A Bearing with Adjustable Stiffness for Application in Machine-Tools

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SUMMARY

The design of a hydrostatic bearing of which the radial stiffness can be adjusted within wide limits is described in detail. The bearing potential is such that it can be used in machine tools in two ways:

- By adjusting the stiffness the static deflections of the machine tool can be reduced or even compensated. This will result in improved geometric accuracy of products.
- The adjustability of the stiffness of the bearing offers the opportunity to quickly change the dynamic characteristics of the machine tool during operation. This will result in suppressing dynamic instabilities such as chatter in lathes.

Some recommendations for further work are given.

INTRODUCTION

An important function of bearings in a machine tool is the positioning of the shaft as independently of the machining forces as possible. This requirement leads to the application of very stiff bearings. When using hydrostatic bearings a high radial stiffness can be obtained by applying a high supply pressure. This, however, results in high energy consumption. Several methods to improve the radial stiffness of hydrostatic bearings have already been developed (Rowe [1], de Gast [2], Mayer [3]).

The application of very stiff bearings in a machine tool improves the geometric accuracy of the product, but mainly as far as static machining forces are concerned.
In the case of dynamic machining forces, the shaft - bearing - support system can be excited at a natural frequency, as a result of which appreciable vibration amplitudes may occur, decreasing the quality of the product.
Vibration problems can be solved by changing the natural frequency of the system until it differs sufficiently from the exciting force associating frequency. Until now, this solution has been applied in turbomachinery, but as far as we know, it has not yet been applied very much to machine tools.

Should bearings with adjustable radial stiffness be applied, the natural frequency of a dynamic system can be varied by changing the stiffness of those bearings.

The bearing described in this paper has originally been developed as a bearing with infinite stiffness. An additional property of this bearing is that its radial stiffness can be adjusted over a rather wide range in a simple way. This becomes more important should it be necessary to change the vibrational characteristics of the system with the machine tool operating.

NOMENCLATURE

F_s	lubricant film force on the shaft
f_s	lubricant film force on the shaft (dimensionless)
F_r	lubricant film force on the ring
f_r	lubricant film force on the ring (dimensionless)
h_b	radial clearance of the bearing
h_r	radial clearance of the ring
l_b	half axial length of bearing
l_r	half axial length of ring
p_d	pressure in the discharge channel of the auxiliary bearings
p_o	ambient pressure
p_r	lubricant pressure in the slot
p_s	supply pressure
q	lubricant film flow per unit width
ε	relative excentricity
μ	viscosity
index b	at bottom of bearing
index t	on top of bearing

OPERATING PRINCIPLE OF THE BEARING

Fig. 1 shows a simplified model of the "TNO adjustable stiffness bearing" (see also the patent description by Hirs [4]). The bush of the bearing is an accurate hollow cylinder with radial slots in the middle. A floating ring covers these slots and there is a narrow clearance between ring and bush. The positioning of the ring is described further on. The pressurized lubricant enters the chamber around the ring and from there it successively flows via the clearance between ring and the outside of the bush, through the slots and finally through the actual bearing clearance.

As long as shaft and ring are in a concentric position with respect to the bush the pressure distribution in the bearing will be rotationally symmetric. Assume that the ring is fixed in a concentric position with respect to the bush. Should a load be applied to the shaft in the downwards direction, the shaft is forced to move downwards. As the bearing clearance above the shaft increases, the flow resistance will decrease. As a result of this, the axial flow over the top of the shaft increases, the local pressure drop across the ring clearance increases and the mean pressure above the shaft decreases.
Below the shaft the bearing clearance decreases, resulting in a lower axial flow. This will cause a lower pressure drop across the ring clearance and a higher mean pressure below the shaft. The difference between the mean pressure below and above the shaft results in an upwards lubricant film force, which counteracts the external load on the shaft.

In the present design, however, the ring is not fixed, but mounted in "springs". In the situation described above, the pressure drop across the ring at the top is larger than same at the bottom. This results in a lubricant film force on the ring directed downwards. So the ring will move downwards until the lubricant film force is in equilibrium with the spring forces.
This displacement of the ring increases the pressure drop across the ring at the top and reduces same at the bottom. As a result the mean pressure above the shaft decreases again and below the shaft it increases, resulting in a higher lubricant film force on the shaft directed upwards as would be in the case of a fixed ring.
The amplification of load carrying capacity and stiffness of the bearing depends on the stiffness of the springs. It is even possible to design this bearing in such a way that, within a certain load range, the stiffness of the bearing is infinite. This can be demonstrated by a calculation applied to the following simplified model.

Assumptions:

- The loaded shaft maintains the concentric position.
- Only the ring is free to move.
- Only local axial flow at top and bottom of the bearing (i.e. at minimum and maximum film thickness between ring and bush) will be considered. Tangential flow will be neglected.
- The flow resistance of the radial slots will be neglected.

Further assume that the pressure in the slots is p_{rt} (at top) and p_{rb} (at bottom of the bearing).

Moving the ring downwards (displacement $\varepsilon . h_r$) yields a flow:

- at top : - across ring : $q_t = \frac{1}{12\mu}\frac{p_s - p_{rt}}{l_r} . h_r^3\ (1-\varepsilon)^3$

- across bearing: $q_t = \frac{1}{12\mu}\frac{p_{rt} - p_o}{l_b} . h_b^3$

- at bottom: - across ring : $q_b = \frac{1}{12\mu}\frac{(p_s - p_{rb})\ h_r^3\ (1+\varepsilon)^3}{l_r}$

- across bearing: $q_b = \frac{1}{12\mu}\frac{(p_{rb} - p_o)\ h_b^3}{l_b}$

which results in:

$$p_{rt} = \frac{p_s - p_o}{\frac{l_r}{l_b}\left(\frac{h_b}{h_r}\right)^3 \frac{1}{(1-\varepsilon)^3} + 1} = \frac{p_s - p_o}{\frac{c}{(1-\varepsilon)^3} + 1}$$

$$p_{rb} = \frac{p_s - p_o}{\frac{c}{(1+\varepsilon)^3} + 1}$$

The lubricant film force on the ring (downwards direction positive) is:

$$F_r = \tfrac{1}{2}(p_{rb} - p_{rt})\, l_r$$

dimensionless $f_{ring} = \frac{F_r}{l_r} = \tfrac{1}{2}(p_{rb} - p_{rt})$

The lubricant film on the shaft is:

$$F_s = \tfrac{1}{2}(p_{rt} - p_{rb})\, l_b$$

dimensionless $f_{shaft} = \frac{F_s}{l_b} = \tfrac{1}{2}(p_{rt} - p_{rb}) = -f_r$

Fig. 2 shows lubricant film forces as a function of relative displacement of the ring.

It is apparent that the lubricant film force on the ring is in the same direction as the displacement of the ring. The film force on the shaft is in the opposite direction. Within the load carrying capacity range of the bearing, the shaft can be maintained in the centre of the bearing (infinite stiffness) if a counteracting force $-f_r$ is applied on the ring.
According to fig. 2 a linear spring can perform this duty till about $\varepsilon_r = 0{,}6$, which corresponds with about 80% of the load carrying capacity (f_b) of the bearing.

It is obvious that for a real cylindrical bearing the calculations are much more complicated, so that they cannot be carried out without the assistance of a computer programme. Fig. 2, however, offers sufficient insight in the operating priciples of this bearing to design a prototype bearing for preliminary experiments.
The nature of the spring which has to produce the restoring force on the ring has not yet been discussed.

It will be possible to use metal springs. Difficulties may arise with the production of at least three identical springs to ensure a symmetrical bearing response.
A more elegant way to solve this problem is the use of a hydraulic spring. In fact, each hydrostatic bearing is such a hydraulic spring.
Fig. 3 shows a design with two hydrostatic step bearings positioned on each side of the floating ring. These auxiliary bearings act independently of the pressure distribution in the central part of ring and in main bearing.
The auxiliary bearings can be designed in such a way that their force-displacement-characteristic corresponds for the greater part with the lubricant force-displacement-curve of the central part of the ring according to fig. 2.

New opportunities arise by designing auxiliary bearings very stiffly, in which case the stiffness of the auxiliary bearings can be easily reduced. This is done by increasing the pressure in the chamber into which the lubricant from the auxiliary bearings is discharged.
This method offers two advantages:

- Since the stiffness of the main bearing depends on the characteristics of the springs, which in turn determine the equilibrium position of the ring, a bearing with adjustable stiffness has been obtained. By throttling the discharge flow of the auxiliary bearings the stiffness of the main bearing can be adapted to the dynamic requirements of the complete machine system.
- When an infinite stiffness bearing has to be manufactured, very narrow tolerances have to be maintained, as small deviations from the design will greatly influence the performance of the bearing. The adjustability of this bearing offers the opportunity to increase manufacturing tolerances because the infinite stiffness can be adjusted after the bearing has been assembled.

EXPERIMENTS

A bearing according to fig. 3 (diameter 40 mm, length 40 mm) has been thouroughly tested under static load conditions. The results of these experiments are given in fig. 4 and fig. 5.
Each line in fig. 4 corresponds to a value of the discharge pressure/supply pressure ratio $\frac{p_d}{p_s}$

The figure clearly shows that with increasing p_d/p_s ratio (resulting in a decreasing stiffness of the auxiliary bearings) the stiffness of the bearing increases.
It even proved to be possible to obtain a negative stiffness without leading to instability problems. This phenomenon can be used to compensate deflection of the bearing support which results from static load.
The negative stiffness curve (E) shown in fig. 4 is the stability limit. When the discharge channel flow of the auxiliary bearing is throttled further, the bearing will become unstable.
It is obvious that a high stiffness only exists below a certain value of the static load. In the diagram the load curves will not exceed the dotted lines representing the situation of the floating ring touching the bearing bush. These lines have been established experimentally by closing the valve in the discharge channel of the auxiliary bearings. It is a typical unstable situation.

Fig. 5 shows the influence of rotation on the load curves of the bearing adjusted for infinite stiffness at zero speed. It appears that the stiffness remains infinite with rotating shaft.

In addition to this a load curve has been established at 1000 rpm with the floating ring in a fixed concentric position, thus acting like a normal hydrostatic bearing with fixed external restrictors. Comparison with curve A in fig. 4 learns that the hydrodynamic effects cannot increase the stiffness of a normal hydrostatic bearing to a value competitive with the stiffness of the bearing described in this paper.

DYNAMIC BEHAVIOUR OF THE BEARING

The dynamic behaviour of the bearing is determined by stiffness and damping coefficients.
The stiffness of the bearing is equal to the slope of the curves in fig. 4 in case the floating ring follows the movements of the shaft without any phase shift. However, the mass of the ring and the damping effect of the lubricant film between ring and outer diameter of the bush will certainly result in a phase shift. This may reduce the movement of the ring, particularly at high frequencies of load variations and/or discharge channel pressure variations. Indications are that under extreme conditions (high damping and high frequencies) the displacement of the ring may become negligible, in which case the stiffness of the "infinite-stiffness" bearing is in the same order of magnitude as the stiffness of a normal hydrostatic bearing with fixed external restrictors, as far as dynamic load components are concerned.
In any case, the stiffness will decrease with increasing frequency of load and discharge channel pressure variations. The extent of this effect has still to be determined.
The difference between the "static" stiffness and the stiffness for quickly varying loads may be one of the most attractive aspects of this bearing. This will offer the possibility to design a bearing with the following properties:

- a sufficiently high (but not necessarily infinite) "static" stiffness to contain the displacement of the shaft due to the static load components of machining forces within acceptable limits
- a sufficiently low stiffness for quickly varyinging loads to influence the natural frequency in such a way that the latter differs sufficiently from the frequency of the dynamic components of the machining forces.

There is no reason why the stiffness of the bearing should be adjusted prior to a complete machining operation. The stiffness can also be readjusted at the moment vibration problems arise, in which case the stiffness has to be changed in such a way that it shifts the natural frequency of the machine to a higher or lower value.
It is still to be determined experimentally whether this can be done quickly enough.

CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER WORK

It has experimentally been proven that the radial stiffness of the "TNO adjustable stiffness bearing" can be adjusted in a simple way over a wide range up to infinite. Even a negative stiffness is possible.
To change the stiffness only one valve in an auxiliary system has to be operated.

The bearing offers the opportunity to readjust the stiffness, thus influencing the dynamic behaviour of the complete machine tool, as required by the operating conditions.

To arrive at an optimum design a computerprogramme is being developed to calculate the stiffness and damping characteristics of the bearing. The results of these calculations have to be used in a computerprogramme which contains a dynamic model of a complete machine tool. (e.g.a. lathe).

More experiments have to be carried out, particularly on actual machinetools, to study the effectiveness of influencing the dynamic behaviour.

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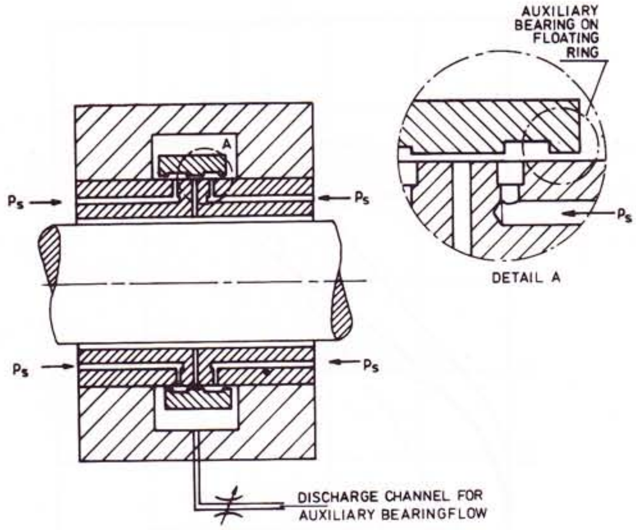

Fig. 3 Bearing with auxiliary bearings on floating ring.

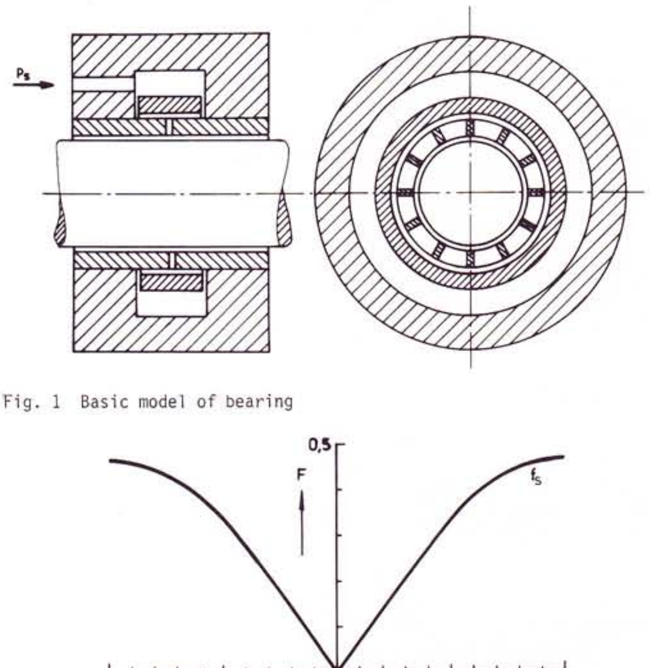

Fig. 1 Basic model of bearing

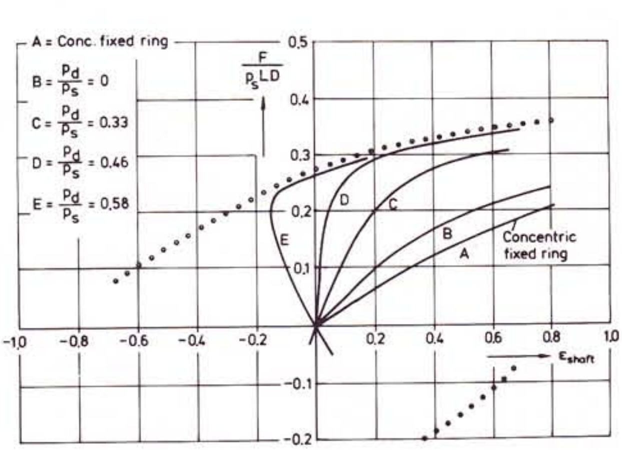

Fig. 4 Experimental results at 0 r.p.m.

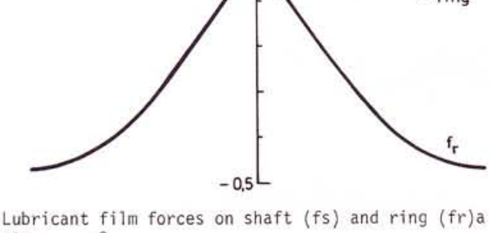

Fig. 2 Lubricant film forces on shaft (fs) and ring (fr)a at $\varepsilon_{shaft}=0$

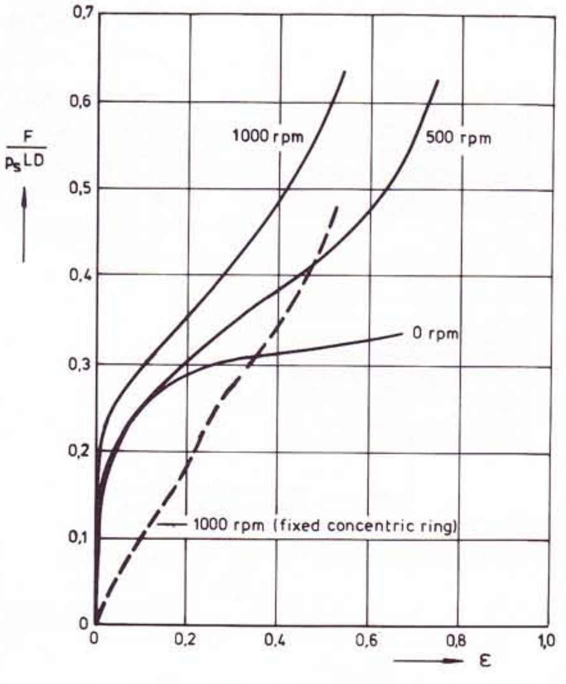

Fig. 5 Infinite stiffness bearing, influence of rotation.