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(54) **Gear pump**

Zahnradförderpumpe

Pompe à engrenages

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Description

[0001] The present invention relates to the sector of positive-displacement rotary pumps. There are known various types of rotary pump, including gear pumps, lobe pumps and screw pumps.

[0002] Gear pumps generally comprise two gears, one of which, known as the driving gear, is connected to a drive shaft and causes the other wheel, known as the driven gear, to rotate. The pumps of this type for high pressures are generally produced with a so-called "balanced" or "equilibrated" configuration, in which the two opposing faces of the bushings for supporting the gears are subjected to pressures over areas which, although they are large in absolute terms, are not very different from each other in order to generate a moderate differential force which tends to keep each bushing in contact with the gears.

[0003] Figures 1 and 2 illustrate an example of a gear pump of known type and comprising the features of the preamble of claim 1. In particular, Figure 1 is a longitudinal section along a plane which extends through the axes of rotation of the two gears, and Figure 2 is a section taken on line II-II in Figure 1. The driving gear 13 and the driven gear 14, whose shafts are supported by two bushings 15, are housed inside a shell 10 which is closed by a front cover 11 and a rear cover 12. Omega-shaped ($\overline{\omega}$) seals 16 which separate the intake zone (A), at lower pressure, from the output zone (M), at higher pressure, are housed on the outer face 17a of the bushings 15. During use, the bushings 15 are subjected to a pressure both on the outer faces 17a thereof and on the inner faces 17b thereof. The omega-like configuration of the seals 16 is such that the portion of outer face 17a of each bushing 15 on which the output pressure, which is greater than the intake pressure, acts is greater than the portion of inner face 17b of the bushing which is subjected to the same output pressure. Since the area on which the output pressure acts in the region of the inner face 17b of each bushing cannot be determined with accuracy, the optimum configuration of the omega-like seal 16 is usually identified by trial and error.

[0004] Owing to the difference between the pressures which act on the two faces, the outer face 17a and inner face 17b, the bushings 15 are urged with a force which is moderate, and controlled, against the gears 13 and 14 so as to minimize the leakages over the faces of the gears themselves as a result of the difference in pressure between the intake and output. In the prior art, therefore, the two bushings are floating in an axial sense.

[0005] Obtaining good leak-tightness between the intake and output is one of the principal objectives in the production of gear pumps. In fact, the efficiency of pumps of this type declines rapidly if the leak-tightness is not total. Another problem which the manufacturers of pumps have to deal with is the noise of the pumps themselves, owing to irregular phenomena, or "ripples", in the transfer of the fluid. A study of the above-mentioned problems linked to the design of gear pumps is set out in "C. Bonacini, Sulla portata delle pompe ad ingranaggi (On the efficiency of gear pumps), L'ingegnere, 1961 n. 9".

[0006] The above-mentioned solutions of the prior art have the common problem consisting in the noise of operation caused by the instantaneous oscillations of the output over time, better known as ripple noise. The above-mentioned oscillations generate a pulsating wave which, by way of the fluid, is transmitted to the surroundings and, in particular, to the walls of the pump, to the pipes and to the output ducts. The noise produced can reach levels which are also unpredictable where the above-mentioned members begin to resonate with the frequency of oscillation or ripple.

[0007] In addition to this, the rotation of the gears causes a periodic variation in the area of the inner face 17b of the bushings 15 that is exposed to the output pressure. This variation determines oscillations in the axial loads on the bushings, which contributes to an increase in the noise of the pump, besides reducing the total efficiency thereof. This oscillation of the axial loads, which is normally of small magnitude in gear pumps having straight teeth, becomes significantly greater in gear pumps having helical teeth, in which the meshing between the gears is the cause of both mechanical and hydraulic axial loads such that the balance and the taking-up of clearances on the bushings illustrated in Figures 1 and 2 is not completely satisfactory, since the hydraulic axial loads have perceptible pulses.

[0008] Prior art documents GB908687, FR1197750, US3447472 and US2981200 disclose a gear pump with shafts supported by bushings having two faces subjected in use to pressure.

[0009] The object of the present invention is to provide a positive-displacement rotary pump which overcomes the disadvantages of the prior art and, in particular, which substantially reduces the noise without resulting in a substantial increase in the cost and complexity of production in comparison with pumps of known type. A further object of the invention is to provide a pump which has good leak-tightness characteristics between the intake and output, which is simple and economic to produce and maintain and which has good reliability over time.

[0010] In order to achieve the above-mentioned objects, the subject-matter of the invention is a positive-displacement pump which comprises the features indicated in the claims appended to the present description.

[0011] One advantage of the present invention consists in that the axial position of the rotors is unambiguously defined even in the event that they are subjected to axial loads or pressures owing to mechanical contact with the shell or portions thereof. In fact, it is known that, in the running-in stages of pumps of known type, it is accepted and desirable for there to be contact of portions of the rotors with the shell so that the rotors remove an extremely small layer of material until an individual seat has been "scooped out", in such a manner that, when the pump is used after the running-in operation, the clearance between the teeth of the rotors and the shell has minimal dimensions. This slight interference between the helical teeth of the rotors and the shell of the pump produces additional axial loads on the rotors, which mainly have an unknown value. The present invention, by providing a fixed plane of reference, also ensures the correct positioning of the rotors in the initial running-in stage of the pump, and even in the event of interference between the rotors and the shell, when unknown axial forces resulting from the above-mentioned mechanical contact are added to the axial forces expected in normal operation of the pump.

[0012] Other characteristics and advantages of the invention will become clear from the following detailed description which is given with reference to the appended drawings which are provided purely by way of non-limiting example and in which:

- Figures 1 and 2 illustrate, as described above, a gear pump of known type,
- Figure 3 is a longitudinal section of a gear pump according to the present invention, substantially similar to Figure 1, in which identical reference numerals identify corresponding elements,
- Figure 4 is a perspective view of a bushing set of the pump in Figure 3,
- Figure 5 is a plan view of an intermediate plate of the pump, taken in the plane of the line V-V in Figure 3,
- Figure 6 is a plan view of the front cover of the pump, taken in the plane of the line VI-VI in Figure 3,
- Figure 7 is a top view of a variant of one of the two components of a bushing set, configured to promote the hydrodynamic lubrication thereof, and
- Figure 8 is a cross-section, drawn to an enlarged scale, taken on line VII-VII in Figure 7, in which the depth of the depressions or channels of the bushing has been greatly increased for clarity of description.

[0013] Now with reference to Figure 3, a gear pump 20 comprises, as already described above with reference to the known pump in Figures 1 and 2, a shell 10 having an output opening and an intake opening for a fluid, inside which are housed the driving gear 13 and the driven gear 14. The gears 13 and 14 are of the cylindrical type having helical teeth.

[0014] The shell 10 is closed at the two ends by the front cover 11 and the rear cover 12. The end 21 of the shaft 23 of the driving gear 13 protrudes from the front cover 11. For ease of illustration, the seals between the shaft 23 and the front cover 11 have been omitted in Figure 3. Inside the shell 10, the shafts 23 and 24 of the gears 13 and 14, respectively, are supported by two bushing sets, a front bushing set 15a and rear bushing set 15b. Each of the bushing sets 15a and 15b can be produced in one piece, as in the case of the known pump in Figure 1, or preferably in two separate pieces 22a, 22b, as illustrated in detail in Figure 4. This latter preferred solution is more economical and precise from the point of view of production since it is easier to obtain very high levels of working precision without any necessity for resorting to special machine tools. Furthermore, such a configuration of bushing sets minimizes the axial output passages for fluid, in the region of the central zones 27 of the bushing sets 15a, 15b which correspond to the meshing zone of the gears 13, 14. Longitudinal channels 25 are preferably, but not in a limiting manner, provided on the two flanks of each bushing set 15a, 15b and promote the distribution of the output pressure over the flanks of the bushing sets so as to keep the two separate pieces 22a, 22b close together.

[0015] The bushing sets 15a, 15b, both in the one-piece version and in the version produced by means of separate pieces 22a, 22b, can preferably have passages having variable width in order to allow the hydrodynamic lubrication thereof. One embodiment is illustrated in Figures 7 and 8, wherein the face 17b of each piece 22a, 22b of one or both of the bushing sets 15a, 15b has depressions or channels 50 which extend in a radial direction and whose profile-section is slightly concave in a direction orthogonal to the radius of the piece 22a, 22b. A slightly deeper slot or channel 51 is preferably provided in a substantially central position in respect of each depression or channel 50 for better distribution of the lubricating fluid. The behaviour in use of a pump with bushing sets configured in this manner allows the performance thereof to be further improved, especially during use at high operating pressures.

[0016] At the end adjacent the rear cover 12, the shafts 23, 24 react against a pair of check pins or balancing pistons 29, 30 which are mounted for axial sliding in a close-fitting manner in respective axial housings 31, 32 which are provided in an intermediate plate 26. The ends of the check pins 29, 30 that are remote from the shafts 23, 24 are directed towards a common chamber 34 which is provided in the rear cover 12 and which, in use, is preferably in communication with the output of the rotary pump. In this manner, the pressurized fluid which will occupy the chamber 34 acts on the check pins 29, 30 so as to oppose the axial load produced by the gears 13, 14.

[0017] As is visible in Figure 5, a groove 27 for accommodating a seal 16 which is, for example, substantially configured in an "omega"-like manner, is provided in the face of the intermediate plate 26. An opening 28 is provided at the side of the plate that, when the pump is in use, communicates with the intake A. The configuration of the intermediate plate 26, which is similar to that of the lower cover of the pumps of the prior art in Figures 1 and 2, allows, during use of the pump, a distribution of output pressure M to be obtained over the outer face 17a of the rear bushing 15b which substantially affects the area P_{MAX} indicated with hatching in Figure 5. This area, as is known, is greater than the area of the inner face 17b of the rear bushing 15b which is subjected to the output pressure, as long as the force differential owing to the pressure provides for the production of an axial load on the rear bushing 15b directed towards the gears 13, 14, as indicated by arrow S' in Figure 3.

[0018] With reference now to Figure 6, the inner face of the front cover 11, unlike the pumps of known type, does not have the omega-like seal, and is instead provided with a large opening 36 which communicates with the intake A of the pump. Two limbs 37 which communicate with the openings 38, 39 for housing the shafts 23, 24 of the gears 13, 14 extend from the opening 36. This configuration of the face of the front cover 11 ensures that the outer face 17a of the front bushing 15a is subjected only to the intake pressure, which affects, by way of indication, the hatched area, which is denoted P_{MIN} in Figure 6. Since, however, a portion of the inner face 17b of the same front bushing 15a is also subjected to the output pressure, the pressure differential acts on the front bushing 15a in a manner counter to that seen previously for the rear bushing 15b, as long as during use of the pump, the front bushing 15a, and in particular the two pieces 22a, 22b thereof, are urged firmly against the face of the front cover 11, as indicated by arrow S" in Figure 3.

[0019] At all times during use of the pump, the pressure applied to the shafts 23, 24 by the check pins 29, 30 is such that the gears 13, 14 are in turn thrust axially onto the front bushing set 15a, as indicated by arrow S"'. Consequently, the axial forces S', S" and S"' which are produced during the operation of the pump all act in the same direction and contribute to keeping the gears 13, 14 and the front bushing set 15a and rear bushing set 15b as a whole in abutment with the reference plane, indicated by line VI-VI in Figure 3, which is constituted by the face of the front cover 11 that is directed towards the inside of the shell 10.

[0020] In this manner, there is obtained a taking-up of the axial clearances which is complete and constantly defined in spite of the oscillation of the axial forces produced during the rotation and the meshing of the gears 13, 14.

[0021] As is visible in Figure 3, the check pins 29, 30 have different diameters in order to apply different axial loads to the two gears 13, 14. This is because, in the example of the Figure, the driving gear 13 and driven gear 14 are of a helical type and therefore, during operation, produce per se axial loads whose direction is counter to and in accordance with the direction of the axial load applied by the check pins 29, 30. Naturally, in the case of gears having straight teeth which do not produce per se axial loads, the check pins 29, 30 can have substantially corresponding diameters so as to apply an axial load of equal intensity to both of the gears.

[0022] Naturally, the principle of the invention remaining the same, the forms of embodiment and details of construction may be varied if without departing from the scope of the appended claims.

Claims

1. Positive-displacement rotary pump comprising a pair of meshing gears (13, 14) or rotors, consisting of a driving gear and a driven gear, which are contained in a shell (10) having an output opening and an intake opening for a fluid, the gears (13, 14) comprising shafts (23, 24) which are supported by bushings (15a, 15b) having two faces (17a, 17b) subjected, in use, to pressures which bring about an axial load (S', S") on the bushing itself, wherein the resultant of the axial loads (S', S") on the two bushings has a predetermined direction so as to move the bushings (15a, 15b) and the gears (13, 14) as a whole into close abutment with a predetermined reference plane (VI-VI), **characterized in that** the pair of meshing gear (13, 14) or rotors are of the type having helical teeth and **in that** means are provided in the pump such that the pressure acting on the bushings (15a, 15b) provides, in use, axial loads (S', S") on these bushings (15a, 15b) which always act in the same direction as one another towards the reference plane (VI-VI), wherein the outer face (17a) of one (15a) of the two bushings is directed towards the reference plane (VI-VI) and is affected only by the intake pressure (PMIN) and a portion (PMAX) of the outer face (17a) of the other one (15b) of the two bushings is affected by the output pressure (M), the portion (PMAX) being separated from the remaining portion of the outer face (17a), which is affected by the intake pressure (A), by a separating seal (16) which is connected to a plate (26) directed towards the outer face (17a) of the bushing (15b), in such a manner that the resultant of the pressures on the outer face (17a) of the bushing (15b) is constantly greater than the resultant of the pressures acting on the inner face (17b) and **in that** it comprises a system for compensating for the loads, including axial load means (29, 30) which act on the same end of each shaft (23, 24) of the gears (13, 14), the axial load means (29, 30) causing, in use, an axial load (S"') on the gears (13, 14) which prevails over the axial forces produced by the meshing and having the same direction as the resultant of the axial loads (S', S") on the bushings (15a, 15b).

2. Positive-displacement pump according to claim 1, **characterized in that** the axial load means comprise check pins or balancing pistons (29, 30) which are mounted for axial sliding in a tight manner in respective housings (31, 32) which are provided in a plate (26) fixed to one of the two covers (11, 12) of the shell (10) of the pump.

3. Positive-displacement pump according to claim 2, **characterized in that** the ends of the check pins (29, 30) press, at one side, on the ends of the shafts (23, 24) of the gears (13, 14) and, at the other side,

are directed towards a common chamber (34) which is provided in the cover (12) of the shell (10), the common chamber (34) being in communication with the output opening of the rotary pump.

4. Positive-displacement pump according to claim 1, **characterized in that** the axial load means (29, 30) have different dimensions so as to apply different axial loads (S''') to each of the gears.

5. Positive-displacement pump according to claims 2 and 4, **characterized in that** the check pins have different diameters.

6. Positive-displacement pump according to any one of the preceding claims, **characterized in that** at least one bushing (15a, 15b) comprises at least two separate pieces (22a, 22b), each of which is intended to support in rotation an end of one of the two shafts (23, 24) of the gears (13, 14).

7. Positive-displacement pump according to claim 6, **characterized in that** at least one longitudinal channel (25) is provided on the flanks of each separate piece (22a, 22b) of the at least one bushing (15a, 15b) and promotes the distribution of the output pressure over the flanks of the bushing (15a, 15b), so as to keep the two separate pieces (22a, 22b) close together.

8. Positive-displacement pump according to claim 1, **characterized in that** depressions or localized channels (50) are provided on one (17b) of the two faces of at least one of the bushings (15a, 15b) in order to allow the hydrodynamic lubrication thereof.

9. Positive-displacement pump according to claim 8, **characterized in that** the depressions or channels extend radially over the face (17b) of the bushing, the profile-section of each depression being slightly concave in the direction orthogonal to the radius, a slightly deeper slot or channel (51) being provided in a substantially central zone of each depression (50).

Patentansprüche

1. Verdränger-Rotations-Pumpe umfassend ein Paar ineinander greifender Zahnräder (13, 14) oder Rotoren, bestehend aus einem Triebwerk und einem getriebenen Rad, die in einer Hülle (10) mit einer Ausstoßöffnung und einer Einlassöffnung für ein Fluid enthalten sind, wobei die Zahnräder (13, 14) Wellen (23, 24) umfassen, die durch Laufbuchsen (15a, 15b) mit zwei Flächen (17a, 17b) getragen sind, die im Gebrauch Drücken ausgesetzt sind, die eine Axiallast (S', S") auf die Laufbuchse selber zustande bringen, wobei die Resultierende der Axiallasten (S', S") auf die zwei Laufbuchsen eine vorherbestimmte Richtung aufweist, um die Laufbuchsen (15a, 15b) und die Zahnräder (13, 14) als Ganzes in dichte Widerlagerung mit einer vorherbestimmten Referenzebene (VI-VI) zu bewegen, **dadurch gekennzeichnet, dass** das Paar ineinander greifender Zahnräder (13, 14) oder Rotoren von dem Typus mit spiralförmigen Zähnen ist und dass Mittel in der Pumpe bereitgestellt sind, so dass das Ausüben von Druck auf die Laufbuchsen (15a, 15b) im Gebrauch Axiallasten (S', S") auf diese Laufbuchsen (15a, 15b) bereitstellt, die immer in die gleiche Richtung wirken wie gegenseitig in Richtung zur Referenzebene (VI-VI), wobei die äußere Fläche (17a) einer (15a) der zwei Laufbuchsen in Richtung zu der Referenzebene (VI-VI) gerichtet ist und nur durch den Einlassdruck (P_{min}) beeinflusst ist und ein Abschnitt (P_{Max}) der äußeren Fläche (17a) der anderen (15b) der zwei Laufbuchsen durch den Ausstoß-Druck (M) beeinflusst wird, wobei der Abschnitt (P_{Max}) von dem bleibenden Abschnitt der äußeren Fläche (17a) getrennt ist, der durch den Einlassdruck (A) von einer trennenden Abdichtung (16) beeinflusst wird, die mit einer in Richtung zu der äußeren Fläche (17a) der Laufbuchse (15b) gerichteten Platte (26) in solch einer Weise verbunden ist, dass die Resultierende der Drücke auf die äußere Fläche (17a) der Laufbuchse (15b) konstant größer als die Resultierende der auf die innere Fläche wirkenden Drücke (17b) ist, und dass sie ein System zum Kompensieren der Lasten aufweist, beinhaltend Axiallast-Mittel (29, 30), die auf das selbe Ende jeder Welle (23, 24) der Zahnräder (13, 14) wirken, wobei die Axiallastmittel (29, 30) im Gebrauch eine Axiallast (S''') auf die Zahnräder (13, 14) bewirken, die gegenüber den Axialkräften vorherrscht, welche durch das Ineinandergreifen und das Aufweisen der selben Richtung wie die Resultierende der Axiallasten (S', S") auf die Laufbuchsen (15a, 15b) hergestellt sind.

2. Verdränger-Pumpe nach Anspruch 1, **dadurch gekennzeichnet, dass** die Axiallastmittel Kontrollstifte oder Ausgleichkolben (29, 30) umfassen, die für axiales Gleiten in einer eng anliegenden Weise in entsprechenden Gehäusen (31, 32) befestigt sind, die in einer an einer der zwei Abdeckungen (11, 12) der Hülle (10) der Pumpe fixierten Platte (26) bereitgestellt sind.

3. Verdränger-Pumpe nach Anspruch 2, **dadurch gekennzeichnet, dass** die Enden der Kontrollstifte (29, 30) an einer Seite auf die Enden der Wellen (23, 24) der Zahnräder (13, 14) drücken und an der anderen Seite in Richtung zu einer gemeinsamen Kammer (34) gerichtet sind, die in der Abdeckung (12) der Hülle (10) bereitgestellt ist, wobei die gemeinsame Kammer (34) in Kommunikation mit der

Ausstoßöffnung der Rotationspumpe steht.

4. Verdränger-Pumpe nach Anspruch 1, **dadurch gekennzeichnet, dass** die Axiallastmittel (29, 30) verschiedene Dimensionen aufweisen, um verschiedene Axiallasten (S''') auf jedes der Zahnräder anzuwenden.

5. Verdränger-Pumpe nach den Ansprüchen 2 und 4, **dadurch gekennzeichnet, dass** die Kontrollstifte verschiedene Durchmesser aufweisen.

6. Verdränger-Pumpe nach einem der vorhergehenden Ansprüche, **dadurch gekennzeichnet, dass** mindestens eine Laufbuchse (15a, 15b) mindestens zwei separate Stücke (22a, 22b) umfasst, wovon jedes dafür beabsichtigt ist, während der Rotation ein Ende einer der zwei Wellen (23, 24) der Zahnräder (13, 14) zu tragen.

7. Verdränger-Pumpe nach Anspruch 6, **dadurch gekennzeichnet, dass** mindestens ein longitudinaler Kanal (25) an den Flanken jedes separaten Stückes (22a, 22b) der mindestens einen Laufbuchse (15a, 15b) bereitgestellt ist und die Verteilung des Ausstoß-Drucks über den Flanken der Laufbuchse (15a, 15b) fördert, um die zwei separaten Stücke (22a, 22b) dicht zusammenzuhalten.

8. Verdränger-Pumpe nach Anspruch 1, **dadurch gekennzeichnet, dass** Vertiefungen oder lokalisierte Kanäle (50) auf einer (17b) der zwei Flächen mindestens einer der zwei Laufbuchsen (15a, 15b) bereitgestellt sind, um die hydrodynamische Schmierung davon zu ermöglichen.

9. Verdränger-Pumpe nach Anspruch 8, **dadurch gekennzeichnet, dass** die Vertiefungen oder Kanäle sich radial über die Fläche (17b) der Laufbuchse erstrecken, wobei der Profilabschnitt jeder Vertiefung leicht konkav in der Richtung orthogonal zu dem Radius ist, wobei ein leicht tieferer Schlitz oder Kanal (51) in einer im Wesentlichen zentralen Zone jeder Vertiefung (50) bereitgestellt ist.

Revendications

1. Pompe rotative à déplacement positif comprenant une paire d'engrenages engrenés (13, 14) ou rotors, constituée d'un engrenage d'entrainement et d'un engrenage entrainé, qui sont contenus dans une enveloppe (10) ayant une ouverture de sortie et une ouverture d'entrée d'un fluide, les engrenages (13, 14) comprenant des arbres (23, 24) qui sont supportés par des bagues (15a, 15b) ayant deux faces (17a, 17b) soumises, en fonctionnement, à des pressions qui entrainent une charge axiale (S', S") sur la bague elle-même, dans laquelle la résultante des charges axiales (S', S") sur les deux bagues a une direction prédéterminée de manière à faire bouger les bagues (15a, 15b) et les engrenages (13, 14) ensemble, en butée étroite avec un plan de référence prédéterminé (VI-VI), **caractérisée en ce que** la paire d'engrenages engrenés (13, 14) ou rotors est du type ayant des dents hélicoïdales et **en ce que** des moyens sont prévus dans la pompe de telle sorte que la pression agissant sur les bagues (15a, 15b) fournit, en fonctionnement, des charges axiales (S', S") sur ces bagues (15a, 15b) qui agissent toujours dans le même sens les unes comme les autres dans la direction du plan de référence (VI-VI), dans laquelle la face externe (17a) d'une (15a) des deux bagues est dirigée vers le plan de référence (VI-VI) et est affecté seulement par la pression d'entrée (P_{MIN}), et une partie (P_{MAX}) de la face externe (17a) de l'autre (15b) des deux bagues est affectée par la pression de sortie (M), la partie (P_{MAX}) étant séparée de la partie restante de la face externe (17a), qui est affectée par la pression d'entrée (A), par un joint de séparation (16) qui est connecté à une plaque (26) dirigée vers la face externe (17a) de la bague (15b), de telle manière que la résultante des pressions agissant sur la face externe (17a) de la bague (15b) est continuellement supérieure à la résultante des pressions agissant sur la face interne (17b) et **en ce qu'**elle comprend un système de compensation des charges, comprenant des moyens de charge axiale (29, 30) qui agissent sur la même extrémité de chaque arbre (23, 24) des engrenages (13, 14), les moyens de charge axiale (29, 30) provoquant, en fonctionnement, une charge axiale (S" ') sur les engrenages (13, 14) qui l'emporte sur les forces axiales produites par l'engrènement et ayant la même direction que la résultante des charges axiales (S', S") sur les bagues (15a, 15b).

2. Pompe à déplacement positif selon la revendication 1, **caractérisée en ce que** les moyens de charge axiale comportent des broches de contrôle ou des pistons d'équilibrage (29, 30) qui sont montés pour glisser axialement de façon étanche dans des logements respectifs (31, 32) qui sont prévus dans une plaque (26) fixée à l'un des deux couvercles (11, 12) de l'enveloppe (10) de la pompe.

3. Pompe à déplacement positif selon la revendication 2, **caractérisée en ce que** les extrémités des broches de contrôle (29, 30) pressent, d'un côté, sur les extrémités des arbres (23, 24) des engrenages (13, 14) et, de l'autre côté, sont dirigées vers une chambre commune (34) qui est prévue dans le couvercle (12) de l'enveloppe (10), la chambre commune (34) étant en communication avec l'ouverture de sortie de la pompe rotative.

4. Pompe à déplacement positif selon la revendication 1, **caractérisée en ce que** les moyens de charge axiale (29, 30) ont des dimensions différentes de manière à appliquer des charges axiales différentes (S" ') sur chacun des engrenages.

5. Pompe à déplacement positif selon les revendications 2 et 4, **caractérisée en ce que** les broches de contrôle ont des diamètres différents.

6. Pompe à déplacement positif selon l'une des revendications précédentes, **caractérisée en ce qu'**au moins une bague (15a, 15b) comporte au moins deux pièces séparées (22a, 22b), chacune étant destinée à soutenir en rotation une extrémité de l'un des deux arbres (23, 24) des engrenages (13, 14).

7. Pompe à déplacement positive selon la revendication 6, **caractérisée en ce qu'**au moins un canal longitudinal (25) est prévu sur les flancs de chaque pièce séparée (22a, 22b) de ladite au moins une bague (15a, 15b) et favorise la répartition de la pression de sortie sur les flancs de la bague (15a, 15b), de manière à garder les deux pièces séparées (22a, 22b) proches l'une de l'autre.

8. Pompe à déplacement positif selon la revendication 1, **caractérisée en ce que** ces creux ou canaux localisés (50) sont prévus sur l'une (17b) des deux faces d'au moins l'une des bagues (15a, 15b) afin de permettre la lubrification hydrodynamique de celui-ci.

9. Pompe à déplacement positif selon la revendication 8, **caractérisée en ce que** les creux ou canaux s'étendent radialement sur la face (17b) de la bague, la section de profil de chaque creux étant légèrement concave dans la direction orthogonale par rapport au rayon, une fente ou canal (51) légèrement plus profonde étant prévu dans une zone sensiblement centrale de chaque creux (50).

FIG.1

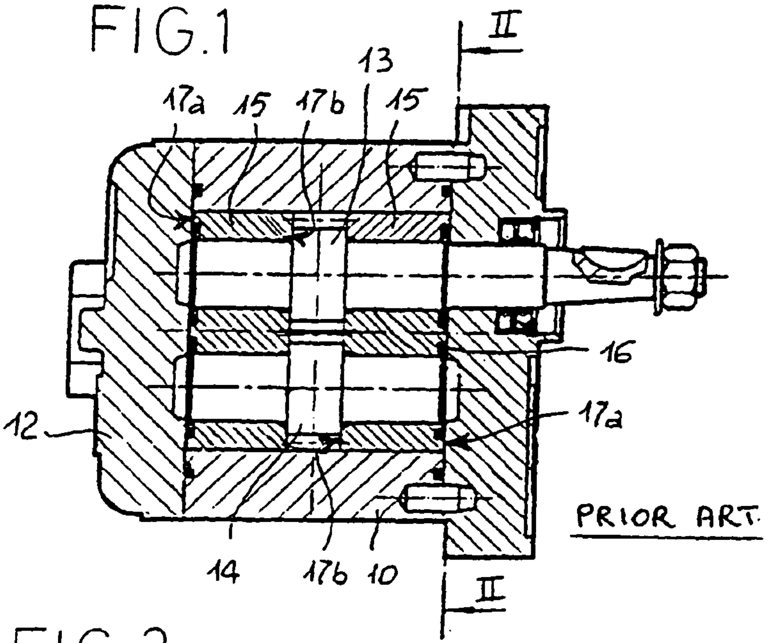


FIG.2

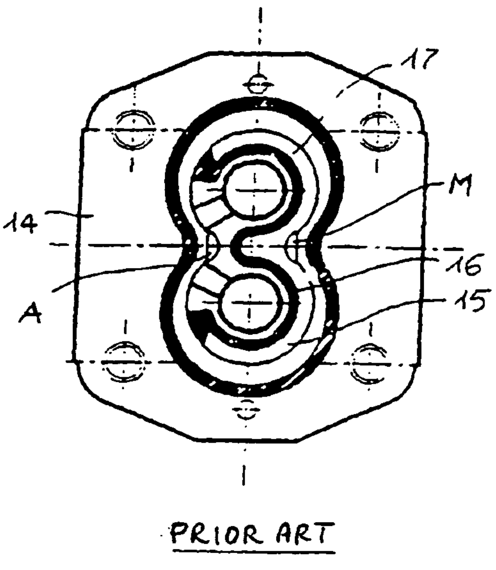
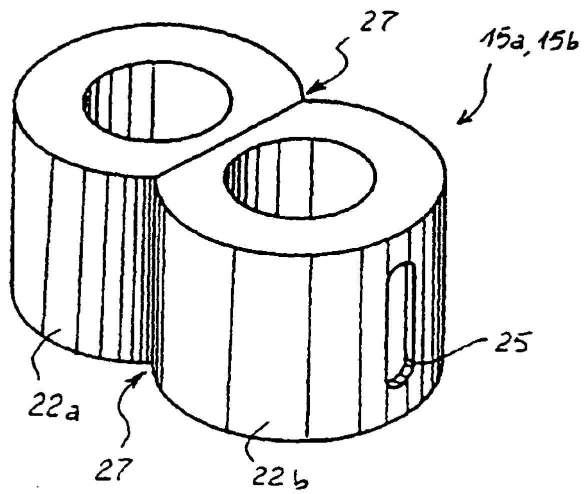


FIG.4



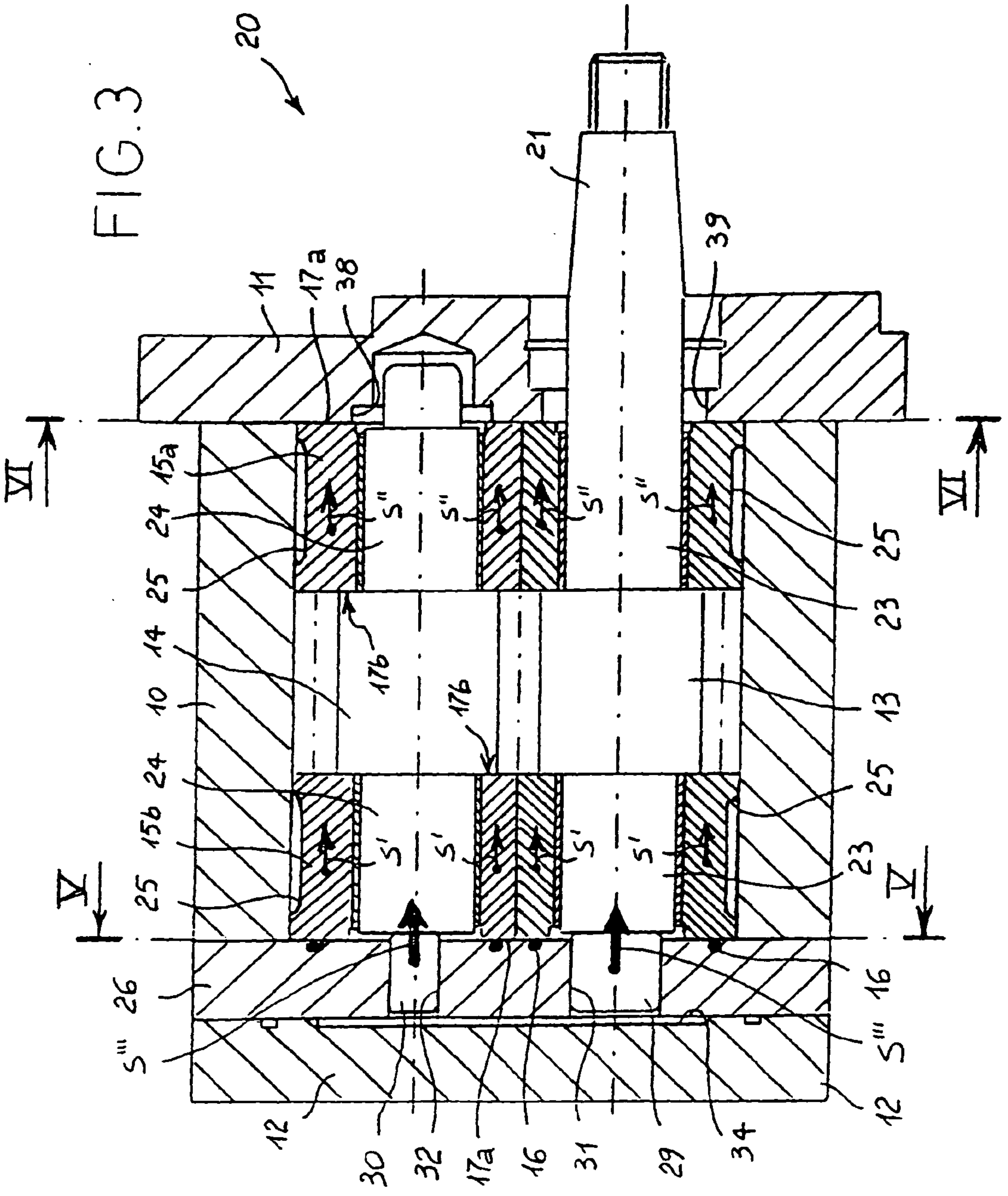

20
21
39
11
17a
38
VI
15a
24
25
14
10
17b
13
23
15b
V
26
S'''
12
30
32
16
31
29
34
S'
S''

FIG. 3

FIG.6

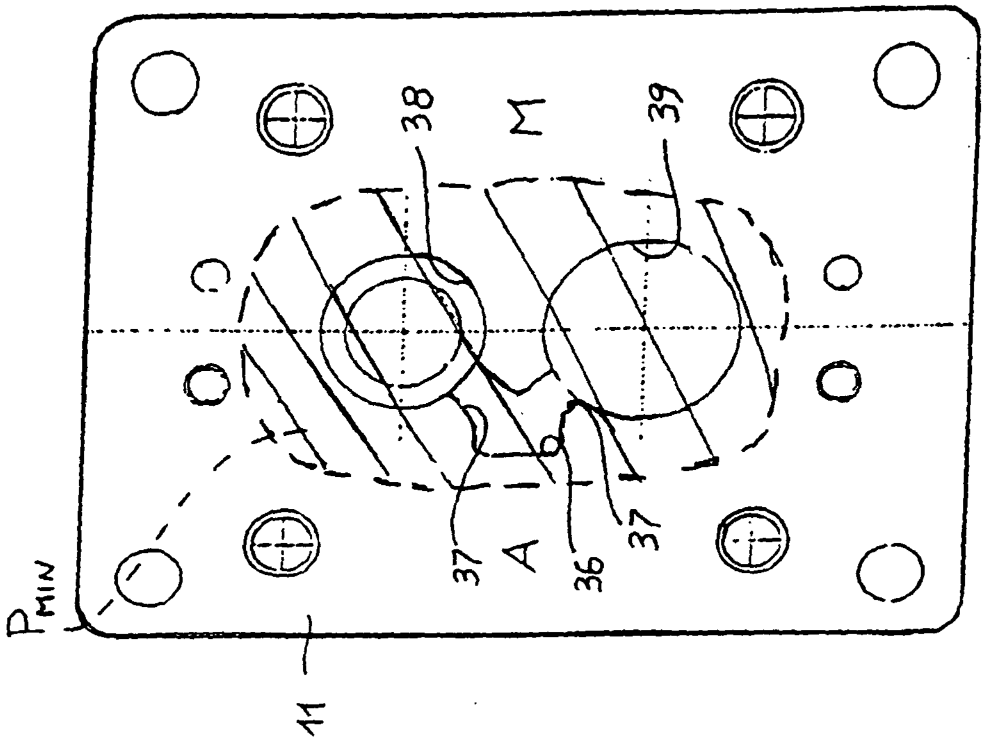


FIG.5

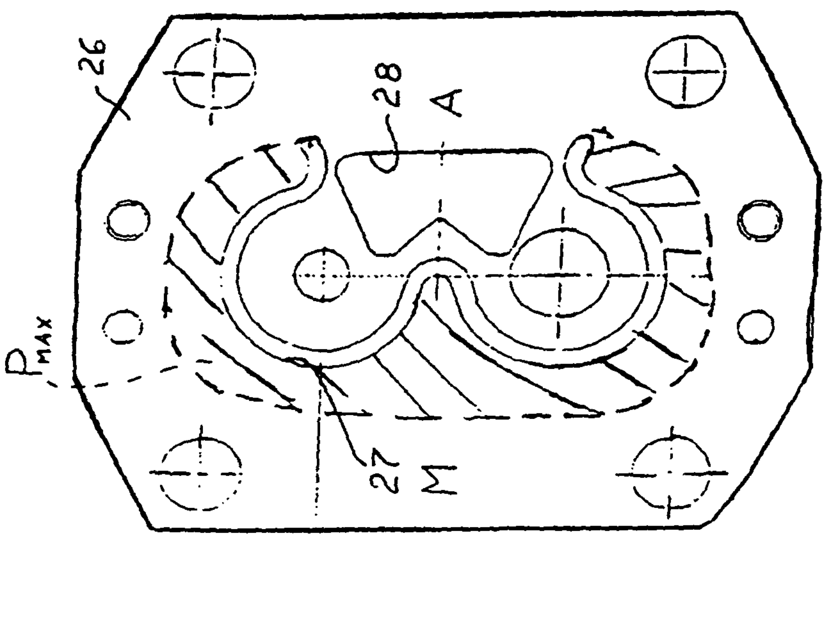


FIG.7

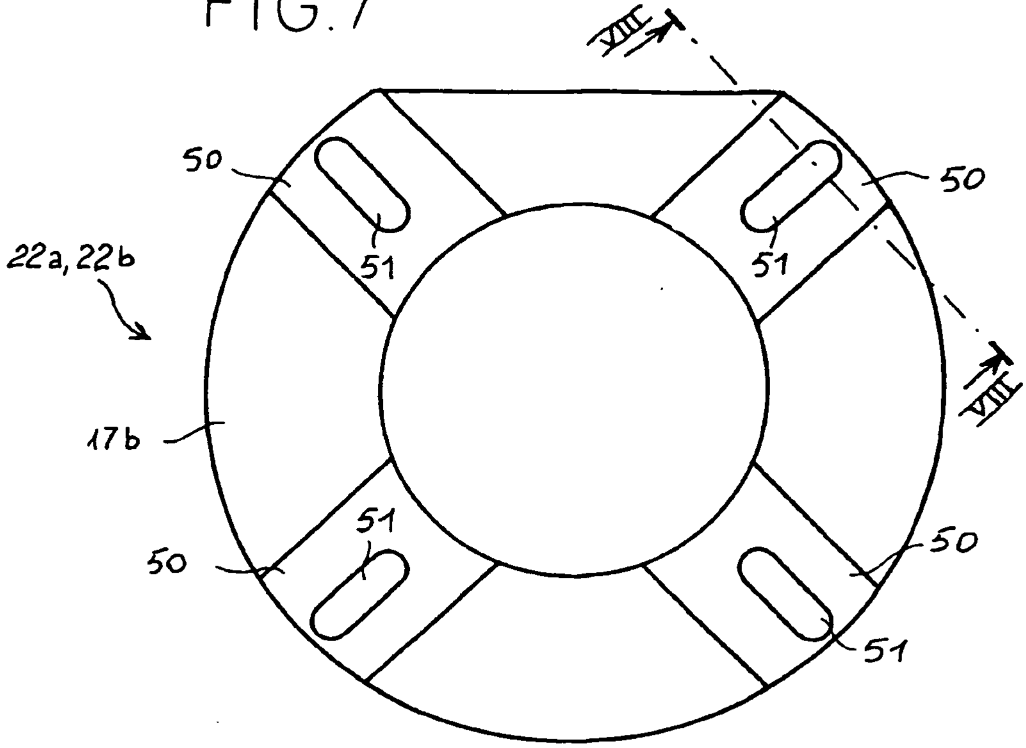
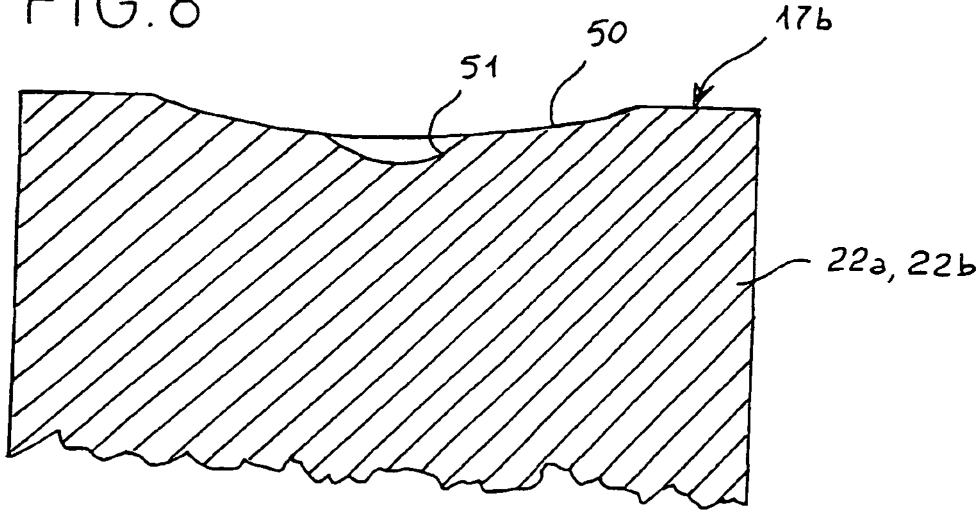


FIG.8



REFERENCES CITED IN THE DESCRIPTION

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