Feb. 24, 1970 O. ECKERLE ET AL 3,496,877

INTERNAL GEAR HYDRAULIC PUMP OR MOTOR

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Fig.1

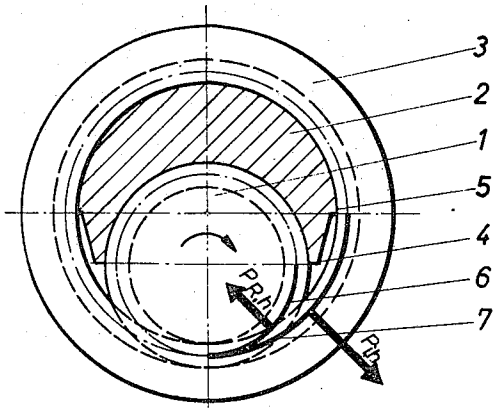
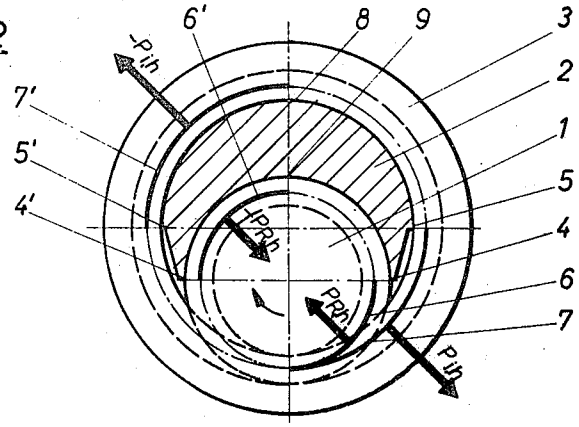


Fig.2



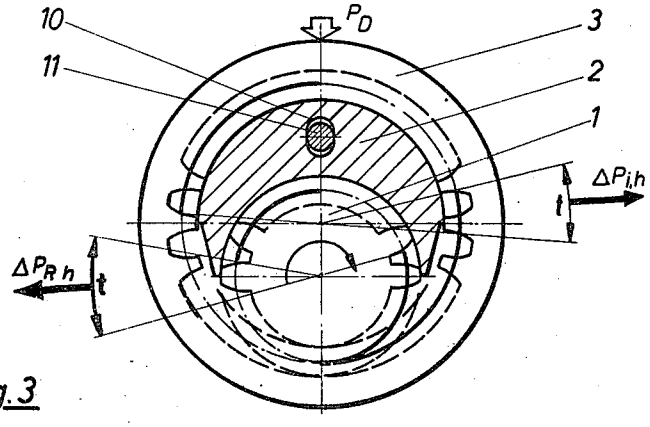


Fig.3

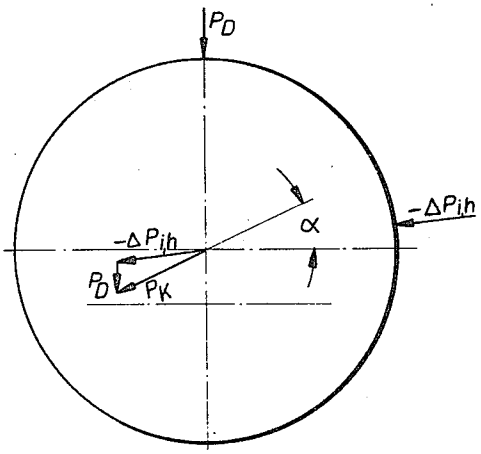
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INTERNAL GEAR HYDRAULIC PUMP OR MOTOR

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Fig.4



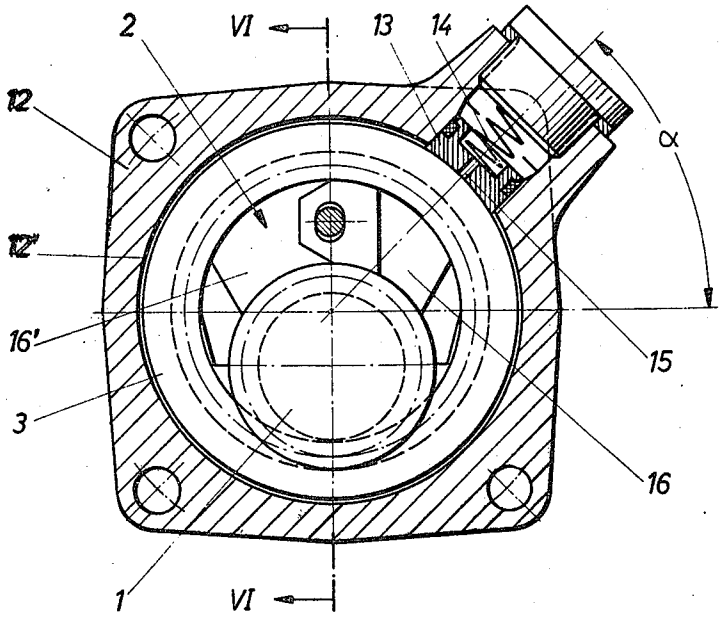


Fig.5

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Fig.6

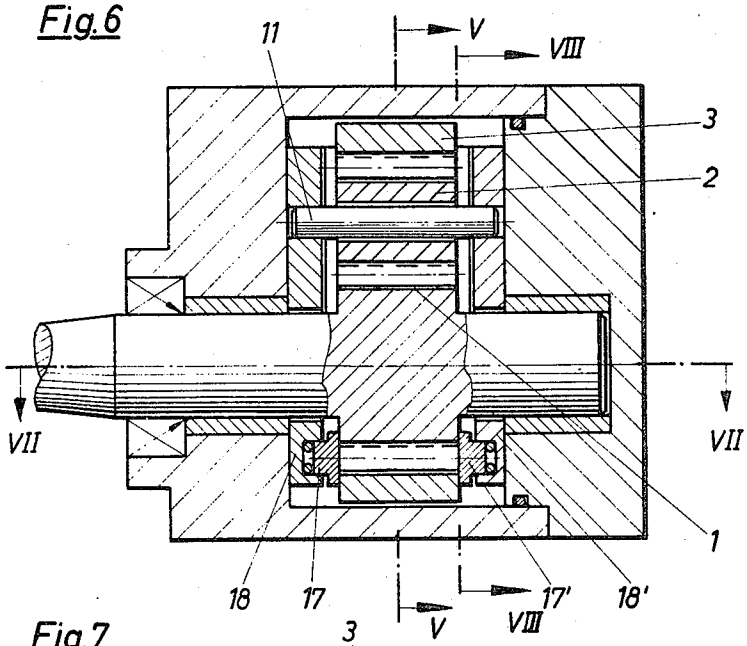
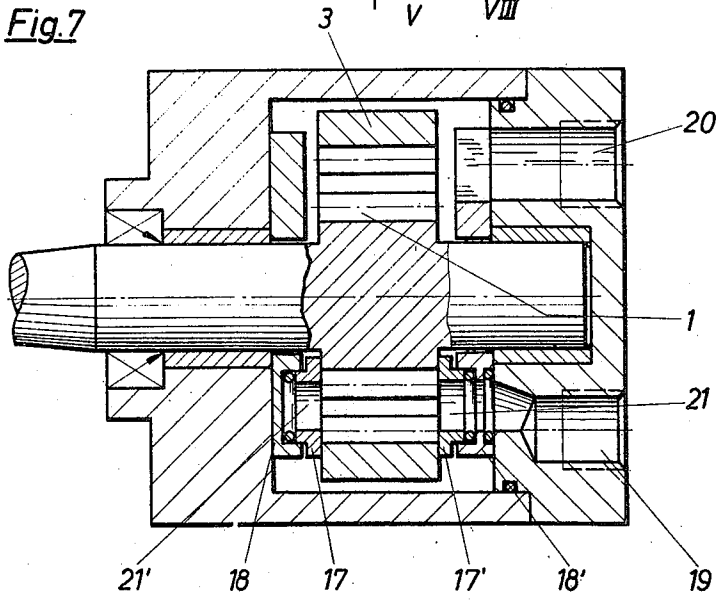


Fig.7

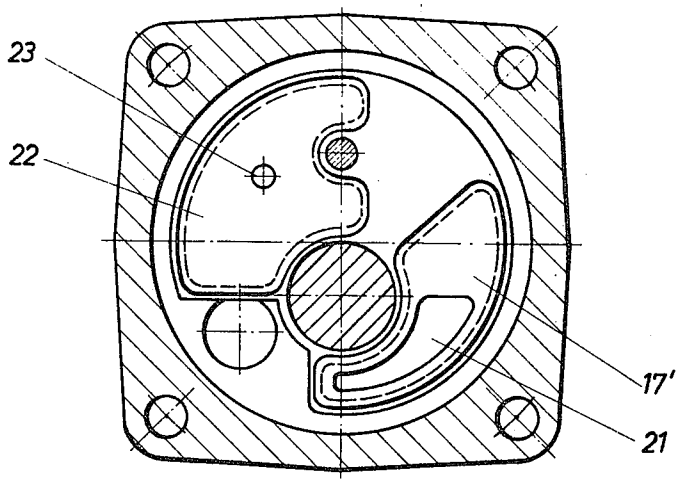


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Fig.8



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Fig.9

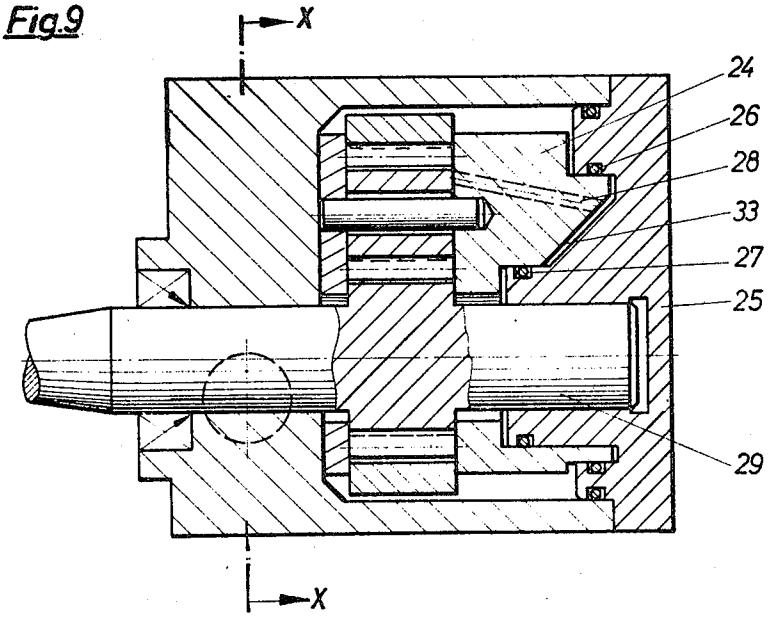
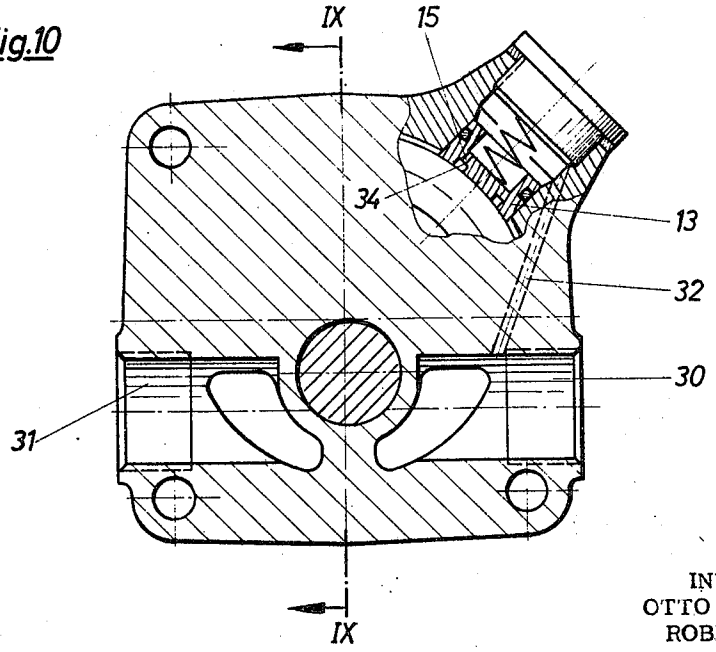


Fig.10



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INTERNAL GEAR HYDRAULIC PUMP OR MOTOR

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Int. Cl. F04c 1/06
U.S. Cl. 103—126 10 Claims

ABSTRACT OF THE DISCLOSURE

A high pressure internal gear hydraulic device of the kind which can be used as a pump or as a motor, in which the externally-toothed inner gear or pinion is driven, the internally toothed outer gear or ring gear is an idler, and a crescent-shaped filling piece or crescent seal is disposed between the two gears.

BACKGROUND OF THE INVENTION

The present invention relates to the problem of compensating for wear, internal forces, and gap leakage losses in internal gear hydraulic pumps or motors. It is known that gap leakage losses are caused by the fact that the crescent or filling piece and the inner gear or pinion are fixedly mounted, and the internally-toothed ring gear or outer gear is mounted with its entire outer circumference closely fitting in a bore in the casing of the device, so that gaps may be formed between the tooth tip circles of the gears and the crescent through which liquid may flow back from the pressure chamber into the suction chamber of the device.

In order to compensate for wear, internal forces, and gap leakage losses, a floating ring gear operating in conjunction with a movable crescent has already been proposed. In this arrangement the ring gear is maintained in close contact with both the crescent and the pinion by means of a control piston. In this manner freedom from flank clearance between the two gears is achieved, although this is possible only if the teeth are produced with the greatest accuracy.

The problem underlying the present invention is to provide an internal gear hydraulic device which has the same properties but in which the control piston is eliminated, so that the gears may be run with flank clearance and thus the teeth need not be formed with great accuracy.

The present invention, then, consists in a high pressure internal gear hydraulic device which has a driven pinion, a ring gear meshing therewith, and a crescent disposed between the two gears, wherein the ring gear is supported only by the crescent, and wherein pressure zones for balancing the forces exerted by the fluid in the pressure chamber on the ring gear and the pinion are created in recesses connected by suitable passages to the pressure chamber of the device, said recesses being disposed on the opposite side of the device from the pressure chamber. Such recesses may be situated between the ring gear and the crescent, and between the crescent and the pinion, respectively.

The invention, accordingly, comprises the features of construction, combinations of elements, and arrangements of parts, which will be exemplified in the constructions hereinafter set forth, and the scope of the invention will be indicated in the claims.

BRIEF DESCRIPTION OF THE DRAWINGS

For a fuller understanding of the nature and objects of the invention, reference should be had to the following detailed description taken in connection with the accompanying drawings, in which:

FIGURES 1 through 4 are a schematic illustration of the principal parts of a device according to the invention and the forces acting thereon;

FIGURE 5 is a sectional view of a device according to the invention taken along line V—V of FIGURE 6;

FIGURE 6 is a sectional view of an embodiment of the present invention taken along line VI—VI of FIGURE 5;

FIGURE 7 is a sectional view of an embodiment of the present invention taken along line VII—VII of FIGURE 6;

FIGURE 8 is a sectional view of an embodiment of the present invention taken along line VIII—VIII of FIGURE 6;

FIGURE 9 is a sectional view of the second embodiment of the present invention taken along line IX—IX of FIGURE 10; and

FIGURE 10 is a sectional view taken along line X—X of FIGURE 9.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to the drawings, FIGURES 1 through 3 illustrate the relative positions of the pinion **1**, the ring gear **3**, and the crescent **2**. When pinion **1** rotates in the direction of the arrow pressure is built up in the pressure chamber and creates a pressure zone **6** on pinion **1** and a larger pressure zone **7** on ring gear **3**. These pressure zones extend from the bottom of the gears where their teeth mesh to the edges **4** and **5** of crescent **2**. Pressure zone **6** applies a resultant force $P_{R,h}$ to pinion **1**, this force being taken by the pinion bearings in conventional internal gear pumps. Pressure zone **7** applies a resultant force $P_{i,h}$ to ring gear **3**, which is supported only by crescent **2**. Force $P_{i,h}$ tends to force ring gear **3** away from pinion **1**.

In order to compensate for these two resultant forces, a counter-pressure zone **7'**, as illustrated in FIGURE 2, is created between ring gear **3** and crescent **2**. In addition, a counter-pressure zone **6'** is created between crescent **2** and pinion **1**. These counter-pressure zones may, by way of example, be formed by means of recesses in the crescent or in the gears into which fluid under pressure is introduced through channels in the casing of the device. The counter-pressure zones **6'** and **7'** are so selected in respect of their size and position that their resultant forces $-P_{i,h}$ and $-P_{R,h}$ cancel the forces $P_{i,h}$ and $P_{R,h}$.

In order to avoid radial gap leakage losses it is necessary that the teeth of pinion **1** should lie tightly against the surface and the edges **4** and **4'** of crescent **2**, and that the teeth of ring gear **3** should lie tightly against the surface **8** and the edges **5** and **5'** of crescent **2**. In order to fulfill this condition, as illustrated in FIGURE 3, a force P_D is applied to ring gear **3** and crescent **2** is so mounted as to be slidable in the direction of this force. For this purpose the crescent is provided with a slot **10**, through which there passes a pin **11** which is in turn fastened in the casing of the device. In order to produce the force P_D use may be made, for example, of a spring, or a piston acted on by fluid from the pressure chamber of the device.

When the force P_D is used to compensate for gap leakage, complete compensation for the forces $P_{R,h}$ and $P_{i,h}$ is no longer possible, since the tips of crescent **2** would have to be extended farther downwards than is illustrated in FIGURE 3. The consequence of such added extension of these tips would be that the edges **4** and **4'** of crescent **2** would be lifted off the teeth of pinion **1** and the edges **5** and **5'** would be pressed too firmly against the teeth of ring gear **3**. It is therefore necessary

that the circular arcs between edges 4 and 4' and 5 and 5' respectively should not exceed an angle of 180°. This condition of incomplete compensation causes a residual force $\Delta P_{i,h}$ to act on ring gear 3, and a residual force $\Delta P_{R,h}$ to act on pinion 1. The last-mentioned force can be ignored, because it is taken by the pinion bearing.

The two residual forces are proportional to the tooth pitch t, that is to say, the smaller the tooth pitch the smaller the residual force.

The residual force $\Delta P_{i,h}$ acting on ring gear 3 must be compensated. This compensation can be effected with the aid of a spring or with the aid of a piston acted on by the pressure medium. Since as illustrated in FIGURE 4, the force P_D together with a compensating force $-\Delta P_{i,h}$ gives a resultant force P_K which forms an angle α with the horizontal, a spring or a piston exerting a force corresponding to the force P_K will be sufficient to apply the force P_D and the compensating force $-\Delta P_{i,h}$.

A gear pump fitted with a radial piston of this type is illustrated in FIGURES 5 to 8. Ring gear 3, which is supported only on crescent 2, and which meshes with pinion 1, is mounted in a bore 12' which is provided in casing 12 and has an inside diameter slightly larger than the outside diameter of ring gear 3. A radial piston 13 which presses against ring gear 3 is disposed in casing 12 at an angle α to the horizontal. The rear side of piston 13 is in communication with the pressure chamber of the pump through a passage (not shown). A spring 14 provides the necessary pressing pressure during the starting of the pump. Piston 13 may be provided with a bore 15, the purpose of which is to provide a controlled gap or relief between piston 13 and ring gear 3 due to fluid under pressure continuously forcing itself there between. Since crescent 2 is movably mounted, it is advantageous for its surface sliding against the laterally adjoining pump elements to be kept small. For this reason the surfaces 16 and 16' of the crescent 2 are cut back on both sides.

The axial sealing of the pressure chamber of the pump is effected by pistons 17 and 17', which are guided in recesses in respective intermediate discs 18 and 18' and have apertures 21 and 21' to allow the pressure medium to pass to the pressure connection 19 and to load the rear of the pistons. As can be seen particularly in FIGURE 7, ring gear 3 moves freely in the suction chamber which is in communication with the suction connection 20.

FIGURE 8 shows that corresponding pistons 22, which are hydraulically loaded through bores 23, may also be provided on the suction side of the pump in order to compensate for the axial gaps in the pressure chamber which are caused by the counter-pressure zones 6' and 7'. The counter-pressure zones 6' and 7' may in this arrangement be built up from the pressure chamber through the axial pressure zone of the piston 22 and the bore 23.

The embodiment illustrated in FIGURES 9 and 10 shows another possible way of balancing the axial forces occurring in the pump. The relatively complicated pistons 17 and 22 of FIGURES 6 to 8 are replaced by a common piston 24 which is formed of cylindrical surfaces and thus can be produced by simple means. O-rings 26 and 27 bound the axial pressure zone 33, which is in communication through bores both with the pressure chamber of the pump and with the radial counter-pressure zones 6' and 7' (FIGURE 2). The feeding of the radial counter-pressure zones is thereby achieved in a simple manner. The piston 24 is guided in a bearing cover 25 in such a manner that the bearing load is not taken by the pistons 24 but by the stationary bearing cover 25 in which the pinion shaft 29 is mounted. Axial balance therefore cannot be influence by bearing load.

As FIGURE 10 shows, the pressure connection 30 and the suction connection 31 are accommodated in the pump casing. This affords the advantage that the gearing parts of the pump can be dismantled without dismantling the pipes and the casing. Radial piston 13 can be fed through an inclined bore 32 leading to the pressure connection.

The prevously mentioned bore 15 in the radial piston 13 leads into an annular relief groove 34.

The area of the axial pressure zone 33 of the piston 24 is slightly larger than the inner pressure zone acting in the axial direction, and the center of area of the axial pressure zone 33 corresponds to the common center of area of the pistons 17' and 22.

What is claimed is:

1. An internal gear hydraulic device, comprising: an externally-toothed pinion; means for driving said pinion; an internally-toothed ring gear meshing with said pinion; crescent-shaped sealing means disposed between said pinion and said ring gear, said ring gear being supported only by said crescent-shaped sealing means; axial sealing means to abut sealingly against the pinion, ring gear, and crescent-shaped sealing means, the axial sealing means, pinion, ring gear, and crescent-shaped sealing means forming together a pressure chamber; and pressure zone creating means to create pressure zones (6', 7') connected to the pressure chamber and disposed opposite to the pressure chamber, between said ring gear and said sealing means and between said sealing means and said pinion, respectively, for at least partially balancing the forces exerted by fluid in the pressure chamber on said ring gear and said pinion.

2. A device as claimed in claim 1 including means for applying to said ring gear a force (P_D) directed along a line which passes through the center of said pinion and through the center of said crescent-shaped sealing means, said crescent-shaped sealing means being mounted for sliding movement in the direction of said force.

3. A device as claimed in claim 1 wherein the surfaces of said crescent-shaped sealing means which coact with said pinion and said ring gear to form seals therewith extend around said pinion and said ring gear by no more than 180°.

4. A device as claimed in claim 3, pressurized fluid in the pressure chamber creating a residual force ($\Delta P_{i,h}$) acting on said ring gear, wherein means is provided which acts on said ring gear oppositely to the direction of said residual force.

5. A device as claimed in claim 4 in which said means for applying a force (P_D) and said means which acts on said ring gear are combined into a force producing means disposed in the device so as to act in the direction of the resultant (P_K) of said forces P_D and $\Delta P_{i,h}$.

6. A device as claimed in claim 5 in which said means disposed to act in the direction of resultant force (P_K) includes a spring biased piston means operable by fluid under pressure, said piston having a bore by means of which pressurized operating fluid may reach the space between said piston and said ring gear.

7. A device as claimed in claim 1 in which the axial sealing means comprise axially slidable, hydraulically actuatable piston means.

8. A device as claimed in claim 7, further comprising pressure connection (19) means to deliver pressurized fluid, said hydraulically actuatable piston means provided with apertures, one of said apertures providing communication between said pressure chamber and the pressure connection means.

9. A device as claimed in claim 1, in which, for the compensation of axial gap leakage and to form the axial sealing means, piston means (24) is provided, further comprising a bearing cover means, a shaft (29) carrying the pinion, the shaft (29) bearing on the bearing cover means, the piston means (24) being mounted in the bearing cover means on two cylindrical surfaces and slidable into sealing abutment on the pinion, ring gear, and crescent-shaped sealing means, a pressure zone (33) between the piston means (24) and the bearing cover means, whereby the piston means (24) may be urged into said abutment by pressurized fluid, the piston (24) being free of the bearing load of the shaft (29).

10. A device as claimed in claim 9, in which the area

of said pressure zone (33) acting on said piston means (24) is greater than the corresponding area of the internal pressure zone between pinion, ring gear, and crescent-shaped sealing means acting in the axial direction.

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