text{out,rev}}^{(m)} \) of the system can be expressed as:

\[
P_{\text{out,rev}}^{(m)} = \frac{E_{m}}{n_{m}} h_{\text{tot},m} = \frac{E_{m}}{n_{m}} \left( h_{m}^{a} + \sum_{n} h_{nm} \right)
\]

(28)

where \( h_{\text{tot},m} \) is the sum of the power output of the subsystem to the other subsystems. \( h_{m}^{a} \) is the power lost through the connection between subsystem and deterministic subsystem. According to Equations (27) and (28), the dissipated power \( P_{\text{diss,}\text{m}} \) of the system can be expressed as:

\[
P_{\text{diss,}\text{m}} = M_{m} \frac{E_{m}}{n_{m}}
\]

(29)

where \( M_{m} \) is the modal overlap factor of the statistical subsystem.

### 3. Full Parameter FE-SEA Model Modeling of Heavy Truck Commercial Vehicle

Commercial vehicle cab modeling and the connection handle uses the hybrid FE-SEA model calculation accuracy of key factors. These include the stiffness of the larger bridge skeleton structure, a megastructure with a thickness of more than 1 mm with FE modeling, and the sheet metal structure with a thickness of 1 mm or less with the SEA subsystem modeling. These are incorporated in the process of modeling simple local characteristics of the cab. The CAD model of the commercial vehicle cab is shown in Figure 2. The prepared quality cab is composed of the cab body in white, the cab interior decoration, the driver’s outdoor decoration, the cab electrical appliances, and other structures.

![Figure 2. CAD model of the commercial vehicle cab.](image-url)

#### 3.1. Subsystem Modeling and Partitioning

According to the commercial vehicle cab CAD model for the hybrid FE-SEA model, the first subsystem division is comprised of many parts: the bridge model has a bridge metal...
frame, beam, high rigidity structure that is divided into FE subsystems, such as sheet metal pieces that are less than 1 mm thick. The structure is divided into SEA subsystems, and the local irregular surface is divided into the FE subsystem, and aspects of the curved surface are divided into SEA subsystems. The hybrid FE-SEA model subsystem division results are shown in Table 1. The bridge structure was divided into 40 subsystems, including 28 SEA substructures and 12 FE substructures. The internal vocal cavity is divided into the chief driver’s head, chief driver’s waist and chief driver’s legs, and the co-driver’s head, chief driver’s waist, and chief driver’s legs. The external vocal cavity is composed of 10 evenly distributed cavities according to the external contour size of the cab.

| Sequence Number | Subsystem Name                  | Structure Attribute | Sequence Number | Subsystem Name                  | Structure Attribute |
|-----------------|---------------------------------|---------------------|-----------------|---------------------------------|---------------------|
| 1               | Sidewall inner plate left       | SEA                 | 21              | The floor in the right           | SEA                 |
| 2               | Side coaming inner plate right  | SEA                 | 22              | After the floor left             | SEA                 |
| 3               | Door inner plate left           | SEA                 | 23              | The floor right after            | SEA                 |
| 4               | Door inner panel right          | SEA                 | 24              | Skylight left                    | SEA                 |
| 5               | Door glass left                 | SEA                 | 25              | Skylight right                   | SEA                 |
| 6               | Door glass right                | SEA                 | 26              | Side girder left                 | FE                  |
| 7               | Front windshield                | SEA                 | 27              | Side girder right side           | FE                  |
| 8               | Car door plate left             | SEA                 | 28              | Side surround top beam left      | FE                  |
| 9               | Car floor board right           | SEA                 | 29              | Side surround top beam right     | FE                  |
| 10              | Inside board of storage box     | SEA                 | 30              | Rear floor rail left             | FE                  |
| 11              | Storage box outside plate       | SEA                 | 31              | Rear floor rail right            | FE                  |
| 12              | Interior ceiling left           | SEA                 | 32              | Front floor rail left            | FE                  |
| 13              | Interior ceiling right          | SEA                 | 33              | Front floor rail right           | FE                  |
| 14              | Coaming after right             | SEA                 | 34              | Front coaming top beam left      | FE                  |
| 15              | Right after the coaming         | SEA                 | 35              | Front coaming top beam right     | FE                  |
| 16              | Rear coaming glass to the left  | SEA                 | 36              | Pillar A outer plate left        | FE                  |
| 17              | Rear coaming glass right        | SEA                 | 37              | Pillar A outer plate right       | FE                  |
| 18              | Floor left before               | SEA                 | 38              | External spoke                   | SEA                 |
| 19              | Right before the floor          | SEA                 | 39              | Internal spoke                   | SEA                 |
| 20              | The floor in the left           | SEA                 | 40              | The door spoke                   | SEA                 |

### 3.2. Subsystem Connection Relationship

The connection relation of commercial vehicle cab includes point connection, line connection, surface connection, and the hybrid connection between three kinds of connection. In general, the subsystem connection relationship is modeled after the bridge FE subsystem and bridge SEA subsystem models are established. The correctness of the hybrid FE-SEA model connection relation affects the energy transfer of the whole system. Whether the hybrid FE-SEA model is effective depends on the connection relation of the hybrid model to a large extent. Combined with the hybrid FE-SEA model of commercial vehicle cab, the connection relation modeling is carried out.

Figures 3 and 4 are the connection schematic diagram of the hybrid FE-SEA model. SEA point connection exists in the node connection between different subsystems, hybrid point connection exists in the connection between some FE substructure nodes and SEA substructure, SEA line connection exists in different regions of the same subsystem, and also exists in the connection between artificially segmented SEA subsystems. Such as the line connection between the front coaming subsystem, the line connection between the cab roof subsystem, also includes the line connection between different components, such as the line connection between the door subsystem and the side enclosure subsystem, etc. The hybrid line connection exists at the interface of the FE subsystem and SEA subsystem, and also exists at the external boundary of the SEA subsystem. The SEA subsystem plane connection exists between the cavity subsystem and the SEA subsystem, and it belongs to the structure-cavity plane connection and the connection between different cavities of the SEA, and it belongs to the acousto-cavity connection. The hybrid plane connection exists between the FE substructure and SEA cavity subsystem, which belongs to the structure-cavity plane connection. Connection relation between subsystems through a theoretical method for correction of the third chapter, the basic method of correction for fixed point, mix line connection between the radiation radius of the shape function, line connection,
thus to correct the subsystem connection relations, and through the PIM power input method for the vibration of the subsystem energy, error test before and after modification. The connection model established by the hybrid FE-SEA model of commercial vehicle cab is shown in Tables 2 and 3. A total of 632 SEA point connections, 721 hybrid point connections, 715 SEA line connections, 724 hybrid line connections, 409 SEA surface connections, and 375 hybrid surface connections have been established.

Figure 3. Commercial vehicle cab connection model. (a) SEA point connection model; (b) SEA point-line connection model; (c) SEA line connection model; (d) FE wire connection model; (e) hybrid line connection model; (f) SEA surface connection and hybrid surface connection model.
Table 2. Part of the point connection model.

| Name-ID | Mass | Connections-1                  | Connections-2                  | Connections-3                  |
|---------|------|--------------------------------|--------------------------------|--------------------------------|
| Point-1 | None | Sidewall inner plate left      | Door inner plate left          | The small plate                |
| Point-2 | None | Side coaming inner plate right | Door inner panel right         | The floor right after           |
| Point-3 | None | Door inner plate left          | Car door plate left            | After the floor left            |
| Point-4 | None | Door inner panel right         | Car door board right           | Right before the floor          |
| Point-5 | None | Door glass left                | Car door plate left            | Skylight left                   |
| Point-6 | None | Door glass right               | Car door board right           | Skylight left                   |
| Point-7 | None | Front windshield               | Cowl panel                     | Skylight left                   |
| Point-8 | None | Car door plate left            | Door inner plate left          | Skylight left                   |
| Point-9 | None | Car door board right           | Door inner panel right         | Skylight left                   |
| Point-10| None | Inside board of storage box    | Storage box outside plate      | Front windshield                |
| Point-11| None | Storage box outside plate      | Interior ceiling right         | Skylight right                  |
| Point-12| None | Interior ceiling left          | Interior ceiling left          | After the round glass           |
| Point-13| None | Interior ceiling right         | Interior ceiling left          | After the round glass           |
| Point-14| None | The coaming after left         | Right after the coaming        | After the round glass           |
| Point-15| None | Right after the coaming        | The coaming after left         | After the round glass           |
| Point-16| None | Rear coaming glass to the left | The coaming after left         | After the round glass           |
| Point-17| None | Rear coaming glass right       | Right after the coaming        | Side of the top beam            |
| Point-18| None | Right before the floor         | The floor left before          | Side of the top beam            |

Table 3. Part of the hybrid wire connection model.

| Name-ID     | Mass | Connections-1                  | Connections-2                  | Connections-3                  |
|-------------|------|--------------------------------|--------------------------------|--------------------------------|
| Hybrid Line-1| None | Side enclosure outer plate left | Side girdor left                | Floor front rail left           |
| Hybrid Line-2| None | The floor left before          | Front middle floor left         | Floor front rail left           |
| Hybrid Line-3| None | The floor left before          | After the floor left            | Floor front rail right          |
| Hybrid Line-4| None | Right before the floor         | Front middle floor right        | Floor front rail right          |
| Hybrid Line-5| None | Right before the floor         | The floor right after           | Floor front rail right          |
| Hybrid Line-6| None | Side enclosure outer plate left| Side surround top beam left     | Side surround top beam right    |
| Hybrid Line-7| None | Side outer panel right         | Side surround top beam right    | Side surround top beam right    |
| Hybrid Line-8| None | The floor left before          | Before the floor left           | Before the floor left           |
| Hybrid Line-9| None | Right before the floor         | Before the floor right          | Before the floor right          |
| Hybrid Line-10| None | Front outer trim left          | Front windshield                | Front top girder left           |
| Hybrid Line-11| None | Front outer trim right         | Front windshield                | Front top girder top right      |
| Hybrid Line-12| None | Dash panel on the left         | Front top girder left           | Front top girder top right      |
| Hybrid Line-13| None | Dash panel right               | Front top girder top right      | Front top girder top right      |
| Hybrid Line-14| None | Dash panel on the left         | Front inner plate left          | Front inner plate left          |
| Hybrid Line-15| None | Dash panel right               | Front inner plate right         | Front inner plate right         |
| Hybrid Line-16| None | The floor left before          | Frame left                      | Door frame inner panel left     |
| Hybrid Line-17| None | The floor left before          | Frame to the right              | Door frame inside panel right   |
| Hybrid Line-18| None | The coaming after left         | Rear enclosure top beam left    | After the plate inside          |

3.3. Acquisition of Modal Density Modeling Parameters

Modal density generally refers to the number of modals in a unit frequency band of a structure within a certain frequency range. Methods to measure modal density include the sinusoidal scanning method, percussion method, random excitation method, etc. The admittance method is one of the most widely used methods to obtain modal density parameters of structures. Figure 5 to test the bridge side surround and floor, ceiling, and
acceleration sensor arrangement, structure modal density by impedance force hammerhead, parts of the test point for incentives, such as cab side surround and cabin floor, acceleration admittance method based on the parts of the modal density, the results are added to the cab hybrid FE-SEA model.

Figure 5. The layout of modal density measurement points. (a) Side measurement point of the cab; (b) Floor measurement point of the cab; (c) Modal test table of the cab; (d) Roof measurement point of the cab.

For flat plate structure, the real part of the spatial and frequency average input admittance \( Y \) is:

\[
< G > = \frac{1}{\Delta \omega} \int_{\omega_2}^{\omega_1} \text{Re}[Y] d\omega = \frac{n(f)}{4M} = \frac{\pi n(\omega)}{2M}
\]  (30)

Further sorting out Equation (30), we can get:

\[
n(\omega) = \frac{2M}{\pi} \frac{1}{\omega_2 - \omega_1} \int_{\omega_2}^{\omega_1} \text{Re}[Y] d\omega
\]  (31)

wherein, \( \omega_2 \) and \( \omega_1 \) are the lower and lower limits of the frequency of \( \Delta \omega = (\omega_2 - \omega_1) \). \( \text{Re}[\overline{Y}] \) is the spatial average of \( \text{Re}[Y] \). It can be seen from Equation (31) that the method of testing modal density can be obtained by testing the input admittance of excitation points. \( Y \) can be expressed as [12–15]:

\[
Y(\omega) = \frac{V(\omega)}{F(\omega)} = \frac{-iA(\omega)}{\omega F(\omega)}
\]  (32)
In the formula, $F(\omega)$, $V(\omega)$ and $A(\omega)$ respectively represent the Fourier transform of excitation force, velocity, and acceleration. When the ideal system ignores external noise or feedback noise, the admittance of the input point can be expressed as:

$$Y(\omega) = \frac{V(\omega)}{F(\omega)} = -\frac{iA(\omega)}{\omega F(\omega)} = \frac{G_{fv}(\omega)}{G_{ff}(\omega)}$$

(33)

where, $G_{fv}(\omega)$ and $G_{ff}(\omega)$ represent the unilateral cross-spectral density and self-spectral density of excitation force and response velocity, respectively. The modal density of commercial vehicle cab tested according to the above measurement method is shown in the figure below:

It can be seen from the modal density test results that when the structure stiffness is large, the modal density of the subsystem is small in a certain frequency range. In the SEA model of commercial vehicle cab, the modal density of the front floor, front wall side outer panel, front coaming, front wall connecting plate, and the rear floor is relatively low when the frequency is 0–1000 Hz. In the hybrid FE-SEA modeling, the flexibility of some components can be appropriately carried out. When building the vehicle SEA model, many complex structures need to be simplified, and the modal density obtained by the modal density theoretical calculation formula is equal in each frequency band. However, it can be seen from Figure 6 that the modal density obtained from the test varies in value within each frequency band. The hybrid FE-SEA model was adjusted and modified through the test curve, and the accuracy of the model prediction was improved.

![Figure 6. Modal density test results. (a) Test results of storage bins and ceilings; (b) Test results of side coiling and door panels; (c) Test results of the front windshield and front floor; (d) Test results of the instrument panel and floor.](image-url)
3.4. Obtaining Modeling Parameters of Damping Loss Factor

Damping loss factor (DLF) is a physical quantity used to express the damping characteristics of a structure. It refers to the ratio of the loss and stored energy of a structure per unit frequency and time. There are two methods to measure the internal loss factor: the steady-state method and the transient method. The transient attenuation method is to apply random excitation to the system and then remove the excitation, and obtain the internal loss factor through vibration attenuation. This method is suitable for rapidly estimating the average internal loss factor of structure and sound volume, and its statistical error is relatively small. The acquisition impulse response is converted to 1/3 octave by using Equation (34) [13–15]:

\[ x(t, f_c, \Delta f) = y(t) \cdot h(t, f_c, \Delta f) \]  

where \( y(t) \) represents the pulse signal collected. \( h(t, f_c, \Delta f) \) is the 1/3 octave filtering impulse response. Signal transformation is used to solve the average signal envelope, and its expression is as follows:

\[
\begin{align*}
\epsilon^2(t, f_c, \Delta f) &= (x(t, f_c, \Delta f) + x(t, f_c, \Delta f))^2 \\
\epsilon(t, f_c, \Delta f) &= H(x(t, f_c, \Delta f))
\end{align*}
\]

(35)

The curve within a period of time (generally 10 ms) is processed to obtain a smooth RMS attenuation curve, which is expressed as:

\[ \epsilon^2(t \rightarrow t + \Delta t, f_c, \Delta f) = \epsilon^2(t, f_c, \Delta f) \cdot r(\Delta t) \]

(36)

where \( r(\Delta t) \) is the unit vector. \( f_c \) is the center frequency. Root mean square attenuation curve is output in logarithmic form, and the curve is converted into acceleration level (dB) unit, which can be expressed as:

\[ 10 \log_{10}(\epsilon^2(t \rightarrow t + \Delta t, f_c, \Delta f)) \]

(37)

According to the above formula and the attenuation method, the internal loss factor of the structure can be deduced as follows:

\[ \eta(f_c, \Delta f) = \frac{D_R}{27.3 f_c} \]

(38)

When the pulse responses of \( N \) accelerometers are collected in a system, the average internal loss factor of the system can be obtained. Equation (39) can be used to obtain the average internal loss factor of the subsystem under \( K \) times excitation, which is expressed as:

\[ \eta(f_c, \Delta f) = \frac{1}{KN} \sum_{i=1}^{k} \sum_{j=1}^{n} (\eta_{ij}(f_c, \Delta f)) \]

(39)

When measuring the internal loss factor, a short-time pulse signal is generally selected, and the data tends to be smooth after the pulse signal fluctuation. Therefore, the data on the attenuation curve within 0.4 s is usually selected for least square processing to obtain the attenuation rate. The internal loss factor of the subsystem can be calculated through Equation (39). Figure 7 for cabin parts within the loss factor testing figure, including within the dashboard whole cabin internal loss factor testing, servicing the internal loss factor testing, cab within coaming after loss factor testing, cab front bumper at the bottom of the internal loss factor testing, etc., according to the above method to test values of the measured points test results for multiple averaging systems within the average loss factor.
Figure 7. Internal loss factor test. (a) The dashboard loss factor testing; (b) The door inside loss factor testing; (c) Loss factor testing within coaming after; (d) Loss factor testing in the bottom of the front bumper.

Figure 8 shows the internal loss factor test results of different structures. It can be seen from Figure 8a,b that the internal loss coefficient of the outside and inside panels on the cab side is greater than that of the ceiling, and is also greater than the damping coefficient of the left rear floor and the right rear floor. The Rear left and right sides of the two floors of the internal loss factor size are similar. It can be seen from Figure 8c,d that the internal loss factors of the front bumper of the cab > instrument panel and the inside plate of the door, the inside plate of the storage box, and the outside plate of the storage box are all small and equal, and the value of the internal loss factor is 0.01. Figure 8e,f for the first floor, the floor, after floor, sunroof, outside the small act the role of the internal loss factor testing curve, the front and in the middle of the floor loss factor is larger, the results as an input parameter added to the mixture of FE-SEA model, the parameters of the related component model is modified, can improve the calculation accuracy of hybrid model.
Figure 8. Internal loss factor test results of different structures. (a) Internal loss factor of side plate; (b) Roof and other structures of the internal loss factor; (c) Internal wear factor of bumper and other structures; (d) Internal loss factor of storage box and other structures; (e) Internal wear factor of rear floor and other structures; (f) The internal loss factor of floor and other structures.
3.5. Obtaining Modeling Parameters of Coupling Loss Factor

The coupling damping factor is an important parameter used to express the energy exchange between Coupling systems. It is used to represent the power flow or damping effect of one system to another system. The test method testing the coupling loss factor can be used for algorithm modification and model modification. The cab system is divided into 40 subsystems. For $N = 40$ subsystems, the PIM power input method can be used to test the coupling loss factor of the system and measure $\eta_{ij}(i,j = 1,2,3,\ldots,N)$ unknowns, which can be expressed as:

$$\{\eta\} = (\eta_{11}\eta_{12}\ldots\eta_{1N}\eta_{21}\eta_{22}\ldots\eta_{NN})^T \quad (40)$$

In order to solve the unknown quantity of the system, $N$ independent tests are carried out. When only the subsystem $i$ is excited, the power flow balance equation of the system can be obtained:

$$[E_i^1 B_1, E_i^2 B_2, \ldots, E_i^N B_N] \{\eta\} = \begin{pmatrix} \omega_1 p_i \\ 0 \\ \vdots \\ 0 \end{pmatrix} \quad (41)$$

where Equation (41) is the matrix equation obtained from the $i$th test. After completing $N$ independent tests, the matrix equation can be expressed as [11–15]:

$$[\mathcal{B}] = \{\eta\} = \{p\} \quad (42)$$

$$[\mathcal{B}] = \begin{bmatrix} E_1^1 B_1 & E_1^2 B_1 & \cdots & E_1^N B_N \\ E_2^1 B_1 & E_2^2 B_2 & \cdots & E_2^N B_N \\ \vdots & \vdots & \ddots & \vdots \\ E_N^1 B_1 & E_N^2 B_2 & \cdots & E_N^N B_N \end{bmatrix} \quad (43)$$

$$B_i = \begin{bmatrix} -1 & 0 & 0 & \cdots & 0 \\ 0 & -1 & 0 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 1 & 1 & 1 & \cdots & 1 \\ 0 & \cdots & 0 & -1 & 0 \\ 0 & \cdots & 0 & 0 & -1 \end{bmatrix} \quad (44)$$

$$\{p\} = \left( \frac{p_1}{\omega_1}, 0, \frac{p_2}{\omega_2}, 0, \ldots, 0, \frac{p_N}{\omega_N} \right)^T \quad (45)$$

In the case that power flow cannot be directly measured, the modal density of the subsystem can be tested. In this case, the reciprocity principle is used to obtain:

$$\eta_{112} = \eta_{211} \quad (46)$$

Combined with Equations (40)–(46) and the system energy and power tested by the PIM method, the coupling loss factor matrix of $N$ subsystems can be obtained. Due to the small order of magnitude of coupling loss factors in the testing process, some of the coupling loss factors may lose their physical significance in the final results, which need to be modified mathematically to achieve acceptable test accuracy. Using the reciprocity
principle in Equation (46), the power flow balance equation of the system composed of \( N \) subsystems in the cab can be expressed as:

\[
\frac{P_i}{\omega} = \sum_{k=1}^{N} \eta_{ik} E_i - \sum_{j=1 \atop j \neq k}^{N} \eta_{ji} E_j
\]

\[
= \sum_{k=1}^{N} \eta_{ik} n_i \frac{E_i}{\pi_i} - \sum_{j=1 \atop j \neq k}^{N} \eta_{ji} n_j \frac{E_j}{\pi_j}
\]

\[
= \varepsilon_i \sum_{k=1}^{N} C_{ik} - \sum_{j=1 \atop j \neq k}^{N} C_{ji} \varepsilon_j \quad (i = 1, 2, \ldots, N)
\]

(47)

where the expressions of \( \varepsilon_i \) and \( C_{ji} \) are as follows:

\[
\begin{cases}
\varepsilon_i = \frac{E_i}{\pi_i} (i = 1, 2, \ldots, N) \\
C_{ji} = n_i n_{ji} = n_j n_{ij} = C_{ij}
\end{cases}
\]

(48)

It is assumed that the excitation sources among different subsystems are independent of each other, so the system responses can be superimposed, and the power flow balance equation of the whole system can be expressed as:

\[
[B] \{ \varepsilon \} = \{ p \}
\]

(49)

where, \( \{ \varepsilon \} \) and \( [B] \) can be expressed as:

\[
\begin{cases}
\{ \varepsilon \} = (\varepsilon_1, \varepsilon_2, \ldots, \varepsilon_N)^T \\
B_{ij} = C_{ij} = \sum_{k=1}^{N} C_{ik} \quad i = j \\
-C_{ij} \quad i \neq j
\end{cases}
\]

(50)

Based on Equations (49) and (50), the power flow balance equation of \( N \) systems can be rewritten as:

\[
\{ \varepsilon \} = [B]^{-1} \{ p \} = [A] \{ p \}
\]

(51)

Matrix \( [A] \) can be obtained by using the test method, and then matrix \( [B] \) can be obtained by inverse operation, and then the coupling loss factor with a certain accuracy can be obtained. The specific implementation steps are as follows:

1. Steady-state excitation of a substructure of the cab is carried out, and the input power of the test is output, as well as the modal energy of all the substructures.
2. Steady-state excitation is carried out to the other substructures one by one, and the input power of each subsystem and the other substructures are output Structural modal energy of the system.
3. After the steady-state excitation of all subsystems is completed, matrix \( A_{ij} \) obtained from the experimental test is used to edit matrix \( [A] \), and the matrix \( [B] \) is obtained by the inverse operation. For the operation with partial irreversible matrix, refer to the algorithm for solving the coupling matrix of hybrid FE-SEA model in Section 2 and the parameter iteration of the ill-posed matrix, and replace the ill-posed matrix to get the revised \( [A] \).
4. Because of the test error and the ill-conditioned matrix in the process of finding the inverse, the correction principle for matrix \( [A] \) satisfies the following relation.
\[
\begin{align*}
A_{ij} &= A_{ji} \quad i = 1, 2, \ldots, N \\
A_{ij} &> A_{ji} \quad i = 1, 2, \ldots, N \\
\sum_{j=1}^{N} B_{ij} &> 0 \quad i = 1, 2, \ldots, N \\
B_{ii} &> 0
\end{align*}
\] (52)

The parameters of the coupling loss factor were obtained according to the above method. The measurement points of the coupling loss factor of the cab subsystem were arranged as shown in Figure 9. The acceleration sensor adopts the PCB model from the USA, measuring range ±2 g–±50 g, output voltage 0.5 V–4.5 V. The acceleration signals of different measuring points are collected, and the output power and energy matrix of the measuring points are measured by processing the acceleration admittance signals. Through formula (40–46) conversion, the coupling loss factor of the measured point is obtained. The measuring range of the force hammer sensor is 5000 N, the resonant frequency is >22 kHz, and the output impedance is <100 Ω. The measurement points were mainly distributed in the front coaming of the cab, rear coaming of the cab, outside plate, inside plate, floor, and other structures. Bridge structure of the coupling loss factor testing results as shown in Figure 10, from Figure 10a,b shows that the coupling loss factor between different structures is more concentrated in the \(10^{-2}–10^{-3}\) orders of magnitude, side round plate transfer car door plate between the coupling loss factor and the results are consistent, in the process of reverse transfer in the process of power flow transmission, round plate and the car door side coupling loss factor between the board structure with the increase of frequency, coupling loss factor decreases, reducing trend during 800–1000 Hz frequency curve to flatten out. It can be seen from Figure 10c,d that the coupling loss factor transferred from the front floor to the front coining plate is greater than the power transfer between the inside sidewall and the outside plate. With the increase of frequency, the coupling loss factor between the front floor, inside sidewall, and the outside plate decreases gradually. The above test results of the coupling loss factor were compared with the hybrid FE-SEA model of cab, to realize the coupling loss factor of the modified model. The accuracy of the connection relationship between the coupling loss factor and the internal loss factor is one of the key factors affecting the accuracy of the model.

**Figure 9.** Measurement points of coupling loss factor of different structures. (a) Front coaming measuring point; (b) Rear coaming measuring point; (c) Door measuring point; (d) Floor measuring point.
Figure 10. The coupling loss factor of different structures. (a) Coupling loss factor of roof and other structures; (b) Coupling loss factors of door panels and other structures; (c) Coupling loss factor for front floor and other structures; (d) Coupling loss factors for structures such as rear floors; (e) Coupling loss factor of rear coaming and other structures; (f) Coupling loss factor of front bumper and other structures.
4. Model Recognition in Volume Source Mode

In the cab model verification, the volume sound source is used as the excitation source. The volume sound source is arranged in the left front wheel excitation, the right front wheel excitation, and the engine compartment excitation just below the floor of the cab, respectively. The excitation method is white noise sweep excitation. Acoustic microphones were uniformly arranged on the surface of the inner and outer sound cavities of the hybrid FE-SEA model in the cab to receive the sound from the excitation source. The internal and external sound fields of the cab and the whole vehicle are shown in Figure 11. The internal sound field separates the seat sound field from the cab sound field. The external sound field is shown in Figure 11b,c which represents the sound pressure distribution of the external sound field at different sound velocities. According to the subsystem modeling method of the hybrid model, 8–12 microphones are uniformly arranged on the surface of each internal sound field. The distance between the microphone and the surface of the internal sound cavity is 100–150 mm. The sound pressure value at the surface position of each internal sound cavity is taken as the input sound load of the hybrid FE-SEA model of the cab. In the test process, the sound pressure level of the background noise is required to be 20 dB lower than the sound pressure level of the test condition. In the vehicle test area, except for the engine compartment, the acoustic characteristics of the left and right sides of the vehicle are considered to be symmetrical. During the test, only one side of the vehicle is measured, and the arrangement of measuring points is shown in Figure 12.

Figure 11. Internal and external sound fields. (a) The internal sound chamber of the cab; (b) External sound fields at different velocities (70 km/h); (c) External sound fields at different velocities (80 km/h).
The cab hybrid FE-SEA model validation results are shown in Tables 4–8. Figure 13a,b shows the left front wheel for acoustic excitation, where sound transmission loss within 0–2000 Hz frequency has high precision, there is a simulation value and test value maximum error smaller than 2.66 dB, and an increase of frequency at 2000–7000 Hz frequency range. When the model simulation value and test value of error increases gradually, the individual frequency point error value is bigger. This is mainly due to the errors caused by the relatively dense structural modal density of the finite element subsystem in the hybrid FE-SEA model at medium and high frequencies, as well as the test errors caused by noise signals and external interference signals in the test of the statistical energy parameters of the SEA board, such as input power and energy parameters of each subsystem. At the same time, when building the complex model of vehicle hybrid FE-SEA, the complex structure is properly simplified without considering the influence of partial leakage and holes, and the coverage rate of the interior accessories of the cab is properly simplified, which will also affect the prediction error in the high-frequency band. Figure 13c,d shows the distribution curve of acoustic transmission loss and error of the main driving when the right front wheel is excited. It can be seen from the figure that the error trend is the same as that of the left front wheel excitation, and the intermediate frequency accuracy is high. When the frequency increases, the front end model of the cab can be modified by the test method under the condition of front-wheel excitation, which can be used for the simplification of the subsystem and the errors in the high-frequency band caused by the dense mode density. Figure 13 e–j shows the engine compartment, left rear wheel incentives, and right rear wheel under the incentive of the rear seat sound transmission loss in the 0–7000 Hz of the whole band simulation value. The test value is relatively close
to, in addition to the individual frequency maximum error is less than 3 dB. The results show that the test-measured SEA parameters and acoustics parameters are used to fix the hybrid model using the in-vehicle hybrid FE-SEA high-frequency simulation analysis. This method is accurate and reliable.

Table 4. Left front wheel drive test results.

| Analysis Frequency | Simulation (dB) | Test (dB) | Error Value (dB) |
|--------------------|----------------|----------|-----------------|
|                    | Left Front Wheel-Near Driver's Ear | Left Front Wheel-Near Driver's Ear |                  |
| 400                | 40.67          | 38.01    | 2.66            |
| 500                | 41.34          | 39.81    | 1.53            |
| 630                | 41.48          | 39.07    | 2.41            |
| 800                | 43.26          | 40.84    | 2.42            |
| 1000               | 45.36          | 47.47    | 2.11            |
| 1250               | 47.28          | 47.99    | 0.71            |
| 1600               | 49.42          | 50.21    | 0.78            |
| 2000               | 51.82          | 49.58    | 2.24            |
| 2500               | 57.41          | 47.86    | 9.55            |
| 3150               | 59.38          | 55.00    | 4.38            |
| 4000               | 60.90          | 51.35    | 9.55            |
| 5000               | 63.60          | 57.61    | 5.99            |
| 6300               | 66.12          | 58.99    | 7.13            |

Table 5. Right front-wheel-drive test results.

| Analysis Frequency | Simulation (dB) | Test (dB) | Error Value (dB) |
|--------------------|----------------|----------|-----------------|
|                    | Right Front Wheel-Near Driver's Ear | Right Front Wheel-Near Driver's Ear |                  |
| 400                | 42.72          | 37.28    | 5.43            |
| 500                | 42.87          | 41.57    | 1.30            |
| 630                | 42.72          | 39.76    | 2.95            |
| 800                | 44.21          | 37.03    | 7.18            |
| 1000               | 46.14          | 44.35    | 1.59            |
| 1250               | 47.98          | 46.35    | 1.63            |
| 1600               | 50.08          | 46.83    | 3.25            |
| 2000               | 52.49          | 50.70    | 1.79            |
| 2500               | 58.88          | 48.60    | 10.29           |
| 3150               | 60.94          | 58.19    | 2.75            |
| 4000               | 62.82          | 52.76    | 10.05           |
| 5000               | 65.29          | 59.10    | 6.19            |
| 6300               | 67.58          | 59.27    | 8.31            |

Table 6. Engine compartment excitation test results.

| Analysis Frequency | Simulation (dB) | Test (dB) | Error Value (dB) |
|--------------------|----------------|----------|-----------------|
|                    | Engine Nacelle-Driver's Ear | Engine Nacelle-Driver's Ear |                  |
| 400                | 39.19          | 34.67    | 4.51            |
| 500                | 40.42          | 39.25    | 1.17            |
| 630                | 41.44          | 39.01    | 2.43            |
| 800                | 43.17          | 40.16    | 3.01            |
| 1000               | 44.78          | 48.81    | 4.03            |
| 1250               | 46.64          | 47.81    | 1.17            |
| 1600               | 48.53          | 48.08    | 0.26            |
| 2000               | 50.10          | 51.55    | 4.6              |
| 2500               | 52.93          | 49.17    | 3.76            |
| 3150               | 54.49          | 57.53    | 3.05            |
| 4000               | 56.15          | 53.96    | 2.19            |
| 5000               | 57.62          | 58.73    | 1.10            |
| 6300               | 59.06          | 59.88    | 0.82            |
Table 7. Left rear wheel excitation test results.

| Analysis Frequency | Simulation (dB) | Test (dB) | Error Value (dB) |
|--------------------|----------------|-----------|-----------------|
|                    | Left Rear Wheel-Middle Rear | Left Rear Wheel-Middle Rear | |
| 400                | 32.02          | 32.20     | -0.18           |
| 500                | 35.70          | 35.49     | 0.21            |
| 630                | 37.05          | 37.57     | -0.54           |
| 800                | 38.17          | 43.08     | -4.91           |
| 1000               | 40.57          | 43.70     | -3.13           |
| 1250               | 43.55          | 44.48     | -0.93           |
| 1600               | 45.70          | 45.80     | -0.10           |
| 2000               | 47.50          | 46.30     | 1.20            |
| 2500               | 49.20          | 51.50     | -2.30           |
| 3150               | 51.80          | 52.00     | -0.20           |
| 4000               | 55.10          | 53.20     | 1.90            |
| 5000               | 57.90          | 57.20     | 0.70            |
| 6300               | 59.10          | 59.10     | 0.00            |

Table 8. Right rear wheel excitation test results.

| Analysis Frequency | Simulation (dB) | Test (dB) | Error Value (dB) |
|--------------------|----------------|-----------|-----------------|
|                    | Right Rear Wheel-Middle Rear | Right Rear Wheel-Middle Rear | |
| 400                | 32.02          | 31.98     | 0.22            |
| 500                | 35.90          | 35.84     | 0.06            |
| 630                | 37.20          | 35.56     | 1.64            |
| 800                | 38.40          | 40.74     | -2.34           |
| 1000               | 40.80          | 43.80     | -3.00           |
| 1250               | 43.70          | 45.50     | -1.80           |
| 1600               | 45.90          | 46.10     | -0.20           |
| 2000               | 47.70          | 49.70     | -2.00           |
| 2500               | 49.30          | 52.10     | -2.80           |
| 3150               | 51.80          | 53.80     | -2.00           |
| 4000               | 55.10          | 55.50     | -0.40           |
| 5000               | 57.70          | 58.30     | -0.80           |
| 6300               | 59.00          | 60.20     | -1.20           |

Figure 13. Cont.
Figure 13. Cont.
Figure 13. Verification results of cab model. (a) Acoustic transmission loss under left front wheel excitation; (b) Acoustic transmission loss under left front wheel excitation; (c) Acoustic transmission loss under right front wheel excitation; (d) Acoustic transmission loss under right front wheel excitation; (e) Acoustic transmission loss under engine room excitation; (f) Acoustic transmission loss under engine room excitation; (g) Acoustic transmission loss under left rear wheel excitation; (h) Acoustic transmission loss under left rear wheel excitation; (i) Acoustic transmission loss under right rear wheel excitation; (j) Error distribution under right rear wheel excitation.

5. Acoustic Package Optimization

5.1. Vehicle Acoustic-Acoustic Transfer Function

The acoustic-acoustic transfer function is a method to obtain the sound transmission of the whole system by stimulating the external or internal sound field, which has a certain effect on optimizing the acoustic performance of the cab. When making sound incentive-sound transfer function, select the left front, right front wheel incentives, engine compartment, left rear wheel incentives as incentive points, the main drive is chosen as the receiver, the cab acoustic test results of the noise transfer function is shown in Figure 14, from Figure 14a,b, in the left front wheel Hz frequency range of 0–3000 to the main driver acoustic noise transfer function value is low, especially within the 2000 Hz, the sound transmission loss in 36–42 dB. In the right front turn, the acoustic-acoustic transfer function of the main driver is consistent with the trend of the left front wheel in the frequency range of 0–3000 Hz. Therefore, it is necessary to optimize the performance of the acoustic package in the cab. Meanwhile, in the low-frequency range of 0–500 Hz, the damping design and layout of the cab body should be carried out to meet the noise isolation at low frequency. Figures 14c and 14d respectively represent the acoustic transfer function from the engine compartment to the main driver and the acoustic transfer function from the left rear wheel to the middle rear seat. It can be seen from Figure 14c that the frequency range is 0–2000 Hz, the acoustic transfer function from the engine compartment to the main driver is small, and the acoustic transfer loss is 34–48 dB. In Figure 14d, when the acoustic transfer function of the left rear wheel excitation is transferred to the middle of the back row, the acoustic transfer loss is relatively low when the frequency is less than 2000 Hz. According to the acoustic-acoustic transfer function at each excitation point, the airtightness of the whole cab and the performance of the acoustic package need to be further optimized.
Figure 14. Test results of the acoustic-acoustic transfer function. (a) Acoustic-acoustic transfer function under left front wheel excitation; (b) Acoustic-acoustic transfer function under right front wheel excitation; (c) Acoustic-acoustic transfer function under engine room excitation; (d) Acoustic-acoustic transfer function under left rear wheel excitation.

5.2. Optimization Scheme and Verification

According to the acoustic noise transfer function of the test result of cab air tightness test and optimization and testing to seal cab, sealed cabin pressure relief valve in two places, to smoke tests are carried out from the co-pilot wind window, in the smoke reaches 125 pa pressure for pressure test and record the cab leak, the test process is shown in Figure 15. According to the leakage point smoke screen need to optimize the structure of the cab, air-tightness optimization results as shown in Table 9, through the airtightness test, a total of 19 screening leak, concentrated leakage hole under the wind window 4, side surrounds discharge hole 4, under the small leakage hole 3, 8 from rat hole under the C column, a total of 56.8 SCFM leak value, according to the different leak point respectively adopt partition, glue, glue sealing cavity optimization method for leak tightness.
Table 9. The sealing scheme of the cab.

| Sequence Number | Let the Cat Out of the Area | Leakage Rate/SCFM | Let the Cat Out of the Path | Optimum Proposal                      |
|-----------------|----------------------------|------------------|----------------------------|---------------------------------------|
| 1               | Leakage hole under air window × 4 | 8.8              | The hole communicates with the cavity | PVC Glue sealing                      |
| 2               | Leakage hole is enclosed on the side × 4 | 18               | The hole communicates with the cavity | Optimize the partition position       |
| 3               | Leakage hole is enclosed in front × 3 | 10               | The hole in the electrophoresis fluid communicates with the cavity | Finger pressure glue sealing          |
| 4               | Mouse hole under pillar C × 8 | 20               | Leakage of sheet metal lap joint | Optimize the gluing quality           |

Figure 15. The cab sealing test. (a) Seal of cab; (b) Seal measuring point; (c) Cab bottom measuring point; (d) Measuring point of connection.

High frequency energy distribution simulation results in the reference bridge panel acoustic package scheme determined, based on the modal superposition method of bridge panel energy simulation results shown in Figure 16a it can be seen from the picture, when in 300–5000 Hz frequency range after all modal superposition bridge side surround most area of the energy concentration, need to optimize acoustic package and in the same way according to the Figure 16b,c, the bridge at the top of the interior trim plate position, after the bridge coaming interior trim panels on both sides of the regional distribution of strain energy is larger, the cab floor front strain energy is relatively concentrated. According to the simulation results of acoustic package optimization scheme as shown in Figure 17, in Figure 17a position in the cab roof lining 1–3 area increased absorption material used to absorb the cab at the top of the energy, in Figure 17b in the areas of circumference interior trim panels of 1 to 5, increase the sound-absorbing cotton material, in Figure 17c, lateral confining interior plate left and right side respectively increase 1 area of sound-absorbing material. The optimization scheme of the acoustic package and acoustic material parameters are shown in Table 10.
Figure 16. Simulation of an acoustic enclosure covering an area of the cab. (a) Simulation of acoustic coverage of cab side; (b) Simulation of acoustic coverage of cab rear; (c) Simulation of acoustic coverage of cab floor.

Table 10. Acoustic optimization scheme for the cab.

| Sequence Number | Scheme                        | Thickness/mm | Gram Weight/g/m² | Fraction of Coverage/% |
|-----------------|-------------------------------|--------------|-------------------|------------------------|
| 1               | The top is lined with attracting cotton | 20           | 232               | 70                     |
| 2               | Side interiors attract cotton | 20           | 232               | 70                     |
| 3               | The rear enclosure interior attracts cotton | 15           | 672               | 80                     |
| 4               | Door interior panels attract cotton | 15           | 232               | 60                     |
Verification results of airtightness and acoustic package optimization scheme are shown in Figures 18 and 19. The noise test method is adopted to verify the improvement effect of cab acoustic performance under typical full load and no-load conditions. As can be seen from Figure 18, in the full load condition of 70 km/h with uniform speed, the cab noise performance has a significant effect at frequencies above 400 Hz, and the ear noise of both the main driver and the co-driver is significantly reduced. In the frequency range of 400–8000 Hz, the ear noise of the main driver and the co-driver is reduced by 2.56 dB (A) and 2.4 dB (A). As can be seen from Figure 18c, d in the full load condition of 80 km/h with uniform speed, the ear noise of the main driver and the co-driver is reduced by 2.0 dB (A). With the increase of the vehicle speed, the noise performance level tends to be stable. Figure 19 shows the cab acoustic verification results under the no-load condition of 70 km/h and 80 km/h at constant speed. It can be seen from the figure that, in the range of 400–8000 Hz at a constant speed of 70 km/h, the ear noise of the main driver and the co-driver decreases by 3.1 dB (A) and 4.6 dB (A) respectively, and that at a constant speed of 80 km/h decreases by 3.6 dB (A) and 4.9 dB (A) respectively. As can be seen from the test results in Figures 18 and 19, within the frequency range of 200–8000 Hz, the RMS value of the driver’s ear noise is less than 70 dB after the optimization of the cab acoustic package scheme and the vehicle airtightness, indicating a significant optimization effect, which further verifies the effectiveness of the cab acoustic package optimization and airtightness optimization methods adopted in this paper.
Figure 18. Optimization of acoustic performance under typical load conditions. (a) 70 km/h uniform full load condition, comparison before and after optimization of main drive noise; (b) 70 km/h uniform full load condition, comparison before and after optimization of first drive noise; (c) 80 km/h uniform full load condition, comparison before and after optimization of first drive noise; (d) 80 km/h uniform full load condition, comparison before and after optimization of first drive noise.
Figure 19. Optimization of acoustic performance under typical no-load conditions. (a) 70 km/h uniform no-load condition, comparison before and after optimization of main drive noise; (b) 70 km/h uniform no-load condition, comparison before and after optimization of first drive noise; (c) 80 km/h uniform no-load condition, comparison before and after optimization of first drive noise; (d) 80 km/h uniform no-load condition, comparison before and after optimization of first drive noise.
After the optimization scheme in Tables 9 and 10, the acoustic package and airtightness performance of the cab are optimized. The acoustic performance of the cab is significantly improved. Frequency extraction of drivers’ ear noise at different speeds (70 km/h, 80 km/h) is carried out, as shown in Figure 20. From Figure 20a,b, it can be concluded that the acoustically optimized noise waterfall diagram near the driver’s ear has no obvious noise order. However, the noise in the initial state without acoustic optimization is larger in order 2.6, 3, 6, and 10. Therefore, acoustic package optimization and airtightness optimization for noise have significant effects.

Figure 20. Noise frequency extraction at different speeds. (a) Spectrum extraction of driver’s ear noise (80 km/h); (b) Spectrum extraction of driver’s ear noise (70 km/h).

6. Conclusions

In this paper, a hybrid FE-SEA structural model and a commercial vehicle cab acoustic model are established by using the subsystem modeling and connection method of the hybrid FE-SEA model. Experimental statistical energy parameters and model validation of the FE-SEA hybrid model are studied. In addition, the hybrid model with statistical energy parameters (internal loss coefficient, modal density, coupling loss factor, etc.) was tested and modified. The model verification results show that under engine excitation, left rear wheel excitation, and right rear wheel excitation, the sound transmission loss of the rear seat is in the 0–7000 Hz frequency band, and the simulation value of the hybrid model is
close to the experimental value. Except for a few frequency points, the maximum error is less than 3 dB, and the model is accurate and reliable. Finally, the acoustic performance of the cab is optimized by using an airtight acoustic package scheme. After vehicle validation, the acoustic scheme was reduced by 2–3 dB on average.

**Author Contributions:** J.S. and Z.D.; methodology, J.S.; software, J.S.; validation. All authors have read and agreed to the published version of the manuscript.

**Funding:** This research was funded by [Research on key technologies of auxiliary Tools for Body design and analysis] grant number [H20200380].

**Informed Consent Statement:** Written informed consent has been obtained from the patient(s) to publish this paper.

**Acknowledgments:** The research was financially supported by the National Natural Science Foundation of China.

**Conflicts of Interest:** The authors declare no conflict of interest.

**References**

1. Langley, R.S.; Bremner, P. A hybrid method for the vibration analysis of complex structural-acoustic systems. *J. Acoust. Soc. Am.* **1999**, *105*, 1657–1671. [CrossRef]

2. Langley, R.S.; Cordioli, J.A. Hybrid deterministic-statistical analysis of vibro-acoustic systems with domain couplings on statistical components. *J. Sound Vib.* **2009**, *321*, 893–912. [CrossRef]

3. Musser, C.T.; Rodrigues, A.B. Mid-frequency prediction accuracy improvement for fully trimmed vehicle using hybrid SEA-FEA technique. *SAE Tech. Pap.* **2008**. [CrossRef]

4. Manning, J.E. Hybrid SEA for mid-frequencies. *SAE Tech. Pap.* **2007**. [CrossRef]

5. Mattson, S.G.; Labyak, D.; Praetz, J.; Connelly, T. Prediction of muffler insertion loss by a hybrid FE acoustic-SEA model. *SAE Int. J. Passeng. Cars-Mech. Syst.* **2009**, *2*, 1323–1329. [CrossRef]

6. Shorter, P.; Zhang, Q.; Parrett, A. Using the hybrid FE-SEA method to predict and diagnose component transmission loss. *SAE Tech. Pap.* **2007**. [CrossRef]

7. Yun, W.G.; Wu, Z.F.; Deng, C. Analysis and Control of Structural Noise of Heavy Commercial Vehicle Cab. *Noise Vib. Control.* **2015**, *5*, 87–90.

8. Cicirello, A.; Langley, R.S. The vibro-acoustic analysis of built-up systems using a hybrid method with parametric and non-parametric uncertainties. *J. Sound Vib.* **2013**, *332*, 2165–2178. [CrossRef]

9. Cicirello, A.; Langley, R.S. Efficient parametric uncertainty analysis within the hybrid Finite Element/Statistical Energy Analysis method. *J. Sound Vib.* **2014**, *333*, 1698–1717. [CrossRef]

10. Yin, H.; Yu, D.; Lü, H.; Yin, S.; Xia, B. Hybrid finite element/statistical energy method for mid-frequency analysis of structure-acoustic systems with interval parameters. *J. Sound Vib.* **2015**, *353*, 181–204. [CrossRef]

11. Zeng, X.K. *Simulation Calculation and Research of Low Frequency Acoustic Modes in Heavy Commercial Vehicle Cab*; Jilin University: Changchun, China, 2008.

12. Zhang, Y.T. *Simulation and Experimental Study on Low Frequency Acoustical Vibration Characteristics of Heavy Commercial Vehicle Driving Room*; Jilin University: Changchun, China, 2009.

13. Zhang, Y.B. *Analysis and Control of Structural Noise of Commercial Vehicle Cab Based on Acoustic-structure Coupling Model*; Hunan University: Changsha, China, 2013.

14. Zheng, J.; Yang, S.L.; Yu, J.H. Experimental Research on Active Noise Control Technology of Commercial Vehicle Cab. *Automot. Technol. Technol.* **2009**, *4*, 58–61.

15. Jing, X.R. *Research on Active Acoustic Quality Control Technology in Heavy Commercial Vehicle Driver Room*; Jilin University: Changchun, China, 2018.