United States Patent

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[54] **GEAR-TYPE HYDRAULIC MACHINE**
22 Claims, 5 Drawing Figs.

[52] U.S. Cl. **418/73,** 417/310, 418/81, 418/133, 418/171
[51] Int. Cl. **F01c 1/10,** F01c 19/08, F01c 21/00
[50] Field of Search 418/73, 74, 81, 133, 171; 417/310

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ABSTRACT: A gear-type hydraulic machine, wherein fluid is conveyed between gear teeth of two mating gearwheels, includes an internal gear rotatably mounted in the machine and a smaller inner external gear mounted in the machine on journals and meshing with the internal gear. The pressure and the suction sides formed between the cooperating gear teeth are separated exclusively by the mating or meshing teeth. The numbers of teeth of the two gears differ sufficiently that a substantial part of the gear peripheries run out of engagement to define a crescent-shaped zone between the gears. This crescent-shaped zone communicates, through suitably dimensioned leakage passages or gaps, with the respective pressure and suction sides of the machine in order for intermediate pressure to build up in the crescent-shaped zone. This intermediate pressure operates to reduce substantially the radial thrust applied to the smaller inner gear by the pressure on the pressure side.

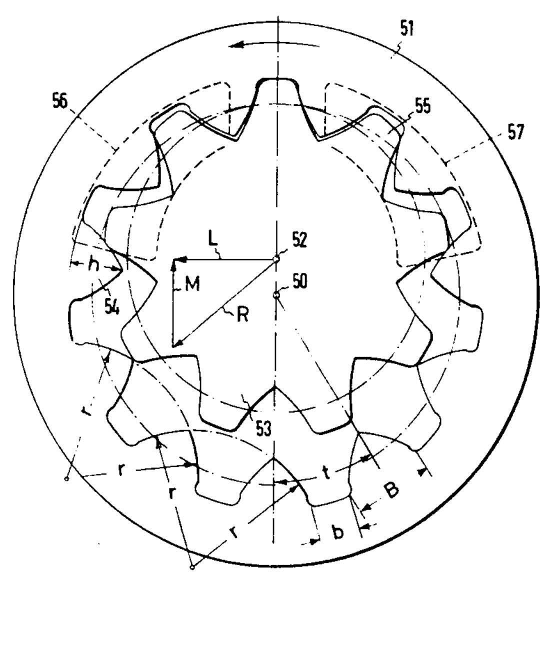

FIG.1

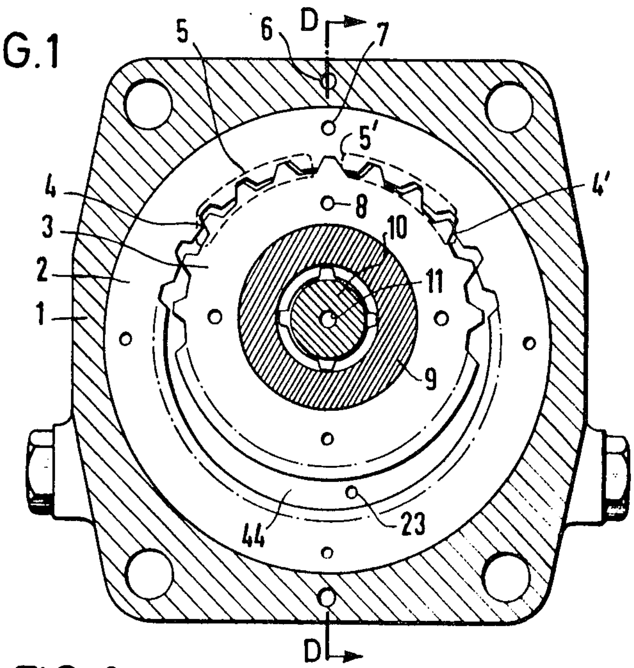

FIG. 2

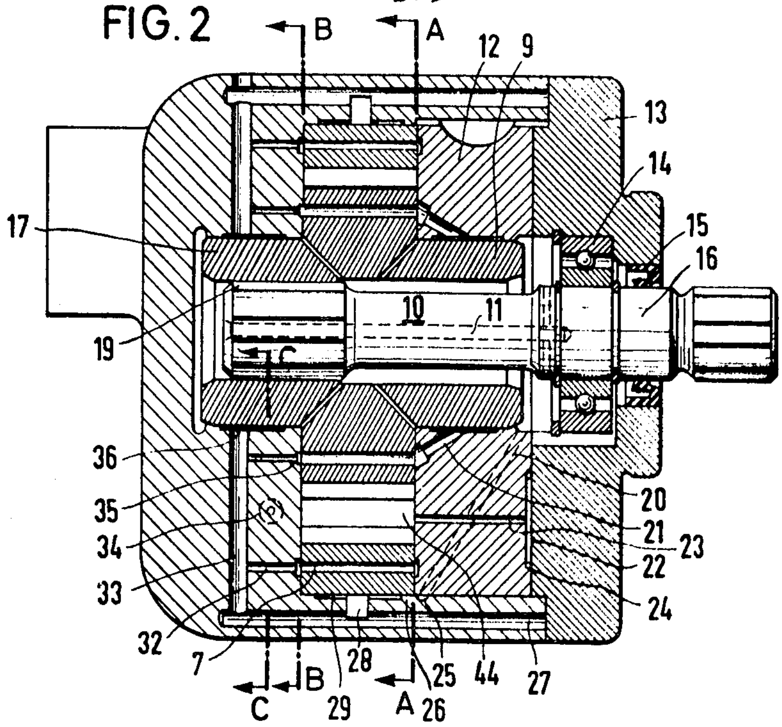

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FIG.3

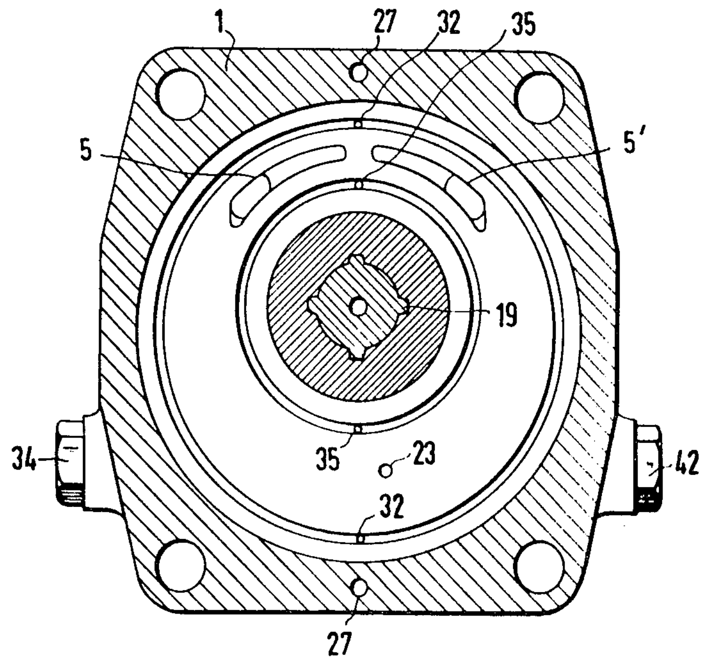

FIG.4

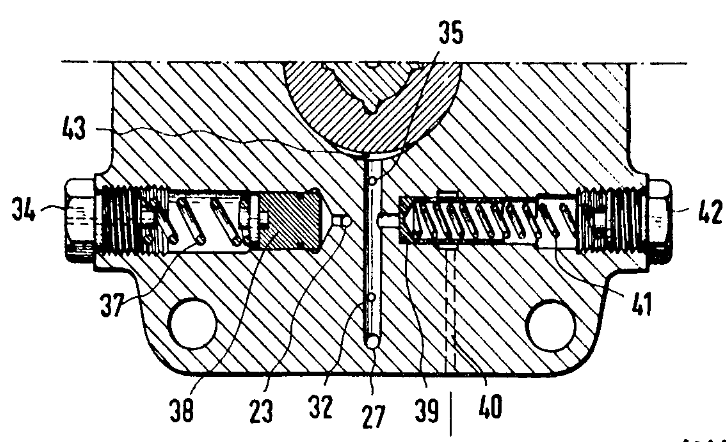

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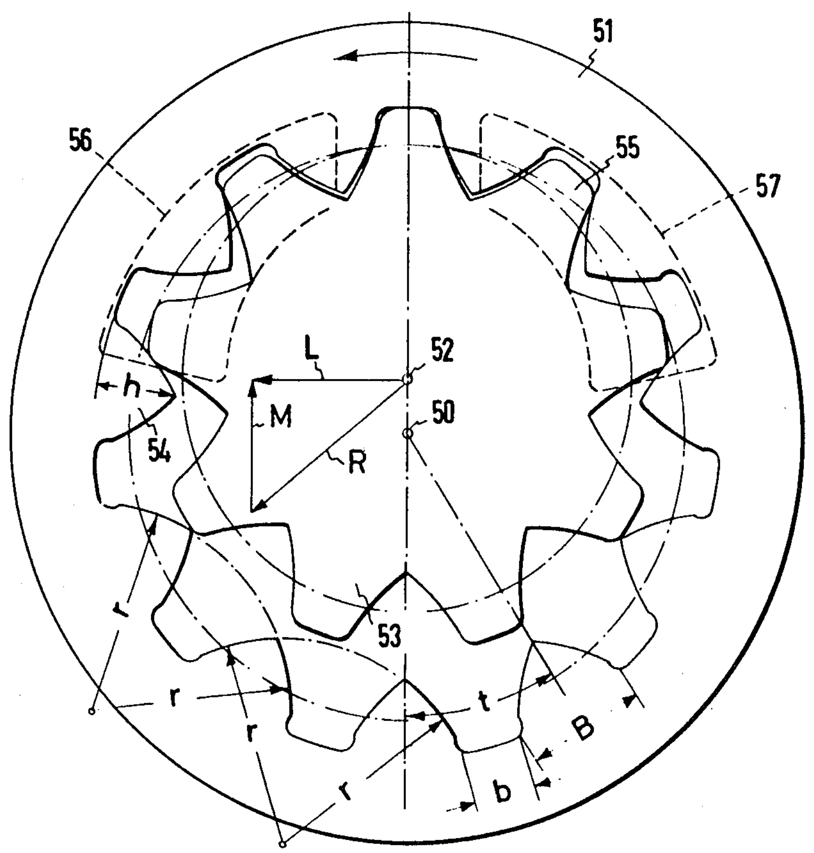
FIG. 5



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GEAR-TYPE HYDRAULIC MACHINE

BACKGROUND OF THE INVENTION

Gear-type hydraulic machines, wherein a fluid is conveyed between the gear teeth of two mating gears, of which one is an internal gear rotatably mounted in the machine and the other is a smaller inner external gear mounted in the machine on journals, are well known in the art. They are low-pressure pumps, which are built normally for pressure heads not exceeding 10 atmospheres gauge. In these pumps, the teeth of the internal gear are circular arcuate teeth, and the inner gear has one tooth less than the outer gear. The pressure heads which can be generated by such pumps are not very high because, at a point diametrically opposite the point where the pitch circles of the inner and outer gears meet, the suction and pressure sides of the machine are separated exclusively by the clearance between the crest of one tooth of the inner gear and the crest of one tooth of the outer gear.

There are also known high-pressure pumps in which an internal gear meshes with an inner external gear. In such high-pressure pumps, the difference between the numbers of teeth, or the design of the gear teeth, or both, is such that, at a point diametrically opposite the point where the pitch circles of the two gears make contact, there is a gap between the internal gear and the inner external gear. This gap is filled by a substantially crescent-shaped body, which has an internal arcuate surface in sliding contact with the crests of the teeth of the inner gear, and an external arcuate surface in sliding contact with the crest of the teeth of the outer gear. In this type of pump, the filling body creates the seal between the suction and the pressure slides.

The efficiency of pumps of this type, which are high-pressure pumps, intended to generate pressure heads of 100 atmospheric gauge and more, remains satisfactory for as long as there is no wear. However, apart from wear being liable to occur, these pumps also have other drawbacks.

In the first place, the production cost of such machines is fairly high because the filling member must be machined very precisely. Normally, this member is intended to cooperate with the ground tooth crests of the inner and outer gears in the addendum cylinders of the gears. Increasing wear of the filling member causes a considerable reduction in the pumping efficiency of such pumps.

In order to overcome this difficulty, it has already been proposed to mount the external peripheral surface of the internal ring gear in a cradlelike member surrounding the outlet port, with the filling member interposed movably between the inner and outer gears. However, this form of construction is likewise costly, and it has the further drawback of the direction of rotation of the pump not being reversible, as is now frequently required. Another major drawback of this form of construction is that the inner gear must withstand the full delivery pressure on the pressure side, which tends to bend the shaft which carries the inner gear and which is journaled in the body of the machine. This also has the effect of promoting wear. Finally the sealing surface between the pressure and suction sides on the outside of the internal ring gear is confined to approximately the width of a single tooth crest. This is also a defect of this form of construction.

SUMMARY OF THE INVENTION

This invention relates to gear-type hydraulic machines and, more particularly, to a gear-type hydraulic machine wherein the fluid is conveyed between the teeth of a rotatably mounted internal ring gear in a smaller diameter inner external gear, with the pressure and suction sides between the cooperating gear teeth of the two gears separated exclusively by the mating teeth. Even more particularly, the invention is directed to a gear-type hydraulic machine of this kind which is economical to produce, which is free of the above-mentioned disadvantages, which can be used as a high-pressure gear pump of simple design, and which is capable of being so constructed that pressure fluctuations of the fluid delivered by the pump are low for this kind of pump.

In accordance with the invention, the gear-type hydraulic machine is so designed that the thrust of the high-pressure fluid, which applies a one-sided bending moment to the shaft of the inner gear, is compensated by the generation of a counterthrust. Also, the machine is designed to be suitable for high-pressure operation, for operation with a wide range of speeds, or for both.

In further accordance with the invention, a gear-type hydraulic machine of the mentioned kind is so designed that the numbers of teeth of the two gears differ sufficiently that a substantial part of the peripheries of the two gears when out of engagement with each other, defining a crescent-shaped zone, between the gears, in which the gear teeth are not in meshing engagement. This zone communicates, through suitably designed leakage gaps or passages, with the pressure and suction sides, respectively, to develop an intermediate pressure in the crescent-shaped zone, this intermediate pressure operating substantially to reduce the radial thrust effective on the inner gear due to the pressure existing on the pressure side.

In such pumps, the inlet and outlet ports for the fluid, which are provided at least in one and preferably only in one of the sidewalls between which the gears revolve, should be as close to the point of contact of the pitch circles of the gears as the creation of a satisfactorily separating seal will permit. In the opposite directions, these ports should, in principle, extend far enough for a pumping chamber, between two teeth on the inner gear and two teeth of the outer gear, not to open after having passed across the entry port until this chamber has ceased to be in communication with the port. Conversely, the same applies on the side of the outlet port.

In such a pump, relatively favorable conditions arise if the angular distance, about the axis of the internal gear, of the far ends of the usually roughly arcuate suction and pressure ports, from the point where the pitch circles of the two gears make contact, is less than 90° and preferably between 40° and 60°. The pressure existing in the region of the port through which the fluid leaves or enters at high pressure will then exert a thrust on the inner gear directed away from the port and towards the axis of the inner gear. The circumstance that, in the invention construction, the crescent-shaped zone is filled with fluid at an intermediate pressure operates to generate a further inward thrust which acts on the inner gear. The line of action of this thrust is likewise radial and extends through the center of the inner gear and the point of contact of the two pitch circles. By correctly dimensioning the leakage clearances this thrust, which is generated by a relatively low pressure acting within a relatively large angular region, can be made to compensate exactly that component, of the first above-mentioned radial thrust generated on the pressure side, which is directed radially inwardly from the point of contact of the two pitch circles. The total inward thrust acting on the inner wheel thus can be reduced by as much as about 30 to 40 percent. Another advantage of the invention construction is that the forces acting on the internal gear likewise can be partly compensated. However, this effect on the internal gear is of secondary importance, because the internal gear can be supported around its entire periphery so that its specific bearing pressure is relatively low.

Tests have disclosed that, despite the leakage which, in the invention construction, is deliberately allowed in the regions of the ends of the ports remote from the point of contact of the pitch circles, a very satisfactory pumping efficiency nevertheless can be achieved because the loss due to this leakage is relatively small. This advantage over conventional pumps having circular arc tooth profiles on the internal gear is primarily due to the fact that the separating seal, between those ends of the pressure and suction sides remote from the point of pitch circle contact, is not limited to a single point but is divided between two points, namely one at each end of the crescent-shaped zone. This creates a kind of labyrinth effect.

Preferably the arrangement may be such that the inlet and outlet regions, i.e. the angular regions occupied by the suction and pressure ports, are separated in the peripheral direction by at least a half tooth division from the point of pitch circle contact, and that at least one of these regions extends peripherally to a point that is slightly less than one tooth division away from the point where the two gears run out of or into engagement. This form of construction ensures that a fluid-filled chamber, between two neighboring teeth of the internal gear and two teeth of the inner gear, will open already into communication with the crescent-shaped zone while it still slightly overlaps the end of the relative inlet and/or outlet port. The leakage thus can be determined exactly by choosing the degree of this overlap. It is unnecessary that, in this region, there should be a certain amount of clearance between cooperating tooth flanks. In fact it is permissible, and in practice this will usually be the case, for the meshing teeth even in this region to maintain close sliding contact, at least at one of the ports.

The required leakage naturally can be created also by suitably designing the gear teeth. However, this is more difficult to do. If the wall which contains the inlet and the outlet ports is arranged to be angularly slightly adjustable, the intermediate pressure can be controlled by appropriate adjustment which permits the overlap on the low- or high-pressure side to be increased and the other reduced, or the two overlaps to be made equal on both sides.

It follows from what has been said that the pressure in the crescent-shaped zone can be regulated according to the requirements of the design by increasing or reducing the leakage at the suction or pressure side until this pressure is of the desired magnitude.

Although the gear-type hydraulic machine according to the invention can be operated to function as a hydraulic motor, it is primarily intended to work as a gear pump. In this case the pressure side is that at the outlet port and the suction side is at the inlet port. The intermediate pressure zone then will extend between these two sides.

Preferably a gear-type machine according to the invention is arranged to be reversible. This can be done by designing the chamber in which the two meshing gears revolve, including the inlet and outlet ports, so that the layout is symmetrical about a plane containing the axes of rotation of the two gears.

The choice of an appropriate gear tooth profile is of paramount importance in a correctly designed machine according to the invention. Involute tooth flanks introduce certain difficulties. The inner gear must have a reasonable diameter to provide a sufficiently long peripheral region for the accommodation of the inlet and outlet ports between the point of contact between the pitch circles and the points where the two gears run out of or into engagement. In the case of a relatively large inner gear and a consequent small difference between the numbers of teeth on the internal gear and the inner gear, involute tooth flanks would create interference in the regions where the two gears run out of and into engagement.

This difficulty can be overcome by tip relief of the involute gear flanks. A cycloidal tooth flank is even better. However, it is preferred to provide the internal gear with gear teeth having circular arc flanks which meet at the tip. In such teeth, the tooth crests therefore are reduced to zero width. This form of construction provides not only favorable sealing tooth flank engagement but also substantially pressure-free sliding contact between the teeth of the gears for as long as they remain in mesh. It is of major importance that it should be possible to grind such tooth flanks on the internal gear with a relatively simple gear grinding machine and in a gear-type hydraulic machine comprising an internal gear the invention enables the tooth flanks of such a gear to be produced with a heretofore unattained degree of precision. This precision permits backlash to be diminished, wear to be reduced and the contact pressure to be more accurately controlled. Undesirable clearances and leakage are also reduced.

That this novel type of gear tooth can be used is the result of the proposed design features of the machine since, in conventional pumps, the employment of similar teeth would create very considerable sealing difficulties. In a pump according to the invention the clearance curve at the roots of the teeth, that is to say the bottoms of the gaps between neighboring teeth, is slightly increased to ensure that the grinding tool will clear the teeth at the roots when grinding the flanks.

Another major advantage of the proposed tooth profile is that the gaps between neighboring teeth are relatively large, in other words that the fluid-filled space available between two teeth on the internal gear and two teeth of the inner gear is relatively large. This, in turn, permits a finer circular pitch to be used, and nonuniformity of flow of the delivered fluid to be substantially diminished.

Another advantage of the proposed tooth profile is that during the displacement of the fluid trapped between the gear teeth as these come into mesh, there is no wide tooth crest abruptly penetrating into the fluid to displace a relatively large volume but only the sharp edge at the tip enters the fluid. Finally, yet another advantage of the proposed tooth profile is that it permits a wider choice of the difference between the numbers of teeth on the internal and inner gears.

More particularly, the tooth flanks may meet at the tip at an angle between 80° and 140°. An angle between 100° and 110° is preferred. It is also preferred that the relatively remote flanks of two neighboring teeth should form parts of the same circular cylinder surface.

The radius of the circle defining the curvatures of the internal gear teeth may with advantage be about 75 percent to 90 percent, preferably between 80 percent and 85 percent, of the radius of the addendum circle of the internal gear.

Moreover, the height of the teeth of the internal gear may with advantage be equal to about half the pitch of the internal gear teeth.

It is also preferred that the width of the gaps between the teeth at the roots of the teeth of the internal gear should be about 40 percent to 60 percent of the thickness of the roots of the teeth.

The above-specified dimensions are by no means obligatory, but they have proved to be essential to a satisfactory design of a pump according to the invention.

The profile of the teeth of the inner gear is preferably generated by rolling the inner gear on the internal gear. Such an inner gear may with advantage be produced by first providing a correctly shaped mold and then sintering the gear. Alternatively, the inner gear may be produced with the aid of a suitable gear-cutting machine of the nongenerating shaping or of the generating type. Finally the inner gear also may be produced by hobbing and honing on a machine using the generating roll principle.

With reference to the number of teeth, it should be mentioned that the internal gear preferably may have only one or two teeth more than the inner gear. The lesser the number of teeth of the internal gear, the lesser will be the difference between the numbers of teeth. The more teeth on the internal gear the greater will be the difference between the numbers of teeth of the two gears.

In order to locate the gear axes precisely, and in order to prevent wear in the bearings, particularly during starting up and at low angular velocities, in accordance with a further feature of the invention, the fluid forced through the clearances between the side faces of the gears, the side face of the casing and the side face of a thrust member is collected in annular grooves in the casing, thrust member or both. The collected fluid is then conducted through bores in the casing or in the thrust member or in both, and partly also axially through bores in the gears, and supplied at limited pressure to hydrostatic bearing surfaces or to collecting pockets associated with hydrodynamic bearings, or to both, the fluid being conducted to the hydrodynamic bearings alone if only hydrodynamic bearings are provided. This arrangement may be employed also, with the same advantages, in other types of gear pumps or hydraulic motors. Conveniently, the thrust member has the form of a plate which is urged by fluid pressure against one face of each of the gears.

In order to assure maintenance of uniform clearances over the entire side faces and irrespective of the difference between the pressures of the fluid escaping from the working chambers, ports and intermediate zone, areas defined by O-rings for biasing the thrust member against the sides of the gears, are pressurized through axial ducts traversing the thrust member and communicating with the working chambers, the ports, or the crescent-shaped intermediate zone, respectively, with which these areas are axially aligned.

An object of the invention is to provide an improved gear-type hydraulic machine wherein a fluid is conveyed between teeth of two mating gears including an internal ring gear and a smaller diameter external gear and wherein the pressure and suction sides between the meshing gear teeth are separated exclusively by the meshing teeth.

Another object of the invention is to provide such a machine which is economical to produce and which can be used as a high-pressure gear pump of simple design.

A further object of the invention is to provide such a machine in which pressure fluctuations of the fluid delivered by the pump are low.

Another object of the invention is to provide such a machine in which the thrust of the high-pressure fluid, applying a bending moment to the shaft of the inner gear, is compensated by the generation of a counter thrust.

A further object of the invention is to provide such a machine which is suitable for high-pressure operation, for operation within a wide range of angular velocities, or both.

Another object of the invention is to provide such a machine in which the teeth of the internal ring gear have a novel profile.

For an understanding of the principles of the invention, reference is made to the following description of typical embodiments thereof as illustrated in the accompanying drawing.

BRIEF DESCRIPTION OF THE DRAWINGS

In the Drawings: X

FIG. 1 is a pump according to the invention shown in a section taken on the line A—A in FIG. 2;

FIG. 2 is a pump according to the invention shown in a section taken on the line D—D in FIG. 1;

FIG. 3 is a pump according to the invention shown in a section taken on the line B—B in FIG. 2;

FIG. 4 is a fragmentary section taken on the line C—C in FIG. 2; and

FIG. 5 is a pump according to the invention showing a preferred gear tooth profile.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

The theoretical considerations upon which the design of a pump embodying the invention is based will first of all be described with reference to FIG. 5. In this drawing an internal ring gear 51 which revolves about a center at 50 meshes with the external teeth of a smaller diameter inner external gear 53 which revolves about a center at 52. The illustrated internal ring gear has eleven internal teeth 54, whereas the inner gear 53 has nine external teeth 55. The inlet and outlet ports 56 and 57, in the illustrated embodiment, are located, symmetrically with reference to a symmetry plane containing the two centers 52 and 50, in one sidewall of a chamber containing the two gears.

It will be understood from FIG. 5 that the flanks of the teeth 54 of the internal gear 51 form parts of cylinder surfaces having the same radius r. Moreover, the teeth are so disposed and designed that the relatively remote flanks of each pair of adjacent teeth form parts of the same cylinder surface. In the sectional drawing, these flanks appear as arcs of a common circle. Moreover, the height h of the teeth, the thickness B of the roots of the teeth and the width b of the clearance gaps between the roots of adjacent teeth of the internal gear are marked in the drawing. The pitch t of the teeth is likewise shown. The pitch of the teeth is understood to be the distance between similar points on adjacent teeth, for instance, in the present case, between the crests of the teeth if these were assumed to be a gearwheel of infinite diameter. In practice, in an internal gear, this distance can be measured on the pitch circle, as indicated in FIG. 5.

The compensation, according to the invention, of the radial pressure generated on the pressure side will now be explained. Let it be assumed that the two gearwheels work in a pump and revolve in the direction of the arrow in FIG. 5. The port 56 then will be the suction port and the port 57 the pressure port through which the pump delivers the fluid. The full pressure head of the pump then will be generated around an angular region of the inner gear which corresponds roughly to the angular extent of the port 57. The resultant thrust R then will act at the center of the inner gear. Since, in the lower part of FIG. 5, there will be generated a medium pressure which is intermediate the suction pressure and the delivery pressure, this intermediate pressure generates a thrust M at the center of the inner gear. By correctly dimensioning the available leakage cross sections between the teeth at the transition between the suction side and the intermediate pressure zone, it is possible to ensure that the magnitude of that force M will be exactly equal to the magnitude of that component of force R acting in the symmetry plane in FIG. 5. When this is the case, the only remaining radial thrust on the bearings of the inner gear 53 will be that marked L. This is only about ⅔ of the total magnitude of R. The proposed pressure compensation not only relieves the load on the bearings of the inner gear 53 but also that on the internal ring gear 51. A force equal in magnitude to R but in the opposite direction, as well as a force equal to M but opposed thereto in direction, act simultaneously on the internal gear, so that there is also a partial compensation of these two forces.

In the embodiment of the invention shown in FIGS. 1 through 4, an internal ring gear 2 is rotatably mounted within casing 1, with the external peripheral surface of ring gear 2 providing radial location thereof. An inner gear 3, having external teeth, is rotatably mounted in casing 1 and is axially located therein by means of its side faces. Gear 3 is radially located by a stub axle 17 journaled in a bearing in casing 1 and the stub axle 9 journaled in a bearing in a thrust member 12. The external peripheral surface of the thrust member 12 preferably has the same overall diameter as the outer diameter of internal ring gear 2. Thrust member 12 is axially displaceable within casing 1, but cannot rotate therein. The axial movement of thrust member 12 is limited by a cover 13 facing the outer surface of thrust member 12 and threadedly secured to casing 1 to form a tight seal with the latter.

A torsionally elastic drive shaft 16 is secured to rotate with inner gear 3, being connected to the latter by a gear coupling 19 and stub shaft 17. Shaft 16 is rotatably supported in cover 13, as by antifriction bearings 14, for example, which are externally sealed by sealing means 15 surrounding shaft 16.

The involute gear teeth, which are shown in FIGS. 1 through 4 as an alternative to the previously described preferred type of gear teeth, are provided with slight tip relief in such a way that, within the entire region of tooth engagement between the points of intersection 4, 4', of the addendum circles, the clearance between the tooth flanks is only 0.02 to 0.05 mm. The points of intersection of the addendum circles and the point of contact of the pitch circles, or briefly the point of full engagement, thus sharply separate the suction side, the pressure side and the intermediate pressure zone of the machine. The ports 5 and 5' through which the pumped fluid enters or leaves, in accordance with the direction of rotation, are located in the sidewall of casing 1. However, it is naturally also possible, in a manner known in the art, to admit or discharge the pump fluid peripherally into and from internal ring gear 2, through openings in the periphery of this gear into the space between the teeth or correspondingly to admit or discharge the pumped fluid from the inside in a similar way between the external gear teeth of inner gear 3. This may have advantages under certain working and design conditions, for instance when the gears are cylinder gears of considerable axial length, for handling large volumes, or if the gear diameters are small.

In the embodiment of the invention illustrated in FIGS. 1 through 4, the external periphery of internal ring gear 2 is mounted in a bearing recess in casing 1, and the stub shafts of inner external gear 3 are mounted in bearing recesses in casing 1 and thrust bearing 12, respectively. The bearing recesses form two hydrodynamic bearing surfaces 26 separated by an interposed hydrostatic bearing surface 29 communicating with ducts 27 through bores 28. Fluid that has been forced through the clearances between the side faces of the gears and the side faces of casing 1 and thrust member 12, respectively, and which has been collected in annular grooves 32 and 35 in casing 1, enters ducts 27 through bores 33 which communicate with grooves 32 and 35. This fluid also enters partly through axial bores 7 and 8 in the gears, and is thus conducted at limited pressure to hydrostatic bearing surfaces 29, 36, or into pockets 43 (FIG. 4), when only hydrodynamic bearings are used.

This limited pressure may be controlled by a pressure regulating valve 42 in casing 1, valve 42 comprising a plunger 39 which, when necessary, uncovers an outlet opening 40 against the bias of a spring 41. This limited pressure also may be admitted into areas 22 on the outer surface of thrust member 12, for generating axial thrust on thrust member 12, the areas 22 being defined and tightly sealed by the O-rings 24. However, in a preferred form of construction the pressure-generating areas are pressurized through axial bores 23 traversing thrust member 12, to transmit the pressures existing in working chambers or ports 5 and 5′, or the intermediate zone that are in axial alignment with the corresponding thrust-generating areas.

Considerable pressure fluctuations during operation, in the crescent-shaped intermediate zone, also may be damped and equalized by a pressure-equalizing device 34 (FIGS. 2 and 4), including a piston 38 exposed to the pressure in the crescent-shaped zone and displaceable against the bias of a spring 37. At the same time, this arrangement also suppresses pressure fluctuations in the leakage gaps. The pumped fluid, which is expelled from the hydrodynamic bearings, is collected in annular grooves 25 and delivered through bores 20, 21 and 11 to a tank or the like.

While specific embodiments of the invention have been shown and described in detail to illustrate the application of the principles of the invention, it will be understood that the invention may be embodied otherwise without departing from such principles.

What is claimed is:

1. In a gear-type hydraulic machine, wherein fluid is conveyed between gear teeth of two mating gearwheels, one of which is an internal gear rotatably mounted in the machine and the other of which is a smaller diameter inner external gear mounted in the machine on journals, and wherein the pressure side and the suction side, formed between the intermeshing teeth of the two gears, are separated solely by the mating teeth: the improvement comprising the numbers of teeth of said two gears differing sufficiently for a substantial part of the gear peripheries to run out of engagement with each other to define a crescent-shaped zone between said gears and in which the gear teeth are not in engagement; and suitably dimensioned leakage paths connecting said crescent-shaped zone with the pressure side and with the suction side to build up, in said crescent-shaped zone, an intermediate pressure operating substantially to reduce the radial thrust applied to said smaller diameter inner gear by the pressure on the pressure side.

2. In a gear-type hydraulic machine, the improvement claimed in claim 1, in which the fluid inlet and fluid outlet regions are separated peripherally, by at least a half tooth division, from the point where the pitch circles of said two gears make contact; at least one of said fluid inlet and fluid outlet regions extending peripherally, in a direction away from said point, to a point that is spaced slightly less than one tooth division from the adjacent peripheral end of said crescent-shaped zone.

3. In a gear-type hydraulic machine, the improvement claimed in claim 1, wherein said machine is a gear pump.

4. In a gear-type hydraulic machine, the improvement claimed in claim 1, wherein said machine is reversible.

5. In a gear-type hydraulic machine, the improvement claimed in claim 4, in which said machine has inlet and outlet ports; the space in which said gears revolve, including said inlet and outlet ports, having a symmetrical configuration with reference to a symmetry plane including the axes of rotation of said gears.

6. In a gear-type hydraulic machine, the improvement claimed in claim 1, in which said internal gear has gear teeth with circular arc tooth flanks meeting at a common tip.

7. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which said tooth flanks meet at said tip at an angle of 80° to 140°.

8. In a gear-type hydraulic machine, the improvement claimed in claim 7, in which said tooth flanks meet at said tip at an angle of 100° to 110°.

9. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the radius of the circular arc of said tooth flanks of said internal gear is about from 75 percent to 90 percent of the radium of the addendum circle of said internal gear.

10. In a gear-type hydraulic machine, the improvement claimed in claim 9, in which the radium of the circular arc of said tooth flanks of said internal gear is about from 80 percent to 85 percent of the radium of the addendum circle of said internal gear.

11. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the height of said teeth of said internal gear is approximately equal to half the pitch of said teeth of said internal gear.

12. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the width of the gaps between the teeth at the roots of said teeth of said internal gear is equal to about from 40 percent to 60 percent of the thickness of said teeth at the roots thereof.

13. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the shape of the external teeth of said inner gear is generated by rolling on said internal gear.

14. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the number of teeth of said internal gear is one more than the number of teeth of said inner gear.

15. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the number of teeth of said internal gear is two more than the number of teeth of said inner gear.

16. In a gear-type hydraulic machine, the improvement claimed in claim 6, in which the relatively remote tooth flanks of adjacent gear teeth of said internal gear from parts of a common cylinder surface.

17. In a gear-type hydraulic machine, the improvement claimed in claim 1, including a casing in which said gears are rotatably mounted, said casing having a sidewall member engaging corresponding first sides of said gears; a thrust member mounted in said casing and engaging corresponding second sides of said gears; at least one of said members being formed with annular grooves communicating with the clearances at the side faces of said gears, to collect fluid forced into said clearances; said casing and said thrust member having bearing recesses defining, with said gears, hydraulic bearing recesses; ducts communicating with said hydraulic recesses; and bores connecting said annular grooves to said ducts to supply the collected fluid to said hydraulic bearing recesses at a limited pressure.

18. In a gear-type hydraulic machine, the improvement claimed in claim 17, in which said bores include bores extending radially of said wall member of said casing.

19. In a gear-type hydraulic machine, the improvement claimed in claim 17, in which said bores include bores extending axially of said internal gear.

20. In a gear-type hydraulic machine, the improvement claimed in claim 17, including a cover plate secured to said casing and engaging with the outer side of said thrust member; said outer side of said thrust member being formed with pressurized areas defined by O-rings for urging said thrust member against the second sides of said gears; and axial ducts formed in said thrust member and connecting said pressurized areas with working chambers, inlet and outlet ports, or said crescent-shaped zone of said machine then axially aligned with said pressurized areas.

21. In a gear-type hydraulic machine, the improvement claimed in claim 17, including a pressure relief valve communicating with said bores.

22. In a gear-type hydraulic machine, the improvement claimed in claim 20, including a pressure-equalizing device communicating with said crescent-shaped zone.

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