

Hydrodynamic axial bearing

DESCRIPTION

Technical Field

5 The invention relates to the field of hydrodynamic axial bearings for rotating shafts, such as those used in exhaust-gas turbochargers.

The invention relates to a hydrodynamic axial bearing having a floating disk.

Prior Art

10 Load-bearing axial bearings are used when axial thrust forces are applied to rotors which rotate at high speed. In the case of exhaust-gas turbochargers by way of example, hydrodynamic axial bearings are used to absorb high axial forces resulting from the flow, and to guide the turbine shaft in the axial direction. In order to improve the capability to compensate for incline positions and the wear behavior in applications such as these, free-floating disks, so-called floating disks, can be used between a bearing comb, which rotates at the shaft rotation speed, and a bearing housing, which does not
15 rotate, in hydrodynamic axial bearings.

Examples relating to this can be found, inter alia, in GB1095999, EP0840027, EP1199486 and EP1644647. The floating disk is radially guided either on the rotating body, that is to say on the shaft or on the bearing comb by a radial bearing which is integrated in the floating disk, for example as disclosed in EP0840027, or else on a
20 stationary bearing collar, which concentrically surrounds the rotating body, for example as disclosed in EP1199486. In general, a hydrodynamic axial bearing such as this is lubricated by means of lubricating oil from a dedicated lubricating-oil system or, in the case of exhaust-gas turbochargers, via the lubricating-oil system of an internal combustion engine which is connected to the exhaust-gas turbocharger.

25 During operation, a load-bearing lubricating film is formed between the floating disk, which rotates at only about half the shaft rotation speed, and the shaft or the bearing

comb which is arranged on the shaft. Profiled annular surfaces are generally provided for this purpose on both sides of the floating disk and form the lubricating gap in each case together with one smooth sliding surface. The profiled annular surfaces comprise wedge surfaces which are oriented at least in the circumferential direction and which, together with the smooth sliding surfaces, form a converging gap. If sufficient lubricant is drawn into this converging gap, the load-bearing lubricating film is formed. The lubricant propagates in the radial direction because of the effect of the centrifugal force from the floating disk, which is rotating at high speed.

The friction moments on the axial and radial sliding surfaces of the floating disk influence the rotation speed of the floating disk. At high shaft rotation speeds, this is typically less than 50% of the shaft rotating speed, that is to say the floating disk rotates less than half as fast as the shaft. This results in different relative speeds in the two axial lubricating gaps. The relative speed of the shaft with respect to the floating disk is in this case greater than the relative speed of the floating disk with respect to the bearing housing.

The gap heights which occur in the two axial lubricating gaps are of different magnitude, because of the different relative speeds and the different centrifugal-force effects. Since the bearing size is designed for the smallest lubricating gap that in each case occurs, one bearing gap is oversized, and this can lead to an unnecessarily high power loss and to an unnecessarily high oil throughput.

Brief description of the invention

The object of the present invention is therefore to optimize the power loss and oil throughput for a hydrodynamic axial bearing with a floating disk.

According to the invention, this is achieved by the two bearing gaps on the two sides of the floating disk having supporting capabilities of different magnitude, thus resulting in the minimum lubricating gaps being the same at the design point. The supporting capability of a bearing is defined as the supporting force as a function of the rotation speed, the minimum lubricating gap and the oil characteristics.

According to the invention, the different supporting capability is achieved by lubricating gaps of different geometric design. In this case, the lubricating gap is defined as the

area bounded by a profiled annular surface and a smooth sliding surface between the components which form the axial bearing, that is to say between the bearing housing and the floating disk on one side, and the floating disk and the bearing comb, which rotates with the shaft, on the other side.

- 5 A radial step on the floating disk makes it possible to individually design the surfaces of the two axial lubricating gaps in such a way that the minimum gap height in the two lubricating gaps is of the same magnitude at the design point. In this case, that side of the floating disk which faces the bearing housing is advantageously equipped with a larger supporting surface than the side facing the bearing comb.
- 10 The size of one of the profiled circular surfaces can also be reduced by shortening it radially. In this case, the shortening can be carried out radially internally, radially externally or radially internally and externally.

- A supporting surface of the axial bearing is defined as the profiled annular surface and the smooth sliding surface which together form the lubricating gap. The size of the
- 15 supporting surface can also reduce, for example, when only one of the two profiled annular surfaces is shortened in the radial direction, while the smooth sliding surfaces have the same dimensions. The same effect is achieved when two profiled annular surfaces of the same size interact with smooth sliding surfaces, one of which does not extend over the entire annular surface for example in the radial direction.

- 20 A variation in the supporting capability can also be achieved by a geometric change to the profile in the circumferential direction. For example, the number of segments may be reduced from six to five. Alternatively, the extent of the oil grooves in the circumferential direction can be increased.

- The cited embodiments make it possible to compensate for the minimal lubricating-gap
- 25 heights on both sides of the floating disk, despite different relative speeds between the floating disk and the bearing housing on the one side and the floating disk and the bearing comb on the other side.

- Alternatively, one or both sides of the floating disk may be in the form of a smooth sliding surface, and the profiled annular surface may be arranged on the shaft and/or
- 30 the bearing comb. By way of example, if the side of the floating disk which faces the shaft is in the form of a smooth sliding surface and the annular surface is profiled to

correspond to the shaft, annular surfaces, which rotate at higher speed, are each profiled from both bearing parts.

Brief description of the drawings

Embodiments of the invention will be explained in more detail in the following text with
5 reference to the drawings, in which:

Figure 1 shows a first embodiment of an axial journal bearing designed according to the invention and having a stepped floating disk,

Figure 2 shows a second embodiment of an axial journal bearing designed according to the invention and having a floating disk with differently profiled annular
10 surfaces, and

Figure 3 shows a section through the lubricating-oil supply for a third embodiment of an axial journal bearing designed according to the invention and having a stepped floating disk.

Approach to implementation of the invention

15 Figure 1 and Figure 2 show two embodiments of the hydrodynamic axial bearing according to the invention, with the center of each of the figures showing a section along the shaft axis through the axial bearing. The axial bearing comprises a floating disk 30 which is arranged axially between the bearing housing 20 and the bearing comb 11, which is arranged on the shaft 10 and rotates with the shaft. Optionally, the bearing
20 comb may be integrated in the shaft as a radially protruding projection, resulting in the floating disk being arranged axially between the bearing housing and the shaft projection. The left-hand and right-hand areas of each of the figures show a view of the floating disk from the respective side. The left-hand side shows the floating disk 30A viewed in the direction A, and the right-hand side shows the floating disk 30B viewed in
25 the direction B.

The profiled annular surface of the floating disk 30A facing the bearing housing rotates at the absolute speed v_s , that is to say in the counterclockwise direction when viewed in the direction A in the illustrated embodiment. In this case, the lubricating oil which is introduced radially via lubricating grooves 31 into the area of the profiled annular

surface of the floating disk between the floating disk 30 and the bearing housing 20, as indicated by the broad arrows, is drawn into the wedge surfaces 32 in the opposite direction to the rotation direction of the floating disk. The pressure required for the supporting capability of the axial bearing is built up by the narrowing of the lubricating gap between the wedge surfaces 32 and the opposite smooth sliding surface on the bearing housing. The greatest pressure results in the area of the transition from the wedge surface 32 to the latching surface 33.

The profiled annular surface of the floating disk 30B facing the bearing comb rotates at the absolute speed v_s , in the clockwise direction when viewed in the direction B in the illustrated embodiment. However, since the bearing comb 11 rotates in the same direction at more than twice the speed v_w , this results in a relative speed of the profiled annular surface v_R , which runs counterclockwise, when considered in the direction B. The relative speed v_R is in this case greater than the absolute speed v_s . Once again, the broad arrows indicate how the lubricating oil is carried radially outward via the lubricating grooves, and in the process is drawn into the wedge surfaces in the circumferential direction.

In the embodiment shown in Figure 1, the floating disk 30 has a radial step, as a result of which the side facing the bearing housing 20 projects in the radial direction beyond the side facing the bearing comb 11. The radial extents of the two sides of the floating disk are therefore different. The profiled annular surface on the side facing the bearing housing has an annular width r_G which is greater than the annular width r_w of the annular surface on the side of the floating disk facing the bearing comb.

Because of the higher relative speed with which the wedge surfaces of the profiled annular surface rotate along the smooth sliding surface on the bearing comb on the side of the floating disk facing the bearing comb, a lubricating gap is formed, despite the smaller supporting area, which corresponds to that between the bearing housing and the side of the floating disk facing the bearing housing.

In the embodiment shown in Figure 2, the floating disk 30 has two sides of equal size, but whose profiled annular surfaces are different. The profiled annular surface is subdivided into a plurality of segments 34, wherein one segment comprises a lubricating groove 31, a wedge surface 32 and an adjacent latching surface 33. The annular surface, which rotates more slowly in comparison to the stationary bearing housing, on

the left-hand side has a greater number of segments 34 than the annular surface, which rotates at a higher relative speed, on the side facing the bearing comb.

The supporting capability can optionally be varied for example by varying the inclination angle of the wedge surface, thus making the area of the greatest supporting force smaller or larger. The transition between the wedge surface 32 and the latching surface 33 may be provided by means of an edge, or as a continuously running surface without an edge. In the latter case, there is no need for any difference between the wedge surface and the latching surface, as a result of which, for example, the wedge surface can also rise continuously, at an angle which becomes continually smaller, as far as the next lubricating groove.

In the embodiment shown in Figure 3, the floating disk 30 once again has a radial step, as a result of which the side facing the bearing housing 20 projects in the radial direction beyond the side facing the bearing comb 11. In addition, an incline is applied on the radially inner side of the side facing the bearing comb, and the internal radius of the profiled annular surface is enlarged somewhat outward. The inner edge of the smooth sliding surface on the bearing comb is offset radially outward, thus additionally reducing the supporting area of the lubricating gap between the floating disk 30 and the bearing comb 11, in comparison to the first embodiment.

Both sides of a floating disk for a hydrodynamic axial bearing must be supplied with lubricating oil. According to the invention, at least one supply hole 35 is incorporated in the floating disk for this purpose. The supply hole 35 connects the two sides of the floating disk, and allows lubricating oil to be supplied from one side to the other. Optionally, the floating disk has a supply hole for each lubricating groove on the side facing the bearing comb.

In the illustrated embodiment, the lubricating oil is passed via a lubricating oil supply line 22 in the bearing housing into the area of the lubricating gap between the floating disk 30 and the bearing housing 20. The rotation of the profiled annular surface feeds the lubricating oil along the lubricating grooves 31 radially outward, and in the process draws it into the wedge surfaces in the circumferential direction. The lubricating oil is also fed via the supply holes 35, which are arranged in the area of the lubricating grooves 31, into the area of the lubricating gap between the floating disk 30 and the bearing comb 11. If, in this case, starting from the lubricating gap, the supply holes with

the lubricating-oil supply are directed at an angle toward the outer edge, rather than in the axial direction, the lubricating-oil flow is assisted by the rotation of the floating disk.

The supply hole means that there is very largely no need for any supply grooves in the area of the radial bearing of the floating disk. This reduces the lubricating-oil throughput through the decoupling gap between the bearing comb 11 and the bearing collar 21, which is present when the floating disk 30 is borne on the stationary bearing housing. This is particularly advantageous when the decoupling gap leads into the injection-oil area, for example of the exhaust-gas turbocharger, in the area of the radial bearing of the floating disk, without any additional sealing element.

List of Reference Symbols

10	Shaft
11	Bearing comb
20	Bearing housing
21	Bearing collar
22	Lubricating oil supply line
30	Floating disk
30A	Floating disk, bearing housing side (viewed in direction A)
30B	Floating disk, bearing comb side (viewed in direction B)
31	Lubricating groove
32	Wedge surface
33	Latching surface
34	Segment of the profiled annular surface
35	Supply hole
r_G	Annular width on the side of the bearing housing
r_W	Annular width on the side of the bearing comb
v_R	Relative speed of the floating disk with respect to the bearing comb
v_S	Speed of the floating disk
v_W	Speed of the bearing comb

PATENT CLAIMS

1. A hydrodynamic axial bearing for a shaft (10) which is mounted such that it can rotate in a bearing housing (20), comprising a floating disk which is arranged axially between the bearing housing and a bearing comb (11) which is arranged on the shaft, wherein a lubricating gap, which is bounded by a profiled annular surface and a smooth sliding surface is in each case formed between the bearing housing (20) and the floating disk (30), and the floating disk (30) and the bearing comb (11), wherein the profiled annular surfaces comprise a plurality of wedge surfaces (32) which each narrow the lubricating gap in the circumferential direction, characterized in that the lubricating gaps which are formed by the profiled annular surfaces and the smooth sliding surfaces on the two sides of the floating disk have different geometric dimensions to one another.
2. The hydrodynamic axial bearing as claimed in claim 1, wherein the lubricating gaps on the two sides of the floating disk have different radial dimensions.
3. The hydrodynamic axial bearing as claimed in claim 1 or 2, wherein the two profiled annular surfaces which form the lubricating gap on both sides of the floating disk (30) are each subdivided into a different number of segments (34), wherein one segment (34) in each case comprises one wedge surface (32).
4. The hydrodynamic axial bearing as claimed in claim 1 or 2, wherein the two profiled annular surfaces which form the lubricating gap on both sides of the floating disk have wedge surfaces (32) of different design.
5. The hydrodynamic axial bearing as claimed in claim 1 or 2, wherein the profiled annular surfaces comprise wedge surfaces (32) and smooth latching surfaces (33), and the two profiled annular surfaces which form the lubricating gap on both sides of the floating disk have a different area ratio from the latching surface (33) to the wedge surface (32).
6. The hydrodynamic axial bearing as claimed in one of claims 1 to 5, wherein one side of the floating disk (30) is radially stepped with respect to the other side.
7. The hydrodynamic axial bearing as claimed in one of claims 1 to 6, wherein at least one supply hole (35) is incorporated in the floating disk (30), wherein the at least one

supply hole (35) connects the two sides of the floating disk to one another.

8. The hydrodynamic axial bearing as claimed in claim 7, wherein the floating disk has radial lubricating grooves (31), and the at least one supply hole (35) opens into a lubricating groove (31) at least on one side of the floating disk.
- 5 9. The hydrodynamic axial bearing as claimed in claim 7 or 8, wherein the at least one supply hole (35) is directed at an angle radially outward.
10. An exhaust-gas turbocharger, comprising a shaft with a hydrodynamic axial bearing as claimed in one of the preceding claims.