FRICITION IN LINEAR ROLLING SYSTEMS
FOR MICRO ACTUATORS

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ABSTRACT
As a result of small dimensions friction on the moving surfaces in the micro actuators becomes critical and is one of the fundamental limitation in the design and the fabrication of reliable MEMS. The use of the micro linear or rotating ball bearings in the micro actuators applications implies the simplify in construction, low level of friction and high stability, so that the micro ball bearings seems promising for future MEMS applications. Based on the results of the experiments realised in [1] and [3], authors developed two analytical models to evaluate the friction coefficient in a micro linear rolling system.

Keywords: Micro linear rolling system, adhesion, capillarity, friction coefficient

INTRODUCTION
The nanotechnology, described often as the technology of the future focused the researches and development activities in the last two decades. A lot of miniaturized devices, known as MEMS (micro electromechanical systems) was realised both for research in the laboratories and for various applications in the automotive industry, medical instrumentation, informatics technology as: micro sensors, linear and rotary micro actuators, micro motors, micro pumps, micro gear transmissions, micro grippers. The MEMS devices include mobile components with dimensions having the order of magnitude of about 10^-6 m. When the dimension of a machine component decreases from millimetres to microns, the area decreases by a factor of 10^6 and the volume (mass) decreases by a factor of 10^9. In the motion of such components the resistive forces as friction, viscous drags or surface tensions that are proportional to the area increase by a factor of 10^3 or more that the inertial forces that are proportional with the mass of the elements. As a result of small dimensions friction on the moving surfaces in the MEMS becomes critical and is one of the fundamental limitation in the design and the fabrication of reliable MEMS. Most of the MEMS devices have contact-type bearings with sliding friction, when the friction coefficient can be usually higher than 0.5…1 and the stick-slip instabilities can appear in the low speed conditions. Also, the wear limits the performances of such devices. Non contact-type bearings with more complicated support mechanisms like electrostatic or pressurized air have much less friction and non wear compared to contact-type bearings but are more complex systems. The use of the micro linear or rotating ball bearings in the MEMS applications implies the simplify in construction, low level of friction and high stability, so that the micro ball bearings seems promising for future MEMS applications.

FRICTION IN THE MICRO ROLLING TRIBOSYSTEMS
In macrotibology the processes are developed in components with relatively large mass under heavily loaded conditions. In these conditions, wear is inevitable and the bulk properties of mating components dominate the tribological performance. In micro/nanotribology, the tribological processes are developed in systems with relatively small mass under lightly loaded conditions. In this situation, negligible wear occurs and the surface properties dominate the tribological performance.

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micrograms to a few milligrams). As a result, friction and wear (on a nanoscale) of lightly loaded micro/nanocomponents are highly dependent on the surface interactions (few atomic layers). These structures are generally lubricated with molecularly thin films. In the micro and nanotribosystems a lot of interfacial forces as adhesion, van der Waals, electrostatic, capillary forces can be important and have an important contribution on the friction losses. Must be observed that in micro and nanotribosystems the friction forces and moments can stopped the normal function.

Adhesion between two solid surfaces based on the thermodynamic interfacial free energy can develop attraction forces of (200 – 300)μN or more [4-7]. For a ball on a flat surface the adhesive force Fa is given by relation [4]:

$$F_a = 3 \cdot \pi \cdot R \cdot \gamma \quad (1)$$

where R is the radius of the ball and $\gamma$ is the interfacial energy (J/m²). For two balls in contact with radius $R_1$ and $R_2$, the adhesive force is given by relation:

$$F_a = 3 \cdot \pi \cdot \gamma \cdot \left[1/R_1 + 1/R_2\right]^{-1} \quad (2)$$

As a result of adhesion, a micro-rolling tribosystem can be loaded supplementary with normal force and the contact area $A_c$ increases. According to the Johnson-Kendall-Robert (JKR) model [4], the contact area between a ball and a plane with including the adhesion effect is given by relation:

$$A_c = \pi \left[\frac{R}{E^*} + \frac{6\pi R \sigma}{Q_0} + \sqrt{\frac{12\pi R \sigma}{Q_0} + \left(6\pi R \sigma\right)^2}\right]^{3/2} \quad (3)$$

where $Q_0$ is normal force applied to the ball. Even in absence of an applied a normal load, the ball stick to the surface.

$E^*$ is the equivalent elastic modulus for the two solids in contact:

$$E^* = \frac{4}{3} \left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}\right]^{-1} \quad (4)$$

where $\nu$ and $E$ are the Poisson ratio and Young’s modulus, respectively.

The capillary forces are presents as a result of the condensed water from atmosphere on the solids. The most of the solids are hydrophilic surfaces and the atmospheric water cover these surfaces with molecular layers. In the contact zone between the two solids, by the capillary effect the adhered water lead to increase of normal force. A lot of experiences evidenced the influence of the pressure, temperature and humidity of air on the thickness of the condensed water films. So, Opitz, A. et al [6] measured for hydrophilic silicon the water layers between 0.7 nm to 2.6 nm, depending of the pressure. Other experiments presented in [4-6] indicated the water layers of 10 to 50 nm, for various materials. The capillary effects are dominant for the water layers more than (0.7…1)nm [4]. For a ball on a flat surface, as in Fig. 1, the ball is attracted on the flat surfaces by a capillary force $F_c$ given by relation [4]:

$$F_c = 4 \cdot \pi \cdot R \cdot \sigma \quad (5)$$

where $\sigma$ is the surface tension (N/m).

![Fig. 1: Ball-plane interaction as result of capillary force](image)

For a ball-ball contact the radius R in relation (5) will be changed by the equivalent radius $R^*$ given by relation:

$$R^* = \left[\frac{1}{R_1} + \frac{1}{R_2}\right]^{-1} \quad (6)$$

Sliding friction results in micro and nanotribosystems

In the last years a lot of researches was realised to determine the realistic friction forces in sliding microsystems. With the AFM technique ultra small forces (less than 1 μN), both normal and frictional was determined. A lot of new microtester was developed in the last years. Important results regarding friction was obtained in the Institut fur Physik, TU Ilmenau and IAVF Antriebstechnik AG, Karlsruhe, Germany [4-6] using the sliding ball–plane tribosystem and materials based on silicium.

Some important effects regarding the sliding friction in microrobosystem can be observed:
- The water from the atmosphere adhere on the surfaces in contact and influences the friction forces.
- For high water layers (more than 10nm) friction is dominate by capillary. The capillary bridge increases the normal force and imposes resistance against shear. High values for the friction coefficient was experimentally obtained ($\mu = 1…4$ and more).
- For very low water films (less than 0.2 nm) the low values of the coefficient of friction was determined ($\mu = 0.3…0.4$).

A QUASI STATIC ANALYTICAL MODEL TO EVALUATE FRICTION LOSSES IN A MICRO LINEAR BALL BEARING

In the Fig. 2 is presented the geometry of a micro linear ball bearing experimentally studied in [1]. The slider is in relative linear motion from the stator with the linear speed and the ball has a angular velocity $\omega_b$.

Under the external load G all the four contacts of the ball with slider and stator are loaded with a normal force $Q_0$ given by relation:
According to the JRK adhesion model the normal force for every ball–race contact $Q$ will be higher than $Q_0$ and is computed by following relation [4]:

$$Q = \left( Q_0 + 6\pi\gamma R + \sqrt{12\pi\gamma RQ_0 + (6\pi\gamma R)^2} \right)$$ (8)

Considering thin water layers on the surfaces of balls and races as lubricant, in all four contacts between ball and races are developed followings forces[2]: hydrodynamic rolling forces $FR$, pressure forces $FP$, sliding contact forces $FS$. The directions of these forces acting on the ball are presented in Fig. 3. Note that the index 1 refer to slider and index 2 refer to stator.

The hydrodynamic forces $FR$ are computed with relation:

$$FR = 2.86 \cdot E \cdot RX^2 \cdot k^{0.348} \cdot G^{0.22} \cdot U^{0.66} \cdot W^{0.47}$$ (9)

where $E$ is the equivalent Young’s modulus of the materials in contact, $RX$ is the equivalent radius in the rolling direction, $k$ is the radii ratio.

$G$ is the dimensionless material parameter, $G = E \cdot \alpha_p$ (10)

where $\alpha_p$ is the piesoviscozitat coefficient.

$U$ is the dimensionless speed parameter,

$$U = \frac{\eta_v \cdot \nu}{E \cdot RX}$$ (11)

where $\eta_v$ is the viscosity of the water at the atmospheric pressure and at the contact temperature, $\nu$ is the tangential speed in the ball-races, in the rolling direction.

$W$ is the dimensionless load parameter definite by relation:

$$W = \frac{Q}{E \cdot RX^2}$$ (12)

$FP$ is pressure forces due to the horizontal component of the water pressure in the rolling direction. For a ball – ring contact, the pressure force acting on the centre of the ball can be expressed as a function of hydrodynamic rolling force $FR$:

$$FP = 2 \cdot FR \cdot \frac{R_R}{R_R + R_b}$$ (13)

where $R_b$ is the ball radius and $R_R$ is the ring radius. For the linear system, $R_R \to \infty$ and the pressure force acting on a linear race is null ($FP = 0$).

The capillary force between balls, $F_{cb}$ is computed with relations (5).

The friction forces $FS$ in a ball-race contact is the sliding traction forces due to local micro slip occurring in the contact, and can be calculated explicitly as the integral of the shear stress $\tau$ over the contact area:

$$FS = \int \tau \cdot dA$$ (14)

Considering the contact deformations between the ball and the slider and the relative speed, Xiaobo Tan et.al [3] developed a sophisticated viscolastic model for the ball-race friction force. The model developed by Tan et al. need some materials characteristics as damping and spring constants, difficult to be obtained.

The moments acting on the ball are presented in Fig. 4. In each ball-race contact, two resistance moments are developed: the elastic resistance moments MER and and
the pivoting moments $MP$.

For a ball-plan contact $MER$ can be determined by relation [2]:

$$MER = 4.78 \times 10^{-7} \left( \frac{d}{2} \right)^{0.33} Q^{1.33}$$  \hspace{1cm} (15)

where $d$ is the ball diameter.

Considering a constant friction coefficient for the sliding motion in a ball-race contact $\mu_s$, the pivoting moment $MP$ can be evaluated by relation:

$$MP = \frac{3}{8} \mu_s \cdot Q \cdot a$$  \hspace{1cm} (16)

The semi major contact ellipse axis $a$ is computed considering both normal load $Q_0$ and adhesion, according to JRK model. So, from equation (3) results:

$$a = \left[ \frac{R}{E} \left( Q_0 + 6\pi R + \sqrt{12\pi R Q_0 + (6\pi R)^2} \right) \right]^{1/3}$$  \hspace{1cm} (17)

where $R$ is ball radius ($R = d/2$).

As a result of contacts between balls, for each ball acts a resistance moment $Mb$ evaluated by relation:

$$Mb = \mu_b \cdot Fcb \cdot d$$  \hspace{1cm} (18)

where $\mu_b$ is the friction coefficient between two balls and $Fcb$ is the capillary force determined with relations (5).

For the same geometry and material for slider and stator the forces and the moments are the same values both for contact (1) and contact (2).

Based on the equilibrium of the forces and moments acting on the ball, the friction force for a ball-race contact $FS$ can be determined as result of all the other forces and moments acting on the ball and following relation was obtained:

$$FS = \frac{1}{d} \left[ 2MP_{tg}(\theta) + 2MER + \frac{Mb}{\cos(\theta)} \right] + FR$$  \hspace{1cm} (19)

For every ball-race contact, the ball acts on the race with the total tangential force in rolling direction: $FS + FR$.

Considering that every ball acts on the slider or stator in two contact points it can be obtained the total tangential resistance force given by a ball to the stator or to the slider in rolling motion:

$$F_{ball} = \frac{1}{d} \left[ 4MP_{tg}(\theta) + 2MER + \frac{2Mb}{\cos(\theta)} \right] + 4FR$$  \hspace{1cm} (20)

For a system with $z$ balls on the slider or on the stator acts a total tangential force $F_{total}$ given by relation:

$$F_{total} = z \cdot F_{ball}$$  \hspace{1cm} (21)

A global friction coefficient can be obtained by dividing the total tangential resistance of the slider by the normal load $G$:

$$\mu_{global} = \frac{z \cdot F_{ball-race}}{G}$$  \hspace{1cm} (22)

Numerical results

The numerical results are performed for the condition used in the paper [1]:

- stainless steel balls with diameter $d = 285$ $\mu$m;
- silicon slider and stator with two $V$ - grooves realised at an angle $\theta = 54.7^o$;
- temperature of 27 degree and a relative humidity of 40%RH;
- relative speed between the slider and stator was between zero to 150 mm/s;

The number of the balls and the load $G$ was considered in two variants:

- variant A with 4 balls without contact between balls and a mass of the slider $G = 0.9$ grams;
- variant B with 18 balls on the two $V$ - grooves (9 balls for each $V$-groove) with the balls in contact one each other and the mass of slider $G = 0.4$ grams.

The elastically properties of the ball and races was:

- $E_{steel} = 2.1 \times 10^{11}$ $Pa$ , $E_{silicon} = 1.5 \times 10^{11}$ $Pa$ ,
- $\nu_{steel} = 0.3$ , $\nu_{silicon} = 0.3$ .

The viscosity of water $\eta = 0.001$ $Pa\cdot s$ and the piezoviscosity coefficient $\alpha_p = 10^{-8}$ $Pa^{-1}$.

A. Friction coefficient for a system with 4 balls without contacts between balls

If the balls are not in direct contact, the capillary effect is not included: $Fcb = 0$ and $Mb = 0$.

The global friction coefficient was computed for two values of the friction coefficient in pivoting motion, so for $\mu_s = 0.5$ and for $\mu_s = 1$.

If it was neglected the adhesion between ball and race, was obtained values for the global friction coefficient between 0.002 and 0.005 , as in Fig. 5-a. Including the adhesion effects was obtained increasing of the contact friction coefficient with values between 0.007 and 0.012, as in Fig. 5-b.

Following remarks can be made:

1. According to our analytical model, by including the adhesion in ball-race contacts was obtained accepted values for friction coefficient with a magnitude between...
0.007 to 0.012. Comparison with experimental values obtained by [1] in the similar conditions evidenced a good correlations between our numerical results and the average values obtained by experiment.

\[ \mu_{\text{global}} = \frac{z}{d \cdot G} [4MP\tan(\theta) + 4MER] \]  

(23)

2. No important variation of the friction coefficient with relative speed was obtained by analytical model. It can be explained by the small influence of the hydrodynamic effect given by the forces FR (less of 5%). In this conditions, if are not contact between balls the global friction coefficient in a micro linear rolling system presented in Fig. 1 can be evaluated with following simplified relation:

\[ \mu_{\text{global}} = 0.5 \]

\[ \mu_{b} = 1 \]

The numerical results of global friction coefficient are presented in Fig. 6. The obtained values for global friction coefficient are between 0.25 to 0.55 for the condition imposed. In the similar conditions, by the experiments realized in [1] it was obtained variation of the friction coefficient between 0.2 and 0.6.

Fig. 6: The global friction coefficient determined by analytical model for 18 balls

It can be observed that the contacts between balls dominates the losses for the given conditions. The influence of the pivoting friction is smaller that the influence of friction between balls.

A DYNAMIC ANALYTICAL MODEL TO EVALUATE FRICTION LOSSES IN A MICRO LINEAR BALL BEARING

The quasi static analytical model presented in the section 2 can not evidences the variation of the friction coefficient with the relative speed between slider and stator in the vicinity of zero speed, as are resulted by experiments [1], see Fig. 7.

Fig. 7: Coefficient of friction in the vicinity of zero speed obtained experimentally in [1] for variant A

To evidence the variation of the friction coefficient with the speed in the vicinity of zero speed Xiaobo Tan et al. [3] developed a sophisticated viscoleastic model for the ball-race friction force. So, the model developed by Xiaobo Tan et al. was realized for a rolling contact between a ball and a plan and leads to zero value for friction coefficient when the sliding speed is zero, see Fig. 8 [3].
We propose a simplified dynamic model by including a modified Langevin function to the quasi static model. So, the global friction coefficient for variant A (without contact between balls) can be expressed by relation:

\[
\mu_{\text{global}} = \frac{4\pi G}{d} \left[ M P g(\theta) + \text{MER} \right] \cdot f(v)
\]  

(24)

where \( f(v) \) is the modified Langevin function:

\[
f(v) = K \left( \frac{1}{\alpha v} - e^{\alpha v} + e^{-\alpha v} \right)
\]  

(25)

where \( K \) and \( \alpha \) are constants determined by the experimental results and \( v \) is the relative speed.

Imposing the limits of the friction coefficient for the relative speed \( v = 0.01 \text{m/s} \) and \( v = -0.01 \text{m/s} \) with the values obtained by the quasi static model was determined the constants \( K \) and \( \alpha \): \( K = 0.9 \text{ to } 1.0 \) and \( \alpha = 2000 \text{ to } 3000 \).

The variation of the global friction coefficient with relative speed as result of equation (24) is presented in Fig. 9 for two values of the sliding friction coefficient for pivoting motion of the ball: \( \mu_s = 0.5 \) and \( \mu_s = 1 \).

More accurate experiments must be realised to obtain the best values for these constants.

3. Analysing the relation (24) it can be observed that the global friction can be obtained as a sum of two components: a friction coefficient as result of pivoting motion and a friction coefficient as result of rolling friction. The rolling component of the friction coefficient is too smaller that the pivoting component and can be neglected in a first approximation.

The model developed by Xiaobo Tan is applied only in rolling friction without pivoting. The pivoting friction is characterized by a constant friction coefficient non depending of the sliding speed.

As a new approximation the global friction coefficient can be expressed by relation:

\[
\mu_{\text{global}} = \frac{4\pi G}{d} \left[ M P g(\theta) \right] \cdot f^*(v)
\]  

(26)

where \( f^*(v) \) is a particular expression of the Langevin function with constant \( K = 1 \) and \( \alpha > 10^4 \). The variation of the friction coefficient with speed realised by equation (26) is presented in Fig. (10), for \( \mu_s = 0.5 \).

CONCLUSIONS

A complex quasi static analytical model to evaluate the friction coefficient in a micro linear ball system has been developed. The model includes following losses sources: pivoting motion between ball-race, elastic resistance in rolling of the ball over the races, hydrodynamic effects in ball-races and the losses in the ball-ball contacts. Both adhesion between balls and the races and capillary effects in ball-ball contacts was considered.

To validate the model, the numerical results was performed for a micro linear ball bearing studied
experimental by [1]. The numerical results are in the same range of values with experimental results.

When the balls are not in direct contact one with other, the friction coefficient have values between 0.007 to 0.012 For this condition the most important source of losses is the ball-race pivoting motion.

When the balls are in direct contacts one with other, the friction coefficient increases with an order of magnitude. So, values for friction coefficient between 0.25 and 0.55 was obtained. The friction losses between the ball-ball contacts are dominate.

Including a modified Langevin function, the authors realised a simple dynamic model for friction coefficient in a micro linear ball system with the friction coefficient – relative speed dependence.

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