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(54) **Görgőscsapágy elrendezés**

Antifriction bearing assembly

The invention focuses on an antifriction bearing assembly, in particular for a rotary joint, with two against each other concentric, at least to some extent into one other arranged rings as well as a gap between these rings so that they are rotating against each other around an imaginary perpendicular axis in the center of those rings whereby in the area of the gap between the rings at least two sets of rolling elements with in each case to a defined rotational axis symmetrical, in particular cylindrical roller, tapered roller, needle, or barrel-shaped form are provided that are rolling between these rings, and whereby the rotational axes of the rolling elements of a first row are (approximately) directed radially relative to the rotational axis of the rings, while the rotational axes of the rolling elements of a second row are (approximately) directed parallel to the rotational axis of the rings.

Roller bearings provide the advantage of having linear contact between each rolling element and each guide surface in contrast to the just approximately punctiform contact in ball bearings. Because of the larger areas of contact vastly greater forces can be transmitted by roller-shaped rolling elements than with ball-shaped rolling elements. A disadvantage however consists in the fact that with roller shaped rolling elements only forces perpendicular to their lateral surface can be transmitted. Therefore pure roller bearings are either radial bearings or axial bearings. It has been tried already to achieve a certain guidance in direction of the rotational axis of the rolling elements by designing the running surface(s) trough-shaped, in particular groove-shaped. The friction between the facing sides of the rollers and these groove shoulders is not rolling friction but sliding fiction and therefore does not just only entail a comparably great energy loss but in addition also great wear and tear. Finally adjustment of the bearing clearance is insufficient and hardly possible – either rollers can move back and forth between groove shoulders dimensioned too wide, or they squeeze between groove shoulders of undersized distance.

Therefore if one does not want to waive the advantages of roller bearings, in many cases of application one is forced to combine two separately manufactured bearings – a radial bearing and an axial bearing – in order to be

able to transmit forces in different directions. This however implicates sometimes complicated connection and adjustment work in order to combine altogether four rings with the different machine components while still ensuring the ease of movement to be expected from antifriction bearings. At that particularly with large-diameter antifriction slewing rings of 1000 mm and more it proves aggravating to bring the rotational axes of both antifriction bearings in exact alignment. If this fails the circulating rolling elements will not find a smooth and even running surface and are subject to increased wear and tear.

Seite 2a eingefügt:

FR2387373 A1 reveals an antifriction bearing assembly for a rotary joint with two against each other concentric, at least to some extent into one other arranged rings, comprising one each joint plane for connection to one each component part of vehicle, machine or equipment or foundation as well as connecting elements each in form of fastening or blind hole drilled holes, furthermore with a gap between these rings so that they are rotating around an imaginary axis perpendicular to the ring plane in the center of those rings, whereby in the area of the gap between both rings additionally one row of ball-shaped rolling elements is provided, where one ring exhibits an approximately L-shaped cross section similar to a short piece of tubing or sleeve as tubular section with a ring flange or collar attached at the face side, whereby the gap exhibits an approximately L-shaped cross section, whereby the other ring to some extent finds space in the cross sectionally hollow area between both L-shaped legs of the ring of L-shaped cross section and thereby retains the rolling elements in place inserted inside the gap, whereby the path of the ball-shaped rolling elements in the gap cross section is arranged in the area of the tubular ring section, and whereby the raceways of each ring for the row of rollers respectively roller shaped rolling elements and for the row of ball-shaped rolling elements are shaped at one single joint undivided solid of revolution.

Einfügung Seite 2a Ende

From these disadvantages of the described state of the art results that the problem initiating the invention, to advance a generically antifriction bearing assembly so that the transmission of great forces in radial as well as in axial direction is enabled, whereby axial forces in both directions must be borne so that the tilting forces can be absorbed. Hereby it would be desirable if assembly

would not be more complicated than in the case of a comparable ball bearing and if wear and tear would be as little as possible.

The solution of the problem succeeds with an antifriction bearing assembly according to claim 1.

Therefore two rows of roller-shaped rolling elements with rotational axes differently oriented so that one row of rolling elements can transmit radial forces and the other one axial forces are jointly integrated with one row of ball-shaped rolling elements. By such integration the necessity of adjustment of both rotational axis to obtain joint alignment of the same is eliminated at the assembly. In this connection the term "approximately radial" means at least "predominantly radial" in terms of "angle between the rotational axis of a rolling element and the rotational axis of the rings is greater than 50 degrees" preferably however "close to radial" in terms of "angle between the rotational axis of a rolling element and the rotational axis of the rings is greater than 85 degrees", and the term "approximately parallel" means at least "predominantly parallel" in terms of "angle between the rotational axis of a rolling element and the rotational axis of the rings is less than 40 degrees", preferably however even "close to parallel" in terms of "angle between the rotational axis of a rolling element and the rotational axis of the rings is less than 5 degrees, or a point of intersection does actually not exist (ideal parallelism)". The additional row of balls serves the purpose of absorbing axial pulling forces between both rings in opposite direction to the axial compressive force absorbed by the axially loadable rollers so that for example horizontally installed antifriction bearings with the axial load on the same predominantly vertical pointing down can also absorb tilting effects whereby one part of the antifriction bearing tends to tilt upwards.

It is within the scope of the invention that the raceways of both rings for the ball-shaped rolling elements are offset against each other so that a bearing angle of more than 0 degrees is resulting, for example of more than 15 degrees, in particular of more than 30 degrees. There such offset results from the different heights of the centers of the raceways in relation to the raceway of the roller bearing absorbing the axial forces, whereby in the ideal case assumed in this

connection both rings are brought together as close as possible in axial direction corresponding to the diameter of the rollers of that roller bearing. Thereby the line through both contact points or areas formed by a ball together with both ball raceways slants down towards the raceways of the roller bearing absorbing the axial forces. At one ring exhibiting the lower raceway of the roller bearing the contact point or area between the balls and the ball raceway moves upward, at the other ring carrying the upper raceway of the axial roller bearing in contrast thereto it moves downward. The relevant inclination angle against the bearing plane being the support angle is preferably less than 60 degrees, and is in most cases approximately 45 degrees.

It has proven advantageous to have rolling elements with rotational axes oriented (approximately) parallel to the rotary axis of the rings in one plane, which is penetrated by the rotary axis of the rings approximately perpendicular, corresponding to a single row radial bearing.

The invention furthermore recommends to design the arrangement so that the plane of rolling elements with rotational axes being (approximately) parallel to the rotary axis of the rings divides the surface facing the common gap of each ring in two sections, of which only one exhibits guide surface(s) for rolling elements with rotational axes running approximately radial referred to the rotary axis of the rings. Such an arrangement facilitates the assembly of an antifriction bearing according to the invention.

The same purpose serves a design instruction according to which each ring exhibits guide surface(s) facing only one single connecting face for rolling elements with rotational axes running radial referred to the rotary axis of the rings. Therefore the rings with the rolling elements inserted in between can easily be assembled in axial direction.

Especially preferred is an arrangement at which all rolling elements with rotational axes running radial referred to the rotary axis of the rings are coplanar in one single joint plane. Resulting thereof is a particularly simple cross sectional geometry of the rings.

The invention can even be advanced so that one row of rolling elements with rotational axes running radial referred to the rotary axis of the rings is comprised of rollers being divided crosswise to their longitudinal axes. In such case the rollers circulating farther out can rotate at a different speed than the rollers circulating inside so that in spite of an altogether rather large contact area between rolling elements and rings hardly any wearing sliding friction occurs, but only low-wear rolling friction.

Since all rolling elements of an in each case to a defined rotational axis symmetric form, in particular cylinder roller, tapered roller, needle, or barrel-shape form, perhaps with curved faces and edges as the case may be, exhibit almost equal diameters independent from the orientation of their rotational axes, they can be produced from the same rod-shaped raw material, which not only facilitates their production, but also their assembly; they will then even have identical diameters. The term "identical diameters" in this respect is understood so that the deviation of one specified diameter (variance) is in each case smaller than a specified value, for example less than 10 µm. preferably less than 6 µm, in particular less than 2 µm. Such condition can be achieved by sorting. Different loads in axial respectively radial direction can be allowed for either by differently long rollers, or by stringing together face-to-face of a corresponding number of identical rollers, perhaps for running in the same cage. However also rollers of different cross sections can be used.

Preferably are all rolling elements of an in each case to a defined rotational axis symmetric as well as roller or barrel-shaped form in exactly two planes that are parallel to each other, namely in one plane the rolling elements with (approximately) radial rotational axis, and in another plane coplanar with but in axial direction along the rotational axis of the rings offset against the first one, the rolling elements with (approximately) axial rotational axis. In this way the design height of the antifriction bearing according to the invention is minimized, which is of advantage in some cases of application.

Since according to the invention the guide surfaces of each ring for all rolling elements are accomplished at one each single undivided solid of rotation, the manufacturing effort can be further minimized, as altogether only two ring parts

have to be assembled and not three as it would be necessary in case of a divided ring.

The invention recommends to design at least one guide surface for one row of roller or barrel-shaped rolling elements having their rotational axes axially parallel to the rotary axis of the rings as perhaps circumferential groove of approximately rectangular cross section, In that groove the rolling elements contained in it are guided in axial direction during operation, and also already during assembly.

When the groove-shaped guide surface is shaped at the outer lateral surface of the ring radially located inside, it is easier to equip it with the rolling elements, which are preferably inserted piece by piece, than it is into a groove provided at the inner lateral surface of a ring.

According to the invention the gap between both rings exhibits an approximately L-shaped cross section. To each of both sides of such a L-shaped cross section at least one row of rolling elements with accordingly oriented rotational axes is allocated.

For generating such a L-shaped gap cross section according to the invention one ring, preferably the internal ring, exhibits an approximately L-shaped cross section similar to a short piece of tubing or sleeve with a ring flange or collar attached at the face side. The other ring finds its place partly inside the cross sectionally hollow area between both L-sides and there keeps the rolling elements inserted in the gap in place.

Specific advantages can be accomplished by the fact that according to the invention at least one row of roller or barrel-shaped rolling elements with rotational axes running axially parallel referred to the rotational axis of the rings is located in a section of the gap between the row of roller or barrel-shaped rolling elements with rotational axes running axially parallel referred to the rotational axis of the rings at the one hand, and the additional row of ball-shaped rolling elements at the other hand. Because in such case the axial distance between both planes of roller or barrel-shaped rolling elements is

minimal so that they are supporting each other and disadvantageous tilting of both rings involved is almost excluded.

In the area of at least one row of rolling elements, preferably in the area of all rows of rolling elements, one cage each and/or set of distance elements can be arranged in order to guide the rolling elements at uniform distances and to ensure consistent load bearing capacity.

The connection elements in the area of a ring should be designed as blind hole drilled holes so that they will not collide with the other – cross section-wise L-shaped – ring.

In that case the direction of the longitudinal axes of the blind hole drilled holes is allowed to intersect with or at least be tangent to the raceway for at least one row of roller or barrel-shape rolling elements with rotational axes running radial referred to the rotary axis of the rings. In this way the space requirement of the antifriction bearing according to the invention is reduced.

Finally it corresponds to the essence of the invention that at one ring a circumferential tooth system is provided, preferably at the radially outer ring, in particular at its outer lateral surface, so that a defined relative distortion is enabled, for example through a motor-driven worm or pinion intermeshing with such tooth system.

Further characteristics, details: Advantages und effects on basis of the invention result from the following description of a preferred embodiment of the invention as well as by reference to the drawing. The only figure of it shows a section through an antifriction bearing assembly according to the invention along a radial plane parallel to the rotary axis of the rings.

The antifriction bearing assembly 1 according to the invention from the drawing consists of two against each other concentric, at least to some extent into one other arranged rings 2, 3 with a gap 4 between these rings 2, 3 so that they are rotating against each other around an imaginary perpendicular axis in the center of those rings 2, 3.

In the area of the gap 4 between the rings 2, 3 several rows of rolling elements 5, 6, 7 are located, which are rolling off on raceways 8, 9, 10, 11, 12, 13 in each case opposite to each other in both rings 2, 3 and thus enable a free-moving relative distortion.

Ring 2 radially located further out in the depicted example has a basic shape of approximately quadratic cross section on which however the raceways 8, 10, 12 for the rolling elements 5, 6, 7 are machined out.

Ring 3 radially located further inside on the contrary has an approximately L-shaped cross section on which however the raceways 9, 11, 13 for the rolling elements 5, 6, 7 are machined out.

Thereby the cross-sectional part 14 oriented vertically in the figure parallel to the rotational axis of that ring 3 represents a ring section with a geometry similar to a very short tube 14, while the horizontally oriented cross part 15 represents a circular disk ring 15 at the face side of that tube 14 in the style of a circumferential collar, flange, or brim. Between the tube-shaped section 14 and the brim-like section 15 remains a hollow 16, into which the other ring 2 radially located further outward overlaps. Simultaneously the surfaces 17, 18 delimiting this hollow 16 surround the gap 4 and incorporate the raceways 9, 11, 13 for the rolling elements 5, 6, 7. Between both surfaces 17, 18 is a circumferential step 19 so that both surfaces 17, 18 do not directly border on each other, but are separated by this step. Height and width of that step19 are slightly smaller than the diameter of a rolling element 5, 6, 7 and partly fill the gap 4 in the area of the L-kink so that the gap volume is minimal.

In the described exemplary embodiment the entire surface 17 forms the raceway 9 for the roller-shaped rolling elements 5 of which their rotational axes are oriented radial referred to the not depicted rotary axis of the rings. These rolling elements 5 preferably have a cylindrical form (cylindrical rollers), however they also can be designed frustoconical (tapered rollers), corresponding to the differing raceway circumference at the inner and outer diameter of the raceway 9. In the described example however cylindrical rollers

5 are used, which instead each are divided across to their longitudinal direction into an inner and an outer roller that can rotate at different speeds. However they are jointly held at equidistant distances by a cage 20, which is only one-sided, namely at its radially outer rim, closed to a ring. The raceway 8 of the outer ring 2 for these rolling elements 5 and the cage 20 is designed as circumferential step at the face side 21 towards gap of that ring 2, in which the rolling elements 5 and the cage 20 almost completely plunge in.

At the face side 22 of ring 2 located opposite to this face side 21 provided with the raceway / step 8 are blind hole drilled holes 23 with internal threads spread around annular for the connection of this ring 2 to a machine part or foundation.

At the radially outer side 24 of that ring 2 a circumferential tooth system 25 is provided, which may extend almost over the entire height of ring 2, preferably however not completely up to its connecting face 22.

The concave respectively radially inlying side 26 of that ring 2 is facing the gap 4. At its lower end is a rectangular, or – in the present example – trapezoidal recess respective step as raceway 10 for one row of cylindrical roller-shaped rolling elements 6 with rotational axes parallel to the rotary axis of the rings. Raceway 11 in the opposite surface 18 of the inlying ring 3 is groove-shaped with rectangular cross section. The rolling elements 6 of this row are held at equidistant distance by another cage 27. The distance between the groove-shaped raceway 11 and the circular raceway 9 measured in direction of the rotary axis of the rings is a little (slightly) greater than the height of the stepping 19 plus a circumferential ring 28 of cage 27 circulating at the edge.

The cylindrical roller-shaped rolling elements 5, 6 with – referring to the rotary axis of the rings – radially oriented rotational axes and with the in contrast axially oriented rotational axis preferably have equal or at least almost equal diameters (± 10 %), and preferably equal or at least almost equal lengths (± 10 %) so that the forces transmittable by a rolling element 5 are approximately commensurate to the forces transmittable by a rolling element 6. Since each time two of the rolling elements 5 are arranged behind one other in the present example, the thrust force transmittable by this antifriction bearing assembly 1 in

axial direction is approximately twice as great as the force transmittable in radial direction. By using identical rollers 5, 6 for the rows with radial and axial rotational axis orientation the production of an antifriction bearing according to the invention can be simplified.

To enable also transmission of pulling forces in axial direction an additional row of ball-shaped rolling elements 7 is provided. That row is located within the gap 4 aside from both rows with roller-shaped rolling elements 5, 6 in the area of the tubular ring section 14 respectively of the rimmed lateral surface 18 in the area of the gap 4. The diameter of a ball 7 corresponds roughly to the diameter of the rollers 5, 6. The raceways 12, 13 in both rings 2, 3 for the ball-shaped rolling elements 7 each have approximately the same cross section, namely the shape of a circular segment with a radius slightly greater than the one of the balls 7. As both raceways 12, 13 are slightly offset against each other in the assembled condition of the antifriction bearing 1 according to the drawing – raceway 13 in ring 3 with L-shaped cross section is relative to the rotary axis of the rings slightly closer to the radially oriented section of the gap 4 than the opposite raceway 12 at the other ring 2 – an axial compressive force is generated between both rings 2, 3, which presses the same tightly against the radially oriented rollers 5. A separate cage holds the balls 7at equidistant distances.

Beyond the row of ball-shaped rolling elements 7 the gap 4 is closed by a seal 30. As the face side 31 of the inner, respectively by cross section L-shaped ring 3 bordering here with the gap 4 is set back against the connection face 22 of the outer ring 2 by a slight amount of some millimeters, that seal 30 can be realized as elastic gasket ring, which is inserted and anchored at a circumferential grove 32 at the inner side 26 of the outer ring 2 and which is pressing with at least one sealing lip – in the present example with 2 sealing lips – against the face side 31 of inner by cross section L-shaped ring 3. Also at the other end of the gap 4 a seal could be provided, which however is not necessarily required.

The facing side 33 of the radially inner by cross section L-shaped ring 3 delimiting the brim-shape ring section 15 at the outside located opposite to facing side 31 serves as its connection face to a component part of vehicle,

machine or equipment or foundation. That connection face 33 is provided with drilled fastening through holes 34. These fastening through holes 34 are equally spaced around the circumference of ring 3, run perpendicular to connection face 33 and extend through the tubular ring section 14 to the face surface 31 at the opposite side so that machine bolts can be stuck through these drill holes 34.

Görgőscsapágy elrendezés

Szabadalmi igénypontok

1. Görgőscsapágy elrendezés (1) forgó kapcsolathoz, két, egymáshoz képest koncentrikusan, valamint legalább tartományokban egymásban elrendezett gyűrűvel (2, 3), melyek rendelkeznek egy jármű-, gép- vagy berendezésrészhez vagy alaphoz való csatlakozáshoz egy-egy csatlakozófelülettel (22, 33), továbbá egy-egy rögzítő- vagy zsákfuratokként (23, 34) kiképzett csatlakozóelemmel, továbbá egy ezen gyűrűk (2, 3) között réssel (4), úgy, hogy a gyűrűk (2, 3) egy képzeletbeli, a gyűrűk (2, 3) központjában húzódó, a gyűrűk síkjára hozzávetőlegesen merőleges tengely körül egymáshoz képest elforgathatók, ahol a gyűrűk (2, 3) közötti rés (4) tartományában legalább két, egy meghatározott forgástengelyre szimmetrikus valamint hengergörgő-, kúpgörgő-, tű- vagy hordószerű alakú gördülőtest (5, 6) sor található, és ahol egy első sor gördülőtesteinek (5) forgástengelyei a gyűrűk forgástengelyéhez képest túlnyomórészt sugárirányban húzódnak, míg az egy második sor gördülőtesteinek (6) forgástengelyei túlnyomórészt a gyűrű forgástengellyel párhuzamosan húzódnak, ahol a gyűrűk (2, 3) közötti résben (4) továbbá egy golyó alakú gördülőtest (7) sor van elhelyezve, ahol egy gyűrű (2, 3) hozzávetőlegesen L keresztmetszetű egy rövid cső- vagy karmantyúdarabként kialakított csőszerű szakasszal (14), valamint egy homlokoldalon kiképzett gyűrűperemmel vagy -szegéllyel (15), ahol

a) a rés (4) L keresztmetszetű, ahol

b) az L keresztmetszetű gyűrűnek (2, 3) az L két szára közötti, üreges keresztmetszeti tartományában van a másik gyűrű (3, 4) részben behelyezve és ennek révén a résen (4) belül elhelyezett gördülőtesteket (6) helyükön rögzíti,

c) ahol a cső alakú gyűrűszakasz (14) tartományában a rés keresztmetszetben a golyó alakú gördülőtest (7) sor a legalább két görgő alakú gördülőtest (5, 6) sor túloldalán van elrendezve,

d) és ahol minden egyes gyűrű (2, 3) futópályái (8, 13) a legalább két hengergörgő-, kúpgörgő-, tű- vagy hordószerű alakú gördülőtest (5, 6) sora számára valamint a golyó alakú gördülőtestek (7) sora számára egy-egy egyetlen, közös, osztatlan forgástesten (2, 3) vannak kialakítva.

2. Az 1. igénypont szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a gyűrűk (2, 3) futópályái a golyó alakú gördülőtestek (7) számára tengelyirányban úgy vannak egymáshoz képest eltolva, hogy egy 0°-nál nagyobb, például egy 15°-nál nagyobb, különösen 30°-nál nagyobb tartószög adódik.

3. Az 1. vagy 2. igénypont szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a gördülőtestek (6), melyek forgástengelyei túlnyomórészt a gyűrűk forgástengelyével párhuzamosan húzódnak, egy olyan síkban fekszenek, amelyen a gyűrűk forgástengelye körülbelül merőlegesen hatol át.

4. A 3. igénypont szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a gyűrűk forgástengelyével túlnyomórészt párhuzamos forgástengelyű gördülőtestek (6) síkja mindegyik gyűrű (2, 3) közös rés (4) felé eső felületét (17, 18, 21, 26) két szakaszra osztja, melyek közül csupán egy-egy (17, 21) rendelkezik a gyűrűk forgástengelyére túlnyomórészt sugárirányban húzódó forgástengelyű gördülőtestek (5) számára futópályával (8, 9).

5. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a gyűrűk forgástengelyére sugárirányban húzódó forgástengelyű gördülőtestek (6) (mindegyik) futópályája (8, 9)

mindegyik gyűrűn (2, 3) csupán egyetlen homlokoldal felé, előnyösen a vonatkozó (a gyűrűk forgástengelyéhez képest tengelyirányban túlnyúló) csatlakozófelülettel (22, 33) szemben elhelyezkedő homlokoldal (21, 31) felé néz.

6. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a gyűrűk forgástengelyére sugárirányban húzódó forgástengelyű összes gördülőtest (5) egyetlen közös síkban fekszik.

7. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy egy a gyűrűk forgástengelyére sugárirányban húzódó forgástengelyű gördülőtest (5) sort hossziránykra keresztirányban osztott görgők alkotnak.

8. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy az egy meghatározott forgástengelyre szimmetrikus, különösen hengergörgő-, kúpgörgő-, tű- vagy hordószerű alakú összes gördülőtest (5, 6) forgástengelyének irányától függetlenül közel azonos, azaz ±10%-nál kisebb, különösen ±5%-nál kisebb, különösen ±2%-nál kisebb eltérésű átmérőjű.

9. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy az egy meghatározott forgástengelyre szimmetrikus, valamint gőrgő- vagy hordószerű alakú összes gördülőtest (5, 6) pontosan két egymással párhuzamos síkban található.

10. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy az egy meghatározott forgástengelyre szimmetrikus, valamint hengergörgő-, kúpgörgő-, tű- vagy hordószerű alakú, a gyűrűk forgástengelyével tengelypárhuzamosan húzódó forgástengelyű gördülőtest (6) sor legalább egy futópályáját (10, 11) egyenes, előnyösen egymással párhuzamos oldalélű és különösen hozzávetőlegesen négyszögletes keresztmetszetű körbefutó horony (11) képezi.

11. A 10. igénypont szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a horony alakú futópálya (11) a sugárirányban belső gyűrű (3) domborúan ívelt palástfelületén (18) van kialakítva.

12. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a sugárirányban belső gyűrű (3) megközelítőleg L keresztmetszetű.

13. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy legalább egy gördülőtest (5-7) sor tartományában, előnyösen az összes gördülőtest sor (5-7) tartományában egy-egy kosár (20.27, 29) és/vagy távtartó elemek vannak elhelyezve.

14. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy az egy meghatározott forgástengelyre szimmetrikus, valamint hengergörgő-, kúpgörgő-, tű- vagy hordószerű alakú gördülőtesteknek (5, 6) domború homlokoldalai vagy homlokélei vannak.

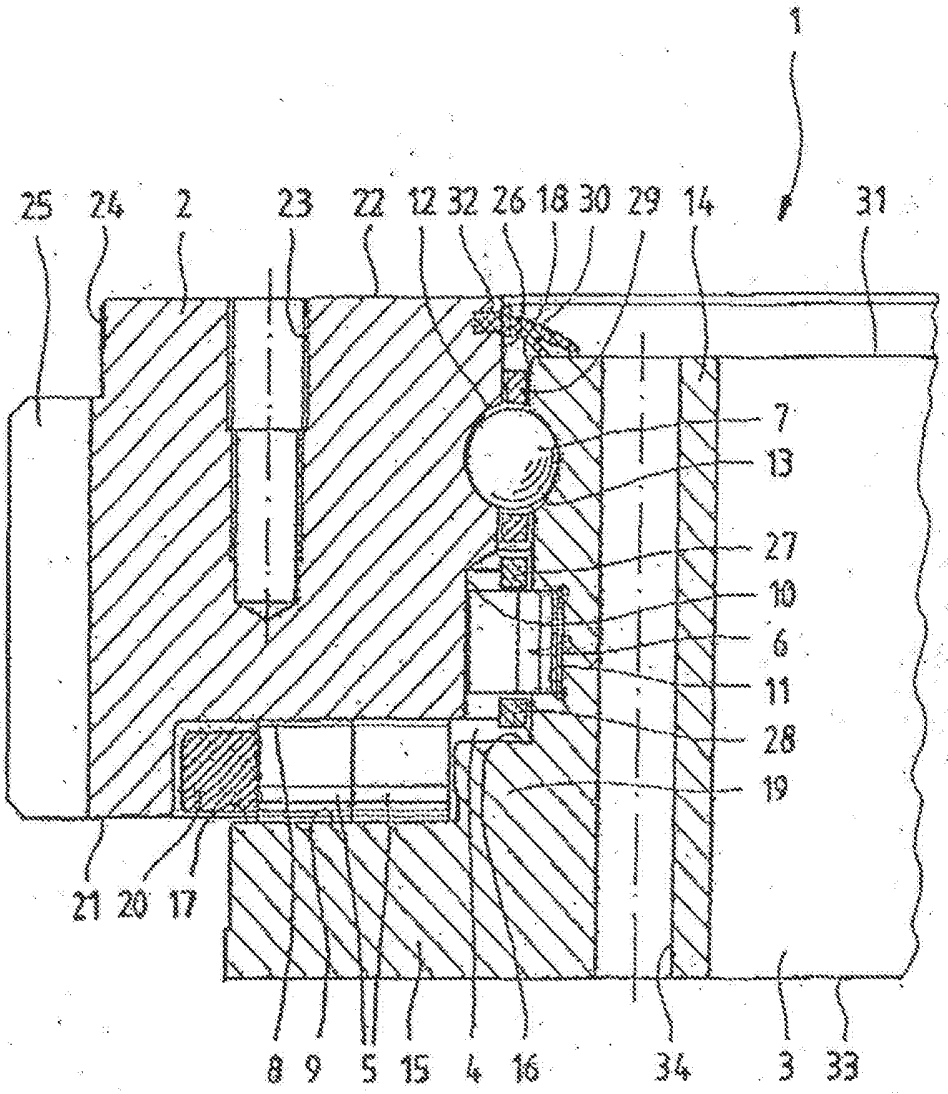
15. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy a zsákfuratok (23) hossztengelyének iránya a legalább egy, egy meghatározott forgástengelyre szimmetrikus, valamint hengergörgő-, kúpgörgő-, tű- vagy harangszerű alakú, a gyűrűk forgástengelyére sugárirányban húzódó forgástengelyű gördülőtest (5) sorhoz tartozó érintett gyűrű (2) futópályáját (8) metszi vagy legalább érinti.

16. Az előző igénypontok bármelyike szerinti görgőscsapágy elrendezés, azzal jellemezve, hogy egy gyűrűn (2, 3), előnyösen a sugárirányban kívül fekvő gyűrűn (2), különösen annak egy domborúan ívelt palástfelületén (24) körbefutó fogazás (25) van kiképezve.

Tisztelettel a meghatalmazott:

DR. KRAJNYÁK & TÁRSA
ÜGYVÉDI IRODA

Fig. 1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