Test setup for the characterization of hydrostatic bearings with micro gap sizes

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Abstract

Hydrostatic and aerostatic bearings are state of the art in precision engineering applications. These hydrostatic bearings are working with a gap size between 15 and 60 µm whereas aerostatic bearings are working with a gap size ranging from 2 to 10 µm. For the characterization of smaller gap sizes less than 10 µm by experiments the KERN company has developed a test setup to evaluate the static and the dynamic behaviour of hydrostatic bearings with micro gap sizes. This paper presents the design of the test bench.

1. Introduction

Smaller gap sizes in hydrostatic bearings improve the stiffness, the damping and reduce the power losses of the bearing. For aerostatic bearings it is state of the art to handle the close machining tolerances, so this should also be possible for hydrostatic bearings. The test set ups for the characterisation of the hydrostatic gaps in the literature are not suitable for gap sizes smaller than 10 µm, in the following called micro gap sizes [1,2].

2. Requirements for the test bench

The test bench is designed for the static and dynamic characterization of hydrostatic micro gaps. The static analysis should be possible up to a load of 6 kN per pocket resulting in a bearing pressure of 10 MPa. The additional dynamic load shoud be 500 N peak to peak with a frequence up to 2500 Hz. During the experiment the gap size, the load, the supply and the pressure should be measured.

Different hydrostatic compensating devices, e.g. fixed restrictors and variable restrictors like diaphragm controlled valves, should be able to be tested on the test bench. Also the hydrostatic pocket geometry should be replaceable.

Because of the micro gap size the alignment of the bearing part has to be very accurate to avoid effects of non parallel gaps.

3. Design of the test setup

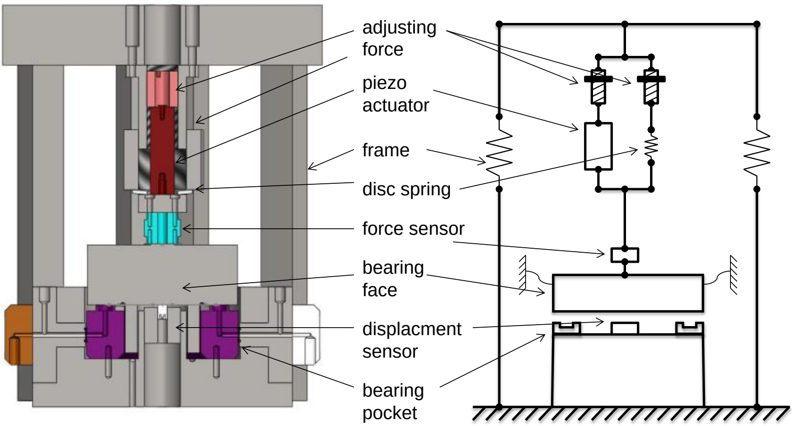

Figure 1: Test setup design

Figure 1 shows the design of the test setup. The first interesting point is the application of the load.

On the one hand the static load on the hydrostatic bearing has to be constant over the changing gap sizes. One the other hand the actuator for the dynamic load needs a very stiff coupling to reduce power losses at high frequencies. For this reason a parallel load application is selected for the test bench design with a soft disc spring coupling for the static force and a direct stiff coupling for the piezo actuator. The two forces are merged together and are measured by the force sensor before they are applied to the bearing face. The alignment of the bearing face is difficult, because of the micro gap size.

3.1 Self alignmet of the bearing face

The bearing face is only guided by a leaf spring. This spring fixes the position in the bearing plane and the rotation around the force axis. There is no additional fixing of the rotation around the axis in the bearing plane. Because of this guidance for the bearing plane, the bearing gap is self aligned. A second advantage is the negligible influence of the leaf spring guide way to dynamic behavior of the hydrostatic gap. The gap is mesured by a capacitive displacement sensor, which is in the center of the 3 bearing pockets, see figure 2.

3.2 Modular design

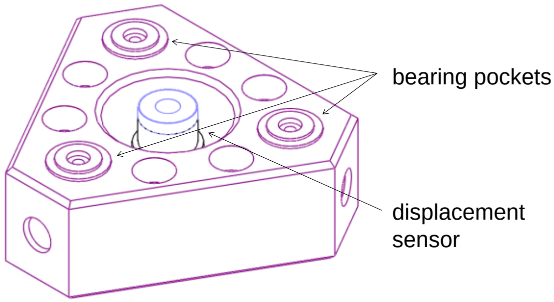

Figure 2: Test setup design

The bearing pockets are in one plane so it is easy to manufacture the part by lapping. It is also possible to replace the pocket modul by another pocket modul to study the effects of different pocket geometries and dimensions. The compensating device for the control of the oil flow is replaceable, too. So it is possible to experiment with different restrictor and diaphragm controlled valves. The disadvantage of the modular design is that the force loop and the measurement loop are not completely separated. Because of this the stiffness of the common loop has to be mapped and compensated.

4. Experiments

To map the stiffness of the common parts and the force loop of the measurement the oil supply is turned off and different static loads are applied to this contact situation. The load and the displacement of the bearing face are measured. The data can then be fitted with polynomials by using the method of least mean squares.To evaluate the dynamic capabilities of the set up the transfer function from piezo input signal to the force sensor output signal is measured.

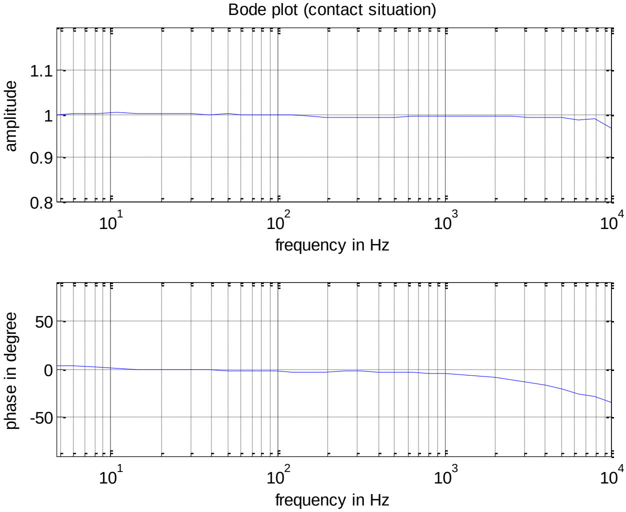

Figure 3: Bode plot in contact situation

Up to a frequency of 2500 Hz, there is only an amplitude decrease of less than 1 % detected, see figure 3. The phase shift of -15 degree is due to the lag element characteristic of the piezo actuator.

5. Conclusion

The paper presents the design for a test setup to characterize hydrostatic bearings with micro gaps under static and dynamic loads. In the test the dynamic capabilities are proven and the next step will be the characterization of different bearings and compensating devices in micro gap application.

References:

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