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SIMULATION INVESTIGATION OF BALL SCREWS FOR HIGH SPEED TRANSMITTING SYSTEMS

The ball screws are used in feed systems in CNC machines working with constantly higher cutting speed. When the speed of the balls reaches certain levels, the repetitive chock (impact forces) generated by the balls in the "transient" phase of the balls' motion and forces acting in the return channel cause damage to the recirculation ball system. A simulation of impact forces for variable length of transient phase in function of the rotational speed was made. The static axial rigidity is a very important factor to be considered in defining ball screws performance and exerts a major influence on positioning accuracy of NC machine tools. One of rigidity components is the balls/balltrack area rigidity, dependent on Hertz's deflections. However, due to machining inaccuracies, the ball screws properties (especially rigidity) are considerably lower than the theoretical ones. A coefficient known as the geometric correction factor makes it possible to predict the rigidity of the ball screw. A computer simulation of the influence of machining inaccuracies on contact deflections, rigidity and servo drive resonance was carried out. It was examined what rotational speeds set off the harmful phenomenon of the agreement of the frequency of the arrangement of the impact forces with the frequency of the servo drive system resonance. The presented method allows for the calculation of the impact forces, total rigidity and resonance of NC machine feed system, in the preliminary phase of technical project.

1. INTRODUCTION

Ball screws are used as elements of feed systems in CNC machinery and precision machine tools. Allowable rotational speed is determined by examining two aspects - $d{\cdot}n$ value (where: d - nominal diameter in mm, n - rotational speed of the screw shaft in rpm) and critical speed. Current standard $d{\cdot}n$ value are defined within the range 70 000 to 80 000, but for innovative ball screws could reach value of 150 000 and feed rate has increased to 180 m/min [1]. Since the screw shaft is usually fixed on its two ends, its begin to vibrate due to resonance. This vibration could develop into vibration or noise in the entire machine. The calculation for critical speed was presented in [5].

The $d{\cdot}n$ value express the velocity of the balls moving in circulation circuit. During the operation of a ball screw the balls are subject to the load variables .The full cycle of the

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circulation of each ball consists of the following three phases: the working phase (the loaded balls roll between coils of the screw and the nut), the transient phase (the balls come in to the working circulation subjected to the growing load or they leave it and the load diminishes to zero), and the phase of the return movement - the non-loaded balls come back to the beginning of the working circulation. As the driven shaft in the mechanism operates at high rotating speed, in transient phase appear the impact forces which are considerably larger the than forces N_φ which come into being during quasi static entry in load (Fig. 1). In case when the frequency of appearing of the impact forces (definite with angle θ) agrees with the frequency of the vibration resonance of the drive the growth of vibration and the noise of mechanism will happen and it may also cause damage to the ball return elements of the ball screws [1]. Therefore it necessities the examining of what turns can set this phenomenon.

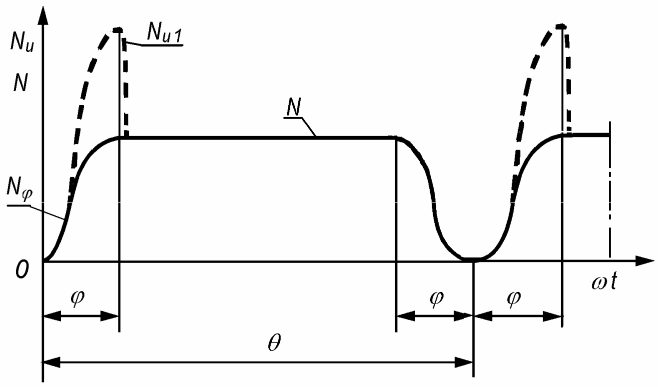

Fig. 1. The course of normal forces and impact forces in the function of screw rotation angle for a full cycle of one ball circulation: N – normal force acting on the ball, N_{u1} – impact force, θ – screw rotation angle corresponding to a ball rolling a distance equal to its diameter

The frequency of the balls entering in loading can be counted as follows:

$$f_\theta = \frac{60n}{\theta} \tag{1}$$

where: n – rotating speed [rpm], θ – screw rotation angle corresponding to a ball rolling a distance equal to its diameter [4].

To calculate the resonance vibration we need the values of the rigidity of the drive cinematic elements.

2. IMPACT FORCES SIMULATION

Entry of the ball in the loading state for higher velocity can be compared to the system of two bodies, which is the ball and nut. Fig. 2a shows the simplified mathematical model of the transit zone described in [4]. Angle φ determines the position of the ball and the acting direction of the variable force N. Relationship between total contact deflections of the system ball/screw raceway and the ball/nut raceway (marked as w) and the normal force N acting on the ball can be determined by Hertz's formula:

$$N_\varphi = f(w),$$

$$\text{for } \varphi = 0,\ N_\varphi = N = k \cdot w^{3/2} \tag{2}$$

where: w – hertzian deflection in the ball/balltrack area, k – rigidity characteristic [6].

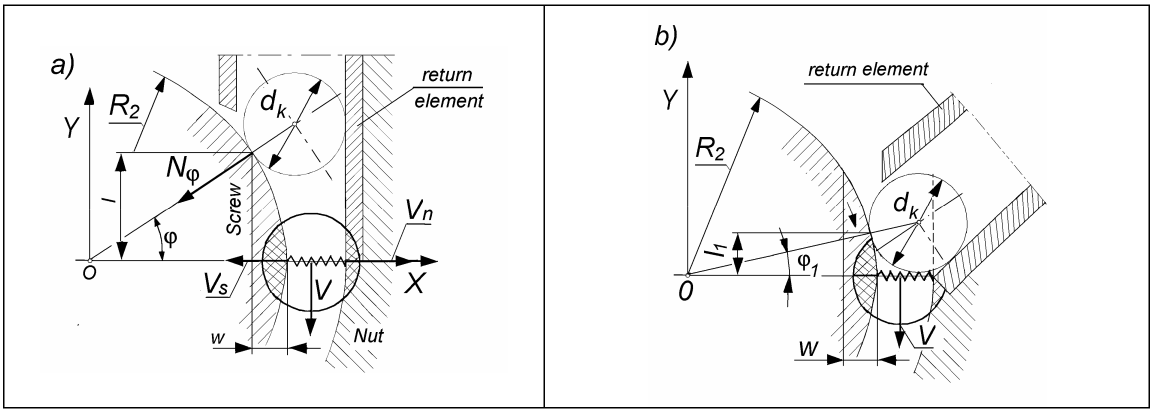

Fig. 2. Mathematical model of transition zone:
a) Ball's entrance to circulation circuit (return element tangential to the helix), b) defective (real) position of return element: d_k – ball diameter, R_2 – one of the main radiuses of curvature (proportional to ball diameter), φ, φ_1, – angles determines the position of the ball, l , l_1 – length of transition zone, $V_{s,n}$ – velocity of contact deflection properly for screw shaft and nut, $V_{s,n}$ – velocity of contact deflection, V – ball velocity

For simplification, it was assumed that the total elastic deflection w is realized only by contact deflection of the ball treated as a spring without mass.

The detailed description of the transient phase of the dynamic phenomena is given in [4]. This analysis is based on the elastic impact of the spherical bodies (Hertz theory) given in i.e., in [1]. According to this theory the contact deflections can be described as:

$$w = \left[\frac{5V_0^2 m_s m_n}{4k(m_s + m_n)} \right]^{2/5} \tag{3}$$

where: V_o – velocity of ball deflection, $m_{s,n}$ – mass of screw and nut respectively.

In the discussed case the speed of the elastic ball deflection V_0 is a vector perpendicular to the surface of the ball and at the same time perpendicular to the component of the vector of center ball speed V (moving tangent to helix).

From the condition of the time identity for the process in the transient phase we can calculate the time of impact:

$$t_u = \frac{l}{V} = \frac{w}{V_0}, \qquad \text{and:} \quad V_0 = \frac{w \cdot V}{l} \tag{4}$$

where: $l = R_2 \cdot sin\varphi$ – for small values of the angle φ.

In the central moment of the impact, which is at maximum approach (for $\varphi=0$) you will find the maximum force of impact and according to (2):

$$N_{u1}=\left(\frac{5k^{2/3}m_s\cdot m_n}{4(m_s+m_n)}\right)^{3/5}\left(\frac{\pi\cdot D_t\cdot n\cdot w\cdot sin\alpha}{120R_2\cdot sin\varphi}\right)^{6/5} \tag{5}$$

where: α – contact angle, D_t – diameter determined the balls contact points with the screw raceway.

As a result of deviations of position and the mounting and machining errors of the return element, the transient phase undergoes shortening (Fig. 2b). Because $\varphi < \varphi_1$, (as well as $l_1 < l$) , then the time of impact according to (4) diminishes and impact forces according to (5) grow.

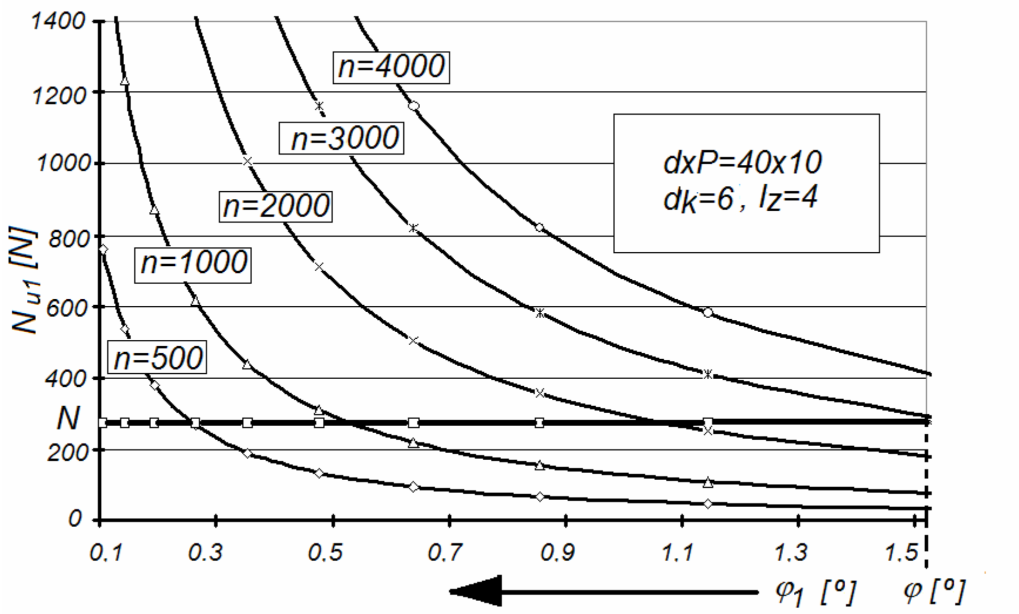

Fig. 3. The relationship between the impact forces N_{u1} and the rotational speed n [rpm]: φ, φ_1 – angles determines the ball position in the transition zone: d – screw shaft diameter, P – lead, d_k – ball diameter, I_z – number of loaded turns, n – rotational speed [rpm]

A simulation of impact forces for variable length of transient phase in function of the rotational speed was made. The results of the simulation are shown in Fig. 3 for ball screw with shaft diameter d=40 mm, lead P=10 mm and ball diameter d_k=6 mm. For calculation we took the contact deflection w equals 0.45 Ca (dynamic axial load ratings), mass of screw that was 1000 mm long and mass of a nut of a length equal to for turns of the thread (I_z = 4). The results of simulation show, that the value of impact forces for higher rotational speeds and shortening of the transient zone, are considerably higher than the normal forces N.

The research proves that impact forces increase not only with rotational velocity increase, but also with the increase of the ball diameters (Fig. 4).

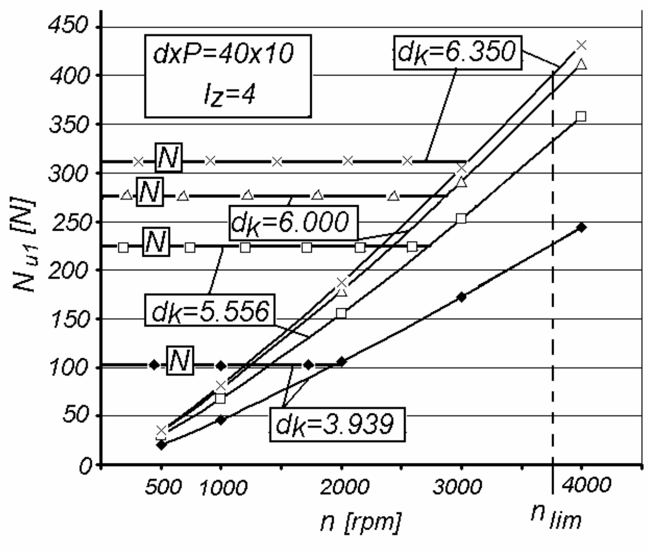

Fig. 4. The relationship between the impact forces N_{u1} and the rotational speed n [rpm] for variable ball screw diameter d_k; N – the normal forces, n_{lim} - upper limit of rotational speed

The dashed line n_{lim} in Fig. 4 shows the upper limit of rotational speed is obtained using the equation:

$$n_{\lim} = \frac{150000}{d}$$

3. STATIC AXIAL RIGIDITY SIMULATION

The static axial rigidity of the ball screws exerts a major influence on positioning accuracy of NC machine tools. The Fig 5 presents the helpful model for calculation of the rigidity elements entering in composition of chain of cinematic feed system. The rigidity R constitutes the resistance to deformation and denotes the force ΔF which is requires to effect a component deflection Δl in the axial direction on load application:

$$R = \frac{\Delta F}{\Delta l} \tag{6}$$

where: ΔF – force increase [N], Δl – deflection [μm].

The total axial rigidity of the driving system is obtained using the equation [7]:

$$\frac{1}{R} = \frac{1}{R_b} + \frac{1}{R_m} + \frac{1}{R_{bs}} \tag{7}$$

where: R_b – rigidity of the support bearing, R_m – rigidity of the nut bracket (table montage), R_{bs} – overall rigidity of the ball screw.

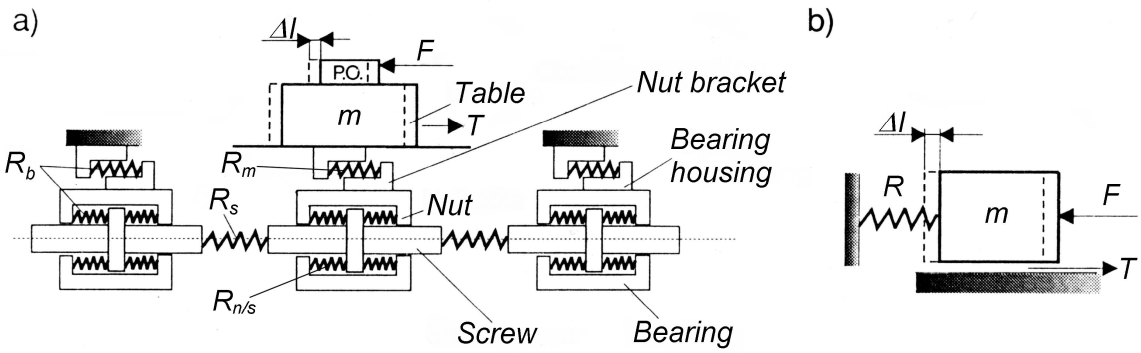

Fig. 5. Calculating model of ball screw axial rigidity (a) end substitute model (b) [2]: R_b –rigidity of the support bearing, R_m – rigidity of the nut bracket (table montage), R_s –rigidity of the screw shaft, $R_{n/s}$ – rigidity of nut body and screw shaft

The axial static overall rigidity of the ball screw is arrived at by adding the pertinent rigidity values of the components [6]:

$$\frac{1}{R_{bs}} = \frac{1}{R_s} + \frac{1}{R_{nu,sf}} \tag{8}$$

where: R_s – rigidity of the screw shaft, $R_{nu,sf}$ – rigidity of the ball nut unit takes into account the machining inaccuracies.

The axial rigidity of the screw shaft varies depending on the method for mounting the shaft. The rigidity of the ball nut unit is obtained from the following equation [6]:

$$\frac{1}{R_{nu,sf}} = \frac{1}{R_{n/s}} + \frac{1}{R_{b/t,sf}} \tag{9}$$

where: $R_{n/s}$ – axial rigidity of nut body and screw shaft, $R_{b/t,sf}$ – axial rigidity of the ball/ball track area calculated with the corresponding geometry correction factor.

The calculation of balls/balltrack area rigidity is the main problem because in a result of machining inaccuracies this rigidity is considerably smaller from theoretical. In work [4] the geometry correction factor was introduced in order to estimate the influence of machining inaccuracies and dimensions of ball screw on lowering the level of rigidity:

$$R_{b/t,sf} = R_{b/t} \cdot s_f \tag{10}$$

where: s_f - geometry correction factor:

$$s_f = \frac{\Delta l_{b/t}}{\Delta l_{b/t,sf}} \tag{11}$$

where: $\Delta l_{b/t}$ – the axial deflection due to Hertz stress , $\Delta l_{b/t,sf}$ – deflection due to machining inaccuracies.

The computer simulations, described in [4], made for 1st standard tolerance grade (according to ISO [6]) and for confidence level Cl=0.98 show that geometry correction factor depend non only on accuracy grade but also on geometrical parameters of the ball screw thread (Fig. 6). The investigation was made for ball screw d=40 mm, I_z=4, length of screw l_s=1000 mm (ball screw shaft mounting at both ends), variable lead P and balls diameters d_k.

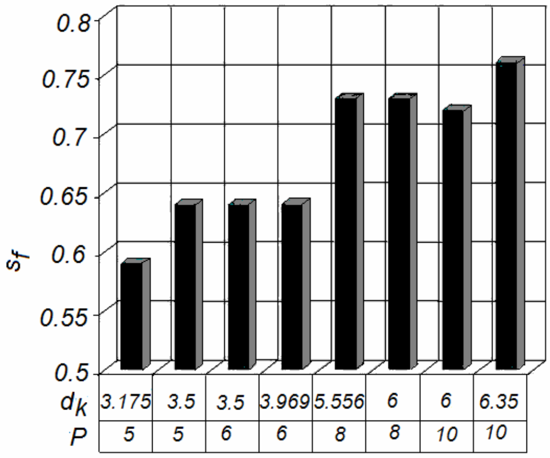

Fig. 6. The relationship between geometry correction factor s_f and geometrical parameters of ball screw for d=40 (simulation for 1st standard tolerance grade for confidence level Cl=0.98): d_k – ball diameter, P – lead

The table 1 and Fig. 7 shows the results of rigidity components calculation for ball screw d=40, lead P=10, and ball diameter d_k=6. (Fig. 7 does not show the value of $R_{n/s}$ – it is too high for this figure). The axial rigidity of the support bearing (depending on the screw shaft diameter) and the nut bracket are taken from the ball screw catalog [7].

Table 1. Results of rigidity components [N/μm] calculation for screw shaft diameter d=40 mm, lead P =10 mm and balls diameters d_k=6 mm: s_f - geometry correction factor, $R_{b/t}$– axial rigidity of the ball/ball track area (theoretical), $R_{b/t,sf}$ – axial rigidity of the ball/ball track area (simulated), $R_{n/s}$ – axial rigidity of nut body and screw shaft, R_{nu} – rigidity of the ball nut unit (theoretical), $R_{nu,sf}$ – rigidity of the ball nut unit (simulated), R_{bs} – overall rigidity of the ball screw, total axial rigidity of the driving system, R - total rigidity

d x P	d_k	s_f	$R_{b/t}$	$R_{b/t,sf}$	$R_{n/s}$	R_{nu}	$R_{nu,sf}$	R_{bs}	R
40x10	6	0.72	785	565	15540	747	545	323	200

After calculation of total rigidity R it becomes possible to calculate the resonance frequency of team the table of machine tool - the ball screw [2]:

$$f_r = \frac{1}{2\pi}\left(\frac{R \cdot 10^6}{m}\right)^{1/2} \tag{12}$$

where: f_r – resonance of the table-ball screw ensemble [Hz], R - total rigidity [N/μm], m – payload mass (of the work and table) [kg].

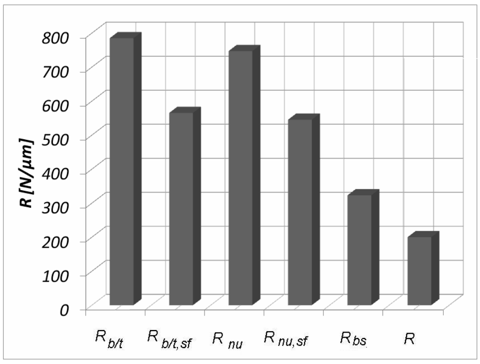

Fig. 7. The rigidity components calculation for ball screw d=40, lead P=10, and ball diameter d_k=6

The table 2 shows the results of total rigidity R and resonance f_r calculation (payload mass m=1000 kg) for variable lead P and balls diameters d_k. The criterion of the resonance is one of the most important criteria of projecting the ball screw drives. It is dependent on the motor driving type [2].

Table 2. Results of total rigidity R and resonance f_r calculation for screw shaft diameter d=40 mm, variable lead P and balls diameters d_k

d_k	3.175	3.5	3.5	3.968	5.556	6	6	6.35
P	5	5	6	6	8	8	10	10
R [N/µm]	167	174	189	190	200	200	200	204
f_r [Hz]	92,3	94	98	98	101	101	101	102

The maximum angular speed of the screw should be considerably smaller from critical speed, near which transverse resonance vibration can happen. Moreover one should check if there is no phenomenon of agreement of the resonance vibrations with the frequency of the balls entering the load, that is:

$$f_r = f_\theta$$

where: f_θ - frequency acc. to (1).

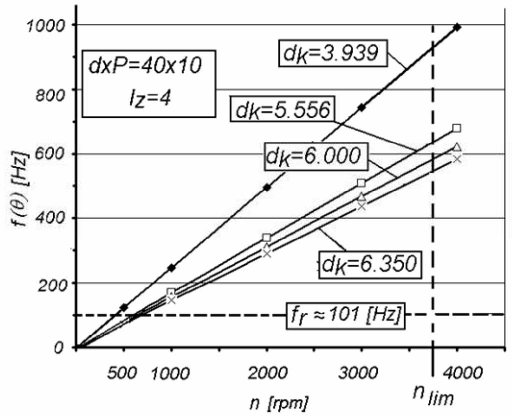

Fig. 8. The relationship between frequency of the balls entering the load and rotational speed n for variable balls diameters d_k: n_{lim} - upper limit of rotational speed

The Fig. 8 shows the results of frequency calculation. For investigation made in this paper the value of drive system resonance (~101 Hz) is considerably less then the frequency of balls entering the load for upper limit of rotational speed (for $d{\cdot}n$ =150 000).

4. CONCUSIONS

For certain levels of high rotational speed the repetitive shocks caused by the balls entering into load (impact forces) make growth of vibration and noise. The frequency of impact forces can not be equal to the resonance frequency of driving system because it could cause a incorrect work of the system and damage to the ball screw mechanism. The investigations show, that for definite diameter of the screw and the definite turns the frequency of impact forces can change by selection of diameter of balls.

During preliminary phase of design it is hardest to estimate the balls/balltrack area rigidity because it is dependent not only on the value of axial forces (the hertzian deflection) but also on the machining inaccuracies and on the dimensions of ball thread. The proposed geometric correction factor appointed by simulation investigations enables the calculation of the hertzian deflection and then the rigidity of the drive system in preliminary phase of the technical project.

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