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Sliding bearing with adjustable friction properties

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ABSTRACT

Sliding bearings have good damping properties and a high stiffness. However, stiction and poor sliding friction characteristics limit their use in applications, where high requirements are defined on the smoothness of motion. By inducing ultrasonic oscillations into the sliding contact it is possible to actively control the friction. The adjustable friction bearing uses this principle. This leads to a new bearing type that has the stiffness and damping properties of a sliding bearing, while at the same time, offering a linear friction characteristic without stiction. This paper presents basic principles for the design of ultrasonic bearings along with experimental results of a friction-adjustable machine table.

1. Introduction

The friction characteristics of bearing systems are usually defined by the material and lubricants within the contact region. Roller bearings, sliding bearings and fluido static bearings are state of the art. Sliding bearings have good damping and stiffness properties but suffer from high stiction and nonlinear friction characteristics [1]. Roller bearings have much lower friction and stiction but poor damping properties, lower stiffness and vibrations from compressing and decompressing of rolling elements. Furthermore, roller bearings are often mounted with wipers to protect the bearing from dust and fluids, hence stiction is reintroduced by wiper setups [2].

The idea of a bearing with adjustable friction properties comes from industrial processes where ultrasonics are used to reduce the process forces, like in ultrasonic cutting, milling and drawing [3]. Theoretical models for the effect are given in [4] explaining that the high frequency microscopic motion is averaging the friction on a microscopic scale resulting in a loss of friction on the macroscopic scale. Section 2 gives a short review of the important oscillation parameters from the theoretical friction reduction model. The design approach to combine a sliding bearing with ultrasonics is described in Section 3. Experimental results of the oscillation and friction properties are presented in Section 4.

2. Friction theory and parameters of an oscillating slider

The effect of high-frequency oscillations reducing the macroscopic friction can be described using the coulomb friction model. In this model, the friction force depends on the normal force F_N and an experimentally determined friction coefficient μ. With the direction of friction opposing the relative velocity v_{rel} between the surfaces, the friction force F_R becomes

$$F_R = -sign(v_{rel})\cdot\mu F_N. \tag{1}$$

Through oscillation of the slider the relative velocity becomes a superposition of the constant feed velocity v_f and an oscillating velocity v_s.

For a harmonic oscillation with an amplitude of x_s at the frequency f_s, the relative velocity of the motion becomes

$$v_{rel}(t) = v_f + v_s(t) = v_f + 2\pi f_s x_s \cos(2\pi f_s t). \tag{2}$$

If the oscillation velocity amplitude is higher than the feed velocity the direction of the friction (1) will change within one period. In [4] it is shown that averaging the friction force over one oscillation period gives

$$F_R^* = -\mu F_N\cdot \underbrace{\frac{2}{\pi}\arcsin\left(\frac{v_f}{v_s}\right)}_{\gamma}. \tag{3}$$

Eq. (3) can be interpreted as the normal friction force with unchanged friction coefficient and unchanged normal force modified by a friction reduction factor γ. In this model, the friction reduction is solely depending on the ratio of the feed velocity to the vibration velocity controlled by the oscillation

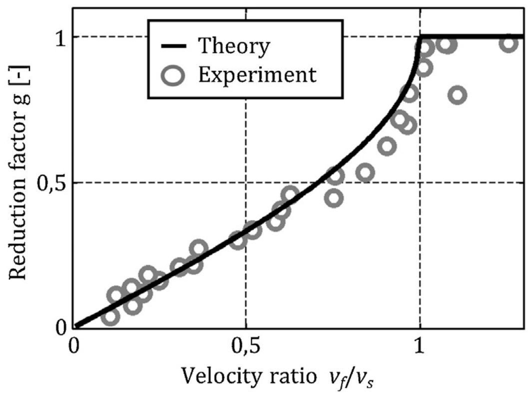

Fig. 1. Friction reduction: theoretical and experimental results using a steel–plastic combination with 50 Hz and x_s within 30–80 μm.

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parameters x_s and f_s. An experiment was made to validate this model in respect to material combinations used in the bearing design. Fig. 1 shows the theoretical friction reduction factor γ as a function of the velocity ratio along with experimental results.

Within the experiment the drive was used for superposing oscillations onto the feed velocity. However, considering the bearing design, the goal is to reduce the oscillation amplitudes to minimize interference with the macroscopic motion. Using ultrasonic, the frequency can be increased significantly while at the same time the amplitude can be small (2).

Table 1 gives examples of oscillation parameters to achieve a reduction factor of 0.3 at a feed velocity of 100 mm/s.

Table 1
Oscillation parameters and friction reduction.

Friction reduction example and oscillation parameters 70% reduction → $\gamma = 0.3 \rightarrow v_f/v_s = 0.45 \rightarrow v_s = 220$ mm/s			
f_s	300 Hz	3 kHz	30 kHz
x_s	116 μm	11.6 μm	1.2 μm

Table 1 shows that reducing the oscillation amplitude to the order of a few microns, ultrasonic frequencies >20 kHz are required. For the bearing design, ultrasonics allows reaching high vibration velocities with only little oscillation amplitudes. For generating oscillations within these dynamic ranges, piezo actors are most suitable. Hence, the design goal is to integrate a piezo actuator that can vibrate the sliding surfaces of a sliding bearing.

3. Bearing design

The design goals of the adjustable friction bearing are high friction reduction properties and high stiffness. To achieve high stiffness the load forces need to be transferred to the guiding rail without any bending elements. At the same time, the contact region must be able to vibrate within the plane of the guiding rail to achieve high vibration velocities. Fig. 2 shows our concept of an adjustable friction setup on a prismatic rail. Normal and side forces are transmitted onto the rail going from the mounting plate directly through the sliding area onto the guiding rail, but not through the piezo stacks. The vibration module contains two resonators, piezo stacks with electric contacts and bolts to apply in-between the inertial masses of the resonator and the piezo actors. Both stacks of piezos are used to vibrate and excite the longitudinal mode of the resonator. The vibration waves travel from the inertial masses to the resonator tip. The change of intersection from the inertial masses to the resonator tip amplifies the vibration amplitude.

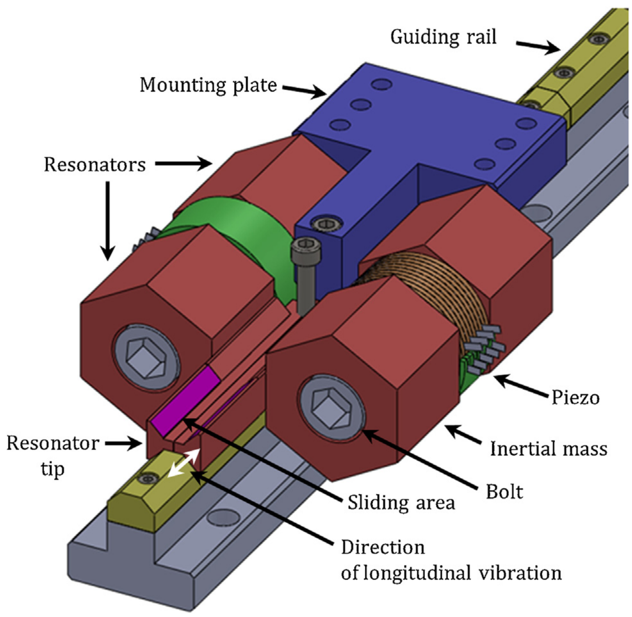

Fig. 2. Components of a prismatic adjustable friction bearing [5].

The operating point of the adjustable friction bearing is defined by the frequency and voltage amplitude. Limits for the operating point are given by the piezo amplifier; in our case the possible frequency is limited to 20–30 kHz. Placing the resonance frequency within the specifications of the amplifier therefore is another design goal. For analysing the vibration modes a FE-model is created that simulates the behaviour of the vibration module. Important model parameters are derived (Fig. 3) and ranges for the parameters are defined.

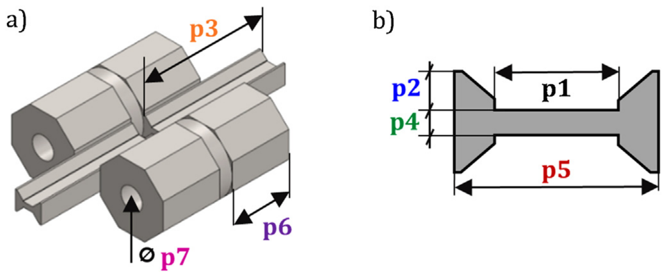

Fig. 3. Geometric parameters of the vibration modules.

Harmonic analyses are run to calculate the variation of the first longitudinal frequency due to a change of the geometric parameters. This knowledge is used for, designing the bearing to match a certain resonance frequency within the amplifier bandwidth and for adjusting the resonance of the built bearing to compensate manufacturing tolerances. To shift up the resonance frequency, for example, the sensitivity of the resonator tip length (gradient of p3) is −282 Hz/mm or 314 Hz/mm for the sensitivity of the length of the inertial mass (p6). Thus it is possible to shift up the resonance frequency by cutting short p3 or shifting down the resonance by cutting p6 (Fig. 4).

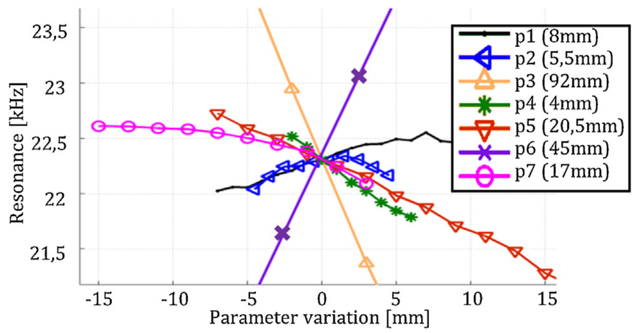

Fig. 4. Influence of parameter variations on the resonance frequency using FE simulation (initial model parameters in brackets).

4. Experimental results

A prismatic and a flat profiled bearing were designed and built using steel for the resonators. Fig. 5 shows the assembled prismatic

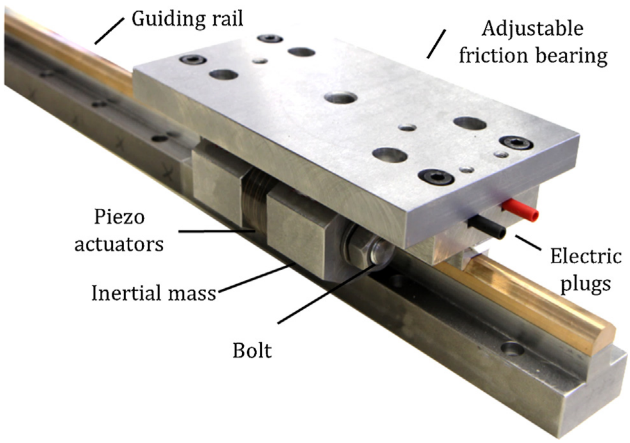

Fig. 5. Single adjustable friction bearing with prismatic rail profile.

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