A Study on the Ball Screw Friction Torque

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Abstract

Tato studie se zaměřuje na zlepšení současných modelů tření v kuličkovém šroubu za účelem jejich efektivnějšího využití v Ecodesignu obráběcích strojů a jejich enegretické optimalizace. Modely použité v této studii vycházely z modelů tření v kuličkových ložiskách (Harris) a ve valivých lineárních vedeních (Tan). Data pro porovnání výsledků modelů byla převzata z disertace "Precision Control of High Speed Ball Screw Drives" Amina Kamalzadeha, která byla zaměřena na řízení lineárních os s kuličkovým šroubem.

Celkem byly studovány tři modely. Prvním z nich byl dřívější model tření v kuličkovém šroubu ad autora Olaru. Byl zjištěn výrazný rozdíl mezi výsledky experimentu a tímto modelem. Nebyl nalezen způsob, jak výsledky tohoto modelu zlepšit.

Druhým model byl založen na tření viskoelestických materiálů, který byl úspěšně použit pro modelování tření v lineárních valivých vedeních s mikro kuličkami. Bylo zjištěno, že tento model není použitelný pro modelování tření v kuličkových šroubech obvyklých velikostí.

Třetí model byl založen na modelech tření v kuličkových ložiskách. Původní model určený pro valivá ložiska byl zjednodušen několika předpoklady, pro zrychlení výpočtu. Výsledky tohoto modelu přijatelným způsobem aproximovaly výsledky experimentu a ukázalo se tak, že použitá zjednodušení mohou být aplikována.

Klíčová slova

Kuličkový šroub, EHL, tření

1. Introduction

The ball screw is a mechanism which converts rotational motion to translational motion. It consists of three basic components - shaft, nut and balls in thread between them. The ball screw is widely used in machine tool and other industries. The most important advantage of ball screw is high positioning accuracy which is influenced by many parameters, one of which being friction inside the system. It is important to be able to predict magnitude of the friction, to achieve better positioning accuracy. Another reason to study friction is the fact that friction is directly related to efficiency and energy loss in the system. So a reliable and simple model of friction would help to make an image of energy lost in the system during its operation. Such a model could then be used to optimize machine tool's energy consumption. Energetic optimization and Ecodesign of all machines, not only machine tools is gaining importance as the energy prices are rising and making environment friendly machines clearly became a trend.

There has been a lot of research trying to create a reliable model of friction in ball screw and similar devices like rolling bearings and linear guides. Although a lot of parameters are considered and the research has been conducted for a long time, there still is room for improvement in accuracy and simplicity of the friction models.

The research related to friction in ball screw can be categorized into three areas. The first of which is rolling friction, which was well known and research on it started a long time before the first ball screw was manufactured. The second important area of research is friction caused by lubricant – hydrodynamic friction. And the last one is friction in bearings. The researchers who investigate friction in bearings try to apply results from the two previously mentioned areas as well as other researches.

All models of friction for ball screw are based on research done on friction in ball bearings and theory mentioned above.

A ball screw friction model have been developed by Wei [1] based on present models of friction in ball bearings. Powerful model to estimate friction in radial and angular contact bearings without integrating of the shear stress on contact ellipses was introduced by Houpert [2]. Based on this work Olaru [3] have developed a new model of friction in ball screw.

There are three models studied in this paper.

The first is model presented by Olaru [3]. This model was used as a starting point. Other models were applied to ball screw, based on experience with this model.

The second of them is so called "Viscoelastic model" which was originally developed by Poschel [4] for rolling friction prediction in viscoelastic materials. This model was later used by Tan [5] to predict rolling friction in linear guides with micro balls. One of goals of this paper was to try to investigate if the viscoelastic model can be used for friction prediction in common size ball screw.

The third model presented in this paper is called "Simplified model". The Simplified model was an attempt to simplify previous model of friction in rolling bearings from Harris [6].

2. Models used for simulation

2.1 Olaru's model

Olaru's model of friction in ball screw was presented in [3]. Results of this model were not compared to experimental data. So the first step in the search for better friction models for ball screw was its verification.

This model of friction in ball screw is based on model of friction for bearings. The resulting friction force is the sum of forces acting on a ball. Such as the hydrodynamic rolling forces, the pressure forces, friction force between the ball and the nut, friction force between the ball and the screw and the friction force between the balls.

2.2 The Definition of the Viscoelastic Model

The viscoelastic model was proposed by Tan [5] to model friction in micro ball bearings based on model of rolling friction for viscoelastic materials. The basic equations remained the same but it was applied to ball screw here.

A system of independent mass-spring-damper elements moving in the z direction is used to model the viscoelasticity of the plane. The force acting on an element at the contact area as presented in Tan's paper is

$$f(x, y)dxdy = m\ddot{z}(x, y)dxdy + \gamma\dot{z}(x, y)dxdy + kz(x, y)dxdy \tag{1}$$

where m is the mass of springs per unit area, γ is the damping constant per unit area, and k is the spring constant per unit area.

Besides the force acting on the contact plane, there is also force F_B on the leading edge. This force accelerates elements which enter the contact area from zero velocity to $\dot{z}$. To calculate

this force, Tan first defined angle θ as shown in the *Fig. 3.19* the interval of angle theta $\theta \in [-\theta_0, \theta_0]$ covers the leading edge i.e. the section of r^+ circle between intersections with r^- circle.
After integration, the resulting force acting on the leading edge is

$$F_B = \int_{-\theta_0}^{\theta_0} f_B(\theta) d\theta = -m\tau v^2 [2\theta_0 + \sin(2\theta_0)] \quad (2)$$

The only unknown in F_B and f is the depth of penetration τ which can be solved based on equilibrium of forces

$$\iint_{A_c} f(\xi_x, \xi_y) d\xi_x d\xi_y + F_B + F_N = 0 \quad (3)$$

This is an implicit function of τ. The best way seems to be to solve the whole problem numerically.

After the τ is evaluated it is possible to calculate the kinetic energy per time transferred to leading edge P_B as derived in Tan's research

To evaluate the friction force between the ball and plane one uses the energy balance equation.

$$F_{fric} v = \iint_{A_c} f(\xi_x, \xi_y) \dot{z}(\xi_x, \xi_y) d\xi_x d\xi_y + P_B \quad (4)$$

The resulting friction force on the whole screw is sum of forces F_{fric} on all balls in contact. The friction moment on the whole screw is then total friction force multiplied by the ball radius *R*. Friction moment on the nut can be obtained in the same way.

2.3 The Simplified Model Based on Friction in Bearings

The purpose of this model is to try to estimate friction based on simplified equations given by Harris [6]. The friction model as described by him accounts for friction caused by contact pressure as well as friction caused by lubricant properties.

The Friction Equation

The main equation describing the friction in the lubricated contact is

$$F_{fric} = \int \tau dA = ab \int_{-1}^{+1} \int_{-\sqrt{1-q^2}}^{+\sqrt{1-q^2}} c_v \frac{A_c}{A_0} \mu_0 p + \left(1 - \frac{A_c}{A_0}\right) \left(\tau_N^{-1} + \tau_{\lim}^{-1}\right)^{-1} dt dq \quad (5)$$

A very complicated procedure is needed to solve this equation. One of goals of this paper is to try to simplify it and get an approximate solution using appropriate assumptions. The way that the simplified equation was defined follows.

The Friction due to Asperity Contact

The first thing to do is to determine the Herzian pressure in the contacts. This is calculated using curve fitted formulas from Houpert's paper [7]. Unknowns in these equations were calculated in the same way as in the Olaru's paper [3].

– The A_c to A_o ratio

Exact value of A_c to A_o ratio is difficult to calculate, it is a complex problem depending on the contact micro geometry, load, lubricant properties, velocity, probability etc. An alternative approach was however used by Zhou [8] and later by Olaru [3]. The following equation is based on Olaru's equation.

$$A_c / A_o = \exp(-B \cdot \lambda^c) \tag{6}$$

where λ is ratio which indicates the condition of lubricant film formation. The value of constant B is: *1.42*. Value of the friction coefficient μ_0 is 0.1.

– The coefficient of sliding

The reason why microslip occurs in the ball bearings is that contacting surfaces elastically deform and the contact area develops as the load is applied to the contact. The rolling cannot occur in all contact ellipse and only two lines of pure rolling are formed. These lines divide the contact ellipse to slip regions. The slip directions are different in different slip regions. The situation is showed in *Fig. 1*.

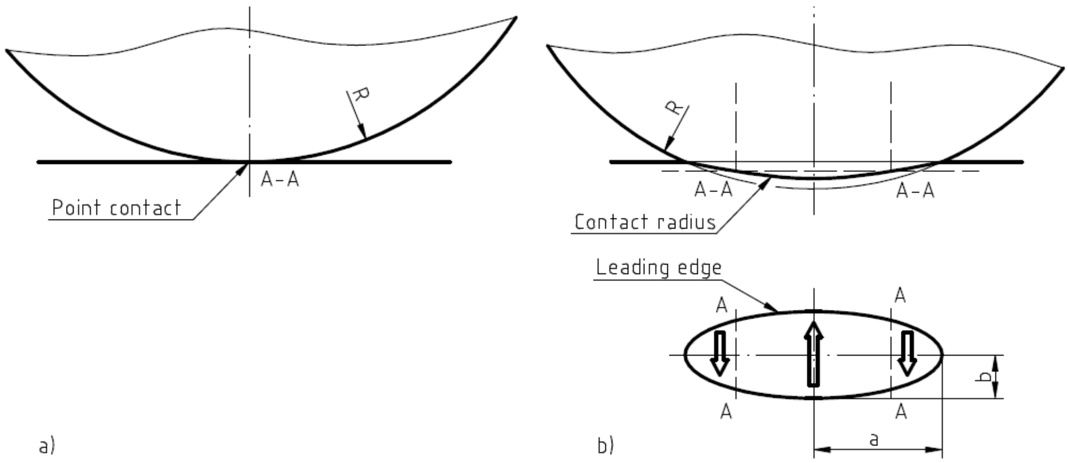

Fig. 1. *Microslip in sphere-plane rolling contact a) No load applied, no deformation and no slip occurs, b) Load applied, the contact ellipse is divided to three slip regions.*

The coefficient of sliding equals 1 or -1 depending on the sliding direction so the integration is divided to three integrals one with $c_v=1$ and two with $c_v=-1$.

It is assumed in this study that the area of slip acting against the rolling is 65% of area of contact ellipse, the area of the other two regions is 35% therefore, the coefficient c_v can be approximated by factor 0.3 and the three integrals are substituted by one. A similar assumption has been made in Olaru's friction model for ball screw [3].

The Friction due to Hydrodynamic Forces

The lubricant behavior in Elastohydrodynamic (EHL) lubrication regime is non Newtonian.

Several models of non Newtonian behavior have been developed. So called general model can substitute or closely approximate the others and that is why, it was used in this study. The equation of general model is given by

$$\tau_s = \tau_{\lim}\left[1-\left[\frac{\dot{\gamma}\cdot\eta}{\tau_L}\right]^{-n}\right]^{\frac{1}{n}} \tag{7}$$

where $\tau_{\lim}$ is the limiting shear stress and η is the viscosity of lubricant in the contact area, $\dot{\gamma}$ is shear strain rate and τ_s is shear stress.

– Shear strain rate

General expression for the shear strain rate in lubricant is derivative of velocity with respect to distance between lubricated surfaces $\frac{\partial v}{\partial h}$. Dependence of lubricant velocity v on the depth of lubricant is difficult to obtain. That is why the general equation was simplified to

$$\dot{\gamma} = \frac{v_c}{h_{\min}} \tag{8}$$

where v_c is the surface speed and h_{min} is minimal thickness of the lubrication film.

3. Results

Simulation data were taken from PhD thesis "Precision Control of High Speed Ball Screw Drives" by Kamalzadeh [9]. Static frictional characteristic of a high speed ball screw driven stage was measured in this thesis. The stage didn't use linear guides or solid ways, it used air brushings instead. So the characteristic shape was influenced only by the rolling friction. The resulting graph is showed in *Fig. 4.1.*

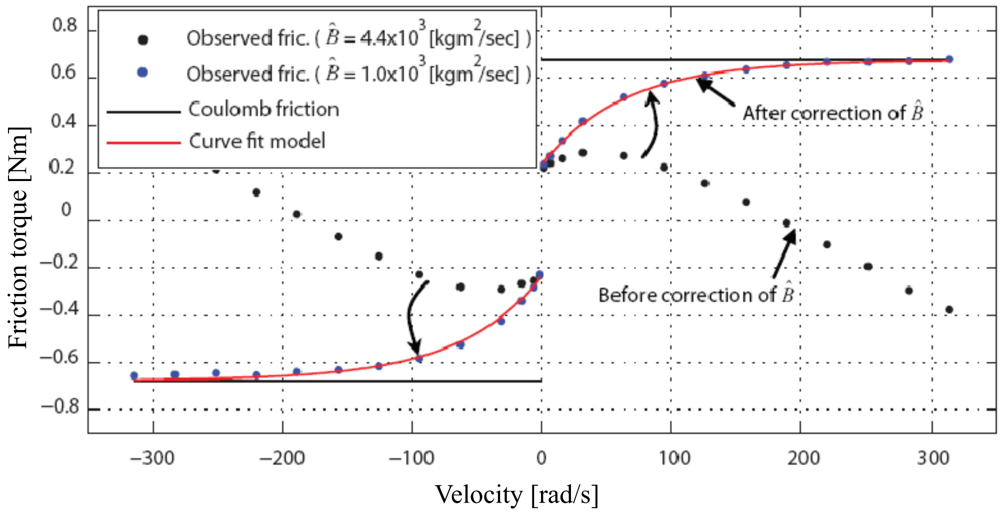

Fig. 2. *Friction characteristics, experimental data [9]*

3.1 Results of the Olaru's model

The model was evaluated for simulation data given in Kamalzadeh's PhD thesis to see how well it will fit in reality. The results are shown in the *Fig. 3.*

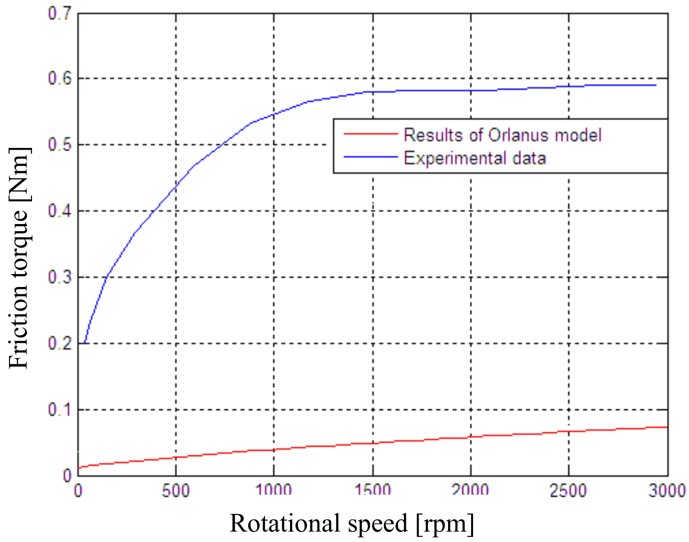

Fig. 3. *Comparison of the results of Olaru's model [3], and the experimental data from [9].*

The *Fig. 3* shows that the Olaru's model doesn't fit to the experimental data by Kamalzadeh [9]. The disagreement is relatively big. Given the fact that this model was not compared to any experimental results in the original paper [3], it was concluded, that this model is wrong.

3.2 Results of the Viscoelastic model

It is necessary to calculate constants of viscoelastic model first. It is assumed that there is elastic layer which is 100nm thick like in Tan's paper, this should be acceptable because the size of the elastic layer is compensated by the value of the constant γ. The constants m and k are calculated based on density and Young's modulus of the used material respectively.

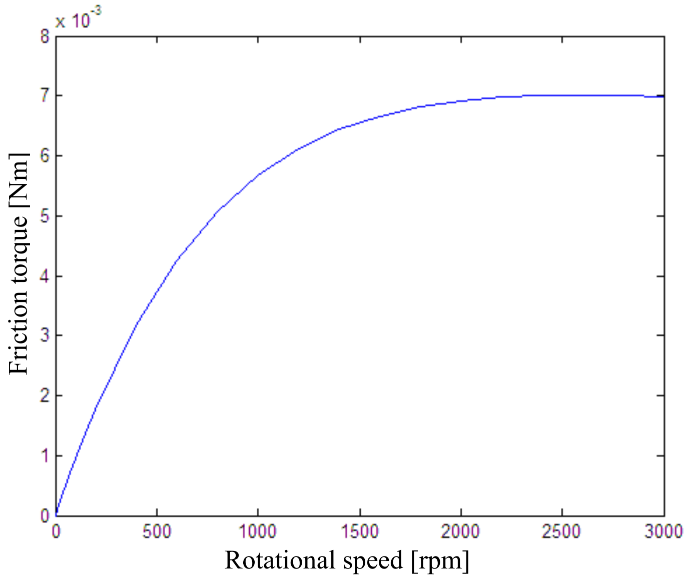

***Fig. 4**. Friction characteristics for k=2.1·1018N/m^3, m=7.8·10-4kg/m^3, γ =3.5·1013N·s/m^3*

It can be seen in the *Fig. 4* that the trend of data from vicoelastic model and the trend of the experimental data shown in *Fig. 2* is the same. The values calculated by the viscoelastic model are however much lower than the experimental data and the graph in *Fig. 4* lacks shift from the X-axis as it is in the *Fig. 2*(the experimental data start at value 0.2Nm).

The values calculated by the model can however be influenced by some factors not considered before. This will be studied in the following sections.

Influence of Preload on Results of the Viscoelastic Model

The estimation of the friction moment was based on mean torque calculated from the experimental data. It is however not clear what exactly the value of the preloading force was. The value of friction torque in the experiment varies from 0.2Nm to 0.6Nm. Preloading forces corresponding to these values will be used to see the influence on results of the viscoelastic model.

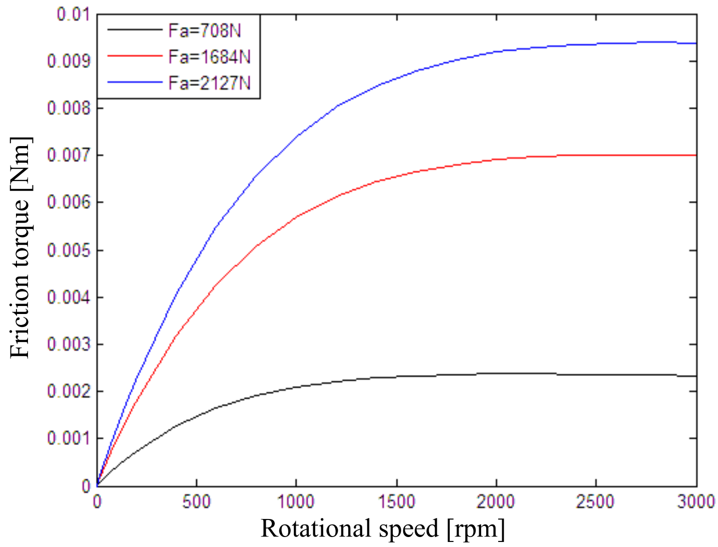

Fig. 5. *Influence of different preload on the results of viscoelastic model*

It is obvious from the *Fig. 5* that the influence of the uncertainty of the preloading force is significant. It can’t however explain big difference between the experimental data and model data.

3.2 Results of the Simplified Friction Model

The simplified model based on the bearing friction was described in section 2.3. Results of this model are presented in the *Fig. 6.*

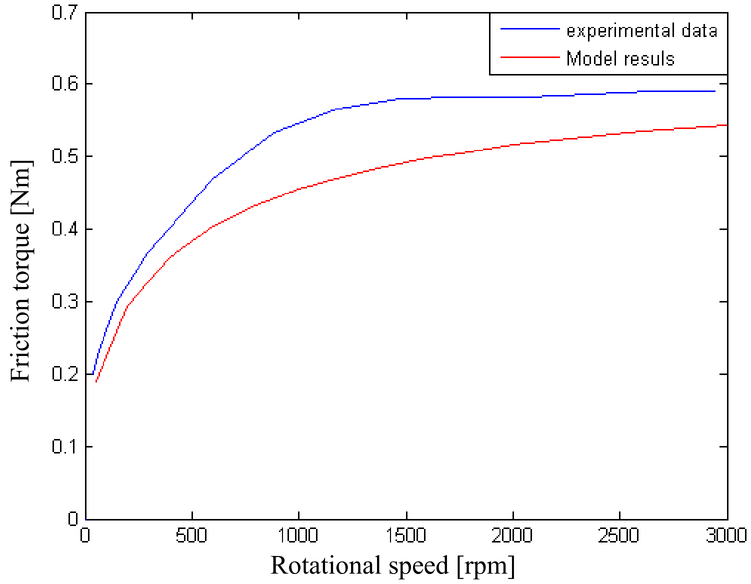

Fig. 6. *The comparison of experimental results and the Simplified model simulation results*

It is clear that the model values are lower than experimental data but the values are closer to experimental data than values of viscoelastic model.

The reasons why the results are lower are most likely absence of some parameters affecting the friction moment and the uncertainty of some input parameters especially the preloading force and limiting shear stress. Also some simplifying assumptions can contribute to differences in results. Influence of some of the mentioned variables will be investigated in following sections.

Influence of Different Preload on the Simplified Friction Model

As mentioned above, the estimation of the friction moment was based on mean torque calculated from the experimental data. It is however not certain what exactly the value of the preloading force was. The values of friction torque in the experiment varies from 0.2Nm to 0.6Nm so the preloading forces corresponding to these border values were used to see the influence on results of the simplified friction model.

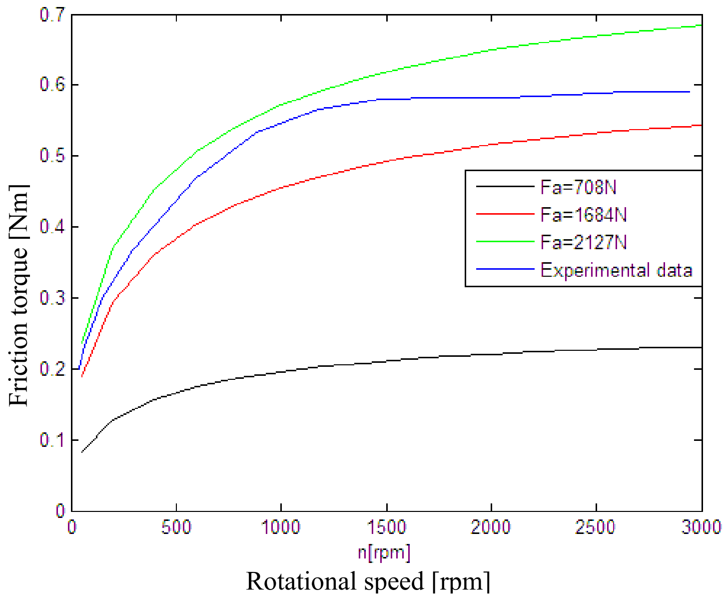

Fig. 7. *Influence of different preload on the results of simplified friction model*

The influence of different preloading forces is shown in the *Fig. 7*. It is clear that there is a big difference in estimated friction torque values and some influence on the curve shape.

Nevertheless uncertainty of the real preloading force can't explain slight difference of the curve shape between the model and the experimental data. This will be studied in the next section.

The Influence of the Lubricant Viscosity on the Simplified Friction Model Results

The simplified friction model was in this section evaluated with assumption of lubricant temperature 20°C. Because it was not clear which lubricant was actually used in the experiment, PAO ISO 68 was chosen based on viscosity recommended by the ball screw manufacturer. However, the ball screw manufacturer gives a range of viscosity. So that more lubricants can be chosen as suitable, for example SAE 20. Also the measurement of the friction characteristic took quite a long time, because it had to be measured in steady state. Presence of friction in the ball screw results in increased temperature and so the working temperature could be higher than 20°C. The ball screw working temperature can according to Heisel's research [12] rise up to 45°C.

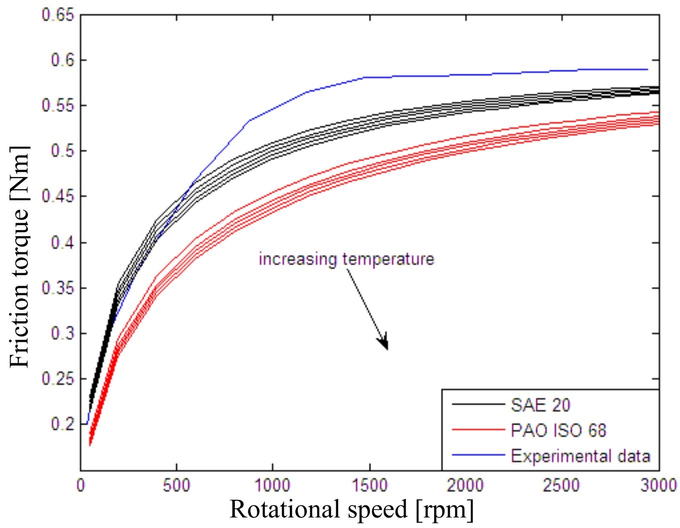

Fig. 8. *Influence of different lubricants and temperatures on the results of simplified friction model*

Table 1. *- Temperatures corresponding to results in Fig. 7*

Lubricant	temperatures at which was the viscosity calculated [°C]				
PAO ISO 68	20	30	35	40	45
SAE 20	24	30	35	40	45

The influence of the different lubricants and temperatures can be seen in the *Fig. 8*. The corresponding temperatures are shown in the *Table 1*. It can be seen that the increasing temperature slightly decreases friction and doesn't affect the curve shape (This model doesn't account for the increased preload effect possibly caused by increased temperature). Choosing the lubricant with higher viscosity can also increase the calculated friction torque.

4. Conclusion

This paper has tried to improve currently available models of friction in ball screw. The models applied were based on models used for prediction of friction in ball bearings and linear guides. The data for simulation were taken from PhD thesis "Precision Control of High Speed Ball Screw Drives" by Kamalzadeh which was aimed to ball screw control. The ball screw friction characteristic measured in his study was used here to compare ball screw friction models.

Three models were evaluated for the available simulation data. The first was ballacrew friction model by Olaru [3]. A significant miss match between the simulation data and the experimental data was found.

The second model was based on model of friction for viscoelastic materials which was successfully used for prediction of friction in linear guides with micro balls. Although the viscoelastic model is quite complicated and the solution takes a lot of time, it proved to be unsatisfactory to predict friction in ball screw accurately.

The reason why this model works for linear guides with micro balls and not for common size ball screw is most likely different dominant source of friction. While the most important source of friction in ball bearings and ball screws is microslip and hydrodynamic forces, the most important source of friction in linear guides with micro balls was probably elastic deformation.

The third model used for simulation was based on friction in ball bearings. The original model used for ball bearings was simplified by several assumptions to increase computational speed. The results of this model were in a good agreement with the experimental data. And thus the assumptions proved to be acceptable.

Resulting model is fast to compute and its accuracy is good enough for the purpose of energy loss estimation in ball screw. It is expected that a similar model would be used in a general model for energetic optimization of machine tool. This would be an important step forward in Ecodesign as the demand for energetic optimization techniques of machine tools grows.

List of Symbols

a	length of the longer semi axe of the contact ellipse	[m]
A_c	area associated with asperity-asperity contact	[m^2]
A_0	total contact area	[m^2]
b	length of the shorter semi axe of the contact ellipse	[m]
B	constant	[1]
c_v	coefficient of sliding	[1]
f	force acting on an element at the contact area	[N/m^2]
f_B	force on the leading edge per length	[N/m]
F_b	force on the leading edge	[N]
F_{fric}	friction force	[N]
F_n	total tangential force between a ball and the nut	[N]
h_{min}	minimal lubrication film thickness	[m]
k	spring constant per unit area	[N/m^3]
m	the mass of springs per unit area	[kg/m^2]

PB	kinetic energy per time transferred to leading edge	[J/s]
q	transformed coordinate	[1]
t	transformed coordinate	[1]
v	velocity of the ball centre	[m/s]
x	coordinate in contact plane	[m]
y	coordinate in contact plane	[m]
z	penetration of rigid ball	[m]
$\dot{z}$	first time derivative of penetration of deformable ball	[m/s]
$\ddot{z}$	second time derivative of penetration of deformable ball	[m/s^2]
y	damping constant per unit area	[kg/s/m^2]
n	dynamic viscosity of lubricant in the contact area	[Pas]
θ	position angle of elements on the leading edge	[°]
θ_0	angle limiting the leading edge	[°]
λ	lambda ratio	[1]
μ_0	friction coefficient for boundary lubrication	[1]
ξ	transformed coordinate	[1]
ξ	transformed coordinate	[1]
τ	maximal penetration of deformable ball	[mm]
$\tau_{\lim}$	limiting shear stress	[Pa]
τ_N	Newtonian shear stress	[Pa]
τ_S	shear stress	[Pa]

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