The Vibration Characteristics Analysis of Damping System of Wall-mounted Airborne Equipment Based on FEM

Changqing Hu\textsuperscript{1,2}, Yiyong Yang\textsuperscript{1}, Kexia Peng\textsuperscript{2}, Yu Hu\textsuperscript{2}

\textsuperscript{1}School of Engineering and Technology, China University of Geosciences(Beijing), Beijing 100083, China
\textsuperscript{2}Beijing Institute of Aerospace Control Devices, Beijing 100039, China

*Corresponding author e-mail: 18910801138@189.cn

Abstract. Because of the structural stiffness coupling, the vibration responses of wall-mounted vibration damping system are difficult to analyze. Therefore, a research method based on FEM (finite element method) analysis is proposed. The method builds a finite element model of damping system by setting up elastic damping connection parameters. Then the damping system parameters are obtained by combining vibration reduction tests and vibration theory formula. Finally, the random vibration simulation results are compared with the test results, verifying the feasibility of FEM to study wall-airborne vibration damping system. Based on the finite element method, the effects of gravity center offset and connection position on vibration responses of damping system are studied.

1. Introduction

AWACS (Airborne Warning and Control System) aircrafts play a significant role in modern warfare. With the development of modern electronic technology, the kinds and amounts of electronic equipment installed in the cabin are continuously improving, and the installation spaces are becoming constrained. For saving spaces, some small equipment can be installed on the bulkhead [1]. Meanwhile, the mechanical environment in the aircraft cabin is strictly required during flight. Absorbers need to be added to the connection between the wall-mounted equipment and aircraft bulkhead, for vibration damping [2]. However, the mass center of damping system equipment gravely deviates from the stiffness center of absorbers. Consequently, the stiffness coupling will occur during vibration [3], which will enhance the design difficulty.

Nowadays, FEM has been widely applied in engineering [4]. According to the vibration characteristics of wall-mounted vibration damping system, FEM can be used to complete the simulation and analysis of vibration responses at full frequency, and conduct the vibration responses tests with multiple positions and multiple types. Consequently, the paper combines vibration theories and random vibration test results of damping system to predict the absorber parameters. Then Ansys work bench software is used to set up bushing to connect damping elements simulating absorber to simulate FE random vibration. By comparing the simulations results with vibration tests, the absorber parameters can be checked, and accurate equivalent values can be gained. Based on the parametric model, the vibration responses of wall-mounted damping system can be simulated and analyzed with different gravity offset center sand connection positions.
2. **Simplify of model and environment**

2.1. *Introduction to damping system model*

One type of wall-mounted airborne damping system is mainly composed of five parts: airborne electronic equipment, transfer support, large-damping T-shape rubber absorber assembly, hanger’s assembly and bulkhead locking device, shown in Fig. 1. Airborne equipment directly connects with transfer support, composing the loading base. While the rear hanger’s assembly tightly connects with bulkhead through the locking device composing the vibration base. The bases connect with each other through rubber absorber.

![Figure 1. Structural composition of damping system and connection of shock absorber](image)

2.2. *Simplify of test model*

The research priority is vibration response of wall-mounted damping system. To optimize the structure, the loading base is replaced with modal vibration component with higher fundamental frequency. The modal vibration component is supposed to have the same mechanical parameters as the loading base. Similarly, the vibration base is replaced with uniforms with higher fundamental frequency. The uniforms are designed as L-shape structure, as shown in Fig. 2. When vibration table is connected with mounting surface of vibration base, a typical damping system symmetric about four points is composed, as shown in Fig. 3a). When vibration Table relates to the vertical base of vibration Table, a typical wall-mounted damping system is composed, shown in Fig. 3b).

![Figure 2. Construction drawing after optimization](image)

![Figure 3. Two types of installation of vibration table](image)

2.3. *Airborne vibration conditions*

According to what caused vibration excitations, the vibration spectrum of airborne equipment was obtained, shown in Fig. 4.
3. Finite element modeling

3.1. Equivalent modeling of absorber

At present, the modeling method of shock absorber includes three-dimensional modeling and data equivalent modeling. The 3D modeling method can easily generate element expansion self-locking [5]. Hence, the data equivalent modeling method is used in this paper. The modeling of shock absorber was conducted by setting ‘bushing connection’ between the loading base and vibration base [6]. When setting up bushing connecting elements, the connection surfaces between equipment support and hangers are selected as the acting surfaces, for axial damping, shown in Fig. 5.

3.2. FE modeling of damping system

In order to prevent large-deformation elements that will cause iterative calculation divergence, the structures of large curvature like threaded holes and chamfering are deleted. Meanwhile, the computations brought by structural contact need to be reduced to improve efficiency. This can be realized by putting the assembly between the structures with fixed positions. The simplified structure is shown in Fig. 6.

Meshing methods on the platform of Ansys workbench software mainly divide into six types [7]. Sweep meshing divides the highest quality of hexahedrons. The premise behind the method is that divided body has the regular geometry. And the surface meshing can be used to acquire higher quality of hexahedral grid, shown in the Fig. 6. The results show the grids are uniform and smooth. The average value of unit quality factor 0.899 implies the meshing has good quality.
4. Material parameter settings

4.1. Parameter estimation
The paper carries out the test with traditional method. In order to improve the accuracy, two-points diagonal controlling and one-point monitoring are adopted. Specifically, two control sensors are placed on the upper surface of the vibration base where they are diagonal positions with respect to the loading base, while monitoring sensor is placed at the center of upper surface of the loading base, shown in Fig. 7.

![Figure 7. Locations of vibrating sensor](image)

4.2. Parameter calculations
According to relative directions of vibration base in Fig. 7, vibration tests in the X, Y, and Z directions are respectively conducted. Input the acceleration power spectrum density (PSD), then the response acceleration PSD can be measured by monitoring sensor in three directions, shown in Fig. 8.

![Figure 8. Acceleration PSD of loading base](image)

Through analysis, resonant frequency and amplification factor of the damping system in X, Y, Z directions can be concluded, as shown in Table 1:

| Direction | X    | Y    | Z    |
|-----------|------|------|------|
| Resonant frequency/Hz | 52.5 | 52.5 | 52.73 |
| Amplification factor  | 2.74 | 3.11 | 3.6  |

In the testing model, the geometry center of absorber installation location is coincident with the loading base mass center, and the shock absorber is symmetric about the inertial principal axis. Model contains four groups of shock absorber. The whole damping system has the stiffness value of 4k and damping value of 4c [8]. The mass of load base is M. The common formula of vibration Angle frequency and the damping ratio can be given:

\[
\begin{align*}
\omega_n &= \sqrt{\frac{k}{M}} - 2\pi f_n \\
\xi &= \frac{4c}{2\sqrt{k}\cdot 4k}
\end{align*}
\]
The formula can be given of the relationship between amplification factor and damping ratio:

\[ q = \frac{1}{2\varepsilon} \]  

(2)

By solving the above equations:

\[ \begin{cases} k = \varepsilon^2 \frac{n}{M} \\ \varepsilon = \frac{1}{2Q} \end{cases} \]  

(3)

According to Table 1, the axial and tangential stiffness value and damping ratio of absorber can be preliminarily estimated, shown in Table 2.

**Table 2. Initial parameters of stiffness and damping ratio**

| Directions | X         | Y         | Z         |
|------------|-----------|-----------|-----------|
| Stiffness (N/m) | 272030.97 | 272030.97 | 274419.70 |
| Damping ratio \(\xi\) | 0.182     | 0.161     | 0.139     |

4.3. Material parameter settings of other structure models

According to the parameters of shock absorber obtained by the test, the stiffness parameters of X, Y and Z are input to the elastic damping element bushing. In the assembly model of damping system, the specific material parameters of vibration base and loading base are shown in Table 3.

**Table 3. Material parameters vibration base and loading base**

| Materials        | Density/kg/m³ | Poissions ratio | Elasticmodulus/GPa |
|------------------|----------------|-----------------|--------------------|
| Aluminium Alloy  | 2770           | 0.33            | 71                 |
| Structural Steel | 7850           | 0.3             | 200                |

5. FE simulation and verification

According to the installation form of the damping system, the gravity center of the load base is perpendicular to the shock absorber installation axis, while the shock absorber and the vibration base are connected via the excitation input surface. The random vibration in Ansys work bench is simulated by modal superposition method. Considering the influence of gravity in practical vibration environment, this paper adopts pre-stressed modal simulation.

Firstly, statics analysis is conducted. The environment gravity direction is set up to be vertical to the installation axis, to complete the simulation. Next, the modal analysis is carried out. The analysis range is set up to 10-2000Hz, to finish the simulation. Finally, random vibration is analyzed. Use the results of pre-stressed modal analysis as basic parameters, take the fixed constraint surface as input surface of airborne vibration excitation, select the upper surface center of loading base as monitoring point, and preliminarily set the corresponding damping parameters when vibrations are occurring in different directions. The simulation is completed.

Through the simulation, the acceleration response PSD of the monitoring point can be obtained, as shown in Fig. 9(the left). To verify the feasibility of FE simulation, the results are compared with the response PSD of the tests. To improve the experiment accuracy, two-point diagonal control and one-point monitoring are adopted as shown in Fig. 9(the right).

The Fig. 10 displays the comparison between simulation and test, making two conclusions: firstly, the simulation results are basically consistent with the test PSD, with the same number of resonance
peak; secondly, the simulation and test PSD curves for three directions are essentially coincident in low frequency range (within 100Hz), but they have larger deviations in the range of medium-high frequency. The main reason is that the geometric center of shock absorber installation coincides with the mass center of loading base, and the shock absorber is symmetric with respect to the inertial principal axis. But for the damping system, the loading base seriously deviates from the stiffness center of shock absorber. This makes the stiffness coupling in the vibration process.

![Figure 9. Response acceleration PSD of simulation and Sensor installation location](image1)

![Figure 10. Comparison between the simulation and test](image2)

a) Vibration in X direction b) Vibration in Y direction c) Vibration in Z direction

In order to improve the accuracy of simulation results, the bushing connecting element stiffness parameters are corrected (shown in Fig. 11).

![Figure 11. Corrected comparison curves:a)X direction b)Y direction c)Z direction](image3)

From Fig. 11, it can be seen that with corrected parameters of damping system, the simulation and test response curves are coincident, and the response deviations are decreased in the medium-high frequency range. Through analysis of response curves in the graph data, the error values of resonant frequency, amplification factor and vibration level for three directions can be gained, as shown in Table 4. By the table it can be drawn the FE simulation analysis has good feasibility.
Table 4. Error analysis of simulation and test results

| Directions | Test   | Simulation | Errors (%) |
|------------|--------|------------|------------|
| x          | 55.00  | 51.39      | 6.5        |
| y          | 50.00  | 52.73      | 5.46       |
| z          | 48.37  | 52.29      | 8.1        |
| x          | 55.00  | 51.39      | 6.5        |
| y          | 50.00  | 52.73      | 5.46       |
| z          | 48.37  | 52.29      | 8.1        |

6. Vibration analysis

The influence of vibration environment on the damping system of airborne equipment mainly includes large displacement and higher vibration magnitude. The former could cause the collision of the equipment, and the latter could destroy the equipment and reduced the strength of component.

6.1. The influence of gravity center

According to the gravity positions of airborne equipment, the gravity center offset values of 12mm, 16mm, 20mm and 25mm are selected respectively for research. By means of quantitative analysis, the center of gravity is changed only when the installation position, installation spacing, and mass of equipment remain constant. The different gravity centers are gained by changing density of loading base to change its vertical thickness. By FE analysis of four different gravity center offsets for random vibrations, get the maximum displacement response of loading base $L_{max}$ and vibration magnitude RMS, as shown in Table 5. Along the X and Y directions, response levels decrease, and the maximum response displacements increase with the increase of gravity center offset. Along the Y direction, the response levels and maximum displacements both show the decrease trend with the increase of gravity center offset. This is due to the effect of stiffness coupling. For this reason, the balance could be made by adjusting the relative positions between gravity center and installation geometric center.

Table 5. Vibration displacement and vibration magnitude with different gravity centers

| Vibration directions | Response values | 12    | 16    | 20    | 25    |
|----------------------|----------------|-------|-------|-------|-------|
| X                    | RMS/G          | 2.823 | 2.609 | 2.41  | 1.90  |
|                      | $L_{max}$/mm   | 0.070 | 0.072 | 0.0824| 0.118 |
| Y                    | RMS/G          | 2.487 | 2.486 | 2.4864| 2.31  |
|                      | $L_{max}$/mm   | 0.178 | 0.178 | 0.1787| 0.1778|
| Z                    | RMS/G          | 2.839 | 2.627 | 2.4267| 2.13  |
|                      | $L_{max}$/mm   | 0.067 | 0.069 | 0.079 | 0.1189|

6.2. The influence of connection position

In order to study the effects of vertical connection position of installation geometric center relative to equipment gravity center on the vibration response characteristics, the mass center deviations of -16mm, -8mm, 0mm, 8mm and 16mm are chosen, where negative values refer to downward deviation and positive refer to upward. By FE simulation of the damping system with five connection positions, maximum displacement response $L_{max}$ and vibration magnitude RMS of the loading base can be acquired, shown in Table 6. The vibration response results of 0mm are the same as the results when gravity center offsets are 25mm in Table 5.

From Table 6, it can be seen that the vibration response magnitude and maximum displacements increase with the increase of the mass center deviation distances along the X and Y directions. The vibration response along the Z direction shows the opposite characteristic to that along the X and Y.
directions. When upward and downward displacements are same along the Y and Z directions, the vibration magnitudes are close to the maximum displacements. For the vibration response at the medium center, the deviations are the least along the X direction and are the largest along the Y and Z directions. From the data results, the downward connection is better than upward, and the medium connection position has the best damping effects along the X direction.

Table 6. Vibration displacement and magnitude responses with different connection positions

| Response values | -16  | -8   | 8    | 16   |
|-----------------|------|------|------|------|
| X RMS/G         | 2.326| 2.1808| 1.98557| 2.1105|
| Lmax/mm        | 0.136| 0.12466| 0.11915| 0.12426|
| Y RMS/G         | 2.17 | 2.131 | 2.133 | 2.181 |
| Lmax/mm        | 0.273| 0.22521| 0.22504| 0.27406|
| Z RMS/G         | 2.054| 2.10558| 2.11 | 2.05646|
| Lmax/mm        | 0.10166| 0.11242| 0.11241| 0.10161|

7. Summary

To solve the problem that vibration characteristics are difficult to analyze for wall-mounted damping system, this paper puts forward the FEM analysis. The method simulates the damping characteristics by setting up the bushing connection elements, and conducts the vibration test to achieve the estimation of absorber parameters. Then the simulation results and test results are compared to correct the absorber parameters. Finally, the error values of resonant frequencies and amplification factors are within 10% and vibration response levels within 5%, verifying the feasibility of FEM.

Quantitative analysis is then used to study the influence of the gravity center and the shock absorber positions to the vibration characteristics. The results show that, with the increase of the gravity center distance along X or Z direction, the response magnitude decreases, and the maximum response displacement increases. The response magnitude and the maximum response displacement are insignificantly decreasing along Y direction. With the increase of connection center distance, the magnitude of the vibration response and the maximum displacement value increased along X and Y directions. However, the magnitude of the vibration response and the maximum displacement value decrease along the z direction.

In the end, through the research, the horizontal and vertical deviation of installation geometric center with respect to the gravity center should be adjusted, so that the structure damping system can be optimized, and the damping efficiency can be improved. This provides references for the future design for damping system of wall-mounted airborne equipment.

References

[1] Ji, F.Y. and Ji X. Elastic Properties Design of Wall-hung Vibration Isolator [J]. Electro-Mechanical Engineering, 2012, 6 (28): 1-4
[2] Yang, W. F, Wei, Q. and Zhu, L.Q. The design of airborne electronic equipment damping system based on FEM analysis [J]. Vibration and Shock, 2010, 5 (29): 230-234.
[3] Yan, X.L. and Wang, L.C. A Method of the Vibration Decoupling Based on Finite Frequency Domain [J]. Electro-Mechanical Engineering, 2012, 6 (28):1-4.
[4] Zou, G. P. and Liu, Z. Finite Element Simulation of Metal Rubber Damper Random Vibration [J]. Chinese Journal of Mechanical Engineering, 2016, 14 (27): 1960-1963.
[5] Yin, Y. T. and Zhang, B. Finite Element Modeling Method Based on FOG Inertial Navigation System with Dampers [J]. Aviation Precision Manufacturing Technology, 2007, 43 (6): 23-26.
[6] Tian, Y. and Zhang, B. Parameters estimate for damper of strap-down inertial navigation system and stress analysis of optic-fiber rings [J]. Machinery Design & Manufacture, 2011, 1: 233-235.
[7] Pu, G.Y. Basic Tutorials and Examples of ANSYS Workbench [M]. Beijing: China WaterPower Press, 2014.

[8] Ma, S.Q. Damping Design of Onboard Electronic Equipment [J]. Noise and Vibration Control, 2014, 2: 185-187.