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Correlation between preload and no-load drag torque of ball screws

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This paper examines the relationship between the preload and no-load drag torque of ball screws. Based on a new equation to calculate the normal contact load between balls and the screw raceway under no external load, a new correlation between preload and no-load drag torque is proposed. Meanwhile, a novel preload-adjustable ball screw mechanism and a drag torque measuring system are constructed. Three LCM300 load cells and a force sensor are implemented for obtaining the experimental data from the constructed systems. Preload and no-load drag torque are obtained and analyzed. It is found that there is a large gap between the calculated preload by traditional formula and the measured value. The experimental results, agreeing well with the theoretical values calculated in this paper, show there exists a linear correlation between preload and drag torque in a preloaded ball screw mechanism. Furthermore, the derivation of the drag torque would be decreased when increasing the preload, which can improve the stability of the screw during its operation. This study provides an accurate correlation to obtain the preload through the no-load drag torque for preloaded ball screws, which is significant for better performance of ball screws as well as the CNC machine tools.

1. Introduction

Ball screws, which can transform force and motion, are the machine component most frequently used for computer numerical control (CNC) machining equipment [1]. Two most important features of the mechanism are its high positioning accuracy and heavy load carrying capacity. In recent years, the growing demand for high precision, high speed and high durability in CNC machine tools has propelled high performance and reliability to be the most important index for ball screws.

In order to eliminate backlash and provide the required rigidity for dynamic processes, most applications in the field of production technology use preloaded ball screws of different levels [2]. For example, oversized balls are often used in the single-nut ball screw (4-point-contact) to provide preload for light to medium loading applications and a pin is inserted between the double-nut ball screw (2-point-contact) to give proper preload for medium and higher loading applications [3–5]. Increasing the pretension will raise the achievable dynamics as well as the maximal allowable thrust load on the feed drive. However, the service life of ball screws would be decreased due to excessive wear and heat generation if a superabundant preload was applied. An appropriate and constant preload is the basic requirement for a ball screw to retain its reliability in position accuracy and dynamic characteristic, which is significant for the performance of CNC machine tools. Therefore, it is important for researchers and engineers to know the preloading value of ball screws precisely.

So far, a direct identification of the preload is generally associated with high complexity and expenses, and therefore, is not used in practice [6]. Usually the preload level of a ball screw resulting from assembly is determined by measuring the drag torque at 100 rpm when no external loads are applied [7]. In the most commonly used formula of drag torque and preload of ball screws, for example, according to the product technology manuals of NSK [8], only the helix angle is taken into consideration, other parameters, such as the contact angle or the coefficient of friction are not considered.

Verl and Frey studied the correlation between feed velocity and effective preload in ball screw drives through a preload sensing mechanism, while the preload level is un-adjustable for a certain ball screw [6]. With a preload-adjustable ball screw feed drive system, Feng and Pan examined the relationship between ball screw preload variation and vibration signals [9]. However, the drag torque of the mechanism presented in their study cannot be diagnosed. Xu, et al. develop a new systematic creep analysis model to calculate the friction of ball screws. Their study can be used to calculate the mechanical efficiency of ball screws (under axial load) other than the drag torque, and the experimental verification is totally in doubt [10].

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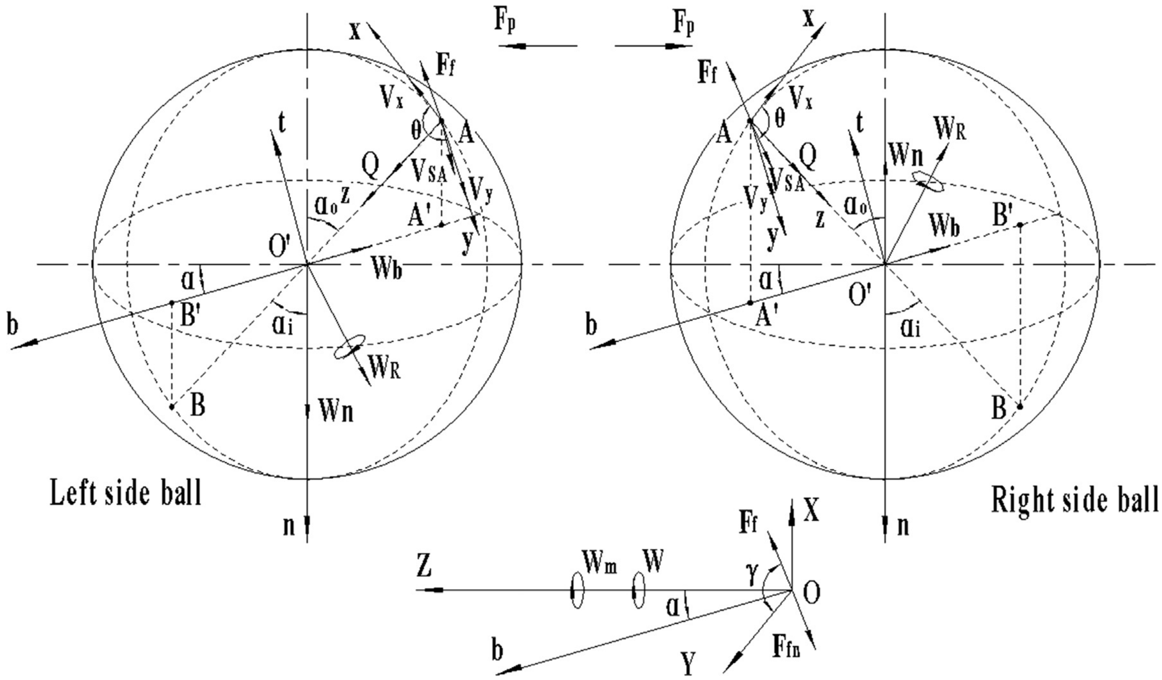

Fig. 1. Normal force, sliding velocity and friction force formed at the ball-nut contacts.

There is a lack of literature discussing the correlation between preload and drag torque of ball screws. And the existing formula is not applicable to the cases of both the double-nut preloaded type and the single-nut preloaded type with an offset pitch. Thus, a new equation to calculate the normal contact load between balls and the screw raceway under no external load was derived, and then a new accurate correlation between no-load drag torque and preload was built. With a novel preload-adjustable ball screw mechanism and a drag torque measuring system constructed in this research, the preload and drag torque at different preloading levels and positions of the screw were measured. The experimental results verify the correlation derived in this paper.

2. Preload and no-load drag torque of ball screws

According to DIN ISO 3408–3 [7], the test instructions of preloaded ball screws are as follows

1. System preloaded (with or without end seals).
2. For the recording of the radial preload force, couple the ball nut body to a load cell at the distance l (mm) from the axis of rotation.
3. Take recordings of the force indicator at a rotational speed 100 rpm in both directions of rotation.
4. For lubrication, use a lubricant of ISO viscosity grade 100.

It can be seen from the test instructions that the rotational speed is very low (100 rpm). According to Ref. [11], the contact angle between the ball and the screw or the nut can be regarded as 45 degree when the rotational speed is lower than 1000 rpm.

2.1. Problem of current formula

According to Ref. [8], the relationship between preload and drag torque in ball screws can be expressed as follows:

$$T_p = \frac{0.05}{\sqrt{\tan\alpha}} \bullet \frac{F_p \bullet P_h}{2\pi} \tag{1}$$

where α, T_p, F_p and P_h are the lead angle, drag torque, preload and lead of the ball screws, respectively.

Eq. (1) is based on the energy conservation principle, which means the work done by the drag torque ($T_p \bullet 2\pi$) is equal to that done by the preload ($F_p \bullet P_h$) with a coefficient of $\frac{0.05}{\sqrt{\tan\alpha}}$. However, the preload is an internal force and does not do any work. And for a certain applied preload, the contact angle would directly determine the normal force between the ball and the raceway, and therefore, influence the friction and drag torque, which is not reflected in Eq. (1). Besides, for a ball screw mechanism of the double-nut preloaded type and the single-nut preloaded type with an offset pitch, the drag torque is applied by two nuts or two sides of balls, which is not applicable for Eq. (1). This would lead to a large gap between the calculated preload by Eq. (1) and the actual value under the above conditions.

2.2. New correlation between preload and no-load drag torque

It has to be pointed out that, according to DIN ISO 3408–3 [7], the ball nut body is coupled to a load cell during the measurement of the drag torque. Thus, the obtained drag torque is just the moment applied by the friction force on the ball-nut contact point at a distance 1 (mm) from the screw axis, independent of the moment between the balls or the ball and the screw. The friction model in Ref. [10] includes those above three kinds of moment, while the experiment verification is the drag torque test of the preloaded ball screw under no load, which is the drag torque between the ball and the ball nut track. Thus the experiment verification is obviously not right. Other studies on the preload of ball screws are based on the drag torque through Eq. (1) [6]. Thus, there's no right formula of the correlation between no-load drag torque and preload of ball screws in literature. Based on the kinematics and dynamics analysis between the ball and the nut, a completely new model was built. As shown in Fig. 1, three coordinate systems are established first: the global coordinate system, Frenet–Serret coordinate system, and contact coordinate system. The global coordinate system, (X,Y,Z), is fixed in space with its Z-axis coincident with the axis of the screw. The Frenet–Serret coordinate system [12], (t,n,b), is fixed at the ball center (O′) whose