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## (54) A GEAR PUMP OR MOTOR

(71) We, ROBERT BOSCH GmbH, a German company of Postfach 50, 7000 Stuttgart, Germany, do hereby declare the invention, for which we pray that a patent may be granted us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to a gear pump or motor comprising a housing, two externally meshing gear wheels, shafts on which the gear wheels are mounted and bearings in which the shafts are journalled, all arranged in the housing. The bearings are arranged in pairs, the bearings of each pair being situated on a different side of the gear wheels. In some cases, a pressure plate may be urged against the side surfaces of the gear wheels, at least on one side thereof, by an hydraulically influenced axial pressure field.

Gear pumps or motors and gear pumps in particular, have the disadvantage that the delivery flow and the torque fluctuate between a delivery maximum and a delivery minimum, that is to say they are non-uniform, as a result of the finite number of delivery elements formed by the gear teeth. Thus, among other things, pulsation and undesirable noise occur in an installation connected to the pump or motor.

The gear pump or motor in accordance with the invention comprises a housing, two externally meshing gear wheels, shafts on which the gear wheels are mounted and bearings in which the shafts are journalled, all arranged in the housing, the bearings being arranged in pairs, the bearings of each pair being situated on a different side of the gear wheels and being provided with mutually opposed flat peripheral surfaces, means being provided which, when the pump or motor is in operation, generate a radially acting pressure field between the mutually opposed flat peripheral surfaces of each pair of bearings, urging each pair of bearings apart.

With that arrangement, the flank clearance of the teeth on the gear wheels can be zero and the gear wheels are pressed against one another by a relatively small force, thus increasing the uniformity of delivery flow and torque. So that the pressure force of the gear wheel tooth flanks on one another due to the radial pressure forces of the gear wheels is not too large, the radially acting pressure fields are designed to provide a counter force through the bearings. This is so calculated that the other forces are practically eliminated, that is to say the gear wheels are only pressed against one another by a relatively small force.

However, it can happen that frictional forces, for example between the bearings and housing covers when provided or between the bearings and the gear wheels or sealing means under the influence of an axial pressure field, adversely affect the adjustment of the distance between the shaft axes for a clearance free meshing of the gear wheels. Moreover, such sealing means may comprise a seal arranged in a groove between the bearings and the gear wheels, or the pressure plates if provided, and provided with some degree of pre-stressing.

Thus, in a preferred form of gear pump or motor the bearings are in the form of bearing bushes each mounted in a bearing member.

With such an arrangement, frictional forces acting on the bearing members themselves can no longer exert an influence on the adjustment of the distance between the shaft axes for clearance free meshing of the gear wheels.

The preferred arrangement is further improved if an auxiliary radially acting pressure field, located within the low pressure region of the pump or motor, is generated between each bearing bush and the bearing member in which it is mounted. Desirably, a radial clearance is provided between each bearing member and the bearing bush mounted in it and which is sufficient to permit a variation in the distance between the shaft axes, up to a flank clearance of zero for the teeth on the gear wheels.

That has the advantage that the bearing bush is free from axial forces so that, even at high pressures, it is radially movable and permits corrections of the distance between the shaft axes for a clearance free meshing of the gear wheels, even with tooth errors or if the gear wheels run out of true.

In order that the invention may be clearly understood and readily carried into effect, two embodiments thereof will now be described with reference to the accompanying drawings, in which:

Figure 1 is a longitudinal section through a gear pump in accordance with the invention, and which can also be operated as a motor,

Figure 2 is a section taken on the line II—II in Figure 1 showing a pair of bearings in the form of bearing members and including a triangle of forces,

Figure 3 is a section taken on the line III—III in Figure 1,

Figure 4 is a triangle of forces,

Figure 5 is a longitudinal section through a modified gear pump,

Figure 6 is a section taken on the line VI—VI in Figure 5 showing a pair of bearings in the form of bearing bushes,

Figure 7 is a section taken on the line VII—VII in Figure 5, and

Figure 8 shows the bearings of Figure 6 in more detail and to a larger scale.

In Figure 1, the housing 10 of the gear pump is closed at each end by covers 11 and 12. The inner space 13 of the housing is formed by two intersecting bores so that it takes the cross-sectional shape of a numeral eight. Two bearings, in this instance in the form of a pair of bearing members 14 and 15 are arranged on one side of the recess 13 and a pair of bearing members 16 and 17 of exactly the same shape are arranged on the other side of the recess 13. Shafts provided with journals 22 to 25 on which are mounted two gear wheels 26 and 27 meshing in external engagement with one another, are mounted in bearing bores 18 to 21 in the bearing members 14 to 17. The shaft journal 25 has an extension 28 which passes through a bore 29 in the cover 12 and serves for driving the machine. It also forms an output shaft when the pump is being operated as a motor.

Pressure plates 30 and 31 are arranged one on each side of the gear wheel surfaces, between the gear wheels and the bearing members. These plates 30 and 31 are urged against the side surfaces of the gear wheels by fluid pressure forces and thus reduce the leakage losses within the machine. The fluid forces are generated in pressure fields 32 and 33, the pressure field 32 being formed at the plate 30 and the pressure field 33 at the plate 31. The pressure fields 32 and 33 are facing the bearing members and are mainly influenced from the high pressure side through small bores 38. Such pressure plates with pressure fields are known and are therefore not described in further detail. All that needs be said is that, the plates 30 and 31 have through bores 34 to 37 the diameters of which are somewhat larger than the diameter of the shaft journals passing through them.

As Figure 2 shows, the bearing members 14 and 15 are provided with mutually opposed flat peripheral surfaces which, in this embodiment, abut one another along flat sides 39 and 40. Between these flat sides, an hydraulically generated radially acting pressure field 47 is built up which extends in the imaginary section plane which passes through the centres of the shaft journals 22 and 24. Moreover, the centres of the mutually opposed flat peripheral surfaces of the bearing members 14 and 15, lie on a line A—A passing through the centres of the shafts. The pressure field 47 is formed in a cylindrical recess 41 in the bearing member 14 and a similar cylindrical recess 42 in the bearing member 15. These recesses have exactly the same diameter and lie diametrally opposite one another. A piston 43, which can consist of an elastic material, is arranged to fit in the recess 42. It extends into the recess 41 and therein has an indentation at its inner surface. From the recess 41, a bore 45 leads to the pressure field 32 in which high pressure prevails. This also transmitted to the recess 41 through the bore 45. Thus, when the pump is in operation, an hydraulic pressure field 47 is generated between the bearing members through which a radially acting force $F_D$ is produced which urges the two bearing members apart in opposite directions. A similar radially acting pressure field 48 is generated between the bearing members 16 and 17.

As can be better seen in Figure 2, two coaxial bores 49 and 50 enter the housing 10 and terminate at the point of contact of the gear wheels. The bore 50 represents the low pressure or suction side of the pump whereas the bore 49 represents the high pressure side of the pump.

Due to the hydraulic forces $F_{hydr}$ acting on the gear wheels, the gear wheels, together with the bearing members, are forced towards the suction side of the pump against the housing wall on that side. In the pressure fields 47 and 48, the force $F_D$ is generated which acts outwardly towards the centres of the gear wheels. The resultant force from these two forces is shown dotted and is referenced $F_R$. The force $F_R$ may be resolved into a normal force $F_G$ and into a pressure force $F_A$ which forces the bearing members, and with them the gear wheels, towards one another with a slight excess force. Thus, the flank clearance between the teeth on the gear wheels is reduced or even becomes zero. This resolution of forces is shown in Figure 4.

The value of the pressure force $F_D$ can be chosen in accordance with the size of the pressure fields 47 and 48. When the flank clearance is practically zero, the bearing point of the bearing members in the housing must lie at the points in Figure 2 referenced 0. Then, the normal force $F_G$ can no longer influence the pressure force $F_A$.

In some circumstances with pumps having a large delivery, the radially acting pressure fields between the bearing members are no longer sufficient to retain the pressure force $F_A$ small. However, since there is no longer sufficient room between the bearing members for the size of pressure field required, an additional auxiliary radially acting pressure field can be arranged at the outer periphery of the bearing members in the low pressure region of the pump. In that manner, the pressure force $F_A$ can once again be maintained within limits. In addition, there is the possibility of reducing the normal force $F_G$ by an appropriate size and position of the radially acting pressure fields at the outer periphery of the bearing members. Wobbling of the gear wheels is prevented by the forcing of the bearing members against the housing.

In the embodiment illustrated in Figures 5 to 8, in which many like parts are given the same reference numerals as in Figures 1 to 4, the housing 10 of the gear pump is also closed at both ends by covers 11 and 12. The interior 13 of the housing is formed by two intersecting bores 13′ and 13″ whereby it adopts the cross-sectional shape of a numeral eight.

Two bearing members 14′ and 15′ are arranged in the bore 13′ on opposite sides of the gear wheel 32′. Bearing bushes 18′ and 19′ mounted respectively in the bearing members 14′ and 15′ form bearings for the shaft journals 30′ and 31′ respectively. Similarly, two bearing members 16′ and 17′ are arranged in the bore 13″ on opposite sides of the gear wheel 33′ and bearing bushes 20′ and 21′ are mounted respectively in the bearing members 16′ and 17′, the bearing bushes 20′ and 21′ forming bearings for the shaft journals 34′ and 35′ respectively.

The bearing members are each provided with a stepped bore 22′ to 25′ on which one of the bearing bushes 18′ to 21′ is mounted. Each bearing bush 18′ to 21′ is axially shorter than the bearing member in which it is mounted and its end nearest to the gear wheels 32′ and 33′ abuts against one of the shoulders 26′ to 39′ formed by a step in each of the stepped bores.

Shaft journals 30′, 31′ forming a shaft for the gear wheel 32′ and shaft journals 34′, 35′ forming a shaft for the gear wheel 33′, meshing with the gear wheel 32′, are each mounted in a respective bearing bush 18′ to 21′. Furthermore, the bearing bushes 18′ to 21′ are mounted in the stepped bores 22′ to 25′ with sufficient radial clearance to permit a variation in the distance between the axes of the shafts formed by the shaft journals 30′, 31′ and 34′, 35′, up to a flank clearance of zero for the intermeshing teeth on the gear wheels 32′ and 33′. The shaft journal 31′ has an extension 31″ passing through the cover 12 and serving as a driving shaft for the pump or as an output shaft if the pump is used as a motor.

A pressure plate 36′ is arranged between and in engagement with the side surfaces of the gear wheels 32′ and 33′ and the bearing members 14 and 16, and a similarly formed pressure plate 37′ is arranged between and in engagement with the other side surfaces of the gear wheels 32′ and 33′ and the bearing members 15′ and 17′. The shape of these sealing plates follows the contour of the interior 13 of the housing 10. On the side surfaces of the pressure plates 36′ and 37′ facing the bearing members, there are recesses 38′ and 39′ which are shaped somewhat like a numeral three (see Figure 3). A sealing member 40′ or 41′ having a U-cross-sectional shape is located in a respective recess 38′ or 39′, the limbs of the U facing the gear wheels 32′ and 33′. In this manner, axial pressure fields 42′ and 43′ are produced when the pump is in operation and which are influenced through bores 44′ and 45′ in the pressure plates 36 and 37′, the bores 44′ and 45′ being connected to the high pressure side of the pump.

As Figure 6 shows, two housing bores 46′, 47′ enter the interior 13 of the housing from opposite sides at the level of the gear wheels, the bore 47′ forming the high pressure bore and the bore 46′ forming the low pressure or suction bore.

As can be seen from Figures 5 and 6, auxiliary pressure fields 48′ to 51′ are each located between a bearing bush and the bearing member in which it is mounted. These pressure fields are facing the low pressure bore 46′, that is to say, they are located within the lower pressure region of the pump, and are formed by sealing rings 52 and 55 which lie in continuous, somewhat circular, grooves 56 and 59 which are formed in the outer periphery of each of the bearing bushes 18′ to 21′. The diameter of these grooves amounts to about two-thirds of the length of the bearing bushes.

Once again, a respective radially acting pressure field 60 or 61 is generated between each pair of bearing members 19′, 21′ and 18′, 20′ when the pump is in operation. The pressure fields 60 and 61 are each formed between mutually opposed flat peripheral surfaces provided on the bearing bushes 18′, 20′ and 19′, 21′. The pressure fields 60 and 61 are each generated within two telescopically slidable pistons 66 and 67 shown in detail in Figure 8. The pistons 66 and 67 pass through apertures in the mutually opposed flat peripheral surfaces of the bearing members and into engagement with the bearing bushes. Moreover, the pistons 66 and 67 seat in recesses 62, 63 or 64, 65 (see Figure 5) formed in the mutually opposed flat peripheral surfaces of the bearing bushes. Once again, the centres of the flat surfaces lie on a line A—A (Figure 6) passing through the centres of the gear wheel shafts.

Figure 8 shows how the pressure fields 48′ to 51′ and 60 and 61 are generated. An inlet duct 70 in the bearing member 15′ extends from the high pressure bore 47′ to the pressure field 60 through a bore 71 in the piston 66. Outlet ducts 72 and 73 extend through the bearing bushes 19 and 21 from the pressure field 60 to respective auxiliary pressure fields 48′ and 49′ generated between the bearing bushes and the bearing members and located within the low pressure region of the pump facing the low pressure bore 46′. The ducts 72 and 73 lead respectively from bores 74 and 75 also formed in the pistons 66 and 67. The pump delivery pressure prevails in the bores 70, 72 and 73. A similar arrangement of inlet and outlet ducts leading to and from the pressure fields 61, 50′ and 51′ is provided in respect of the other pair of bearing bushes 18′ and 20′.

When the machine is operating as a pump, pressure medium is drawn through the bore 46′, which represents the low pressure side, and is delivered along the walls of the bores 13′ and 13″ to the high pressure side, that is to say to the bore 47′. Due to the pressure build up along the gear wheels an hydraulic pressure force $F_{hydr}$ (see Figure 2) acts on the gear wheels, urging them towards the low pressure side. Pressure medium under high pressure is supplied to the pressure fields 48′ to 51′ between the bearing bushes 18′ to 21′ and the bearing members 14′ to 17′, whereupon a force $F_2$ (see Figure 6) is exerted on the bearing bushes 18′ to 21′ which counteracts the force $F_{hydr}$ but does not extend exactly in the effective direction of the latter.

Likewise, high pressure is applied to the pressure fields 60 and 61 so that a force $F_1$ is created which acts in a plane A—A on which the axes of the wheel shafts lie. This force $F_1$ combines with the force $F_2$, but is not large enough for the triangle of forces to close. The force $F_1$ acts in the plane A—A to force the bearing bushes apart. However, a small residual force $F_R$, remains which opposes the force $F_1$ in the plane A—A, that is to say the force $F_R$, urges the bearing bushes towards one another and also the gear wheels so that the latter mesh clearance-free with one another. A reduction in the non-uniformity of the delivery flow and torque is produced due to this clearance-free meshing.

WHAT WE CLAIM IS:—

1. A gear pump or motor comprising a housing, two externally meshing gear wheels, shafts on which the gear wheels are mounted and bearings in which the shafts are journalled, all arranged in the housing, the bearings being arranged in pairs, the bearings of each pair being situated on a different side of the gear wheels and being provided with mutually opposed flat peripheral surfaces, means being provided which, when the pump or motor is in operation, generate a radially acting pressure field between the mutually opposed flat peripheral surfaces of each pair of bearings, urging each pair of bearings apart.

2. A gear pump or motor according to claim 1, in which the bearings are in the form of bearing members.

3. A gear pump or motor according to claim 2, in which a piston, responsive to the radially acting pressure field, is arranged between the mutually opposed flat peripheral surfaces for urging the bearing members apart.

4. A gear pump or motor according to claim 3, in which the piston is arranged in two cylindrical recesses each formed in an opposite one of the mutually opposed flat peripheral surfaces.

5. A gear pump or motor according to claim 1, in which the bearings are in the form of bearing bushes each mounted in a bearing member.

6. A gear pump or motor according to claim 5, in which two telescopically slidable pistons, responsive to the radially acting pressure field, are arranged between the bearing bushes, each piston being in engagement with an opposite one of the mutually opposed flat peripheral surfaces on the bearing bushes.

7. A gear pump or motor according to claim 5, in which the two telescopically slidable pistons are hollow.

8. A gear pump or motor according to claim 6 or claim 7, in which the telescopically slidable pistons are in engagement with the bearing bushes, through apertures formed in the bearing members.

9. A gear pump or motor according to claim 8, in which the apertures are formed in mutually opposed flat peripheral surfaces on the bearing members.

10. A gear pump or motor according to any one of claims 6 to 9, in which the pistons seat in recesses formed in the mutually opposed flat peripheral surfaces on the bearing bushes.

11. A gear pump or motor according to any one of claims 5 to 10, in which, when the pump or motor is in operation, an auxiliary radially acting pressure field, located within the low pressure region of the pump or motor, is generated between each bearing bush and the bearing member in which it is mounted.

12. A gear pump or motor according to claim 11, in which an outlet duct extends from the pressure field generated between a pair of bearings to an auxiliary radially acting pressure field associated with the said pair of bearings.

13. A gear pump or motor according to claim 12, in which each outlet duct extends through a bearing bush.

14. A gear pump or motor according to any one of claims 10 to 13, in which each auxiliary pressure field is bounded by a seal arranged in a continuous groove in the outer periphery of the respective bearing bush.

15. A gear pump or motor according to any one of claims 5 to 14, in which a radial clearance is provided between each bearing bush and the bearing member in which it is mounted, the radial clearance being sufficient to permit a variation in the distance between the axes of the shafts, up to a flank clearance of zero for the teeth on the gear wheels.

16. A gear pump or motor according to any one of claims 5 to 15, in which each bearing member is provided with a stepped bore and the bearing bush is mounted in the stepped bore with its end nearest to the gear wheels abutting against a shoulder formed by a step on the stepped bore.

17. A gear pump or motor according to any preceding claim, in which an inlet duct extends from the high pressure side of the pump or motor to the radially acting pressure field generated between a pair of bearings.

18. A gear pump or motor according to claim 17, in which the inlet duct extends in a bearing member associated with the said pair of bearings.

19. A gear pump or motor according to any preceding claim, in which the centres of the mutually opposed flat peripheral surfaces of each pair of bearings, lie on a line passing through the centres of the shafts.

20. A gear pump or motor according to any preceding claim, comprising pressure plates engaging opposite sides of the gear wheels, at least one of the pressure plates being urged against the respective sides of the gear wheels by an axial pressure field generated when the pump or motor is in operation.

21. A gear pump or motor according to any one of claims 2 to 4, in which an auxiliary radially acting pressure field is arranged at the outer periphery of each bearing member, in the low pressure region of the pump or motor.

22. A gear pump or motor substantially as herein described with reference to Figures 1 to 4 or Figures 5 to 8 of the accompanying drawings.

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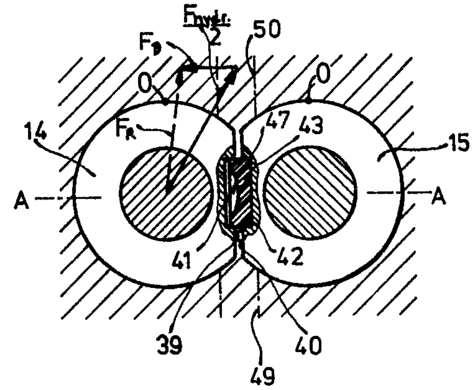
Fig.2



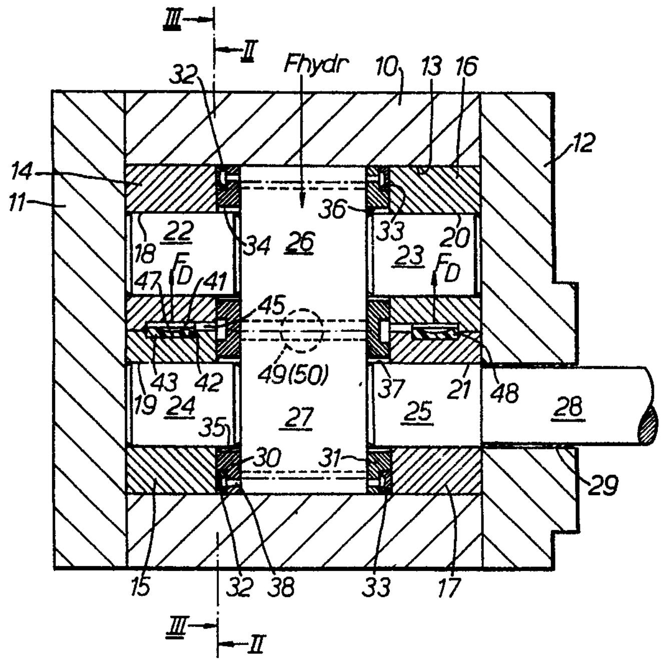

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FIG. 1.

Fig. 3

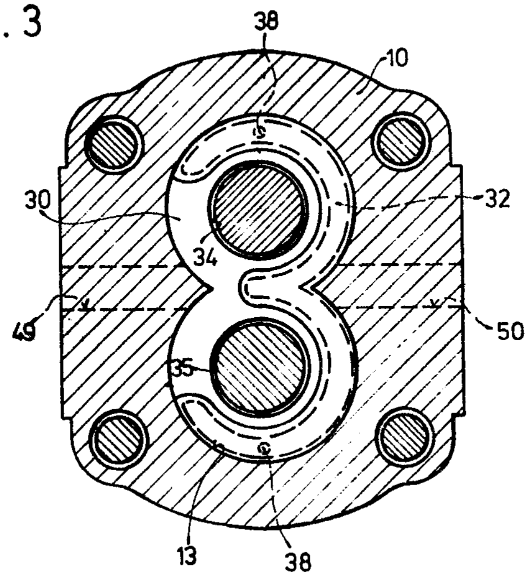
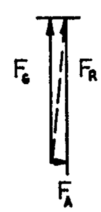



Fig. 4



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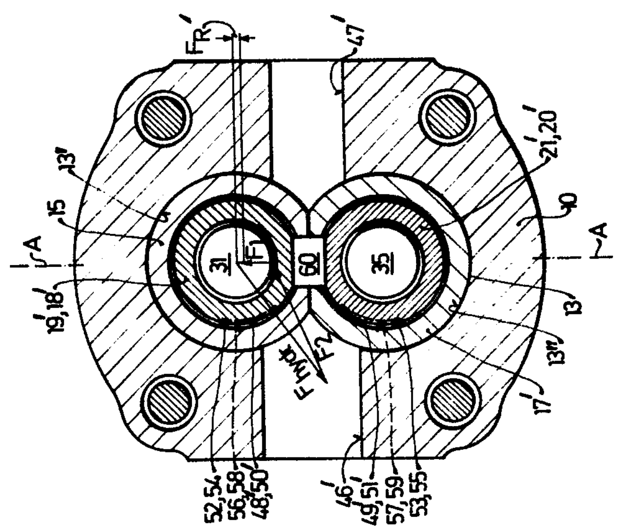

Fig.6

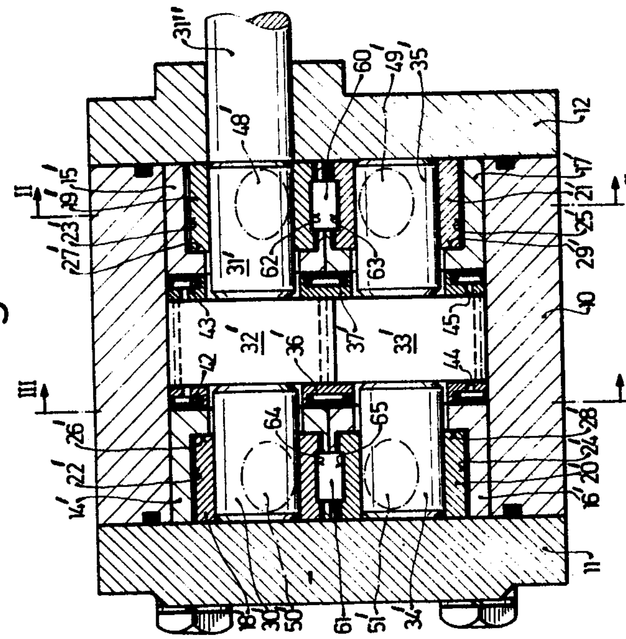

Fig.5

Fig. 8

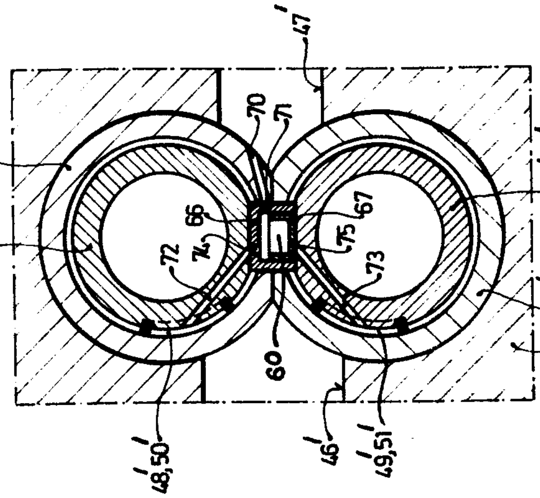
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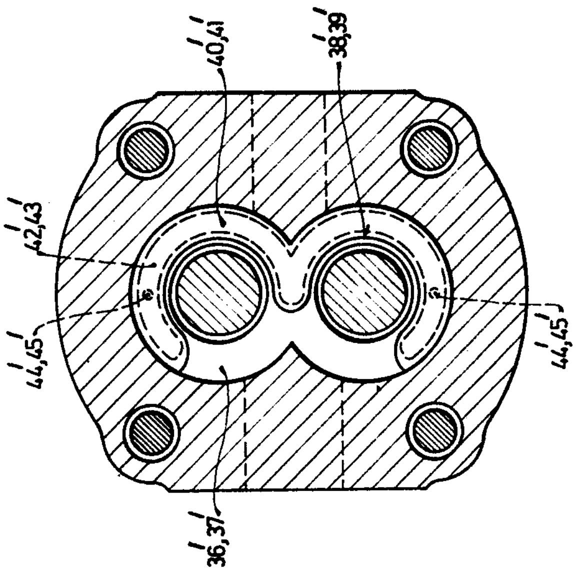
Fig. 7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