

Hydrodynamic axial bearing

DESCRIPTION

Field of the Invention

The invention relates to the field of hydrodynamic axial mounting of rotating shafts, as
5 are used, for instance, in turbomachines, in particular in exhaust gas turbochargers.

Prior Art

If rapidly rotating rotors are loaded with axial shearing forces, load-bearing axial bearings are used. For example in the case of turbomachines, such as exhaust gas turbochargers, hydrodynamic axial bearings are used to absorb axial forces which are
10 high as a result of the flow and to guide the shaft in the axial direction. In order to improve the oblique position compensation capability and the wear behavior in applications of this type, disks which float freely in the lubricating oil, what are known as floating disks, can be used in hydrodynamic axial bearings between a bearing comb which rotates at the shaft rotational speed and a non-rotating axial stop on the bearing
15 housing. The lubricating gaps between a rotating bearing comb and the floating disk and between the floating disk and the stationary axial stop on the bearing housing are advantageously delimited in each case by a profiled circular ring face and a plane sliding face which lies opposite the profiled circular ring face. The profiled circular ring face serves to optimize the pressure build-up in the lubricating gap, which pressure
20 build-up is decisive for the load-bearing force of the axial bearing. In order to distribute the lubricating oil which is supplied in the radially inner region of the profiled circular ring face, there are lubricating oil grooves which lead radially to the outside. Wedge faces which constrict the lubricating gap in the circumferential direction and via which the lubricating oil which is introduced into the lubricating oil grooves exits are formed
25 adjacently with respect to the lubricating oil grooves. Here, the lubricating oil is guided into the wedge face as far as possible over the entire radial height of the lubricating oil grooves. The pressure build-up which is necessary for the load-bearing capability of the axial bearing takes place substantially in the region of the wedge faces. Rest faces which comprise a planar face and are provided by the load-bearing face of the profiled

circular ring face are formed adjacently with respect to the wedge faces in the circumferential direction.

Examples of axial bearings of this type are found, inter alia, in GB 1095999, EP0840027, EP1199486, EP1644647 and EP2042753. The radial guidance of the floating disk takes place either on the rotating body, that is to say on the shaft or on the bearing comb by way of a radial bearing which is integrated into the floating disk, as is disclosed, for example, in EP0840027, or else on a stationary bearing collar which surrounds the rotating body concentrically, as is disclosed, for example, in EP1199486. The lubrication of a hydrodynamic axial bearing of this type takes place as a rule 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.

In the cold state, at a standstill, all the load-bearing faces of conventional axial mountings lie perpendicularly with respect to the rotational axis of the rotor or else at least parallel to one another. During operation, the load-bearing faces can be deformed on account of temperature gradients, centrifugal, shearing and other forces. A deformation of this type of the bearing load-bearing faces can impair the load-bearing force of the mounting. Temperature gradients over the comb of the comb bearing can have particularly great effects. The comb which protrudes radially with respect to the shaft is deformed in an umbrella-shaped manner on account of the temperature difference between the load-bearing face and the rear side. This deformation can lead to rubbing of the comb bearing on the floating disk, particularly in the case of a low oil supply pressure. The deformation on account of the temperature gradient is particularly critical in a conventional comb bearing construction, since said deformation causes a lubricating gap which widens to the outside. This constellation firstly reduces the load-bearing capability for geometric reasons and secondly reduces the centrifugal force-induced pressure build-up in the radial direction, since the outflow resistance for the lubricating oil radially to the outside is reduced.

Brief Summary of the Invention

It is therefore an object of the present invention to improve the load-bearing capability of a hydrodynamic axial bearing for mounting a shaft which is mounted rotatably in a bearing housing.

- 5 If the gap which is formed between the load-bearing faces of the axial bearing is configured so as to be constricted to the outside in the radial direction, by the load-bearing faces being arranged obliquely relative to one another at least in the radially outer region, a reduction in the relative oblique position of the load-bearing faces results during operation on account of the abovementioned deformation of the rotating load-
- 10 bearing face. The constriction in the radially outer region is reduced, with the result that the load-bearing faces rest more uniformly on one another during operation.

If, for instance, the bearing comb is manufactured with a conical load-bearing face, that is to say a load-bearing face which is inclined toward the load-bearing face which lies opposite it, the temperature deformation in the comb bearing can be compensated for.

- 15 During the compensation, the deformations on account of centrifugal, shearing and further forces likewise have to be taken into consideration.

Since the comb bearing deformations are dependent on the operating point, the lubricating gap becomes smaller in the radial direction under certain operating conditions. This situation is more favorable than the current one with a widened

20 lubricating gap, since the load-bearing capability is reduced to a lesser extent and the centrifugal force-induced pressure build-up in the radial direction is aided.

The compensation on account of load-bearing face deformations as a result of temperature gradients, centrifugal, shearing and further forces can also take place at the floating disk, or at the axial stop of the bearing housing in the case of an axial

25 bearing without floating disk. Any temperature-induced deformations which occur in the region of the axial stop on the bearing housing can be carried out in a similar way as on the comb bearing.

If a floating disk which is conical on both sides or a very thin floating disk which is adapted to changing geometric conditions during operation is used, the comb bearing

30 deformation can also be compensated for by way of a conical configuration of the axial stop on the bearing housing.

Thanks to the compensation for the deformation, the axial mounting becomes more robust at the adjacent bearing parts with respect to rubbing of the floating disk or the bearing comb, or, in the case of an axial bearing without floating disk, of the axial bearing. The turbocharger becomes more operationally reliable and wear-induced costs can be reduced.

Brief Description of the Drawings

In the following text, exemplary embodiments of the invention will be explained in detail using drawings, in which:

- fig. 1 shows, in the right-hand part, a section which is guided along the rotational axis through an embodiment (configured according to the prior art) of an axial sliding bearing with a rotating bearing comb, a stationary axial stop and a floating disk, and shows, in the left-hand part, a frontal view in the axial direction of the corresponding floating disk with a profiled circular ring face,
- fig. 2 shows a diagrammatically illustrated axial sliding bearing according to fig. 1, the bearing comb being shown in each case in the cold state in this figure and in all following figures, and additionally the deformation of the bearing comb in the operating state on account of the heating and the rapid rotation and the resulting lubricating gap being indicated by way of dashed lines,
- fig. 3 shows a diagrammatically illustrated axial sliding bearing according to a first embodiment according to the invention, with a conically shaped bearing comb and a lubricating gap which results therefrom and tapers radially toward the outside,
- fig. 4 shows a diagrammatically illustrated axial sliding bearing according to a second embodiment according to the invention with a floating disk which is shaped conically on the bearing comb side and a lubricating gap which results therefrom and tapers radially toward the outside,
- fig. 5 shows a diagrammatically illustrated axial sliding bearing according to a third embodiment according to the invention, with a conically shaped axial bearing and conically shaped bearing comb, and two lubricating gaps which result therefrom and taper radially toward the outside,

- fig. 6 shows a diagrammatically illustrated axial sliding bearing according to a fourth embodiment according to the invention with a conically shaped axial bearing and a floating disk which is shaped conically on the bearing comb side, and two lubricating gaps which result therefrom and taper radially toward the outside,
- fig. 7 shows a diagrammatically illustrated axial sliding bearing according to a fifth embodiment according to the invention with a floating disk which is shaped conically on both sides, and two lubricating gaps which result therefrom and taper radially toward the outside,
- fig. 8 shows a diagrammatically illustrated axial sliding bearing according to a sixth embodiment according to the invention with a conically shaped bearing comb and a floating disk which is shaped conically on the axial bearing side, and two lubricating gaps which result therefrom and taper radially toward the outside,
- fig. 9 shows a diagrammatically illustrated axial sliding bearing according to a seventh embodiment according to the invention, without a floating disk, with a conically shaped bearing comb, and a lubricating gap which results therefrom and tapers radially toward the outside, and
- fig. 10 shows a diagrammatically illustrated axial sliding bearing according to an eighth embodiment according to the invention, once again without a floating disk, with a conically shaped axial stop, and a lubricating gap which results therefrom and tapers radially toward the outside.

Way of Implementing the Invention

Fig. 1 shows by way of example a hydrodynamic axial bearing according to the prior art, the three essential components of the axial bearing being made visible in the right-hand part of the figure in a section which is guided axially along the rotational shaft. The bearing comb 10 is placed on the rotating shaft 40, or is optionally connected in a material-to-material manner to the shaft or is produced with the shaft from one piece, and rotates with the shaft. A floating disk 30 is arranged axially between an axial stop 21 on the bearing housing 20 and the bearing comb. In each case one lubricating gap is formed firstly between the axial stop and the floating disk and secondly between the

floating disk and the bearing comb, in which lubricating gap a thin lubricating oil layer is situated between the load-bearing faces. In the embodiment which is shown, the load-bearing face 22 on the axial stop and the load-bearing face 11 on the bearing comb in each case have a sliding face which is of planar configuration in the circumferential direction, whereas the two load-bearing faces of the floating disk are part of a profiled circular ring face. This basic construction of the two lubricating gaps is also adopted in all the embodiments described in the following text of the hydrodynamic axial sliding bearings according to the invention with floating disk. It applies to all these embodiments that the sliding faces and the profiled circular ring faces in the case of one or both of the lubricating gaps can also be arranged on the respectively other side of the lubricating gap, with the result that, for example, the floating disk has in each case one planar sliding face on both sides whereas the profiled circular ring face is attached on the load-bearing face of the bearing comb and the axial stop of the bearing housing. In an embodiment without floating disk, the profiled circular ring face would correspondingly be arranged on the rotating bearing comb and the planar sliding face would be arranged on the axial stop of the bearing housing or at any rate vice versa, that is to say the planar sliding face on the rotating bearing comb and the profiled circular ring face on the axial stop of the bearing housing.

The construction of the profiled circular ring face can be seen from the left-hand part of fig. 1, in which the floating disk is rotated by 90°, with the result that one of the end sides of the floating disk can be seen in a plan view.

The profiled circular ring face serves to optimize the pressure build-up in the lubricating gap between the load-bearing faces, which pressure build-up is decisive for the load-bearing force of the axial bearing. The profiling of the circular ring face comprises a plurality of segments with in each case one lubricating oil groove 33 which is led radially to the outside in order to distribute the lubricating oil which is supplied in the radially inner region of the profiled circular ring face. Counter to the rotational direction (indicated by way of the black arrow) of the profiled circular ring face, wedge faces 34 which constrict the lubricating gap in the circumferential direction are formed adjacently with respect to the lubricating oil grooves 33, via which wedge faces 34 the lubricating oil which is introduced into the lubricating oil grooves 33 exits in accordance with the thick arrows. Here, the lubricating oil is guided into the wedge face 34 as far as possible

over the entire radial height of the lubricating oil grooves 33. The pressure build-up which is necessary for the load-bearing capability of the axial bearing takes place substantially in the region of the wedge faces. Rest faces 35 are formed adjacently with respect to the wedge faces 34 in the circumferential direction, which rest faces 35
5 comprise a planar face which is at the smallest spacing from the corresponding contact, as the above-described sliding face. The axial extent (thickness) of the lubricating gap can therefore be described as the spacing between the rest faces 35 and the sliding face which lies opposite. In order to optimize the pressure build-up in the radial direction in the lubricating oil groove and over the wedge faces, the lubricating oil groove and
10 wedge face can be closed radially to the outside by way of a web which constricts the lubricating gap. Here, the web typically comes to lie as far as the height of the rest face, with the result that the rest face and web lie in one plane.

The configuration of the lubricating oil groove and the wedge face is disregarded for the embodiments which are described in the following text. Accordingly, the expressions of
15 the profiled circular ring face and the sliding face are no longer used in the following text. For the practical implementation, however, reference is made to the fact that the lubricating gaps, as described above, are advantageously delimited in each case by a profiled circular ring face and a planar sliding face. The expression used in the following text of the active load-bearing face means that region of the profiled circular ring face
20 which is generally called rest face. The rest faces are typically situated so as to adjoin the wedge faces as viewed in the flow direction of the lubricating oil.

As indicated in fig. 1 and in the detail which is shown on an enlarged scale according to fig. 2, in the cold state, that is to say at a standstill of the rotor, the load-bearing faces of the axial mountings are configured perpendicularly with respect to the rotational axis of
25 the rotor or else at least parallel to one another. During operation, the load-bearing face in the bearing comb can be deformed on account of temperature gradients, centrifugal, shearing and further forces. The comb which protrudes radially with respect to the shaft is deformed in an umbrella-shaped manner on account of the temperature difference between the load-bearing face which is relevant for the axial bearing and the rear side
30 which faces away from said load-bearing face. This deformation (indicated in fig. 2 by way of dashed lines) can lead to rubbing of the comb bearing on the floating disk in the radially inner region, since the load-bearing force of the lubricating gap diminishes on

account of the radially outwardly diverging load-bearing faces 31 and 11' of the axial bearing and the associated unimpeded escape of the lubricating oil, in particular in the case of a low oil supply pressure, at which sufficient lubricating oil cannot be replenished.

5 Fig. 3 shows a diagrammatically illustrated hydrodynamic axial sliding bearing according to a first embodiment according to the invention. Here, the active load-bearing face 31 on that side of the floating disk 30 which faces the bearing comb is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40. In contrast, the load-bearing face 11 of the bearing comb is shaped so as to be inclined
10 toward the floating disk 30, which results in a constriction in the axial direction in the radially outer region of the lubricating gap 52. In this embodiment, just as in the further embodiments which will be described in the following text, the inclination of the load-bearing face 11 of the bearing comb can be realized by way of a uniform, straight inclination or by way of a curved inclination. In the figures, the deformations of the
15 rotating components and the constrictions of the lubricating gaps are illustrated in a greatly exaggerated manner. In fact, the inclination angles which are provided according to the invention move over the entire radius of the inclined component in the range of a few hundredths of a degree, which results in a constriction of the lubricating gap at the radially outer edge of a few hundredths of a millimeter in the case of a disk with a
20 diameter of 200 mm. During operation, a deformation of the bearing comb which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 11, which is inclined towards the floating disk in the cold state, of the bearing comb stretches in such a way that the angle of the
25 constriction of the lubricating gap 52' is reduced during nominal operation and the two load-bearing faces 31 and 11' of the bearing run parallel to one another or, while maintaining a lubricating gap constriction which is less pronounced than in the cold state, run at least virtually parallel to one another. The fact that, in the cold state, that is to say at a standstill and also at small rotational speeds, the configuration according to
30 the invention of the axial sliding bearing leads to a constriction of the lubricating gap in the radial outer region is not a problem, since the accumulated lubricating oil ensures an additional pressure build-up.

Fig. 4 shows a diagrammatically illustrated hydrodynamic axial sliding bearing according to a second embodiment according to the invention. Here, the load-bearing face 11 of the bearing comb is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40. For this purpose, the load-bearing face 31 on that side of the floating disk 30 which faces the bearing comb is configured so as to be inclined toward the bearing comb 10 in this embodiment, which results once again in the constriction in the axial direction in the radially outer region of the lubricating gap 52. The floating disk is therefore of conical configuration on the side which faces the bearing comb, whereas it is oriented perpendicularly with respect to the rotational axis of the shaft 40 on the other side which faces the axial stop on the bearing housing. During operation, a deformation of the bearing comb which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 11, which is oriented perpendicularly with respect to the rotational axis of the shaft 40 in the cold state, of the bearing comb is bent in such a way that, during nominal operation, the angle of the constriction of the lubricating gap 52' is reduced and the two load-bearing faces 31 and 11' of the bearing run parallel or virtually parallel to one another.

In the embodiments according to figs. 5 to 8, in addition to the lubricating gap 52 between the floating disk 30 and the bearing comb 10, the lubricating gap 51 between the axial stop 21 and the floating disk 30 is also configured with a constriction in the axial direction in the radially outer region.

Fig. 5 shows a diagrammatically illustrated hydrodynamic axial sliding bearing according to a third embodiment according to the invention. Here, the load-bearing face 31 is oriented strictly radially on that side of the floating disk 30 which faces the bearing comb, that is to say perpendicularly with respect to the rotational axis of the shaft 40. In contrast, the load-bearing face 11 of the bearing comb is shaped such that it is inclined toward the floating disk 30, which results in a constriction in the axial direction in the radially outer region of the lubricating gap 52. The second lubricating gap which is likewise provided with a constriction in the axial direction in the radially outer region extends between the load-bearing face 32 which is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, on that side of the

floating disk 30 which faces the axial stop and the load-bearing face 22 of the axial stop 21 on the bearing housing, which load-bearing face 22 is inclined toward the floating disk 30. The floating disk is therefore provided with two sides which run parallel to one another and are oriented perpendicularly with respect to the rotational axis of the shaft 40. During operation, a deformation of the bearing comb 10 which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 11, which is inclined toward the floating disk in the cold state, of the bearing comb stretches in such a way that, during nominal operation, the angle of the constriction of the lubricating gap 52' is reduced and the two load-bearing faces 31 and 11' of the bearing run parallel to one another or virtually parallel to one another.

Fig. 6 shows a diagrammatically illustrated hydrodynamic axial sliding bearing according to a fourth embodiment according to the invention, which hydrodynamic axial sliding bearing differs from the preceding one in that the load-bearing face 11 of the bearing comb is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, and, for this purpose, the load-bearing face 31 is configured so as to be inclined toward the bearing comb 10 on that side of the floating disk 30 which faces the bearing comb. The second lubricating gap which is likewise provided with a constriction in the axial direction in the radially outer region once again extends between the load-bearing face 32 which is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, on that side of the floating disk which faces the axial stop and the load-bearing face 22 of the axial stop 21 on the bearing housing, which load-bearing face 22 is inclined toward the floating disk 30. The floating disk is therefore of conical configuration on the side which faces the bearing comb, whereas it is oriented perpendicularly with respect to the rotational axis of the shaft 40 on the other side which faces the axial stop on the bearing housing. During operation, a deformation of the bearing comb which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 11, which is oriented perpendicularly with respect to the rotational axis of the shaft 40 in the cold state, of the bearing comb bends in such a way that, during nominal operation, the angle of the constriction of the lubricating gap 52' is

reduced and the two load-bearing faces 31 and 11' of the bearing run parallel to one another or virtually parallel to one another.

Fig. 7 shows a diagrammatically illustrated hydrodynamic axial sliding bearing according to a fifth embodiment according to the invention. Here, the load-bearing face 11 of the bearing comb is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40. In contrast, on that side of the floating disk 30 which faces the bearing comb, the load-bearing face 31 is configured so as to be inclined toward the bearing comb 10, which results in a constriction in the axial direction in the radially outer region of the lubricating gap 52. The second lubricating gap which is likewise provided with a constriction in the axial direction in the radially outer region extends between the load-bearing face, which is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, of the axial stop 21 on the bearing housing and the load-bearing face 32 which is inclined toward the axial stop on that side of the floating disk which faces the axial stop. The floating disk 30 is therefore configured so as to be conical on both sides. During operation, a deformation of the bearing comb 10 which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 11, which is oriented perpendicularly with respect to the rotational axis of the shaft 40 in the cold state, of the bearing comb bends in such a way that, during nominal operation, the angle of the constriction of the lubricating gap 52' is reduced and the two load-bearing faces 31 and 11' of the bearing run parallel to one another or virtually parallel to one another.

Fig. 8 shows a diagrammatically illustrated hydrodynamic axial sliding bearing according to a sixth embodiment according to the invention, which hydrodynamic axial sliding bearing differs from the preceding one in that the load-bearing face 31 on that side of the floating disk 30 which faces the bearing comb is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, and, for this purpose, the load-bearing face 11 of the bearing comb is shaped so as to be inclined toward the floating disk 30, which results once again in a constriction in the axial direction in the radially outer region of the lubricating gap 52. The second lubricating gap which is likewise provided with a constriction in the axial direction in the radially

outer region once again extends between the load-bearing face 22, which is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, of the axial stop 21 on the bearing housing and the load-bearing face 32 which is inclined toward the axial stop on that side of the floating disk which faces the axial stop. The floating disk is therefore of conical configuration on the side which faces the axial stop on the bearing housing, whereas it is oriented perpendicularly with respect to the rotational axis of the shaft 40 on the other side which faces the bearing comb. During operation, a deformation of the bearing comb which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 11, which is inclined toward the floating disk in the cold state, of the bearing comb stretches in such a way that, during nominal operation, the angle of the constriction of the lubricating gap 52' is reduced and the two load-bearing faces 31 and 11' of the bearing run parallel to one another or virtually parallel to one another.

The two last figures in each case show a hydrodynamic axial sliding bearing without a floating disk, in which a load-bearing face 12 is arranged on the rotating bearing comb 10 and a load-bearing face 22 is arranged on the axial stop 21 of the bearing housing 20. According to the invention, the lubricating gap 53 which results between them is once again configured so as to converge radially to the outside, that is to say the lubricating gap tapers in the radially outer region.

The seventh embodiment according to the invention (shown in fig. 9) of a hydrodynamic axial sliding bearing has a load-bearing face 12 of the bearing comb 10, which load-bearing face 12 is shaped so as to be inclined toward the axial stop 21 of the bearing housing 20, which results in the constriction in the axial direction in the radially outer region of the lubricating gap 53. The load-bearing face 22 of the axial stop 21 of the bearing housing 20 is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40, in this embodiment. During operation, once again a deformation of the bearing comb which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 12, which is inclined toward the load-bearing face of the axial stop 21 in the cold state, of the bearing comb stretches in such a way that, during nominal operation, the angle of

the constriction of the lubricating gap 53' is reduced and the two load-bearing faces 12' and 22 of the bearing run parallel to one another or virtually parallel to one another.

The eighth embodiment according to the invention (shown in fig. 10) of a hydrodynamic axial sliding bearing has a load-bearing face 12 of the bearing comb 10, which load-bearing face 12 is oriented strictly radially, that is to say perpendicularly with respect to the rotational axis of the shaft 40. For this purpose, in this embodiment, the load-bearing face 22 of the axial stop 21 on the bearing housing 20 is configured so as to be inclined toward the bearing comb 10, which results once again in the constriction in the axial direction in the radially outer region of the lubricating gap 53. The axial stop is therefore of conical configuration on the side which faces the bearing comb. During operation, once again a deformation of the bearing comb which is once again indicated by way of dashed lines results on account of the above-described heating of the bearing comb and as a result of the action of the stated forces. According to the invention, the load-bearing face 12, which is oriented perpendicularly with respect to the rotational axis of the shaft 40 in the cold state, of the bearing comb 10 bends in such a way that, during nominal operation, the angle of the constriction of the lubricating gap 53' is reduced and the two load-bearing faces 12' and 22 of the bearing run parallel to one another or virtually parallel to one another.

In all the embodiments, in each case one of the load-bearing faces is described as deviating from the plane which is oriented perpendicularly with respect to the rotational axis of the shaft and the other load-bearing face is described as running strictly radially, that is to say along precisely this plane which is oriented perpendicularly with respect to the rotational axis of the shaft. According to the invention, the narrowing lubricating gaps can also be realized by the respective load-bearing faces both deviating from respective planes which are oriented perpendicularly with respect to the rotational axis of the shaft, but being at an angle with respect to one another. For example, in the embodiment with a floating disk, both the load-bearing face on that side of the floating disk which faces the bearing comb and the load-bearing face on the bearing comb can run so as to be inclined toward the lubricating gap in comparison with the plane which is oriented perpendicularly with respect to the rotational axis of the shaft, and can thus delimit the narrowing lubricating gap.

Even if in each case only load-bearing faces were mentioned in all the abovementioned embodiments, it is to be noted once again here that, if one or both of the components which delimit a respective lubricating gap have a profiled surface with a lubricating oil groove, wedge faces and rest faces, the expression load-bearing face means in each

5 case that region of the profiled surface which is called rest face. In the absence of a rest face, the load-bearing face extends along the maximum elevation of the wedge faces in the transition region to the respectively next lubricating oil groove.

List of Designations

10	Bearing comb
11, 12	Load-bearing face on the bearing comb
11', 12'	Load-bearing face on the bearing comb (in the operating state)
20	Bearing housing
21	Axial stop
22	Sliding face
30	Floating disk
31, 32	Load-bearing face of the floating disk
33	Lubricating oil groove
34	Wedge face
35	Rest face
40	Shaft
51	Lubricating gap between the axial stop and the floating disk
52	Lubricating gap between the floating disk and the bearing comb
52'	Lubricating gap between the floating disk and the bearing comb (in the operating state)
53	Lubricating gap between the axial stop and the bearing comb
53'	Lubricating gap between the axial stop and the bearing comb (in the operating state)

PATENT CLAIMS

1. A hydrodynamic axial bearing for mounting a shaft (40) which is mounted rotatably in a bearing housing (20), comprising an axial stop (21) of the bearing housing (20) and a bearing comb (10) which rotates with the shaft, at least one lubricating gap (51, 52, 53) which is loaded with lubricating oil and is delimited by a profiled circular ring face and a planar sliding face (22) which lies opposite the circular ring face being formed between the axial stop (21) and the bearing comb (10), the profiled circular ring face being configured so as to rotate around or with the shaft (40), the profiling of the circular ring faces having a plurality of segments with in each case one radially running lubricating oil groove (33), a wedge face (34) which is connected to the lubricating oil groove (33) in the circumferential direction, and a rest face (35) which adjoins the wedge face (34) in the circumferential direction, wherein, in the case of at least one lubricating gap (51, 52, 53), the rest faces (35) and the planar sliding face (22) are configured in such a way that the lubricating gap (52) which is delimited by the rest faces (35) and the planar sliding face (22) is constricted radially to the outside with regard to the axial direction.
2. The hydrodynamic axial bearing as claimed in claim 1, a planar sliding face (22) of the axial stop, which planar sliding face (22) delimits the lubricating gap (51, 53) which is constricted radially to the outside, being configured to be inclined toward the bearing comb (10) at least in a radially outer part in a manner which deviates from the plane which lies perpendicularly with respect to the rotational axis.
3. The hydrodynamic axial bearing as claimed in either of claims 1 and 2, a planar sliding face (22) of the bearing comb (10), which planar sliding face (22) delimits the lubricating gap (52, 53) which is constricted radially to the outside, being configured to be inclined toward the axial stop (21) in a manner which deviates from the plane which lies perpendicularly with respect to the rotational axis.
4. The hydrodynamic axial bearing as claimed in one of claims 1 to 3, a floating disk (30) being arranged axially between the axial stop (21) and the bearing comb (10), and a lubricating gap (52) which is constricted radially to the outside and is delimited by a profiled circular ring face and a planar sliding face (22) which lies opposite it being formed between the floating disk (30) and the bearing comb (10).

5. The hydrodynamic axial bearing as claimed in claim 4, the profiled circular ring face of the floating disk (30) and a planar sliding face of the bearing comb (10) delimiting the lubricating gap (52) which is constricted radially to the outside, and the planar sliding face being configured to be inclined toward the axial stop (21) at least in a radially outer part in a manner which deviates from the plane which lies perpendicularly with respect to the rotational axis.
6. The hydrodynamic axial bearing as claimed in either of claims 4 and 5, a further lubricating gap (51) being delimited by the axial stop (21) and the floating disk (30), the planar sliding face (22) of the axial stop (21), which planar sliding face (22) delimits said further lubricating gap (51), being configured to be inclined toward the floating disk (30) in a manner which deviates from the plane which lies perpendicularly with respect to the rotational axis.
7. A turbomachine, comprising a shaft (40) which is mounted rotatably in a housing (20), having a hydrodynamic axial bearing as claimed in one of claims 1 to 6.
8. An exhaust gas turbocharger, comprising a shaft (40) which is mounted rotatably in a housing (20), having a hydrodynamic axial bearing as claimed in one of claims 1 to 6.
9. The exhaust gas turbocharger as claimed in claim 8, the bearing comb (10) and the shaft (40) being connected in a material-to-material manner or being manufactured from one piece.