Modal and Random Vibration Fatigue Analysis of a Certain Type of EMU Bogie Sanding Device

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Abstract

The EMU bogie sanding device is subjected to random vibration disturbances from the wheel and rail during service, to evaluate the fatigue life, to guide the design selection and structural optimization, combined with the modal calculation and modal test benchmarking, and to verify the finite element. Model, obtain modal parameters, and synthesize random response analysis and fatigue stress assessment Dirlik method, study random vibration fatigue analysis method based on modal analysis, and finally guide the selection of sanding device mounts, the analysis method formed on the EMU The limited life design of the vehicle mounting structure is instructive.

Keywords

Sanding device, Modal analysis,Random response, Fatigue assessment.

1. Introduction

In order to increase the adhesion coefficient between the wheel and the rail of the rain and snow weather, the EMU bogie is equipped with a sanding device. As a cantilever structure, the sanding device bears the load from the wheel and rail during the service phase, and its dynamic performance and fatigue strength directly affect the safety of the structure. The traditional fatigue design method performs static analysis on typical working conditions, extracts the principal stress, calculates the stress amplitude and stress mean, and uses the Goodman fatigue limit diagram to evaluate the fatigue strength of the structure to find the weak area. The sanding device is subjected to broadband random vibration loads during train operation, and its dynamic response should be considered in fatigue analysis. Therefore, the random vibration fatigue analysis of the sanding device should be divided into two stages: random response analysis and fatigue strength assessment. Random response analysis usually includes both time domain and frequency domain methods. The time domain method is to input random load data (usually acceleration) in the time domain, and the output stress response is expressed in the form of time history; the frequency domain method usually uses acceleration power spectrum. The excitation load is input in the form of density, and the stress power spectral density of the output is statistically evaluated for fatigue strength.

2. Modal Analysis

The bogie sanding device is mounted on the axle box body by four bolts, and the end is equipped with heating devices and pipelines. As shown in Fig. 1, in order to verify the parameters of the simulation model and obtain the damping parameters of the structure, the modalities are first developed separately. Calculation and modal test.

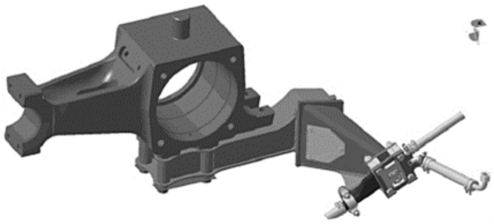


Fig 1. Schematic diagram of sanding device for a certain type of EMU bogie

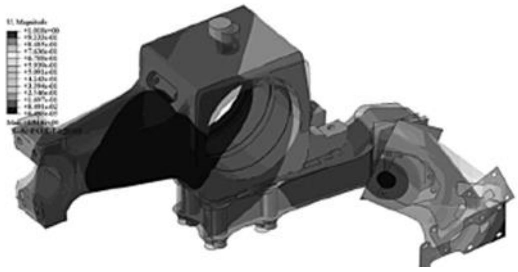
The modality is an intrinsic property of the structure. The modal parameters such as the natural frequency and mode shape of the structure can be obtained by calculation or experimental analysis. In the case of ignoring the sniffer, the modal parameters of the structure can be solved by:.

$$([K]-W^2[M])\{\varphi\}=0 \tag{1}$$

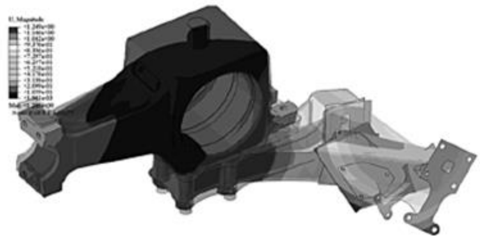
Where K is the stiffness matrix, M is the mass matrix, ω is the natural frequency, and φ is the modal matrix.

The modal calculation is carried out by ABAQUS software, the main structure is S4 unit and C3D8R unit; the bolt between the mounting arm and the bracket is simulated by the B31 unit; the axle box bearing is between the axle box and the axle, and the two are simulated by the Hinge connection. The rotation relationship between the two; use Busing to connect the analog boom node and set its stiffness. The modal test is carried out by means of a small hammer hammering excitation method on the sanding device of the current vehicle installation state, and the acceleration sensor is arranged according to the vibration mode result calculated by the modality.

Fig. 2 shows the calculation results of the first two modes, the frequencies are 156.72 Hz and 341.37 Hz, respectively. The vibration modes are the vertical swing of the sanding device and the lateral torsion. Fig. 3 shows the measured results of the first-order mode. The frequency is 175.7 Hz, the vibration mode is the vertical swing of the head; Fig. 4 is the comparison of the calculation and test frequency of the first four modes. As shown in the figure and Table 1, the second-order frequency result has the largest error (12.57%), the third-order frequency error is the smallest (0.17%). The measured result can be used as the target value, and the parameters such as the mass and elastic modulus of the finite element model can be modified to reduce the error between the modal calculation result and the measured result. In addition, the damping parameters of the structure were obtained by modal test, as shown in Table 1.



(a) First order mode156.72Hz



(b) Second order mode156.72Hz

Fig 2. Calculation results

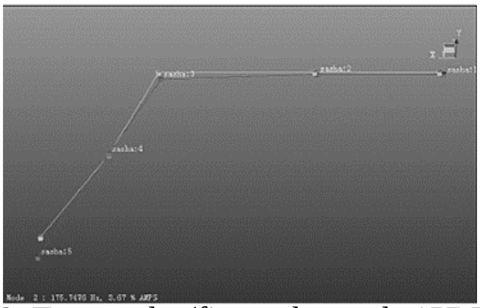


Fig 3. Test results (first-order mode 157.5Hz)

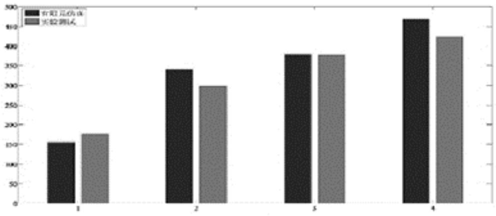


Fig 4. Comparison of finite element simulation and measured results

Table 1 First four-order frequency contrast damping coefficient

Frequency/Hz	1 order	2 order	3 order	4 order
Simulation calculation	156.72	314.37	378.64	468.56
Measured	175.7	298.45	378	424.08
Error/%	10.8	12.57	0.17	9.49
Damping coefficient/%	3.58	2.34	1.69	4.62

3. Random Response Analysis

For a structure with N degrees of freedom, the dynamic equation in the frequency domain is under the action of the random load y(t):

$$(-W^2[M]+jwC+[K])X(w)=Y(w) \tag{2}$$

Where C is the damping matrix, X(ω) is the displacement response vector in the frequency domain, and Y(ω) is the load vector in the frequency domain of y(t).

The modal superposition method is used to analyze the random response of the sand blasting device. Considering the vibration response under the condition of self-weight and bolt pre-tightening, the analysis steps are static strength calculation, modal calculation and random response calculation. Firstly, the influence of gravity and bolt preload is considered in the static strength calculation, then the modal calculation considering prestress is carried out, and finally the random response calculation is based on the modal superposition method.

Fig. 5 shows the acceleration power spectral density curve specified by the impact vibration test standard for IEC 61373-2010 rail vehicle installation equipment commonly used in the railway industry, taking f2=500Hz, X=124.9(m/s2)2/Hz. In the ABAQUS software, enter the acceleration power spectral density curve shown in Fig. 5, add the measured critical damping coefficient, and obtain the dynamic stress response of the sanding device. Figure 6 shows the round hole edge of the mounting arm (Fig. 7 rms Mises stress) The power spectral density curve of the Mises stress in the frequency domain in the maximum position. It can be seen from the figure that there is a peak near 160 Hz. This is because the sanding device has a vibration mode of the first-order head vertical swing at 156.72 Hz. The acceleration excitation has the largest amplitude in the frequency range of 20-100 Hz, and rapidly decreases beyond 100 Hz. The first-order frequency is 156.72 Hz close to 100 Hz, and the second-order frequency is about 300 Hz. Therefore, only one peak appears in Fig. 6.

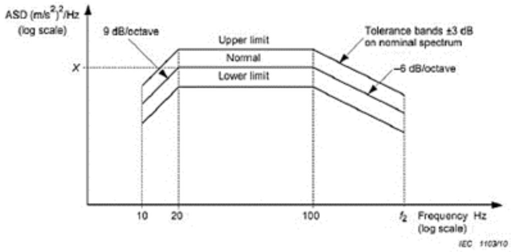


Fig 5. Standard acceleration power spectral density curve

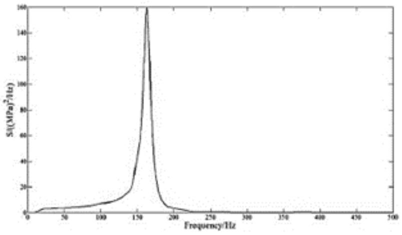


Fig 6. Stress spectrum density of the mounting arm seat

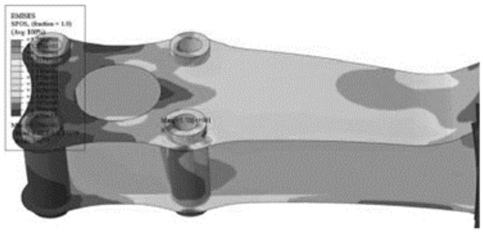


Fig 7. RMS root of the mounting arm of the sanding device Mises stress cloud

4. Fatigue Evaluation

The Dirlik function is an empirical formula describing the characteristics of random vibration stress distribution and is commonly used for the evaluation of random vibration fatigue stress. The Dirlik probability density function can be expressed as:

$$([K]-W^2[Mp(s)=\frac{\frac{D_1}{Q}\exp(-\frac{Z}{Q})+\frac{D_2Z}{R^2}\exp(-\frac{Z^2}{2R^2})+D_3Z\exp(-\frac{Z^2}{2})}{2(m_0)^{1/2}} \tag{3}$$

In the middle: $D_1=\frac{2(x_m-r^2)}{1+r^2}$, $D_2=\frac{1-r-D_1+D_1^2}{1-R}$, $D_3=1-D_1-D_2$, $Q=\frac{1.25(\mathrm{r}-D_3-D_2R)}{D_1}$, $r=\frac{m_2}{\sqrt{m_0m_4}}$, $x_m=\frac{m_1}{m_0}\sqrt{\frac{m_2}{m_4}}$, $R=\frac{r-x_m-D_1^2}{1-r-D_1+D_1^2}$, $z=\frac{s}{2\sqrt{m_0}}$, $m_0=\int_{-\infty}^{+\infty}G(f)f^n df$.

s is the stress range, $G(f)$ is the stress power spectral density, which can be derived from the random response analysis, and m_n is the n order spectral moment of the stress power spectral density.

According to Miner's linear cumulative damage theory, the fatigue damage of P(s) in time T is:

$$D_T=\alpha T\int\frac{P(s)}{\mathrm{N}(s)}ds \tag{4}$$

D_T is the fatigue damage in time T, $\mathrm{N}(s)$ is the fatigue curve, and α is the cycle number of the rain flow cycle unit time count, the value is $\alpha=\sqrt{\frac{m_4}{m_2}}$.

Fig. 6 shows the stress power spectral density curve of the part of interest. When the material of the mounting arm is Q235, the fatigue damage value DT=5.6274>1.0 can be obtained from the S-N curve of the material of formula (3) and Q235. The IEC 61373-2010 standard uses a 5-hour long-life random vibration test to simulate the service life of a train installation equipment for 20 years. The time for the crack damage can be 0.8885h by DT reverse thrust; according to the test result, when the time T=1 Crack occurs at .2h, so the calculated value is relatively conservative; when the material of the mounting arm is Q345, the fatigue damage value DT=0.8999<1.0 can be obtained from the S-N curve of the materials of formula (3) and Q345. No fatigue cracks occur during the design life. Therefore, Q345 material should be used for the mounting arm.

5. Conclusion

The dynamic performance and fatigue strength of the sanding device directly affect the safety of the structure. The modal calculation method and the modal test can be used to check the finite element calculation model, and the damping ratio and other parameters are obtained as the random method based on the modal superposition method. According to the input parameters of the analysis, the vibration response of the structure under the action of random acceleration is calculated, and then the fatigue intensity is evaluated by the Dirlik probability density function, which can effectively support the design selection and structure optimization. The formed analysis process is designed for the limited life of the vehicle installation equipment. It has reference significance.

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