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STRENGTH EVALUATION OF HIGH-STRESS CONNECTING BOLTS FOR HEAVY SUSPENSION EQUIPMENT OF HIGH-SPEED TRAIN BASED ON SUB-MODEL METHOD

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The high stress bolts connecting the heavy-duty suspension equipment of high-speed trains are vulnerable, and the cost of full-scale testing is high and the results are scattered. This study uses the sub-model method to compare the strength evaluation of mechanical design theory and VDI 2230 standard. The results indicate that the sub model method is feasible, but the strength evaluation method should be selected according to the needs. The former is suitable for rapid estimation, while the latter provides more accurate analysis of connection strength through multiple factors such as load and temperature. The results provide reference for the evaluation of complex engineering bolts.

Keywords: high-speed train, heavy-duty suspended equipment, high-stress bolts, sub-model approach, strength assessment

1. Introduction

Heavy-duty suspended equipment is an important bearing component of high-speed train, which is mainly connected with the car body through high stress bolts. Such bolts have the advantages of strong bearing capacity, good fatigue resistance and not easy to loosen under dynamic load [1]. However, in the service process of the product, this kind of high stress bolt will still have different degrees of fatigue damage or loose phenomenon, which brings safety hazards to the product service [2,3]. Currently, many engineering technicians have carried out relevant research on the strength assessment of connection bolts for high-speed trains. Zhang applied the multi-body ADAMS static calculation method to decompose the chassis load of the rubber-tired train and proposed bolt load extraction methods for the multi-bolt connection structure and single-bolt connection structure in the rubber-tired train respectively [4]. Liu used the ANSYS software to simulate the actual

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stress situation of subway anti-climbers' bolts. Through the comparison between finite element simulation and experiments, they summarized the methods for calculation convergence [5]. Qu studied the evolution law of the bolt axial load during the braking process by conducting braking process simulations and braking dynamic tests [6].

Existing studies have made progress in extracting and varying bolt loads for high-speed trains, but the results are different due to complex modeling, low efficiency and inconsistent standards depending on full-size car body models [7]. Therefore, this study builds a calculation model based on the sub-model method, and combines the mechanical design theory with VDI 2230 standard to establish a programmed bolt strength evaluation process. The feasibility of this method is verified by an example of heavy suspension equipment, which provides a standardized solution for the strength evaluation of high stress bolts.

2. Whole-based sub-model technology

The sub-model method can effectively improve the calculation efficiency and ensure the reliability of the results [8,9]. It should be noted that the cutting boundary should be kept away from the region of interest in the sub-model's response [10]. Based on the theory of finite element analysis, any structure has an equilibrium equation:

$$[K]\{U\}=[F] \tag{1}$$

where K is the stiffness matrix of the global model, U is the displacement vector, and F is the load vector.

Assuming that the global model consists of two parts, namely component I and component B, their displacement vectors can be represented as U_I and U_B. If the specific value of U_B is known, eq. (1) can be rewritten as:

$$\begin{bmatrix} K_{II} & K_{IB} \\ K_{BI} & K_{BB} \end{bmatrix} \begin{Bmatrix} U_I \\ U_B \end{Bmatrix} = \begin{Bmatrix} F_I \\ F_B \end{Bmatrix} \tag{2}$$

where K_{II} and K_{BB} are the stiffness matrices of component I and component B, K_{BI} and K_{IB} are the stiffness matrices between the two, and U_I, U_B and F_I, F_B are the displacement vectors and load vectors of the two, respectively. eq. (2) can be written as:

$$[K_{II}]\{U_I\}=\{F_I\}-[K_{IB}]\{U_B\} \tag{3}$$

eq. (3) indicates that the specified displacement U_B is a part of obtaining the U_I load vector. That is to say, for the elastic structures, the specified displacement applied to them can generate load-related effects.

In the sub-modeling method, three typical cases are usually distinguished: a solid global model driving a solid sub-model, a shell global model driving a shell sub-model, and a shell global model driving a solid sub-model [11]. The present study focuses on the second case.

The shell global model driving a shell sub-model is the most common in engineering applications and constitutes the main emphasis of sub-modeling analysis. Its fundamental idea is to employ the mid-surface of the global shell model to drive the boundary response of the sub-model. Specifically, the procedure is as follows: first, determine whether the boundary nodes of the sub-model are located near the boundary of the global shell model; then, using the shell mid-surface as a reference, evaluate the approximate normal distance from the sub-model nodes to the mid-surface to verify whether the accuracy requirement is satisfied. As shown in Fig. 1, when this distance falls within the permissible range, the closest node in the global model can be selected as the driving node. This ensures that the sub-model remains consistent with the global model in terms of both geometric position and mechanical behavior, thereby enhancing the accuracy and stability of the computational results. Compared with the solid-model-driven approach, this method not only better reflects the stress characteristics of shell structures but also significantly reduces the computational scale, offering distinct advantages in the analysis of thin-walled structures.

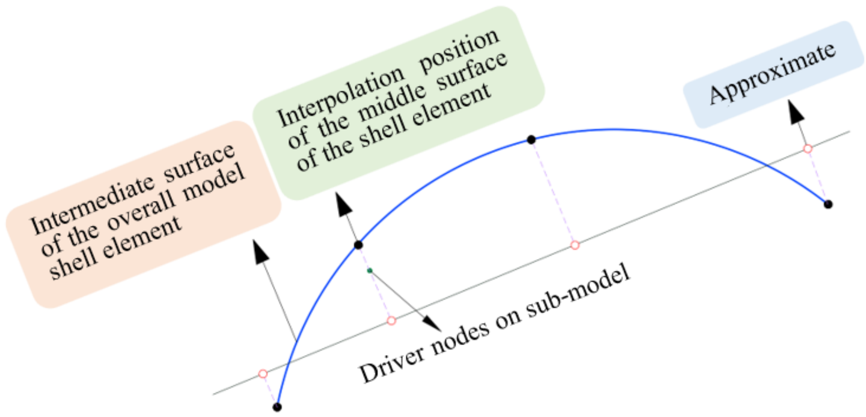


Fig. 1. Shell global model driving a shell sub-model

The principles of a solid global model driving a solid sub-model and a shell global model driving a solid sub-model are similar, both being realized by establishing a correspondence between the nodes of the sub-model and those of the global model, with differences only in the criteria for judgment and the selection of degrees of freedom. As their application scenarios are relatively limited, a detailed discussion is not provided here.

The analysis process of the sub-model method is shown in Fig. 2.

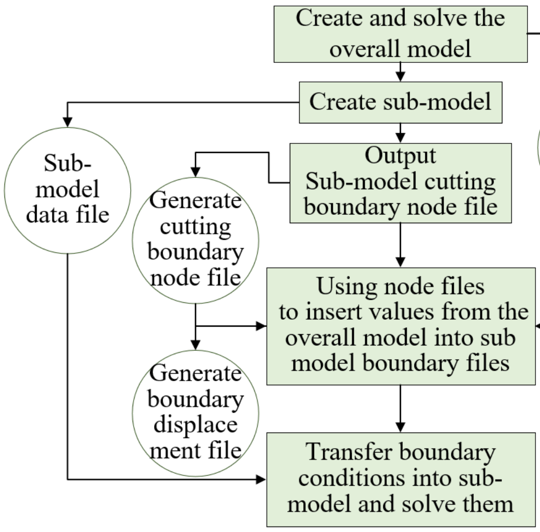


Fig. 2. Analysis process of the sub-model method

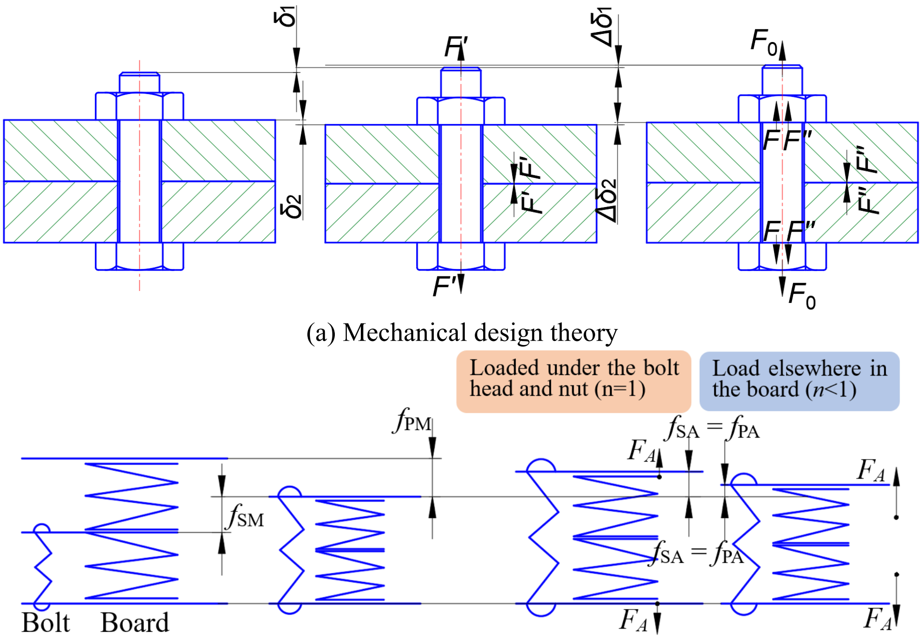
3. Key technology of high-stress bolt strength evaluation

Based on mechanical design theory and the VDI 2230 standard [12], research was conducted on the key technologies and theories applicable to the strength assessment of connection bolts for heavy-duty suspended equipment of high-speed trains.

3.1 Bolt strength evaluation method

Mechanical design theory divides bolt connections into two categories: loose connections and tight connections [13]. The suspension equipment bolts need to be pre tightened, which belongs to the tight bolt connection that bears the pre tightening force F' and the axial working load F. As shown in Fig. 3(a), its deformation under stress is divided into three stages: no deformation when not loaded; Under pre tightened state, the bolt elongates by δ_1 and is compressed by the connected component by δ_2; Under working conditions, the total tensile force of the bolt increases to F_0, and the deformation amount is (δ_1+$\Delta\delta_1$). The remaining pre tightening force decreases to F'' and the deformation amount of the connected part is (δ_2-$\Delta\delta_2$) [14].

The VDI 2230 standard takes into account bolt connection forms, material properties, and loading methods, simplifying bolts and clamped parts into tension springs and compression springs, respectively. As shown in Fig. 3(b), its mechanical behavior is divided into three stages: no deformation in the initial state; When in the assembled state, the bolt is subjected to a pre tightening force F_M and elongates f_{SM}, generating a clamping force F_K and compressing f_{PM} when the connected part is compressed; When in working condition, the bolt tension increases to (F_M+F_{SA}) and the deformation is (f_{SM}+f_{SA}). The compression force of the connected part decreases to (F_K-F_{PA}) and the deformation is (f_{PM}-f_{PA}). According to the deformation coordination condition, (f_{SA}=f_{PA})。



(a) Mechanical design theory

(b) VDI 2230 standard

Fig. 3. Working principle of bolt

When the strength of bolts is checked by using the mechanical design theory, the following calculations are required. When the calculation results meet the following conditions, the bolts are qualified:

(1) Safety factor of static strength $S \geqslant 1.2$

The designer shall provide the diameter, strength grade, bolt model and tightening torque of the bolt, and the preload Q_P of the bolt is:

$$Q_P = \frac{T}{0.2d} \tag{4}$$

where T is the tightening torque of the bolt, and d is the nominal diameter of the bolt.

Considering that the bolt is subjected to shear force during pre-tightening, the axial force is increased by 30% according to the fourth strength theory, so the strength calculation formula of the dangerous section is:

$$\sigma_{ca} = \frac{1.3Q}{A} \tag{5}$$

where σ_{ca} is the equivalent stress; A is the dangerous cross-sectional area of the bolt;

Q is the axial force of bolt.

$$S \le \frac{\sigma_s}{\sigma_{ca}} \tag{6}$$

where S is the safety factor of static strength; σ_s is the yield strength of the bolt material.

(2) Fatigue strength safety factor $S_a \geqslant 2.5$.

When the bolt is subjected to alternating load, the fatigue strength of the bolt needs to be checked. The calculated stress amplitude of the bolt is calculated by the following formula:

$$\sigma_a = \frac{C_1}{C_1 + C_2} \times \frac{2F}{\pi d_1^2} \tag{7}$$

where σ_a is the stress amplitude of the bolt; C_1 is the stiffness of the bolt, C_2 is the stiffness of the connected part, F is the axial tension, and d_1 is the diameter of the bolt.

$$S_a \le \frac{\varepsilon \sigma_{-1t}}{\sigma_a K_\sigma} \tag{8}$$

where S_a is the fatigue strength safety factor; ε is the dimension factor; σ_{-1t} is the tensile fatigue limit of bolt; K_σ is the effective stress concentration factor.

Based on the VDI 2230 standard, an executable strength evaluation process for high-stress bolts is proposed. This process consists of a total of 14 steps, as shown in Fig. 4.

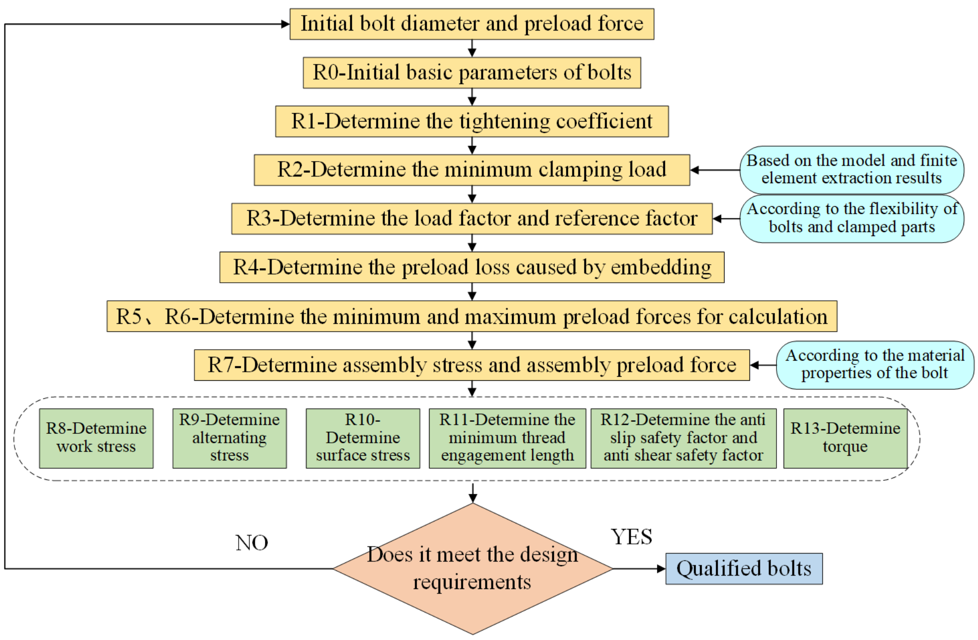


Fig. 4. Strength evaluation process of high-stress bolt based on VDI 2230 standard

In the process of evaluating the strength of bolted connections according to the VDI 2230 standard, the following calculations need to be carried out and the results meet the relevant thresholds:

(1) Work stress safety factor $S_F > 1$

$$\sigma_{red,B} = \sqrt{\sigma_{Z\max}^2 + 3(k_\tau \cdot \tau_{\max})^2} \tag{9}$$

where $\sigma_{red,B}$ is the equivalent stress in the working state; $\sigma_{Z\max}$ is the maximum tensile stress; k_τ introduces a reduction factor for torsional shear stress; $\tau_{\max}$ is the maximum torsional shear stress.

$$S_F = \frac{R_{P0.2\min}}{\sigma_{red,B}} \tag{10}$$

where S_F represents the working stress safety factor; $R_{P0.2\min}$ is the minimum yield stress of the bolt.

(2) Alternating stress safety factor $S_D \geq 1.2$

$$\sigma_a = \frac{(F_{SAo} - F_{SAu})}{2A_s} \tag{11}$$

where σ_a represents the continuous alternating stress amplitude of the bolt; F_{SAo} is the maximum bolt additional load; F_{SAu} is the minimum bolt additional load; A_s is the cross-sectional area of thread stress.

$$S_D = \frac{\sigma_{AS}}{\sigma_a} \tag{12}$$

where S_D represents the alternating stress safety factor; σ_{AS} is the allowable stress amplitude for bolts.

(3) Surface pressure safety factor $S_P \geq 1$

$$P_{M\max} = \frac{F_{Mzul}}{A_{P\min}} \leq P_G \tag{13}$$

where $P_{M\max}$ represents the maximum surface pressure in the assembled state between the bolt and the clamped component; F_{Mzul} is the allowable pre-assembly load; $A_{P\min}$ is the minimum bearing area of the bolt head or nut; P_G is the allowable compressive stress.

$$P_{B\max} = (F_{V\max} + F_{SA\max} - \Delta F_{Vth}) / A_{P\min} \leq P_G \tag{14}$$

where $P_{B\max}$ is the maximum surface pressure between the bolt and the clamped part in the working state; $F_{V\max}$ is the maximum remaining preload load; $F_{SA\max}$ is the maximum axial additional bolt load; ΔF_{Vth} is an additional thermal load.

$$S_p = \frac{P_G}{P_{M/B\max}} \tag{15}$$

where S_p is the surface pressure safety factor.

(4) Anti slip safety factor $S_G \geq 1.2$, anti shear safety factor $S_A \geq 1.1$

$$F_{KR\min} = \frac{F_{Mzul}}{\alpha_A} - (1 - \Phi_{en}^{*})F_{A\max} - F_Z - \Delta F_{Vth} \tag{16}$$

where $F_{KR\min}$ is the minimum residual preload force at the interface; α_A is the tightening coefficient; Φ_{en}^{*} is the load factor for eccentric clamping; $F_{A\max}$ is the maximum axial working load; F_Z is the pre-loaded loss caused by embedding.

$$S_G = \frac{F_{KR\min}}{F_{Kerf}} \tag{17}$$

where S_G represents the anti-slip safety factor; F_{Kerf} is the clamping load required to transmit lateral loads.

$$\tau_{Q\max} = \frac{F_{Q\max}}{A_\tau} \tag{18}$$

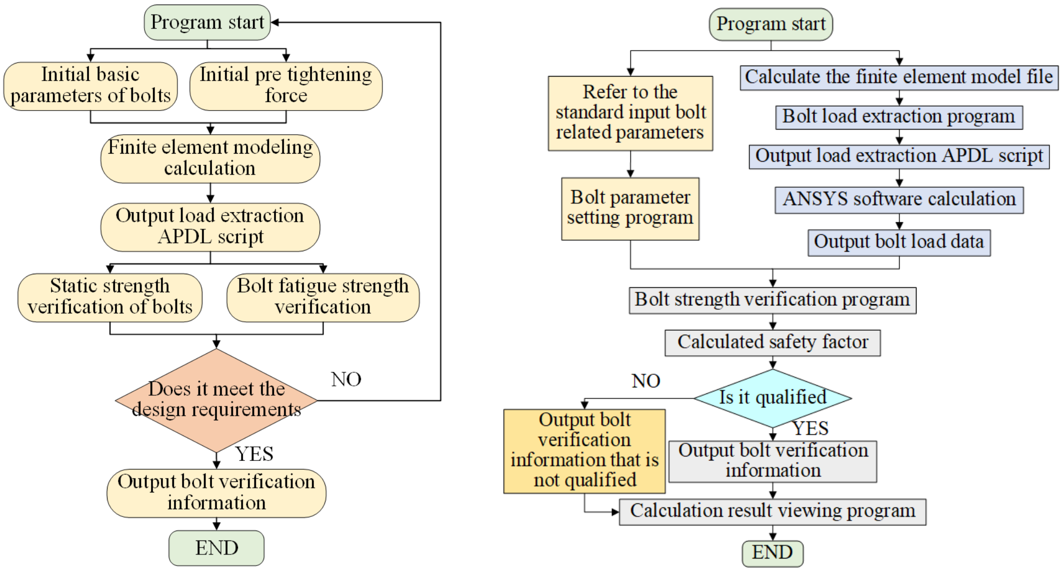
where $\tau_{Q\max}$ is the shear stress of the bolt cross-section; $F_{Q\max}$ is the maximum lateral force that the bolt can withstand; A_τ is the shear area at the contact surface.

$$S_A = \frac{\tau_B}{\tau_{Q\max}} \tag{19}$$

where S_A is the shear safety factor; τ_B is the allowable shear stress.

3.2 Programming of high-stress bolt strength evaluation method

In order to realize the bolt strength evaluation efficiently and conveniently, the bolt strength evaluation process is programmed using Python language. The technical route of bolt strength evaluation program based on mechanical design theory is shown in Fig. 5(a), and the technical route of bolt strength evaluation program based on VDI 2230 standard is shown in Fig. 5(b).



(a) Mechanical design theory (b) VDI 2230 standard

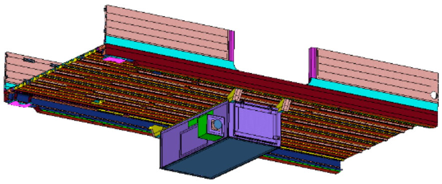
Fig. 5. Technology roadmap

4. Strength evaluation of high-stress connecting bolts for typical suspension equipment of high-speed train

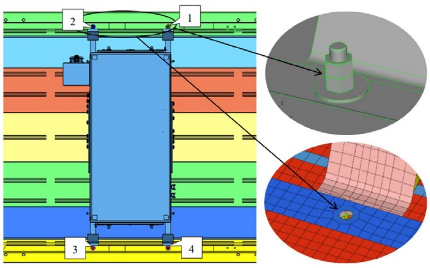
4.1 Finite element modeling and calculation conditions

Taking the high-voltage electrical box, a typical heavy-duty suspended equipment under the carriage of a high-speed train, as the research object, the strength of the bolts connecting it to the train body was evaluated. The established sub-model of the high voltage electrical box area is shown in Fig. 6 (a). The analysis model and numbering of the bolts are shown in Fig. 6(b).

In order to improve work efficiency, the geometric model was simplified for modeling. Simplified modeling should not only conform to the structural stress characteristics but also ensure the reliability of the calculation results [15]. In this paper, the Beam188 beam element was used to simulate the finite element model of the bolt. It was connected to the meshes of the high-voltage electrical box and the side beam area of the train body through the Rigid element in a co-node manner. The feasibility of this modeling method for the bolt structure has been verified in engineering products [16,17].



(a) Sub-model



(b) Bolt and No.

Fig. 6. Finite element model

According to the EN 12663 standard [18], the calculation conditions for evaluating the bolts of the high-voltage electrical box lifting structure are established, as shown in Table 1.

Table 1

Calculation condition

Static Strength					Fatigue Strength			
Number	Acceleration/g				Number	Acceleration/g		
	Longitudinal	Transverse	Vertical			Longitudinal	Transverse	Vertical
1	3		1		1	0.15	0.15	0.15+1
2	-3		1		2	0.15	0.15	-0.15+1
3		1	1		3	0.15	-0.15	0.15+1
4		-1	1		4	0.15	-0.15	-0.15+1
5			3		5	-0.15	0.15	0.15+1
6			-1		6	-0.15	0.15	-0.15+1
					7	-0.15	-0.15	0.15+1
					8	-0.15	-0.15	-0.15+1

The constraint application method for the strength evaluation of the high-voltage electrical box bolt connections is as follows: the vertical linear displacement is constrained at the air springs of the bolster, the lateral linear displacement is constrained at the lateral stoppers of the center pin seat, and the longitudinal linear displacement is constrained at the coupler plate at the second end. The constraint locations are shown in Fig. 7.

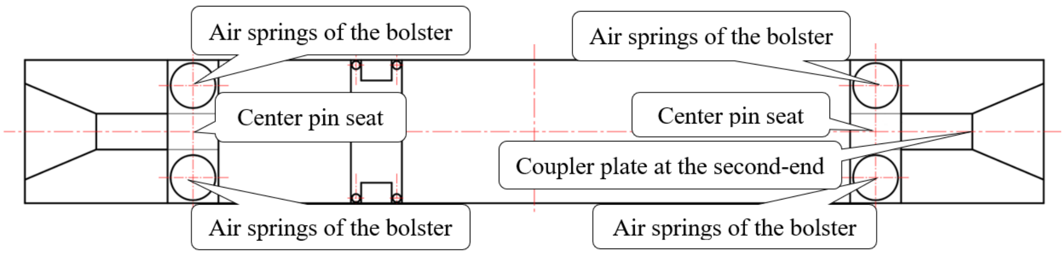


Fig. 7. Constraint location schematic

4.2 Establishment and rationality verification of the sub-model

To verify the reliability of the boundary conditions of the sub-model, it is necessary to compare the displacement and stress values between the sub-model and the overall model. Under the action of static strength condition 1, the comparison of the comprehensive displacement results and Von. Mises stress results at the same nodes of the sub-model and the overall model is shown in Table 2.

Table 2

Comparison of displacement and stress between sub-models and overall models

Position	Comprehensive displacement /mm			Von.Mises stress/MPa		
	Sub-model	Overall model	Error	Sub-model	Overall model	Error
Bottom of high-voltage electrical box	3.162	3.351	5.64%	5.768	6.12	5.75%
High-voltage electrical box bolt holes	2.944	3.08	4.42%	9.19	9.6	4.27%
Vehicle body frame	4.483	4.482	0.02%	7.154	7.14	0.20%
Vehicle side beam	3.001	3.015	0.46%	4.859	4.985	2.53%
Underframe groove	4.713	4.713	0	3.885	3.744	3.77%

It can be seen from Table 2 that the comprehensive displacement and stress values of the sub-model and the overall model of the high-voltage electrical box at the same node are basically the same, which can meet the engineering calculation requirements.

4.3 Bolt strength evaluation results

The No.1 bolt is determined as the most dangerous bolt through the program analysis, and its strength is evaluated according to the process in Fig. 5. The safety factor obtained is shown in Table 3.

Table 3

Safety factor assessment results

	Condition serial number	1	2	3	4	5	6	7	8
Machine Design method	S	1.613	1.573	1.593	1.592	1.608	1.578	-	-
	S_a	21.311	27.71	21.811	28.562	26.86	37.887	27.659	3.498
VDI 2230 standard	S_F	1.634	1.624	1.634	1.634	1.634	1.628	-	-
	S_P	2.015	2.015	2.015	2.015	2.015	2.015	-	-
	S_G	1.36	1.44	11.46	29.71	9.37	27.73	-	-
	S_A	19.06	22.44	88.22	418.01	67.38	202.14	-	-
	S_D	70.44	70.44	70.44	70.44	70.44	70.44	70.44	70.44

It can be seen from Table 3 that the safety factors under all working conditions meet the design requirements.

4.4 Analysis of assessment results

The results of strength evaluation of bolts using two methods meet the design requirements. The differences and applicability between mechanical design theory and VDI 2230 standard are mainly manifested in the following aspects:

(1) Differences in calculation elements: The former simplifies the interaction between bolts and connected components, while the latter comprehensively considers the connection form, material properties, loading method, and preload loss.

(2) Differences in mechanical models: The former is based on classical mechanical formulas, while the latter accurately characterizes mechanical properties through spring models.

(3) Applicability difference: The former is suitable for rapid estimation, while the latter satisfies complex working conditions through high-precision parameter input.

5. Conclusion

In order to conduct the strength assessment of high-stress connection bolts for heavy-duty suspended equipment of high-speed trains during the design phase, reduce the workload of full-scale tests and full-scale structural modeling of high-speed trains, lower the research and development costs, and improve the computational efficiency. Through numerical simulation methods, based on the sub-model method during the design phase, research on high-stress bolt strength assessment was carried out by utilizing mechanical design theory and the VDI 2230 standard.

(1) To address the challenges of full-scale car body testing and the low computational efficiency in bolt strength simulations, this study proposes a method using small-scale sub-models to replace full-vehicle modeling. Numerical simulations demonstrate that this sub-modeling technique can effectively reduce the strength evaluation cycle for high-stress bolts in high-speed trains, confirming its engineering feasibility.

(2) When using sub-models for high-stress bolt strength evaluation, sub-model boundaries must avoid potential critical areas, and their critical points should align with the full-scale structure. Boundary conditions should be constrained at regions with maximum stiffness based on directional stiffness characteristics. By comparing stress and displacement results between the sub-model and full-scale model, an accurate finite element numerical analysis model can be established.

(3) Research indicates that the sub-model method combined with bolt

evaluation theory is feasible for assessing the strength of high-stress connection bolts in high-speed train heavy-duty suspension equipment, but appropriate methods should be selected based on engineering requirements. Mechanical design theory offers simple calculations and low computational effort, suitable for rapid rough estimations. In contrast, the VDI 2230 standard comprehensively considers connection types, loading conditions, and temperature effects, requiring precise input parameters and offering thorough analysis, making it applicable to complex working conditions and high-precision scenarios.

REFERENCES

[1] *C. Petrescu, D. T. Cicic, C. Rontescu, G. Solomon and I. M. Vasile*, Research on welding reconditioning of the friction liner from passenger cars for railway safety improvement, U.P.B. Sci. Bull. Series D, Vol.**83**, Iss.3, pp.239-252, 2021.

[2] *G. Ilie, G. Popa and M. Andrel*, The Influence of Axle Load, Emu Composition and Runway Parameters on Accelerations at Start-up, U.P.B. Sci. Bull. Series D, Vol.**85**, Iss.2, pp.147-162, 2023.

[3] *Q.Y. Wu, M.M. Wang, S.N. Xiao, B. Yang, G.W. Yang, T. Zhu and W.J. Wang*, Study on the Strength of Sandwich Composite Skirt for High Speed Maglev Trains, Urban Mass Transit, Vol.**23**, Iss.9, pp.98-102, 2020.

[4] *J. Zhang, C. Zhao, L.J. Zhu and, X. Gao*, Research on the Loading Method and Strength Verification of Chassis Bolts for Rubber Rail Trains, Mechanical Engineering & Automation, Vol.**37**, Iss.5, pp.46-49, 2024.

[5] *X. Liu, H. Huang, X.Z. Kuang and, R. Lai*, Simulation analysis of anti climbing bolts for standard subway trains based on ANSYS, Electric Locomotives & Mass Transit Vehicles, Vol.**47**, Iss.4, pp.68-72, 2024.

[6] *J. Qu, W. Wang, B. Wang, G. Li, and B. Jiao*, Analysis of the axial load of bolts of wheel-mounted brake discs of high-speed trains, *Engineering Failure Analysis*, Vol.**137**,106250, 2022.

[7] *L. Liu and H. K. Jiang*, A Design Method for Determinning Non-standard Rectangular Spiral Spring Stiffness Based on Similarity Theory and FEM Method, U.P.B. Sci. Bull. Series D, Vol.**85**, Iss.1, pp.149-160, 2023.

[8] *J.Z. Wang, P. Zhang, J. Liu and Y.S. Cheng*, Collaborative optimization method for ship cabin structure proxy model based on sub-model decomposition, Chinese Journal of Ship Research, Vol.**19**, Iss.2, pp.98-106, 2024.

[9] *W.P. Zhao, G.L. Zhang, P.P. Zhi, L.K. Guo, Y. Li and Z.M. Wang*, Optimization and Experimental Verification of the Body Structure of Light Railway vehicles, Journal of Machine Design, Vol.**38**, Iss.10, pp.114-120, 2021.

[10] *G.M. Yuan, Q.B. Wang, G.X. Jing, S.Z. Wang, G.C. Wang, X.C. Zeng and L.Q. Zhang*, Method for fatigue strength analysis of main bearing threads based on sub models, Chinese Internal Combustion Engine Engineering, Vol.45, Iss.1, pp.37-46, 2024.

[11] *Z.F. He*, Fatigue simulation analysis and reliability modeling of wind turbine blades based on sub-model technology, Guangzhou University, 2024.

[12] VDI 2230 Blatt1-2003, “Systematic calculation of high duty bolted joints-Joints with one cylindrical bolt”, Verein Deutscher Ingenieure, 2003.

[13] *D.X. Cheng, D.F. Wang, X.Q. Han, Y. Jiang, C.S. Li and X.Y. Wang*, Mechanical Design Manual (5th edition), Chemical Industry Press, 2008.

[14] *J.F. Jiao, L.X. Fan, Y. Li, Z.Z. Wei and Y. Liu*, Experimental study and simulation analysis of constant amplitude fatigue performance of hinge joint bolt and pin shaft, Journal of Lanzhou University of Technology, Vol.**50**, Iss.2, pp.111-117, 2024.
[15] *S.J. Yu*, Study on fatigue strength of high-speed train body under aerodynamic load, Southwest Jiaotong University, 2016.
[16] *Q. Wang and G. Xu*, 2005, Theoretical basis and selection method of ANSYS beam element, Journal of Three Gorges University (Natural Science Edition), Vol.**27**, Iss.4, pp.336-340.
[17] *L. Wang and D.X. Leng*, Study on the Influence of Shear Coefficient on Timoshenko Beam Model, Missiles and Space Vehicles, Vol.**37**, Iss.3, pp.57-59+68, 2015.
[18] EN 12663, “Railway Applications-Structural Requirements of Railway Vehicle Bodies”, European Committee for Standardization (CEN), 2010.