348. Control of friction forces with stationary wave piezoelectric actuator

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Abstract. In the field of the research on piezoelectric motors, the control of friction forces by ultrasonic waves was studied mainly from an experimental point of view [1]. The principle of friction force reduction by imposed mechanical vibrations in unlubricated contacts was recently studied in order to reduce the friction losses in an internal combustion engine with Langevin type actuators [2]. This article deals with the advantages of flexural stationary wave piezoelectric actuators in the control of the friction forces thanks to their high pressures generated at high frequencies (>10 Khz). The use of specific contact geometry which is defined by the Hertz theory associated with partial slip contact conditions, allows optimizing the lubrication effect. In the case of piezoelectric torque limiter application, the design and the numerical simulation of dedicated piezoelectric actuator are compared. In agreement with the contact modeling, the characterization of the complete actuator on mechanical test bench validates the "torque limiter" function and the optimization of lubrication principle with dedicated contact geometry.

Keywords: piezoelectric actuators, stationary wave, mechanical vibrations, electroactive lubrication, friction force control, friction coefficient reduction.

Introduction

The electroactive lubrification principle is based on the superposition of mechanical vibration within contact in dry friction. Preliminary studies, which are done from an elementary ball/plane contact, made it possible to establish predictive analytical models. In order to obtain higher relative speeds, an optimization of this new concept is necessary. Two improvement ways are possible: the increase of contact frequency by the choice of dedicated piezoelectric structure, and the vibratory amplitude minimization by the decoupling of contact stiffness. A first part is devoted to determine the parameters of mechanical vibration (amplitude and frequency) by an analytical model, in the case of a discretized contact. Following the obtained results, the choice of piezoelectric actuator structure and the one of their specified geometry are carried out. The selected structure as well as the discretized contact principle were implemented and validated within an operational structure of piezoelectric clutch type.

Cylinder/plane contact. Discretization.

Recent developments concerning the study of the electroactive lubrication for classical ball/plane contact geometry [2], [3], shows that specific indentor geometry needs to be took into account in order to optimize the mechanical vibration parameters and the mechanical power of dedicated actuators. A low tangential stiffness associated to a high normal stiffness are necessary in order to keep the sliding contact in partial slip condition with a low magnitude of the mechanical vibration. A necessary mechanical decoupling between the tangential and the normal stiffness by a discretization of the geometry of the indentor is then studied.

Amplitude and frequency of vibrations. To determine vibratory amplitude Z and frequency ω, tangential and normal stiffness have to be expressed. The cylinder/plane contact is considered as the limit of ellipsoidal/plane contact. The major axis “b” is supposed infinite.

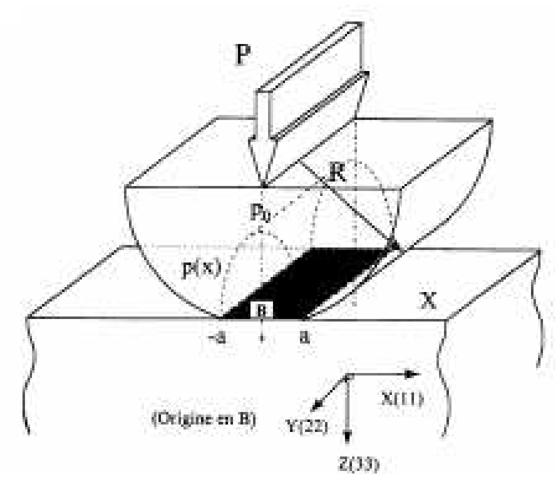


Fig. 1. Cylinder/plane contact with a and l, respectfully half-width and length of contact and R, radius of cylinder [4]

Respectfully, the tangential and normal stiffness expressions are, [5]:

$$For\ \frac{a}{b} \to 0, \quad \frac{K_{CT}}{K_{CN}} = 1$$

and

$$K_{CN} = \frac{\partial P}{\partial \delta_z}$$

P is the static effort and δ_z is the indentation. Some authors propose different expressions of indentation. Thanks to the last contact hypotheses, the equation of indentation is given by [6]. To insure the regular separation of surfaces, the dynamic force, which is produced by the excitation with a vibration, has to be negative or null. This condition implies minimal vibratory amplitude. Moreover, the requirement of partial slip is respected by the definition of a minimal excitation frequency, for a constant relative tangential speed, ν. The relative displacement has to be inferior or equal to the transient displacement. This last one is the limit between partial slip and total slip [2], [3].

$$Z \geq \frac{P}{K_{CN}} = Z_0$$

and

$$\omega \geq \frac{\nu}{\mu_d . K_{CT} . K_{CN}} \cdot \frac{1}{\sqrt{Z^2 - Z_0^2}}$$

After expressing excitation conditions, the instantaneous friction coefficient can be calculated.

Table 1. Instantaneous friction coefficient value in accordance with Coulomb Orowan's law

$\mu(t) = \frac{\delta(t)}{K_{CT}}$	$si\,\delta(t) < \delta_t\ et\, 0 < t < \Delta < T$	Contact and partial slip
$\mu(t) = \mu_d$	$si\,\delta(t) > \delta_t\ et\, 0 < t < \Delta < T$	Contact and total slip
$\mu(t) = 0$	$si\ \Delta < t < T$	Surfaces separation

The tangential force is defined by:

$$Q = \int_0^T \mu(t).F_d(t)dt$$

T is excitation period and Δ is the effective contact time betwwen the two surfaces.

Normal and tangential stiffness allow to determine apparent friction coefficient which is the ratio between normal static effort P and the tangential force Q. To limit dynamic force and the power of used actuators, the electroactive lubrification has to be optimize: vibratory amplitude has to decrease and relative speed has to increase. According to the last study, the improvement of electroactive lubrification corresponds to the augmentation of normal stiffness and to the diminution of tangential stiffness. The discretization of cylinder/plane contact seems to be a good solution to ensure the optimization.

(1)

Contact discretization. The studied contact is illustrated by

Fig. 2.

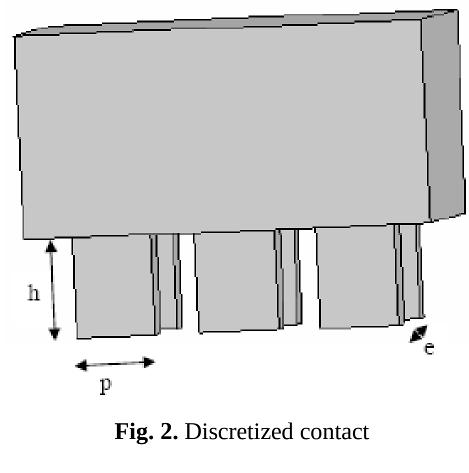


Fig. 2. Discretized contact

In first approximation, the new normal and tangential stiffness are defined by, [2]:

(2)

$$K_{CN} = \frac{E.n.e.p}{h}$$

and

$$K_{CT} = \frac{n.3.E.p.e^3}{12.h^3}. \quad (4)$$

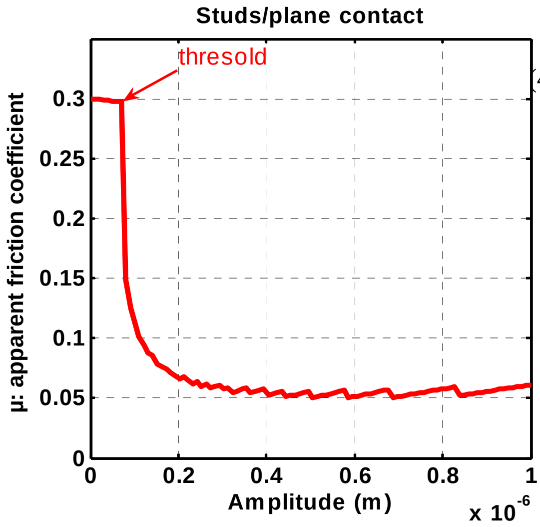
These expressions are inserted in the calculation of instantaneous friction coefficient.

Comparison between discretized and non discretized contact. According to the last study, there are two fundamental parameters of excitation: amplitude and frequency. In order to determine the most efficient contact for electroactive lubrification, Matlab simulations are compared. The two surfaces in contact are composed by the same material (steel). The normal static effort is about 20 N for each type of contact. The dynamic friction coefficient which is experimentally defined without vibrations is chosen about 0.3. Moreover, we equalize the contact areas. The cylinder/plane contact is defined by the length of 7.5 mm and the equivalent radius of 20 mm. The discretized contact is as shown in

Fig. **2**: e=1.5 mm, h=5 mm and p=2.5 mm.

For each curves, it exists threshold which is equal to the dynamic friction coefficient. From the minimal amplitude, there is the separation of contact surfaces. However, friction forces increase for the cylinder/plane contact while they decrease for the studs/plane contact. In the case of cylinder/plane contact, the vibratory amplitude creates a dynamic force which is superior to static effort P, contrary to studs/plane contact.

For a high excitation frequency, the apparent friction coefficient tends to decrease until a low value. For studs/plane contact, the friction forces decrease more quickly for a frequency range less important. The analytical simulations show that studs/plane contact allows a best control of electroactive lubrification thanks to a higher relative tangential speed. So, we chose implant this type of contact on the dedicated piezoelectric actuator.



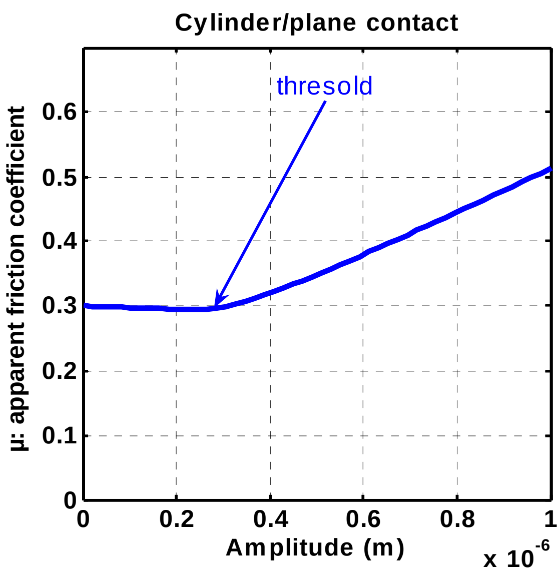
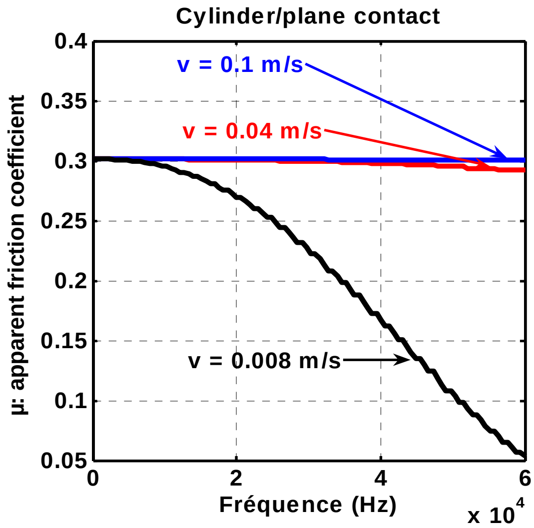


Fig. 3. Evolution of apparent friction coefficient according to vibratory amplitude. For each contact, excitation frequency is 22 kHz and relative speed is 0.05 m.s^{-1}



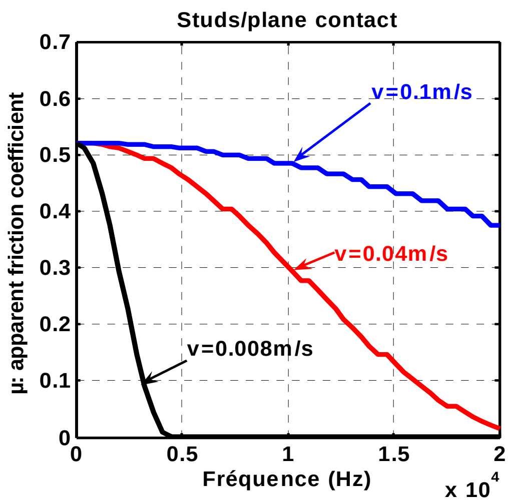


Fig. 4. Evolution of apparent friction coefficient according to excitation frequency. For each contact, vibratory amplitude is 0.3 µm and different tangential relative speed (0.008, 0.04, 0.1 m.s^{-1})

Dedicated actuator

In order to insure the required amplitude and frequency for efficient electroactive lubrification, piezoelectric actuators seem to be the best appropriate. Under high pressure (20 MPa), they generate low vibrations (few µm). When piezoelectric actuators are exited in their mechanical resonance, the frequency becomes high enough to satisfy the frequency condition. The chosen actuator has to have:

- good electromechanical conversion. The longitudinal piezoelectric mode presents the most efficient electromechanical coupling.
- easy processing and good resistance for severe applications. A prestressed structure is preferred to a glued structure.

So, the most appropriate assembly is composed by two masses which prestress piezoelectric ceramic stack. If this actuator presents a normal deformation, it is a Langevin type. The contact frequency is equal to excitation frequency. However, the contact frequency of flexural type actuator, which has a flexural deformation, is twice superior to the excitation contact. According to contact study, the increase of contact frequency is an advantage for the optimization of friction force control. So, the flexional actuator is selected.

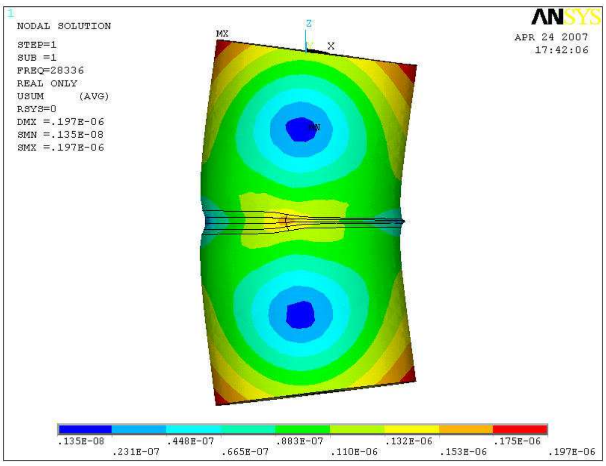


Fig. 5. The Ansys study of specific flexional actuator

Actuator characteristics
Inner Radius: 4 mm
Outer Radius: 12.5 mm
Mass length: 20 mm
Piezoelectric ceramic thickness: 0.5 mm
Material: steel

The study of this actuator is based on [7]. At the mechanical resonance, this actuator is represented by the following schema of Mason.

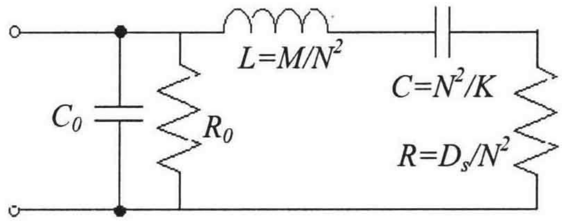


Fig. 6. The equivalent schema of Mason for flexural actuator
C0: locked capacity
R0: electric looses
M: vibratory mass
K: structure stiffness
Ds: mechanical dissipation
N: electromechanical coupling

R_0, C_0, M, K, Ds and N are analytically defined by [7]. They depend on mechanical and electrical properties and on the structure geometry. These parameters allow defining the dynamic behavior and the optimal operating conditions of actuator. R_0, C_0, L, C and R are experimentally determined by the admittance measurement of equivalent circuit while N is defined from the vibratory speed and excitation tension. To do the last measurement, an impedance meter and a laser vibrometer are used. In order to determine the most appropriate actuator geometry, a comparison is done thanks to Ansys simulations between two actuators.

The magnitude of the displacement measured with a laser vibrometer on the surface of structure n°1 is about 1.5 µm, for an electric field about 200 kV/m. The comparison of the two actuators presented on the table 2, shows the influence of actuator geometry on the vibratory amplitude

Table 2. Comparison between two flexural actuators obtained with ANSYS numerical simulation

		Structure 1 (ANSYS)	Structure 1 (Experimental data)	Structure 2 (ANSYS)
Masses length	L (mm)	20	20	40
Inner radius	Rin (mm)	4	4	4
Outer radius	Rout (mm)	12,5	12,5	12,5
Constitutive material		Steel (XC38)	Steel (XC38)	Steel(XC38)
Resonance frequency	f (Hz)	24250	28336	14031
Modal Vibratory mass	M (g)	43,26	58,87	65,89
Modal Stiffness	K ($N.m^{-1}$)	1.10^9	$1,91.10^9$	$5,12.10^8$
Electromechanical coupling	N ($N.V^{-1}$)	1,18	1,45	1,18

and the resonance frequency. The contact optimization, which is obtained by its discretization allows to prefer a high resonance frequency to an important vibratory amplitude, in particular when high relative speeds are required. In addition, the minimization of vibratory amplitude associated to a high resonance frequency leads to the reduction of actuator volume and its electronic converter.

Electroactive thrust bearing

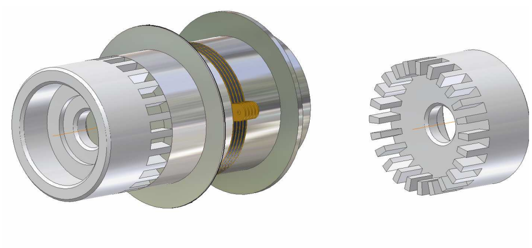
The electroactive lubrification principle constitutes an interesting way when one wants to act on high forces or high torques, possibly by controlling them electrically, without magnetic pollution, and in a reduced volume.[2] The typical industrial application should be an electroactive thrust bearing or a piezoelectric clutch. A first piezoelectric clutch bearing had been designed with the new concept of contact geometry. This piezoelectric actuator is based on the structure of flexural actuator excited in a stationary wave condition. The excitation frequency is about 25 kHz.

With respect to the evolution of friction torque, this piezoelectric clutch (

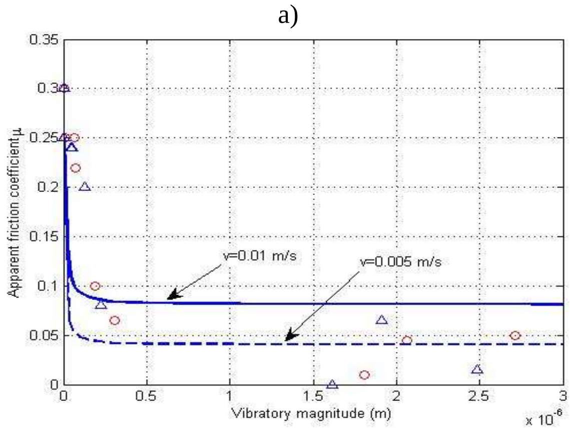
Fig. **7**) was characterized in a specific mechanical test bench in function of the vibratory amplitude for two imposed relative speeds. A comparison with the analytical modeling is presented on the curve of the

Fig. **7**. It shows a good adequacy with the model which is presented on the

Fig. 3 and a reduction from approximately 85% of the apparent friction coefficient.



a)



b)

Fig. 7. a) Design of the piezoelectric clutch - b) Comparison of theoretical and experimental results of the friction torque as a function of the vibratory magnitude

Conclusion

The electroactive lubrification principle which is firstly studied for a ball/plane contact was extended to the cylinder/plane contact and was optimized by its discretization in order to obtain higher relative speeds. According to modelling, excitation frequency has to be maximised while vibratory amplitude presents a contact separation threshold which has to be minimised. The increase of contact frequency is difficult except by the choice of piezoelectric flexural type structure. The vibratory magnitude was decreased by a discretized contact choice. These optimizations are validated by the processing and the characterisation of piezoelectric thrust bearing. The obtained results show, on the one hand, a good correlation with the proposed contact model and on the other hand, a friction coefficient decrease about 85% of its value without imposed vibrations.

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