Research Article

Research on the Numerical Calculation Method for Antiloosening Performance of Screwed Joints under Complex Working Conditions

Yimin Mo,1 Shenghui Guo,1 Xiongzhen Qin,2 Jialiang Qin,2 Kui Zhan,1 and Xinshang Gao1

1School of Mechanical and Electronic Engineering, Wuhan University of Technology, Wuhan 430070, China
2SAIC-GM-Wuling Automobile Co., Ltd., Liuzhou 545007, China

Correspondence should be addressed to Shenghui Guo; shenghui.guo@whut.edu.cn

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This study aimed at addressing the difficulties entailed in accurately determining the working loads of screwed joints (SJs) by establishing mechanical models and verifying the accuracy of the numerical calculation model of antiloosening performance under complex working conditions. First, considering the slip state of the interface and the stress state of the thread surface, a corresponding mechanical model was established to investigate the quantitative model of the interaction amongst structural parameters, complex working loads, and antiloosening performance of SJs. The applicability of existing models is expanded by this new model. Second, a load calibration test, an actual working condition test, and a dynamic simulation were combined to accurately determine the load under complex working conditions. A new experimental scheme for measuring the critical residual preload was employed to verify the reliability and accuracy of the numerical calculation model. The results confirmed that structural safety is ensured and that accident risk is reduced. Finally, based on this model, the transverse load, axial load, bending moment, torque about the bolt axis, clamping eccentricity, loading eccentricity, and coefficient of friction in the thread and at the interface were analyzed in terms of the antiloosening performance. The results of this study are expected to provide significant guidance to engineering practices. Moreover, the numerical calculation model can accurately predict the antiloosening performance and failure and also provide technical support for improving the structural reliability, particularly for key screwd-joint structures (SJSs), under complex working conditions loading.

1. Introduction

Screwed joints (SJs) are widely used in various mechanical structures owing to their advantages of having a simple structure, easy assembly or disassembly and adjustment, and so on. Screwed-joint performances (SJP) are important for obtaining the overall structural characteristics and estimating the probability of the main cause for structural damage and failure. Screwed-joint structures (SJSs) typically undergo loosening failure during service, which may lead to product failure or major accidents. The loss of preload, which is directly linked to loosening, is affected by external loads and structural parameters [1–6]. The external loads break the original force balance at the SJS, and the preload reduction at the SJS is caused by plastic deformation and slipping. The structural parameters and preload exert significant influence on the distribution and transmission of external loads.

In engineering practice, different structural parameters (shape, size, material properties, clamping eccentricity, loading eccentricity, coefficient of friction in the thread and at the interface, and so on) and loads (preload, transverse load, axial load, bending moment, torque about the bolt axis, and so on) are considered under complex working conditions. The study of the interaction between the loads and the structural parameters is complex because the loosening of SJSs is also a complicated process. In studies investigating the influence of loads and parameters
on loosening to establish a simple and effective numerical model and accurately predict the antiloosening performance and failure, the following challenges have been encountered: (1) owing to the complexity of the working conditions and loads, the occurrence of substantial error affects the load measurement by instruments and equipment and makes it difficult to accurately obtain the load values; (2) the consideration of complex working loads, slip state of the interface, stress state of the thread surface, influences of eccentric clamping and eccentric loading on the loads, elastic resilience, and load coefficient make it difficult to establish the mechanical models of SJs; (3) the complexity of working loads and the difficulty of measuring the antiloosening performance complicate the process of constructing a test method for verifying the accuracy of the numerical calculation model.

Research on the loosening problems of SJs has been conducted since the 1940s [1]. Goodier and Sweeney [2] proposed the theory and model for the occurrence of loosening during the dynamic loading of SJs and thus contributed toward understanding the parameters related to loosening, such as the bolt diameter and thread pitch. Clark and Cook [3] investigated the effect of fluctuating torque on the loosening of a tightly seated bolt. Until then, researchers had focused on the loosening caused by axial loads. In contrast, Junker [4] experimentally determined that the transverse loads result in greater risk than the axial loads, and his test apparatus for assessing the loosening according to the vibration, called the “Junker test machine,” has been widely used. However, because this test machine can only apply transverse loading to bolts, it cannot be used for simulation and failure prediction in engineering practice. Finkelston [5] investigated the relationship between the self-loosening of bolted joints and influencing parameters such as vibration amplitude, initial preload, thread pitch, and surface characteristics. However, he did not investigate the interactions amongst these factors. Moreover, the VDI-2230 standard [6] ensures a satisfactory antiloosening performance of the structure by calculating the safety margin against interface slippage during the design of SJSs. The limit value of the safety margin is determined by the user. Gong and Liu developed a three-dimensional finite element model to investigate the effects of the preload generation, vibration parameter, and material model on the loosening [7] and identified the critical transverse force for initiating loosening [8]. Zhao et al. [9] proposed a simplified numerical model of bolt slipping for simulating SJs and better capturing the slipping phenomenon. Zhu et al. developed a high-precision instrument to determine the effect of variables on the initial loss of preload [10] and proposed a torque-preload force formula for evaluating the antiloosening performance of thread fasteners [11]. Jiang et al. [12–14] found that the external lateral load has a critical value. When the load is lower than the critical value, the bolt does not loosen. Pai and Hess [15] pointed out that the overall slippage of the bearing surface under the screw head is a necessary condition for the loosening by rotation of the bolt and nut. In the literature [16–18], it is reported that, under the repeated action of a transverse load, the local slip of the bearing surface and the thread surface under the screw head gradually accumulate elastic strain energy at the contact surface. When energy is accumulated to a certain extent, the entire contact surface exhibits slippage, which leads to bolt loosening. Dinger [19] employed the critical loosening gradient of a screw head (0.01/\text{cycle}) to evaluate the loosening of a bolted joint. Izumi et al. investigated the mechanisms of loosening caused by microbearing-surface slippage under transverse loading within the framework of the three-dimensional finite element method [20] and found that the loosening commenced when the thread surface exhibited complete slippage, regardless of the slip status of the bearing surface [21]. Gong et al. [22] analyzed local slippage accumulation on the bearing surface using the modified Iwan’s model and developed a thorough understanding of the loosening mechanism. Various theoretical models [23–33] have been proposed to understand the mechanism of loosening caused by complete slippage under transversal vibration. Yokoyama et al. [34] investigated the loosening of bolted joints subjected to cyclic torquing.

In summary, existing studies have mainly investigated the influence of the SJS parameters on antiloosening performance under simple working conditions loading. However, the exact quantitative relationship amongst the structural parameters, complex working loads, and loosening has not yet been obtained. Moreover, VDI-2230 [6] provides a method for calculating the safety margin against interface slippage, whose limit value is determined according to experience or by the users to prevent the interface slippage and bolt/nut loosening by rotation. However, the mechanism of loosening by rotation under complex working conditions is not fully considered; thus, the limit value of the safety margin cannot be accurately determined. Other studies have mainly investigated the influence of a single transverse load or torque on the preload loss, interface slippage, or bolt/nut loosening by rotation. However, in engineering practice, the structural parameters and loads on the SJSs are complex, and the results obtained by existing studies do not accurately predict the antiloosening performance and loosening failure. Therefore, in the design process, it is necessary to comprehensively consider the intrinsic relationship amongst the structural parameters, complex working loads, and antiloosening performance.

In this study, an investigation and a verification scheme for the numerical calculation model of antiloosening performance under complex working conditions loading were constructed, as shown in Figure 1. The following process was employed: (1) load data (Figure 1(e)) were obtained by conducting load calibration tests (Figure 1(b)) and tests under actual working conditions (Figure 1(c)) to calibrate the dynamics simulation model (Figure 1(a)); thereafter, the typical, complex working conditions were simulated, and the load data (Figure 1(f)) of the SJSs were output. (2) The mechanical model of the SJSs was established by considering the clamped parts and fastener model (Figure 1(d)), complex working loads, eccentric clamping, and eccentric loading (Figure 1(h)). Based on the slip state of the interface of clamped parts, criteria for determining the interface slip were established,
Figure 1: Research and verification scheme for numerical calculation model of antiloosening performance under complex working conditions loading. (a) Dynamic simulation model. (b) Load calibration tests. (c) Tests under actual working conditions. (d) Clamped parts and fastener model. (e) Load data obtained by load calibration tests and tests under actual working conditions. (f) Load data obtained by dynamics simulation. (g) Preload measurement by the ultrasonic measuring system. (h) The mechanical model of SJSs under complex working loads. (i) The mechanical model of the bolt thread surface under complex working loads. (j) The microarea mechanical model at point B on the thread. (k) Loosening state of the SJS during the tests under actual working conditions.
and the numerical calculation model was derived. Considering the stress state of the thread surface, the mechanical models of the thread surface (Figure 1(i)) and the thread microarea (Figure 1(j)) were established to investigate antiloosening performance. (3) The measuring scheme of the critical residual preload for loosening by rotation under actual working conditions was constructed to verify the numerical calculation model (Figures 1(g) and 1(k)).

Based on the abovementioned scheme, the numerical calculation model of the interaction amongst the structural parameters, complex working loads, and loosening by rotation was established to expand the application scope of existing models and predict the performance and failure of SJs. Subsequently, a new experimental scheme (including the load extraction under complex working conditions, measurement of critical residual preload, and two practical engineering cases) was used to confirm the reliability and accuracy of the numerical calculation model. Finally, based on the model, the effects of transverse load, axial load, bending moment, torque about the bolt axis, clamping eccentricity, loading eccentricity, and coefficient of friction in the thread and at the interface were analyzed in terms of antiloosening performance.

2. Mechanical Models

By considering the slip state of the interface and the local stress state of the thread, mechanical models were established to investigate the numerical calculation model of the interaction amongst the structural parameters, complex working loads, and antiloosening performance. This study divided the loosening process of the SJSs into the following three stages, based on previous studies [12–14], which divided the loosening process into two stages.

Stage 1 (plastic deformation of materials). The attenuation of the preload is mainly caused by the plastic deformation and expansion of the material. If the preload can ensure that relative slip does not occur at the interface of clamped parts, the bolt, nut, and thread surface will not be affected by external transverse loads and will not loosen by rotation.

Stage 2 (relative slip of clamped parts). With the continuous attenuation of the preload in the first stage, the interface of the clamped parts relatively slips owing to the decrease of the friction force, and the forces at the thread surface and bearing surface change. The bolt’s bending deformation can absorb the transverse load, which is small, and the bearing surface and thread surface do not slip and rotate owing to the frictional resistance. Therefore, the bolt and nut do not loosen by rotation.

Stage 3 (loosening by rotation of the bolt/nut). With further preload attenuation, the relative slip and rotation of the bolt and nut occur when the loads on the structure reach a critical value, and the forces exerted on the bearing surface and thread surface are greater than the friction resistance.

In the first and second stages, the preload attenuation is mainly caused by the plastic deformation of the material, and the decrease is slow. In the third stage, the preload rapidly decreases, which results in loosening failure. Therefore, it is necessary to ensure that the SJSs are in the first two stages throughout the service life and avoid them entering the third stage. When the SJSs must satisfy the requirements of stability, sealing, and corrosion resistance, they must be in the first stage and the relative slip of the interface is not allowed. The numerical calculation model for the loosening process of the SJSs under complex working conditions loading was investigated as described below.

Owing to the elastic characteristics of the clamped parts and the bolt/nut, the axial load and bending moment on the clamped parts are proportionally transmitted to the bolt. Simultaneously, the plastic deformation of the bolt and clamped parts reduce the clamp load on the interface. When calculating the residual clamp load $F_{KR}$ and safety margin against the interface slipping $S_C$, VDI-2230 [6] only considers the axial load and loss of preload caused by plastic deformation; it does not consider the influence of the bending moment $M_B$ on the clamping load of the interface under eccentric clamping and eccentric loading, which affects the accuracy of the interface’s antislapping safety verification. Considering the abovementioned factors, the mechanical model was established (Figure 2), and the residual clamp load in the clamping area was calculated using the following equation:

$$F_{KR} = F_M - (1 - \Phi^* \cdot F_A) + \left(\frac{\Phi^* \cdot F_A}{s_{sym}}\right) M_B - F_Z,$$

where $F_Y$ is the preload (general); $F_M$ is the assembly preload; $\Phi^* \cdot F_A$ is the load factor for the eccentric clamping and eccentric loading; $F_A$ is the axial load; $\Phi^*$ is the load factor for the moment loading and eccentric clamping; $s_{sym}$ is the clamping eccentricity, whose sign rule is given in VDI-2230 [6]; $M_B$ is the bending moment; and $F_Z$ is the loss of preload caused by plastic deformations, which can be calculated according to VDI-2230 [6].

Thus, the interface’s antislapping verification criterion is established. First, the residual clamp load in the clamping area $F_{KR}$ is calculated under complex working conditions. Second, the clamp load $(F_Q / (q_F \cdot \mu_T)) + (M_Y / (q_M \cdot r_a \cdot \mu_T))$ is calculated to transfer the transverse load $F_Q$ and the torque about the bolt axis $M_Y$. Third, the residual clamp load is compared with the required clamp load. In particular, if the residual clamp load is less than the required clamp load, the interface is assessed as slip; otherwise, it is assessed as no slip, as expressed in the following equation:

$$F_{KR} > \frac{F_Q}{q_F \cdot \mu_T} + \frac{M_Y}{q_M \cdot r_a \cdot \mu_T},$$

where $F_Q$ is the transverse load; $M_Y$ is the torque about the bolt axis; $q_F$ and $q_M$ denote the number of
force-transmitting and torque-transmitting interfaces, respectively; \( \mu_T \) is the coefficient of friction at the interface; and \( r_a \) is the friction radius at the clamped parts under the action of \( M_Y \).

The critical condition for the loosening by rotation of the bolts/nuts was investigated based on the interface’s antislapping verification criterion. When the interface is assessed to slip according to the verification criteria, that is, when equation (2) is not verified, the axial load \( F_{AS} \) of the bolt bears, which is the proportional transmission of the axial load \( F_A \) and transverse load \( F_Q \), and torsion about the bolt axis \( M_{YS} \), all of which result from the transverse load \( F_Q \) and torque about the bolt axis \( M_Y \) overcoming the friction, can be calculated as expressed by the following equations:

\[
F_{AS} = \Phi_{en}^{\ast} \cdot F_A, \quad (3)
\]

\[
F_{QS} = F_Q + \frac{M_Y}{r_a} - F_{KR} \cdot \mu_T \cdot q, \quad (4)
\]

\[
M_{YS} = M_Y - (F_{KR} \cdot \mu_T \cdot q - F_Q) \cdot r_a. \quad (5)
\]

If the interface does not slip, the bending moment \( M_{SB} \) acting on the bolt can be calculated according to VDI-2230 [6], as expressed by equation (6). By considering that the bolt bears the additional bending moment \( F_{QS} \cdot l_K \) caused by the transverse load \( F_Q \) when the interface slips, the bending moment acting on the bolt can be derived as expressed by equation (7):

\[
M_{SB} = \frac{\beta_p}{\beta_s} \left[ F_A \cdot a - \Phi_{en}^{\ast} F_A \cdot s_{sym} + M_B \left( 1 - \frac{s_{sym} \Phi_m^{\ast}}{s_{sym}} \right) \right], \quad (6)
\]

\[
M_{SB} = F_{QS} \cdot l_K + \frac{\beta_p}{\beta_s} \left[ F_A \cdot a - \Phi_{en}^{\ast} F_A \cdot s_{sym} + M_B \left( 1 - \frac{s_{sym} \Phi_m^{\ast}}{s_{sym}} \right) \right]
\]

\[
+ M_B \left( 1 - \frac{s_{sym} \Phi_m^{\ast}}{s_{sym}} \right) - F_{QS} \cdot l_K \]

where \( l_K \) is the clamping length; \( \beta_p \) is the elastic bending resilience of the clamped parts; and \( \beta_s \) is the elastic bending resilience of the bolt.

Based on [24], a mechanical model of the bolt thread surface under complex working loads is established, as shown in Figure 3. \( DD' \) is a line passing through the center and along the same direction as the transverse load. The different positions of the circumferential direction of the thread surface are represented by the angle \( \theta \) between the radius of the cross-cutting circle and line \( DD' \); the range of \( \theta \) is \([0^\circ, 360^\circ] \). Additionally, \( \theta = 0^\circ \) corresponds to the location closest to the action point of the transverse load, and \( \theta = 180^\circ \) corresponds to the location farthest from the action point of the transverse load. As the radial dimension of the thread surface is small, it is assumed that the stress distribution along the radial direction is uniform. A pair of thread pairs is equivalent to the mass block/bevel model, and the bevel angle is equal to the lead angle of the thread. Figure 4 [24] can be used to analyze the forces acting on the microarea at point \( B \) on the thread, wherein the square represents the bolt thread, and the bevel represents the nut thread.

Moreover, \( S_{Q1} \) and \( S_{Q2} \) are the transverse stresses generated by the transverse loads and torque about the bolt axis on the microarea of the bolt thread; \( S_Q \) and \( S_{sym} \) are the transverse and axial combined stresses on the microarea of the bolt thread, as calculated by the following equations:

\[
S_{Q1} = \frac{F_{QS}}{A_{slip}}, \quad (8)
\]

\[
S_{Q2} = \frac{2M_{YS}}{d_2 \cdot A_{slip}}, \quad (9)
\]

\[
S_Q = \sqrt{S_{Q1}^2 + S_{Q2}^2 + 2S_{Q1} \cdot S_{Q2} \cdot \cos \alpha},
\]

\[
S_A = \frac{F_{AS}}{A_{slip}} - \frac{M_{sb} \cdot d_2 \cdot \cos \theta}{2I_z}, \quad (10)
\]

\[
A_{slip} = \frac{\pi \cdot (d_2^2 - d_1^2)}{4}, \quad (11)
\]
where \( A_{slip} \) is the contact area of a single-thread surface; \( d \), \( d_1 \), and \( d_2 \) are the outside diameter, minor diameter, and pitch diameter of the thread, respectively; \( I_z \) is the moment of inertia of the lateral section of the bolt; and \( \alpha \) is the angle between the two transverse stresses, and it is deduced to be related to \( \theta \), as expressed in the following equation:

\[
\cos \alpha = -\sin \theta. \tag{12}
\]

Moreover, \( \sigma_Q \) and \( \tau_Q \) are the normal and tangential stresses generated by \( S_Q \) on the thread surface, as calculated by equation (13); \( \sigma_A \) and \( \tau_A \) are the normal and tangential stresses generated by \( S_A \) on the thread surface, as calculated by equation (14); \( \lambda \) is the angle between the transverse stress and its projection line on the thread surface; \( \beta \) is the angle between the axial stress and its normal line on the thread surface, whose value is equal to the lead angle; \( \gamma \) is the angle between \( \tau_Q \) and \( \tau_A \):

\[
\sigma_Q = S_Q \cdot \sin \lambda, \tag{13}
\]

\[
\tau_Q = S_Q \cdot \cos \lambda, \tag{14}
\]

\[
\sigma_A = S_A \cdot \cos \lambda, \tag{13}
\]

\[
\tau_A = S_A \cdot \sin \lambda. \tag{14}
\]

The angle \( \lambda \) is related to \( \beta \) and \( \theta \), and the angle \( \gamma \) is related to \( \theta \) [35], as expressed in the following equations:

\[
\sin \lambda = \sin \beta \cdot \sin \theta, \tag{15}
\]

\[
\cos \gamma = \begin{cases} \sin^2 \theta, & \theta \in [0^\circ, 180^\circ], \\ -\sin^2 \theta, & \theta \in (180^\circ, 360^\circ]. \end{cases} \tag{16}
\]

The tangential combined stress \( \tau \) and friction stress \( f \) on the thread surface are calculated using the following equations:
loosening by rotation. Flank SJSs are determined, the critical residual preloads for the vehicle road test and dynamics simulation. The testing working conditions by combining a load calibration test and Accurate load extraction was achieved under complex loosening by rotation. Considering the SJSs of a certain vehicle as examples, the 3. Verification of Numerical Calculation Model of Antiloosening Performance

According to previous studies [15, 16], in the process of bolt/nut loosening by rotation, the thread surface rotates first, drives the rotation of the bearing surface of the bolt head, and the local slip of the thread surface causes loosening by rotation. Therefore, the condition for the loosening by rotation not occurring is that the tangential combined stress at any position on the thread surface is less than or equal to the friction stress, as expressed by the following equation:

$$\tau = \sqrt{r_A^2 + r_Q^2 + 2r_Ar_Q\cos \gamma}$$

$$= \begin{cases} \sqrt{r_A^2 + r_Q^2 + 2r_Ar_Q\sin^2 \theta}, & \theta \in [0^\circ, 180^\circ], \\ \sqrt{r_A^2 + r_Q^2 - 2r_Ar_Q\sin^2 \theta}, & \theta \in (180^\circ, 360^\circ]. \end{cases}$$ (17)

$$f = \begin{cases} (\sigma_A - \sigma_Q) \cdot \mu_G, & \theta \in [0^\circ, 180^\circ], \\ (\sigma_A + \sigma_Q) \cdot \mu_G, & \theta \in (180^\circ, 360^\circ]. \end{cases}$$ (18)

$$\tau - f = \begin{cases} \sqrt{r_A^2 + r_Q^2 + 2r_Ar_Q\sin^2 \theta} - (\sigma_A - \sigma_Q) \cdot \mu_G \leq 0, & \forall \theta \in [0^\circ, 180^\circ], \\ \sqrt{r_A^2 + r_Q^2 - 2r_Ar_Q\sin^2 \theta} - (\sigma_A + \sigma_Q) \cdot \mu_G \leq 0, & \forall \theta \in (180^\circ, 360^\circ]. \end{cases}$$ (19)

$$F_{VR}^l = F_{VR} + \frac{P\varphi}{360^\circ(\delta_p + \delta_s)},$$ (20)

where $P$ is the pitch; $\delta_p$ is the elastic resilience of the clamped parts; and $\delta_s$ is the elastic resilience of bolts.

3.3. Application Case for Verification of Numerical Calculation Model. Two sets of SJSs were selected (shock absorber-body SJS and crossarm-subframe SJS) to verify the numerical calculation model through the following key steps:

(1) Determining the SJS parameters, simulating the typical complex working conditions using a dynamic simulation model, and obtaining the ultimate working loads on the SJSs under the working condition of passing through a single-side pothole, as presented in Table 3.

(2) For the assembly preload $F_M$, assessing whether the bolt/nut will loosen by rotation. Calculating the loss of preload $F_Z$ according to VDI-2230 [6]; calculating the residual clamp load $F_{KR}$, axial load $F_{AS}$, transverse load $F_{QS}$, torsion about the bolt axis $M_{YS}$, and bending moment $M_{SB}$ according to equations (2), (4)–(6), and (8); calculating max$|\tau - f|$, $\forall \theta \in [0^\circ, 360^\circ]$ according to equations (9)–(20); if max$|\tau - f| \leq 0$, the bolt/nut will not loosen by rotation; if max$|\tau - f| > 0$, the bolt/nut will loosen by rotation. The calculation and assessment results are presented in Table 4.

(3) Based on the parameters and working loads listed in Table 3, calculating the critical residual preloads for the loosening by rotation $F_{VR}^l$, that is, the solution of $F_M - F_Z$ is obtained when max$|\tau - f| = 0$, according to equations (2), (4)–(6), and (8)–(20). The critical residual preloads of the two SJS sets are 32,450 N and 19,865 N, respectively.

The measurement data of the residual preload and rotation angle of the marking line are presented in Table 5. According to equation (1), the critical residual preloads for

3. Verification of Numerical Calculation Model of Antiloosening Performance

Considering the SJSs of a certain vehicle as examples, the accuracy of the numerical calculation of the antiloosening performance model was verified by conducting a vehicle road test to measure the critical residual preload of the loosening by rotation.

3.1. Load Extraction under Complex Working Conditions. Accurate load extraction was achieved under complex working conditions by combining a load calibration test and vehicle road test and dynamics simulation. The testing systems and procedures are listed in Table 1. The dynamic model comprises the body, front and rear suspension, steering system, power assembly, and tire and braking system and includes 46 rigid bodies in total. The number and types of kinematic constraints are as follows: 14 ball hinges, 10 rotating hinges, 12 constant speed hinges, 6 moving hinges, 3 cylindrical hinges, 2 hook hinges, and 1 gear rack hinge. The number and types of force elements are 44 rubber bushes, 4 suspension springs, 4 dampers, 2 torsion springs in the front and rear stabilizers, 4 tire force elements, and 4 braking forces.

3.2. Measurement of Critical Residual Preload of Loosening by Rotation. The procedures for measuring the critical residual preload of loosening by rotation are presented in Table 2. $F_{VR}^l$ is calculated as follows:

$$\tau = \sqrt{r_A^2 + r_Q^2 + 2r_Ar_Q\cos \gamma}$$

$$= \begin{cases} \sqrt{r_A^2 + r_Q^2 + 2r_Ar_Q\sin^2 \theta}, & \theta \in [0^\circ, 180^\circ], \\ \sqrt{r_A^2 + r_Q^2 - 2r_Ar_Q\sin^2 \theta}, & \theta \in (180^\circ, 360^\circ]. \end{cases}$$ (17)

$$f = \begin{cases} (\sigma_A - \sigma_Q) \cdot \mu_G, & \theta \in [0^\circ, 180^\circ], \\ (\sigma_A + \sigma_Q) \cdot \mu_G, & \theta \in (180^\circ, 360^\circ]. \end{cases}$$ (18)
the loosening by rotation of the two SJS sets are 30,740 N and 18,702 N, respectively. The test and numerical results are presented in Table 6. The comparisons revealed that the test result values are slightly lower than the numerical result values, and the relative errors are 5.56% and 6.22%, respectively. Based on an analysis of the loosening mechanism, in the numerical calculation, using the local slip of the thread surface as an antiloosening condition is a conservative...
approach that yields calculated values that are higher than the values obtained via testing. In engineering applications, a conservative design can ensure structural safety and reduce the likelihood of accidents.

4. Investigation of Parameters Influencing Antiloosening Performance

In the derivation described in the section “Mechanical Models,” it was found that the antiloosening performance of the SJs is closely related to the working loads and structural parameters. Based on the numerical calculation model, the influences of transverse loading, axial loading, bending moment, torque about the bolt axis, clamping eccentricity, loading eccentricity, and coefficient of friction in the thread and at the interface on the antiloosening performance were analyzed. Based on the working loads and structural parameters of the shock absorber-body SJS in Table 1, the influence of each parameter on the critical residual preload of loosening by rotation was successively investigated by

Figure 7: Vehicle road test.

Figure 8: (a) Vertical force spectrum $F_A$ and (b) lateral force spectrum $F_Q$ for the left rear wheel under braking condition.

Figure 9: Dynamic simulation model.
changing the value of one parameter within a certain range while keeping the other parameters unchanged, as shown in Figures 14–21. As the working loads and loading eccentricity increased, the critical residual preload of the loosening by rotation increased while the antiloosening performance decreased, as shown in Figures 14–18. As the friction coefficient at the interface and in the thread increased, the critical residual preloads for failure decreased and the

changing the value of one parameter within a certain range while keeping the other parameters unchanged, as shown in Figures 14–21. As the working loads and loading eccentricity increased, the critical residual preload of the loosening by rotation increased while the antiloosening performance decreased, as shown in Figures 14–18. As the friction coefficient at the interface and in the thread increased, the critical residual preloads for failure decreased and the
Figure 12: Calibration of bolt preload.

Figure 13: Preload measurement by the ultrasonic measuring system.

| Parameter name                              | Shock absorber-body SJS | Crossarm-subframe SJS |
|---------------------------------------------|--------------------------|-----------------------|
| Outside diameter of thread, \( d \) (mm)   | 10                       | 12                    |
| Pitch of the thread, \( P \) (mm)           | 1.25                     | 1.25                  |
| Lead angle, \( \beta \)  (°)               | 2.48                     | 2.04                  |
| Clamping length, \( l_k \) (mm)            | 30                       | 94                    |
| Elastic resilience of the bolt, \( \delta_k \) (mm·N\(^{-1}\)) | \(3.37 \times 10^{-6}\) | \(4.69 \times 10^{-6}\) |
| Elastic bending resilience of the bolt, \( \beta_k \) (mm·N\(^{-1}\)) | \(7.03 \times 10^{-7}\) | \(1.49 \times 10^{-6}\) |
| Coefficient of friction in the thread, \( \mu_c \) | 0.13                     | 0.13                  |
| Clamping eccentricity, \( s_{sym} \) (mm)  | -4                       | 0                     |
| Elastic resilience of the clamped parts, \( \delta_p \) (mm·N\(^{-1}\)) | \(1.39 \times 10^{-6}\) | \(1.92 \times 10^{-6}\) |
| Elastic bending resilience of the clamped parts, \( \beta_p \) (mm·N\(^{-1}\)) | \(1.60 \times 10^{-9}\) | \(1.54 \times 10^{-8}\) |
| Friction radius at the clamped parts, \( r_a \) (mm) | 18.5                     | 13.6                  |
| Coefficient of friction at the interface, \( \mu_T \) | 0.16                     | 0.15                  |
Table 3: Continued.

| Parameter name                              | Parameter value |
|----------------------------------------------|-----------------|
| Number of force-transmitting interfaces, \( q_F \) | 1               |
| Number of torque-transmitting interfaces, \( q_M \) | 1               |
| Transverse load, \( F_Q \) (N)               | 5121            |
| Axial load, \( F_A \) (N)                    | 5704            |
| Bending moment, \( M_B \) (N-mm)             | 33116           |
| Torque about the bolt axis, \( M_Y \) (N-mm)  | 465             |
| Loading eccentricity, \( a \) (mm)           | 17              |
| Load factor for concentric clamping and eccentric loading, \( \Phi_{en} \) | —               |
| Load factor for eccentric clamping and eccentric loading, \( \Phi^*_{en} \) | \( 7.85 \times 10^{-2} \) |
| Load factor for moment loading and concentric clamping, \( \Phi_m \) | —               |
| Load factor for moment loading and eccentric clamping, \( \Phi^*_{m} \) | \( 3.23 \times 10^{-3} \) |

Table 4: Calculation and assessment results for loosening by rotation.

| Calculation/assessment result          | Shock absorber-body SJS | Crossarm-subframe SJS |
|----------------------------------------|--------------------------|-----------------------|
| Assembly preload, \( F_M \) (N)       | 32500                    | 25300                 |
| Loss of preload, \( F_Z \) (N)        | 2311                     | 2268                  |
| Residual clamp load, \( F_{KR} \) (N) | 24911                    | 22782                 |
| Axial load, \( F_{AS} \) (N)          | 30463                    | 23034                 |
| Transverse load, \( F_{QS} \) (N)     | 1161                     | 76                    |
| Torsion about the bolt axis, \( M_{YS} \) (N-mm) | 21469                   | 1034                  |
| Bending moment, \( M_{Sb} \) (N-mm)   | 35037                    | 7862                  |
| max\( \tau - f \) (N-mm\(^{-2}\))     | 93                       | -75                   |
| Assessment results                     | Loosening                | Not loosening         |

Table 5: Measured data for residual preload and rotation angle of marker line.

| Name of SJSs            | Residual preload, \( F_{PW} \) (N) | Rotation angle of marker line, \( \phi \) (°) |
|-------------------------|-------------------------------------|-----------------------------------------------|
| Shock absorber-body SJS | 26800                               | 5.4                                           |
| Crossarm-subframe SJS   | 15500                               | 6.1                                           |

Table 6: Test and numerical results for critical residual preload for loosening by rotation.

| Name of SJSs            | Test result (N) | Numerical result (N) | Relative error (%) |
|-------------------------|-----------------|----------------------|--------------------|
| Shock absorber-body SJS | 30740           | 32450                | 5.56               |
| Crossarm-subframe SJS   | 18702           | 19865                | 6.22               |

Figure 14: Relationship between transverse load and critical residual preload.
antiloosening performance improved, as shown in Figures 19 and 20. The influence of the clamping eccentricity on the critical residual preload of loosening by rotation depends on the distance $|a - s_{sym}|$ between the action line of the axial load and the bolt axis (Figure 5). As the clamping eccentricity increased, the distance $|a - s_{sym}|$ decreased, the critical
residual preload of loosening by rotation decreased, and the antiloosening performance improved, as shown in Figure 21. Notably, the antiloosening performance is sensitive to transverse loading, axial loading, torque about the bolt axis, and coefficient of friction in the thread and at the interface. Therefore, attention should be paid to the influences of these factors in the SJS design and failure prediction.

These influences of the preload, transverse load, and coefficient of friction in the thread on the antiloosening performance are consistent with those reported in the literature [8, 24]. However, previous studies did not consider the effects of axial loading, bending moment, torque about the bolt axis, clamping eccentricity, loading eccentricity, and coefficient of friction at the interface. Therefore, existing numerical models [8, 24] are not suitable for calculating the critical residual preloads of the SJSs considered in this study and used in engineering practice.

In summary, the antiloosening performance of SJSs is influenced by the working loads and structural parameters. Hence, by reasonably controlling the parameter values, the antiloosening performance can be improved and loosening failure can be avoided.

5. Conclusions

A numerical calculation model of antiloosening performance is proposed to obtain the exact complex working loads of SJSs. The accuracy and reliability of the proposed model were verified by testing. From the results obtained by this study, the following conclusions can be drawn:

(1) Load data were obtained through load calibration testing and load spectrum measurement tests under actual working conditions, and the dynamic simulation model was calibrated to ensure the satisfactory accuracy of the loads used as the data for numerically calculating and measuring the critical preload required for failure.

(2) By considering the complex working loads, slip state of the interface, stress state of the thread surface, influences of eccentric clamping and eccentric loading on the load, and the elastic resilience and load coefficient, mechanical SJS models were established, and the numerical calculation models of antiloosening performance were obtained. The proposed calculation method is simple and effective for use in engineering applications and can obtain the quantitative relationship and interaction amongst the structural parameters, load under complex working conditions, and antiloosening performance.

(3) Because it is difficult to validate the numerical calculation model’s accuracy, the scheme for measuring the critical residual preload of loosening by rotation under actual working conditions was constructed, and the accuracy of the numerical calculation model was more efficiently verified in terms of time and cost. Because the maximum error between the test results and numerical results is 6.22% and the numerical results are conservative, the reliability and safety of the structure are ensured, and the accident risk is reduced.

(4) The numerical analysis results revealed that the transverse load, axial load, torque about the bolt axis, friction coefficient of the interface, and coefficient of friction in the thread greatly influence the antiloosening performance. The working loads, loading eccentricity, and distance between the action line of the axial load and bolt axis are negatively correlated with the antiloosening performance. The coefficients of friction in the thread and at the interface are positively correlated with the antiloosening performance. These results can be useful as guidelines in engineering practice.

As can be seen, the factors considered in the numerical calculation model are more comprehensive and suitable to practical engineering scenarios. The results obtained by this study expand the application scope of existing numerical calculation models of antiloosening performance and can accurately predict the antiloosening performance and loosening failure of SJSs under complex working conditions loading. In future work, to further improve the prediction accuracy of antiloosening performance, the dynamic contact state and micromechanical behavior of the interface, thread surface, and bearing surface should be investigated by experiment or using the finite element method.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.
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