EXPERIMENTAL STUDY OF FRICTION LOSSES IN BALL SCREWS

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ABSTRACT

In contrast with conventional sliding lead screws, ball screws with rolling friction between the screw and nut have high efficiency, accurate positioning and smooth, quiet operation at low friction that make this kind of systems ideal for use in a wide variety of equipments (precision measuring, transportation, office automation, medical) in many industries such as machine-tools building, robots, fine mechanics, automotive, truck and tractors, aircraft, nuclear reactor, printing, paper-processing, automatic, textile, cutting and grinding machines.

The safety of mechanical systems that use ball screws depends on friction phenomena on the ball and tracks contact surfaces which can significantly reduce the positioning accuracy.

An experimental equipment was designed and tested in order to measure friction losses in a ball screw transmission. The mechanical device has two hydrostatic pretensioned ball nuts which ensure a highly accurate transmission of motions with an increased rigidity, especially for accurate positioning. Experimental tests were carried out on a 32 mm diameter worn-out ball screw in dry friction conditions, at low speed and thrust loads.

KEYWORDS: Rolling friction losses, ball screw system.

1. INTRODUCTION

The ball screw system is a force and motion transfer device belonging to the family of power transmission screws. Power screws are used to convert rotary motion to linear motion of the meshing member along the screw axis and are used to lift weights (screw-type jacks), exert large forces (presses, tensile testing machines) or obtain precise positioning of the axial movement.

The main feature of ball screws is that the sliding friction produced by conventional screws has been replaced by a rolling friction, in the same way as it is occurring in ball bearings. This characteristic is obtained constructively by a free movement of the balls in helical grooves made on both screw shaft and nut surfaces and a recirculating system. Figure 1 shows one solution for closed circuit of the balls in a ball screw.

Fig. 1. External recirculation nut

Ball screw systems are normally preloaded in order to eliminate backlash, to increase the overall rigidity of the ball and nut system and to improve the positioning accuracy. Preloading should be done with
care (e.g. too high preload on the nuts would result in an increase in the system driving torque, which in turn would mean a shorter operational life and a higher temperature).

In this way, ball screw converts rotary motion to linear motion, or torque to thrust, and vice versa. It is primarily a power screw with a train of ball bearings riding between the screw and the nut in a recirculating track. The screw has a rounded shape to conform to the balls. Due to the extremely low rolling friction coefficient of the ball screws, in comparison with the sliding friction of conventional screws, the efficiency of the former is much higher, in some cases as much as 97% and 98%.

All these characteristics contribute to ensuring a long service life of the ball screws and their working with high positioning and motion accuracy. They also allow for transfer speeds much higher than those obtained with conventional trapezoidal thread screws. Other interesting features are fewer vibrations and displacements, less wear and overheating and, finally, lower drive powers. Because of their metal-against-metal contact and subsequent sliding friction, conventional screws require a high initial driving force. By contrast, only a small force is needed by the balls of a ball screw to overcome the rolling friction.

In this paper we will examine the friction losses in ball screw for average loads and low speeds, peculiar for NC machine-tools, on especially designed equipment in Machine Design Laboratory.

2. TESTING EQUIPMENT DESCRIPTION

The experimental equipment for the investigation of friction losses in the ball screw system has four units: the rotating motion drive, the transmission itself, the axial loading and measuring systems.

The rotary motion unit (as seen in figure 2) is composed of a direct current engine 1 and two reduction gears. The first is a planetary gear which consists in a pinion 2, satellites 3, planetary carrier arm 4, planetary wheel 5 and the second one is a cylindrical worm gear 6-7.

The ball screw transmission illustrated in figure 3 is made of ball screw 1 on which two nuts are mounted fixed with a key joint in the cylinder 6. The ball screw is supported on two bearings 15, 17 and it receives turning movement through a rigid coupling 14.

The axial loading is accomplished hydrostatically by the circuit in figure 4 with the following elements: oil tank 1, pump 2, pressure valve 3, distributor 4, manometer 5, supply flange 6, cylinder 7, piston 8, nut 9, bolt 10.

Data acquisition was made with a VISHAY P3 bridge and the diagram of the transducers mounting is presented in figure 6.
3. EXPERIMENTAL RESULTS

The ball screw transmission used in the laboratory test rig has the following geometrical parameters: the outer diameter of the shaft 29 mm, the nut interior diameter 34 mm, the pitch 10 mm, the nominal diameter \( d_0 = 32 \text{ mm} \), the ball diameter \( d_w = 6 \text{ mm} \).

The friction torque was determined under the following two testing conditions: constant axial load and constant speed, both with low values.

The experimental testing was carried out on a region with better quality of both ball screw and nut surfaces, operating continuously. Data recorded by the bridge were inserted into an Excel program in order to determine some mean values for the measured force for each testing conditions.

3.1. Testing at Constant Oil Pressure

The constant pressure of oil is indicated by the manometer on hydraulic circuit which preloads the two nuts of the ball screw transmission. The experimental values for pressure were of 0.35, 0.55 and 9 MPa which induced the axial loading of the ball screw about 612 N, 963 N and 1575 N, respectively.

Figure 7 represents the recording data of the measured force in the case of minimum oil pressure and in figure 8 there is the diagram of the friction torque in terms of screw’s rotational speed.

It has been found that there is a zone at the beginning of the run with high values of measured force which decrease after a period maintained constant at significant lower values. Also the level of measured forces depends on the rotational speed of the ball screw.

The entire friction torque in ball screw was established in terms of measured force considering the location of the gauges on the deformed lamella.

3.2. Testing at Constant Rotational Speed

The screw’s rotational speed is established by measuring in terms of the driving d.c. motor power (speed may be adjusted varying the voltage of direct current source).

Three rotational speeds were used for the leading element of the ball screw transmission: 6.38, 7.4 and 11.11 rpm, corresponding to 15 V, 20 V and 25 V respective.

The experimental data in the case of 15 V constant electrical tension is illustrated in figure 9.

It can be observed that measured forces are increased by increasing pressure, therefore the friction torque has the linear dependence plotted in figure 10.
Fig. 10. Friction torque varying with axial loading.

Figures 11, a and b, show the development of the friction torque on a single nut during the entire screw’s run at constant pressure and constant rotational speed respective.

4. CONCLUSIONS

For the power screws in which high requirements to keep kinematical precision in terms of time, efficiency and reliability are imposed, the use of classical types with sliding between screw and nut becomes dissatisfactory.

Into industrial practice, the ball screws induce directly the performances of an equipment or a tool machine, being one of the most expensive unit. Wear and damage are greatly influenced by the ball screw operating conditions, lubrication, maintenance, cleanliness, etc.

This paper proposes an experimental equipment to measure the friction torque in a ball screw transmission. The testing was conducted for two important operating parameters: axial carrying load and rotational speed of the screw.

From the experimental results, a special behaviour was assessed at low speeds and loads: the more marked increasing of the friction torque in contrast with high speed.

Also the high values of the friction with the screw and nut rolling tracks may result from a low quality of the contact surfaces (both elements are quite worn) as well as lubrication regime (boundary almost dry). In the future, the experiments will be carried out be on the test rig with a new ball screw transmission.

To verify the experimental results the efficiency can be computed.

If it is neglected, the friction in a ball screw system to transfer axial load $F$, the active moment on the screw is:

$$M_{as} = F \cdot \frac{d_s \cdot p}{2 \cdot \pi \cdot d_0} \quad (1)$$

in which $d_s$ is the screw rolling track and ball contact point diameter given by the relation:

$$d_s = d_0 - d_c \cdot \cos \theta \quad (2)$$

with $\theta$ contact angle. After the computing, it results $d_s = 27.757 \, \text{mm}$.

Considering only the friction between balls and rolling tracks, the system efficiency is:

$$\eta = \frac{M_{as}}{M_{as} + M_r} \quad (3)$$

where $M_r$ denotes the enduring moment.

A calculus of the efficiency based on experimental data results in values for $\eta$ between 0.9 and 0.83 corresponding to those recommended in literature (taking into account that the effective ball screw transmission is worn).

REFERENCES