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REGULAR PAPER

Influence of clearance and structural coupling parameters on shimmy stability of landing gear

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Abstract
There are many factors causing the shimmy of the aircraft landing gear and structural clearance of landing gear is a typical factor. Some aircraft in service or operation did not shimmy before, but suddenly appeared after a period of use. To solve the problem of clearance shimmy during the service of a certain aircraft, we established a dynamic model of rotating gear with clearance based on the flexible multi-body dynamics model of landing gear and L-N contact theory. We defined different types of clearance and established a mechanical model of aircraft pendulum vibration considering the clearance of landing gear structure for dynamic simulation, and studied the effects of different clearance types, clearance size of motion pair and different clearance positions on the stability of pendulum. The results show that the axial clearance has little effect on the shimmy performance of landing gear; the radial clearance has a certain effect on the shimmy performance of medium speed running, which slightly improves the shimmy damping required by medium speed running; the rotational clearance affects the shimmy performance of the nose landing gear by affecting the force transmission of structural components. The required shimmy damping coefficient increases at low speed and decreases at high speed. The main reason for the return clearance is that during the return, the shimmy damper does not work, which leads to the decrease of the shimmy damping performance and the increase of the required shimmy damping coefficient in the whole speed range. Meanwhile, the structural clearance will increase the shimmy frequency of the nose landing gear. By analysing the yaw angle of the nose landing gear and the time domain curve of the yaw angle of the yaw damper, we can determine which structure of the landing gear and which type of clearance is the cause of the yaw. Finally, the coupling effect caused by the main structural parameters of the landing gear in "gap shimmy" was analysed according to different mechanical stability distances and strut stiffness of the nose landing gear, providing reference for aircraft anti-shimmy design, nose landing gear fault diagnosis and nose landing gear maintenance support.

Nomenclature

D_c	hole diameter
d_c	shaft diameter
θ_s	the torsion angle of the front wheel around the strut axis
$\bar{\theta}_1$	the actual angular displacement of the damper
θ_1	equivalent angular displacement
K_{Te}	equivalent linear stiffness coefficient
C_{ce}	equivalent coulomb friction damping coefficient
C_{De}^{L}	equivalent linear shimmy damping coefficient
δ	insert depth
Kst	the contact stiffness coefficient of the collider
$C(\delta)$	damping coefficient

$\dot{\delta}$	seepage velocity
E	Young's modulus
R	radius of curvature
v	Poisson's ratio
Ce	recovery coefficient
$Veps$	conversion speed
θ	the angle between the contact surfaces
f_T	dynamic friction coefficient
v_τ	relative tangential velocity
v_τ	critical speed
e	mechanically stable distance

1.0 Introduction

Landing gear shimmy is the violent vibration of the nose landing gear wheel deviating from the centre position when the aircraft is taxiing on the ground. Shimmy is a self-excited vibration coupled with the movement of the wheels and landing gear with various degrees of freedom [1, 2]. For more than half a century, aircraft shimmy and landing gear dynamic instability have been a difficult problem [3].

Due to the various nonlinearities of real landing gear systems, the nonlinear dynamics of landing gear has aroused great research interest in academic circles [4]. Among them, the shimmy caused by structural clearance factors caused by long-term use wear, improper product processing or improper maintenance and repair is defined as "clearance type" shimmy, which has been one of the research priorities so far [5, 6].

As for the clearance shimmy, there are some theoretical methods such as piecewise linearisation, description function method and bifurcation theory, which can quickly analyse the influence of relevant parameters on the shimmy. Among them, the mathematical extension based on bifurcation theory can better analyse the clearance shimmy of the strongly nonlinear landing gear. When analysing the influence of nonlinear terms on the stable zone of shimmy, Wang X used the description function method to give a complete nonlinear equation for describing the 5-dof shimmy system [7]. Literature [8] considers nonlinear problems caused by torsional-free clearance. Bifurcation theory utilises a simple method based on harmonic balance (HB), which can capture unstable and stable LCO branches [9]. Rahmani M found that the presence of free gaps led to the emergence of a new region in the stability diagram, characterised by low amplitude rotational oscillation and zero steady-state transverse amplitude [10]. There are also studies on geometric nonlinearity caused by free clearances based on perturbation analysis technology [11] and incremental harmonic balance method [12]. More examples of bifurcation analysis applied to nonlinear systems can be found in references [13–15] and references [16–18]; pay special attention to aircraft shimmy analysis.

The research of dynamics module is also progressing with the maturity of computer technology. WR Krüger summarised the latest progress of numerical simulation of landing gear dynamics and how to deal with vibration in landing gear through numerical analysis [19]. At present, the research on the gap-type shimmy of landing gear in the service process of a certain aircraft mainly used the results of the dynamic test of landing gear to verify the model basing on the flexible multi-body dynamics of landing gear [20]. Dynamics simulation software was used to deeply study the impact of landing gear structural clearance on shimmy stability, define the clearance form of landing gear structure [21], summarise some aircraft landing gear structural clearance models and solution methods and provide reference for further research on the impact of clearance on shimmy stability of aircraft landing gear [22]. Through dynamic simulation analysis, it is found that clearance will have a negative effect on inhibiting the shimmy of landing gear by coupling with the strut bending and torsion of landing gear [23].

The above work is the simple processing of clearance analysis, and then the simple quantitative analysis of the continuous contact model, or the theoretical qualitative solution of its trend. There are few articles on the detailed study of the gap type shimmy. In view of this, based on different gap structural forms, the paper defined the clearance, studied the influence of different clearance types, clearance size

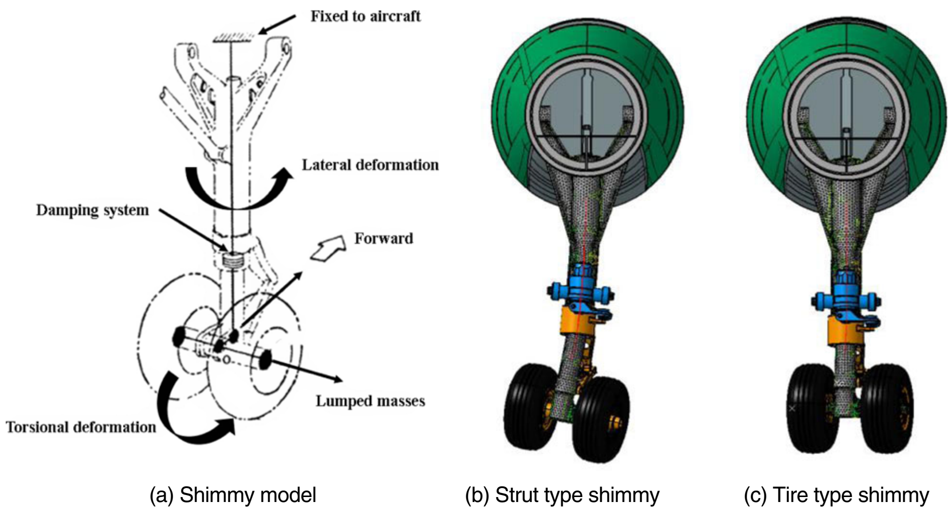

Figure 1. *Schematic diagram of landing gear shimmy.*

of moving pair and different clearance positions on the shimmy stability, analysed the time domain curve of the swing angle and considered the coupling effect caused by the main structural parameters of the landing gear and the structural clearance.

2.0 Dynamic principle and modelling of landing gear structural clearance

2.1 Overview of shimmy of landing gear with clearance

The main reason for shimmy is that the front wheel is excited by the external ground, resulting in violent lateral swing away from the neutral position, which is gradually transmitted upward through the wheel, piston rod and torsion arm, resulting in the shaking of landing gear strut and fuselage. Generally, the front landing gear shimmy includes structural shimmy and tire shimmy. They are vibration in the side-bending direction and vibration in the torsion direction respectively. Although they often occur in the case of coupling, one of them will appear as the main feature. Figure 1 is a typical phenomenon in the front landing gear shimmy.

There must be a gap between rigid parts with relative motion. The influence of the gap between parts on the shimmy stability is gradually superimposed in the transmission process, thus affecting the shimmy stability. After taking off and landing for many times, due to the inevitable wear and extrusion deformation in use, the gap value exceeds the critical gap value of shimmy, resulting in shimmy accidents. This is called "gap type shimmy". The gap is an unavoidable error in processing and installation, and it will be produced after long-term work in the workshop. Therefore, "gap-type shimmy" is very necessary.

Dynamics model of mechanism with clearance according to the analysis method can be divided into two categories. The first category is the massless rod equivalent method, also called "continuous contact" method, which will be equivalent to a clearance without quality fixed rigid rod, and ignore the elastic deformation of the contact surface, so that the original institutions into multiple pole zero clearance degree of freedom system. Although this method is simple, it ignores the structural elastic deformation of the contact surface, and can not accurately reflect the contact and collision characteristics of the mechanism with structural clearance. The second kind of force description method includes two kinds of typical models, "contact – separation" two-state model and "separation – collision – contact" multi-state model. Although this method is complicated, it can truly reflect the states of contact, separation,

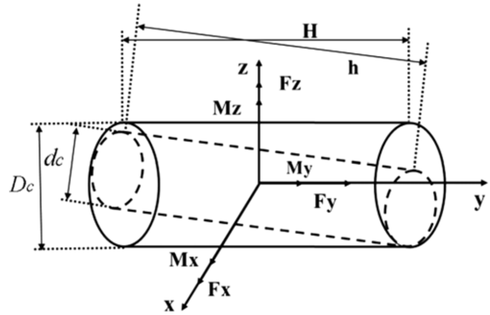

Figure 2. *Stress diagram of rotating pair with clearance.*

collision and transition between two components with clearance, so this paper adopts force description method to analyse the dynamics of structural clearance.

In this paper, most of the swing model motion pairs of the nose landing gear discussed are in the form of the rotating pair of the axle pin and the axle sleeve. Due to the existence of clearance, when the axle pin is subjected to different forces and torques, its position and pose in the axle sleeve are different. As shown in Fig. 2.

The rotating pair is a typical plane pair, which has one degree of freedom. However, after the gap exists, the two components can move relative to each other within a certain range on the other five degrees of freedom. Considering the radial and axial clearance between shaft and shaft sleeve, based on the mechanical properties of the motion pair analysis, the component between the contact pattern and its existence conditions are given, give four kinds of clearance. The first is the radial clearance, the second is axial clearance, the third one is rotating in the radial clearance of shaft, and the last one is in the process of the axial rotation axis rotation return clearance.

Considering the installation requirements between the landing gear and the aircraft and the requirements of oil seal between the strut and the piston rod, even if there is a gap in some places, it is relatively small. According to the analysis of force transmission path and gap equivalent transmission of the front landing gear, the influence of structural gap on shimmy at these positions is ignored. Therefore, the clearance between the front lug and the outer cylinder, the clearance between the rear lug and the strut, the clearance between the strut and the outer cylinder, the clearance between the outer cylinder and the sleeve, the clearance between the sleeve and the upper torsion arm, the clearance between the upper torsion arm and the lower torsion arm and the piston rod are only considered below. There are altogether seven structural gaps (Fig. 3). The radial clearance①, axial clearance②, radial rotation clearance③ and axial rotation return clearance of each structure④ are studied in Table 1.

2.2 Rotational kinematics model with structural clearance

2.2.1 Radial clearance

In the ideal state, the rotating pairs between the nose landing gear constrain the translational motion in three directions, but in the actual state, the translational motion constraint in three directions is invalid due to the existence of clearance. The radial two degrees of freedom are shown in Fig. 4.

The clearance vector model is expressed by introducing a clearance vector into a plane rotating hinge. In this model, the clearance vector represents the exact relative position of the two adjacent components connected by the rotating hinge, which can effectively deal with the change of the relative position of the clearance moving pair. The clearance vector takes the rotation centre of the shaft sleeve as the reference starting point and points to the potential contact point when the shaft and the shaft sleeve move relative to each other, which constitutes the relative collision point of the shaft and the shaft sleeve. The size of the clearance vector is strictly limited in the clearance circle with the rotation centre of the shaft sleeve

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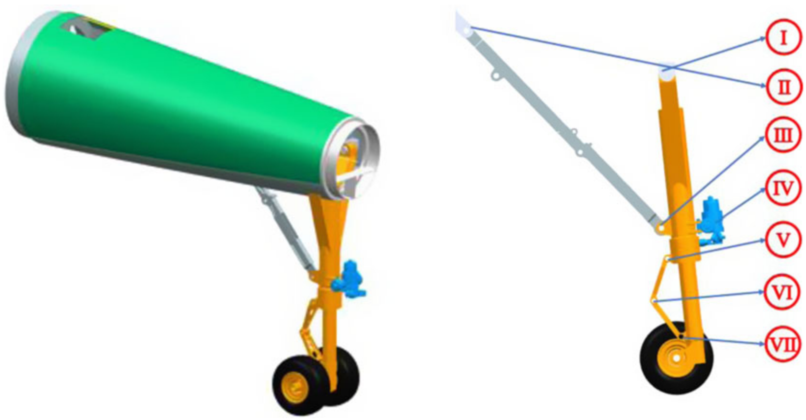

Figure 3. *Nose landing gear structure and clearance position diagram.*

as the centre and the radial tolerance of the shaft and the shaft sleeve as the radius, so the change of the clearance vector size can reflect the machining error of the component (Fig. 5).

For such a gap, D_c is hole diameter, d_c is shaft diameter. The gap size is described by the difference between the hole diameter and the shaft diameter, and the gap is

$$\Delta = D_c - d_c \tag{1}$$

2.2.2 Axial clearance

In addition to two degrees of freedom in the radial direction, there is also one degree of freedom clearance in the axial direction. The schematic diagram is shown in Fig. 6.

For the reinforcement of the holes and shafts of the rotating pair, it is necessary to tighten them left and right. If the installation is too compact, the rotary damping of the rotating pair will be too large to rotate. Too loose, and will lead to the existence of left and right clearance, so need to be considered.

$$\Delta = \Delta_1 + \Delta_2 \tag{2}$$

2.2.3 Radial rotation around the axis clearance

In addition to the free clearance in the translational direction, there is also a clearance of the third degree of freedom; that is, the clearance of radial rotation around the axis. For the simulation of the rotation pair with clearance, the axis is discretised into several spheres to simulate the pore-axis coordination. This allows the gap to be simulated. As shown in Fig. 7, the surface represents the hole wall structure and the size is D_c; the multi-sphere represents the shaft structure, and the size is d_c; the gap is $\Delta_c = (D_c - d_c)\big/2$; The rotation angle is $2\Delta_c/\mathrm{L}$.

2.2.4 Axial rotation about axial return clearance

Gear or connecting rod mechanism can be regarded as a rotating pair, if the force or torque transmission, when the steering suddenly reversed, if there is a certain return clearance, there will be a short period of failure phenomenon. Especially when there is high frequency and small angle rotation between the outer cylinder and the sleeve of the landing gear, the force transmission between the dampers connecting the two is mostly a connecting rod mechanism or a gear mechanism, and the existence of clearance has a great influence on the shimmy (Fig. 8).

The existence of clearance will affect the structure of landing gear and its force transmission properties. Through the motion characteristics, the nonlinear equations can be quasi-linearised by the

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Table 1. Clearance position of landing gear structure

Code name	Interstitial site	Clearance type
I	Front lug and outer tube	①②③
II	Rear lug and brace	①②③
III	Brace and outer cylinder	①②③
IV	Outer tube and sleeve	④
V	Sleeve and upper torque arm	①②③
VI	Upper and lower torsion arms	①②③
VII	Lower torsion arm and piston rod	①②③

Figure 4. *Schematic diagram of the radial clearance.*

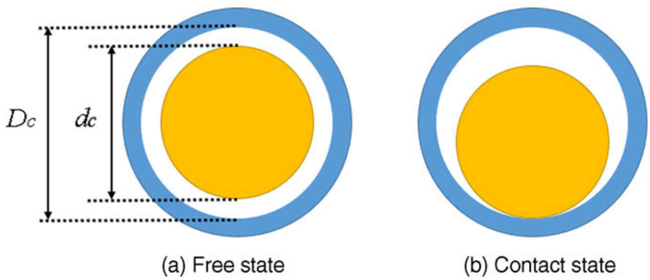

Figure 5. *Radial clearance contact model.*

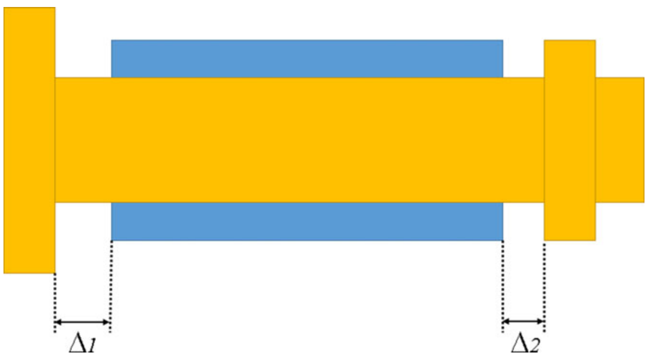

Figure 6. *Schematic diagram of the axial clearance.*

description function method, and the structural force transmission in the non-existent stiffness, Coulomb friction, damping and non-clearance interval can be equivalent to the value without clearance in the whole process.

Quasi-linearisation of a nonlinear term is a method of approximating the nonlinear term by a corresponding linear term, which depends on the input properties of the original nonlinear term, i.e. as a function of amplitude and frequency. Description function method is a convenient quasi-linearisation

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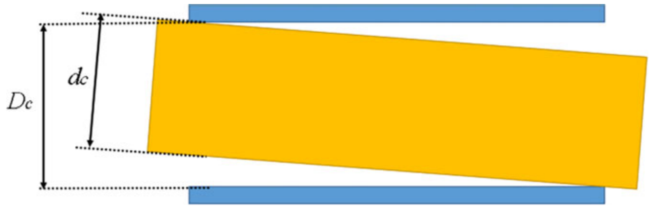

Figure 7. *Schematic diagram of radial-axis rotation clearance.*

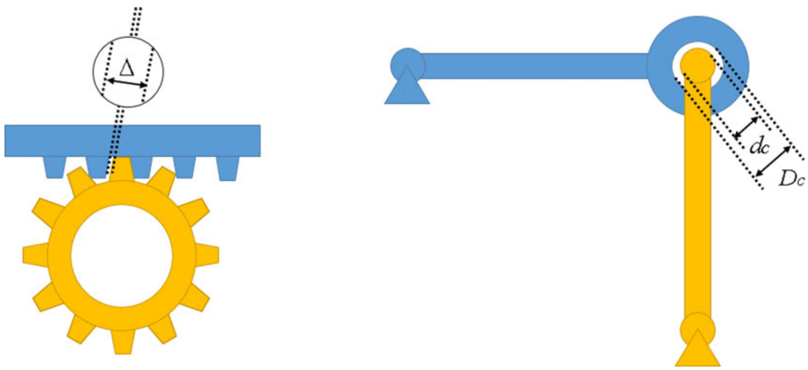

Figure 8. *Schematic diagram of the transmission clearance.*

method for the analysis of self-excited vibration. In this method, a linear function is used to replace the original nonlinear term and the error between them is minimised. The input signal is generally sinusoidal signal or Gaussian random signal, because it is only limited to the study of self-excited vibration of the front wheel, so the input signal is $x = A\sin(\Omega t)$.

For a given non-linear term $F_N(x, x)$, A sinusoidal input signal is $x = A\sin(\Omega t)$, The nonlinear term can be quasi-linearised by using the descriptive function method:

$$F_L = N_1 \bullet x + \frac{N_2}{\Omega} \bullet \dot{x} \tag{3}$$

Where:

$$N_1 = \frac{1}{\pi A}\int_0^{2\pi} F_N(A\sin\Omega t, A\Omega\cos\Omega t)\sin\Omega t d\Omega t \tag{4}$$

$$N_2 = \frac{1}{\pi A}\int_0^{2\pi} F_N(A\sin\Omega t, A\Omega\cos\Omega t)\cos\Omega t d\Omega t \tag{5}$$

Although there exists a nonlinear term in the system, using the describing function method, its steady solution under certain condition is still close to harmonic. According to the front wheel shimmy test, we can know that the front wheel shimmy is basically simple harmonic motion, using the describing function method, the three nonlinear term on the pillar structure can be linearisation, The corresponding equivalent stiffness coefficient, equivalent Coulomb friction and equivalent linear.

Torsional stiffness with clearance. Set the torsion angle of the front wheel around the pillar axis $\theta_s = A_{\theta_s}\sin\Omega t$. Due to the influence of the clearance, the elastic recovery moment generated by the front

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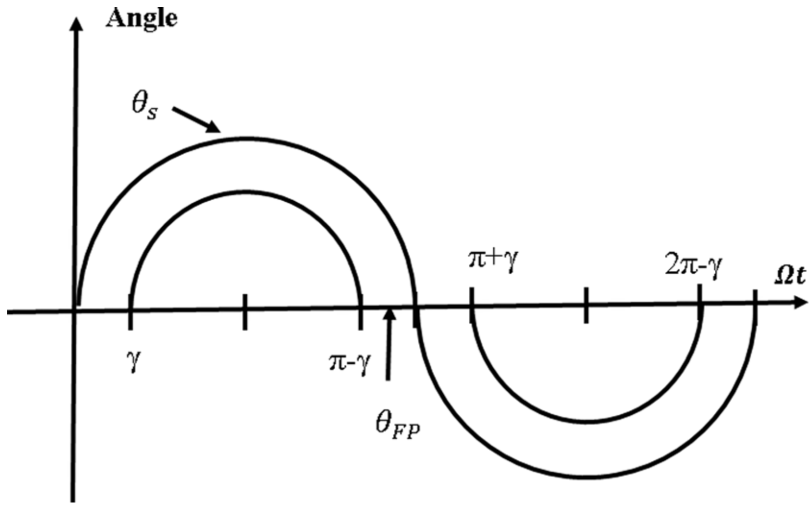

Figure 9. *Torsional stiffness with clearance.*

wheel swing angle θ_s can be expressed as:

$$M_T(\theta_s) = \begin{cases} 0 & (k\pi - \gamma \le \Omega t \le k\pi + \gamma) \\ K_T(\theta_s - \theta_{FP}) & (k\pi + \gamma \le \Omega t \le (k+1)\pi - \gamma) \\ K_T(\theta_s + \theta_{FP}) & ((k+1)\pi + \gamma \le \Omega t \le 2k\pi - \gamma) \end{cases} \tag{6}$$

Where, k=1,2, ..., γ is the phase angle due to clearance effect:

$$\gamma = \arcsin\left(\frac{\theta_{FP}}{A_{\theta_s}}\right) \tag{7}$$

In the formula A_{θ_s} is the amplitude of front wheel swing angle. It can be known from the descriptive function method:

$$\begin{aligned} N_1 &= \frac{1}{\pi A}\int_0^{2\pi} M_T(\theta_s)\sin\Omega t d\Omega t \\ &= K_T\left\{1 - \frac{2}{\pi}\left[\arcsin\left(\frac{\theta_{FP}}{A_{\theta_s}}\right) + \frac{\theta_{FP}}{A_{\theta_s}}\sqrt{1-\left(\frac{\theta_{FP}}{A_{\theta_s}}\right)^2}\right]\right\} \end{aligned} \tag{9}$$

$$N_2 = 0 \tag{10}$$

Thus the equivalent linear stiffness coefficient can be expressed as:

$$K_{Te} = K_T\left\{1 - \frac{2}{\pi}\left[\arcsin\left(\frac{\theta_{FP}}{A_{\theta_s}}\right) + \frac{\theta_{FP}}{A_{\theta_s}}\sqrt{1-\left(\frac{\theta_{FP}}{A_{\theta_s}}\right)^2}\right]\right\} \tag{11}$$

Coulomb friction with clearance. As can be seen from Fig. 9, Coulomb friction torque generated between pillars and tires and the ground can be expressed as:

$$M_c(\dot{\theta}_s) = (T_{CF} + G_{CF})\mathrm{sgn}(\dot{\theta}_s) \tag{12}$$

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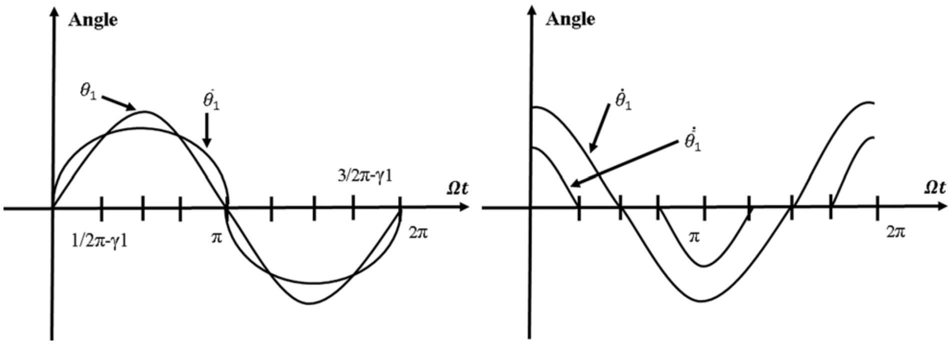

Figure 10. *Damping with clearance.*

The equivalent Coulomb friction damping coefficient C_{ce} can be obtained by the descriptive function method.

$$C_{ce} = \frac{4}{\pi A_{\theta_s} \Omega}(T_{CF} + G_{CF}) \tag{13}$$

Linear term damping with clearance. As shown in the Fig. 10, $\bar{\theta}_1$ is the actual angular displacement of the pendulum damper and θ_1 is the equivalent angular displacement.

$$\theta_1 = A_{\theta_1} \sin(\Omega t + \beta) \tag{14}$$

Where β is the phase difference between θ_s and θ_1.

Thus the damping moment of the linear term with clearance can be expressed as:

$$M_{DL}(\dot{\theta}_1) = \begin{cases} 0 & \left(\frac{1}{2}k'\pi - \gamma_1 - \beta \le \Omega t \le \frac{1}{2}k'\pi + \gamma_1 - \beta\right) \\ C_t\dot{\theta}_1 & \left(\frac{1}{2}k\pi + \gamma_1 - \beta \le \Omega t \le \frac{1}{2}(k+2)\pi - \gamma_1 - \beta\right) \end{cases} \tag{15}$$

Where $k' = 1, 3, 5, \ldots\ldots, k = 1, 2, 3, \ldots\ldots, \gamma_1$ is the phase angle caused by the clearance:

$$\gamma_1 = \arcsin\left(\frac{\theta_{FP}}{A_{\theta_1}}\right) \tag{16}$$

θ_{FP} of the equation (16) is the clearance, A_{θ_1} is the amplitude of the angular displacement θ_1 of the pendulum damper.

The equivalent linear pendulum damping coefficient C_{De}^L can be obtained by using the descriptive function method.

$$C_{De}^L = C_t \bullet \left[1 - \frac{2}{\pi}\arcsin\left(\frac{\theta_{FP}}{A_{\theta_1}}\right) + \frac{2}{\pi}\frac{\theta_{FP}}{A_{\theta_1}}\sqrt{1 - \left(\frac{\theta_{FP}}{A_{\theta_1}}\right)^2}\right] \tag{17}$$

2.3 Mechanical model of contact impact of structures with clearance

The collision theory is based on Hert contact theory, which has developed from rigid contact to the present finite element theory of flexible contact. The simulation results become more accurate, but at the same time the amount of calculation, the complexity is also rapidly increasing, so there are many theoretical model based on Hertz contact theory, can be used to don't care about the contact component in the study of the overall deformation, not only to ensure certain accuracy. Model complexity can also be greatly reduced.

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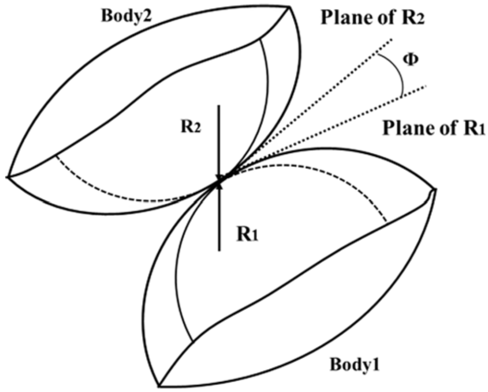

Figure 11. *A contact model of two surfaces.*

The contact force between elements in a moving pair is composed of two parts: the force perpendicular to the contact plane (contact collision force) and the force within the contact plane (friction force).

The accuracy and computability of the contact collision force model are very important for the dynamic simulation of the pore-shaft fit with structural clearance. The material properties of the contact surface, the mechanical shape, the convergence of the mathematical model and other factors should be considered. The most widely used contact force is The Hertzian model as shown in Fig. 11. This model simplifies contact collision into a spring damping system, but the stiffness term is a nonlinear function embedded deeply in the contact object. Meanwhile, the damper is used to simulate energy loss in the contact process, and the normal contact force is expressed as follows:

$$\mathrm{F_n} = |\delta|^{1.5} K_{st}\mathrm{sgn}(\delta) + \mathrm{C}(\delta)\,\dot{\delta} \tag{18}$$

The contact stiffness coefficient K_{st} can be expressed by the following expression:

$$K_{st} = \frac{\sqrt{K_{st}}}{G_e \lambda^{1.5}}\left(1 - \frac{1 - C_e^2}{1 + C_e^2}\right)\tanh\frac{2.5\dot{\delta}}{V_{eps}} \tag{19}$$

$$\lambda = 0.75|1 - |\cos\theta|^{2.17657}|^{0.24586} \tag{20}$$

$$K_D = \frac{1.5}{\frac{1}{R_1} + \frac{1}{R_1{}'} + \frac{1}{R_2} + \frac{1}{R_2{}'}} \tag{21}$$

$$G_e = \frac{1 - v_1^2}{E_1} + \frac{1 - v_2^2}{E_2} \tag{22}$$

The damping coefficient $\mathrm{C}(\delta)$ can be expressed as follows:

$$\mathrm{C}(\delta) = \frac{3(1 - C_e^2)}{4}\frac{\dot{\delta}}{\dot{\delta}^{(-)}} \tag{23}$$

In fact, the Hertzian contact force ignores the friction of the contact surface, which cannot fully express the tangential contact characteristics of the contact surface, so the tangential friction of the contact surface is discussed. After much study by scientists, the dry friction of objects in contact with each other satisfies the following three basic laws: the direction of friction is opposite to the relative direction of motion of two objects in contact. The frictional force is proportional to the normal contact force. Friction is independent of the actual contact area. The modified Coulomb friction model is used to describe the tangential friction model of cylinder contact with clearance. The original Coulomb friction model believed that the friction force was proportional to the normal contact force, but the tangential

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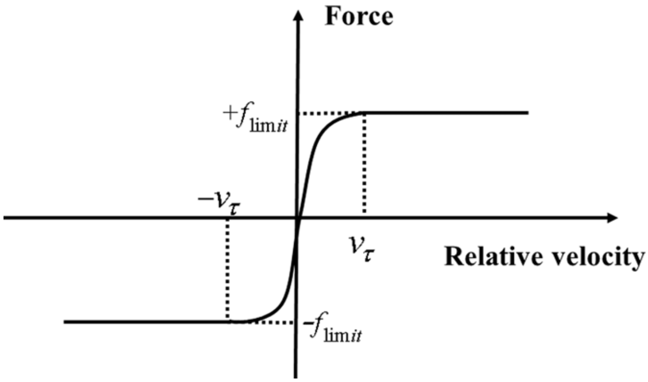

Figure 12. *The Threlfall friction force model.*

velocity was not considered, so it could not deal with the friction state switch of different tangential velocities. Therefore, the expression of the friction force considering the direction switch of tangential velocity was as follows:

$$f_{\mathrm{T}} = -\mu_f F_N \frac{v_\tau}{v_\tau} \tag{24}$$

The defect of the above expression is that it cannot handle the case of zero tangential velocity in the collision contact process. Therefore, many modified Coulomb friction models are proposed. The model used in this paper is Threlfall model, and its expression is as follows:

$$f_{\mathrm{T}} = -\mu_f F_N \frac{v_\tau}{v_\tau} \left(1 - \mathrm{e}^{-3|v_\tau|/v_\tau}\right) \tag{25}$$

The critical velocity of friction independent of tangential velocity. The model of the contact force is shown in Fig. 12.

2.4 Establishment of clearance simulation model of rotating pair structure

In the process of aircraft ground sliding, the existence of structural clearance will destroy the precise motion relationship between original structural components, resulting in complex changes in relative positions, as well as mechanical vibration, impact load and Coulomb friction. In dynamic analysis software, the contact force between parts is defined by the Contact tool. Contact used Restitution type, equivalent spring damping model, to calculate the impact force. It reduces the contact model to a spring damping system.

2.4.1 Realisation of radial clearance

In dynamics software, by adding more than one point of contact to simulate the aircraft landing gear structure in the gap of the two connection parts, as is shown in Fig. 13, under the above torque arm and the rotation of the space between the torque arm as an example, the torque arm along the rotating hinge arrangement several contact points, the radius of the contact point, numerical slightly smaller than the torque arm on the radius of the shaft sleeve. The difference between them is the size of the radial clearance (Fig. 13).

2.4.2 The realisation of axial clearance

The contact points at the left and right positions outside the axis centre of the lower torque arm were set to simulate the axial clearance by contacting the two virtual stretching planes of the upper torque arm,

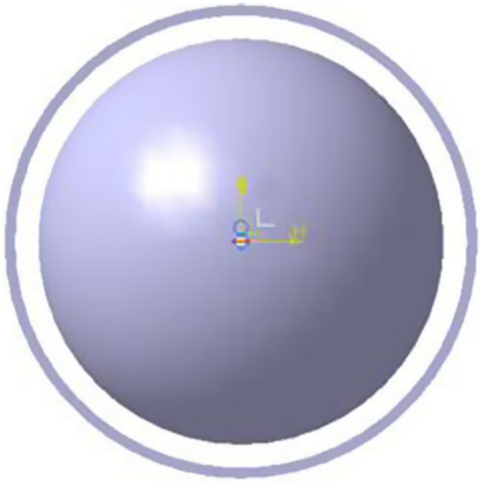

Figure 13. *Schematic diagram of the radial clearance.*

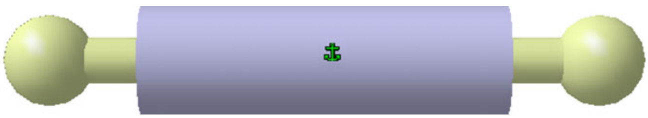

Figure 14. *Schematic diagram of the axial clearance.*

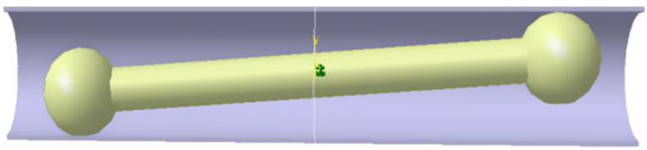

Figure 15. *Schematic diagram of clearance of radial rotation around axis.*

so that both of them would have a maximum displacement constraint on both sides of the axial. The schematic diagram of axial structural clearance model is shown in Fig. 14.

2.4.3 Realisation of radial rotation clearance around axis

In the clearance range of the radial structure, deflection occurs between the shaft of the lower torque arm and the sleeve of the upper torque arm, and the contact points arranged on the lower torque arm and the sleeve of the upper torque arm repeatedly contact and separate. When the two deviates from each other in the axial direction, the contact force between the contact point on the lower torsion arm and the upper and lower planes of the upper torsion arm will limit the displacement once the gap is beyond the range of the axial structure clearance (Fig. 15).

2.4.4 The realisation of the return clearance of axial rotation around axis

When the two parts rotate around the rotation axis, a club and a drawing surface are set in the middle of the two parts. When the rotation direction changes, there will be a return clearance, so as to simulate the shimmy damper and some inevitable clearance at the joint, as shown in Fig. 16.

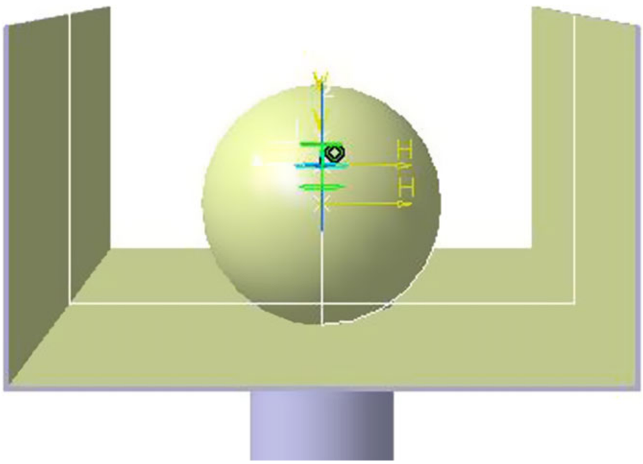

Figure 16. *Schematic diagram of return clearance.*

After the simulation model is established, different clearance conditions can be quickly adjusted by adjusting the contact point radius in the dynamics software to realise the simulation conditions under different sizes of return clearance.

2.5 Shimmy simulation model of flexible nose landing gear with clearance

In the analysis of the mechanical characteristics of the nose landing gear with clearance swing, the landing gear was simplified, some non-bearing components irrelevant to the analysis of the swing were deleted, and only the body lugs, buffer outer barrel, buffer piston rod, sleeve, upper torque arm, lower torque arm and wheel were retained (Fig. 17).

Figure 18 is the schematic diagram of landing gear structure installation and force transmission. For the shimmy stability analysis of flexible front landing gear, the accuracy of structural mode is a necessary condition to ensure the accuracy of the results. This time, the finite element modal analysis information of the dynamic model is calculated to ensure the comparative analysis of model modes under the condition of the same motion relationship.

Considering the overall modal stiffness of wheel and landing gear, the analysis is per Table 2.

Finite element software analyses the structure stiffness of the landing gear as a whole. It can be seen from Fig. 19 that there is bending torsion coupling in the structure of the landing gear. Due to the simplified model, the calculated result will be smaller than the real model. The dynamics software adopts the principle of linear superposition of various parts, and the modal orders of each part is limited, so the result will be slightly larger. The error is within 10% –20%, so the simulation of the model after flexible treatment can be considered to be accurate.

2.6 Verification of mechanical proxy model of pendulum with clearance

In order to enable the landing gear model to accurately simulate the subsequent complex working conditions, we must test the accuracy of the model, so as to lay a foundation for the subsequent simulation, in the absence of clearance simulation analysis was carried out on the landing gear shimmy before, first rigid dynamics modelling, and then to the landing gear flexible processing, starting from the tire touchdown is measured. In the theoretical research, the convergence of nose gear tire swing angle is regarded as stable; however, in the actual design requirements of shimmy reduction, the system needs sufficient stability margin, the damping provided by the system should be able to make the front wheel swing caused by external interference after three cycles, its amplitude attenuation to 1/4 or less than the initial

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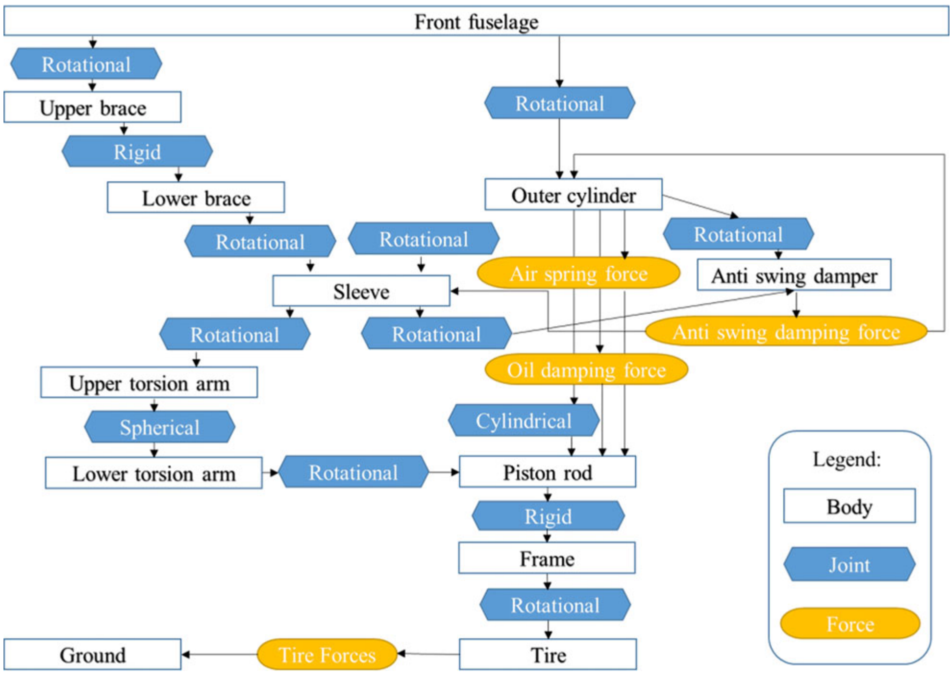

Figure 17. *Nose landing gear assembly connection block diagram.*

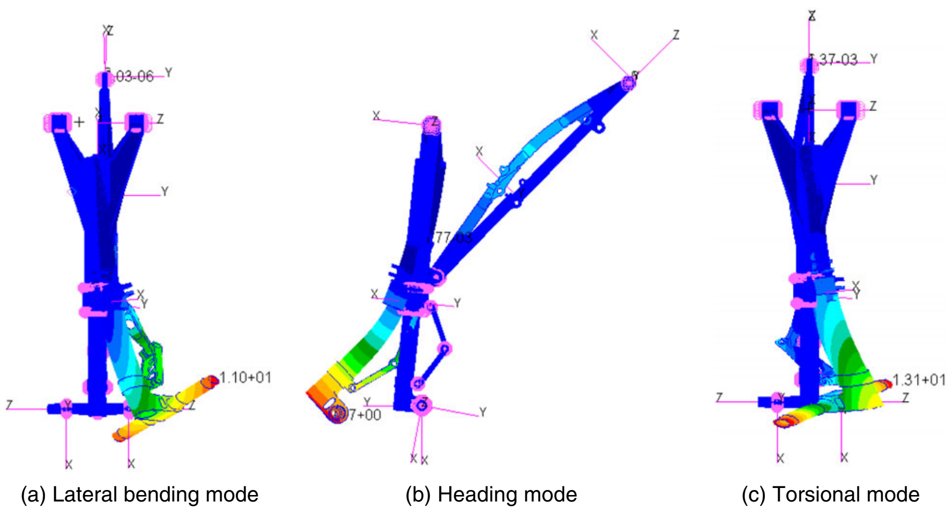

Figure 18. *Schematic diagram of landing gear side bending and torsion modes.*

disturbance. Under the condition that the flexible landing gear has no clearance, the sliding speed of 60m/s is set. After the sliding speed is stable for 2s, the maximum swing angle of the tire of the front landing gear is 3° after the front wheel is disturbed by the excitation, and the curve converges to 0.772° after three cycles.

At the same time, the rotating pair model mentioned above was established by bushing simulation, the clearance of the virtual model was also set as zero, and the same excitation force interference and the same shimmy damping coefficient were used to carry out the test. The curves can be obtained through simulation (Fig. 20).

Table 2. Modality and stiffness contrast

Operating condition	Finite element solution	Dynamic linear operation
The 1st-order lateral mode	52.4	53.8
Order 2 heading mode	73.2	75.1
The 3rd order torsion mode	78.6	80.4
Lateral bending stiffness (N/rad)	1.62e6	1.75e6
Torsional stiffness (N•m/rad)	4.3e4	4.4e4

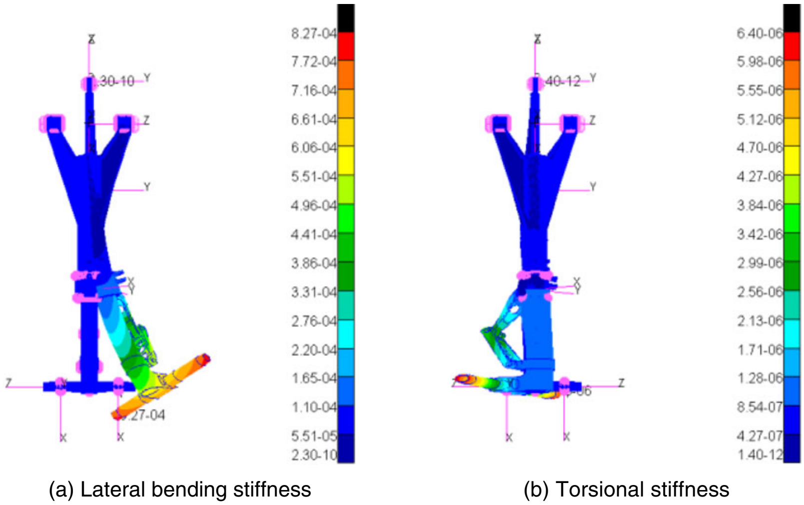

Figure 19. *Schematic diagram for calculation of lateral bending stiffness and torsional stiffness.*

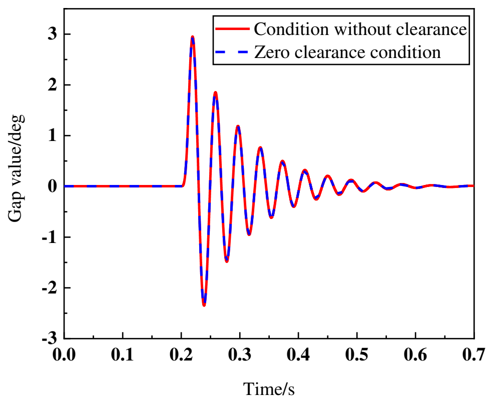

Figure 20. *No gap swing angle curve comparison.*

By comparison, it can be seen that the two curves are almost identical, and the error is within the permissible range. Therefore, the model is basically accurate and can be used for the following mechanical analysis of the pendulum with clearance.

Through the control variable method, the clearance values of different types are changed, and the accuracy of clearance is verified by measuring the clearance values (Fig. 21).

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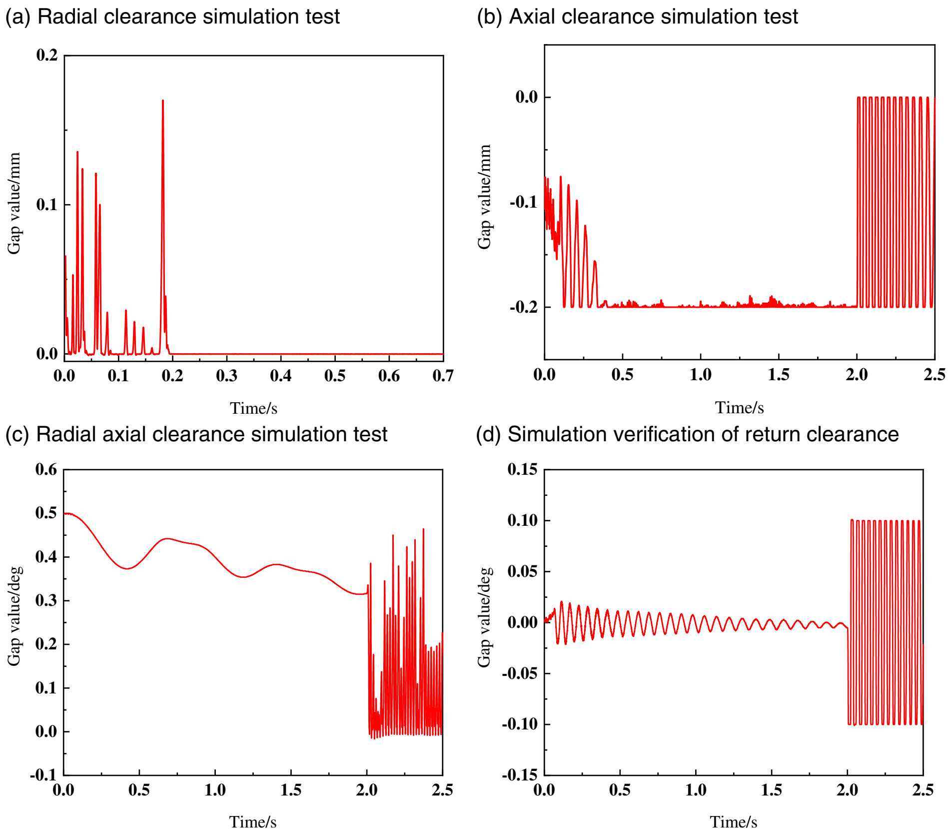

Figure 21. *Clearance simulation correction.*

3.0 Influence of structural clearance on shimmy stability

In the process of aircraft sliding, the clearance between structures will cause the loosening of the connection, resulting in internal collision and friction between structures. Structural clearance makes the movement of the actual mechanism deviate from that of the ideal mechanism, which reduces the kinematic precision of the mechanism, and easily causes the impact dynamic load, affects the load transmission of the system and causes the damage and failure of the motion pair. Different types of structural clearances on different structures have different influences on the shimmy of the nose landing gear. Therefore, the change of the shimmy angle curve of the nose landing gear under different types of clearances is analysed to provide a reference for the specific clearance types of the aircraft in the subsequent experimental stage.

3.1 Influence of clearance position on shimmy stability

Based on the precise aircraft landing gear shimmy model considering the landing gear structural clearance, the effect of landing gear structural clearance on the pendulum vibration mechanics was studied for different structural clearance conditions. Before considering specific to the clearance between the outer barrel and bolts, bolts and brace after clearance, poles, and the clearance between the outside tube, outer cylinder and the clearance between the sleeve, sleeve and torsion on the arm of the gap, the torsion torque arm under arms and clearance and the torque arm and the clearance between the piston rod, the influence of setting, different types of clearance on before landing gear angle of comparison.

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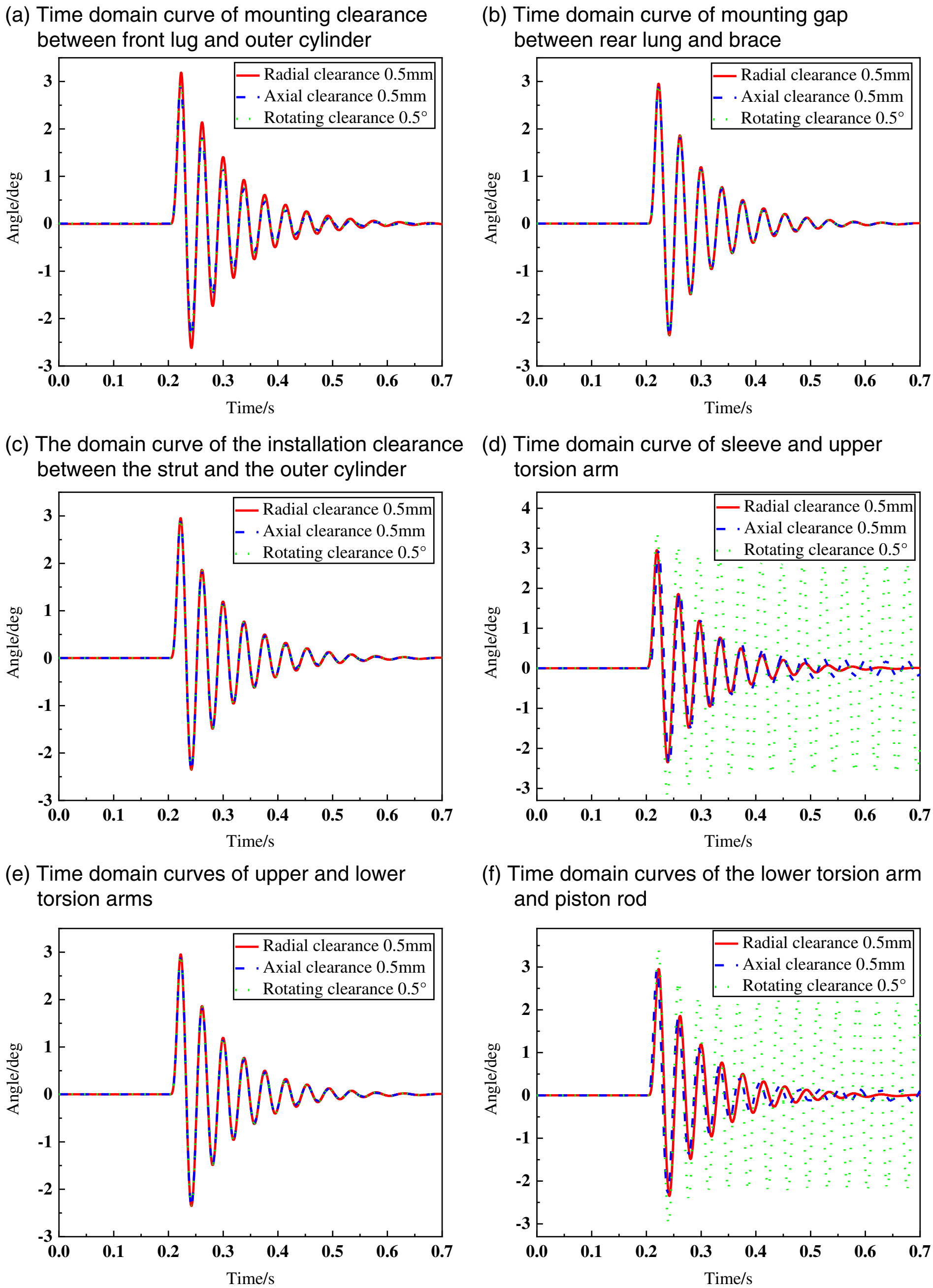

Figure 22. *Time domain curve comparison of front wheel swing angle.*

For the radial clearance, axial clearance and radial rotating clearance between each structure, 0.5mm, 0.5mm and 0.5° were set respectively to simulate and analyse the yaw performance of the nose landing gear with clearance during medium speed sliding. The simulation results are in Fig. 22.

Through comparative analysis of six different structure installation clearance, after which the bolts were installed with poles and poles and outer cylinder clearance gap has little effect on the landing gear shimmy performance, especially for the torque arm and the rotation of the torque arm under joint vice

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Table 3. *Clearance position of the landing gear structure*

Type	Position	Clearance size	Frequency (Hz)
Zero clearance	Nowhere	0	24.5
Radial clearance	Front lug and outer tube	0.5mm	24.6
Axial clearance	Sleeve and upper torque arm	0.5mm	25.9
Axial clearance	Lower torsion arm and piston rod	0.5mm	26.4
Radial rotation around the axis clearance	Sleeve and upper torque arm	0.5°	25.7
Radial rotation around the axis clearance	Lower torsion arm and piston rod	0.5°	26.9

clearance, pivoting radial clearance is not obvious effects on the landing gear shimmy before, so it is sometimes the ball deputy said is reasonable. Other installation gaps with obvious influences are shown in Table 3.

The most obvious effect of radial clearance on shimmy is the installation position of front lug and outer cylinder. When the radial clearance is 0.5mm, the shimmy convergence of the nose landing gear will be slow, and the shimmy frequency will increase slightly; the maximum shimmy angle of the front wheel will expand about 10% under the same excitation condition.

The axial clearance has a great influence on the installation positions of sleeve and upper torque arm as well as lower torque arm and piston rod, but the influence on the installation positions of sleeve and upper torque arm is slightly greater than that of lower torque arm and piston rod. When there is an axial clearance, the swing frequency of the landing gear increases from 24.5 to 25.9Hz and 26.4Hz. After rapid convergence, the pendulum angle will swing at a small constant sliding angle.

The radial rotation clearance also has a greater impact on the installation positions of the sleeve and upper torque arm as well as the lower torque arm and piston rod, and the impact on the installation positions of the sleeve and upper torque arm is slightly greater than that of the lower torque arm and piston rod. The pendulum frequency of the landing gear increases from 24.5 to 25.7Hz and 26.9Hz.

In terms of frequency, the existence of the above three clearances increases the pendulum frequency. For the installation position of sleeve and upper torsion arm and the installation position of lower torsion arm and piston rod, the pendulum frequency of axial clearance and radial rotation clearance are 25.9, 25.7 and 26.4, 26.9Hz respectively. The overall gap between the upper and lower torsion arms and the installation position of the piston rod will increase the swing frequency more. The frequency of the axial gap between the sleeve and the installation position of the upper torsion arm is 25.9Hz higher than that of the radial rotation gap 25.7Hz. However, the frequency of the axial clearance at the installation position of the lower torque arm and piston rod is 26.4Hz, which is lower than that of the radial axial clearance at 26.9Hz.

At the final stable swing angle, the radial clearance does not appear small angle swing and gradually converges with the increase of running time. There is a constant swing angle in the axial clearance and the radial rotation clearance after the sliding stability. The swing angles of the clearance at the installation position of the sleeve and the upper torque arm are 0.204° and 2.61° respectively, and the swing angles of the clearance at the installation position of the lower torque arm and the piston rod are 0.145° and 2.181° respectively. The mounting position of the upper sleeve and the upper torsion arm will increase the tire swing angle after the swing slip stability with clearance.

The initial axial rotation return clearance of 0.1° was set. The simulation results are shown in Fig. 23, including the angle time curve and the swing damping moment time curve.

When the rotary pair at the connection between the outer cylinder and the sleeve has a return clearance, the damping damper will fail in the return clearance. As can be seen from the curve of damping force value generated by the damping damper, when the swing angle is less than the clearance value, it is equivalent to the case of no damping and no damping force is provided.

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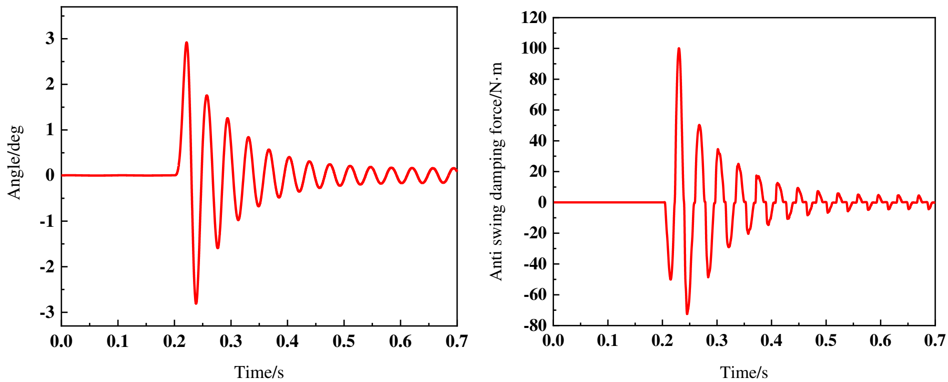

Figure 23. The time domain curve of the axial rotation of the return clearance and the drag curve generated by the shimmy damper.

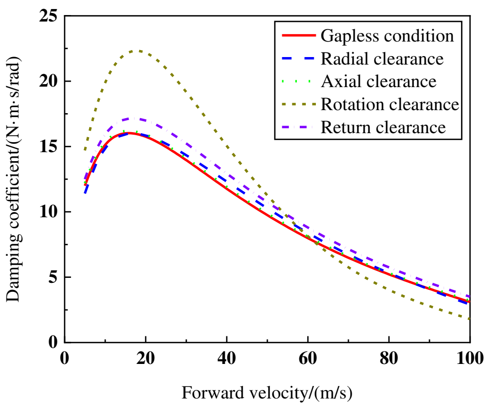

Figure 24. Pendulum damping coefficient and velocity with different clearances.

3.2 The relationship between clearance form and shimmy stability margin

According to the research and analysis in the above section, several types with obvious influences are selected: The radial clearance between 0.2mm front lug and outer cylinder, 0.2mm axial clearance between lower torsion arm and piston rod, 0.5° radial rotary clearance sleeve and upper torsion arm and 0.5° return clearance between outer cylinder and sleeve were analysed respectively to obtain the yaw damping coefficient – velocity curve as shown in Fig. 24.

It can be seen from the Fig. 24 that the axial clearance has little effect on the nose landing gear, and the yaw damping is almost similar to that required in the case of no clearance at all speeds. At 40m/s, the required damping increases by about 5% when the radial clearance increases from 11.7 to 12.4N·m·s. At low speed, the damping of the radial rotary clearance increases to 22.4N·m·s, which is 43.6% higher than the 15.6N·m·s required for no clearance. However, the damping coefficient of the radial rotary clearance decreases at high speed and decreases from 3.2 to 1.8N·m·s at 100m/s. About 44% less; The return clearance increases the damping coefficient at all speeds, with an average increase of 0.8N·m·s and a maximum increase of 1.4 N·m·s near 20m/s. The main reason is that the return clearance leads to the failure of the damping damper for a period of time.

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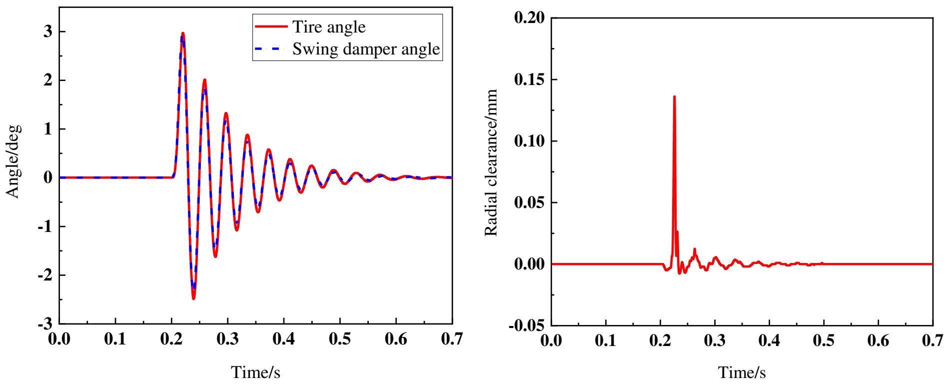

Figure 25. *Time domain curve and radial clearance size curve of 0.5mm radial clearance between front lug and outer cylinder of landing gear.*

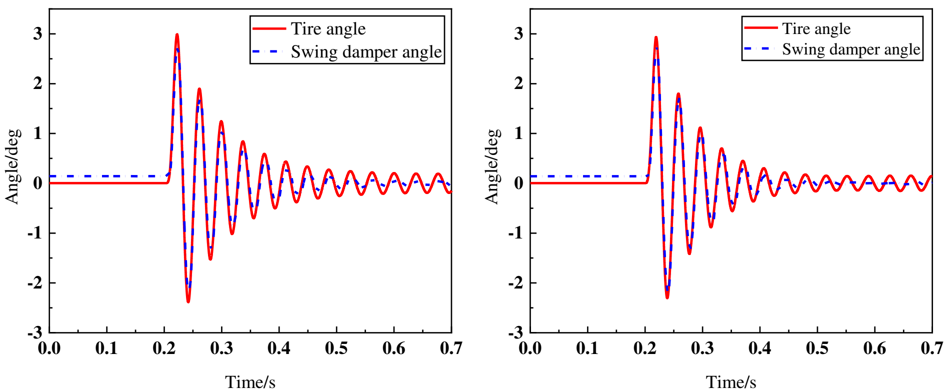

Figure 26. *Time domain curve of axial clearance between installation position of sleeve and upper torque arm and installation position of lower torque arm and piston rod 0.5mm.*

3.3 Research and analysis of swing angle of tire and shimmy damper

The existence of the gap is a complex nonlinear problems, mainly reflected in its different position and the differences of degrees of freedom, in order to study the different gap in different positions on the mechanism of the influence on shimmy and weight, by studying the shimmy with clearance, the former of the gear wheel angular movement of the pendulum angle and damper (influence of shimmy damper performance) of the curve. From the above section, the installation positions and clearance forms which have obvious influence on the shimmy are analysed and studied one by one.

Figure 25 is the swing angle curve of landing gear wheel and damper motion of 0.5mm radial clearance between front lug and outer cylinder.

As can be seen from Fig. 25, the angle of damper is markedly lower than the tire rotation, because of the radial clearance for landing gear swinging and torque arm stiffness deformation reason, as the role of shimmy damper, shimmy angle became smaller, the side force of the ground is reduced, the tire suffered with radial clearance with the landing gear swinging also decrease, tire angular angle gradually close to the damper. Nose landing gear shimmy stabilises.

The installation positions of the sleeve and the upper torque arm with an axial clearance of 0.5mm and the installation positions of the lower torque arm and the piston rod were taken to analyse the tire angle and the swing damper angle. The simulation curves are as in in Fig. 26 and Table 4.

Table 4. *Swing angle under 0.5mm axial clearance*

Position	Maximum angle of tire (°)	Maximum angle of damper (°)	Tire stability angle (°)	The damper stabilises the angle (°)
Sleeve and upper torque arm	3	2.724	0.193	0
Lower torsion arm and piston rod	2.974	2.774	0.154	0

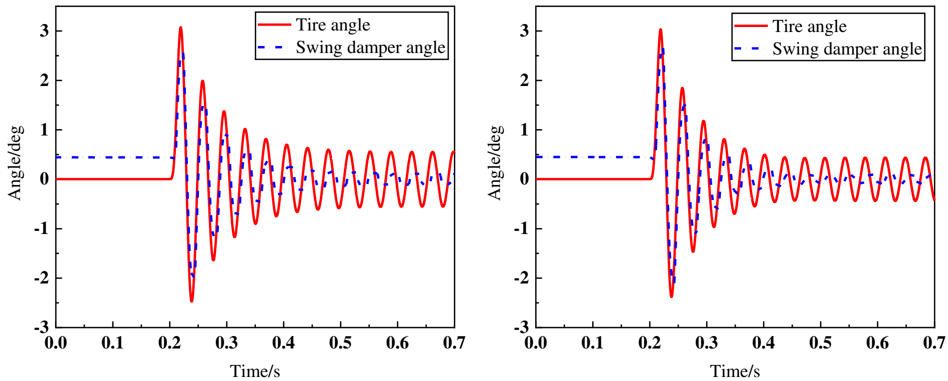

Figure 27. *Time domain curve of 0.5° radial rotary clearance between installation position of sleeve and upper torque arm and installation position of lower torque arm and piston rod.*

When the axial clearance is 0.5mm, after the installation position of sleeve and upper torque arm is excited, the tire angle of nose landing gear and the swing damper angle are 3° and 2.724° respectively, and the tire angle after stabilisation is 0.193°, and the damping angle of the swing damper decreases gradually. After the lower torque arm and piston rod are excited, the tire angle of the nose landing gear and the swing damper angle are 2.947° and 2.774° respectively, and the tire angle of the stabilised tire is 0.154°, and the swing damper angle gradually approaches zero. The angle difference caused by the installation position of the sleeve and the upper torque arm is 0.276°, while the installation position of the lower torque arm and the piston rod is 0.173°. Meanwhile, after stabilisation, the final swing angle of the former is 0.039° larger than the latter, indicating that in the transmission process, the clearance of the same size. The influence of axial clearance on the installation position of sleeve and upper torsion arm is worse than that of lower torsion arm and piston rod.

Similarly, the tire angle and swing damper angle were analysed when there was a 0.5° radial rotation gap. The simulation results of installation positions of sleeve and upper torque arm as well as lower torque arm and piston rod were shown in Fig. 27 and Table 5.

When there is a 0.5° radial rotation gap between the sleeve and the upper torque arm installation position, the maximum swing angle of the tire is 3.084°, the maximum swing angle of the damper is 2.566°, the difference of amplitude is 0.518°, and the time difference is 0.005s. The maximum swing angle of the tire and the maximum swing angle of the damper at the installation position of the lower torque arm and piston rod are 3.044° and 2.684° respectively, with a difference of 0.36° and a time difference of 0.004s. After the swing angle tends to be stable, the final tire swing angle and the swing angle of the damper are 0.555°, 0.123° and 0.438°, 0.097°. The results show that for the same size of clearance, the influence of radial rotation clearance on the installation position of sleeve and upper torsion arm is more severe than that of lower torsion arm and piston rod.

Figure 28 is a schematic diagram of swing angle curve of landing gear wheel and damper movement when the 0.5° return clearance between outer cylinder and sleeve is shown.

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Table 5. *Swing angle at 0.5°radial rotation clearance0.5°*

Position	Maximum angle of tire (°)	Maximum angle of damper (°)	Tire stability angle (°)	The damper stabilises the angle (°)
Sleeve and upper torque arm	3.084	2.566	0.555	0.123
Lower torsion arm and piston rod	3.044	2.684	0.438	0.097

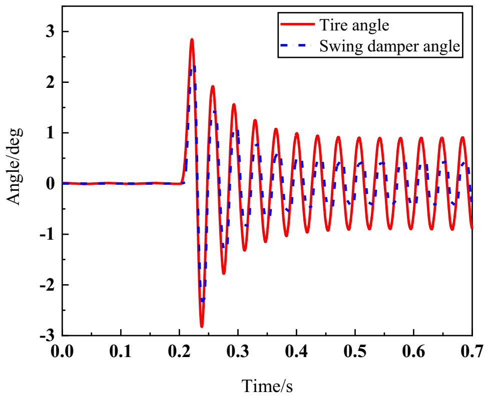

Figure 28. *5° return clearance between outer cylinder and sleeve.*

As shown in the Fig. 28, after the nose landing gear is excited, the maximum swing angles of tire and damper are 3.021° and 2.355° respectively. When the landing gear shimmy finally stably slides, the swing angles of tire and damper are 0.977° and 0.437° respectively. When the maximum angle of tire and damper is reached, the phase difference gradually changes from 0.004 to 0.008s. The pendulum damper has certain hysteresis when the mechanism returns. Through analysis, motivated and stability after sliding before running the landing gear shimmy damper rotating angle and the angular maximum pendulum angle is 0.666° and 0.504°. Angle difference value is greater than 0.5° clearance due to its torsional deformation and gap setting is similar to the spring damping structure, there will be a large deformation under big incentives lead to poor clearance increases.When running steadily, the 0.504° difference angle relative to the 0.5° return clearance is less th an 1%, including small structural torsional deformation. On the whole, the shimmy damper follows the tire swing well.

3.4 Analysis of several influencing factors of clearance shimmy

3.4.1 Influence of gap size on shimmy stability

The initial axial rotation return clearance of 0.1°, 0.2° and 0.5° was set. The simulation results are shown in Fig. 29, including the angle time curve and the swing damping moment time curve.

By analysing the time domain curves of 0.1°, 0.2° and 0.5° return clearance, the pendulum frequency of landing gear increases from 24.6z to 26.7, 27.3 and 27.5Hz. The larger the clearance, the higher the pendulum frequency of nose landing gear, and the frequency gradually converges. The last constant swing angles of landing gear tires were 0.162°, 0.386° and 0.977°, respectively (Table 6). Results show that the greater the return gap, finally and stable running of the pendulum angle, the greater the angle of the pendulum and clearance ratio of 1.62, 1.93, 1.954, respectively, in order to prevent the landing gear tires have been in front of a big swinging angle of swinging forward, should minimise the shimmy damper and the outer cylinder sleeve connection between gear and linkage of return clearance.

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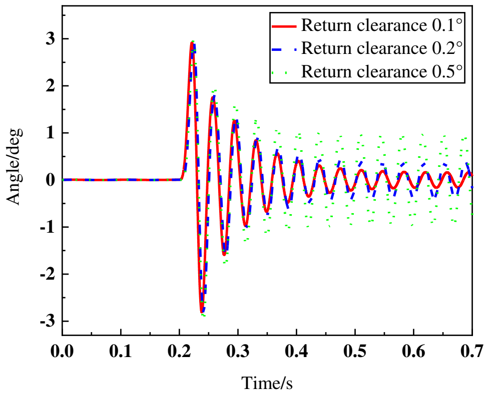

Figure 29. *Time domain curve of return clearance rotated in different axial directions.*

3.4.2 The influence of damping coefficient on shimmy stability

When the return clearance of the outer cylinder and sleeve is 0.5°, the original damping coefficient of shimmy is increased from 11 to 20N·m·s, and the simulation curve is as in Fig. 30.

As shown in the Fig. 30, after the nose landing gear is excited, the maximum swing angles of tire and damper are 2.733° and 2.12° respectively. When the landing gear shimmy finally staves, the swing angles of tire and damper are 0.753° and 0.243° respectively. With the increase of damping force, the final difference between the two angles is 0.51°. It is slightly larger than the set return clearance value, but the maximum swing angle decreases from 0.977° to 0.753°. It can be said that the structural clearance value determines the difference between tire swing angle and the swing angle of the damper, and the increase of the damping coefficient of the tire swing angle will simultaneously reduce the angle after the stable run.

3.4.3 The influence of running speed on shimmy stability

Considering the influence of the sliding speed on the clearance, the speed was reduced to 20m/s. Since each speed required a different damping coefficient, 30N·m·s damping coefficient was set so that its three cycles could converge to 1/4 of the maximum swing angle. The simulation results are as follows:

As shown in the Fig. 31, after the nose landing gear is excited, the maximum swing angles of tire and damper are 2.932° and 2.283° respectively. When the landing gear shimmy finally stably skidding, the swing angles of tire and damper are 0.866° and 0.352° respectively. After the speed decreases, the final difference between the two angles is 0.514°. The results show that at different speeds, the structural clearance determines the difference between the tire swing angle and the damper swing angle, and the speed only affects the angle after the stable run.

3.5 Analysis and summary of clearance type shimmy

After the above analysis of the influence of various conditions, it can be found that “clearance shimmy” is not only related to clearance position, clearance type, speed and shimmy damping coefficient, but also has different influence effects. According to the weight analysis and comparison of various clearance of landing gear, conclusions can be obtained as in Table 7.

It can be seen from Table 7 that although there are many landing gear installation positions and clearance types, there are six clearance types in Table 7 that have significant influence. The clearance size will affect the tire swing angle after stabilisation as well as the difference between tire swing angle and swing damper angle. In the case of small and small clearance, when the radial clearance, the front wheel swing angle of landing gear tends to 0°; when the axial clearance, the swing angle of the damper

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Table 6. *Oscillation frequency and stable oscillation angle under return clearance*

Type	Clearance size (°)	Frequency (Hz)	Constant angular (°)
Zero clearance	0	24.5	0
Axial rotation about axial return clearance	0.1	26.7	0.162
Axial rotation about axial return clearance	0.2	27.3	0.386
Axial rotation about axial return clearance	0.5	27.5	0.977

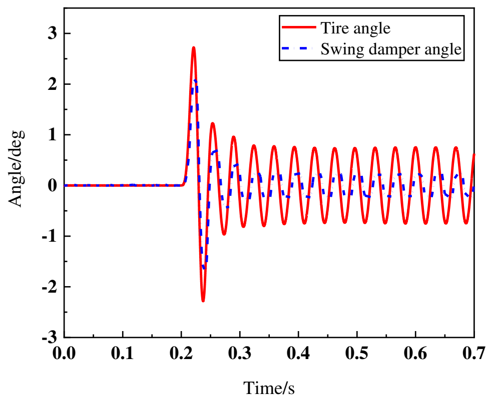

Figure 30. *20N·m· S shimmy damping coefficient 0.5° return clearance of outer cylinder and sleeve.*

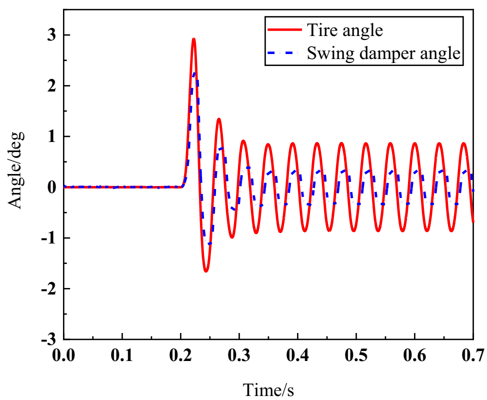

Figure 31. *20m/s running speed outer cylinder and sleeve 0.5° return clearance.*

tends to 0°; when the axial clearance, the influence of the radial rotation clearance is weaker than that of the axial rotation return clearance. Analyse the tire stability swing angle and the post-stabilisation angle and angle difference of the tire stability swing angle at the clearance at the installation position of the sleeve and the upper torque arm and the clearance at the installation position of the lower torque arm and the piston rod, and the ratio is 1.253, 1.267 and 1.268. The influence of the clearance at the installation position of the sleeve and the upper torque arm is about 126% of that at the installation position of the lower torque arm and the piston rod (Table 8).

Through the analysis of the condition of same return clearance, the plane taxiing speed and shimmy damping coefficient, the influence of such factors as the simulation results show that various factors will have an effect on the aircraft landing gear shimmy and its display frequency, tires and stable running of pendulum angle and swing angle difference of dampers are not the same.

Table 7. *The effect of different clearances*

Type	Position	Clearance size	Stable angular (°)	Angle difference (°)
Radial clearance	Front lug and outer tube	0.5mm	0	0
Axial clearance	Sleeve and upper torque arm	0.5mm	0.193	0.193
Axial clearance	Lower torsion arm and piston rod	0.5mm	0.154	0.154
Radial rotation around the axis clearance	Sleeve and upper torque arm	0.5°	0.555	0.432
Radial rotation around the axis clearance	Lower torsion arm and piston rod	0.5°	0.438	0.341
Axial rotation about axial return clearance	Outer tube and sleeve	0.5°	0.977	0.504

Table 8. *Influence of return clearance under different working conditions*

Operating condition	Clearance size (°)	Frequency (Hz)	Stable angular (°)	Angle difference (°)
Zero clearance	0	24.5	0	0
With clearance	0.5	27.5	0.977	0.504
Different damping coefficients	0.5	28.4	0.753	0.510
Different speed situation	0.5	23.4	0.866	0.514

Through the above analysis and research, when the aircraft generates “clearance type shimmy”, the actual measured data is used to conduct theoretical analysis first, qualitatively give the origin of the clearance and the size of the clearance value during the taxiing process of the aircraft, and finally quickly use the simulation software to conduct simulation analysis, which can save time and cost, and is conducive to rapid inspection and problem solving by maintenance personnel. If the coupling problem exists in the case of multiple clearances, a more reliable theoretical analysis can be given by analysing the influence weight of clearances and data simulation fitting, which has certain guiding significance for inspection and maintenance.

4.0 Influence of structure, strength, clearance and mutual coupling

The shimmy of the nose landing gear is affected by many aspects. Different structural clearances have different effects on the shimmy of the nose landing gear, the magnitude of the shimmy damping coefficient and the critical shimmy damping curve at different speeds. And mechanical stability of the nose gear pitch and the nose gear strut stiffness is one of the main factors that affect the landing gear shimmy, especially under the condition of the landing gear of clearance, the landing gear shimmy before there could be some coupling effect, so the article aiming at different speeds in “clearance type shimmy” the critical demand of shimmy damping coefficient. Considering the different mechanical stability distance and strut stiffness of the nose landing gear, as well as the condition of too large clearance, the main structure and strength induced shimmy effect of the landing gear in the “clearance shimmy” was analysed and studied.

4.1 Influence of mechanical stability distance on shimmy stability

Aircraft taxiing on the ground at a certain speed, free to deflection of the tire and the elastic vibration of pillar intermingled with the rotation of wheel, appear a kind of violent vibration phenomenon, it will cause the nose intense shaking, the gear itself is elastomer, when the speed to the excitation energy is

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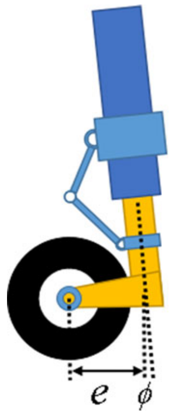

Figure 32. *Schematic diagram of stability distance.*

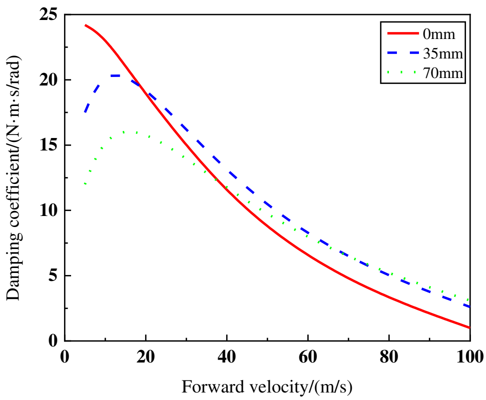

Figure 33. *Pendulum damping coefficient and velocity curve at different stability intervals.*

greater than the damping force shimmy occurs, combined with self-excited vibration, more and more intense until broken. So front wheel stability is necessary. The nose landing gear, which is responsible for shock absorption and steering, must have strong elastic resilience, so its stability is not good.

Landing gear wheel stability is apart from the access method is mainly a front wheel strut tilt or front wheel in the back, at the same time, the stability of the size of the also changes with different ground conditions, stable distance is too small, the stability of the aircraft ground movement is not good, stable distance is too large, pillar under bending moment increases, so, in the front wheel shimmy stability is an important influencing factors (Fig. 32).

Keep the nose landing gear inclination angle of 6°, change the mechanical stability distance of the nose landing gear, namely initial 70mm, 35mm and without mechanical stability distance respectively, and obtain the yaw damping coefficient required by the aircraft nose landing gear under the condition of stability margin at different speeds through dynamic simulation (Fig. 33).

Under the condition of the same landing gear stiffness, the stability distance of the nose landing gear has a certain influence on the shimmy of the aircraft nose landing gear. With the decrease of the mechanical stability distance of the nose landing gear, the shimmy performance of the aircraft nose landing gear tends to increase at low speed, but the damping required by the aircraft decreases sharply at high speed. Therefore, reducing the mechanical stability distance has a certain inhibitory effect on the shimmy of the nose landing gear during high speed sliding.

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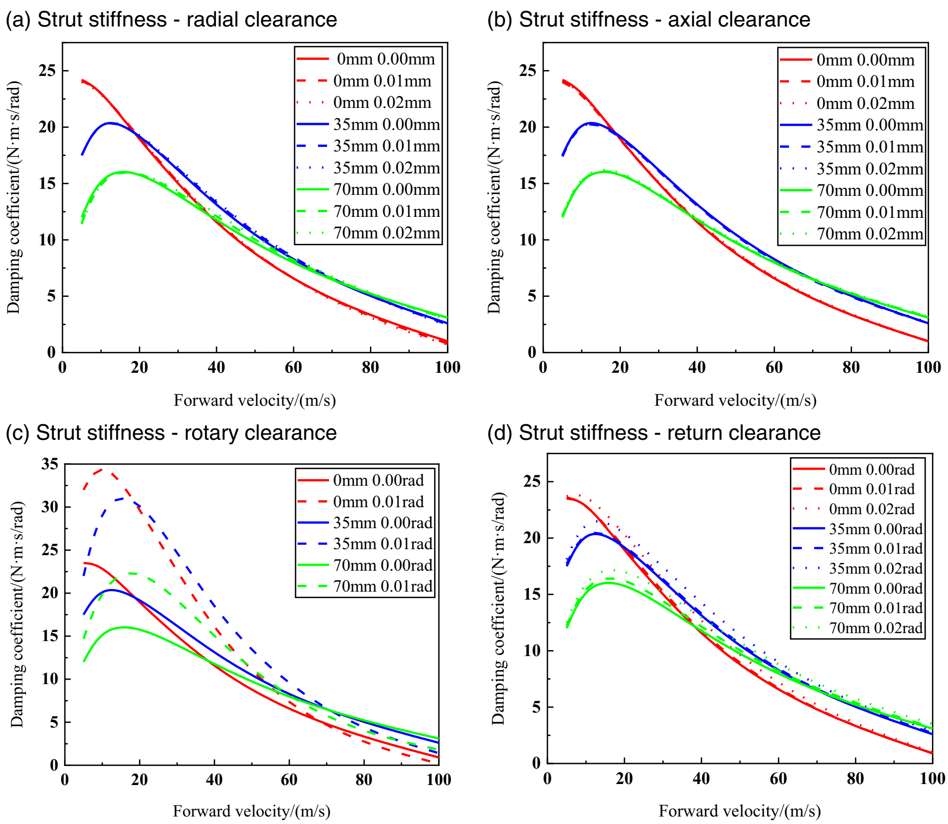

Figure 34. *Velocity curves of damping coefficients of different clearance sizes under different mechanical stability distances.*

4.2 Coupling effect analysis of clearance and stability distance

In the case of low speed and high speed running, the radial clearance plays a certain role in restraining the front swing, while in the case of medium speed running, the damping required for reducing the swing will increase. The greater the mechanical stability of the landing gear, the more obvious the increasing trend of the anti sway damping coefficient at medium speed. (Fig. 34).

In a small range, the axial clearance between the mounting parts of the nose landing gear has no obvious influence on the shimmy of the landing gear. Under the condition of different mechanical stability distances, different axial clearance will slightly affect the size of the shimmy damping coefficient at different speeds.

Nose gear gap between radial pivoting installation parts of landing gear shimmy is very obvious, the influence of low and middle speed skating in the process of running, pivoting radial clearance before will largely increase the aircraft landing gear shimmy need critical damping, but with the increase of speed, needed to shimmy damper coefficient decreased again. Therefore, the radial axial clearance has a bad influence on the shimmy of low and medium speed running, and has a certain inhibition on the shimmy of high speed running. As the stability distance increases, the swing-damping coefficient curves corresponding to the axial clearance of different sizes increase from 60 to 68m/s and then decrease to 62m/s at the velocity intersection.

Nose gear installed axial pivoting return clearance between components of the landing gear shimmy mainly lies in the influence of the return clearance cannot transmit real time during the period of force and moment, especially of shimmy damper, so with the increase of axial pivoting return clearance, the

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Table 9. *Influence of return clearance under different working conditions*

Code name	The stiffness value (N/rad)
High strut stiffness	6.02e6
Middle strut stiffness	1.75e6
Low strut stiffness	7.68e5

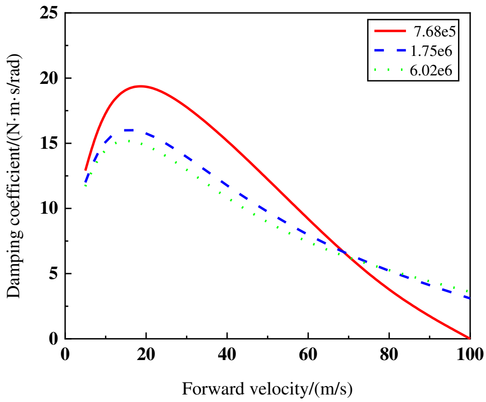

Figure 35. *Pendulum damping coefficient and velocity curve of different strut stiffness.*

aircraft landing gear shimmy before the required damping increases, but have less effect on the low speed and high speed, large influence on run in speed skating. When the stability distance is small, the return clearance will play a promoting role at low speed.

4.3 Influence of strut stiffness on shimmy stability

The lateral bending stiffness of struts has a great influence on the shimmy of landing gear. The structural shimmy of many aircraft nose landing gear during skid is caused by the weak lateral stiffness of the aircraft nose landing gear. Meanwhile, the structural stiffness also affects the frequency of shimmy and the deflection angle under the same excitation. By changing the material and material properties of the outer cylinder and piston rod of the nose landing gear, the three lateral bending stiffness models of the nose landing gear were obtained (Table 9).

The dynamic simulation results are as in Fig. 35.

As shown in Fig. 35, the lateral bending stiffness of the prop has a certain influence on the shimmy stability. When the lateral bending stiffness is large enough and the sliding speed is low, the damping coefficient required for anti-swing increases slightly with the increase of the lateral bending stiffness. When the sliding speed continues to increase, the damping curve of the shimmy damper required by the smaller strut lateral bending stiffness decreases more sharply. It can be seen that the anti-swing performance of the landing gear can be improved to some extent by increasing the lateral bending stiffness of the strut during low and medium speed running.

4.4 Coupling effect analysis of clearance and strut stiffness (Fig. 36)

When the overall stiffness of the nose landing gear is large, the radial clearance has a certain inhibitory effect on the forward shimmy in the case of low speed and high speed, while the required shimmy damping tends to increase in the case of medium speed. When the stiffness is weakened to a certain

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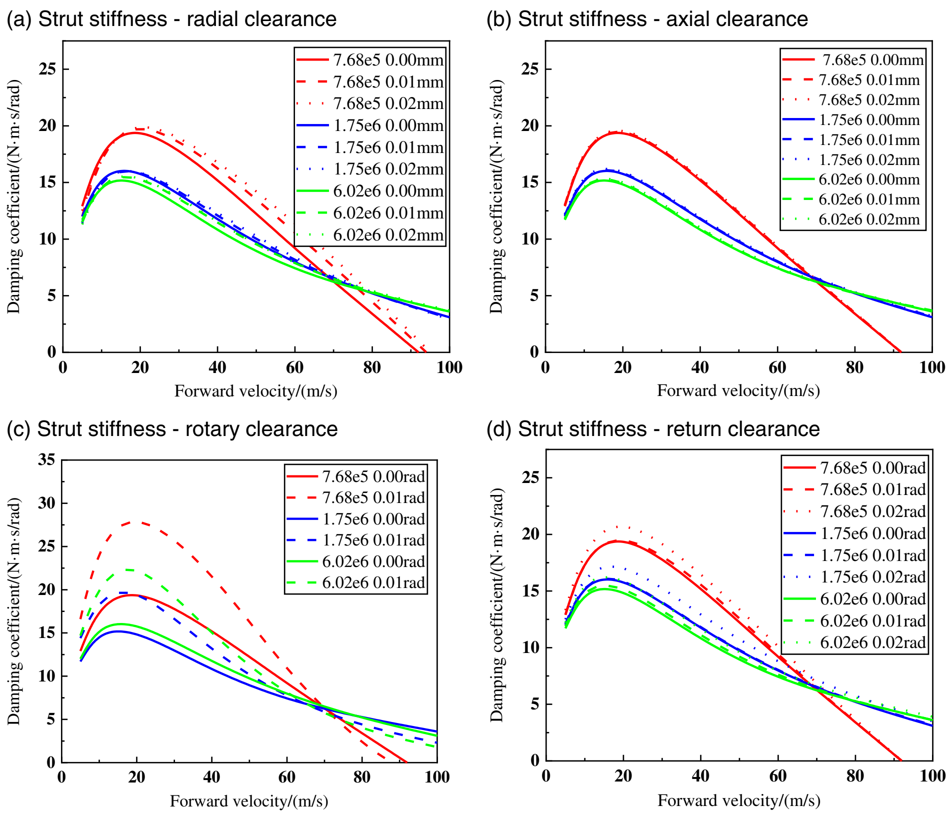

Figure 36. *Pendulum damping coefficient and velocity curve of different strut stiffness and gap size.*

extent, the radial clearance has a certain inhibitory effect on the shimmy of low speed running, while in the process of high speed running, the larger the radial clearance is, the larger the required shimmy damping coefficient is, and the smaller the strut stiffness is, the more obvious the influence of clearance is.

In a small range, the axial clearance between the mounting parts of the nose landing gear has no obvious influence on the shimmy of the landing gear. Under the condition of different strut stiffness, different axial clearance will slightly affect the size of the shimmy damping coefficient at different speeds.

Nose gear gap between radial pivoting installation parts of landing gear shimmy is very obvious, the influence of low and middle speed skating in the process of running, pivoting radial clearance before will largely increase the aircraft landing gear shimmy need critical damping, but with the increase of speed, needed to shimmy damper coefficient decreased again. Therefore, the radial axial clearance has a bad influence on the shimmy of low and medium speed running, and has a certain inhibition on the shimmy of high speed running. When the stiffness of the pillar increases, the swing damping coefficient curves corresponding to the axial clearance of different sizes increase first and then decrease at the intersection point of medium-high speed. The radial axial clearance deteriorates the stiffness of the flexible strut in the process of shimmy, which requires a larger shimmy damping coefficient.

The axial rotary return clearance between the mounting parts of the front landing gear, which mainly affects the shimmy during the return, cannot transmit the real-time force and torque due to the clearance, especially the shimmy damper torque, with the increase of axial pivoting return clearance, the aircraft landing gear shimmy before the required damping increases, but have less effect on the low speed and high speed, large influence on run in speed skating.

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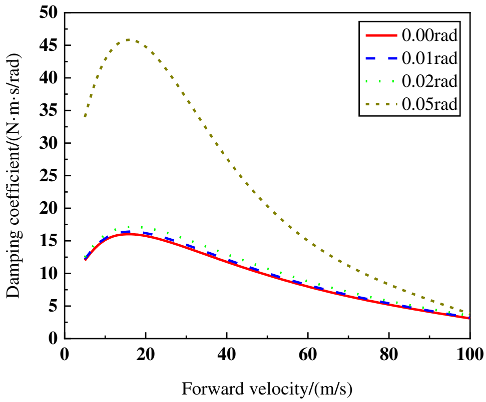

Figure 37. *Pendulum damping coefficient and velocity curve of excessive axial rotation clearance.*

4.5 Influence of excessive structural clearance on shimmy stability

In the case of extreme damage to the aircraft, there may be a large clearance, so it is very meaningful to study a large clearance. Considering the influence of excessive axial rotation clearance on the swing damping coefficient change, the simulation analysis is shown in Fig. 37.

The increase of the axial rotation return clearance in the nose landing gear structure of aircraft does not increase linearly to the increase of the shimmy damping curve as a whole, and the required damping coefficient reaches its maximum about 18m/s. In the absence of clearance, 0.01, 0.02 and 0.05rad, the corresponding damping coefficients of shimmy are 16.7, 17.3, 18 and 49N·m·s. At 100m/s, the corresponding shimmy damping coefficients are 3.1, 3.2, 3.5 and 3.8N ·m·s. It can be found that in the process of shimmy of low- and medium-speed aircraft, increasing the clearance will sharply and badly damage the anti-shimmy performance of aircraft forward lift, but in the process of shimmy of high-speed aircraft, excessive clearance has little influence.

5.0 Conclusion

Based on L-N contact theory, the flexible dynamics models of nose landing gear with different clearance of moving pair at different installation positions were established. The accuracy of the models was verified and various working conditions were analysed. The “landing gear clearance shimmy” was studied and the following conclusions were obtained.

Different types of clearance have different influences on different installation positions of structures. The clearance at the connection of nose landing gear and fuselage has little influence on the shimmy performance, and the clearance at the transmission of the shimmy damper mechanism and the existing clearance at the connection between the return clearance and the upper and lower torsion arm has a great influence on the “gap type shimmy”. The influence of the clearance between the upper torque arm and the sleeve is greater than that between the lower torque arm and the piston rod.

Study and analyse the time domain curve of the front landing gear swing angle. In addition to the radial clearance, in the case of large axial clearance, rotary clearance and return clearance, there will be a constant swing angle after the shimmy glide is stabilised, and the existence of clearance will increase the shimmy frequency of the front landing gear. Change the gap size, speed and damping coefficient of shimmy, analysis of tire angular curve and angle curve of shimmy damper, results show that the gap size determines the maximum angular amplitude and stable amplitude and amplitude and phase difference of the two, running speed and shimmy amplitude damping coefficient will only change both but will not change its amplitude difference.

By comparing and analysing the shimmy stability curves of different clearance types, the results show that the axial clearance has little effect on the shimmy performance of landing gear; radial clearance has a certain effect on the shimmy performance of medium speed running, which slightly improves the shimmy damping required by medium speed running. Rotational clearance affects the shimmy performance of the nose landing gear by affecting the force transmission of structural components. The required shimmy damping coefficient increases at low speed and decreases at high speed. The return clearance is mainly caused by the failure of the shimmy damper during the return, resulting in the decrease of the shimmy performance and the required shimmy resistance in the full speed range.

Through the calculation results of different types of clearances, strut stiffness and mechanical stability distance of the front landing gear, the results show that the radial clearance and axial clearance have little influence on the critical curve of shimmy damping speed, which is mainly to slightly increase the shimmy damping coefficient required for medium and low speed, and the rotating clearance around the shaft will increase the damping coefficient required for low speed and reduce the damping coefficient required for high speed The influence of clearance can be offset by increasing the mechanical stability distance and the lateral bending stiffness of the strut. At the same time, a large clearance will greatly increase the requirements for landing gear anti-swing, especially in the low-speed range. Therefore, special attention should be paid to the return clearance generated by the transmission mechanism and installation of the pendulum reducer.

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