ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1217239

TESTING OF EFFICIENCY AND DYNAMIC EFFECTS OF A DRIVE SYSTEM WITH A LEAD SCREW

Filip Lisowski

Cracow University of Technology
Department of Mechanical Engineering
Jana Pawła II 37, 31-864 Cracow, Poland
tel.: +48 12 6383334, fax: +48 12 6383360
e-mail: flisow@mech.pk.edu.pl

Abstract

The paper presents experimental studies of a drive system with a lead screw. The concept and the construction of the test stand, its parameters as well as the applied measuring apparatus and the scope of tests are described. Furthermore, a measuring system and a methodology of determining the loads on the nut of the lead screw, based on the strain gauge measurements, were developed. A special nut bracket with specimens for the strain gauge measurement was designed. Since the specimens are simultaneously subjected to tension, two plane bending and non–free torsion, a coefficient essential for determining the torque on the nut of the lead screw was estimated based on the finite element analysis of the nut bracket under complex load condition. The first part of experimental studies includes the determination of the efficiency of the lead screw and the efficiency of the typical drive system with the lead screw. The impact of the load level and the constant resistance in the system are included in the research. The second part of the study concerns the determination of dynamic load coefficients for various operating parameters. The efficiency of the lead screw and the efficiency of a typical drive system with the lead screw as well as the dynamic factors at start–up and braking were determined based on experimental tests referred to a ball screw.

Keywords: *lead screws, efficiency, dynamic effects*

1. Introduction

Lead screws are mechanisms used for the conversion of rotary motion to linear motion or in the other way round. A screw shaft, a nut and rolling elements, which can be balls (ball screws) or rollers (roller screws) are the main elements of the mechanism. The use of rolling elements allows replacing the sliding friction between bearing surfaces of the screw and the nut with the rolling friction. Consequently, the efficiency, which depends also on the helix angle and load level, can reach more than 90% [12]. High rigidity, durability similar to the durability of rolling bearings and high load carrying capacity can be indicated as the other advantages of the mechanism. Therefore, lead screws are used in drives of numerically controlled machine tools, space equipment, industrial robots and manipulators. Lead screws are also used in the mechanisms of working machines, measuring devices as well as plane drives.

In the recent years, authors of several publications, who referred to ball screws, considered both theoretical and experimental problems. The paper [3] provides an approach to monitor the health status of the ball screw. The authors examined the relationship between the ball screw preload variation and the detected vibration signals using spectrum analysis of the processed signals. In the paper [4] the microscopic behaviour of a preloaded ball screw was investigated based on experimental results and simulation. The authors demonstrated experimentally that the nonlinear microscopic behaviour of a positioning mechanism driven by a ball screw is also affected by the ball bearings supporting the screw shaft. The paper [10] presents analysis of forced vibration of the drive system with lead screw. The authors paid attention to the influence of damping the coupling and bearings on vibration amplitude values. A theoretical analysis of ball screw kinematics taking into account the lubrication is presented in the study [18].

Also dynamic effects were concerned by the several authors of theoretical studies. In the paper [15] a relation between impact force and design parameters of transient phase in function of rotational speed was developed. The author examined a phenomenon of a drive system resonance frequency. A methodology for stress state limit recovering in return system is developed in paper [1]. The author of the article [5] analysed the occurrence of the periodic force variations and explained this phenomenon. The authors of the publication [16] presented a high-frequency dynamic model of a ball screw drive. Using delivered model, the axial and angular components of mode function were studied to determine the degree of coupling between them.

The authors of recent publications related to roller screws have focused mainly on the theoretical problems concerning principles of operation, kinematics and efficiency, load distribution and contact analysis or dynamics. In the paper [6] the authors investigated capabilities and limitations of roller-screw and explained the slip phenomenon. The presented results were confirmed by experiments. The article [17] studied the efficiency and kinematics of planetary roller screw mechanism. In the paper [7] the contact between surfaces transferring load is investigated. In the next article [9], the same authors used direct stiffness method to build a stiffness model of the roller screw mechanism. The model allows to predict overall stiffness of the mechanism and also calculate the load distribution. In the paper [8], the dynamic equations of motion for the planetary roller screw are developed. The equations of motion are derived using Lagrange's Method with the assumption of viscous friction. In the paper [13] the authors proposed computational model of the load distribution for an arbitrary number of rollers, which can be used for the preliminary design. The paper [11] presents a model for determining the load distribution taking into consideration the contact deformation of threads as well as deformations of the screw and the nut cores.

The few authors of the earlier papers focused on testing problems. The paper [14] considered dynamic load test under large loads, which demonstrated that roller screw mechanism could be damaged by dynamic loading with load magnitudes that are well for the static load range. The article [2] presents the results of the lifetime tests of planetary roller screw under oscillatory motion. The authors investigated the effect of the type of lubrication from solid to liquid under a large number of oscillating loads cycles.

2. The design and construction of the test stand

The test stand for lead screws is designed to test all types of the lead screws (ball screws and roller screws). The concept of the stand is shown in Fig. 1. A structure consists of two frames: a supporting frame (1) and a bearing frame (2). The tested screw (10) is mounted to the bearing frame using bearings. For the drive, side this is a flanged thrust bearing (7) whereas the other end of the screw shaft is supported by a self-aligning bearing with an adapter sleeve (11). The load, up to 20 kN, is performed by a hydraulic system which consists of two hydraulic actuators (13) mounted on spherical joins and a hydraulic power supply (3). The load is transferred to the nut of the leading screw through a carriage (9), moving on linear guides (12). The nut is connected to the carriage by nut's bracket (8) intended for strain gauge measurements. A shaft of the screw is driven by a gear motor (4) controlled by an inverter. A rotating torque sensor (5) is placed between the motor gear and the screw shaft. Furthermore, a capacity displacement sensor (6) is mounted to the supporting frame to measure the displacement of the thrust bearing supporting plate. The axial force in the screw can be determined as a function of plate’s displacement. This requires scaling of the measuring system when the rotation of the screw is fixed.

3. Actuators and measuring devices

The constructed test stand for lead screws including actuators and measuring devices is shown in Fig. 2. The measurements of torque and rotational speed of the screw shaft as well as displacement of supporting plate of the thrust bearing are performed by rotational torque sensor (MT100) and capacity displacement sensor (CP1-3). In addition, a wire encoder was added to the measuring system in order to determine the position of the carriage. Measurements of these values can be recorded by the digital meter, working with computer software. Deformations of the nut's bracket were measured using a strain gauge bridge with six rosette sensors. Parameters of the applied sensors are summarized in Tab. 1, whereas parameters of drive devices in Tab. 2.

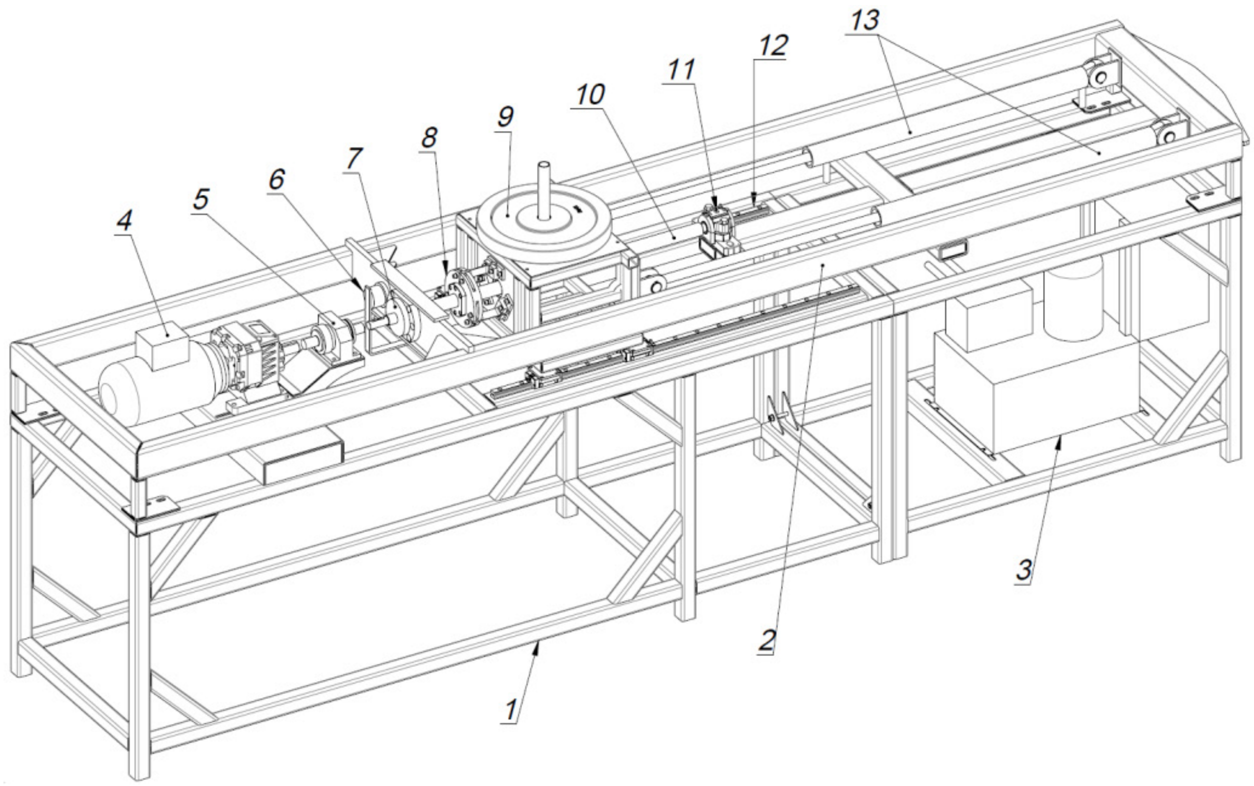

Fig. 1. A concept of the test stand for lead screws

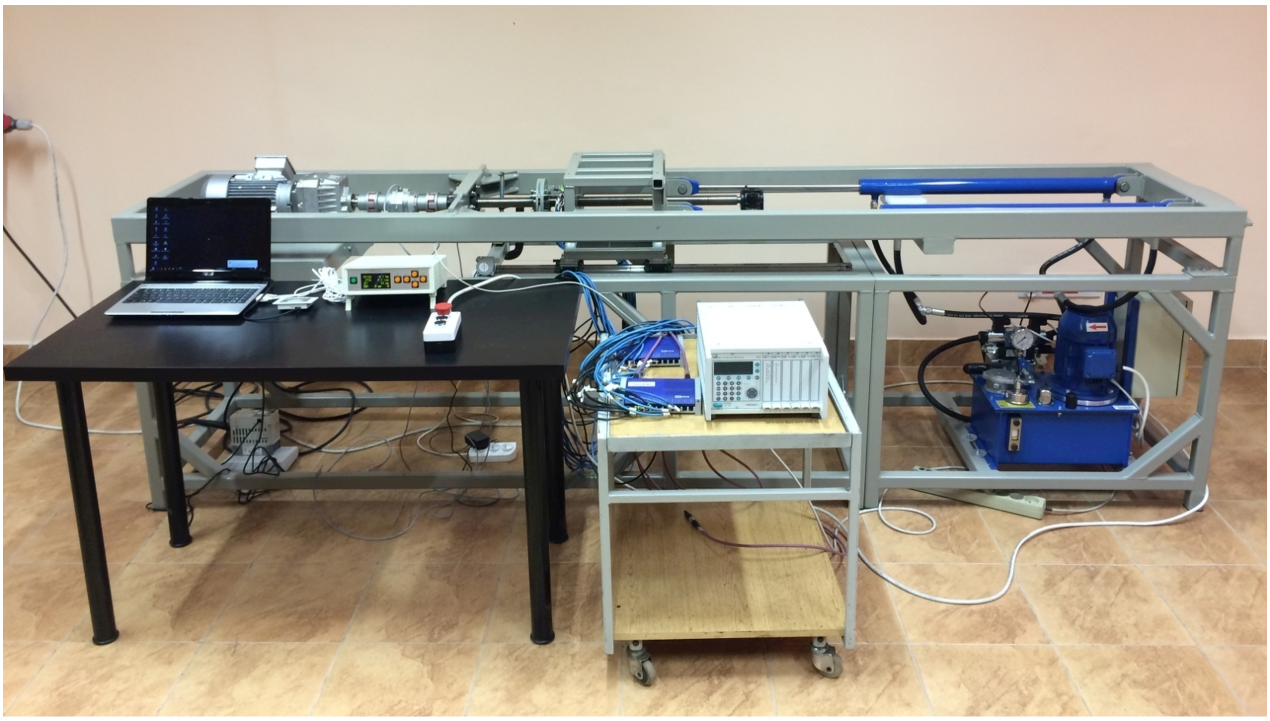

Fig. 2. A test stand for lead screws

Tab. 1. Measuring sensors

Sensor	Measured value	Sensor parameters
Rotating torque sensor (MT100)	Torque and rotational speed at shaft of motor gear	Torque measuring range: ±100 Nm Accuracy: 0.2% Rotational speed measuring range : ±200 rpm Accuracy: ±1 rpm
Capacity displacement sensor (CP1-3)	Displacement of the thrust bearing supporting plate	Measuring range: 1-3 mm Accuracy: ±0.3 %
Strain gauge rosette (TFr8/120)	Normal and angular strain of plates in nut's bracket	Resistance: 120 ±0.2 Ω
Wire encoder	Carriage displacement	Accuracy: ±2 mm

Tab. 2. Drive devices

Gear motor	Motor: 3-phase 0.75kW – 1400 rpm Output revolutions: n2 = 53.2 rpm Output torque: M2 = 150 Nm
Hydraulic power supply	Nominal oil tank capacity: V = 10 dm^3 Motor power: N = 1.5 kW Pump capacity: Q = 1.6 cm^3/rot
Hydraulic actuators	Ø40/22, lead: 640mm

4. Determination of the torque and the axial force on the nut of the lead screw based on strain gauges measurements

The axial force and torque on the nut of the lead screw can be determined on the basis of normal (ε_x) and shear (γ_{xy}) strains, obtained from the strain gauge measurements (Fig. 3). Strain gauge rosettes are placed on both sides of each of three specimens of the nut bracket (Fig. 4). The normal strain (ε_x) is obtained directly from the measurements. In contrast, shear strain on surfaces of specimens (γ_{xy}) is determined on the basis of strain gauge measurements in the other directions of the strain gauges rosettes, which are 120° and 240° (Fig. 5). A relationship between the strain, measured in an arbitrary direction, specified by angle α, and components of strain in *xy* local system is given by eq. 1.

$$\varepsilon_\alpha = \frac{\varepsilon_x + \varepsilon_y}{2} + \frac{\varepsilon_x - \varepsilon_y}{2}\cos(2\alpha) + \frac{\gamma_{xy}}{2}\sin(2\alpha). \tag{1}$$

Shear strain (τ_{xy}) can be obtained based on the system of equations related to direcions specified by angles 120° and 240°:

$$\gamma_{xy} = \frac{2}{\sqrt{3}}(\varepsilon_{240^\circ} - \varepsilon_{120^\circ}). \tag{2}$$

The torque and an axial force on the nut of lead screw can be determined with the following formulas:

$$T = C_1 \cdot \tau_{xy}, \quad C_1 = \eta_1 a b^2, \quad \tau_{xy} = \gamma_{xy}\frac{E}{2(1+\nu)}, \tag{3}$$

$$Q = C_2 \cdot \sigma_x, \quad C_2 = ab, \quad \sigma_x = E\varepsilon_x, \tag{4}$$

where:
T – torque on nut,
τ_{xy} – shear stress,

C_1, C_2 – constants,
a, b – dimensions of plate cross section (a – the longer side of rectangle),
η_1 – coefficient depending on the size of plates,
E – Young's modulus,
ν – Poisson's ratio.

The specimens of the nut bracket with the strain gauge rosettes are rigidly constrained. The first ends of each specimens are mounted in the slot of stiffness plate, screwed to the nut flange. Whereas the second ends of specimens are mounted in the slots of holders, placed on the front side of the carriage. During the operation of the lead screw, which is moving the carriage, loaded by the hydraulic system, the specimens in the nut bracket undergo tension and two-plane bending. In view of the above, the coefficient η_1 must be estimated taking into account the complex state of load.

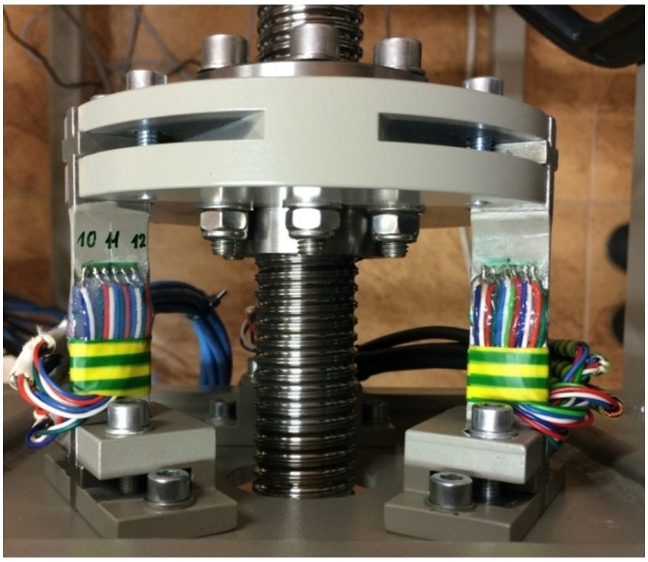

Fig. 3. Nut bracket for strain gauge measurements of axial force and torque on a nut of the lead screw

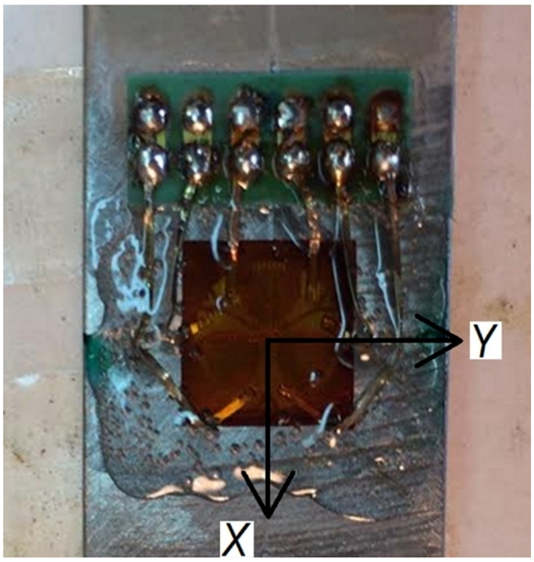

Fig. 4. Orientation of strain gauge rosette

5. Determination of the efficiency of a drive system with the lead screw

The efficiency of the drive system with the lead screw (ball screw) was determined for the load case, where the specimens of the nut bracket, and thus the screw shaft were tensioned. Assuming that the power loss in the system is proportional to the actual transmitted power and that the a constant coefficient of power loss exists, the efficiency of the drive system, for the direct path of power is given by eq. 5. The efficiency for the reverse path of power is given by eq. 6

$$\eta_i = \frac{1}{(1/\eta) - s + (s/\alpha_i)}, \tag{5}$$

$$\eta_r = 2 - (1/\eta), \tag{6}$$

where:
η – efficiency of lead screw under nominal load,
s – coefficient of constant resistance,
α_i – coefficient of transmitted power.

The coefficient of transmitted power is the ratio between the actual load and the nominal load of the system as given by. eq. 7.

$$\alpha_i = \frac{Q_i}{Q_n}. \tag{7}$$

For the load case, in which the screw shaft is tensioned, the efficiency of the drive system with the lead screw, under nominal load, can be calculated using eq. 8.

$$\eta = \frac{tg(\gamma)}{tg(\gamma+\rho')} = \frac{L_s / \pi\, d_p}{2T / Q\, d_p} = \frac{P_{MW} L_s}{2\pi\,(T_{MGC} + T_b)}, \tag{8}$$

where:
γ – helix angle,
ρ' – angle of friction,
L_s – lead of the screw thread,
d_p – pitch diameter of screw,
T – input torque on the shaft,
P_{MW} – axial load measured on the carrying frame,
T_{MGC} – torque on the nut determined from strain gauge measurements,
T_b – frictional moment in a thrust bearing

In order to determine the torque and the axial force for various load levels, experimental tests of typical drive system with lead screw were conducted. A ball screw with pitch diameter $d_p = 35$ mm and thread pitch p = 5 mm was used. The axial load range from 4.38 to 20.16 kN was accepted. That corresponds to the oil pressure in the hydraulic system from 25 to 115 bar.

The efficiency of the drive system with the lead screw and the efficiency of the ball screw was determined for two values of the carriage speed: 2 and 9 mm/s.

6. Determination of the efficiency of the lead screw

Accepting the torque and the axial force on the nut, obtained from the strain gauge measurements, the efficiency of the ball screw can be determined using eq. 9.

$$\eta = \frac{tg(\gamma)}{tg(\gamma+\rho')} = \frac{L_s / \pi\, d_p}{2T_{MGC} / Q_{MGC}\, d_p} = \frac{P_{MGC} L_s}{2\pi\, T_{MGC}}, \tag{9}$$

where:
Q_{MGC}, T_{MGC} – axial load and torque on the nut determined from strain gauge measurements.

Figure 5 presents the of efficiency of the drive system with the lead screw and the efficiency of the ball screw for the carriage speed 2 mm/s and 9 mm/s.

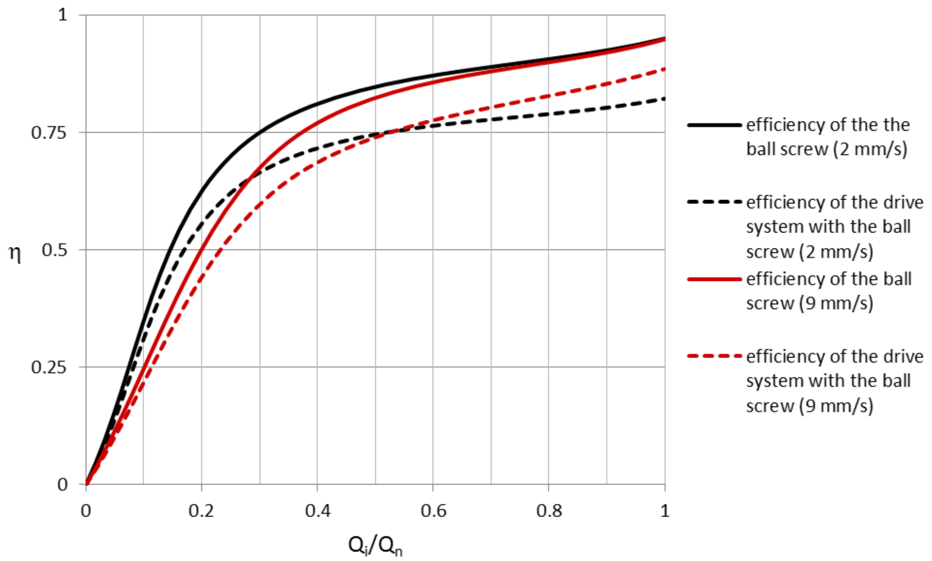

Fig. 5. The efficiency of the drive system and the lead screw

It can be noticed that for low load level (about 10-25% of the nominal load) the efficiency of the drive system and lead screw is $\eta < 0.5$. In such a situation, in reference to eq. 6, the drive system or lead screw is self-locking.

7. Determination of the dynamic factors for start-up and braking

To determine dynamic factors for start-up and braking of the drive system with the lead, four levels of axial load: {4.38, 10.52, 15.78, 20.16} kN were accepted. That corresponds to the pressure in the hydraulic system: {25, 60, 90, 115} bar. The experimental tests were carried out for three value of the carriage speed {2, 4.5, 9} mm/s. Dynamic factors were determined using the eq. 10.

$$K_d = \frac{Q_{\max}}{Q_{ave}}, \tag{10}$$

where:

$Q_{\max}$, Q_{ave} – maximum and average values of the axial load measured on the thrust bearing supporting plate.

Figure 6 presents the dynamic factors for start-up, whereas Fig. 7 for braking of the drive system with the ball screw. It can be noticed, that dynamic factors decreases with increasing the axial load. For the high load level, close to the nominal load of the system, both for the start-up and braking, dynamic factors have constant value, approximately $K_d = 1.17$.

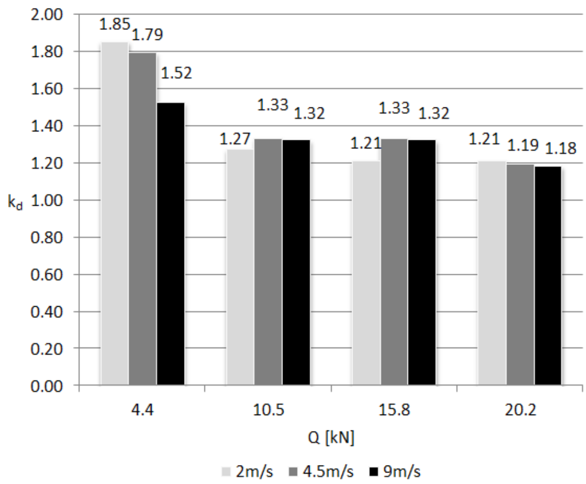

Fig. 6. Dynamic factors for start-up of the drive system with the lead screw for various load and carriage speed

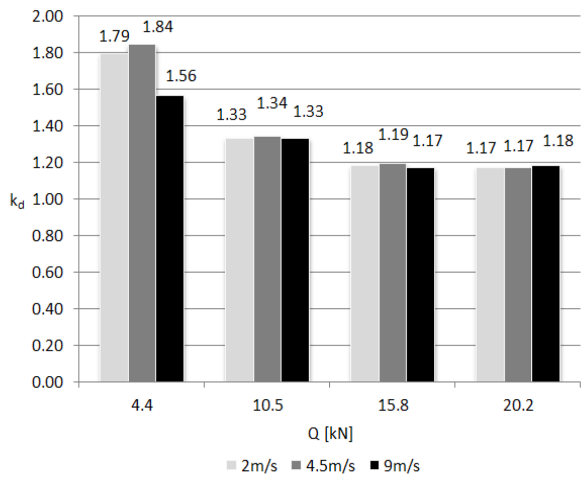

Fig. 7. Dynamic factors for braking of the drive system with the lead screw for various load and carriage speed

8. Conclusion

The analysis of the static and dynamic loads of the drive system with the lead screw is of a fundamental importance to determine the load capacity and durability of lead screws. The designed and constructed test stand allows testing the efficiency of the drive system with the lead screw and the efficiency of lead screws (ball screws or roller screws) in any variant of design. It is also possible to investigate the dynamic effects and determine dynamic factors for start-up and braking of the drive system.

The efficiency of the drive system with the lead screw and the efficiency of the ball screw were determined based on the experimental tests. It was found that for low load level the efficiency is lower than 0.5. Therefore, at the reverse power flow the system is self-locking. This means that the external load is not able to force the linear motion of the nut. It can be important notification, since actually the producers of lead screws do not provide that operating parameter, which is the minimum load at which the specific lead screw is self-locking.

References

[1] Braccesi, C., Landi, L., *A general elastic-plastic approach to impact analysis for stress state limit evaluation in ball screw bearings return system*, International Journal of Impact Engineering, Vol. 34, pp. 1272-1285, 2007.

[2] Falkner, M, Nitschko, T, Supper, L, Traxler, G, Zemann, J.V., Roberts. E.W., *Roller screw lifetime under oscillatory motion: from dry to liquid lubrication*. In: Proceedings of the 10th European Space Mechanisms and Tribology Symposium (ESMATS '03), pp. 297-301, San Sebastian, Spain 2003.

[3] Feng, G.-H., Pan, Y.-L., *Investigatnion of ball screw preload variation based on dynamic modeling of a preload adjustable feed-drive system and spectrum analysis of ball-nuts sensed vibration signals*, International Journal of Machine Tools & Manufacture, Vol. 52, pp. 85-96, 2012.

[4] Fukada, S, Fang, B., Shigeno, A., *Experimental analysis and simulation of nonlinear microscopic behaviour of ball screw mechanism for ultra-precision positioning*, Precision Engineering, Vol. 35, pp. 650-668, 2012.

[5] Frey, S, Walther, M., Verl, A., *Periodic variation of preloading in ball screws,* Production Engineering Research and Development, Vol. 4, pp. 261-267, 2010.

[6] Hojat, Y., Agheli, M., *A comprehensive study on capabilities and limitations of roller screw with emphasis on slip tendency*, Mechanism and Machine Theory, Vol. 44, pp. 1887-1899, 2009.

[7] Jones, M.H., Velinsky, S.A., *Contact kinematics in the roller screw mechanism*, Journal of Mechanical Design, Vol. 135, 2013.

[8] Jones, M.H., Velinsky, S.A., *Dynamics and efficiency of the planetary roller screw mechanism*, Journal of Mechanisms and Robotics, 2014.

[9] Jones, M.H., Velinsky S.A., *Stiffness of the roller screw mechanism by the direct method*, Mechanics Based Design of Structures and Machines, Vol. 42, pp. 17-34, 2014.

[10] Varansi, K.K., Nayfeh, S.A., *The dynamic of lead-screw drivers: low-order modeling and experiments*, Journal of Dynamic Systems, Measurement and Control, Vol. 126, pp. 388-396, 2004.

[11] Lisowski, F., *The analysis of displacements and the load distribution between elements in a planetary roller screw*, Applied Mechanics and Materials, Vol. 680, pp. 361-364, 2014.

[12] Oberg, E., *Machinery's Handbook*, 29th ed. Industrial Press, 2012.

[13] Ryś, J., Lisowski, F., *The computational model of the load distribution between elements in planetary roller screw*, Journal of Theoretical and Applied Mechanics, Vol. 52, pp. 669-705, 2014.

[14] Schinstock, D.E., Haskew, T.A., *Dynamic load testing of roller screw EMA's*, in: Proceedings of the 31st Intersociety Energy Conversion Engineering Conference (IECEC '96), Vol. 1, pp. 221-226, Washington, USA 1996.

[15] Sobolewski, J.Z., *Vibration of the ball screw driver*, Engineering Failure Analysis, Vol. 24, pp. 1-8, 2012.

[16] Vicente, D.A., Hecker, R.L., Villegas, F.J., Flores, G.M., *Modeling and vibration mode analysis of a ball screw drive,* The International Journal of Advanced Manufacturing Technology, Vol. 58, pp. 257-265, 2012.

[17] Velinsky, S.A, Chu, B., Lasky, T.A., *Kinematics and efficiency analysis of the planetary roller screw mechanism*, Journal of Mechanical Design Vol. 131, 2009.

[18] Wei, C.C, Lin, J.F., Horng, J.-H., *Analysis of a ball screw with a preload and lubrication*, Tribology International, Vol. 42, pp. 1816-1831, 2009.