

[54] **HYDRAULIC GEAR PUMPS AND MOTORS**

[72] Inventor: Robert Gelin, Lyon, France
 [73] Assignee: Societe Industrielle Generale de Mecanique Appliquee S.I.G.M.A., Paris, France
 [22] Filed: Sept. 8, 1969
 [21] Appl. No.: 856,108

[30] **Foreign Application Priority Data**

Sept. 9, 1968 France.....165536

[52] U.S. Cl.....418/206
 [51] Int. Cl.....F01c 1/18, F03c 3/00, F04c 1/08
 [58] Field of Search418/131, 133, 206, 132; 91/87; 123/12

[56]

References Cited

UNITED STATES PATENTS

1,940,410	12/1933	Fitch et al.....	418/133
2,627,232	2/1953	Lauck.....	418/131
2,672,100	3/1954	Harvey.....	418/206
2,769,396	11/1956	Norlin.....	418/131
3,043,230	7/1962	Molly.....	418/131
3,173,374	3/1965	Beimfohr.....	418/131
3,395,646	8/1968	Gelin.....	418/133

Primary Examiner—Carlton R. Croyle
 Assistant Examiner—John J. Vrablik
 Attorney—Gerard J. Weiser

[57]

ABSTRACT

Hydraulic gear pumps and motors having different displacements are provided by means of families of interchangeable parts. Leakage pathways from internal zones of high pressure are provided to a low-pressure zone in the vicinity of the intake.

7 Claims, 4 Drawing Figures

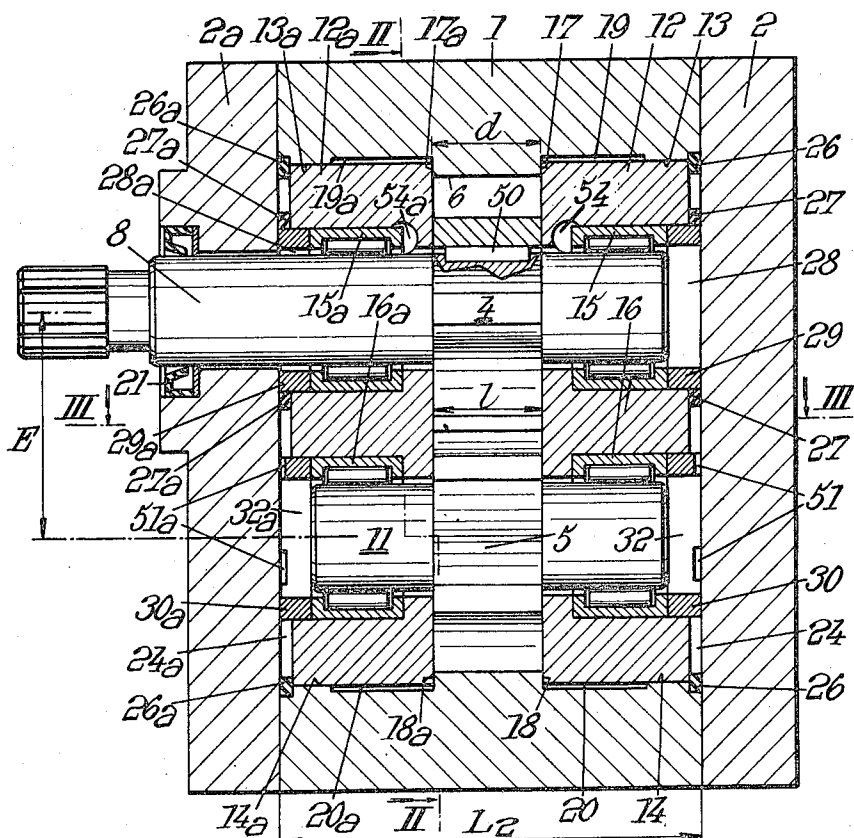


Fig. 2.

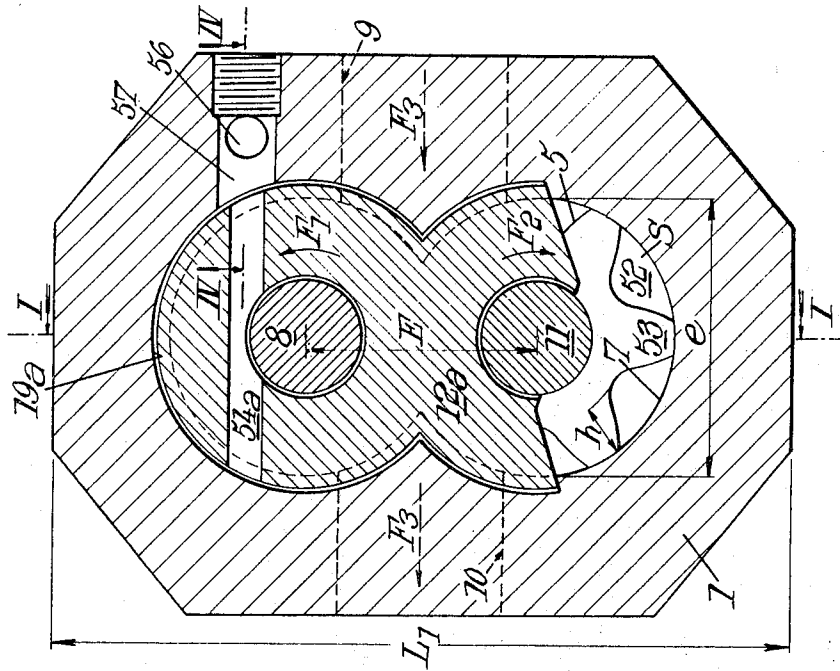
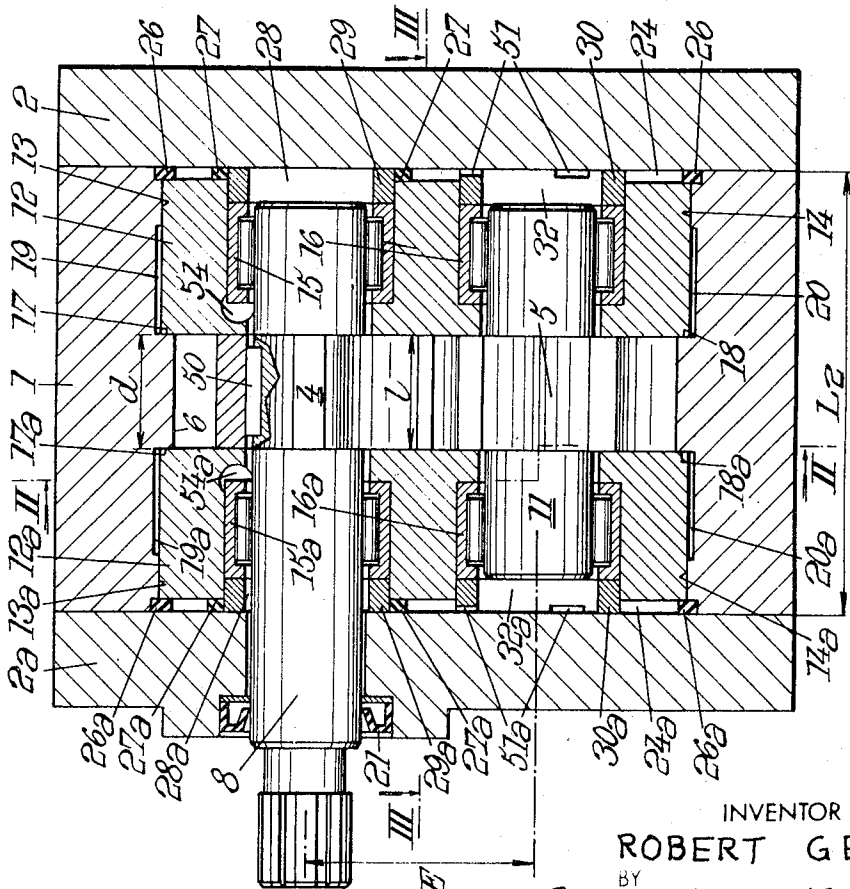


Fig. 1.



INVENTOR
 ROBERT GELIN
 BY
 McChesney, Weiser & Millman
 ATTORNEYS

Fig. 3.

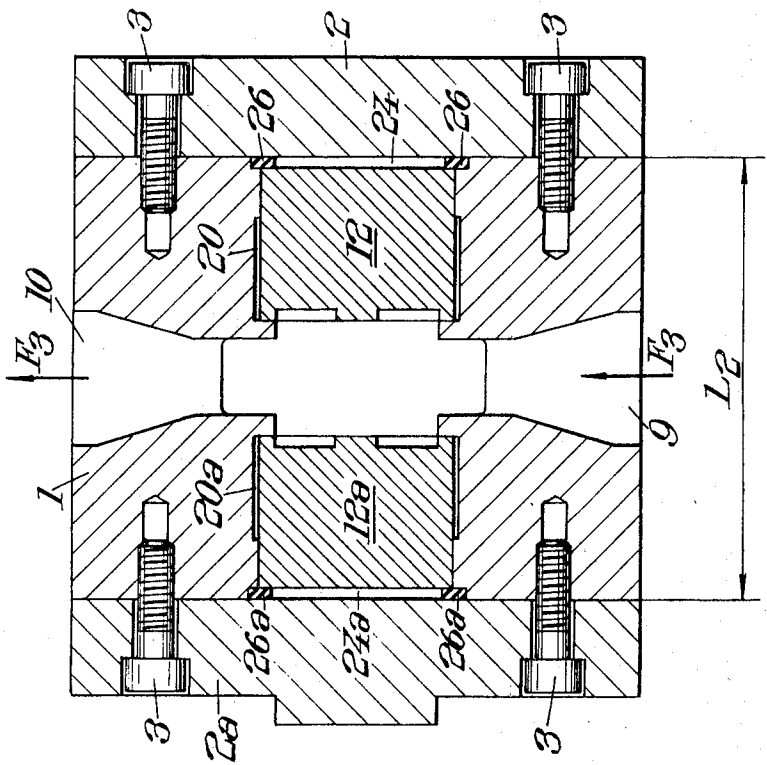
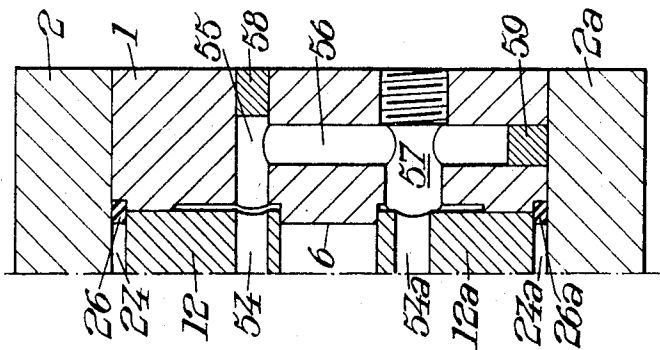


Fig. 4.



INVENTOR
ROBERT GELIN
BY
McClure, Weiser & Millman
ATTORNEYS

HYDRAULIC GEAR PUMPS AND MOTORS

Reference is made to the same applicant's U.S. Pat. No. 3,395,646 granted Aug. 6, 1968, which corresponds to French Pat. 1,433,479, and U.S. Pat. application Ser. No. 720,817 filed Apr. 12, 1968, which corresponds to French Pat. 1,526,323.

This invention relates to improvements in hydraulic gear pumps and motors having component parts in common and being capable of different displacements or deliveries per revolution, particularly though not exclusively those disclosed in the aforementioned patents, as representative of a class of hydraulic volumetric machinery having a generally analogous concept.

Such machinery comprises essentially a casing, two intermeshing gears inside the casing and mounted, respectively, on a driving shaft protruding through one side of the casing and on a driven shaft entirely within the casing. The shafts, whose axes are spaced apart an interaxial distance E , are carried on each side thereof by a common bearing, which is rigid and movable in the axial direction toward the casing interior. Other parts, including bearings, gaskets, and side plates (front and rear), complete such machinery, whose displacement depends essentially upon the gears, characterized by a modulus or diametral pitch m , number of teeth N , tooth width l and finally a tooth correction having to do with the geometric profile of the teeth.

It is known that in such machinery internal leakages of liquid are produced from zones of high pressure (discharge zone for a pump, intake zone for a motor) toward zones of low pressure (intake zone for a pump, exhaust zone for a motor). These leakages occur chiefly between gears and bearings. A return pathway for casing leakages toward the zone of low pressure is provided in such machinery.

A primary object of the present invention is provision of families of volumetric machinery of this kind such that the number of parts common to machinery in the same family is increased and that consequently such machinery can be constructed beginning with a more limited stock of parts.

The invention consists principally, in a family of machinery of this kind, in obtaining differences in displacement among different members of the family, by effecting variation in at least one of the following parameters: the modulus m , the number of teeth N , and the tooth width l ; while maintaining constant the interaxial distance E (between the axes of the driving and driven shafts), and minimizing variations in external diameter of the gears, in such a way that the driving and driven shafts in the same family have the same dimensions independent of the displacement and that preferably the overall dimensions of the casings of the members of the same family are identical.

The invention consists in certain arrangements used preferably together but also capable of being used separately when the occasion arises and as will be specified hereinafter, especially in an arrangement relating more particularly to a return pathway for the internal leakages of each member of the family of the kind in question, which consists in providing for this pathway a direct line. It comprises, on the one hand, two ducts bored respectively in each of the bearings in the vicinity of the driving shaft and the gear carried thereby and extending substantially perpendicular to the axial direction and, on the other hand, drains bored in the casing enabling the ducts to be connected to the zone of low pressure.

The invention can, in any case, be well understood from the following description and the accompanying drawings, of a preferred embodiment.

FIG. 1 is an axial sectional elevation, taken along I—I on FIG. 2, of a gear pump according to this invention;

FIG. 2 is a transverse sectional elevation thereof taken along II—II on FIG. 1;

FIG. 3 is a sectional plan, taken along III—III on FIG. 1, with the gears removed; and

FIG. 4 is a fragmentary sectional plan taken along IV—IV on FIG. 2.

In its essentials the pump comprises, according to the arrangements disclosed in the aforementioned patent documents, a casing 1 and two sideplates (front and rear) 2 and 2a, held together by screws 3, and comprises also two intermeshing gears 4 and 5 disposed in two blind bores 6 and 7. Gear 4 is affixed by means of key 50 to driving shaft 8, which protrudes through sideplate 2a only. Shaft 8 turns in the direction indicated by arrow F_1 (FIG. 2). An intake opening 9 and a discharge opening 10 communicate with the region of overlapping of bores 6 and 7. Gear 5 is itself affixed by means of another key (not shown) to driven shaft 11, which does not protrude through either of sideplates 2 and 2a. A packing gland 21 is provided where shaft 8 passes through sideplate 2a.

Shafts 8 and 11 are arranged parallel to one another in bearings 12 and 12a, located on opposite sides of the gears, each being a rigid one-piece bearing having approximately a figure-eight configuration. Interaxial distance E between the axes of the driving and driven shafts is indicated in FIGS. 1 and 2. Bearings 12 and 12a are housed, respectively, in blind bores 13, 14 and 13a, 14a coaxial with bores 6 and 7 in the casing. Roller bearings 15, 16 and 15a, 16a are advantageously placed between bearings 12 and 12a and respective shafts 8 and 11.

The side face of each bearing 12, 12a furthestmost from the side of the corresponding gear forms one of the inside walls of a chamber 24 or 24a which receives liquid under pressure tending to displace the bearing concerned toward the side of the gear. The liquid is conducted away from the discharge of the pump by channels (not visible) traversing the bearings. Stop means are provided limiting, at a value slightly greater than tooth width l (FIG. 1), minimum distance d (also FIG. 1) between the opposing faces of bearings 12 and 12a. The stop means advantageously are made up of shoulders 17, 17a and 18, 18a provided in pump casing 1 between central bore 6 or 7 and side bores 13, 13a or 14, 14a. Shoulders 17, 17a and 18, 18a are thus separated by distance d .

Preferably the shoulders in question are made up of the edges of an extra thickness of the body of casing 1, with bores 6 and 7 each having diameter e (FIG. 2) slightly less than those of side bores 13, 13a and 14, 14a.

Rabbets 19, 19a and 20, 20a cut into the side bores of the casing permit a reduction in the guideways of bearings 12 and 12a.

Gears 4 and 5 are shown with enough axial clearance that they can be centered automatically by abutment of bearings 12 and 12a against shoulders 17, 17a and 18, 18a. Annular gaskets 26, 26a and 27, 27a isolate the liquid under pressure in chambers 24, 24a from the outside or zones 28, 28a under low or intake pressure by a leakage pathway, as will be seen hereinafter.

Spacer rings 29, 29a and 30, 30a function as axial thrust bearings. Spacer rings 30, 30a are designed so as to enable liquid under pressure in chambers 24, 24a to fill freely zones 32, 32a located adjacent the ends of shaft 11. To do so these rings have, on their faces bearing against sideplates 2 and 2a, radial grooves 51, 51a intercommunicating between zones 32, 32a and all of respective chambers 24, 24a.

The operation of this pump is readily understood, as follows.

When gears 4 and 5 are set into rotation in the direction of arrows F_1 and F_2 , respectively, they drive liquid from intake opening 9 toward discharge opening 10 (FIG. 2) confining it in cells 52 having cross section S (also FIG. 2) bounded by the hollows between the gear teeth, the adjacent inside walls of bearings 12 and 12a, and bores 6 and 7. The liquid is displaced in the direction of arrows F_3 .

The discharge pressure which is built up in chambers 24 and 24a forces the bearings against their stops, centering the gears without hindering their rotation. The compressive forces acting upon the opposite ends of shaft 11 balance it.

On one hand, it is obvious that the displacement of such a pump depends essentially upon the volume of cells 52 and the number of such cells.

On the other hand, it is equally obvious that the liquid confined in a cell 52 and whose pressure increases progressively upward to the discharge pressure tends to leak between the bearings toward zones, such as 28 and 28a, at the casing interior. If a return pathway to the intake is not provided, the pressure of the liquid in zones 28 and 28a will increase to such a degree that packing gland 21 will be subjected on one side to the discharge pressure, which it cannot stand.

Heretofore, in order to bring about variations in displacement between pumps in the same family, it was customary to vary the tooth width l of the gears, but the modulus m , the number of teeth N , and the tooth correction (i.e., the tooth profile) remained constant.

An important disadvantage of such variation in tooth width l is a resulting variation in the length of driving shaft 8 and driven shaft 11 within the same family. Accordingly, casing 1 for the same family retained a constant height L_1 (FIG. 2) but had a variable length L_2 (FIGS. 1 and 3) so that a single model of bearings 12, 12a would fit only one such length.

For example, for a family of pumps whose displacements or deliveries per revolution were equal at 10, 12, 14, and 18 cubic centimeters the corresponding tooth widths were l , 1.2 l , 1.4 l , and 1.8 l . Under these conditions it is apparent that a single model of driving shaft, a single model of driven shaft, and a single shaft-fixing key could not fit all the pumps in the same family.

In order to remedy this disadvantage and following the principal arrangement of the invention, to obtain various displacements from different pumps in the family, one adjusts at least one of the following parameters: modulus m , number of teeth N , and tooth correction. Maintained constant are tooth width l and interaxial distance E , and variations in the external diameter D_e of the gears are held to a minimum, so that the driving and driven shafts in the same family have dimensions independent of the displacement and that preferably the overall dimensions of the casings of the members of the same family are the same.

It is well known that a standard gear has by definition a pitch diameter D_p and that the modulus (or diametral pitch) m is related thereto by the formula $D_p = mN$, where N is the number of gear teeth. In practice the product mN is kept substantially constant, N being necessarily limited to integral values and m being normalized.

Under these conditions, for two pumps having gears with different moduli: on the one hand, the displacement, which is proportional to the product $2NSl$, varies appreciably because the product NS depends considerably upon m ; and, on the other hand, there are only slight relative variations in the gears' outside diameters D_e , being chiefly dependent upon the pitch diameter D_p , which remains constant.

For different moduli, the interaxial distance E (FIGS. 1 and 2) is kept constant, with the aid of tooth corrections, i.e., by slightly altering the profile of teeth 53. The tooth correction affects chiefly the tooth height h (FIG. 2) and, therefore, the outside diameter D_e .

These tooth corrections can be accomplished alone, independently of variations in modulus or in number of teeth, so as to produce changes in displacement. For example, for the same modulus, a reduction in tooth height reduces cross section S and hence the displacement.

It is obvious that a very slight, substantially constant clearance, whose value is determined by acceptable pump efficiency, must be maintained within bores 6 and 7 at the apex of the teeth. A reduction in tooth height should then be accompanied by a reduction in diameter e of bores 6 and 7, and conversely.

This is not disadvantageous because casing 1 can be obtained starting with identical rough outlines having an extra thickness provided in the region of bores 6 and 7. Final diameter e is obtained during the fine machining of bore 6.

Displacement variations as a function of tooth height (in practice, as a function of diameter e) are rather great and can be continuous between an upper limit and a lower limit.

The rate of change of these variations can be realized upon noting that cross section S (FIG. 2) has substantially the shape of an isosceles trapezoid and that a variation in tooth height amounts to displacing parallel thereto the long side of the trapezoid, which produces a rapid variation in area.

In this case, slight variations in tooth height, and hence slight variations in e , will suffice to bring about appreciable variations in displacement, and the extra thicknesses provided in the regions of bores 6 and 7 can be relatively slight.

The advantages of a family of pumps obtained in this manner are the following.

First, dimension l (the tooth width) remains constant for the same family, and only one model of driving shaft and one model of driven shaft will be enough for all the pumps in the family. The same is true for the affixing keys 50.

Second, interaxial distance E is kept constant and the parts that were common to pumps in the same family, in the case where displacement variations were obtained by variations in l alone, remain so. Particularly, bearings 12 and 12a and sideplates 2 and 2a are common to all the pumps.

Third, the overall dimensions of the casing remain constant; only diameter e of bores 6 and 7 varies. The same rough-outline model of casing will suit all the pumps in the same family; only the fine machining will be different.

These advantages can be rendered more concrete with the assistance of the aforementioned example of a family of pumps whose deliveries per revolution are 10, 12, 14 and 18 cubic centimeters.

To make up all the pumps in the family:

Formerly Required	Now Necessary
4 models of gear (4 different values of l)	2 models of gear in rough outline (plus 4 fine machinings)
4 models of casing	1 model of casing in rough outline (plus 4 fine machinings)
4 models of driving shaft	1 model of driving shaft
4 models of driven shaft	1 model of driven shaft
4 models of affixing key	1 model of affixing key

Indeed the displacements of 10 to 12 cubic centimeters will be obtained with a first modulus and a first number of teeth, as follows: a first model of gear in rough-outline will be finished (diameter D_e) so as to deliver, for example, 12 cubic centimeters per revolution or any intermediate value, and a model of casing will be finished (diameter e) to deliver therefrom 10 or 12 cubic centimeters, per revolution. The displacements of 14 and 18 cubic centimeters will be obtained with a second modulus and a second number of teeth, i.e., a second model of gear in rough outline, but with diameter e modified as a function of the displacement.

It is thus apparent that a family of pumps is obtained to satisfy the original objective and in which the number of parts in common is increased considerably.

Manufacture and assembly of such pumps are accomplished simply, economically, and rapidly, with stocking of separate parts whose variety is reduced.

According to a preferred embodiment, which will be assumed to be applied to a pump such as described hereinabove, and which relates more especially to a return pathway for internal leakages, a direct line (FIG. 4) is provided for such return pathway, comprising: on the one hand, two ducts 54 and 54a bored, respectively, in bearings 12 and 12a in the vicinity of driving shaft 8 and gear 4 carried thereby and which extend substantially perpendicular to the direction of the driving and driven shafts; on the other hand, drain channels 55, 56, and 57 bored through casing 1, enabling aforementioned ducts 54 and 54a to be connected to the zone of low pressure, i.e., the intake zone for a pump.

Channel 57 is coaxial with duct 54a and opens to the outside so as to be connected to the intake zone. Draining of the

leakages by duct 54a is accomplished then in a straight line, as directly as possible.

Channel 55 is coaxial with and of the same diameter as duct 54, which it continues in casing 1. Plug 58 stops the outlet of channel 55 at the outer wall of the casing.

Channel 56 connects channels 55 and 57. Its end that opens against sideplate 2a is stopped by plug 59.

It will be noted that the diameter of channel 57 is greater than that of ducts 54 and 54a because channel 57 receives the leakages flowing out from the two ducts toward the intake. The cross section of channel 57 is substantially equal to the sum of the cross sections of ducts 54 and 54a.

The internal leakages are recovered very close to the sites where they are produced, particularly between gear 4 and bearings 12 and 12a.

The outflow of liquid toward zones 28 and 28a being drained immediately, the pressure in these zones remains very low, and packing gland 21 is well protected against large increases in pressure therein.

These internal leakages, being hot, travel over a short path in the pump (or motor) and lose little heat thereto, so the heating thereof is reduced.

Such a pathway can be especially beneficial when it is applied to an hydraulic motor. For example, in the event of a rapid reversal of the direction of rotation of such a motor, the internal leakages increase considerably and must be drained rapidly; otherwise the pressure can increase drastically in zones 28 and 28a and deteriorate the packing gland. The leakage pathway permits these increases or instantaneous peaks of pressure to be reduced.

Obtained thereby are a clear improvement in performance of the pumps and motors and an increase in their life and their reliability.

It goes without saying and follows moreover from the above that the invention, having been particularly considered, it not at all limited to the foregoing methods of practicing it, no more than to the foregoing ways of embodying it from its vari-

ous parts. Instead, modifications may be made therein while retaining the benefits and without departing from the concept of the invention, which itself is defined in the following claims.

I claim:

1. In a hydraulic gear pump or motor having a casing with an intake and an outlet forming respectively a high- or low-pressure zone, two intermeshing gears therein, a driving shaft with one of the gears affixed thereon, a driven shaft with the other of the gears affixed thereon, rigid bearings flanking the gears and supporting the shafts for rotation on rotary bearings, the improvement comprising at least one interior leakage passageway opening at the side of said rotary bearings nearest the gears including a duct through the rigid bearings between an interior location adjacent the outlet and an interior location adjacent the intake and thereby communicating with said low-pressure zone, each such duct extending substantially perpendicular to the plane passing through the axes of said shafts.

2. Hydraulic volumetric apparatus according to claim 1, constituting a gear motor with rapid reversal of its direction of rotation.

3. Apparatus according to claim 1, wherein each such duct is located adjacent the driving shaft.

4. Apparatus according to claim 3, wherein each such duct is also located adjacent one side of the gear affixed on the driving shaft.

5. Apparatus according to claim 1, wherein such ducts through the respective bearings are joined by drain channels in the casing, including respective drain channels joining the respective ducts and with at least one such drain channel communicating with the low-pressure zone.

6. Apparatus according to claim 5, wherein at least one such drain channel interconnects the respective drain channels joining the respective ducts.

7. Apparatus according to claim 5, wherein the cross section of said drain channel is substantially equal to the sum of the cross sections of said ducts.

* * * * *

40

45

50

55

60

65

70

75