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Kinematical analyses and transmission efficiency of a preloaded ball screw operating at high rotational speeds

Chin-Chung Wei *, Ruei-Syuan Lai

Department of Power Mechanical Engineering & Center of MEMS Design and Application, Formosa University, Yunlin 63208, Taiwan, ROC

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ABSTRACT

High rotational speeds in ball screws (greater than 1000 rpm) cause more slip motion between balls and raceways than slower speeds. This increase in the slip motion increases the friction at the contact area, thus increasing the driving torque required for a high-speed ball screw. Theoretical analyses of the kinematics of a preloaded single-nut, double-cycle ball screw operating at high rotational speeds are presented in this study. The mechanical efficiency obtained from the theoretical driving torque, axial load and the orbital angular speeds of the ball was confirmed with experimental data. The slipping–rolling behaviour of each contact area was well developed from the distribution of stagnation lines consisting of pure-rolling points. Increasing the initial contact angle can significantly reduce the distance between the pure-rolling point and contact centre and decrease the driving torque, especially when the operating axial load approaches the applied preload. The mechanical efficiency can thus be increased, and wear may also be avoided by reducing the slip motion occurring at contact areas between balls and raceways.

1. Introduction

The increasing demands in precision engineering applications for positioning systems have instigated research into ball screws. The reciprocating ball screw mechanism is a force and motion transfer device. The most important advantages of the mechanism are its good positional accuracy and high driving speeds. These advantages make it a suitable drive mechanism for the feed–drive mechanism of machine tools and high-precision levelling platforms. The high translation speed ball screw is a major component in rapid processing devices, which require the balls to reach a linear velocity of 90 m/min and an acceleration greater than 9.8 m/s^2. In these situations, a high translation ball screw must be designed with a high degree of positioning accuracy and stiffness by applying a higher preload; however, an applied preload increases the initial driving torque, friction and heat. Therefore, important design criteria for a high-translation, preloaded ball screw include a low driving torque and a minimisation of slip motion to maximise mechanical efficiency.

The ball screw mechanism is a closed system. It is difficult to study directly what happens inside a ball screw when it is moving, so numerical analysis may be used to solve several kinetic parameters in terms of the kinematics and dynamics of the system and then build expressions for performance parameters, such as the friction coefficient, slide–roll ratio and mechanical efficiency. Because the kinematic behaviour of a ball screw is similar to a ball bearing, the analytical analysis applied to a ball bearing can thus be exploited and modified to fit the ball screw motion. Harris [1,2] analysed how the skidding behaviour between the balls and the inner raceway induces rolling surface distress, which can eventually lead to bearing destruction. The model developed by Harris was used in the present study to determine the contact angles of the ball screw as well as several angular velocities using geometric transformation.

Lin et al. [3] presented a theoretical study on the kinematics of the ball screw mechanism in which a function was developed to understand the motion of the balls and their contact patterns with the contact elements. These derivations describe the motion of a

* Corresponding author. Tel.: +886 56315414; fax: +886 56312110.
E-mail address: ccwei@nfu.edu.tw (C.-C. Wei).

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ball in a ball-screw system and show that slipping often occurs between the ball and the nut (or the screw), which means that the no-slip condition assumed in two previous studies [4,5] is unrealistic. Lin et al. [6] also introduced three methods that determine the mechanical efficiency of the ball-screw by developing a closed-form solution for the mechanical efficiency of a ball-screw in motion. Using the theory, an optimum design was then created for the mechanism; however, the friction coefficients, normal forces and contact angles created at the ball-screw and ball-nut contact areas were assumed to be equal. Also, the drag force produced by a ball moving in an oil lubricant was not considered in the force balance. Kinematic analysis of the ball screw mechanism that considered variable contact angles and elastic deformations was studied by Wei and Lin [7]. Their theoretical analysis was developed for a ball-screw with a single nut and a single cycle of balls. Therefore, no preload effect on the mechanical efficiency was discussed in the study. An analytical method developed by Takafumi, et al. [8] was used to determine the motion of the ball and the ball load distribution, including the effect of the motion for a given ball-screw geometry and its operating conditions. Here, the motion of the ball is based on raceway control theory introduced by Jones [9].

Nakashima et al. [10] examined ultra-precision positioning of a fine feed system using the elastic deformation of lead screws, which changes with the preload on a double nut. Wei et al. [11] presented an analytical model for a preloaded ball screw system with lubrication, and the numerical results confirmed the mechanical efficiency of their experimental data. They observed that the mechanical efficiency of a ball-screw decreases quickly when the applied axial load approaches the preload. The applied axial load must be at least 2.8-times higher than the preload [10] to maintain a high mechanical efficiency.

Contact geometry and friction in the contact areas vary greatly depending on whether the screw is operating at high or low rotational speeds. Usually, 1000 rpm is the dividing point between the two speed sub-regions. The effects of high rotational speeds cause a significant increase in the centrifugal force of the balls [7] and the slip motion formed at the ball and the raceway. The centrifugal force of a ball operating at a high rotational speed of the screw was determined through a function that related the rotational speed of the screw and the axial load applied to the nut [7]. An increase in the centrifugal force of a ball leads to a decrease in the normal force at the ball-screw contact but an increase in the normal force at the ball-nut contact. The contact angles will vary several degrees, and the other parameters (normal force, friction, slipping-rolling ratio, etc.) are also affected. In the present study, theories for the kinematics of a preloaded single-nut, double-cycle ball-screw were developed under a lubrication condition. The preload of the ball was adjusted by offsetting the centre pitch of the nut on the two ball tracks. An axial load was assumed to be applied at the end of the ball screw, towards the left of the nut and parallel to the screw axis. Also, the difference in contact angles between the left and right ball-nut contact points was investigated. The variations of these two contact angles in relation with the axial load output were studied. The preload is the sub-region division between small and large axial loads. The following parameters were obtained for the two ball tracks under an axial load operating in either of these two sub-regions: contact angles, pure-rolling points, driving torque and mechanical efficiency.

3. Theoretical analysis

A preload can eliminate the axial backlash and increase the stiffness of a ball-screw, but an excessive preload will increase the friction of the ball screw, thus lowering its mechanical efficiency. Single-nut preload in the present study was achieved by providing an offset pitch, λ, to the nut as shown in Fig. 1. The λ value varies with the axial load. Because the axial load is applied to the nut symmetrically with respect to (w.r.t.) the screw axis, all of the balls in the same track should assume the same loading conditions. Therefore, kinematic analyses were only performed for one pair of balls: one ball to the left ball track and the other to the right. Fig. 1(a) represents the ball screw operating under a preload but without an axial load; Fig. 1(b) shows the ball screw operating under both a preload and an axial load, where the preload is greater than the axial load. Fig. 1(c) shows the ball screw operating under a load condition opposite to that shown in Fig. 1(b). The offset-type preload can add a λ offset to the centre pitch of the nut. The nut's centre pitch $(L+\lambda)$ is thus greater than the screw's centre pitch (L). Due to this offset, the nut and the screw generate a preload in the balls.

When an axial load is applied to the left end of the nut, the offset λ will produce an elastic deformation δ_λ at the left end of the nut (see Fig. 1(b) and (c)). When a ball is trapped between the nut and the screw, elastic deformations are generated at the two elliptical contact areas formed where the ball is in contact with the screw and the nut. The largest elastic deformations in these two elliptical contact areas are $\bar{\delta}_o$ and $\bar{\delta}_i$, respectively. The angle formed between the n-axis and the line joining the centre of either of these two elliptical contact areas is the contact angle. The contact angle formed at the nut (α_o) is different from that formed at the screw (α_i), unless no axial load is applied. If a non-zero axial load is applied, either α_o or α_i formed on the right ball track will be different from that formed on the left track. Before applying an axial load to the ball screw system, all four contact angles have the same value. In this situation, the contact angles formed at each contact area are the same and are called initial contact angles α^o. However, a preload in the nut causes the line joining the centres of two elliptical contact areas formed in the right ball track to incline leftwards at an angle α_o, whereas the line in the left ball track will incline rightwards at the same angle. When an axial load is applied (but is less than the preload), the contact angle (α_{o_L}) formed in both the nut and the left ball track will be reduced, moving the contact centre towards the n-axis. However, the contact angle (α_{o_R}) formed at the nut, as well as on the right ball track, will be moved far away from the n-axis. The contact angle α_{o_L} will continue to decrease until the axial load approaches the preload. When this axial load almost reaches the preload, the deformations δ_{o_L} and δ_{i_L}, which originally existed on the left ball track, will disappear, and the normal force acting on the contact area becomes zero. This contact point in left ball-nut contact area will move to the opposite side of the n-axis if the axial load applied to the system is larger than the preload. This abrupt shift of the contact angle α_{o_L}, when the axial load is almost equal to the preload, indicates that a solution does not exist for several parameters.

To study the kinematics and dynamics of the ball screw mechanisms, four coordinate systems [3] are needed to describe the motion of the three components and their contact behaviours. The (x', y', z') coordinate system is fixed in space with its z' axis coincident with the screw axis (see Fig. 2). The rotating coordinate system, (x_r, y_r, z_r), also has its z_r-axis coincident with the screw