Experimental Study on Fatigue Performance of High Strength Bolts of Rail Vehicles Under Axial Alternating Load

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Abstract. Lots of key parts of rail vehicles are connected by screw thread fastening. Therefore, it is very important to evaluate the service performance of bolt correctly and efficiently. Through the evaluation of the service performance of bolts, a series of engineering application problems of bolt connection can be well solved. In this paper, model building, theory checking, simulation analysis and bench test verification are studied in a unified way, so as to get the theory and research method of engineering application. According to the bench fatigue test, the 10.9 grade bolt was fractured after loading about 3 million times, so the bolt could not meet the life cycle requirement under this working condition.

1. Introduction

Most of the connections of the key parts such as suspension, running parts and pantograph under the rail vehicle are fastened by threaded connection, and the safety of threaded connection will directly affect the driving safety. Most of the threaded connections are bolt connections, and the materials used are high-strength bolts. Therefore, it is very important to evaluate the performance of bolt correctly and efficiently. Through the evaluation of bolt service performance, a series of engineering application problems of bolt connection can be solved, such as direct design of bolt connection, fault analysis and diagnosis, optimization of connection process and so on.

After Albert of Germany carried out the fatigue test of the chain of the mine hoist [1], August Wohler of Germany put forward the concept of stress-life curve and fatigue limit [2–4], Miner proposed the theory of linear cumulative damage [5–8], the basic theory of fatigue life research is provided. At the same time, the British Griffith presented the theory of fracture, fatigue crack propagation research was also launched [9–12].

The fatigue research in China is based on the classical theory and experience of foreign countries. According to the definition of Yao Qihang and other researchers, only the resonance damage caused by the excitation in or near the resonance bandwidth belongs to the vibration fatigue damage, and the rest belong to the static fatigue problem [13–16]. In addition, Wang Hong defined three major resistance factors of structural fatigue crack growth, and described the expression model of fatigue crack growth law [17]. Liu Qingfeng [18], Yu Peishi [19], etc. have also carried out fatigue studies around crack growth.

However, the practical fatigue problem of high strength bolts for rail vehicles still needs to be systematized. In this paper, a dialectical analysis is carried out from the angles of model building,
theoretical calculation, simulation analysis and bench test, the theory and research method of fitting engineering practice can be obtained.

2. Principle analysis

The dynamic equation of the excited vibration system is

\[ M \ddot{x}(t) + C \dot{x}(t) + Kx(t) = Pu(t) \]

where \( M \), \( C \) and \( K \) are the mass, damping and stiffness matrices of the system, \( P \) is the coefficient matrix, and \( u \) is the excitation.

By Laplace transformation,

\[ X(s) = H(s)U(s) \]

where the transfer function matrix is

\[ H(s) = \left( Ms' + Cs + K \right)^{-1}P \]

In this paper, the bolt under the primary spring of metro vehicle is taken as the research object. Under the comprehensive action of the bolt, such as the gravity from the car body, the longitudinal force caused by braking, the lateral force caused by wheel rail contact and the self-preloading force, the stress and transmission relationship of the bolt will be particularly complex, and the \( M \), \( C \), \( K \) and other parameters are difficult or impossible to obtain. Therefore, the reverse analysis of fault cases will be used for analysis and verification.

Under the action of alternating load, the stiffness of bolt material decreases and deteriorates, which is expressed by damage degree \( D \) \([20]\)

\[ D = \frac{E - E_D}{E} (0 \leq D \leq 1) \]

where \( E \) is the young’s modulus without damage and \( E_D \) is the young’s modulus with damage degree \( D \). For linear elastic constitutive relation

\[ \sigma_{ij} = \delta_{ij} \lambda \varepsilon_{kl} e_{kl} + 2 \mu e_{ij} \]

where \( \sigma_{ij} \) and \( \varepsilon_{ij} \) are stress component and strain component respectively, \( \lambda \) and \( \mu \) are lame constants. When the damage degree is \( D \), the constitutive relation is

\[ \sigma_{ij} = (1 - D) \delta_{ij} \lambda \varepsilon_{kl} e_{kl} + 2(1 - D) \mu e_{ij} \]

According to the derivation of \([20]\), the relationship between crack initiation life \( N_f \) and maximum stress \( \sigma_{0,max} \) is

\[ \log N_f = \log \left[ \frac{E^n}{\sigma_{fr} (m) \left( 1 - D_{0,1e} \right)^{2n+1}} \right] - m \log \left( \sigma_{0,max} - \sigma_{0,1e} \right) \]

where \( m_f \) is the undetermined parameter, \( \alpha \) is related to \( m_f, E \), etc., \( D_{0,1e} \) is the initial damage, \( \sigma_{0,max} \) is the maximum stress caused by alternating load, \( \sigma_{0,1e} \) is the stress threshold.

The following damage calculations are carried out by using the finite element method.
Set the incentive as

\[ \{ P \} = \{ P_{\text{max}} \} \sin \omega (t - t_0) \]  

So

\[ \phi = \left[ \left( [k] - \sum D_i [a]^T_i [k]_i [a]_i \right) - \omega^2 [M] \right]^{-1} \{ P_{\text{max}} \} \]  

\[ [k] - \sum D_i [a]^T_i [k]_i [a]_i = \omega^2 [M] \]  

It can be seen from the formula that when \( \omega = \omega' \), \( \phi \) is infinite and resonance occurs, so there are two forms of vibration fatigue failure of components, one is damage failure, that is, the damage degree of a unit reaches 1 and the natural frequency after damage \( \omega_D \neq \omega' \). At this time, the number of load cycles is the fatigue crack initiation life. The other is resonance failure, that is, with the increase of load cycles, the natural frequency of the component decreases, the natural frequency after damage \( \omega_D \neq \omega' \), so the resonance occurs under the action of alternating load.

3. Calculation check

First of all, the material and mechanical properties of bolts are analyzed, and it is found that the material properties of 8.8 grade and 10.9 grade bolts meet the relevant standards, so the fracture fault caused by the bolt itself is eliminated.

3.1. Failure Analysis of Bolt Fracture

The fracture analysis results of 10.9 grade bolt show that, typical fatigue texture can be seen on the fracture surface, and the fracture is fatigue fracture. The fatigue instantaneous fracture zone is isometric dimple, which is judged to be fatigue fracture caused by normal stress.

The fracture of 8.8 grade bolt is analyzed. Firstly, the cross-section morphology of the twin-broken bolts presents obvious striation characteristics of fatigue shell, and then the cross-section morphology is more complex, due to the overall structural

Fig 1. Bolt fracture diagram.
instability caused by the fracture. Therefore, the fracture of 8.8 grade bolt is judged as fatigue fracture caused by normal stress.

3.2. Bolt Calculation and Verification

In order to carry out further analysis, VDI analysis method is used to calculate and analyze the worst stressed bolts under various working conditions. The VDI calculation is carried out by the steps in Table 1.

| Calculation Steps | Result |
|------------------|--------|
| **R0** Nominal diameter | \(d=16\text{mm}\) |
| **R1** Tightening factor \(\alpha_A\) | \(\alpha_A=1.4\) |
| **R2** Minimum clamp load \(F_{\text{Kerf}}\) | \(F_{\text{Kerf}}=39620\text{N}\) |
| **R3** Dividing the working load/load factor \(F_{SA}, F_{PA}, \Phi\) | Axial load factor \(\Phi=0.282\), Bending load factor \(\Phi^*=0.0576\) |
| **R4** Preload changes \(F_Z\) | \(F_Z=3588\text{N}\) |
| **R5** Minimum assembly preload \(F_{\text{Min}}\) | \(F_{\text{Min}}=46720\text{N}\) |
| **R6** Maximum assembly preload \(F_{\text{Max}}\) | \(F_{\text{Max}}=65408\text{N}\) |
| **R7** Assembly stress \(F_{\text{Mzul}}\) | \(F_{\text{Mzul}}=137136\text{N}\) |
| **R8** Working stress \(\sigma_{\text{red,B}}\) | \(\sigma_{\text{red,B}}=753\text{MPa}\) |
| **R9** Alternating stress \(\sigma_{\text{ASV}}\) | \(\sigma_{\text{ASV}}=46\text{MPa}\) |
| **R10** Minimum length of engagement \(m_{\text{effmin}}\) | \(m_{\text{effmin}}=12.8\text{mm}\) |
| **R11** Slipping \(S_G\) | Bolts of 8.8: \(S_G<1\) Bolts of 8.8: \(S_G>1\) |

According to R9 alternating stress calculation, the maximum vertical force change range of bolt is 7.5 KN, and the maximum bending moment change range is 22.624 N-M. Based on this calculation, the stress amplitude caused by vertical load is 22 MPa, the stress amplitude caused by bending moment load is 36 MPa, and the total fatigue load stress amplitude is 58 MPa. According to the fatigue curve of BS-7608, the corresponding cycle life is 2x10^6 times, so the bolt can’t be used for infinite life. Combined with the test results of preload, the following results can be obtained.

3.2.1. When the horseshoe gasket is not installed well. the maximum vertical load of 10.9 grade bolt is 54 KN, then the calculated fatigue stress is 170 MPa, and the corresponding bolt fatigue life is about 40000 times according to VDI calculation.

3.2.2. When the horseshoe gasket is installed well. the maximum vertical load change of 10.9 grade bolt is 18 KN, then the calculated fatigue amplitude is 70 MPa, and the corresponding bolt fatigue life is about 1 million times according to VDI calculation.

3.2.3. When the horseshoe gasket is installed well. the maximum vertical load change of 8.8 grade bolt is 16 KN, then the calculated fatigue amplitude is 50 MPa, and the corresponding bolt fatigue life is about 1.5 million times according to VDI calculation.

According to the above calculation, the bolt has yielded under the working load, and does not meet the anti-sliding condition. The pretension is further lost under the alternating load, so the fatigue life can’t be evaluated.
In conclusion, the calculation and verification results show that, under the condition of 150 N·M, 8.8 grade bolt does not meet the pretension requirements, and may slip. Under the condition of 230 N·M, 10.9 grade bolt meets the pretension requirements, and will not slip.

4. Simulation analysis

4.1. Structural Analysis
When the bolt is tightened, the lower gland is bent to make the bolt bear additional bending moment. In the process of train operation, under the condition of traction or braking, the shoulder of the mandrel produces bending moment at the axle box fulcrum, and the load of the two bolts is unbalanced, which bear the repeated alternating load. The schematic diagram of bolt deformation is shown in Fig. 2.

4.2. Simulation Analysis
The whole structure simulation analysis model based on the primary spring bolt is established, as shown in Fig. 3. In the above analysis, due to the yield of 8.8 grade bolt under the preload, the life of grade 8.8 bolt can’t be evaluated. The life of 10.9 grade bolt
is mainly evaluated. According to the life assessment, two dangerous points of fatigue stress are located in screw in area and screw root respectively. The maximum dynamic fatigue stress is 64 MPa. According to the calculation results of BS-7608 and VDI, the fatigue life of the bolt is estimated to be between 1.5 million and 3.3 million times, which does not meet the requirements of infinite life.

5. experimental analysis

5.1. High Frequency Resonance Fatigue Test
Under the condition of static load of 80 KN and dynamic load of ± 9 KN, the fatigue life interval of 10.9 grade bolt is about 1.44-3.8 million times. Under this condition, the bolt does not meet the service requirements of the whole life. The vertical static load is 54.5 KN, the longitudinal dynamic load is ± 45 KN, and the loading frequency is 5 Hz.

5.2. Bench Test for Simulating Operation Conditions

5.2.1. Fatigue experiment. The moving load of the vehicle is applied to the bolts to be studied through the bogie, so the moving load is simplified to the bogie, that is, the vertical static load applied to the axle box of bogie is 50 KN, and the longitudinal dynamic load is 45 KN. The load is applied by the test bench through bogie parameter measurement. The change of the axial force of the studied bolt is tracked by the force measuring bolt. The results show that the axial alternating loads of 8.8 grade and 10.9 grade bolts are ± 8KN and ± 9 KN respectively.

5.2.2. Dynamics experiment. A bench fatigue test model is set up for fatigue test. The main simulated operation condition is the severe condition that the primary spring bears, that is, the vertical static load is 50 KN, and the longitudinal dynamic load is 45 KN. According to the test results, the 10.9 grade bolt breaks after 3.01 million times of fatigue loading, the fracture position and fracture form are close to the line fracture bolt. The results show that the bolt does not meet the requirements of life cycle.

Fig 4. Fatigue test of bolt.

Fig 5. Bolt fracture diagram in bench test.
6. Results and discussion

According to the test verification and analysis, the failure bolt is fatigue fracture, which is judged to be caused by normal stress.

(a) The results of assembly structure analysis show that, in the process of vehicle operation, the shoulder of the mandrel produces a bending moment at the axle box fulcrum, the load of the two bolts is unbalanced, and they bear the repeated alternating load. When using the horseshoe adjusting pad, the bending moment of the bolt which is opposite to the notched direction of the horseshoe adjusting pad is further increased due to the gap. The axial intersection load of 10.9 grade bolt is \( \pm 9 \) KN.

(b) The evaluation results of assembly preload show that the preload provided by 10.9 grade bolt is about 80 KN when the torque is 230 N\( \cdot \)M, which meets the preload requirements of assembly structure clamping.

(c) The results of axial resonance fatigue test show that the fatigue life of bolt is about 1.44-3.8 million times. The fatigue test shows that the 10.9 grade bolt breaks after loading about 3 million times, and the bolt does not meet the requirements of life cycle.

In conclusion, through a series of theoretical analysis, simulation analysis and experimental research, this paper understands the causes of the fracture of the bolt under the primary spring, which provides a reference for the safety and dynamic performance of the train.

References

[1] W. A. J. Albert, “Archive für mineralogie, geognosie, bergbau und hüttenkunde,” vol. 10, pp. 215-234, 1838.

[2] Luca, Susmel, “The modified wöhler curve method calibrated by using standard fatigue curves and applied in conjunction with the theory of critical distances to estimate fatigue lifetime of aluminium weldments,” International Journal of Fatigue, vol. 31, pp. 197-212, 2009.

[3] N. Pugno, P. Cornetti, and A. Carpinteri, “New unified laws in fatigue: From the Wöhler’s to the Paris’ regime,” Engineering Fracture Mechanics, vol. 74, pp. 595-601, 2007.

[4] J. Sandstroem, “Subsurface rolling contact fatigue damage of railway wheels-A probabilistic analysis,” International Journal of Fatigue, vol. 37, pp. 146-152, 2012.

[5] N. Z. Chen, G. Wang, and C. G. Soares, “Miner’s rule and fracture mechanics-based inspection planning,” Engineering Fracture Mechanics, vol. 78, pp. 3166-3182, 2011.

[6] Z. Hashin, “A reinterpretation of the palmgren-miner rule for fatigue life prediction,” Journal of Applied Mechanics, vol. 47, pp. 446-447, 1980.

[7] J. J. Kauzlarih, “The Palmgren-Miner rule derived,” Tribology, vol. 14, pp. 175-179, 1989.

[8] D. Chen, R. El-Hacha, “Behaviour of hybrid FRP–UHPC beams in flexure under fatigue loading,” Composite Structures, vol. 94, pp. 253-266, 2012.

[9] A. A. Griffith, “The phenomena of rupture and flow in solids,” Philosophical Transactions of the Royal Society of London, vol. 221, pp. 163-198, 1921.

[10] B. Atzori, P. Lazzarin, and G. Meneghetti, “Fracture mechanics and notch sensitivity,” Fatigue & Fracture of Engineering Materials & Structures, vol. 26, pp. 257-267, 2010.

[11] J. Campbell, “The origin of griffith cracks,” Metallurgical & Materials Transactions B, vol. 42, pp. 1091-1097, 2011.

[12] H. A. Elliott, “An analysis of the conditions for rupture due to griffith cracks,” Proceedings of the Physical Society, vol. 59, pp. 208, 1947.

[13] Q. H. Yao, J. Yao, “Vibration fatigue in engineering structures,” Chinese Journal of Applied Mechanics, vol. 23, pp. 12-15, 2006.

[14] Q. H. Yao, J. Yao, “The behavior and analysis of structure vibration fatigue,” Mechanical Science and Technology, vol. 19, pp. 56-58, 2000.

[15] Q. Cao, C. Shao, and Q. H. Yao, “Research on vibration environment test technology of aircraft structure combined with fatigue or static load,” Acta Aeronautica Et Astronautica Sinica, vol. 19, pp. 405-409, 1998.

[16] T. H. Le, L. Caracoglia, “Reduced-order wavelet-galerkin solution for the coupled, nonlinear
stochastic response of slender buildings in transient winds,” Journal of Sound & Vibration, vol. 344, pp. 179-208, 2015.

[17] M. R. Joyce, K. K. Lee, and S. Syngellakis, P. A. S. Reed, “Quantitative assessment of preferential fatigue initiation sites in a multi-phase aluminium alloy,” Fatigue & Fracture of Engineering Materials & Structures, vol. 27, pp. 1025-1036, 2010.

[18] Q. F. Liu, J. L. Xie, and L. X. Miao, “Simulating semi-ellipse surface crack by finite element techniques,” Computer Aided Engineering, vol. A1, pp. 400-403, 2006.

[19] K. Ogura, K. Ohji, and K. Honda, “On the three dimensional aspect of fatigue crack closure in plate specimens,” International Journal of Fracture, vol. 32, pp. 524-527, 1977.

[20] M. Zhang, Q. Meng, and X. Zhang, “Damage mechanics-finite element method for fatigue life prediction of flare-free pipeline connection assemblies,” Acta Aeronautica Et Astronautica Sinica, vol. 30, pp. 435-551, 2009.