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(54) Title: COMPACT PINION AND BEARING ASSEMBLY

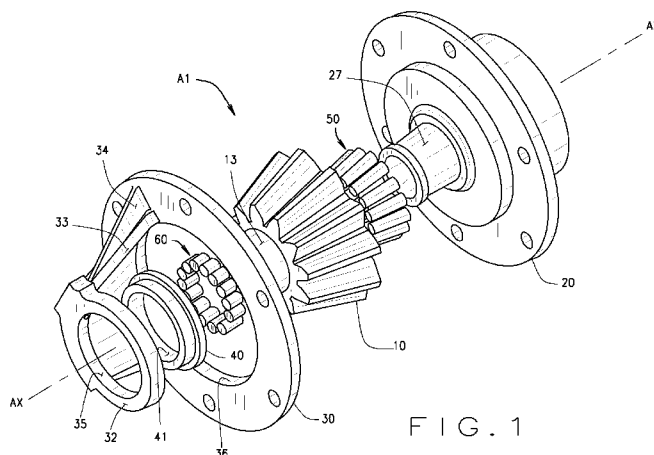


FIG. 1

(57) Abstract: An axially compact and highly integrated bevel pinion and bearing assembly (A1, A2) wherein a pinion gear (10) is integrated with a first support bearing (B1), and the pinion shaft (12) is integrated with a second support bearing (B2) axially displaced from the first support bearing (B1). With the first support bearing (B1) arranged axially under the pinion gear teeth (11), loads from the pinion gear (10) are transferred primarily through the first support bearing (B1). As a result, the second support bearing (B2) can be reduced in size and brought axially closer to the first support bearing (B1) to reduce the axial length of the pinion and bearing assembly. The integration of the pinion gear (10) with the first and second support bearings (B1, B2) optimizes material utilization and reduces overall weight of the pinion and bearing assembly, leading to a power density improvement.

COMPACT PINION AND BEARING ASSEMBLY

CROSS-REFERENCE TO RELATED APPLICATIONS

The present application is related to, and claims priority from, U.S.
5 Provisional Patent Application Serial No. 61/315,576 filed on March 19, 2010,
and which is herein incorporated by reference.

STATEMENT REGARDING FEDERALLY SPONSORED RESEARCH

Not Applicable.

BACKGROUND OF THE INVENTION

10 The present application relates generally to bevel pinion and gear
drives as used in differential transmissions, and in particular, to a compact
pinion and bearing assembly for use in a gear drive transmission.

Pinion and gear drives are commonly used in differentials for land
vehicles as well as in rotorcraft transmissions for aerospace applications. In
15 most pinion and gear drives, the pinion shaft is supported by a pair of rolling
element bearings arranged on one side, usually the back (larger) side, of a
bevel pinion. In these arrangements, the bevel pinion is overhung axially
beyond one end of the supporting bearings and, thus a tilting moment
develops. Adequate bearing support stiffness is very important to maintain a
20 good gear meshing quality, reduced noise level, and a prolonged service life
for the pinion and bearing. To combat the tilting moment, an alternative
configuration employing a back-to-back bearing arrangement with sufficient
bearing spread is often used. This results in a pinion and bearing system that
is somewhat long in axial direction. Additionally, the supporting bearing which
25 is axially disposed closer to the bevel pinion receives greater radial loads than
the pinion itself receives, resulting in the need for a large bearing.

In an effort to reduce the bearing size, some designs employ a straddle
mount arrangement for the bearings and pinion gear. In a straddle mount
arrangement, a nose bearing is installed axially in front of the pinion gear to
30 share the pinion load with a supporting bearing installed axially behind the
pinion gear, and thus improve the bearing supporting stiffness. Such

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arrangements are more complex, requiring a side wall or inwardly projecting structure associated with the pinion gear to host the nose bearing. For example, a nose bearing may be positioned within a recess on a front face of a pinion gear, requiring structural support from a side wall in the transmission casing to support the bearing. One problem associated with employing a side wall of the transmission casing to support a nose bearing is that there is only a limited volume of space available within the interior confines of a differential enclosure or transmission housing, thereby complicating setting of the bearing.

Accordingly, it would be advantageous to provide a compact bevel pinion and bearing assembly for use in a gear transmission which reduces tilting moments and stresses on the pinion bearings, and which does not require excessive axial length.

BRIEF SUMMARY OF THE INVENTION

Briefly stated, the present disclosure provides a compact and highly integrated bevel pinion and bearing assembly wherein the pinion gear is integrated with a first support bearing consisting of a tapered roller bearing, and the pinion shaft is integrated with a second support bearing consisting of a second rolling element bearing. With the first support bearing arranged directly under the pinion gear, loads from the pinion gear are transferred primarily through the tapered roller bearing. As a result, the second bearing can be reduced in size and brought axially closer to the first bearing to reduce the axial length of the pinion and bearing assembly. The integration of the pinion gear with support bearings optimizes material utilization and reduces overall weight of the pinion and bearing assembly, leading to a power density improvement.

In an alternate configuration of the pinion and bearing assembly of the present disclosure, an overrunning clutch is integrated with the pinion and bearings. Integration of the pinion and bearing assembly with an overrunning clutch reduces the total number of bearings required, further improving system power density.

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The foregoing features, and advantages set forth in the present disclosure as well as presently preferred embodiments will become more apparent from the reading of the following description in connection with the accompanying drawings.

5 BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

In the accompanying drawings which form part of the specification:

Figure 1 is an exploded perspective view of a pinion and bearing assembly of the present disclosure;

10 Figure 2 is a sectional view of the pinion and bearing assembly of the present disclosure;

Figure 3 is a perspective sectional view of a bevel pinion gear and integrated pinion shaft of the present disclosure;

Figure 4 is a perspective sectional view of a flange hub of the present disclosure;

15 Figure 5 is a perspective view of a support bracket of the present disclosure;

Figure 6 is a perspective sectional view of an outer race ring for a nose bearing of the present disclosure;

20 Figure 7 is a perspective sectional view of an alternate embodiment of the pinion and bearing assembly of the present disclosure, incorporating an over-running clutch assembly within the flange hub;

Figure 8 is an exploded perspective view of the alternate embodiment shown in Fig. 7; and

25 Figure 9 is an axial sectional view of the over-running clutch assembly shown in Fig. 7 and Fig. 8.

Corresponding reference numerals indicate corresponding parts throughout the several figures of the drawings. It is to be understood that the drawings are for illustrating the concepts set forth in the present disclosure and are not to scale.

30 Before any embodiments of the invention are explained in detail, it is to be understood that the invention is not limited in its application to the details

of construction and the arrangement of components set forth in the following description or illustrated in the drawings.

DETAILED DESCRIPTION

5 The following detailed description illustrates the invention by way of example and not by way of limitation. The description enables one skilled in the art to make and use the present disclosure, and describes several embodiments, adaptations, variations, alternatives, and uses of the present disclosure, including what is presently believed to be the best mode of carrying out the present disclosure.

10 Turning to the figures, and to Figures 1 and 2 in particular, a preferred embodiment of the pinion and bearing assembly of the present disclosure is shown at (A1), comprising a pinion gear (10), a flange hub (20), and a pinion gear support bracket (30). The pinion gear (10) revolves around an axis of rotation (Ax) and is supported on a first support bearing (B1) and a second
15 support bearing (B2). As best seen in Figure 3, the pinion gear (10) includes a set of bevel teeth (11), a cylindrical wall (12) defining a hollow pinion shaft extending axially from the nose of the pinion gear, a tapered inner raceway (14) associated with the first support bearing is concentrically disposed within the pinion gear (10), and a cylindrical outer raceway (13) associated with the
20 second support bearing is formed on the outer circumference of the hollow pinion shaft. As best seen in Figures 2 and 3, the taper of the inner raceway (14) converges to an apex point (AP) on the rotational axis (Ax), axially forward of the pinion gear nose end. The cylindrical outer raceway (13) further includes a rib face (16) axially adjacent the pinion gear. The inner surface
25 (15) of the cylindrical wall (12) defining the pinion shaft may optionally be splined to mate with a drive shaft (not shown). As best seen in Figure 3, the bevel teeth (11), the cylindrical wall (12) defining the pinion shaft, and the rib face (16) may be optionally machined as a unitary body from a single piece of material.

30 Within the pinion and bearing assembly (A1), the pinion gear (10) is mounted concentrically on the axis (Ax) about a portion of the flange hub (20).

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The flange hub (20), best seen in Figure 4, preferably includes an axially disposed tapered boss (21) about which the pinion gear (10) is mounted, a cylindrical base (22), a mounting flange (23), and a axially rearward extending cylindrical boss (24) having an enlarged axial bore (26). The tapered boss
5 (21) includes an axial cylindrical bore (25) which is contiguous with the enlarged axial bore (26), and has a tapered outer circumferential surface (27) which defines a tapered outer raceway associated with the first support bearing (B1), converging to the apex point (AP) on the central axis (Ax). The tapered raceway (27) incorporates a rib face (28) at its large end, adjacent to
10 the cylindrical base (22). The flange hub (20) further includes a set of mounting holes (28) on the flange (23) for attachment to a supporting structure or housing (not shown), and for securing the support bracket (30).

As seen in Figure 5, the support bracket (30) consists of an annular base flange (31) of a first diameter, a support ring (32) of a second and
15 smaller diameter, and a bridge (33) coupling the base flange and the support ring in an axially spaced configuration along the axis (Ax). The bridge (33) consists of a conical surface segment tapering axially inward from the base flange (31) to the support ring (32), and preferably includes a reinforcing rib (34) at each lateral edge to enhance the supporting stiffness of the support
20 ring (32). The base flange (31) has an inner axial opening (36) sized to seat concentrically about the outer circumference of the cylindrical base (22) of the flange hub, and a set of mounting holes (37) aligned with the mounting holes (28) on the flange (23) of the flange hub (20). The inner ring (32) similarly has an axial bore (35), which is sized to receive an outer race ring (40), as shown
25 in Figure 6.

The outer race ring (40) contains a cylindrical inner raceway (41) associated with the second support bearing (B2), an outer rib (42) adjacent a
axially rearward surface configured to engage the inner ring (32) at a matching annular recess, and an inner rib (43) adjacent a forward surface and
30 having a axially rearwardly directed rib face (44).

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To support the pinion gear (10), the first support bearing (B1) includes a set of tapered rollers (50) which are seated and evenly spaced by a first cage (not shown) between the tapered inner raceway (14) in the pinion gear (10), and the tapered outer raceway (27) on the tapered boss of the flange hub (20). The second support bearing (B2) includes a set of cylindrical rollers (60) which are seated and evenly spaced by a second cage (not shown) between the cylindrical inner raceway (41) of the outer race ring (40), and the cylindrical outer raceway (13) on the pinion gear (10). The axial positions of the tapered rollers (50) and the cylindrical rollers (60) are established by the tapered raceways (14) and (27), as well as by the rib faces (16, 28 and 44). Optionally one or more shims (not shown) may be disposed between the mounting flange (23) of the flange hub (20) and the base flange (31) of the support bracket (30) to set the tapered roller bearing formed by the tapered inner raceway (14) of the pinion (10), the tapered outer raceway (27) of the flange hub (20), and the tapered rollers (50). As assembled, the tapered surfaces of raceways (14, 27) and rollers (50) all converge to an apex at point (AP) on the axis (Ax) as shown in Figure 2.

Once assembled, the pinion gear (10) is rotationally fixed to a concentric pinion drive shaft (not shown) extending axially through the flange hub (20) by a spline or any mechanical means seated within the cylindrical wall (12) of the pinion support shaft. For example, the forward end of pinion drive shaft can be keyed to rotationally fix the pinion (10) to the pinion drive shaft. The pinion (10) can also be secured by a laser weld or other suitable fastening means to the pinion drive shaft.

Those of ordinary skill in the art will recognize that when comparing the pinion and bearing assembly (A1) of the present disclosure with a convention pinion gear having an overhung arrangement (not shown), the main support bearings (50) in the present disclosure receives less load, and thus can be down-sized. Reductions in bearing load and size result in reductions in bearing torque, and thus, improved system efficiency.

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Turning to Figure 7, a second embodiment (A2) of the pinion and bearing assembly of the present disclosure integrates an over-running clutch or roller clutch (B) with a pinion drive shaft (71). The clutch (B) is supported coaxially within the enlarged axial bore (26) of the rearward extending cylindrical boss (24) on the flange hub by a first rolling element bearing (100) and a second rolling element bearing (110). The clutch (B) has an outer drive hub (90) coupled to a drive source (not shown), a coaxial inner driven cam assembly (70) coupled to the pinion drive shaft (71), and a set of cylindrical clutch rollers (80) disposed between an inner circumferential surface (92) of the outer drive hub (90), and the outer surface of the inner driven cam assembly (70). The drive hub (90) has a mounting flange (91) for coupling to the drive source. The inner drive cam assembly (70) consists of the pinion drive shaft (71) extending axially through the flange hub to the pinion gear (10), and a concentric drive cam (72). As best seen in Figure 8, the outer circumferential surface of the concentric drive cam (72) is scalloped to form a set of equidistantly disposed wedged spaces (120) relative to the cylindrical inner surface (92) of the drive hub (90). The clutch rollers (80) are arranged annually between the concentric drive cam (72) and the drive hub (90) with each and every roller (80) occupying an associated wedge space (120).

During operation, the roller clutch (B) allows torque and power to be transmitted from drive hub (90) to the concentric drive cam (72) only in one rotational direction, in which the rollers (80) are pushed into the narrow portions of the wedged spaces (120), effectively locking the drive hub (90) and the concentric drive cam (72) together, ensuring that the pinion drive shaft (71) rotates with the drive hub (90). In this rotational direction the concentric drive cam (72) is allowed to over-run the drive hub (90). During rotation in the second rotational direction, the rollers (80) are pushed into the wide portions of the wedged spaces (120), and act as cylindrical roller bearings preventing torque and power from being transmitted from the drive hub (90) to the concentric drive cam (72), or vice versa.

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It will be understood by those of ordinary skill in the art that the roller clutch (B) could be configured with the wedged spaces (120) formed by scallops on the inner circumferential surface (91) of the drive hub (90) and a outer circumferential surface of the pinion drive shaft (71), without the need
5 for the concentric drive cam (72).

As various changes could be made in the above constructions without departing from the scope of the disclosure, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense. For example,
10 those of ordinary skill in the art will recognize that the first and second support bearings (B1 and B2) shown herein and described as employing tapered cylindrical rollers and cylindrical rollers, respectively, may be reconfigured to utilize other types of rolling elements, such as ball bearings, cylindrical bearings, or roller bearings, while still achieving the axially compact
15 configuration as shown and described.

CLAIMS:

1. A compact pinion and bearing assembly (A1, A2), comprising:
a pinion gear (10), said pinion gear having a set of gear teeth (11) and
an axial pinion support shaft defined by a cylindrical wall (12) extending
5 axially forward of said gear teeth;
a first support bearing (B1) disposed within an axial bore of said pinion
gear, concentric with said gear teeth (11); and
a second support bearing (B2) disposed about an outer circumference
of said cylindrical wall (12).
- 10 2. The compact pinion and bearing assembly of Claim 1 wherein
said pinion gear is a bevel pinion gear.
3. The compact pinion and bearing assembly of Claim 1 wherein
said pinion gear (10) and said cylindrical wall (12) are integrally formed.
4. The compact pinion and bearing assembly of Claim 1 wherein
15 said first support bearing (B1) is a tapered bearing having a plurality of
tapered rollers (50), and wherein said second support bearing (B2) is a
cylindrical bearing having a plurality of cylindrical rollers (60).
5. The compact pinion and bearing assembly of Claim 1 further
including a flange hub (20) having a boss (21) extending axially from a
20 cylindrical base (22) into said axial bore of said pinion gear (10); and
wherein said first support bearing (B1) is disposed between an outer
surface (27) of said boss (21) and an inner surface (14) of said bore.
6. The compact pinion and bearing assembly of Claim 5 wherein
said inner surface (14) of said bore of said pinion gear (10) defines an outer
25 raceway of said first support bearing (B1), and wherein said outer surface
(27) of said boss (21) on said flange hub (20) defines an inner raceway of
said first support bearing.
7. The compact pinion and bearing assembly of Claim 6 wherein
said inner raceway and said outer raceways are tapered to converge at a
30 point (AP) on an axis (Ax) of rotation for said pinion gear (10).

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8. The compact pinion and bearing assembly of Claim 7 wherein said plurality of tapered rollers (50) is disposed between said inner and outer raceways.

9. The compact pinion and bearing assembly of Claim 5 wherein
5 said flange hub (20) further includes an annular flange (23) disposed axially rearward of said cylindrical base (22), said annular flange hub configured to receive an annular base flange (31) of a support bracket (30);

wherein said support bracket (30) further includes a support ring (32) spaced axially forward from, and coaxial with, said annular base flange (31),
10 said support ring (32) secured to said annular base flange (31) by a bridge segment (33); and

wherein said second support bearing (B2) is disposed between an inner surface of an axial bore (35) in said support ring and said outer circumferential surface (13) of said cylindrical wall (13) on said pinion gear
15 (10).

10. The compact pinion and bearing assembly of Claim 9 further including an outer race ring (40) secured concentrically within said support ring axial bore (35), said outer race ring having an inner circumferential surface (41) defining an outer raceway of said second support bearing (B2);
20 and

wherein the outer circumferential surface (13) of said cylindrical wall (12) defines an inner race of said second support bearing (B2).

11. The compact pinion and bearing assembly of Claim 10 wherein said second support bearing (B2) further includes a plurality of rolling
25 elements (60) disposed between said inner and outer raceways.

12. The compact pinion and bearing assembly of Claim 1 further including a pinion drive shaft (71) extending axially through said pinion gear (10), said first support bearing (B1), and said second support bearing (B2), said pinion drive shaft rotationally secured to said cylindrical wall (12) of said
30 pinion support shaft.

13. The compact pinion and bearing assembly of Claim 12 wherein said pinion drive shaft (71) is coupled to a rotational drive source by a clutch assembly (B).

14. The compact pinion and bearing assembly of Claim 13 wherein
5 said clutch assembly (B) is an over-running or roller clutch consisting of a plurality of cylindrical rollers (80) disposed in a set of wedged spaces (120) between an outer surface of a concentric drive cam (72) coupled to an axial end of said pinion drive shaft, and a cylindrical inner surface (91) of a drive hub (90);

10 wherein rotational movement of the concentric drive cam (72) relative to the drive hub (90) in a first rotational direction forces said plurality of rollers (80) into a narrow portion of said wedged spaces (120), rotationally locking said drive hub (90) and concentric drive cam (72) together for a transmission of torque and power; and

15 wherein rotational movement of the concentric drive cam (72) relative to the drive hub (90) in a second and opposite rotational direction forces said plurality of rollers (80) into a wide portion of said wedged spaces (120), rotationally releasing said drive hub from said concentric drive cam.

15. The compact pinion and bearing assembly of Claim 13 wherein
20 said clutch assembly (B) is an over-running or roller clutch consisting of a plurality of cylindrical rollers (80) disposed in a set of wedged spaces (120) between an inner cam surface (91) of a drive hub (90), and a cylindrical outer surface at an axial end of said pinion drive shaft (71);

25 wherein rotational movement of the inner cam surface relative to the cylindrical outer surface of the pinion drive shaft (71) in a first rotational direction forces said plurality of rollers (80) into a narrow portion of said wedged spaces (120), rotationally locking said drive hub (90) and pinion drive shaft together for a transmission of torque and power; and

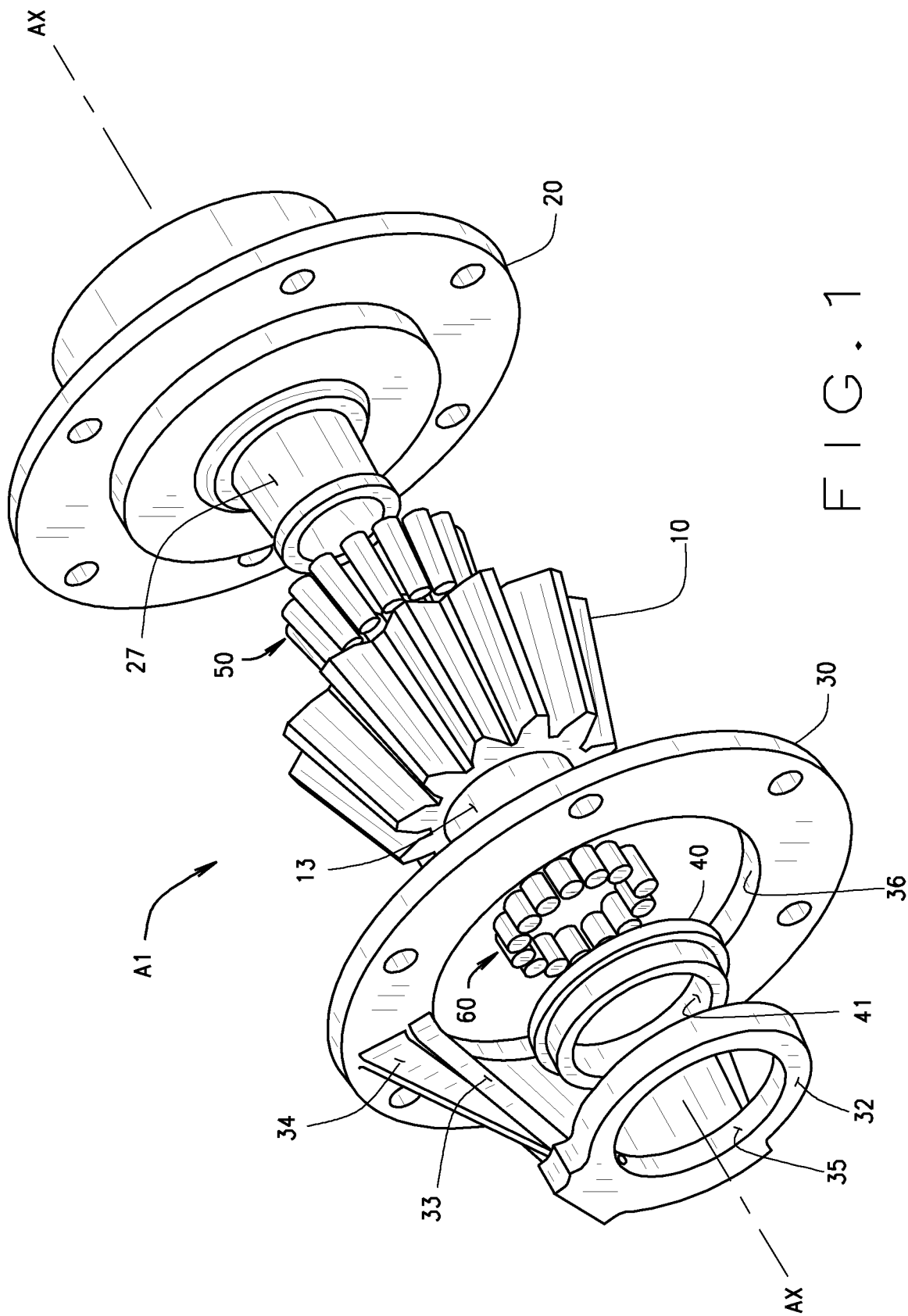
30 wherein rotational movement of the inner cam surface relative to the cylindrical outer surface of the pinion drive shaft (71) in a second and opposite rotational direction forces said plurality of rollers (80) into a wide

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portion of said wedged spaces (120), rotationally releasing said drive hub (90) from said pinion drive shaft (71).

16. The compact pinion and bearing assembly of Claim 1 wherein said first support bearing (B1) is a ball bearing.

5 17. The compact pinion and bearing assembly of Claim 1 wherein said second support bearing (B2) is a ball bearing.



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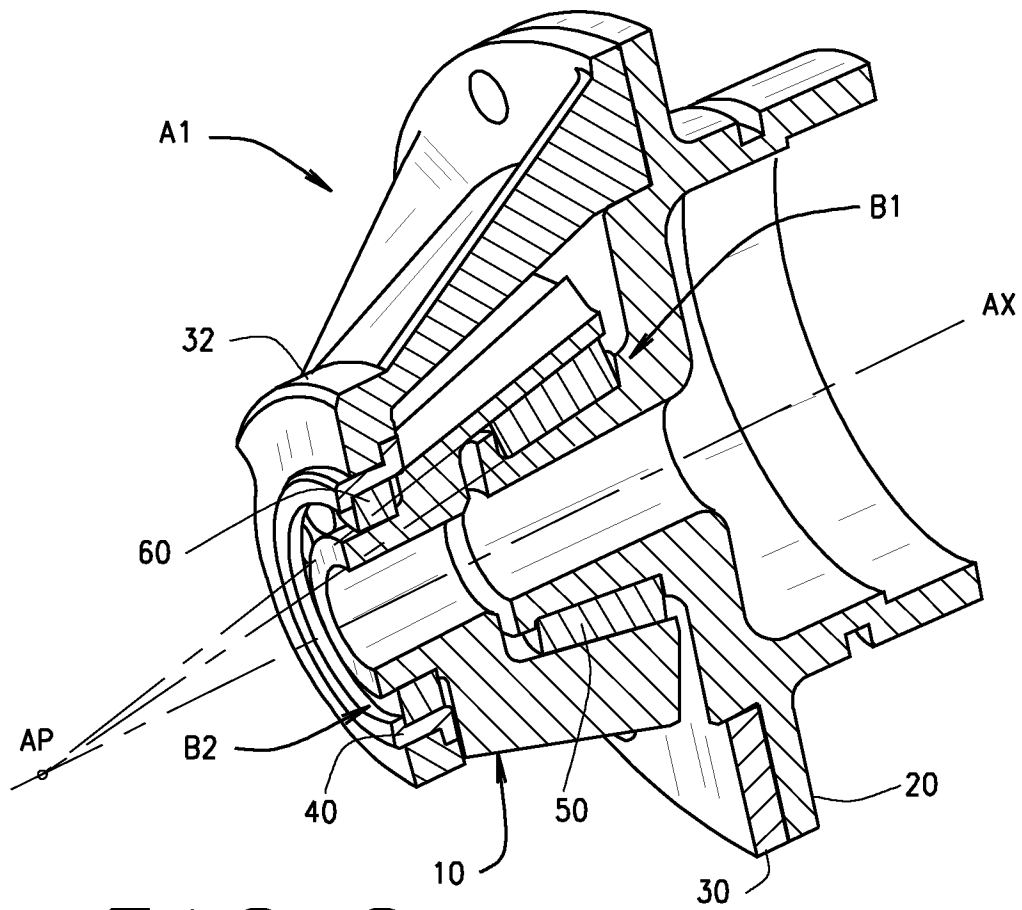


FIG. 2

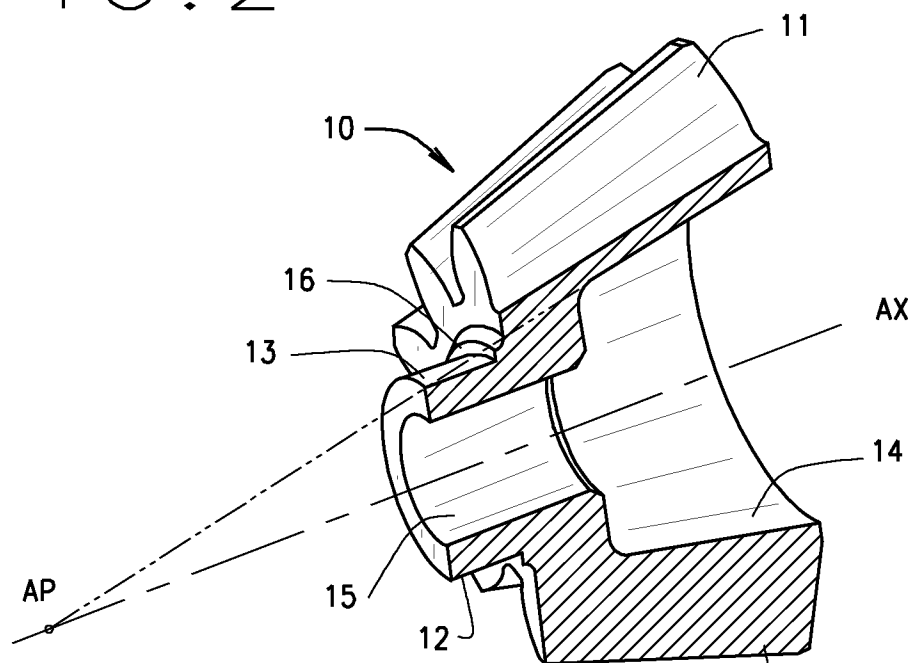


FIG. 3

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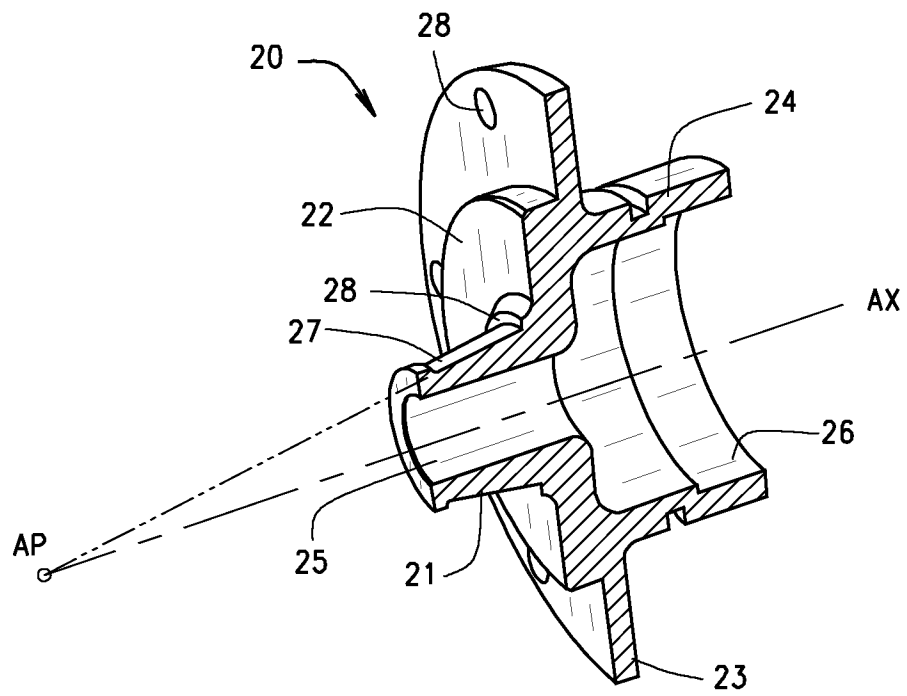


FIG. 4

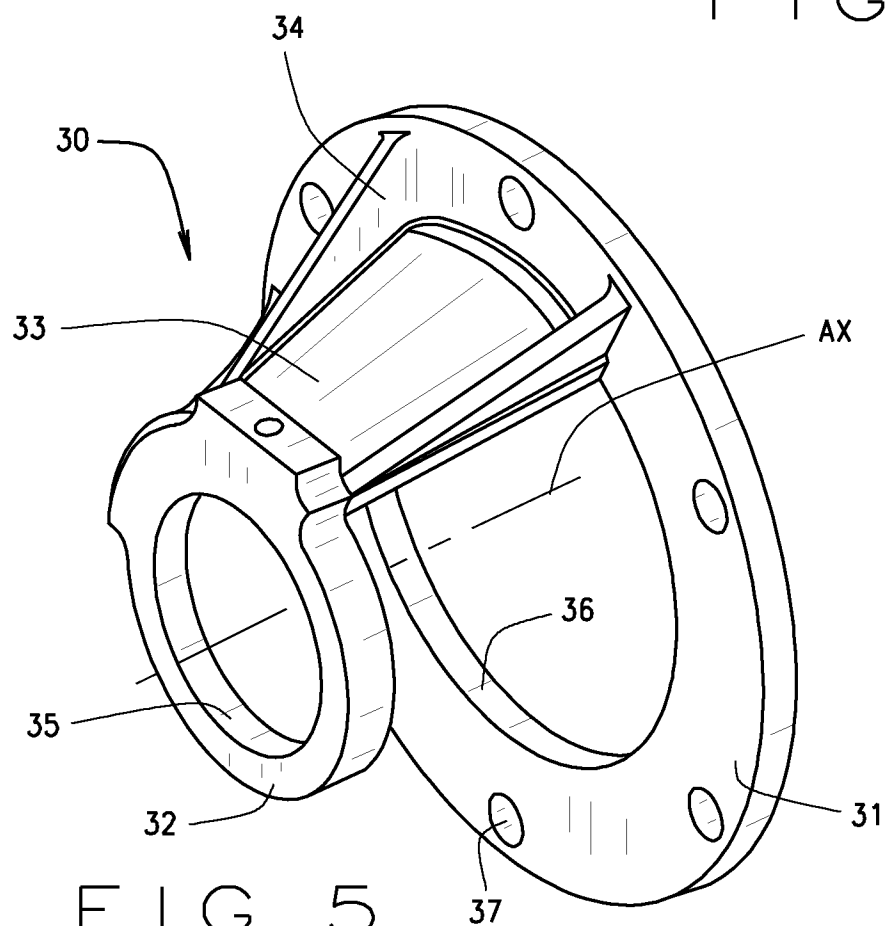


FIG. 5

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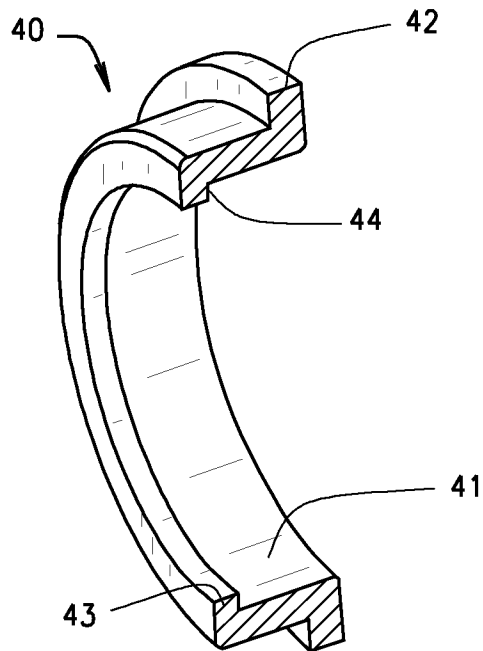


FIG. 6

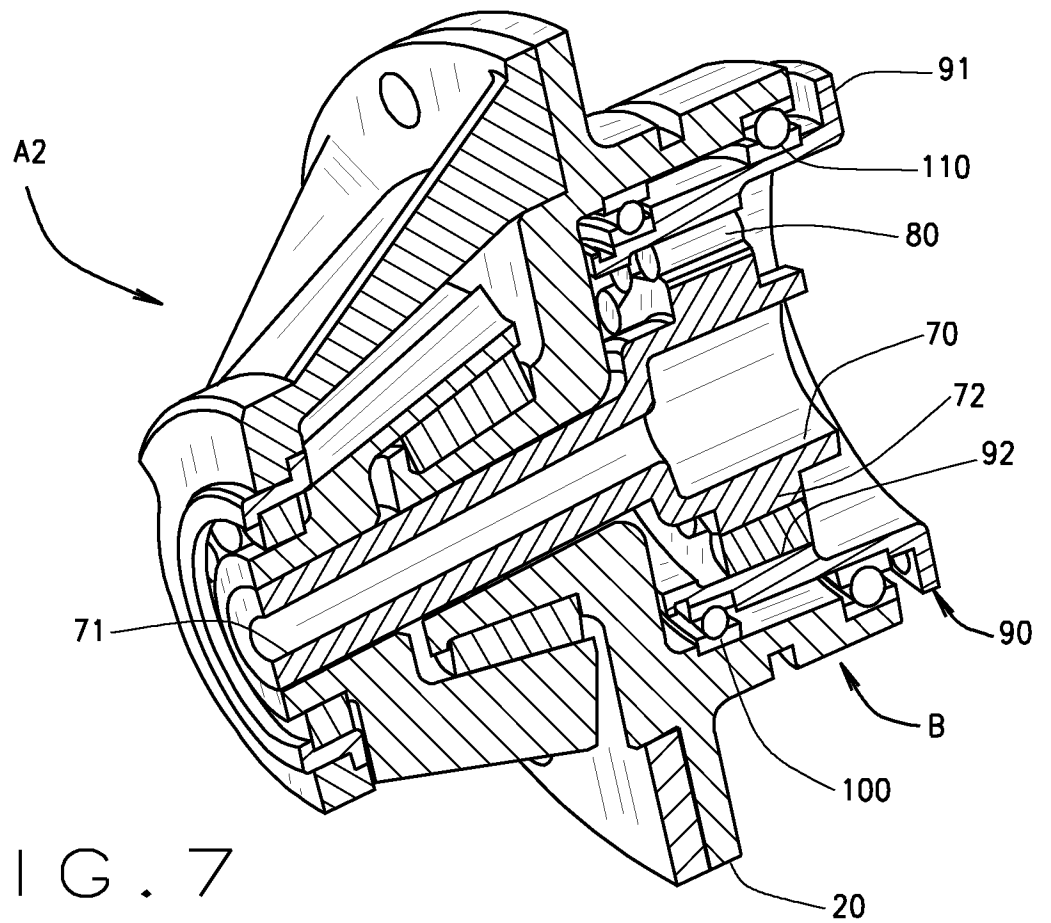


FIG. 7

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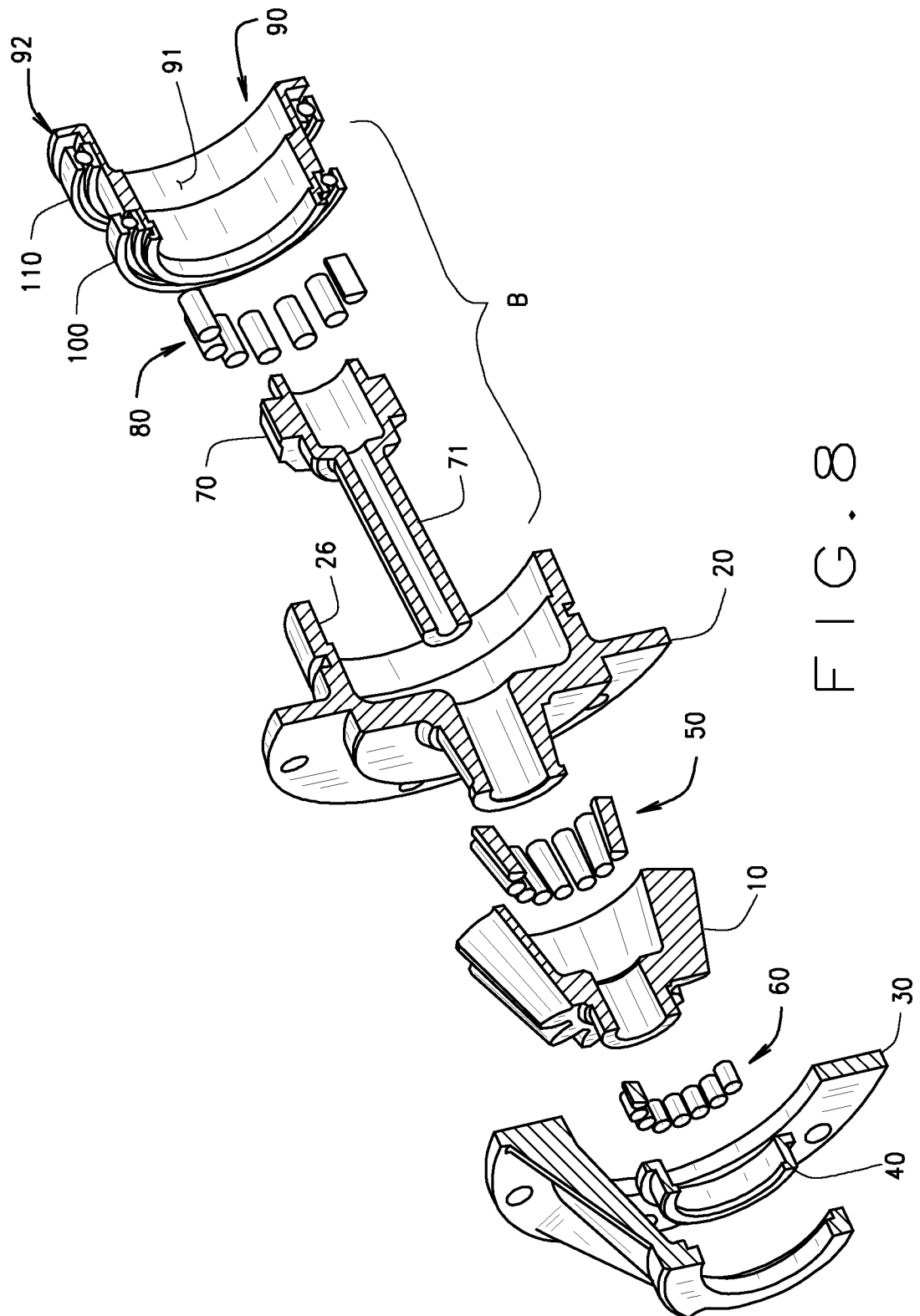


FIG. 8

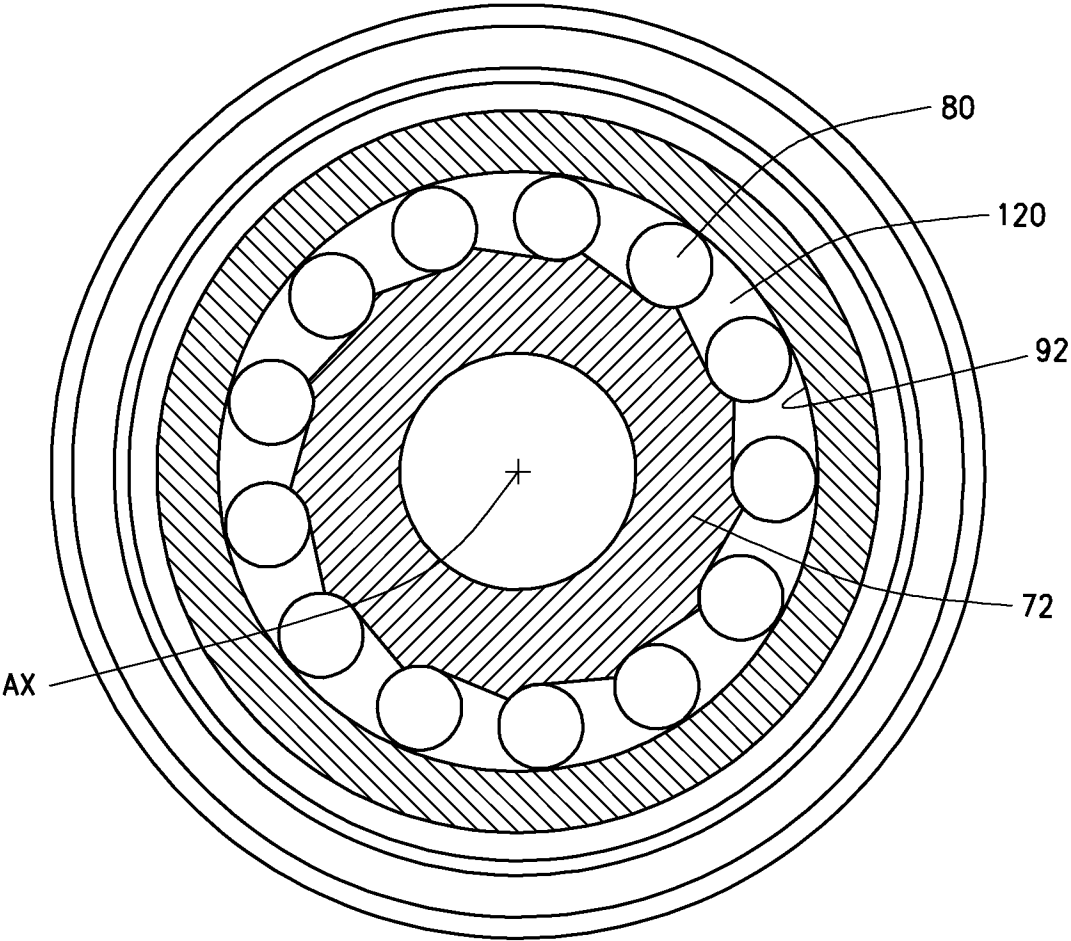


FIG. 9

INTERNATIONAL SEARCH REPORT

International application No

PCT/US2011/028819

A. CLASSIFICATION OF SUBJECT MATTER
 INV. F16H57/02
 ADD.

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
 F16H F16C B60K

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X A	DE 33 46 373 A1 (KIENZLE APPARATE GMBH [DE]) 4 July 1985 (1985-07-04) the whole document	1,3,5,6 2,4,7,9, 12,16,17
X A	----- US 2004/176206 A1 (OATES JACK DARRIN [US] ET AL) 9 September 2004 (2004-09-09) paragraphs [0026] - [0028], [0030] - [0032] figure 3	1-3 4,5,12, 16,17
X A	----- JP 2003 028177 A (TOYOTA MOTOR CORP) 29 January 2003 (2003-01-29) figures 1,2 ----- -/--	1,3,4, 16,17 2,5,12



Further documents are listed in the continuation of Box C.



See patent family annex.

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Date of the actual completion of the international search

27 April 2011

Date of mailing of the international search report

04/05/2011

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INTERNATIONAL SEARCH REPORT

International application No

PCT/US2011/028819

C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 3 330 166 A (ROCKWELL LEONARD C ET AL) 11 July 1967 (1967-07-11)	1,16,17
A	column 3, line 36 - column 4, line 5 figure 1 -----	2-5,12
X	US 2003/029041 A1 (BENDTSEN RANDALL REY [US]) 13 February 2003 (2003-02-13)	1-11,16, 17
A	paragraphs [0019] - [0023] figures 1,2 -----	12
A	US 6 024 666 A (BUNNOW MARCUS [US]) 15 February 2000 (2000-02-15) the whole document -----	1
A	DE 10 2008 000956 A1 (BOSCH GMBH ROBERT [DE]) 8 October 2009 (2009-10-08) paragraphs [0020], [0021]; figure 2 -----	1

INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No

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