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Initial investigations of rolling friction characteristics in planar ball guides with a novel measurement set-up

M. Heyne[1], H.Mehner[2], T.Erbe[1]and R. Theska[1]

1 Ilmenau Univ. of Technol., Fac.of Mechanical Engineering,Institute of Design and Precision Engineering, Gustav-Kirchhoff-Platz 2, 98693, Ilmenau, Germany
2 Ilmenau Univ. of Technol., Fac.of Mechanical Engineering, Dep. Micromechanical Systems, Max-Planck-Ring 12, 9893, Ilmenau, Germany
Corresponding Author / e-mail: hannes.mehner@tu-ilmenau.de, TEL: +49-3677-69-1860, FAX: +49-3677-69-1840

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Detailed knowledge about the rolling friction behaviour is a requirement for the mechanical design and development of control systems for ball guided precision drives. The planar ball guide is an attractive alternative to existing guides and offers many advantages for high precision positioning applications. Existing knowledge derived from linear ball guides cannot be applied because the properties of the rolling contact are different. There is no adequate rolling model for planar ball guides currently available. Consequently, this contribution is focused on an experimental investigation, especially concerning the breakaway characteristics of the balls.

A measurement set-up which allows the simultaneous measurement of these rolling resistance forces on the runner and its displacement is presented. The central aim is the measurement of the varying friction force acting on the runner. For this purpose the measurement of forces inmillinewtons range and displacements in the submicron range is required. Because of these high resolution needs, measurement disturbances and their transmission, especially by the system couplings, are a major concern and their reduction is an important challenge. First measurement results of these experimental investigations are shown and discussed.

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NOMENCLATURE

s = displacement
F_{roll} = friction force
F_N = normal load
a_K = radius contact area of Hertz

1. Introduction

Detailed knowledge about the friction behaviour of guides is crucial for the design and control of positioning systems [1], which is especially important forhigh precision applications (e.g. as wafer handling and measurement systems). Thoseapplications demand planar movements with a high standard of reproducibility and resolution in challenging environmental conditions (e.g. vacuum).

The preferred solutions are stacked linear (mostly ball) guides and vacuum compatible aerostatic guide elements. The deployment of aerostatic guidance systems necessitateselaborate and cumbersome extraction systems for usage within a vacuum environment [1, 2].Stacked linear guides show low as well as varying stiffness and are characterized by a fluctuating balance point [3].

A concept that overcomes these drawbacks is the planar ball guide. It consists of two flat surfaces and (multiple) balls arranged in a flat cage in between (see Fig. 1).

Thanks to its functional principle, the planar ball guide avoids all the major disadvantages of stacked linear guides. A high stiffness of the guide, a compact design of the whole motion system and a movement of therunnerin three DOF relatively to the fixed guide surface can be achieved [4]. Furthermore,unlikeaerostatic guides, the planar ball guide does not need any operating gas.

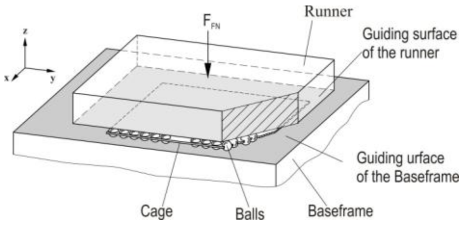

Fig.1Basic set up and main elements of the planar ball guide

An essential feature of the guide is the ball-plane contact.To achieve a high stiffness and a high load capacity more than three balls are preferred. Due to the large number of balls the kinematic position of the runner is over determined. Shape deviations of the balls andplaneslead to a pattern of load taking balls. The quantity and distribution of load bearing balls is strongly dependent on the position of the movement..

Every loaded ball is in contact with both planes of the guide, resulting in two ballplane contacts per ball. The axis of rotation of the rolling elements is parallel to these planes for arbitrary linear movements. Therefore, in contrast to ball-V-groove contacts,drilling friction does not occur. Furthermore, the differential slip of ball-plane contacts is considerably smaller than in ball-V-groove contact situations. These differences render the adaption of experimental findings from previous investigations to planar guideways impossible and require further investigations of the rolling friction behaviour of ball plane contacts.

Rolling friction of ball contacts is mainly caused by the deformationof balls and planes. Those deformations are not completely reversible,as well as the resulting existing energy dissipation [5, 6,7]. Energy dissipationscaused by micro-slip friction, plastic deformation or work against adhesion forces are of minor importance in planarguides because the normal stresses are commonly much smaller than the yield strength of the used materials. Furthermore, the speed of the runner is comparably small, too.

The focusof this paper is the rolling friction behaviour ofball-plane contacts in planar rolling guides.The aim of the measurement is on the one hand, the derivation of a realistic friction model for the controller design of planar motion ball guided systems – taking into account the requirements due toenvironmental limitations of the semiconductor technology applications. On the other hand, the measurement results for the force-displacement interrelation will be used to determine the maximum capacity of the rolling elements and guiding surfaces. Furthermore rule sets or principles for the design of planar ball guides will be concluded addressing various factors (e.g. ball diameter, material combinations) affecting rolling friction behaviour.

2. Rolling friction and breakaway stage –current findings

The distinguishing characteristic of rolling friction behaviour is the motion hysteresis (see Fig. 2) [9, 10].

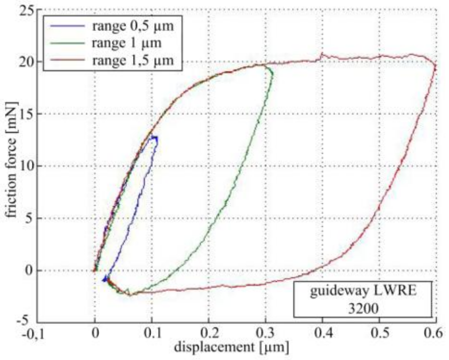

Fig. 2 Motion hysteresis of a linear guideway LWRE 3200 [9]

According to [5] the resisting moment to rolling motion of a ball-plane contact can be estimated by using Hertzian equations as follows:

$$M_{roll} = (3 \cdot F_N \cdot a_K)/16. \qquad (1)$$

with anormal force $F_N$in the ball-plane contact point and the radius of the contact area between spheres and plane a_K.

To reach an excellentpositioning resolution the friction coefficient is less important than the course of the friction force itself.

Based on previous studies concerning rolling friction and the breakaway behaviour of linear rolling guides (see Tab. 1), three areas of incipient rolling friction can be determined (see Fig 3). For this contribution the breakaway motion is defined by the displacement at the transition from the 2nd to the 3rd stage.

Tab. 1 State of the Art studies and results concerning breakaway force

test setup	rolling contact	breakaway path [µm]	normal force [N]
[10]	not documented	100	580
[11]	line contact roller-plane	20 – 55	25 – 290
[12]	point contact – ball in v-groove	40 – 60	46,5 – 123,2

The divergence of the results presented in Tab. 1 can be ascribed to different types of rolling contact and normal loads, highlighting thesignificant influence of those factors to the rolling friction measurement and the breakaway behaviour in particular.

Experimental studies have further shown that the use of lubricants (e.g. grease or oil) have an increasing effect on the rolling friction coefficient of ball-plane contacts [5, 8].

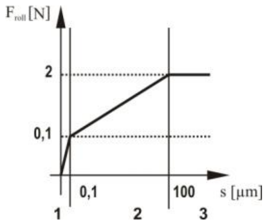

Fig. 3 Characteristic of friction-force-displacement curve [12]

Yet, the qualitative characteristic with the three stages of breakaway behaviour of rolling guides are assumed to be similar for planar guideways as well. However, the two ballplanecontacts per ball with:

- two - in approximation parallel - planes and
- geometrically not defined tracks

cause differences in the sliding friction to rolling friction ratio compared to ball-V-groove contacts and a totally different running-in behaviour.Resulting from these reasons different rolling friction forces and rolling friction coefficients have to be expected. The friction coefficient for ball-plane contacts is given in literature as 10^{-5} [8].

Due to the low friction coefficient, minimal values for resistance force and displacement, especially inthe breakaway stage can be expected [8, 12]. This causes major challenges to minimize disturbances and parasitic influences in the measurement setup.

3. Measurement approach and setup

The aim was the investigation of rolling friction behaviour of a planar ball guide.The resistance to motion of ball guides includes different influences, such as:

- the sliding friction between the balls and the cage,
- the sliding friction between the guide surface(s) and the cage,
- the fluctuations of load of the balls by changing of the load conditions during the motion (e.g. caused by displacement of the centre of gravity of the runner with respect to the balls during motion)
- the fluctuations of load of the balls due to external forces e.g. from the drive chain

With regard to the three DOF of the runner in a planar ball guide there are further influences due tosupporting- and auxiliary-guides as well as by anti-twist measures. In contrast to linear guides an influence by present plastic deformations of both guide surfaces and balls caused by utilization history of the guide can be expected.

In order to measure the onset of motion the following effects are important and have to be taken into consideration:

- lubricants and their effectiveness during small and reversing displacements
- surface quality of the balls and the guide surfaces

Shock or jerk forces and moments caused by the drive chain (e.g. by backlash) at the beginning and – if considered – the reversal point of the movementas well as tilting moments caused by reaction forces of the rotor generatefurther disturbances.

The reduction of these influences was the main focus during the design of the test setup and will be discussed in the following.

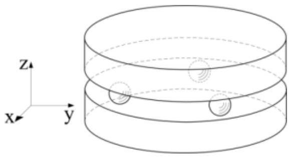

Fig. 4 Minimal configuration of a planar ball guide

To enable the portability of the results, a planar bearing guide reduced to the possible minimum of three balls was used for the measurement setup (see Fig. 4).This setupon the one hand, renders a ball cage unnecessary and avoids the cage related friction. On the other hand, the symmetrically distributed balls bear the same load, which allows a consistent and comparable load situation of all balls within small motion ranges.

The influences of additional guides are of an equal or higher magnitude compared with the expected friction forces of a planar ball guide.Therefore supporting guides have been avoided in contrast to former approaches [12].

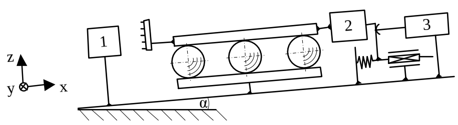

Fig. 5 Principle setup for the investigation of rolling friction with: 1: interferometer; 2: force sensor; 3: piezoelectric actor

Through the tilting of the entire test setup the runner and the balls are inclinedin relation to the gravitational field (cf. Fig.5 and Fig. 6). This arrangement leads to a force vector with stable orientation and magnitude thus defining a constant moving path and preload. In addition, the inclination allows preloading all kinematic couplings. The measurement is therefore not distorted by backlash and the force sensor can be operated with a favourable offset.

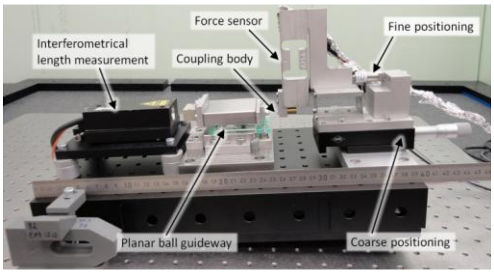

Fig. 6 Measurement setup for the investigation of rolling friction

The force is measuredby a precision load cell(from a precision scale) based on strain gauges. The path of motion is optically measured by a single-beam interferometer. The force is applied by a piezoelectricdrive, directly acting on the load cell. Interferencesfromexternal forces (e.g. by the characteristic line of the load cell,which has to be regarded as a spring guide), was minimized by the elastic coupling between the runnerand the load cell.

4. First measurement results

To conduct the first evaluation series to proof the suitability of the test setup for measurement tasks, the following conditions were used:

- the guiding surfaces of the runner and the fixed base are made of stainless steel with
- overall geometric deviations of 4 µm (flatness, waviness, roughness),
- the balls as rolling elements have a diameter of 4 mm and are made of hardened steel,
- the normal force of approx. 1,5 N on every ball plane contact is applied by the runner weight of 450 g.

Based on the rolling friction values and the noise in force values, the measurement dataof the 1st and 2nd stage cannot be utilised. This is caused by an inconvenient signal to noise ratio and will be addressed in ongoing work by further optimization of electronic components or a new force sensor not based on strain gauges.

However the measurement setup, allows first time to investigate the 3rd stage of the rolling friction characteristic (Fig. 7 and 8). The measured curves vary depending on the measurement position of the runner. Despite this, the curves canbe confirmed with a high reproducibility. The dependence of the measurement results on the position suggests a strong influence of geometric deviations. These geometric deviations can be estimated as the surface roughness.

The roughness changesthe position of the runner normal to the gravity field, which creates an additional force on the force sensor and explains the fluctuation of the force values shown in the diagrams

(Fig. 7).

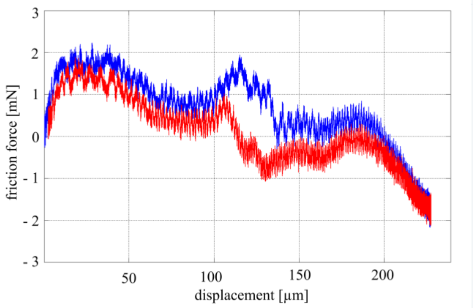

Fig. 7 Friction-force behaviour measured for 225 µm runner displacement – position 1

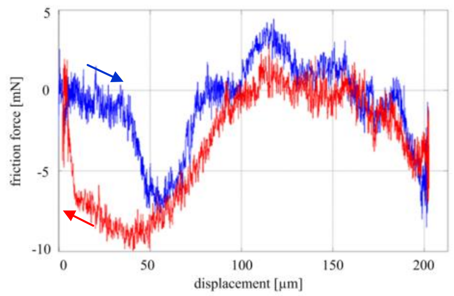

Fig. 8 Friction-force behaviour measured for 205 µm runner displacement– position 2

The friction-force is determined by the force difference between the forward (traversing from 0 micron to 225 micron) and the backward (traversing from 225 micron to 0 microns) trace. High signal noise of the force measurement did not allow an accurate analysis of the rolling friction across all three stages of the onset of rolling friction.

5. Conclusions

In this contribution the concept of the planar ball guideway for high precision applications was presented and the advantages compared to existing conventional guides have been highlighted. The reasons that prevent the application of the current findings on rolling friction and breakaway behaviour to planar ball guideway design have been pointed out.

The presented measurement setup is suitable for experimental investigations of the rolling friction of the contact points of three balls with two plane guiding surfaces.

The efforts for minimising parasitic effects inthe measurement setup, which are necessary to meet the required accuracy, were discussed.

The reduction of the signal to noise ratio in the force measurement is currently the focus of ongoinginvestigations. The focus of future work will therefore be the optimisation of the force sensor and the coupling between the sensor and the runner.

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