Vibrational fatigue failure prediction of a brake caliper used for railway vehicles based on frequency domain method

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Abstract: A brake caliper used for railway vehicles was introduced in this paper. In order to predict the fatigue failure under random vibration condition, firstly a dynamic model was established and proved by comparing the results of modal simulation and modal test. The weak points of the structure were identified by taking random response analysis, and the corresponding PSD (power spectral density) spectrums of stress were obtained. Furthermore three different PDF (probability density function) models of stress amplitude distribution were employed to describe the distributions of stress at weak points, then the fatigue damages were calculated based on Miner’s linear cumulative damage theory and the fatigue failures were predicted. Finally, an accelerated failure test was designed and carried out to verify the validity of the predictions proposed by three models. The result shows that, the Dirlik’s model and Steinberg’s model can provide more exact results and be suitable for vibration fatigue failure prediction of brake caliper.

1. Introduction

Foundation braking equipment, which provides the functions of stopping brake, decelerating brake, continuous brake and stationary brake, is one of the indispensable components of railway vehicles’ braking system. Since it could make the vehicle stopped independently when all other braking devices are out of service, foundation braking equipment is commonly considered as the safest braking way for railway vehicles [1].

When the train is running, especially in the cases of track irregularity, curve negotiation, or passing by a turnout, the wheel-rail excitation will result in the vibration of bogie and vehicle body. Furthermore, due to the elastic structures of brake pads and disc, stick-slip occurs on the friction surfaces while braking, which will lead to self-excited vibration of foundation braking equipment. Thus, the foundation braking equipment is always under vibration in service [2-4].

Brake caliper is designed as the actuator in foundation braking equipment, and it has a lower structural stiffness than brake pad and disc. It makes brake caliper be more affected by vibration, especially the long-time random vibration, will easily lead to structural fatigue failure and bring serious risk for the vehicle operation. Therefore, it is very significant to investigate the probable failure mode and make a prediction of failure for brake caliper under random vibration.
The existing studies mainly provide two approaches for investigating the structural fatigue under random vibration, including time domain method and frequency domain method [5]. The time domain method takes loading course into consideration to obtain the stress-time history, which helps to bring a more precise result. Such as the rain-flow method, is considered as the most accurate counting method. However, enormous data of load and response samples are required and it makes the computation inefficient [6]. In frequency domain method, PSD (power spectral density) is employed to provide a statistical representation for random vibration process [7,8]. By making use of some certain distribution models, such as Bendat’s [9] model, Wirsching’s [10] model, Dirlik’s [11] model, and Steinberg’s model [12], PSD of stress responses can be translated into PDF (probability density function) of stress amplitude distribution, by use of which the fatigue damage of structure can be estimated based on Miner’s linear cumulative damage theory. This greatly improves the computational efficiency and makes the frequency domain method widely used in fatigue damage analysis.

In this paper, a brake caliper used for railway vehicles is introduced, and its structural dynamic model for random vibration analysis is built and verified by modal test. Firstly the weak points of the brake caliper structure are identified by random response analysis and the PSD of stress responses are obtained. Then the vibrational fatigue damage is estimated by Miner’s linear cumulative damage theory and different PDF models of stress amplitude distribution. Finally, an accelerated failure test is taken to prove the validity of the proposed method for vibration fatigue failure prediction of brake caliper.

2. Stress response analysis under random vibration

2.1. Dynamic model for random vibration analysis

The structure of a common brake caliper used for railway vehicles is shown in Fig.1, it consists of two modules: brake cylinder and caliper mechanism. Brake cylinder provides an axial thrust by compressed air. Caliper mechanism, which is assembled from frame, levers, axis pins, pad holders, hangers, hanger bolts, rubber joint, etc., is driven by brake cylinder and promotes brake pads against the rotating brake disc to provide a frictional torque.

The frame and levers amplify the thrust force generated by brake cylinder, this makes levers and frame bearing the maximum load while braking. In addition, brake caliper is mounted on the bogie frame by several suspension points (hanger bolts and rubber join), through which the vibration load transmitted from the bogie to brake caliper, so the components levers and frame should be affected by the vibration load comes from the bogie directly. It will be much more likely to fail on levers and frame than other components.

Fig.2 shows the FEA dynamic model of brake caliper. In order to obtain a higher computational efficiency, some simplifications are introduced into the FEA model as below

- As the brake cylinder has a compact structure with good stiffness compared with caliper mechanism, it can be equivalent substituted by two inertia mass points, which are coupled to the end of each side of lever respectively. Each inertia mass is set to 25kg.
- Set up a cylindrical connection between two inertia mass points with an axial stiffness of 60000N/mm.
- Set up a hinge connection between the components of each revolute pair in the caliper mechanism.
- Set up vertical and horizontal elastic connections between pad holders and the ground to make an approximate representation of the friction interface between brake pads and brake disc. The stiffness of each elastic connection is set to 20000N/mm.
- Ignore the non-critical components that have less contribution to dynamic performance of the structure.
The material properties of main components of the FEA model are shown in Tab. 1.

| Components    | Materials | Density (kg/m$^3$) | Elasticity modulus (GPa) | Poisson ratio |
|---------------|-----------|--------------------|--------------------------|---------------|
| Frame         | QT600-7   | $7.25 \times 10^3$ | 169                      | 0.28          |
| Lever         | QT600-7   | $7.25 \times 10^3$ | 169                      | 0.28          |
| Pad holder    | QT500-7   | $7.12 \times 10^3$ | 162                      | 0.28          |
| Hanger        | QT500-7   | $7.12 \times 10^3$ | 162                      | 0.28          |
| Axis pin      | 42CrMo    | $7.85 \times 10^3$ | 206                      | 0.30          |
| Hanger bolt   | 42CrMo    | $7.85 \times 10^3$ | 206                      | 0.30          |

2.2. Validation of dynamic model

In order to verify the validity of the FEA dynamic model proposed above, modal analysis is carried out by simulation and modal test (see in Fig. 3 and Fig. 4). The results of simulation and test are shown in Tab. 2.

From Tab. 2 one can see that, four out of five modals of simulation result can match with the first four order modals of test respectively by comparing the modal shapes, although there are some acceptable deviations in frequency. In general, the FEA dynamic model is available for vibration analysis.
Table 2. Comparison between simulation result and test result of modal analysis

| Order | Simulated frequency (Hz) | Tested frequency (Hz) | Description of modal shape                      |
|-------|--------------------------|-----------------------|-------------------------------------------------|
| 1     | 27.54                    | 19.36                 | Synchronous oscillation in longitudinal          |
| 2     | 35.87                    | 31.47                 | Synchronous oscillation in transverse            |
| 3     | 48.76                    | 56.07                 | Synchronous oscillation in vertical             |
| 4     | 62.41                    | /                     | Synchronous oscillation in transverse and longitudi nal |
| 5     | 87.14                    | 102.43                | Alternating oscillation in vertical             |

2.3. Stress response analysis

The mechanical, pneumatic, electrical and electronic equipments/components fitted on to railway vehicles are required to satisfy the demands of random vibration and shock test. Especially the random vibration test is very significant to examine the service durability under long-time vibration for the equipments/components.

In order to accelerate the failure exposure of brake caliper, the simulated long-life testing condition of category 3 specified in the standard IEC 61373 is introduced in this paper as the analysis condition of random vibration. The ASD (Acceleration spectrum density) of the vibration load is defined as Fig.5.

By random response analysis, the distribution of RMS (root-mean-square) mises stresses of the main bearing components (frame and levers) can be obtained, and the part of maximum RMS mises stress of each component is identified, see in Fig.6. These parts are considered as the weak points of the brake caliper structure, where the fatigue failures tend to be occurred.
3. Vibration fatigue failure prediction

3.1. PSD analysis of stress responses

In order to estimate the fatigue damage and predict the structure failure of brake caliper under random vibration, the PSD spectrums of stress responses are required. Fig. 7 shows the PSD spectrums of mises stress at the weak points on levers and frame.

![Figure 7. PSD spectrums of mises stress in weak points](image)

Define the relationship between PSD of mises stress and frequency as \( G(f) \), which can be obtained by the results shown in Fig. 7. One can get the corresponding bandwidth coefficient \( \varepsilon \) as below

\[
\varepsilon = \sqrt{1 - \frac{m_i^2}{m_0 m_4}}
\]  

(1)

in which \( m_i (i=0, 2, 4) \) is defined as \( i \)-order spectrum moment of \( G(f) \) and can be obtained by

\[
m_i = \int_0^\infty f^i G(f) df
\]  

(2)

While \( \varepsilon \) tends to zero, the response of stress is approximate to a narrow-band process with single frequency; inversely, while \( \varepsilon \) tends to 1, the response of stress is approximate to a wide-band process as white noise.

Denote the expectation of the times of crossing zero with positive slope per unit time as \( E_0 \), denote the expectation of numbers of peaks appears per unit time as \( E_P \), and they can be obtained by

\[
E_0 = \frac{m_1}{m_0}, \quad E_P = \frac{m_4}{m_2}
\]  

(3)

For a random process of narrow-band, \( E_0 \) is close to \( E_P \).

Substitute the PSD spectrums shown in Fig. 7 into Eqs. (1) and (2), then the bandwidths of the stress responses in the weak points on levers and frame can be obtained, see in Tab. 3.

| Direction Components | Transverse Lever | Vertical Lever | Longitudinal Lever | Transverse Frame | Vertical Frame | Longitudinal Frame |
|----------------------|-----------------|----------------|-------------------|-----------------|-----------------|-------------------|
| \( \varepsilon \)     | 0.47            | 0.58           | 0.45              | 0.38            | 0.27            | 0.58              |

3.2. PDF model of stress amplitude distribution

The PSD spectrum could not describe the stress-time history, so it is required to translate the PSD function \( G(f) \) into PDF function \( P(S) \).

In references [5] and [6], several PDF models of stress amplitude distribution are compared, and the result indicates that, Bendat’s model is fit for the random process whose bandwidth is smaller than 0.3; Dirlik’s model is fit for the random process whose bandwidth is greater than 0.5. From Tab.3 one
can see that bandwidths of PSD spectrums at some weak points are between 0.3 and 0.5, so different PDF models are employed to describe the distribution of stress amplitude.

The PDF model of stress amplitude distribution proposed by Bendat can be represented as

$$P(S) = \frac{S}{\sigma} \exp\left(\frac{-S^2}{2\sigma^2}\right)$$

(4)

where

$$\sigma = \sqrt{m_0}$$

And the PDF model of stress amplitude distribution proposed by Dirlik can be represented as

$$P(S) = \frac{1}{2\sqrt{m_0}} \left[ \frac{D_1}{Q} \exp\left(\frac{-Z}{Q}\right) + \frac{ZD_2}{R^2} \exp\left(\frac{-Z^2}{2R^2}\right) + \frac{ZD_3}{2} \exp\left(\frac{-Z^2}{2}\right) \right]$$

(5)

where

$$\gamma = \sqrt{1-\delta^2}$$

$$D_1 = \frac{2(x_m - \delta^2)}{1+\gamma^2}$$

$$D_2 = \frac{1-\gamma-D_1+D_1^2}{1-R}$$

$$D_3 = 1-D_1-D_2$$

$$Z = \frac{S}{2\sqrt{m_0}}$$

$$x_m = \frac{m_1}{m_0} \sqrt{\frac{m_2}{m_3}}$$

$$R = \frac{\gamma - x_m - D_1^2}{1-\gamma-D_1+D_1^2}$$

$$Q = \frac{1.25(\gamma-D_1-D_2)}{D_1}$$

Substitute the PSD spectrums of stress responses into Eqs.(4) and (5), then the PDFs of stress amplitude distribution at weak points by using Bendat’s model and Dirlik’s model can be obtained respectively, see in Fig.8 and Fig.9.

![PDFs of stress amplitude distribution at weak points obtained by Bendat’s model](image1)

(a) Weak point on lever  (b) Weak point on frame

Figure 8. PDFs of stress amplitude distribution at weak points obtained by Bendat’s model

![PDFs of stress amplitude distribution at weak points obtained by Dirlik’s model](image2)

(a) Weak point on lever  (b) Weak point on frame

Figure 9. PDFs of stress amplitude distribution at weak points obtained by Dirlik’s model

Besides, reference [12] introduces a PDF model proposed by Steinberg. This model considers the stress amplitude to satisfy the Gaussian distribution, the probabilities that the instantaneous stress val-
ue between 0 to 1σ, 1σ to 2σ, and 2σ to 3σ are 68.26%, 27.18%, and 4.30%, respectively. And the probability that the instantaneous stress value beyond 3σ is ignored. This model divides the whole distribution of stress amplitude into three discrete regions and provides a simplified and convenient method for engineering application.

3.3. Evaluation of fatigue damage

After obtaining the PDF of stress amplitude distribution \( P(S) \), one can estimate the fatigue damage results from the random vibration by means of Miner’s linear cumulative damage theory. In this theory, the stress amplitude of random vibration can be dispersed into several intervals \( (S_i, S_i + \Delta S) \), the corresponding cycle number \( n_i \) and cumulative damage \( D_i \) can be obtained by

\[
D_i = n_i / N_i \tag{7}
\]

In which \( T \) denotes duration time of random vibration process; \( N_i \) denotes the cycle life corresponds to \( S_i \), which can be obtained from the fatigue-life curve of material. For common metal material, the relationship between stress \( S \) and fatigue-life \( N \) can be expressed as

\[
NS^m = C \tag{8}
\]

In which \( m \) and \( C \) are the constants related to material. For QT600-7, which is the material of levers and frame, assign \( m \) as 10.51 and \( C \) as \( 1.77 \times 10^3 \) according to experimental data.

The total fatigue damage of random vibration process \( D \) can be estimated as below

\[
D = \sum \frac{n_i}{N_i} \tag{9}
\]

It is determined as failure while \( D \geq 1 \).

For Bendat’s and Dirlik’s models, by substituting Eqs.(6)-(8) into Eq.(9), the total fatigue damage can be obtained by

\[
D = \frac{E_p T}{C} \int_0^\infty P(S) \cdot S^m dS \tag{10}
\]

And for Steinberg’s model, the total fatigue damage can be expressed as

\[
D = \sum \left( \frac{n_i}{N_i} \right) = \frac{E_p T}{C} \sum p_i S_i^m \tag{11}
\]

In which, \( p_1=68.26\% \), \( p_2=27.18\% \) and \( p_3=4.30\% \).

By Eqs.(10) and (11), the cumulative fatigue damages at weak points on levers and frame can be obtained by three models mentioned above respectively, see in Tab.4.

| Model   | Transverse | Vertical | Longitudinal |
|---------|------------|----------|--------------|
|         | Lever      | Frame    | Lever        | Frame    | Lever   | Frame    |
| Bendat  | 3.97×10^-2 | 7.60×10^-4 | 6.03×10^-1 | 7.32     | 9.67×10^-8 | 1.27×10^-7 |
| Dirlik  | 1.30×10^-1 | 2.33×10^-3 | 2.01        | 2.58×10^1 | 3.40×10^-7 | 3.76×10^-7 |
| Steinberg | 1.01×10^-1 | 1.94×10^-3 | 1.54        | 1.87×10^-1 | 2.32×10^-7 | 3.24×10^-7 |

From Tab.4 one can see that the levers and frame will not fail under the transverse and longitudinal random vibration conditions (which are defined in Fig.5) for 0.5 hour, the cumulative fatigue damages estimated by any of the three models are much less than 1. However, under the vertical random vibration conditions, the results obtained by Dirlik’s and Steinberg’s models indicate the structure failure occurred on both levers and frame, but the Bendat’s model only predicts the failure of frame. This shows that Dirlik’s and Steinberg’s models can make approximate predictions.
4. Experimental verification
In order to verify the validity of the predictions result from the three models mentioned above, an accelerated failure test is carried out according to the random vibration conditions proposed in Fig.5. After 0.5 hour transverse and longitudinal random vibration tests, the test sample of brake caliper does not occur any failure in structure. After 0.5 hour vertical random vibration test, fractures appear on both levers and frame, and the fracture locations are very close to the weak points predicted above, see in Fig.10.

![Fracture on Lever](image1)
![Fractures on Frame](image2)

Figure 10. Structure failures of brake caliper after accelerated failure test

By comparison of the predicted result and experimental result, one can find that Dirlik’s model and Steinberg’s model can make the failure predictions be more accordance with testing result.

5. Conclusions
In this paper, the structure fatigue failures under random vibration of a brake caliper used for railway vehicle are investigated in frequency domain by using three stress amplitude distribution models, and the prediction result is verified by an accelerated failure test. The analysis shows that

- The dynamic model of brake caliper proposed in this paper is available for random vibration analysis, which is proved by comparing the simulated result and test result of model analysis.
- The weak points of the structure are identified by random response analysis.
- By PSD analysis of stress responses, the bandwidths of PSD spectrums of weak points on levers and frame range from 0.27 to 0.58, some of them are within 0.3 and 0.5, these do not represent an obviously feature of neither narrow-band nor wide-band. So different PDF models are employed to describe the distribution of stress amplitude.
- The fatigue damages under transverse and longitudinal random vibration conditions are much less than the counterpart results from vertical random vibration condition for a same duration time.
- Dirlik’s model and Steinberg’s model provide similar predictions of failure on levers and frame under a half-hour vertical random vibration, but the Bendat’s model only predicts the failure of frame under the same vibration condition.
- Accelerated failure test obtains approximately the same result of failure as the prediction provided by Dirlik’s and Steinberg’s models. This demonstrates that Dirlik’s model and Steinberg’s model are more suitable for vibration fatigue failure prediction of the brake caliper.

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