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(54) Benævnelse: **Planetgear**

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GB-A- 2 514 167
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Description

The present invention relates to a planetary gearbox, in particular for a wind power plant, with a gearbox housing, a central sun wheel, which is held in the gearbox housing so as to be able to rotate about a central gearbox axis of rotation, and supports an external toothing, a hollow wheel, which is arranged in the gearbox housing concentrically to the central gearbox axis of rotation, and an internal toothing, a planetary carrier, which is mounted in the gearbox housing so as to be able to rotate about the central gearbox axis of rotation, and a number of planetary wheels, which, by means of planetary wheel bearings configured as slide bearings, are mounted on the planetary carrier so as to be able to rotate about planetary wheel axes of rotation and have external toothings, which engage with the internal toothing of the hollow wheel and the external toothing of the sun wheel.

Planetary gearboxes of this type are known in a variety of embodiments in the prior art and serve for instance as transmission gearboxes to convert a low speed from a drive shaft of the planetary gearbox into a significantly higher speed of an output shaft of the planetary gearbox. Accordingly, planetary gearboxes are frequently installed in wind power plants, where a low speed of the rotor shaft has to be converted into a significantly higher speed of the generator shaft. With the use in wind power plants, planetary gearboxes are operated mainly under very changeable operating conditions on account of the variable wind conditions. As a result of temporarily extremely low speeds of the drive shaft and at the same time extremely high force effect on the bearing, roller bearings are predominantly installed in planetary gearboxes for wind power plants in order to mount the planetary wheels.

Alternatively however, planetary wheel bearings in planetary gearboxes can also be embodied for wind power plants as slide bearings. A planetary gearbox of this type for a wind power plant is described for instance in EP 2 383 480 A1 and comprises a gearbox housing, in which a central sun wheel with an external toothing is held so as to be able to rotate about a central gearbox axis of rotation. Furthermore, a hollow wheel with an internal toothing is provided in the gearbox housing concentrically to the central gearbox axis of rotation. A planetary carrier is likewise mounted in the gearbox housing so as to be able to rotate about the central gearbox axis of rotation. A number of planetary wheels are held on the planetary carrier. The planetary wheels have external toothings, which engage with the internal toothing of the hollow wheel and the external toothing of the sun wheel.

The planetary wheels are mounted on planetary wheel bearings configured as radial slide bearings so as to be able to rotate about planetary wheel axes of rotation. For a reliable operation of a radial slide bearing, its bearing clearance must also take into account that during the operation of the radial slide bearing, temperature and/or load-specific extensions and/or deformations may occur. The components of the radial slide bearing and/or the wheel treads of the mounted planetary wheels must therefore be manufactured with high precision, in other words low manufacturing tolerances, and/or postprocessed during assembly, which is associated with high costs.

During the operation of the radial slide bearing, the bearing clearance gradually changes as a result of wear, which may result in a malfunction or failure of the radial slide bearing. Regular maintenance and if necessary replacement of the radial slide bearing is therefore necessary if the bearing clearance of the radial slide bearing threatens to leave an admissible range. This is associated in particular with use in wind power plants with corresponding down times.

Radial slide bearings may exclusively discharge radial forces. In order also to guide the planetary wheels axially and to prevent axial movements of the planetary wheels, axial slide bearings are also required, which discharge axial forces acting on the planetary wheels. Such axial slide bearings can be embodied for instance in the contact range between the solebars of the planetary carrier and the end faces of the planetary wheels and likewise increase the costs of such planetary wheel bearings. One further example of a planetary gearbox of a wind power plant, in which the planetary wheels are supported by engaged radial slide bearings, is shown in the document WO 2014/117197 A1.

Based on this prior art, it is therefore an object of the present invention to create a planetary gearbox which enables the bearing clearance of the slide bearing used to be easily adjusted and have a simple structure.

The inventive object is achieved by a planetary gearbox according to claim 1. The idea thus underlying the invention is to use axially divided double cone slide bearings which are able to discharge both axial and also radial forces. The opposite arrangement of the two cone-shaped sliding surfaces, which are embodied such that the tapered ends point to one another, allows a planetary wheel to be fixed both in the axial and also in the radial direction. Here the radial bearing clearance can be easily adjusted by axially adjusting the cone-shaped bearing element relative to the planetary wheel which has corresponding cone-shaped wheel treads. The torsional strength of the bearing elements can be effected for instance in that bearing elements produced with an oversize are shrunk onto the planetary wheel axis after their positioning.

In an advantageous embodiment, at least one bearing element can be adjusted in the axial direction, in order to adjust a lubrication gap of a defined height between the sliding surfaces of the planetary wheel bearing and the corresponding wheel treads of the mounted planetary wheel. An optimal height of the lubrication gap between the sliding surfaces of the planetary wheel bearing and the corresponding wheel treads of the mounted planetary wheel is an essential prerequisite for reliable operation of the planetary gearbox.

One bearing element is preferably adjustable, while the other bearing element has an axially fixed position. The bearing element with an axially fixed position can serve as a reference for the adjustment of the adjustable bearing element, which contributes to a simple and precise setting of the optimal height of the lubrication gap of the planetary wheel bearing.

In one embodiment of the inventive planetary gearbox, the axial position of the axially fixed bearing element can be defined by an axial stop. An axial stop can in particular serve as a solebar of the planetary carrier or a radial annular shoulder embodied on the planetary wheel axis.

Adjustment means are advantageously assigned to the adjustable bearing element for axial adjustment purposes. Adjustment means of this type facilitate the setting and in particular also the easy readjustment of an inventive planetary gearbox within the scope of maintenance, if the height of the lubrication gap of the slide bearing has changed due to wear.

In one variant of the inventive planetary gearbox, spacer elements which are arranged between a bearing element and an adjacent solebar of the planetary carrier and/or between the bearing elements are provided as adjustment means. By selecting a suitable number of spacer elements and their arrangement on the specified axial positions within the slide bearing, a height of the lubrication gap of the slide bearing can be adjusted, wherein manufacturing inaccuracies of the bearing components can be equalised.

In a further variant, the adjustable bearing element is screwed onto the planetary wheel axis. To this end, an internal thread and a corresponding external thread are embodied on the adjustable bearing element and the planetary wheel axis in each case. The axial position of the adjustable bearing element can be continuously adjusted by rotation about the planetary wheel axis using a screw connection of this type.

Alternatively, the adjustable bearing element can be screwed into an adjacent solebar of the planetary carrier. To this end, an external thread and a corresponding internal thread are embodied on the adjustable bearing element and in an adjacent solebar of the planetary carrier in each case. With a screw connection of this type, the axial position of the adjustable bearing element can be continuously set by screwing into an adjacent solebar of a planetary carrier.

In this way, a torsion-proof securing element can be provided, by means of which the bearing element screwed onto the planetary carrier or into the solebar can be fixed. A torsion-proof securing element allows for the secure fixing of the set axial position of the adjustable bearing element.

In accordance with the invention, a combination of a number of different adjustment means is also possible in order to set the optimal height of the lubrication gap of the planetary wheel bearing.

In one development of the inventive planetary gearbox, at least one lubrication pocket is embodied in each sliding surface, into which a lubricant duct opens, which radially offsets the bearing element, wherein the lubricant duct is connected to an eccentric lubricant supply duct, which is embodied in the planetary wheel axis and axially offsets this. During normal operation of the planetary gearbox, lubricant is supplied to the sliding surfaces of the planetary wheel bearing within the scope of a pressure feed lubrication. The lubricant is introduced into the eccentric lubricant supply duct under pressure, and flows from there, through lubricant ducts into the lubrication pockets, from where it is distributed onto the sliding surfaces.

In a further embodiment, a spacer ring is arranged between the bearing elements, said spacer ring surrounding the planetary wheel axis and defining a minimal axial distance between the bearing elements. Such a spacer ring can prevent the axial distance and thus the height of the lubrication gap from being set too small and from thereby standing in the way of minimising wear and tear on the planetary wheel during operation.

With an inventive planetary gearbox, an annular lubricant collecting groove is embodied on an inner peripheral surface of the spacer ring. This lubricant collecting groove can be used to distribute lubricant between the bearing elements.

In a further development of the present invention, a plurality of lubricant ducts is embodied in the spacer ring, said lubricant ducts opening into the lubricant collecting groove. Lubricant can flow out of the lubricant collecting groove through these lubricant ducts in the direction of the lubrication gap.

Advantageously embodied in the planetary wheel axis is a lubricant supply duct, which opens radially into the lubricant collecting groove of the spacer ring. Lubricant can be supplied to the lubricant collecting groove provided in the spacer ring through such a lubricant supply duct in the form of rolling lubrication. The rolling lubrication allows the planetary gearbox to be operated further during emergency operation if the pressure feed lubrication fails.

Further features and advantages of the present invention will become significant with the aid of the description below of various embodiments of the inventive planetary gearbox with reference to the appended drawing, in which;

FIG 1	shows a schematic axial cross-sectional view of a planetary gearbox according to a first embodiment of the present invention;
FIG 2	shows an axial top view of a planetary carrier of the planetary gearbox shown in Figure 1 along the line II-II;
FIG 3	shows a perspective side view of a bearing element of the planetary gearbox shown in Figure 1;
FIG 4	shows an axial cross-sectional view of a planetary wheel bearing of the planetary gearbox shown in Figure 1;

FIG 5	shows the planetary wheel bearing shown in Figure 4 with identified dimensions;
FIG 6	shows an axial cross-sectional view of a planetary wheel bearing of a planetary gearbox according to a second embodiment of the present invention;
FIG 7	shows an axial cross-sectional view of a planetary wheel bearing of a planetary gearbox according to a third embodiment of the present invention; and
FIG 8	shows an axial cross-sectional view of a planetary wheel bearing of a planetary gearbox according to a fourth embodiment of the present invention.

Figures 1 to 5 show a planetary gearbox 1 according to a first embodiment of the present invention. The planetary gearbox 1 has a gearbox housing 2, which, on opposing end faces, is offset in each case by a drive shaft 3 and an output shaft 4. In the gearbox housing 2, a central sun wheel 5 with an external toothing 6 is held on the output shaft 4 so as to be able to rotate about a central gearbox axis of rotation Z. Arranged in the axial direction corresponding to the sun wheel 5 in the gearbox housing 2 is a hollow wheel 7 which is concentric to the central gearbox axis of rotation Z and has an internal toothing 8, said hollow wheel being fixedly connected to the gearbox housing 2 and surrounding the sun wheel 5. On the drive shaft 3, a planetary carrier 9 is held in the gearbox housing 2 so as to be able to rotate about the central gearbox axis of rotation Z.

The planetary carrier 9 has solebars 10, between which three planetary wheel axes 11 are arranged, of which only one planetary wheel axis is shown in Figure 1 for the sake of greater clarity. The planetary carrier 9 can also support one of three alternative numbers of planetary wheel axes. A planetary wheel bearing 12 embodied as a sliding bearing is provided on each planetary wheel axis 11, in which planetary wheel bearing a planetary wheel 13 is mounted so as to be able to rotate about a planetary wheel axis of rotation A. The planetary wheels 13 have external toothings 14, which engage with the internal toothing 8 of the hollow wheel 7 and the external toothing 6 of the sun wheel 5.

According to the invention, each planetary wheel bearing 12 comprises two annular bearing elements 12a, 12b, which are offset by the planetary wheel axis 11 and held in a torsion-resistant manner thereupon. Cone-shaped sliding surfaces 16 are embodied on the outer peripheral surfaces of the bearing elements 12a, 12b, which, with the central gearbox axis of rotation Z, each span an acute angle α, which preferably amounts to between 5° and 40°. Chamfers or suchlike can be provided in axial edge regions of the sliding surfaces 16, in order to counteract the formation of edges due to wear and tear.

Each bearing element 12a, 12b is radially offset by a lubricant duct 17. The lubricant duct 17 is connected to an eccentric lubricant supply duct 18, which axially offsets the planetary wheel axis 11. The lubricant duct 17 opens into a lubrication pocket 19, which is embodied as a flattening or recess in a minimally loaded region of the sliding surface 16 of the bearing element 12a, 12b. During regular operation of the planetary wheel bearing 12 in the form of a pressure feed lubrication, lubricant is supplied to the sliding surfaces 16 through the lubricant supply duct 18, the lubricant duct 17 and the lubrication pocket 19.

The tapered ends of the bearing elements 12a, 12b point to one another, wherein corresponding wheel treads 20 are embodied on inner peripheral surfaces of the planetary wheel 13 relative to the sliding surfaces 16 of the planetary wheel bearing 12.

A spacer ring 21 is arranged between the bearing elements 12a, 12b of the planetary wheel bearing 12, said spacer ring surrounding the planetary wheel axis 11 and defining a minimal distance between the bearing elements 12a, 12b. An annular lubricant collecting groove 22, into which a plurality of lubricant ducts 23 offsetting the spacer ring open, is embodied on an inner peripheral surface of the spacer ring 21. Corresponding to the lubricant collecting groove 22 of the spacer ring 21, a central lubricant supply duct 24 which opens into the lubricant collecting groove 22 of the spacer ring 21 is embodied in the planetary wheel axis 11. A rolling lubrication of the planetary wheel bearing 12 can take place through the central lubricant supply duct 24, the lubricant collecting groove 22 and the plurality of lubricant ducts 23, this being sufficient for an emergency operation of the planetary gearbox.

Figure 5 shows the planetary wheel bearing 12 of the planetary gearbox 1 according to the first embodiment of the present invention with identified dimensions. The axial widths b_1 and b_2 of the bearing elements 12a, 12b and b_3 of the spacer ring 21 fulfil the relationship $b_1 + b_2 + b_3 = B$, wherein B refers to the width of the planetary wheel bearing 12 between opposite solebars 10 of the planetary carrier 9. In this way the bearing elements 12a, 12b and the spacer ring 21 are axially fixed between the solebars 10 of the planetary carrier 9.

After manufacture, the bearing elements 12a, 12b and the spacer ring 21 firstly fulfil the relationship $b_1 + b_2 + b_3 > B$. During assembly of the planetary wheel bearing 12, the widths of the two bearing elements 12a, 12b and/or the spacer ring 21 are adjusted by machining post-processing, such that both the relationship $b_1 + b_2 + b_3 = B$ is fulfilled and a required height S of the lubrication gap 25 is also set by suitably selecting the widths b_1, b_2 and b_3. Here the relationship $\Delta S = \Delta b * \sin(\alpha)$ exists between a height change ΔS in the lubrication gap 25 and an axial adjustment of the respective bearing element 12a, 12b effected by a change in width Δb.

Figure 6 shows a planetary wheel bearing 12 of a planetary gearbox 1 according to a second embodiment of the present invention. Spacer elements 26 are inserted between a bearing element 12a and the spacer ring 21 and between the bearing elements 12a, 12b and in each case adjacent solebars 10 of the planetary carrier 9. The axial widths b_1, b_2 and b_3 of the bearing elements 12a, 12b and of the spacer ring 21 on the one hand and

D_1, D_2 and D_3 of the spacer elements 26 on the other hand fulfil the relationship $b_1 + b_2 + b_3 + D_1 + D_2 + D_3 = B$, so that the bearing elements 12a, 12b, the spacer ring 26 and the spacer elements 26 are axially set between opposite solebars 10 of the planetary carrier 9.

After manufacture, the bearing elements 12a, 12b and the spacer ring 21 firstly fulfil the relationship $b_1 + b_2 + b_3 < B$ in order to set the desired height S of the lubrication gap 25. Here b_1, b_2, b_3 and B are defined as in the embodiment shown in Figure 4, and D_1, D_2 and D_3 refer to the axial widths of the spacer elements 26.

The axial widths of the bearing elements 12a, 12b and of the spacer ring 21 firstly fulfil the relationship $b_1 + b_2 + b_3 < B$. During assembly, spacer elements 26 of a suitable thickness D_1, D_2 and D_3 are inserted at the specified locations such that the total axial thickness $D_1 + D_2 + D_3$ of all inserted spacer elements 26 is equal to the difference between the bearing width B and the total axial widths $b_1 + b_2 + b_3$ of the bearing elements 12a, 12b and of the spacer ring 21 and the lubrication gap S has the required height S.

Figure 7 shows a planetary wheel bearing 12 of a planetary gearbox 1 according to a third embodiment of the present invention. The adjustable bearing element 12a is screwed onto the planetary wheel axis 11. To this end, an internal thread and a corresponding external thread are embodied on the adjustable bearing element 12a and the planetary wheel axis 11 in each case. This screw connection 27 allows for a continuous adjustment of the axial position of the adjustable bearing element 12a on the planetary wheel axis 11. The axially fixed bearing element 12b is screwed to the adjacent solebar 10 of the planetary carrier 9, as a result of which the solebar 10 of the planetary carrier 9 acting as an axial stop defines the axial position of the axially fixed bearing element 12b. If the required height S of the lubrication gap 25 is reached, the bearing element 12a is fixed in the corresponding axial position by a torsion-proof securing element 28. Pins, bolts or suchlike can be used as the torsion-proof securing element 28. If the height S of the lubrication gap 25 changes over the course of time as a result of wear due to use, the adjustable bearing element 12a can be readjusted accordingly in order to reproduce the required height S of the lubrication gap 25.

Figure 8 shows a planetary wheel bearing 12 of a planetary gearbox 1 according to a fourth embodiment of the present invention. The adjustable bearing element 12b is screwed into an adjacent solebar 10 of the planetary carrier 9. To this end, an external thread and an internal thread are embodied on the adjustable bearing element 12b and in the solebar 10 in each case. This screw connection 27 allows the bearing element 12b to be screwed in the axial direction more or less into the solebar 10 of the planetary carrier 9 and to be fixed in any axial position using a torsion-proof securing element 28. Pins, bolts or suchlike can be used here as the torsion-proof securing element 28. The axial position of the axially fixed bearing element 12a is fixed by a radial annular shoulder 29 embodied on the planetary wheel axis 11 and serving as an axial stop. The required height S of the lubrication gap 25 can be set by screwing the adjustable bearing element 12b in or out. If the height S of the lubrication gap 25 of the planetary wheel bearing 12 has shifted as a result of wear due to use, the planetary wheel bearing 12 can be readjusted by axially adjusting the bearing element 12b accordingly.

For improved flexibility when adjusting planetary wheel bearings 12 of inventive planetary gearboxes 1, the procedure proposed in Figures 5 to 8 for setting an optimal height of the lubrication gap 25 of the planetary wheel bearing 12 can be combined with one another.

During operation of the planetary gearbox 1, the planetary carrier 9 is set to rotate by the drive shaft 3. Due to the engagement of its external toothing 14 into the internal toothing 8 of the hollow wheel 7, the planetary wheels 13 roll along the interior of the hollow wheel 7. By rotating the planetary wheels 13, due to the engagement of its external toothing 14 into the external toothing 6 of the sun wheel 5, the sun wheel 5 and with it the output shaft 4 are in turn set to rotate. In this way the output shaft 4 rotates at a higher speed than the drive shaft 3, because the planetary wheels 13 have a smaller periphery than the circle which the planetary wheel axes of rotation A describe during their rotation about the central gearbox axis of rotation Z of the planetary gearbox 1.

During operation of the slide bearing, lubricant is continuously supplied hereto through the central lubricant supply duct 24. The lubricant is firstly distributed in the lubricant collecting groove 22 of the spacer ring 21 and then flows through the at least one lubricant duct 23 in the direction of the lubrication gap 25.

One major advantage of the inventive planetary gearbox 1 lies in no additional axial slide bearings having to be provided in contrast to cylindrical slide bearings, in order to define the planetary wheel 13 in the axial direction. This renders the developing processing of additional axial sliding surfaces superfluous. A further advantage of the inventive slide bearing lies in the simple ability to set and readjust the height S of the lubrication gap 25, which allows for a larger component tolerance when the components required for the planetary wheel bearing are manufactured. Overall, during the use of an inventive planetary gearbox, cost advantages can be achieved from lower manufacturing costs and a longer service life of the planetary wheel bearing 12 which is increased due to readjustment options.

Although the invention has been illustrated and described in greater detail by the preferred exemplary embodiment, the invention is not limited by the examples disclosed and the person skilled in the art will be able to derive other variations on this basis without departing from the scope of protection of the invention, as it is defined by the appending claims.

Patentkrav

1. Planetgear (1), især til et vindkraftanlæg, med et gearhus (2), et centralt solhjul (5), som holdes roterbart i gearhuset (2) omkring en central geardrejeakse (Z) og bærer en udvendig fortanding (6), et hulhjul (7), som er anbragt koncentrisk til den centrale geardrejeakse (Z) i gearhuset (2) og har en indvendig fortanding (8), en planetbærer (9), som er drejeligt lejret i gearhuset (2) omkring den centrale geardrejeakse (Z), og flere planethjul (13), som ved hjælp af et planethjulleje (12), der er udformet som glideleje, er drejeligt lejret på planetbæreren (9) omkring planethjuldrejeakser (A) og har udvendige fortandinger (14, som er i indgreb med den indvendige fortanding (8) af hulhjulet (7) og den uvendige fortanding (6) af solhjulet (5),
kendetegnet ved, at hvert planethjulleje (12) omfatter to ringformede lejeelementer (12a, 12b), hvorigennem der går en planethjulakse (11), og som holdes drejefast på denne, hvor der på de udvendige omfangsflader af lejeelementerne (12a, 12b) er udformet keglekappeformede glideflader (16) på en sådan måde, at de tilspidsede ender af lejeelementerne (12a, 12b) vender mod hinanden, og hvor der på indvendige omfangsflader af planethjulet (13) er udformet løbeflader (20), som passer til glidefladerne (16) af planethjullejet (12).

2. Planetgear ifølge krav 1, **kendetegnet ved, at** lejeelementerne (12a, 12b) er anbragt med aksial afstand.

3. Planetgear ifølge et af de foregående krav,
kendetegnet ved, at mindst et lejeelement (12a, 12b) kan indstilles i aksial retning for at indstille en smørespalte (25) med defineret højde (S) mellem glidefladerne (16) af planethjullejet (12) og de passende løbeflader (20) af det lejrede planethjul (13).

4. Planetgear ifølge krav 3, **kendetegnet ved, at** præcist et lejeelement (12a, 12b) er indstilleligt, og det andet lejeelement (12a, 12b) har en aksialfast position.

5. Planetgear ifølge krav 4, **kendetegnet ved, at** den aksiale position af det aksialfaste lejeelement (12a, 12b) er defineret af et aksialanslag (10, 29), især en radial ringskulder (29), som er udformet på planethjulaksen (11), eller en vange (10) af planetbæreren (9).

6. Planetgear ifølge et af kravene 3 til 5,
kendetegnet ved, at justeringsmidler (26, 27, 28) til aksial indstilling er tilordnet det indstillelige lejeelement (12a, 12b).

7. Planetgear ifølge krav 6, **kendetegnet ved, at** afstandselementer (26) er tilvejebragt som justeringsmidler, der er anbragt mellem et lejeelement (12a, 12b) og en naboliggende vange (10) af planetbæreren (9) og/eller mellem lejeelementerne (12a, 12b).

8. Planetgear ifølge et af kravene 3 til 7,
kendetegnet ved, at det indstillelige lejeelement (12a, 12b) er påskruet planethjulaksen (11).

9. Planetgear ifølge et af kravene 3 til 7,
kendetegnet ved, at det indstillelige lejeelement (12a, 12b) er skruet ind i den naboliggende vange (10) af planetbæreren (9).

10. Planetgear ifølge et af kravene 8 eller 9,
kendetegnet ved, at der er tilvejebragt en fordrejningssikring (28), ved hjælp af hvilken det indstillelige lejeelement (12a, 12b) kan fastgøres aksialt.

11. Planetgear ifølge et af de foregående krav,
kendetegnet ved, at der i hver glideflade (16) er udformet mindst en smørelomme (19), ud i hvilken der munder en smøremiddelkanal (17), der radialt går gennem lejeelementet (12a, 12b), hvor smøremiddelkanalen (17) er forbundet med en ekscentrisk smøremiddeltilførselskanal (18), som er udformet i planethjulaksen (11) og aksialt går gennem denne.

12. Planetgear ifølge et af de foregående krav,
kendetegnet ved, at der mellem lejeelementerne (12a, 12b) er anbragt en afstandsring (21), som griber om planethjulaksen (11) og definerer en minimal aksial afstand mellem lejeelementerne (12a, 12b).

13. Planetgear ifølge krav 12, **kendetegnet ved, at** der på en indvendig omfangsflade af afstandsringen (21) er udformet en ringformet smøremiddelsamle-not (22).

14. Planetgear ifølge krav 13, **kendetegnet ved, at** der i afstandsringen (21) er udformet en flerhed af smøremiddelkanaler (23), som munder ud i smøremiddelsamle-noten (22).

15. Planetgear ifølge krav 14, **kendetegnet ved, at** der i planethjulaksen (11) er udformet en central smøremiddeltilførselskanal (24), der munder ud i smøremiddelsamle-noten (22) af afstandsringen (21).

FIG 1

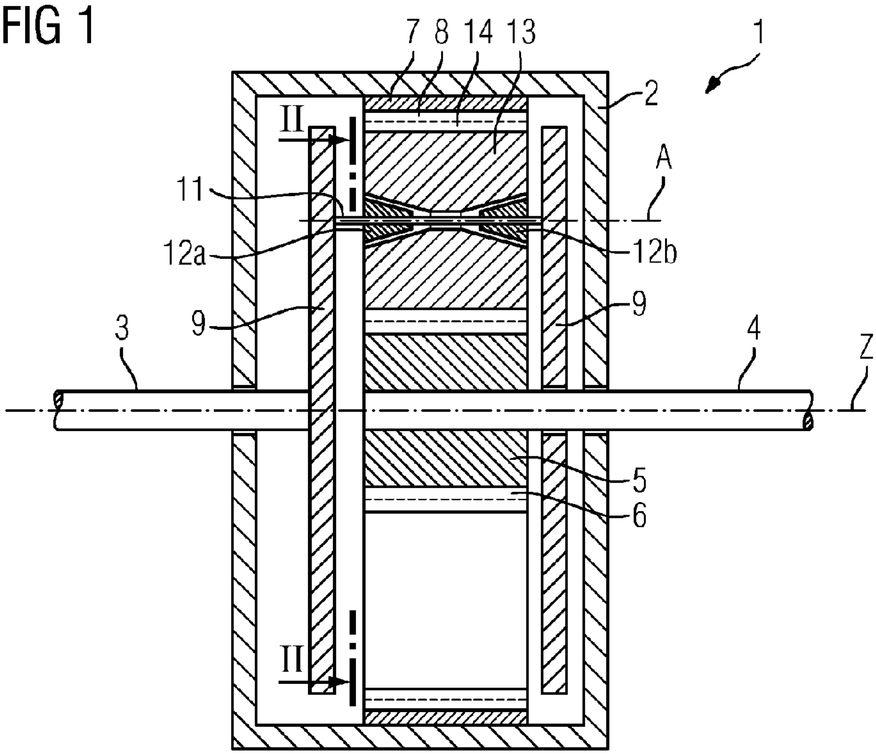


FIG 2

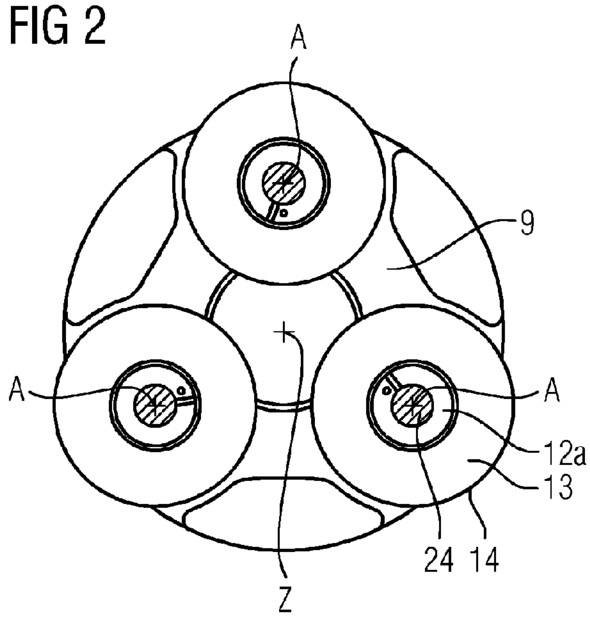


FIG 3

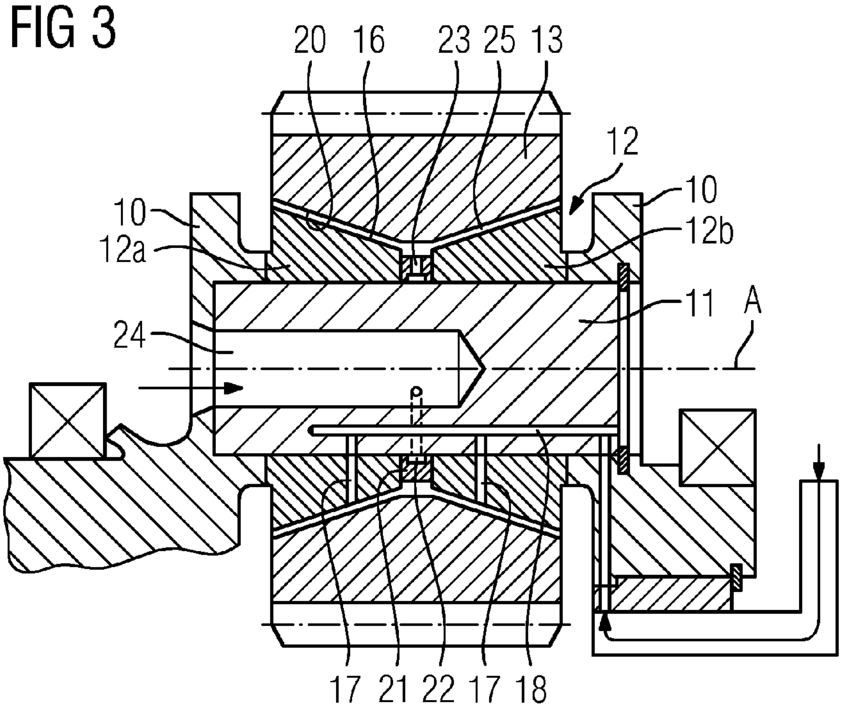


FIG 4

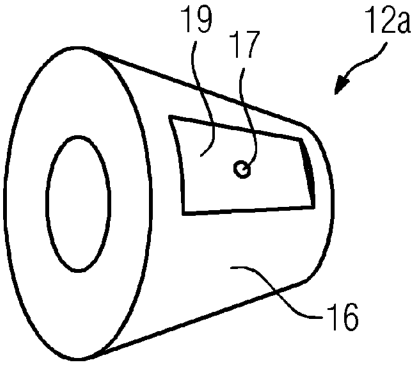


FIG 5

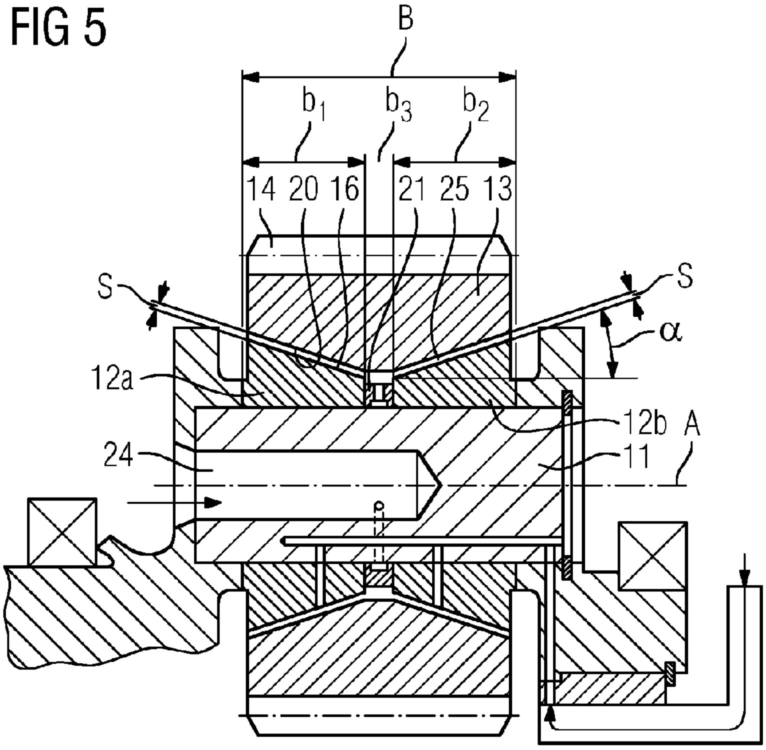

B
b1
b3
b2
14 20 16
21 25 13
S
S
α
12a
12b
A
24
11

FIG 6

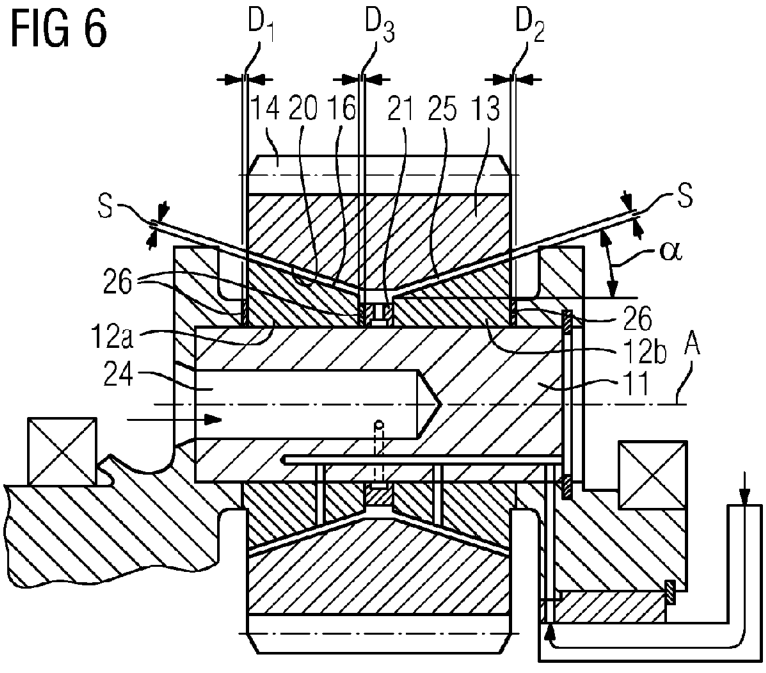
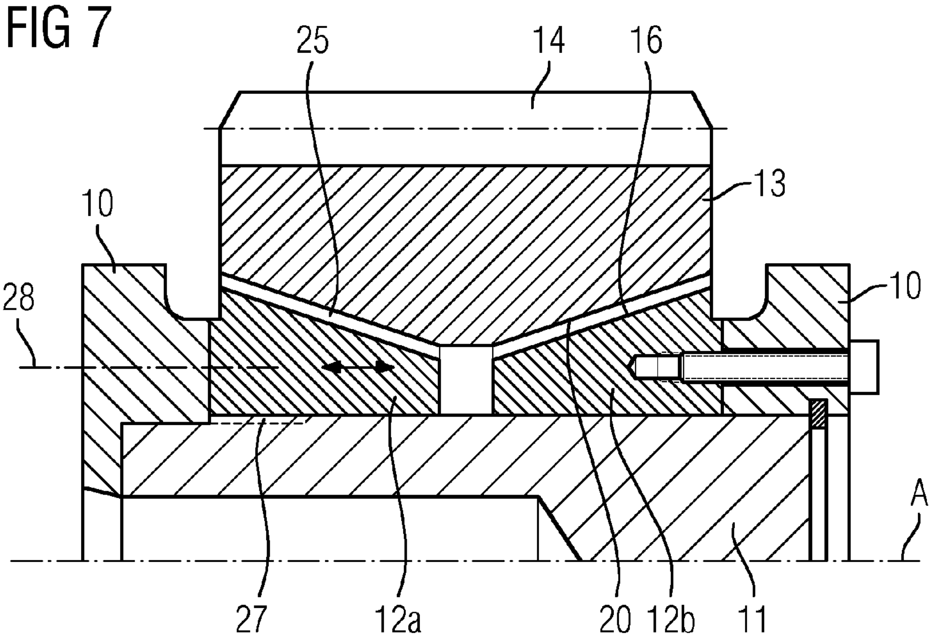

D1
D3
D2
14 20 16
21 25 13
S
S
α
26
26
12a
12b
A
24
11

FIG 7


25
14
16
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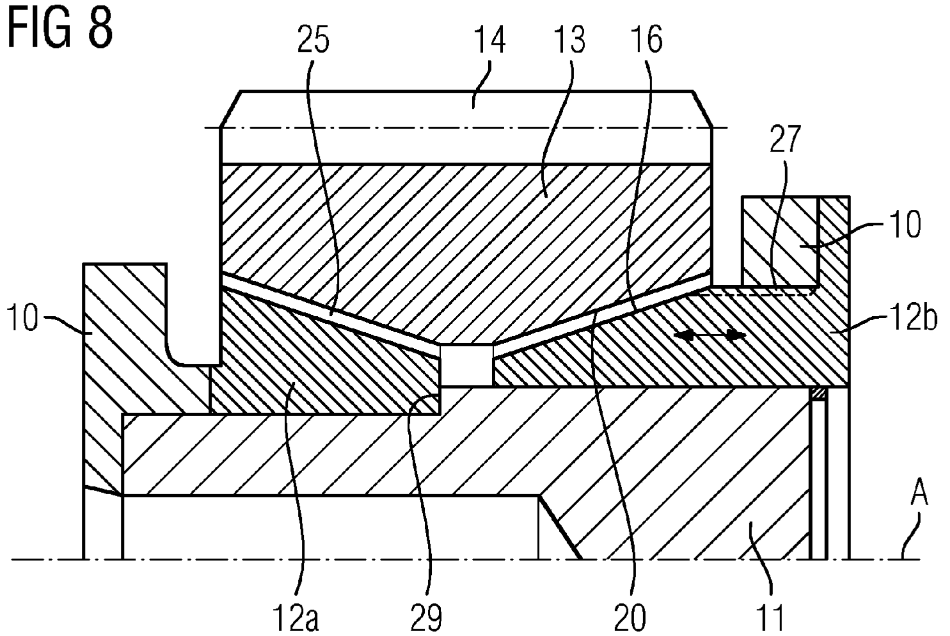
10
10
28
A
27
12a
20
12b
11

FIG 8


25
14
13
16
27
10
10
12b
A
12a
29
20
11