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Finite element analysis of the axial stiffness of a ball screw

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Abstract. The ball screw was developed for high speed and high precision operation; therefore, increasingly greater demands have been placed on the stiffness of the ball screw. Firstly, ANSYS software was used to compare the axial stiffness of a single-nut and single-arc ball screw and a single-nut and double-arc ball screw when the spiral angle is not considered. On this basis, the model of a single-nut ball screw was established taking into consideration the spiral lead angle, and then the variations in displacement and stiffness when the ball screw pair was subjected to an axial force were determined. The axial contact stiffness of the double-nut ball screw pair, subject to a pre-tightening force, was analyzed, according to the above-mentioned steps. The simulation results demonstrated that under the same working conditions, the stiffness of the double-arc ball screw was larger by between 5～100 N/um than that of the single-arc ball screw. The spiral lead angle increased the axial stiffness of the ball screw pair, and the axial stiffness of the double-nut ball screw pair subject to a pre-tightening force was larger by between 790～1360 N/um than that of the axial stiffness of the single-nut ball screw pair.

1. Introduction

The ball screw pair is a critical component in the machine tool feeding system and other areas, due to its high precision and high stiffness. Therefore, it is of great theoretical and practical significance to study the stiffness characteristics between each joint surface in order to improve the static and dynamic performance of the ball screw pair and therefore the machining precision of the machine tool. Kamalzadeh A and Feng G-H *et al* researched the effect of the pre-stressing force and the gap between the contact pair on the overall precision of the ball screw using the experimental method [1,2]. However, the experimental process they used was complex and not conducive to the process. C-G Zhou and H-T Feng et al. analyzed the working condition of the ball screw when a pre-stressing force was applied [3], which was much closer to the real working conditions of the ball screw.

Considering the whole machine tool, B Fernandez-Gauna and T Liang analyzed the factors that influenced the vibration of the ball screw [4,5]. Sobolewski J Z and Garinei A *et al* analyzed the vibration of the ball screw, and carried out flaw detection on the ball screw through the vibration processes [6,7]. However, the displacement variation of the ball screw during the working process could not be quantitatively deduced. The point of least stiffness in the ball screw, i.e. the joint surface among the nut, lead screw, and the balls, was chosen to be studied and this paper has established three-dimensional models with and without the spiral lead angle, starting from the contact surfaces composed of a single ball and nut or lead screw in the joint part of a single-nut ball screw pair. ANSYS finite element software was used to analyze both models and the axial stiffness characteristics of the ball screw were studied. On this basis, taking into consideration the influence of the pre-stressing force on the stiffness of the ball screw, this paper analyzed the variations of the displacement and the stiffness of the double-nut ball screw pair, subject to a pre-stressing force, when the nut was subjected to an axial force. The principle diagram of the ball screw pair has been shown in figure 1.

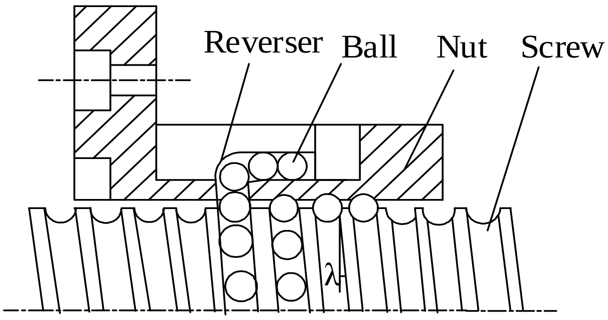

Figure 1. The principle diagram of the ball screw pair.

2. Analysis of axial stiffness without the spiral lead angle

2.1. Establishment of the three-dimensional model

The structural diagram of a nut can be depicted as in figure 2. Since the ball screw is an axisymmetric component, the stress condition of one ball in the ball screw can be taken separately for the analysis; the joint surface structure of a single ball has been shown in figure 3.

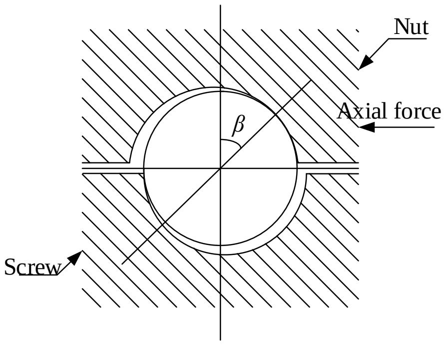

Figure 2. The structural diagram of a single-nut ball screw.

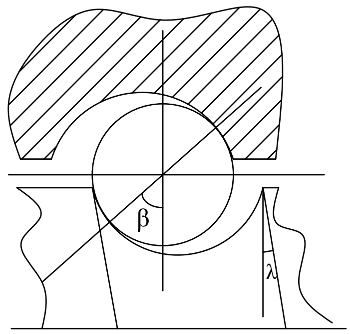

Figure 3. The joint surface structure of a single ball.

Table 1. Material parameter values.

Elastic Modulus	Poisson's Ratio	Density
210GPa	0.3	7800kg/m^3

Table 2. The dimension parameters of the ball screw.

Name	Value	Name	Value
Nominal Diameter of the Ball Screw d_0(mm)	40	Diameter of the Ball d_b (mm)	3.175
Lead Screw P_h(mm)	6	Curvature Ratio of the Raceway	1.04
Overall Length of the Ball Screw(mm)	480	Outer Diameter of the Nut d_n (mm)	63

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The material parameters and dimension parameters of the ball screw have been shown in tables 1 and 2, respectively.

The structural diagram of single nut ball screw has been shown in figure 4.

The structure was analyzed without the spiral lead angle λ, and then the three-dimensional model was established, using the SOLIDWORKS software, as shown in figure 5.

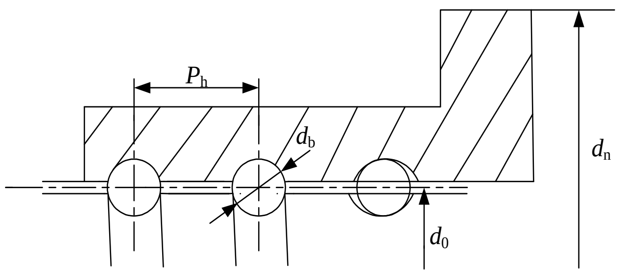

Figure 4. Structural diagram of a single nut ball screw.

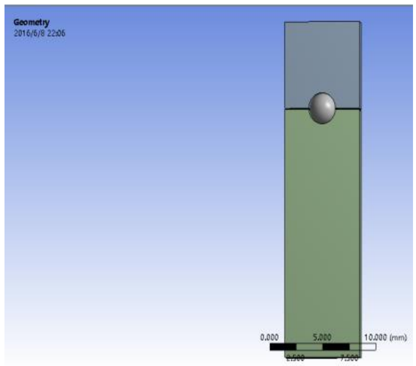

Figure 5. The structural diagram without the spiral lead angle.

2.2. ANSYS analysis process

In ANSYS, the two contact pairs of the ball-lead screw and the ball-nut were set up, respectively, as shown in figure 6. During the simulation process, the lead screw was fully constrained, the displacements for the nut in the Y and Z directions were constrained, and the displacement of the ball in the Z direction was constrained. A 500 N axial force was applied on the right side of the nut in the ball screw model, as shown in figure 7.

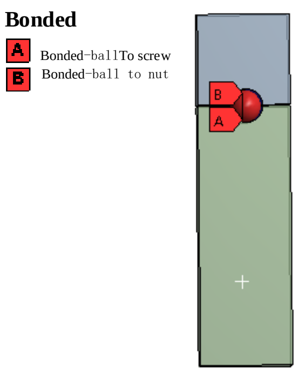

Figure 6. The establishment of the contact pair.

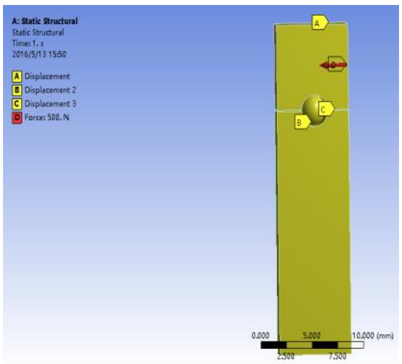

Figure 7. The addition of constraint conditions.

The variations in the axial displacement and the stress distribution of the ball screw have been shown in figures 8 and 9. The maximum stress of the ball screw pair appeared at approximately 45° from the horizontal direction, as can be seen in figure 9.

In this study an applied axial load of 500 N to 6000 N was selected, according to the Hertz point contact theory, the axial contact deformation expression for a single-nut ball screw without the spiral lead angle is:

$$\sigma a = 2.6992 \times 10^{-8} F^{2/3} (m) \tag{1}$$

$$K = \frac{F^{1/3}}{2.6992}\left(N/m\right) \tag{2}$$

where σ_a (m) is the axial contact deformation, F(N) is the axial force, and K(N/m) is the axial contact stiffness.

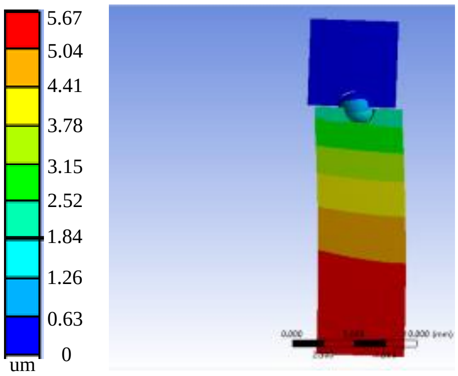

Figure 8. The variation of the axial displacement.

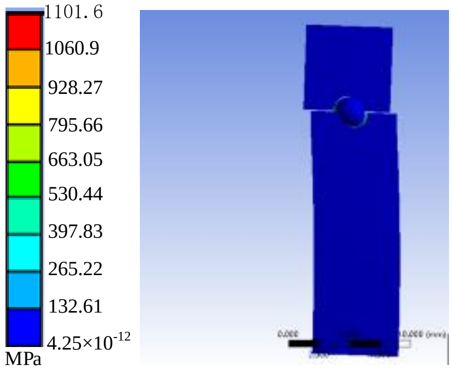

Figure 9. The stress distribution.

The corresponding displacement of the nut was calculated according to this equation. The comparisons of the FEM analysis and the solution of the theoretical model have been shown in figures 10 and 11.

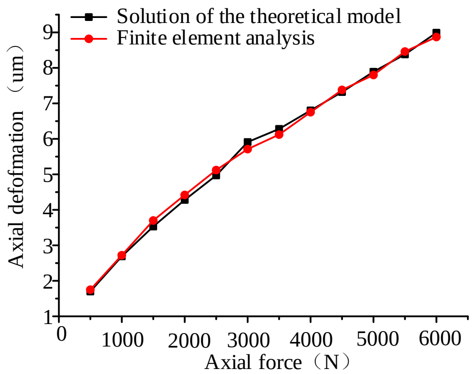

Figure 10. The axial deformation of the ball screw pair.

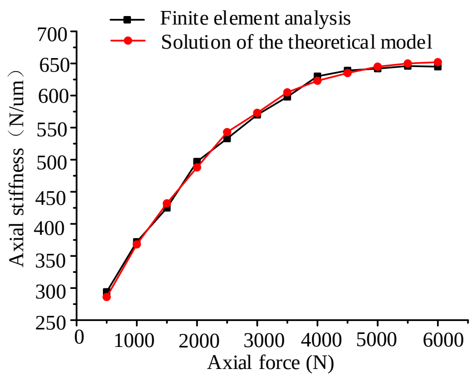

Figure 11. The axial stiffness of the ball screw pair.

It can be seen from figure 10 that the simulation values were basically in accordance with the calculated values from the theoretical model.

The stiffness values have been shown in figure 11, and from this it can be seen that the stiffness of the ball screw pair increased with the increase of the axial load, and the stiffness was relatively low at the start, and it trended to be stable as the load increased. The errors can be analyzed to show that the meshing at non-contact areas was sparse; therefore, the accuracy of the finite element solution was reduced to some extent.

3. Axial stiffness analysis of the single-nut and double-arc ball screw

3.1. Establishment of the three-dimensional model

The structural diagram of the ball screw pair with a double-arc raceway has been shown in figure 12.

The three-dimensional diagram of the ball screw pair with a double-arc raceway was constructed using the SOLIDWORKS software package, as shown in figure 13.

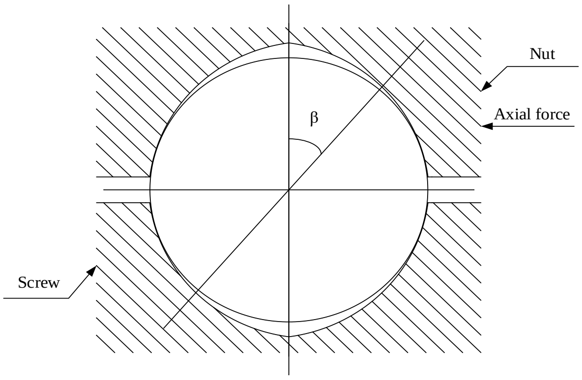

Figure 12. The structural diagram of the ball screw with a double-arc raceway.

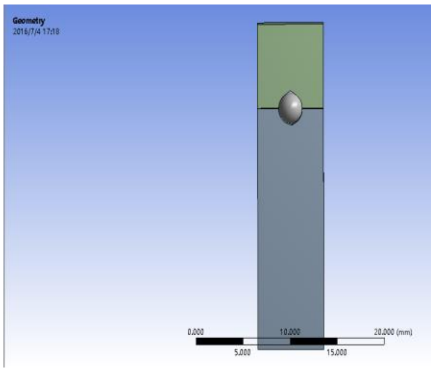

Figure 13. The diagram of the ball screw with a double-arc raceway.

3.2. ANSYS analysis process

The simulation process was the same as mentioned above. Then, the axial displacement variations and the stress distributions under a load of the 500 N axial force were obtained, as shown in figures 14 and 15.

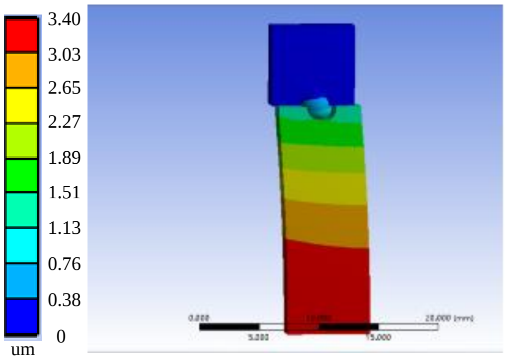

Figure 14. The variation of axial displacement.

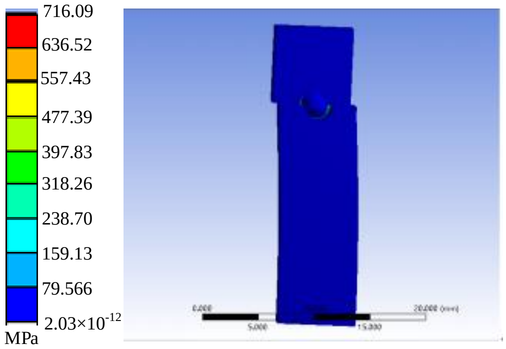

Figure 15. The stress distribution.

Accordingly, the variations in the displacement under the axial load were within the range of 500 N and 6000 N.

3.3. Axial displacement comparison of the single-arc ball screw and the double-nut ball screw

The variations in the axial deformation of the single-arc and double-arc raceway ball screws with the variation of axial load, according to the simulation results, have been shown in figure 16.

It can be seen that under the same axial load, the stiffness of the double-arc ball screw was bigger than that of the single-arc ball screw.

4. Analysis of axial stiffness with the spiral lead angle

The three-dimensional coordinate system of the ball screw pair can be analyzed, as shown in figure 17.

The displacement deformation and the stress distribution under an axial load of 500N have been shown in figures 18 and 19, respectively.

It can be seen that the maximum stress appeared at the locations where contact was made between the ball, the raceways of the lead screw, and the nut, and appeared at approximately 45° from the horizontal direction. Meanwhile, the maximum variation in the axial displacement occurred on the lead screw, which grew larger toward the center, namely, the contact stiffness of the nut's raceway surface was larger than the contact stiffness of the lead screw's raceway surface. Therefore, when checking the strength and fatigue wear of the ball screw pair, it should be based on the deformation at the contact point between the ball and the lead screw. According to the FEM solving steps, the displacement variations for axial loads of between 500 N-6000 N were obtained in turn.

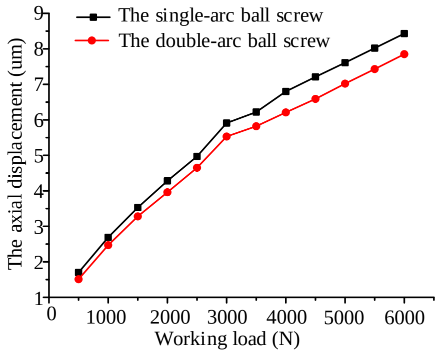

Figure 16. The axial deformation of the ball screw pair.

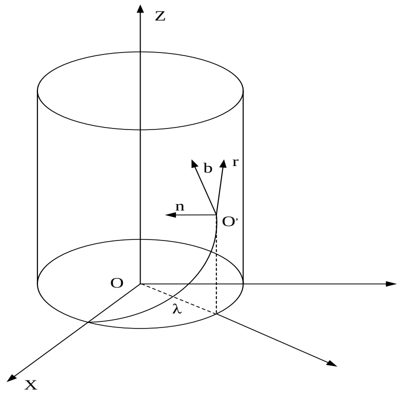

Figure 17. The coordinate system of the ball screw pair.

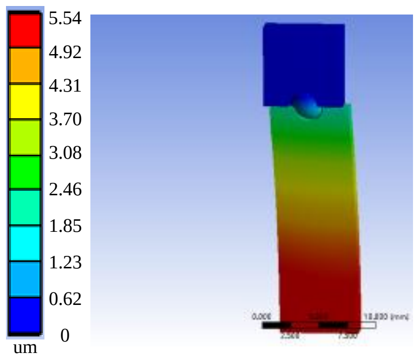

Figure 18. The variation of axial displacement.

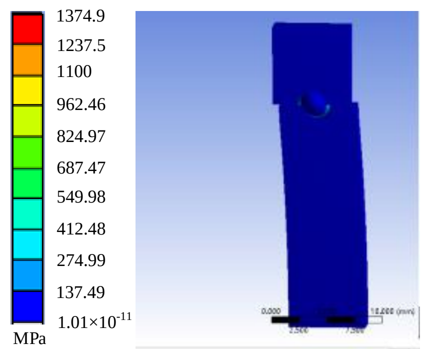

Figure 19. The stress distribution.

5. Comparisons of displacement variations and stiffness for the two models

The displacement variations and stiffness of the nuts in the ball screw pairs, with and without the spiral lead angle, were compared, as shown in figures 20 and 21.

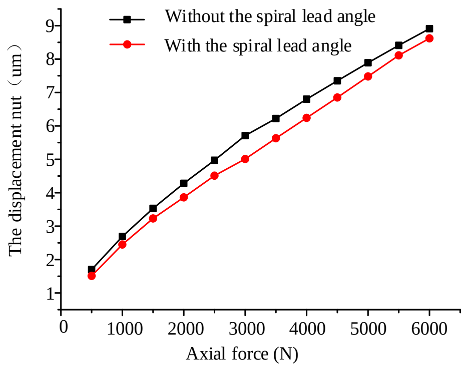

Figure 20. Comparison of the axial deformations.

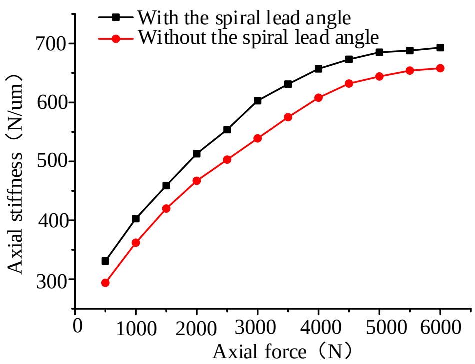

Figure 21. Comparison of the axial stiffness.

It can be seen from figure 21 that the stiffness of the ball screw pair model with the spiral lead

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angle is greater than that of the ball screw pair without the spiral lead angle. The working accuracy of the ball screw pair model with the spiral lead angle is higher, and the stiffness gradually trends to a stable value with the increase in load.

6. Pre-stressed double-nut ball screw

6.1. Establishment of the three-dimensional model

The structural diagram of the double-nut ball screw subjected to an axial force under working conditions can be obtained, as shown in figure 22, and the three-dimensional model was established, as shown in figure 23.

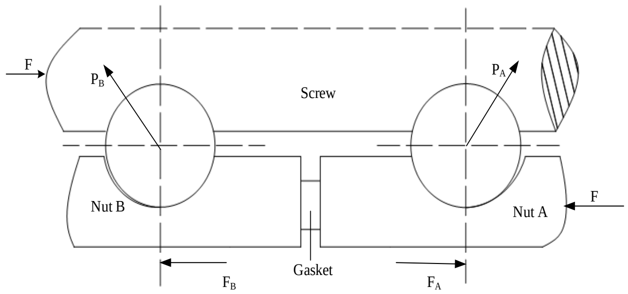

Figure 22. The structural diagram of the double-nut ball screw.

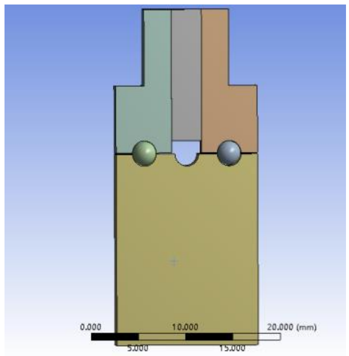

Figure 23. Model of the double-nut ball screw.

6.2. ANSYS simulation process

The simulation process was consistent with the process mentioned above. Preload is a force that is pre loaded on the ball screw to enhance the reliability and tightness of the connections before a working load is applied, in order to prevent gaps or relative slippage between the connected parts after loading. Moreover, the pre-tightening force was approximately one third of the maximum working load F_{max}, i.e. $F_P \approx F_{max}$. Therefore, the pre-tightening force $F_P = 2450$ N. The pre-tightening force was added to the middle of the pre-tightening gasket, based on the pre-tightening unit PRETS179 that was used; meanwhile, the load was applied on the right side of the nut in the ball screw model. The axial displacement variation and the stress distribution under the action of a 500N axial force were obtained, as shown in figures 24 and 25.

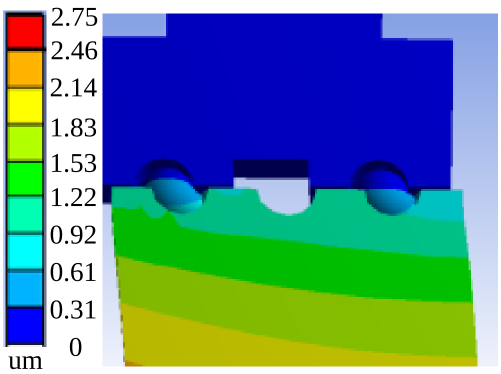

Figure 24. The axial displacement variation.

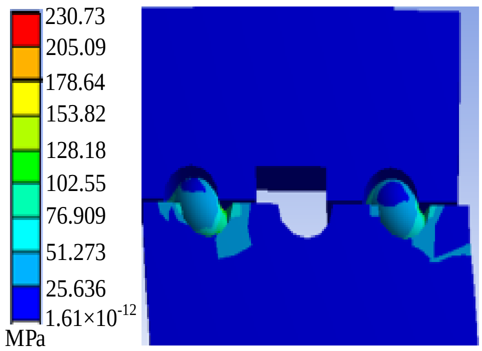

Figure 25. The stress distribution.

The unique variation of the displacement of the nut, based on the sorted simulation data, when an axial force of between 500 N-6000 N was applied in sequence to the right side of the nut, has been shown in figure 26.

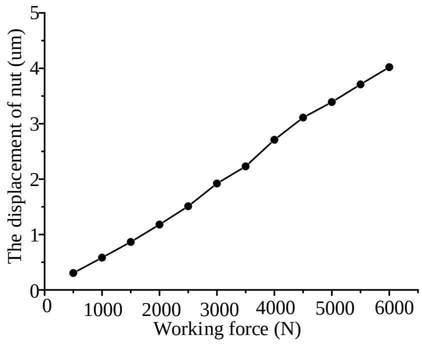

Figure 26. The variation of the displacement of the double-nut ball screw.

It can be clearly seen that the nut displacement is relatively small for the pre-stressed double-nut ball screw under the effect of an axial force, and the corresponding stiffness was larger and the stiffness variation was more moderate, which reflected its structural superiority.

7. Summary

- The location of maximum stress appeared at approximately 45° from the horizontal direction, i.e. the location of the contact angle. The axial stiffness increased with the increase of axial load. Under the same axial load, the stiffness of the double-arc ball screw was larger than that of the single-arc ball screw.
- The contact stiffness of the nut's raceway surface was larger than the contact stiffness of the lead screw's raceway surface. Therefore, when checking the strength and fatigue wear of the ball screw pair, it should be mainly focused on the deformation of the contact point between the ball and the lead screw.
- The stiffness of the ball screw pair model with the spiral lead angle was greater than that of the ball screw pair without the spiral lead angle. The displacement variation of the double-nut ball screw with the spiral lead angle was smaller than that of the ball screw without the spiral lead angle, hence the higher accuracy.
- The working precision of the double-nut ball screw was higher and the stiffness was larger than those of the single-nut ball screw. This reflects the influence of the pre-tightening force, and the nut type of the ball screw on the stiffness and accuracy of the ball screw pair.

Acknowledgments

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