TOOL FOR THE DESIGN AND SIMULATION OF HYDROSTATIC BEARINGS IN MACHINE TOOLS

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Hydrostatic Bearings, in addition to ball bearings and sliding guides, represent a possibility for machine tools to bear moving parts precisely. Compared to the aforementioned types of bearings, hydrostatic bearings are nearly frictionless, have no stick-slip effect, and exhibit very good damping behavior. However, hydrostatic bearings have higher costs due to the complex construction involved. In order to support a hydrostatic bearing with a constant pressure support, a pre-throttle is required. The type of pre-throttling used results in different characteristics of the bearing. A design tool has been developed to characterize various methods of pre-throttling hydrostatic bearing arrangements for machine tools. This tool allows for the selection of the type of pre-throttling, the geometric dimensions of the bearing, and real or assumed load cycles. As a result, the actual behavior of the bearings in the machine tool can be determined during the design phase.

Keywords:
hydrostatic bearing, machine tool, design, modelling and simulation, results

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1 Intruduction

oiseuille Equation, a linear pressure drop across the gap is assumed, as illustrated in Figure 2. This assumption holds true for rectangular pockets, with the exception of the corners of the pockets.

A hydrostatic bearing can exclusively withstand pressure forces on the bearing. To accommodate both tensile and pressure forces, two hydrostatic bearings must be employed (see Figure 1. As a result, the tool slide is clamped and maintained at the center of the two bearings, disregarding its own weight.

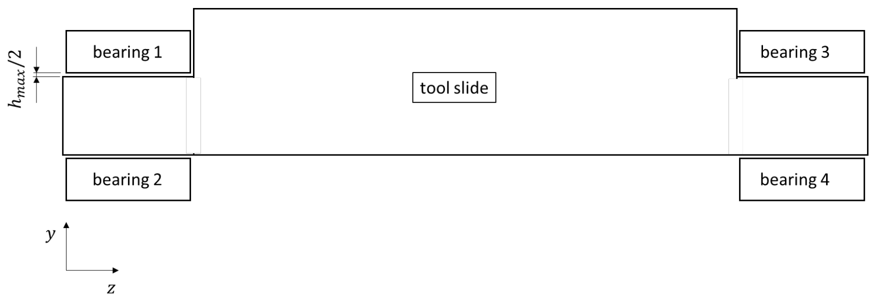

Figure 1: Sketch tool slides of a machine tool.
Source: own

For linear tool slides, mostly rectangular pockets are used. The characteristics of the bearing can be described by the Hagen-Poiseuille Equation for rectangular pockets (see Equation 1). This equation establishes the relationship between the flow rate Q through the bearing, the pressure drop across the bearing Δp, and the geometric dimensions of the bearing b, h, and l. It is notable that the gap height h enters the flow equation with a cubic exponent, and the viscosity η also influences the flow. While the gap height depends on the load and the type of pre-throttling, the viscosity depends on the medium and its temperature.

$$Q = \frac{b\, h^3 \Delta p}{12\, \eta\, l} \tag{1}$$

To support a tool slide in one direction (x, y or z direction), at least three bearings are necessary; normally, four bearings are used. Therefore, the bearing arrangement for one direction consists of a total of 8 hydrostatic bearings, which are combined into four bearing units (see Figure 2). The forces on the individual bearing units can be determined by establishing static equilibrium. To describe the mathematical relationships, only one bearing unit is considered. Figure 2 shows a sketch of a hydostatic bearing and the pre-throttle. The pre-throttle is neccessary to reduce the system pressure of the constant pressure system to the load pressure of the bearing. This system is known in electrical engineering as a voltage divider.

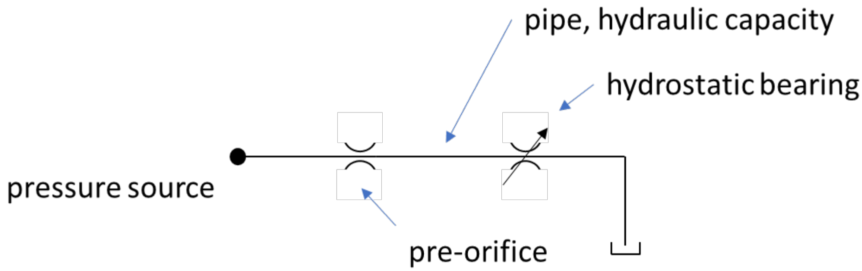

Figure 2: Hydraulic circuit diagram, hydrostatic bearing.
Source: own

The result is a system with a constant pre-throttle (not when a progressive quatity regulator is used) and a variable throttle (hydrostatic bearing). Which type of throttling is used has an affect on the behavior of the hydrostatic bearing. Four different possibilities for pre-throttling can be distinguished:

- by a capillary,
- by a throttle,
- by a orifice,
- by a progressive flow regulator.

Each of the four different types of throttling has distinct characteristics, leading to varying behavior in response to load changes of the hydrostatic bearing.

The capillary can be regarded as a thn tube with a laminar flow inside. The flow depends on the pressure drop and on the viscosity of the fluid, see Equation 2. This provides an advantage when the temperature of the medium is changing, as the flow through the hydrostatic bearing is also dependent on viscosity.

$$Q = \frac{\pi r^4 \Delta p}{8 \eta l} \quad (2)$$

If an orifice is used as a pre-throttle, the flow depends on the geometry of the orifice and the density of the fluid. Additionally, the pressure drop across the orifice follows a quadratic relationship in the equation (refer to Equation 3). One advantage of using an orifice is that it can be placed very close to the hydrostatic bearing, thereby minimizing the dead volume between the orifice and the hydrostatic bearing.

$$Q = \alpha \, A \sqrt{\frac{2 \, \Delta p}{\varrho}} \tag{3}$$

When achieving a load-independent gap height, a progressive flow regulator must be employed. The flow through the hydrostatic bearing then follows the characteristic outlined in Equation 1. In our case, a flow compensator is utilized, as presented by Mörwald [7]. This involves a variable cylindrical annular throttle. The length of this throttle depends on the force equilibrium between a spring and the pressure on the face side of the spool. The gap height is variable and needs to be calculated separately for each distinct hydrostatic bearing.

The shape of the spool can not be calculated analytical, it must be calculated discret. Base of the eauation is the force equilibrium in the spool of the progressive flow regulator. To each pressure drop over the spool a position of the spool is given. Beginning with a start discredization (zero overlab gives a divition by zero) and the equation of continuity each new gab height can be calculated out of the old gab height and the next discret step. Therefore it is necessary to calculate the pressure drop to the last step then the difference between the pressure drop of the last step and the aktualy absolute pressure drop. With this result the new gap height of the aktual step can be calculated.

The actual fow for each step can be calculated with Equation 4. The pressure drop will be quantsized with the same number as the quantization of the maximum stroke of the spool.

$$Q_i = \frac{b \, h^3 \Delta p_{b,i}}{12 \, \eta \, l} \tag{4}$$

Now the pressure drop of the progressive flow regulator must be calculated.

$$\Delta p_{pfr,i} = p_p - \Delta p_{b,i} \tag{5}$$

The nex step is to calculate under the assumption that the flow in each quatization is the same the pressure drop to the last step of calculation $i-1$,

$$\Delta p_{pfr,n=i-1} = \sum_{n=1}^{i-1} \frac{12\, Q_i\, \eta\, \Delta l}{d_{spool}\, \pi\, h_n^3} \tag{6}$$

where $d_{spool}$is the diameter of the spool and h_n is the gap height of the discrete gap.

In the next step the pressure drop of the actual step van be calculated.

$$\Delta p_{pfr,n=i} = \Delta p_{pfr,i} - \Delta p_{pfr,n=i-i} \tag{7}$$

At last the gap height of the actual step i must be calculated.

$$h_i = \sqrt[3]{\frac{12\, Q_i\, \eta\, \Delta l}{d_{spool}\, \pi\, \Delta p_{pfr,n=i}}} \tag{8}$$

The result of this calculation is exemplary shown in Figure 3. In this case is the diameter of the spool 6 mm and the stroke also 6 mm.

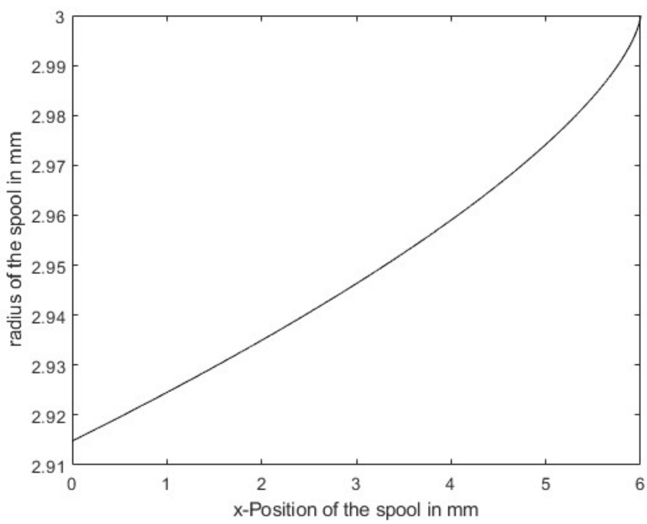

Figure 3: Shape of the gap between spool and housing of the progressive flow controller.
Source: own

To describe the transfer behavior of a hydrostatic bearing a hydraulic-mechanical model must be developed. The hydraulic part of the system will be described with hydraulic resistances, capacitances and inductances. The mechanics of the hydrostatic bearing will be described with Newton's law of motion. In Figure 4 can be seen the model of a hydrostatic bearing consisting of the hydraulic network and the mechanic which is described in [7].

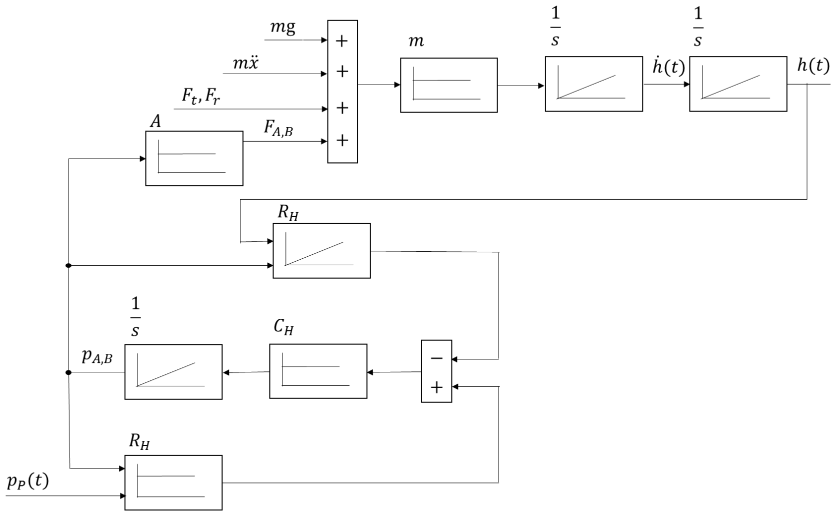

Figure 4: Equivalent circuit, hydrostatic bearing.
Source: [8]

This system is implemented in the Siemens AMESIM software to construct both the hydraulic and mechanical components of a hydrostatic bearing system. The oil volume between the pre-throttle and the hydrostatic bearing is also considered.

3 Model

The AMESIM model consists of different parts. The first one is comprised of the global variables. These variables define the general conditions of the system. A distinction is made here between geometric variables and hydraulic variables. The geometric variable defines the dimensions of the bearings, and the hydraulic variables define the hydraulic conditions.

After defining the global variables, the type of pre-throttling is selected. The precise geometric dimensions of the capillary or orifice are not necessary. The following step involves calculating the exact dimensions of the capillary orifice, which is achieved through the optimization algorithm of the AMESIM software. In situations where a progressive flow controller is employed, the geometry of the spool must be calculated. This task is accomplished using a second software tool, Matlab.

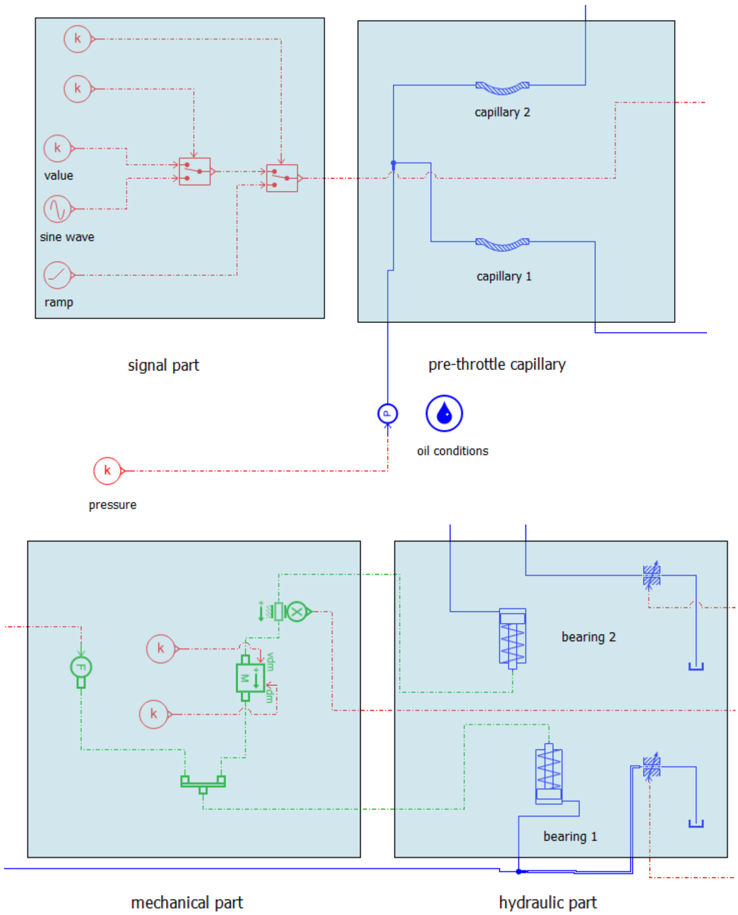

Figure 5: Model of the system hydrostatic beraing with a capillary as pre-throttle.
Source: own

Figure 5 depicts the model for the system with the capillary used as a pre-throttle. The model comprises four parts: the signal part, where the load on the bearing can be selected; the pre-throttle part; the mechanical part of the bearing; and the hydraulic part of the bearing. The hydraulic part of the bearing is divided into a hydraulic cylinder that manages the forces on the bearing, and the bearing gap which generates the oil flow through the bearing.

For the other two versions of the pre-throttle, only the pre-throttle block is displayed in Figure 6. The shape of the control edge of the progressive flow controller is pre-calculated using Matlab, and the flow characteristic is integrated into the sub-model, as depicted in the left part of Figure 6.

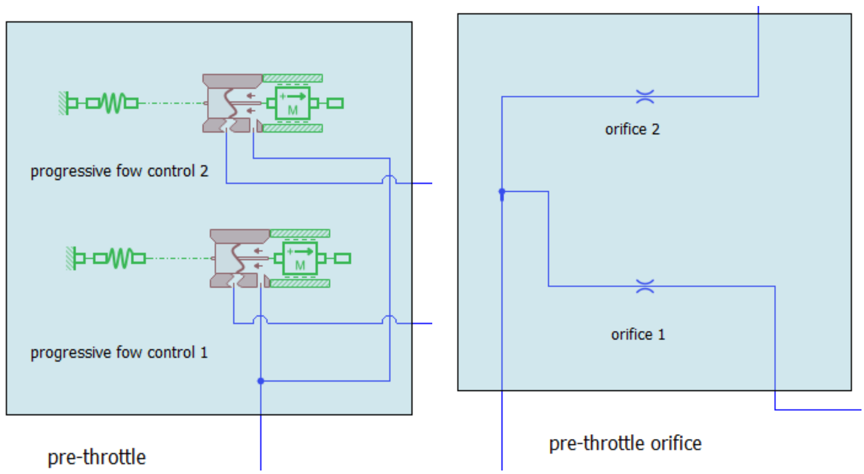

Figure 6: The different models for the pre-throttles.
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4 Results

To compare the various versions of pre-throttle in the hydrostatic bearings, two diagrams are employed. The first one illustrates the bearing gap in relation to the bearing load (Figure 7). It can be observed that using the capillary and the orifice as pre-throttles results in a variable gap height based on the bearing load. The profile of the gap height varies. On the other hand, the progressive flow controller maintains a consistently ideal gap across the middle of the two bearings, although some oscillations are visible in the curve. These oscillations stem from the spring-mass system of the progressive flow controller.

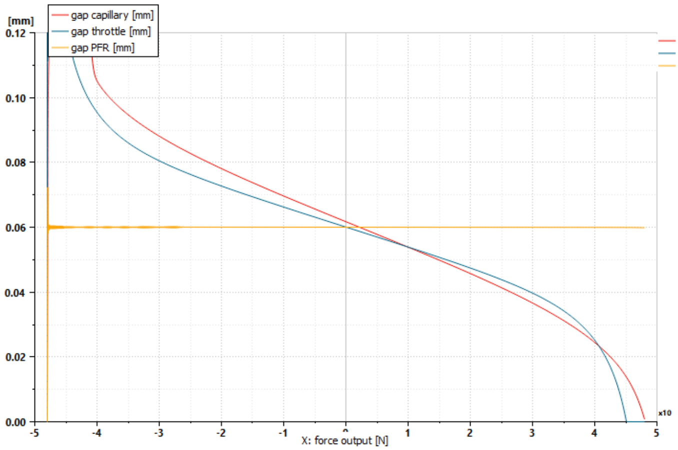

Figure 7: Compatison of the gap height over the load of tifferent types of pre-throttles
Source: own

In the second figure, Figure 7, a dynamic load with a frequency of 25 Hz is applied to the bearing. In this scenario, the two conventional pre-control versions result in an oscillating gap over the bearing's operational time. The amplitude of the gap oscillation is smaller when using the orifice as the pre-control method.

This can also be observed in Figure 6, where the curve's slope is shallower. Additionally, the progressive flow controller yields an oscillating gap with a significantly smaller amplitude. This outcome is due to the mass of the spool within the progressive flow controller.

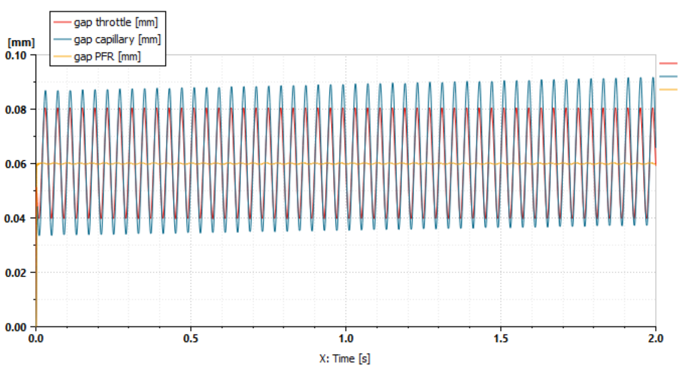

Figure 9: Comparison of the gap height with a dynamic load
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