Random Vibration Simulation Analysis of an Electronic Cabinet

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Abstract: This paper provides a brief introduction to the theoretical foundations of random vibration simulation analysis, using the random vibration simulation of a specific electronic cabinet as a case study. The processes of model building, meshing, boundary condition setting, modal analysis, and random vibration simulation analysis are described. The simulation results verify the rationality of the structural design and identify targets for optimization, providing a solid theoretical basis for subsequent product optimization and engineering practice.

Keywords: Finite Element; Random Vibration; Simulation Analysis.

1. Introduction

The complexity of modern battlefield conditions has raised the bar for the design of electronic cabinets [1]. As carriers for electronic components, electronic cabinets must possess sufficient shock resistance to prevent malfunctions that could impact the combat performance of the entire weapon system [2]. To ensure reliable operation of electronic equipment in vehicles, appropriate protective measures must be taken. Ensuring the work reliability of electronic cabinets under required environmental and usage conditions is essential for maintaining the overall performance of the equipment. Due to the destructive nature and high cost of vehicle-based tests, computer simulations are employed during the design phase to demonstrate the reliability of electronic equipment. Utilizing finite element software for simulation and analysis of the electronic equipment structure provides insights into its dynamic characteristics and responses. In the design phase, reasonable strength and stiffness should be considered, along with effective vibration isolation and buffering measures.

This paper, starting from engineering application practice, employs ANSYS Workbench finite element simulation software to conduct modal and random vibration analyses of an electronic cabinet under development. This approach aims to estimate the cabinet's weak points during the schematic design phase, providing a basis for dynamic optimization design.

2. Theoretical Foundation

2.1 Modal Analysis

Modal analysis is fundamental to dynamic analysis, aiming to determine the system's natural frequencies and modes, providing a theoretical basis for subsequent harmonic response analysis, spectral analysis, and optimized design.

The dynamic differential equation of an object can be expressed as follows [3]:

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\{F(t)\} \tag{1}$$

where $[M]$ is the structural mass matrix, $[C]$ is the structural damping matrix, $[K]$ is the structural stiffness matrix, $\{F(t)\}$ is the time-varying load function, $\{\mathrm{u}\}$ is the nodal displacement vector, $\{\dot{u}\}$ is the nodal velocity vector, and $\{\ddot{u}\}$ is the nodal acceleration vector.

By discretizing this equation and employing suitable numerical computation methods (e.g., second or fourth-order Runge-Kutta methods), the response values under given boundary conditions can be obtained.

2.2 Principles of Random Vibration

The excitation of random vibration is uncertain and unpredictable, and it is not repeatable under the same conditions. Its analysis is a spectral analysis technique based on probability and statistics. It seeks to determine the probabilistic distribution of displacement, stress, etc., under random excitation [4], meaning that both the input and output of the analysis have random probabilistic characteristics. The principle involves first calculating the statistical response of each modal response, then synthesizing them, assuming that the random vibration process is a stationary random process [5].

The power spectral density of a stationary random process $x(t)$ is defined as:

$$S(\omega)=\lim_{T\to+\infty}\frac{1}{2T}\left|X(\omega,T)\right|^2 \quad (2)$$

$S(\omega)$ represents the distribution density of vibration energy at various angular frequencies, and the area between $S(\omega)$ and the ω-axis equals the variance D_x of the process, i.e.:

$$R_x(\tau=0)=\int_{-\infty}^{+\infty}S(\omega)d\omega=D_x \quad (3)$$

3. Random Vibration Simulation Analysis

3.1 Introduction to a Specific Electronic Cabinet

The company's independently developed and designed series of electronic cabinets mainly consist of a skeleton and internal components. They are characterized by high reliability, strong versatility, good environmental adaptability, and friendly human-machine interaction, which have been well received by customers for decades. During the R&D phase, a large number of simulation tests were conducted to validate the cabinet, which not only shortened the R&D cycle but also significantly reduced research costs. The following analysis takes a specific model of the cabinet as an example to simulate its environmental vibration adaptability.

3.2 Modeling

The electronic cabinet is composed of thousands of assembled parts. The actual three-dimensional engineering drawings cannot be directly used for simulation analysis and need to be re-modeled to reduce unnecessary computations and highlight key areas. After the first design, the random vibration simulation results of the cabinet were not satisfactory, mainly showing deformation of the vertical columns under random vibration conditions.

It is necessary to revise the structure by strengthening the weak points, applying reinforcement beams to the front and back pairs of columns as shown in Figure 1. The structure is then meshed.

3.3 Meshing

Plate and shell elements are used to simulate the thin plate structure of the cabinet. These elements have a certain thickness and can resist tension, compression, and bending-twisting deformation. They can also effectively divide the shapes of these structures, fully describing the various characteristics of these parts, making plate and shell elements the most suitable type to describe the features of thick flat plates. In ANSYS, the main element type used is a shell element SHELL63. SHELL63 elements have the following characteristics.

SHELL63 elements can resist bending and include various properties of membrane elements, thus they can withstand in-plane forces as well as loads in other directions. The meshing is shown in Figure 2.

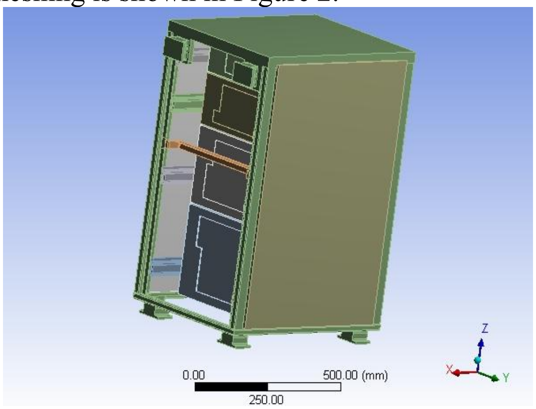

Figure 1. Reinforced Electronic Cabinet

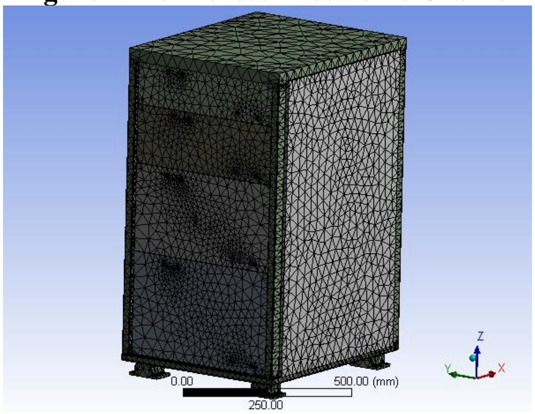

Figure 2. Meshing

3.4 Boundary Condition Setting

When the electronic cabinet is in use or being transported, its bottom and rear are connected to the vehicle's floor and sidewalls through shock absorbers. Therefore, when setting simulation boundary conditions, the lower plane of the shock absorber is fixed, and simulation analysis is carried out under the environment of natural gravity.

3.5 Modal Analysis

After modal analysis of the structure, the first 10 natural frequencies are obtained, as shown in Figure 3 and Table 1.

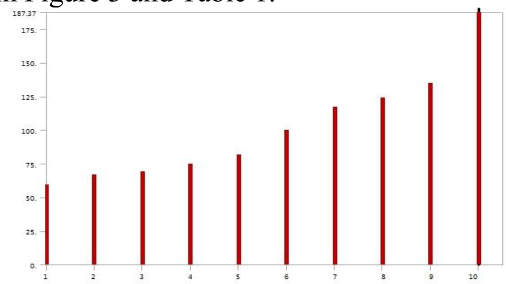

Figure 3. The First 10 Natural Frequencies of a Certain Electronic Cabinet

Table 1. Statistical Table of the First 10 Natural Frequencies of a Certain Electronic Cabinet

Mode	1	2	3	4	5
Frequency (Hz)	59.68	67.05	69.34	74.64	81.49
Mode	6	7	8	9	10
Frequency (Hz)	100.12	117.07	123.90	134.59	187.37

In the process of analyzing the weak points of the cabinet, based on experience, analyzing the first 4 modes is essentially sufficient to understand the dynamic characteristics and weak points of the system.

3.6 Random Vibration Excitation

Referencing GJB150.16A-2009, the vibration excitation for the cabinet is set, with parameters mainly including the random vibration frequency and magnitude in the X, Y, and Z directions. The specific numerical values of the parameters are not detailed here.

3.7 Random Vibration Results

The deformation of the cabinet in the X, Y, and Z directions is analyzed. The deformation in the X direction is shown in Figure 4. In this direction, the maximum deformation occurs on the reinforcement beam, with a type change of 0.8mm, and the key column structure deformation is 0.35mm, which meets the design technical requirements.

The deformation in the Z direction is shown in Figure 6. In this direction, the maximum deformation occurs on the bottom assembly and the guide rail connected to it. Due to the large weight of the bottom assembly, the inertial force generated by the Z-direction vibration is also the largest, resulting in the correspondingly largest deformation of the guide rail, which is 1.2mm, while the main indicator of column deformation is 0.5mm, meeting the technical requirements.

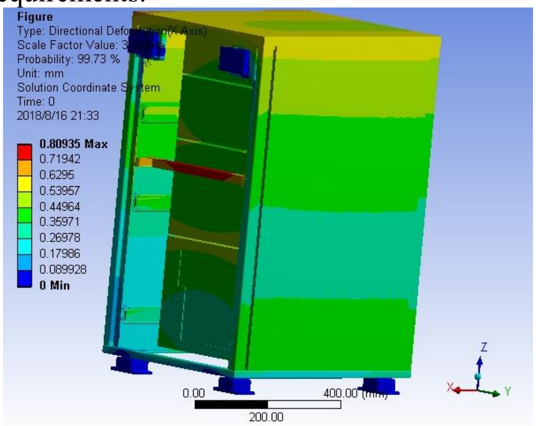

Figure 4. Deformation Schematic in the X Direction

The deformation in the Y direction is shown in Figure 5. In this direction, the maximum deformation occurs on the assembly far from the shock absorber and the skeleton connected to it, with the largest deformation of 0.88mm, also meeting the design technical requirements.

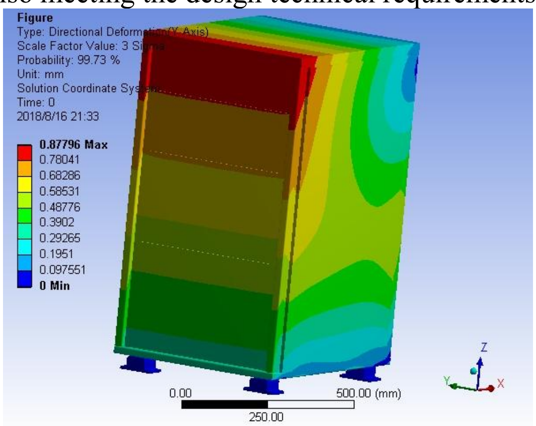

Figure 5. Deformation Schematic in the Y Direction

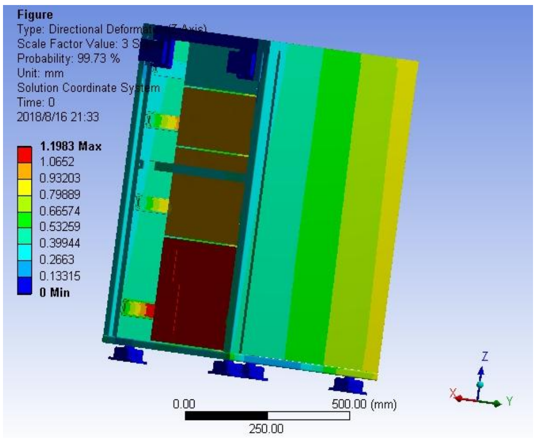

Figure 6. Deformation Schematic in the Z Direction

The focus of the simulation mentioned earlier is on the Y-direction deformation of the column, as the deformation in the Y direction could cause the locking device to malfunction. Through this random vibration simulation analysis, it is shown that the reinforcement structure effectively reduces the deformation of the column, and the impact on the locking device is within an acceptable.

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