Failure analysis and fatigue life prediction of high-speed rail clips based on DIC technique

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Abstract
With the development of rail transportation, the fatigue failure of rail clips has become an issue, which affects the operational safety of trains. In this study, reasons for the fatigue failure of rail clips were investigated to improve their service life. A digital image correlation (DIC) technique was conducted to obtain strain fields, vibration modes, and natural frequencies of a rail clip. The strain and displacement of a rail clip under dynamic cyclic loading were also obtained. A fastener system refinement model was developed to analyze the static, dynamic, and modal responses of the clip. The experimental tests and modal simulation results were mutually verified. The fatigue life was analyzed based on the verified FE model. The results revealed that the maximum strain and minimum fatigue life occur at the heel of the clip, in good agreement with the actual fracture position. As the amplitude and frequency of dynamic cyclic load increased, the fatigue life of the clip decreased sharply. Moreover, the normal wheel–rail force accompanied by high-frequency rail corrugations accelerated crack initiation and reduced the fatigue life. The findings of this study provide guidance for improving the service life of rail clips.

Keywords
Fastening system, digital image correlation, fatigue life, cyclic load, amplitude and frequency

Date received: 1 August 2021; accepted: 24 November 2021

Handling Editor: James Baldwin

Introduction
The fastener system is a critical component in a track structure, providing elasticity that helps reduce vibration and impact on the wheel–rail system. It also plays a significant role in fixing the rail and maintaining the gauge distance.1 However, with the rapid development of rail traffic systems in China, vibrations due to trains have become frequent in recent years. For example, a rail clip is subjected to a combination of bolt preload and wheel–rail cyclic loading, which can cause fatigue fracture, as illustrated in Figure 1.2 The fatigue fracture of an α-type clip can destroy the continuity of the track structure support and directly affect the safety of train operation.3 Therefore, it is necessary to systematically investigate the factors influencing the fatigue failure and accurately predict the fatigue life of rail clips.

Extensive studies have been conducted on estimating the fatigue life of critical components in high-speed railway vehicle and track, especially on the clip in fastener systems.4,5 Lakusšić et al.6 established finite
element (FE) models for SKL-1 and SKL-12 clips to analyze the equivalent stress distribution and critical stress points in clips. Moreover, they conducted a fatigue experiment and found that cracks appeared on the clip after 353,000 cycles. However, because of the simplified boundary conditions of the model, the actual strain condition of the fastener system could not be simulated. Hasap et al.7 performed a fatigue experiment and finite element analysis (FEA) on a clip to determine the cause of its fatigue failure. Based on the FEA and a series of fatigue tests, they found that under normal wheel loading, the toe load does not influence the fatigue life of the clip and that the impact of the wheel load can significantly reduce the fatigue life. However, the impact load applied in the experiment was too simple to simulate the actual operating condition. Shang et al.8 used an FE model to analyze the failure mechanism of a clip. The mechanical behavior of the clip was described using a bilinear model. Based on the FE model, they evaluated the effect of different installation states of the clip and the vertical displacement of the finger on its clamping force and stresses. Crack initiation quickly occurred in the stress concentration area of the clip under loading due to repeated passage of trains. However, the analysis was only based on quasi-static calculations without considering the dynamic load. Xiao et al.9 established a refined model to assess the fatigue life of an e-type clip. Based on the model, static and dynamic analyses were conducted on the fastening clip. The results showed that the stress concentration of the clip due to unreasonable installation and the resonance resulting from rail corrugation could lead to fracture. However, the model was not verified experimentally. Ling et al.10 used a train–track model to investigate the effects of rail corrugation on the dynamic behavior of the fastening system, and the stress in the clips under the excitation of rail corrugation was calculated. With the contribution of the rail corrugation, the maximum von Mises equivalent stress of the clip was found to increase rapidly. The abnormal vibration of the fastening system due to the corrugation has a significant effect on the fatigue damage of the clip. However, no fatigue analysis has been conducted on the clip under high-frequency dynamic loading. Ferreño et al.11 predicted the fatigue of SKL-1 clips through the experimental and FE method. The maximum and minimum loads applied to the fastener were 70 and 5 kN, respectively. The frequency was 5 Hz. The authors concluded that clip failure is unlikely under normal working conditions after the heat treatment process and fatigue limit of the elastic materials. However, the load applied in their study was too simple. Hasap et al.12 evaluated the influences of clip's malposition on the toe load, stress, and friction. The toe load of e-clips increased with the increase in malposition, and the stress was slightly affected by malposition. However, their model is too simple.

The digital image correlation (DIC) technique was proposed in the early 1980s and has been applied to many engineering problems such as displacement measurements and the evaluation of material strengths or stress analysis.13 Ha et al.14 determined the modal parameters (natural frequencies, mode shapes, damping factors) of an artificial wing mimicking a beetle’s hind wing through digital image correlation (DIC) technique. They also compared the results from DIC to finite element analysis results and discussed the difference between them. Their results showed that the DIC method is promising for studying modal analysis in the design process for any mechanical system. Rizo-Patron and Sirohi15 measured the out-of-plane bending deformation of the rotor blade using DIC and determined modal parameters including natural frequencies and mode shapes. The results obtained in this paper show that the DIC method is promising for studying modal analysis in the design for any mechanical system.

The above studies on the fatigue analyses mainly relate to static loading in the installation state, low-frequency dynamic loading, and clip resonance arising from rail corrugation, without considering the effect of the combination of normal wheel–rail force and rail corrugations on the fatigue life of the clip because of the complexity of the simulation. However, rail corrugation is a common phenomenon in many currently running lines, and it can cause wheel–rail vibrations of various frequencies and fatigue failure of fastener systems. Moreover, there has been no static strain distribution analysis or modal analysis based on the digital image correlation (DIC) technology in rail transit industry that can directly provide the full-field strain and displacement with sub-pixel accuracy.

In this paper, the ω-type clip was taken as the research object to study its fatigue life under the effects of dynamic loads with different frequencies. An FE model of the ω-type clip of the W300-1 fastening system was established to analyze the static and dynamic

*Figure 1. The failure position of ω-type fastening clip in the reality.*
characteristics of the clip. First, the mechanical behavior of the material of the clip was determined through a tensile test. Strain tests under different bolt torques, assembly modal tests, and dynamic tests of the fastening system were conducted using the DIC technology. Second, the experimental results were used to validate the FE model. Finally, the stresses and strains obtained from the validated FE model were integrated into the fatigue life analysis software to predict the fatigue life of the \( \omega \)-type clip. The findings of this paper provide guidance for improving the service life of rail clips.

**Materials and methods**

**Material and scope**

Three molded SKL15 clips (\( \Phi = 15 \text{ mm} \)) were detected according to the standard of Q/CR7–2014, so the material composition (38Si7 hot rolled spring steel) of this clip was obtained.\(^{16}\) Tensile tests were performed on two specimens machined from a molded clip to determine the mechanical properties of the clip. To validate the accuracy of FE models, three groups of fastener systems were tested under various conditions. The first group was subjected to a quasi-static test to get the strain field of the clip. The second group was subjected to a modal test to obtain all the vibration modes of the fastener system in the frequency range of 0–1000 Hz through the DIC technology. The last group was subjected to a dynamic fatigue test, and various load–displacement responses were recorded by applying a dynamic load on the rail. Finally, the possible fatigue failure of the clip under different cyclic loads was determined from the simulation.

**Tensile test**

Standard tensile tests were conducted at room temperature in accordance with GB/T 228.1-2010. A computerized SHT4605 universal testing machine was used to determine material properties of the clip (stress–strain curves) at a tensile rate of 1 mm/min, as shown in Figure 2, with a loading capacity of 600 kN. The relative error of the force indication of this test machine is \( \pm 1\% \).

**Digital image correlation measurements**

**Quasi-static tests.** Figure 3 shows the experimental setup. The clip was compressed vertically under various bolt torques to attain several installation states. The measurement section of the outside surface of the clip was covered with a thin-layered white painting, and black speckles of various sizes were sprayed randomly, as shown in Figure 4. Before testing, a standard \( 10 \times 14 \) dot calibration grid with a dot spacing of 10 mm was used to calibrate the low-speed DIC system to determine the relative position of the cameras.\(^{17}\) Two low-speed cameras were placed in front of the fastener system with a fixed separation of approximately 30°. The cameras acquired images at 3906.25 Hz with an exposure of 200 \( \mu \text{s} \) and a resolution of \( 800 \times 800 \) pixels. High-intensity LED light arrays (Visual Instrumentation) illuminated the object during image acquisition.
The strain field of the clip under different bolt torques was measured by the DIC. The strain could be accurately obtained by measuring the strain field of the clip under the torque and relating it to the strain without bolt torques. Torques of 150, 210, 250, 300, and 340 kN m were applied on the bolt, and images of the clip surface strain were captured in real-time using a DIC camera. The time required to make the speckles, set up the camera and acquisition system, and calibrate the cameras was approximately 2 h.

The DIC technique provided the strain expressed in terms of the engineering variable $e_n$. The true variable $e_t$ was obtained using the transformation equation (1).

$$e_t = \ln(1 + e_n) \quad (1)$$

Modal tests. In this study, a high-speed DIC system based on the VIC-3D software was used to measure the mode shape of the clip. The VIC-3D software uses an algorithm to recognize the surface structure of the clip in the images and allocates coordinates to the image pixels. The first image of the clip represents the undeformed state considering the reference image. During the test, further images of the clip were recorded. Subsequently, the correlation algorithm was used to analyze the digitized speckle image to obtain the surface displacement field information of the clip. The surface of the clip is homogeneous. A white color spray was placed onto a polished surface of the clip, and fine dots of black color were sprayed onto the white surface. The black-to-white ratio of the speckles was 1.3:1, providing excellent contrast for pattern recognition. Images of the clip were captured using a pair of high-speed digital cameras.

Figure 5 shows a diagram of the experimental apparatus. Two Phantom FASTCAM SA-X2-1000k high-speed cameras were placed in front of the fastener system and connected to a computer. An LED light source next to the camera illuminated the rail clip. The vibration frequency of the clip could reach 1500 Hz after hammering, requiring the images to be captured at an accurate rate. The images were recorded at 12500 fps with a resolution of $1024 \times 1024$ pixels, and the angle between the optical axes of the two cameras was approximately 30°. The accuracy of the displacement testing was 40 nm, which met the requirements of high-frequency, high-resolution testing. The subset size for performing the DIC was set to $21 \times 21$ pixels. The Hammer-impact method was used to identify the modal features of the clip. A steel hammerhead was applied with ink to hammer the toe area to ensure that each hammer strikes at the same position. The 3D digital image of the vibration mode of the clip series recorded by the high-speed cameras was stored on the computer in the TIFF format. Finally, these images were exported to VIC-3D software to solve the full-field deformation and 3D shape of the clip.

Cyclic loading tests. The boundary conditions for the cyclic loading tests involve fixing the bolt and compressing the rail, as shown in Figure 6. The test was performed on a conventional fatigue testing machine following the technical specification (TB/T 3396.4-2015 High-speed railway fastener system test method). First, the fastener system was compressed under a bolt torque of 250 Nm, and then the rail was subjected to a cyclic load. The maximum and minimum forces were 70 and 5 kN, respectively, and the frequency was 2 Hz. The strain and displacement fields of the clip were recorded through the high-speed DIC system mentioned above. As shown in Figure 7, the external load excitation frequency is 2 Hz, but the sample frequency of displacement is not consistent with load. Due to the limited storage, the displacement signal data were not collected in every cycle. To collect data more effectively, the images were taken at defined phase intervals by locking on to a drive or response signal. Since the camera’s
frame rate may not be fast enough to image several times during a single cycle, several cycles may be skipped before advancing to next phase, and the signal was accurately tracked by the phase locking logic. In Figure 7, the signal in one cycle (red point) is used to replace the signal in 100 cycles or 200 cycles (black line).

**Finite element modeling**

The 3D geometrical models of the components were drawn in the CAD module of the FE software based on the measured dimensions. The parts were assembled based on the physical object of the fastener system, as shown in Figure 8. The material properties of the clip were defined using the stress–strain curve obtained through the tensile test. The material of the other components was considered homogeneous, linearly elastic, and isotropic. Table 1 lists the material details. The final mesh consists of 227,083 nodes and 145,334 elements. Figure 9 shows the element types for all the parts. The element types were 10-node tetrahedral elements for the clip and the gauge block. The element types were eight-node linear brick elements for the components (rail, bolt, bolt gasket, insulation gasket, sleeper, and iron subplate). The element types were 20-node quadratic brick elements for rail subplate and elastic subplate. Three groups of contacts were defined in the model between the following: clip–bolt gasket, clip–gauge block, and clip–insulation gasket. The contacts were defined as frictional contacts with friction coefficients of 0.15, 0.15, and 0.2, respectively. To avoid the interpenetration of the bodies, augmented Lagrange formulation was used in the vertical direction. The nodes on the bottom surface of the sleeper were constrained with displacement as zero in all directions and rotation about all the axes. The bolt was applied with a preload ranging from 0 to 25 kN.

**Fatigue life calculations**

The conventional crack initiation prediction method is based on the nominal stress, referred to as the S-N
method. The second common and more recent method is based on the local strain, called the strain–life ($e$–$N$) method, which is mainly used for the life assessment of components with high-stress levels.\(^{18}\) The strain-based method considers the plastic deformation that occurs in localized regions where cracks nucleate. Similarly, the strain-based method can also be regarded as a comprehensive method to describe the elastic and inelastic cyclic behaviors of a material. This method is now commonly used in fatigue life prediction, particularly in the rail transit industry.\(^{18}\) A plastic deformation will occur in the local area of the clip under a normal installation state.\(^{8}\) Therefore, using the local strain method to conduct fatigue life prediction on the clips can yield more accurate results.

The fatigue analysis software was applied to predict the fatigue life of the clip in this study. Figure 10 shows the process diagram of the clip fatigue life analysis. First, the material properties of the clip were obtained by conducting a tensile test. Subsequently, using the fastener system model generated using the FE software, the residual stress and strain field information of the clip were imported into the fatigue analysis software.\(^{19}\) Final, based on the FEA results, the fatigue life of the clip under various fatigue conditions was analyzed by inputting loading information into the fatigue analysis software.

The material library of the fatigue analysis software does not contain the material 38Si7. Therefore, the fatigue properties of 38Si7 were approximated using Seeger’s method with the help of the ultimate tensile stress (UTS) and elastic modulus of the material.\(^{20}\) The algorithm is suitable for low-alloy steels and can produce accurate results. The UTS of the material of the clip was 1350 MPa, and the elastic modulus was 180 GPa, obtained based on tensile test data.

The fatigue analysis software provides a variety of fatigue life algorithms. It is a critical step in selecting a reasonable fatigue life prediction method for accurately predicting the fatigue life of a material. The material 38Si7 is a ductile metal. The Brown–Miller algorithm is the preferred method for fatigue analysis as recommended by the American Automobile Manufacturers Association. The BM criterion is a multiaxial fatigue criterion based on the strain, and the fatigue damage occurs in the plane where the maximum shear strain occurs. Fatigue damage is the combined action of the maximum shear strain and the principal strain. It assumes that the shear strain controls crack initiation and that the normal strain causes crack propagation.\(^{21}\) Because the wheel–rail cyclic load is asymmetric, it is necessary to consider the effect of the mean stress on the fatigue life. The Morrow mean stress criterion is often used to modify the algorithm in engineering practice.\(^{21}\)

This algorithm uses the strain–life curve defined by the following equation:

$$\frac{\Delta \gamma}{2} + \frac{\Delta e_{nn}}{2} = 1.65 \frac{\sigma_f - \sigma_m (2N_f)^b}{E} + 1.75 \dot{\varepsilon}_f (2N_f)^c$$  \hspace{1cm} (2)$$

Where $\Delta \gamma$ is the shear strain amplitude, $\Delta e_{nn}$ is the normal strain amplitude on the critical plane, $\sigma_m$ is the mean normal stress on the critical plane, $E$ is the elastic modulus, and $\dot{\varepsilon}_f$ is the strain rate.
modulus, $2N_f$ is the number of cycles for crack initiation, $\sigma_f'$ is the fatigue strength coefficient, $\varepsilon_f'$ is the normal fatigue ductility coefficient, $c$ is the fatigue ductility exponent, and $b$ is the fatigue strength exponent. Table 2 shows the values of all the relative parameters for 38Si7 obtained using Seeger’s method.

Figure 11 shows a case of sinusoidal load, which is equivalent to measured normal wheel–rail force (mean value = 240 kN, amplitude = 100 kN). The impact due to irregularities in the rail and wheel, for example, weldment, wear, and rail corrugation, accounts for up to 60% of the wheel–rail load. For a broader analysis, equivalent sinusoidal load amplitudes ranging from 70 to 170 kN were applied to the rail to evaluate the fatigue life of the clip.

A short–pitch rail corrugation causes strong high-frequency vibrations in fastener clips. The equivalent sinusoidal loads (amplitude: 100 N) with various frequencies (4–2000 Hz) were applied to rail fastenings to investigate the influence of rail corrugation on the fatigue life of rail clips.

The actual wheel–rail force is typically composed of normal wheel–rail force and the impact of rail corrugations. To calculate the fatigue life of rail clips more realistically, a wheel–rail force–time curve was obtained by adding loads with various frequencies and amplitudes to the equivalent sinusoidal loads (amplitude: 100 kN, frequency: 4 Hz). For example, the frequency and amplitude of the added load were 270 Hz and 30 kN, respectively, as shown in Figure 12. The amplitude of added loads was 10, 30, and 60 kN. The frequency range was 270–1000 Hz. The fatigue lives of the clip under these conditions were calculated. The fatigue life was defined in terms of the number of cycles.

Results and discussion

Tensile curve and strain distribution

Figure 13 shows the stress–strain curve of the material, to make the test results sufficient and the sample easy to distinguish, two samples were tested, and they were named 1#_New clip and 2#_New clip. As shown, the yield stress and tensile strengths are 1240 and 1350 MPa, respectively. Figure 14 shows the distribution of the strain on the surface of the clip under different installation states, where the maximum strain is at the heel of the clip. The maximum strain was 0.0115 under a bolt torque of 150 Nm, which already exceeds the yield strain of the material (0.00839, see Figure 13). The plastic strain occurred in the local area of the clip, even the bolt in the under-twisted state. Figure 15 shows the maximum principal strain contour obtained from the finite element analysis. The simulation result has a good agreement with the test result (Figure 14(c)), that is the difference of strain is lower than 10%.
Therefore, a bolt torque of 250 Nm in the test corresponds to a preload of 17 kN in the simulation.

Figure 16 shows a comparison between the experimental and simulated curves relating the force and maximum principal strain at the critical location of the clip. The experimental strain results are in good agreement with the FEA results, that is, the difference of strain value is lower than 10%. What’s more, the direction of the maximum principal strain obtained by experiment and simulation is approximately 45° with x-axis. Therefore, the FE model is satisfactorily validated in terms of the static characteristics from comparing the experimental and simulation results.

The strain and stress of rail clips under static loading have been studied by Ferreño et al.11 and Hasap et al.12 who only used FEA. Although they employed the test results obtained from conventional measurement tools, involving the use of strain gauges to validate the FEA,

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**Figure 13.** Stress–strain curve obtained from a new specimen through tensile test.

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**Figure 14.** Strain distribution contour plots of the clip under different installation states obtained through low-speed DIC: (a) 150 Nm, (b) 210 Nm, (c) 250 Nm, (d) 300 Nm, and (e) 340 Nm.
the actual location, and magnitude of the maximum strain and stress of the rail clips could not be accurately detected because of the complex geometry of measuring tools and curved surfaces of rail clips. In comparison, the location of the maximum strain and the full-field strain of the clip can be accurately determined using the DIC technology.

Table 3 summarizes the first two natural frequencies and mode shapes. Through comparison, the maximum error between the simulation output and the test results was 1.6%. The first mode displays as the opposite phase movement of the two arches relative to the toe and heel, and the second mode manifests as the same phase. These two mode shapes make the small arc area at the heel of the clip to be in a typical combined bending and torsion deformation state. Based on the above analysis, it can be concluded that the experimental natural frequencies and shapes are in good agreement with the analytical predictions, thus, the dynamic characteristics of the FE model could be verified.

Figure 15. The simulation strain distribution contour plot of the clip under a bolt preload of 17 kN.

Figure 16. The comparison of maximum principal strain between experimental and simulation results.

Gao et al. focused on the assembly modal response of a clip under different bolt torques. Only two accelerometers were placed on the left and right arms of the clip. Hence, some areas without accelerometers could not be appropriately measured. In comparison, the DIC method can be accurately utilized in experimental modal analyses to extract the natural frequencies and modal shape.

**Strain and displacement under cyclic loading**

Figure 17 shows the experimental and simulated curves relating the time and displacement and strain at the heel of the clip. The experimental displacement and strain values are in good agreement with the FEA results. Therefore, the FEA model can be once again validated. The displacement of rail clips under dynamic loading has been studied by Ferreño et al. who used a displacement transducer. However, it is difficult to measure the full-field displacement using transducers due to their volume. Therefore, the location of the maximum displacement could not be accurately determined. This problem can be solved using the DIC technique.

**Effects of dynamic loading on fatigue life**

Figure 18 shows the contour of the fatigue life of the clip on a logarithmic scale under the equivalent sinusoidal load (mean value = −40 kN, amplitude = 100 kN), estimated using the \( e^{-N} \) curve and the residual stress and strain field data of the clip obtained from the FE software. The red region is the location with a lower fatigue life, which is also the region (the heel of the clip) where the maximum principal strain occurs. The fatigue life of the other areas is close to “infinite life,” that is, the maximum fatigue life limit for the clip is set to 10^20. The lowest fatigue life of the clip under this load condition meets the service requirement of five million cycles.

Figure 19 shows the minimum fatigue life of the clip plotted against the force amplitude. The logarithmic fatigue life of the clip can be approximated to have a linear relationship with the force amplitude. It can also be observed from Figure 19 that fatigue life decreases as the force amplitude increases among the considered range of 70–170 kN. This occurrence of decreasing trend is because the high load amplitude could increase the vibration amplitude of fastener systems. In addition, under equivalent sinusoidal load (amplitude: 100 kN), the fatigue life of the clip is greater than 5 million cycles. However, when the amplitude of the equivalent sinusoidal load was increased to 160 kN due to irregularities between the wheel and rail, the fatigue life was significantly reduced to 4760 thousand cycles. Therefore, according to the analysis above, it can be concluded that the fatigue life of the clip is less than...
5 million cycles that could not meet the service requirement if the force amplitude exceeds 140 kN.

Hasap et al.² studied the fatigue life of a clip under different installation states, that is, low, normal, and high toe loads. They found that under normal wheel loading, the toe load had no influence on the fatigue resistance of the clip. However, with the contribution of the impact considered, the fatigue life reduced significantly. They indirectly proved the influence of wheel–rail load amplitude on the fatigue life of rail clips. The fracture of the clip affected the safe operation of the track; hence, based on this result, that attention should be paid to reducing the impact of the wheel–rail load to improve the service life of the clip in the routine maintenance of tracks.

Figure 20 shows the relationship between the fatigue life of the clip and the frequency of the equivalent sinusoidal load. The load frequency directly affects the

| Order | Test: 515.0 | Error (%) | FEA: 523.3 |
|-------|-------------|------------|-------------|
| First | Test: 515.0 | 1.6        | FEA: 523.3  |
| Second| Test: 578.7 | 0.5        | FEA: 575.5  |

**Figure 17.** Response at the heel of the clip in the time domain (red curve: simulation, black curve: experiment): (a) displacement and (b) strain.

**Table 3.** Comparison between analytical and experimental natural frequencies and modes.
As the load frequency increases, the fatigue life of the clip maintains constant at the frequency range of 4–300 Hz but then decreases sharply when the frequency exceeds 300 Hz. The reason that the life decreases sharply is that the high-frequency and high-amplitude load could excite the response of multiple frequency components of the clip, increasing the local stress and strain, and reducing the fatigue life of a clip. At the frequency of 370 Hz, the fatigue life significantly reduced to 1480 thousand cycles. Hence, it is of great significance to reduce the high-frequency vibration of the fastener system to improve the fatigue life of the clip.

Figure 21 shows the effect of the equivalent sinusoidal load (frequency: 4 Hz, amplitude: 100 kN) with other frequency signals added to it on the fatigue life of the clip. A wheel–rail force–time curve was obtained by adding loads with various frequencies and amplitudes to the equivalent sinusoidal loads (amplitude: 100 kN, frequency: 4 Hz). The amplitude of added loads was 10, 30, and 60 kN, and the frequency range was 270–1000 Hz. According to these curves, the life of the clip decreases with the increase of the frequency under the same load amplitude and decreases with the increase of the load amplitude under the same load frequency. At added load amplitude of 10, 30, and 60 kN, the minimum fatigue lives of the clips are 120 thousand, 14 thousand, and 3 hundred cycles, respectively. They are all less than the required service life (500 million cycles). Compared with the fatigue life (solid red line) of the clip under the equivalent sinusoidal load, the fatigue life decreases significantly under the multiple frequency component loads. The added load amplitudes are smaller than those of the equivalent sinusoidal loads, but still make a great contribution to the fatigue damage of the clip because of increasing a lot of dynamic cycles. In actual, the smaller amplitude high frequency loads can be caused by rail corrugation. Therefore, the
service life of the clip will be reduced if it is operated in a section of a railway with rail corrugation. Rail lines will subject to rail corrugation at the straight-line sections and curved sections after running for several years, considering that the fatigue life of the clip needs to meet the criterion of 5 million cycles, the rail corrugation should be reduced. To make the fatigue life of the clip exceed 5 million cycles, as observed in Figure 21, the corresponding load amplitude should be controlled below 10 kN when the added load frequency is >580 Hz, the corresponding load amplitude should be controlled below 30 kN when the added load frequency is >300 Hz. The pink dotted line in the figure indicates that once the amplitude of the added load due to the corrugation becomes higher than 60 kN, the life of the clip will not meet the service requirements at the frequency range of 270–2000 Hz. It is well known that the frequency and amplitude of the wheel-rail loads are related to the length and depth of the rail corrugation, when the train speed is 250 km/h, the maximum wheel-rail vertical force of the circular curve with wave depth of 0.04 mm increases by 10% compared with that rail lines without corrugation. And when the wavelength is 120 mm, the excitation frequency would reach to 580 Hz in the line. Based on the above analysis, considering that the fatigue life of the clip to meet the criterion of 5 million cycles, the corrugation depth at wavelength of 120 mm should be controlled below 0.04 mm in this case. And with the increase of wavelength, the depth of the rail corrugation meeting the requirements can also increase accordingly. Therefore, the fatigue life of the clip could be improved by griding rail to control the corrugation amplitudes and frequencies.

Ling et al. studied the effects of rail corrugation and found that the maximum von Mises equivalent stress increases rapidly with the increase in the corrugation depth but decreases with the increase in the wavelength. Xiao et al. showed that the resonance due to rail corrugations could lead to fractures in the fasteners. Because of the differences in the train speed and wavelength, rail corrugations can excite the vibration response of the track structure at various frequencies. However, the authors did not predict the fatigue life of the clip under wheel-rail force combined with rail corrugation. This study, which was conducted considering these aspects, evaluated the influence of wheel-rail loads with various frequencies and amplitudes on the fatigue life of rail clips. By adding loads with different frequencies to the equivalent sinusoidal loads, the variation in the fatigue life with the load frequency and amplitude was predicted. The FE model predictions showed that the leading cause of fatigue failure of rail clips is the sustained action of the normal wheel-rail force accompanied by high-frequency rail corrugations. Under this condition, the clip may have surface cracks that would accelerate fatigue failure.

Conclusions

This study applied a novel test method (DIC technology) to identify the static, modal, and dynamic characteristics of a ω-type clip. The fatigue performance of the clip under various dynamic loads was analyzed based on a refined fastening system model with the help of fatigue life analysis software. The conclusions drawn are as follows:

1. DIC techniques can be accurately and effectively used for obtaining full-field displacement and strain of rail clips under static and dynamic loading in a non-contact manner. The global maximum principal strain of the clip under the normal installation state obtained through the DIC was 0.0135, which exceeded the yield strain (0.00839). And the first two natural frequencies of the clip were 515 and 578 Hz at the frequency ranges of 0–1000 Hz, respectively.

2. Under equivalent sinusoidal load, the fatigue life decreased significantly with the increase of the load amplitude. To meet the service requirements (5 million cycles), the corresponding amplitude of the dynamic load should be controlled below 140 kN. As the load frequency increases, the fatigue life maintains constant at first but then decreases sharply. It is of great significance to reduce the high-frequency vibration of the fastener system.

3. The normal wheel-rail force accompanied by high-frequency rail corrugations can accelerate crack initiation and reduce the life of rail clips. Therefore, when the frequency of the added load is >300 Hz, the amplitude should be controlled below 10 kN. When the frequency of the added load is >580 Hz, the amplitude should be maintained below 30 kN. When the amplitude of the added load is >60 kN, the rail corrugation at any frequency should be avoided, which can be achieved by rail grinding.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article.
The National Natural Science Foundation of China under Grant 51708422, Shanghai Rising-Star Program (19QB1401100), and Shanghai Pujiang Program (20PJ1417300).

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