Analysis of Vibration Fatigue of A Subway Shunting Train

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Abstract. In order to research the fatigue life of a subway shunting train, the power spectrum density (PSD) is used to calculate the vibration fatigue based on frequency domain analysis. Through pre-stressed modal analysis in finite element simulation, the natural frequency and vibration mode are obtained. Then harmonic response is used to get the frequency response function (FRF) based on the modal superposition method. Then the solution is imported into Ncode Designlife. After using the Lalanne method to process PSD curve and the Goodman method to correct the mean stress of the estimated S-N curve, the vibration fatigue damage of the shunting train is obtained, so as to evaluate its vibration fatigue. The result will provide a certain theoretical basis for the fatigue study of shunting train.

1. Introduction

The subway shunting train is used for the traction in different conditions [1]. Under long-term cyclic load, the car body will produce fatigue fracture. Sung II SEO et al. established a method for measuring the dynamic load of the body of a subway car, and concluded that it is inaccurate to directly use the structural static strength test results to study the structural fatigue results[2]. Ning Xie calculated the fatigue damage of the car body at different speed levels through modal analysis[3]. Matjaž et al. compared the advantages of frequency-domain fatigue life analysis methods over time-domain analysis methods, especially in numerical calculations [4]. Xing Zhang calculated each response spectrum according to the track irregularity spectrum in the standard, and calculated the frequency response function of each response of the dynamic model to obtain the car-body vertical load spectrum [5]. Zhan L et al. analysed the influence of modal stress on the fatigue strength of the framework based on the standard refractive index and high-frequency vibration conditions [6].

The above literature mainly considers the influence of vibration mode on the fatigue life of high-speed trains, but the literature considering the shunting train body is relatively inadequate which is especially adopt the overall bearing. Therefore, the method can evaluate the life damage of the shunting train body based on frequency domain analysis properly and accurately.

2. Vibration fatigue analysis method

2.1. Frequency domain analysis method

Frequency-domain fatigue calculations are used to calculate the fatigue life through power spectral density stress[7]. The i-th spectral distance is defined as:

$$m_i = \int_0^{+\infty} f G(f) df$$  \hspace{1cm} (1)

Where $f$ is vibration load frequency and $G(f)$ is the power spectral density of the vibration load.

According to Miner’s linear cumulative damage theory, the fatigue damage of the structure is:
\[ D = \sum D_i = \sum \frac{n_i}{N_i} \quad \text{(2)} \]

Where \( n_i \) is the number of cycles under stress level \( S_i \) and \( N_i \) is the fatigue life under stress level \( S_i \).

For continuous state, number of stress cycles in the stress range \((S_i, S_i + \Delta S_i)\) during time \( T \) is:

\[ n_i = E(P)TP(S_i)\Delta S_i \quad \text{(3)} \]

Where \( E(P) \) is the expected value of the peak frequency signal immediately responding to the signal, \( T \) is the random response time and \( P(S_i) \) is the probability density function of the stress amplitude \( S_i \). The S-N curve of material can be described as

\[ N(S) = kS_i^{-b} \quad \text{(4)} \]

Where \( k \) and \( b \) is material constant.

From equation (2) to (4), it is obtained that

\[ D = \frac{E(P)}{k} \int_0^{+\infty} P(S)dS \quad \text{(5)} \]

2.2. Mean stress correction

For the S-N curve, it is limited by conditions in most cases, and curves of different mean stresses cannot be obtained. So, the mean stress correction is required.

This analysis uses the Goodman correction method and it can be described as

\[ \frac{S_a}{S_e(R=-1)} + \frac{S_m}{UTS} = 1 \quad \text{(6)} \]

Where \( S_a \) is stress amplitude under actual working conditions of materials; \( S_m \) is mean stress under actual working conditions of materials; \( UTS \) is the ultimate tensile strength of the material and \( S_e \) (R=-1) is stress amplitude when the stress value is equal to -1.

2.3. Lalanne theory

Lalanne theory can deal with a universal broadband frequency load processing technology and is also a standard used in many industries. The Ncode Designlife platform uses the Lalanne theory for fatigue calculation by default for the generated PSD [8].

\[ p(S) = \frac{1}{2rms} \left\{ \sqrt{\frac{1-y^2}{2\pi}} e^{-\frac{S^2}{2rms^2(1-y^2)}} + \frac{S\cdot\gamma}{4rms} \frac{S^2}{\epsilon^2rms^2} \left[ 1 + \text{erf} \left( \frac{S\cdot\gamma}{2rms\sqrt{2(1-y^2)}} \right) \right] \right\} \quad \text{(7)} \]

Where

\[ \text{erf}(x) = \frac{2}{\sqrt{\pi}} \int_0^x e^{-t^2} dt \quad \text{(9)} \]

3. Establishing of finite element model

3.1. Establishing of 3D model

The body of this type of shunting train is an integral load-bearing structure. The integral load-bearing structure has higher strength because its side walls and roof can share a considerable load compared to the underframe load-bearing structure.

Vibration fatigue analysis requires reasonable simplification of the model. Then the simplified 3D model is created in CREO. The model after meshing is shown in the Figure 1. The nodes number is 394581 and the units number is 159307.

The materials of the overall structural components of the locomotive, including the underframe, cab, and side walls, are all Q345E. Its density is 7850kg / m³, the yield limit is 345MPa, the Young's modulus is 206GPa, and the Poisson's ratio is 0.28. Create the new material in the database based on the parameters.
3.2. Applied load and boundary conditions

This analysis is carried out under vertical load conditions. According to standard [9], 1.3 times of gravity acceleration is required for vertical load conditions. The specific load conditions are in Table 1.

| Component                        | Mass (kg) | Weight (N) | Area (m²) | Load (MPa) |
|----------------------------------|-----------|------------|-----------|------------|
| Console                          | 100       | 1274       | 1.92      | 0.00066    |
| Electrical cabinet               | 90        | 1146.6     | 0.344     | 0.00333    |
| Generator                        | 356       | 4535.44    | 0.105     | 0.04319    |
| Power pack                       | 4647.4    | 59207.87   | 0.71      | 0.08339    |
| Water tank                       | 54.7      | 696.878    | 0.0128    | 0.05444    |
| Diesel engine set                | 165       | 2102.1     | 0.0182    | 0.1155     |
| Battery box                      | 360       | 4586.4     | 0.0612    | 0.0749     |
| Air compressor                   | 165       | 2102.1     | 0.3828    | 0.00549    |
| Total air cylinder               | 217       | 2764.58    | 0.01      | 0.276      |
| Cooling steel                    | 1615      | 20575.1    | 0.139     | 0.148      |
| Fuel tank                        | 690.7     | 8799.51    | 0.12      | 0.0733     |
| Cabs and side walls at both end  | 7365.74   | 93839.52   | 1.75      | 0.0536     |
| Side wall roof                   | 1969.6    | 25092.7    | 4.83      | 0.00520    |
| End plate counterweight          | 309       | 3936.66    | 0.32      | 0.0123     |
| Generator counterweight          | 395       | 5032.3     | 0.42      | 0.0120     |
| Main connecting beam counterweight| 285       | 3630.9     | 0.484     | 0.00750    |
| Counterweight for two-end driver's cab | 1000 | 12740   | 2.771 | 0.00460 |

Under the vertical load condition,, the bottom frame above the secondary spring of the bogie is selected as the boundary condition, and its six degrees of freedom are completely fixed.

4. Vibration fatigue analysis

To analyse vibration fatigue of the car body, the pre-stressed modal analysis is conducted on the model to obtain natural frequency and vibration mode based on ANSYS. Then, the harmonic response analysis is conducted to obtain the frequency response function. The result is imported into Ncode Designlife for calculating the fatigue damage of the car body.
4.1. Modal analysis
The modal analysis is conducted on the car body, and the obtained modal natural frequencies are shown in the Table 2 and the 6 mode graphs are shown in Figure 2.

![Modal Shapes](image)

Figure 2. Modal shape of first 6 modes

| Mode | Frequency (Hz) |
|------|----------------|
| 1    | 11.839         |
| 2    | 13.701         |
| 3    | 20.896         |
| 4    | 21.469         |
| 5    | 23.118         |
| 6    | 23.282         |

4.2. Vibration fatigue simulation
According to GMW3172_AUG2015[10], the acceleration frequency spectrum of random vibration is shown in the Table 3. Input the appropriate frequency range limited by the natural frequency of the car body which is obtained from the modal analysis. In Ncode Designlife, the estimated S-N curve of Q345E can be obtained according to the material's Young's modulus, Poisson's ratio and maximum UTS, as shown in the Figure 3.
Goodman method is used to correct S-N curve of material. The following damage cloud diagram (Figure 4) is obtained through fatigue calculation.

Table 3. Power spectrum density of random vibration

| Frequency (Hz) | 10  | 55  | 180 | 300 | 360 | 1000 |
|---------------|-----|-----|-----|-----|-----|------|
| PSD (G²/Hz)   | 0.1032 | 0.0336 | 0.0013 | 0.0013 | 0.0007 | 0.0007 |

5. Conclusions
The results of the vibration fatigue analysis state that:

- The first-order frequency is lower than 10 Hz which meets the requirements.
- According to the modal graphs, modes are mostly concentrated in 10-25 Hz. The relevant mode is mainly concentrated in the side wall. Due to the overall bearing method, the side wall will also bear the load. So it is the focus and improvement part in the later period.
- From the damage cloud, the maximum damage occurs at node 34097 shown in Figure 5. It is located at the first beam of the top cover which bear the roof load. The maximum damage is $8.33 \times 10^{-6} < 1$. According to the Miner principle, the fatigue fracture doesn’t occur [11].
- The maximum RMS stress is 75.9 MPa which is also occurred at node 34097. And it is lower than the maximum allowable stress which is 207 MPa.
- Other nodes with greater damage are concentrated in the windows and doors of side wall and the welds of the chassis.

After calculation, the fatigue strength of this design meets the requirements. The body structure can withstand all loads without causing permanent deformation. The research results provide a certain theoretical basis for the design of the subway internal combustion shunting train.
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References
[1] Shuxian Zhao, Dan Y and Li L 2016 Overall design of GCY470 shunting locomotive *Railway locomotive and motor Car* 4:0018-22.
[2] Sung II SEO, Choon S P, Ki H KIM, Byung C S and Oak K M 2005 Fatigue strength evaluation of the aluminium car body of urban transit unit by large scale dynamic load test *JSME International Journal* 48(1):27-34
[3] Ning X 2015 *Analysis of vibration modal influence on fatigue life for high speed train car body* (Chengdu: Southwest Jiaotong University)
[4] Matjaž M, Janko S and Miha B 2013 Frequency-domain methods for a vibration-fatigue-life estimation application to real data *International Journal of Fatigue* 47(2): 8-17.
[5] Xing Z 2018 *Research on vibration fatigue analysis method of high-speed train car body under random loads* (Chengdu: Southwest Jiaotong University)
[6] Zhan L, Yuanyuan F, Guohui L and Dachun Y 2020 Comparative analysis on fatigue strength of high speed EMU bogie frame based on standard load working condition and high frequency vibration working condition *Urban Mass Transit* 23(02):133-136
[7] Deyong Li and Weixing Y 2011 Nominal stress approach for prediction of notched specimens under vibration loading *Acta Aeronautica et Astronautica Sinica* 32(11):2036-2041
[8] Kumar S M 2008 Analyzing random vibration fatigue *ANSYS Advantage* 11 (3):39-42
[9] TB/T 2541-2010 *Standard for Testing the Static Strength of Locomotive Car Body*
[10] GMW3172_AUG2008 *General specification for electrical functional components*
[11] Miner M A 1945 Cumulative damage in fatigue *Journal of Applied Mechanics Review* (3):159-164