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Hydrostatic bearings with micro gap sizes – a comparison between theoretical model and measurements

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Abstract

Hydrostatic bearings with a gap size between 10 and 60 µm are state of the art in precision engineering applications. For the use of micro gap sizes less than 10 µm theoretical models still have to be validated by experiments and eventually modified. For this reason the company KERN has developed an experimental hydraulic test setup. With this test setup it is possible to evaluate parametric models by static and dynamic measurements of hydrostatic bearings with micro gap sizes. This paper is a comparison between a new parametric model and measurements. The model is designed regarding a modular structure and for investigation in the influence of different parameters. It can be shown that the structure of the model coincides with the measurements. The parameter variation clarify the main influences on the dynamic behaviour of a hydrostatic bearing like the volume of the pocket.

High precision machine tools, hydrostatic bearings, micro gap size, modelling, sensitivity analysis

1. Introduction

Hydrostatic guideways have many advantages. To further improve its stiffness and damping, micro gap sizes < 10 µm should be realized. Since these are not state of the art yet, measurements on a test setup and a model are created and compared in this paper. The measurements help to understand the different dynamic behaviour and different characteristics of standard hydrostatic bearings compared to bearings with micro gap sizes. The model is explained in section 2. The test setup on which the measurements have taken place are presented in section 3. The model is compared to the measurements in section 4. In section 5 an example of the parameter analysis is presented.

2. Theoretical modelling

The model [1] simulates one bearing pad with a capillary as flow restrictor (red dashed area in **Figure 1**). An oscillating mechanical mass, a constant pump pressure, a constant preload and constant fluid characteristics are assumed.

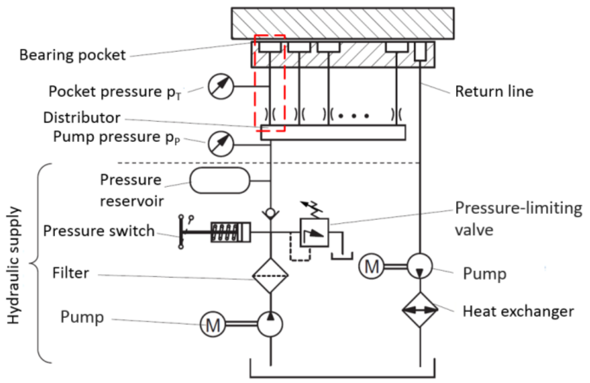

Figure 1. Hydraulic circuit and system limits of the model, see [5].

2.1. Static variables

All static outputs are based on the hydraulic resistances of the bearing gap and the capillary. The pump pressure p_P, the static pressure p_{To} in the bearing pocket, the static load F_0, the effective land area A_{eff}, the hydraulic resistance R_C of the capillary and the static resistance R_{To} of the bearing gap are taken into account to calculate the initial gap size h_0 in equation 1.1.

$$h_0 = \sqrt[3]{(p_P - p_{T_0}) \cdot R_0 \cdot R_C \cdot A_{eff}/F_0} \quad \text{with } R_0 = R_{T_0} \cdot h_0^3 \qquad (1.1)$$

2.2. Dynamic model

The dynamic model (**Figure 2**) is also based on the hydraulic state variables of the system. The central element is the capacity of the hydraulic volume between the flow restrictor and the bearing gap, which changes the pocket pressure p_T depending on the volume flows through the pocket and the capillary. The resulting force of the pocket pressure acts on the moving mass. This force is used in the motion equation as stiffness force of the pocket.

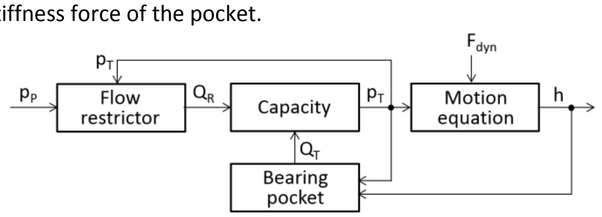

Figure 2. Structure of the dynamic model.

2.3. Characteristics

The compression of the fluid is modelled by the capacity of the hydrostatic bearing. The inertia of the fluid is modelled by the hydraulic inductance of the fluid in the flow restrictor. By observing these two effects, the model represents a mass-spring-Maxwell system, even without a mechanical mass [4].

3. Hydraulic test setup

Specifically for studies on micro gap sizes a test setup was developed during a master thesis [2]. The dynamic force is generated by a piezo force actuator mounted above the bearing plate. The gap height is measurement by a capacitive sensor below the bearing plate. **Figure 3** shows the test setup in section with its actuators and sensors. The dynamic frequency response up to 5 kHz of the test setup is known and shows only its static stiffness up to more than 1 kHz. The first eigen frequency is at 2,5 kHz, so that the following dynamic measurements only taken till 2 kHz. Under this frequency, no relevant influence from the dynamic behaviour of the test setup is assumed [1,3].

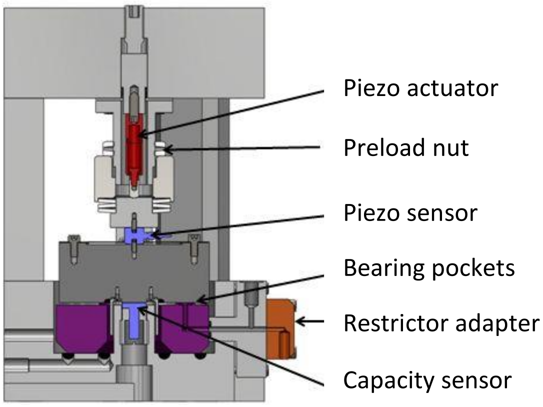

Figure 3. Test setup in section.

4. Comparison

The red lines in **Figure 4** show the results for the dynamic measurements [3] on the test setup for oil VG 32 as pressure fluid. The linearized model [1] is depicted in black lines with pure pocket effects demonstrated in dashed black lines and including the static stiffness of the setup in solid black lines.

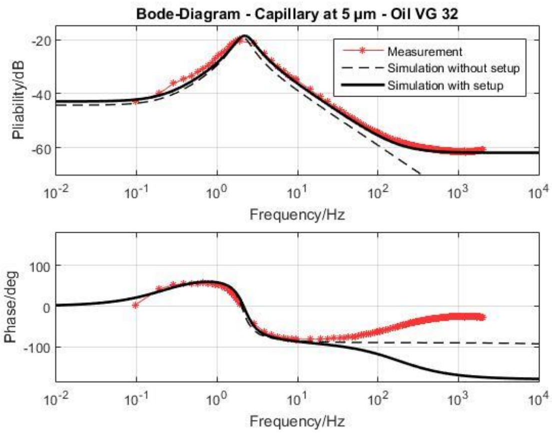

Figure 4. Comparison between model and measurements.

It can be seen that if the stiffness of the test setup is included in the model the amplitude can be represented very well. In addition, the phase seems to be mirrored from the two zero points through the test stand. This difference cannot be determined in the medium of water and appears either to be a linearization error in the simulation or an unconsidered influence of the oil, which is not represented by the model. Nevertheless, the quite accurate corresponding results of the simulation and the measurements show that the model approximates the reality very well.

5. Local sensitivity analysis

A local parameter analysis is made. In the following an example is presented. It was found out that the volume between the capillary and the bearing is an important parameter. The analysis in **Figure 5** shows the frequency response with different pocket volumes V. In addition to slower dynamics, the strongly decreasing damping and the strongly increasing amplitudes are clearly visible when volume is increased. In this way, the phase sequence becomes more positive and starts to increase at lower frequencies. The important influence of compression in the model shows that the assumption of an incompressible fluid is not permitted in the case of micro gap sizes.

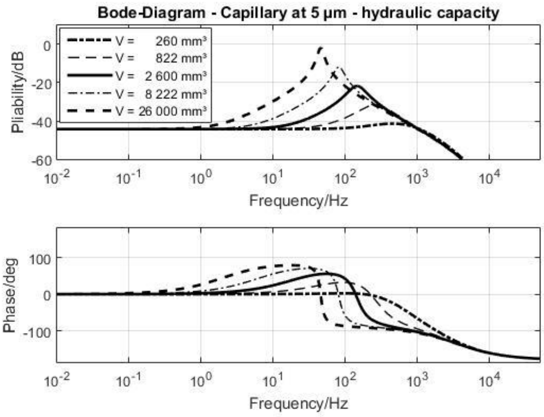

Figure 5. Capacity analysis.

6. Conclusion and future work

It can be shown that the model approximates the measurements well. Therefore, the model could be validated. The analysis of various parameters show that some simplifying assumptions of ordinary gap sizes are not transferable to micro gap sizes under 10 µm. The created model takes this fact into account. It describes micro gaps better than previous models, mainly because it can be validated by measurements in the area of micro gap sizes. Thanks to its modular design, the model can be extended by other types of flow restrictors. It can also be duplicated and used to simulate an entire machine tool axis with multiple hydrostatic pockets. Important influences such as the undissolved air in the fluid can be described as a function of the pressure, adding even more effects to the model.

References

[1] Purkart S 2017. *Dynamische Modellierung von hydrostatischen Lagern mit Mikrospalthöhen*. Bachelor Thesis (Munich University of Applied Sciences)

[2] Adolf M 2012. *Konzipierung und Erstellung eines Prüfplatzes für hydrostatische Lagertaschen.* Master Thesis (Munich University of Applied Sciences)

[3] Reiter M 2015. *Dynamische Untersuchung von hydrostatischen Lagern mit Mikrospalten*. Diploma Thesis (Munich University of Applied Sciences)

[4] Pollmann E and Vermeulen M 1989. *Compressibility and inertia effects on the dynamic behaviour of recessed hydrostatic bearings*. Fillon M, Liang H and Liu W. *Tribology International* **22** (Amsterdam: Elsevier) pp 166-176

[5] Weck M 2006. *Werkzeugmaschinen. Konstruktion und Berechnung.* **8** (Berlin: Springer)