Dec. 11, 1962 G. WIGGERMANN 3,067,691

CONTROLLABLE HYDRAULIC MACHINE OF THE TRIPLE-GEAR TYPE

Filed Dec. 20, 1960 4 Sheets-Sheet 1

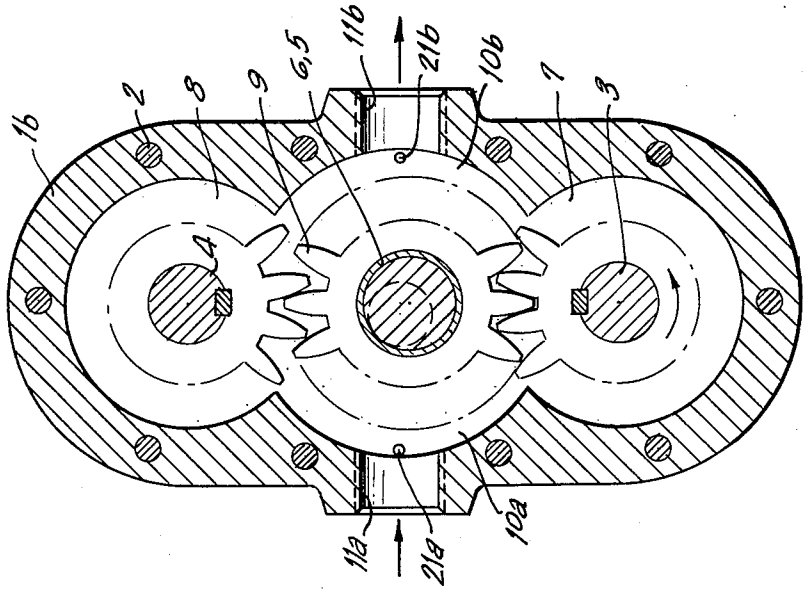


FIG. 2

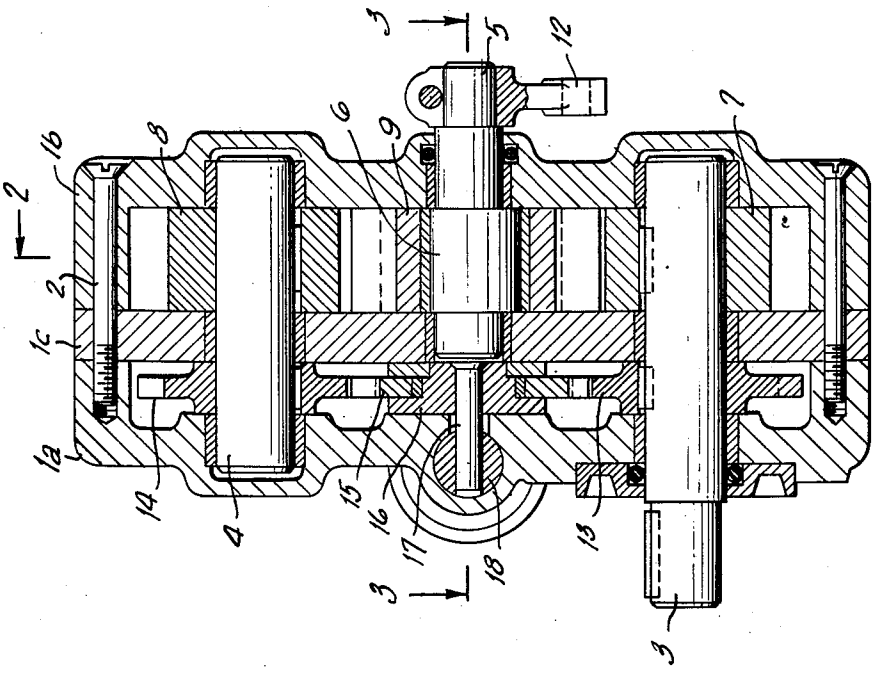


FIG. 1

Dec. 11, 1962 G. WIGGERMANN 3,067,691

CONTROLLABLE HYDRAULIC MACHINE OF THE TRIPLE-GEAR TYPE

Filed Dec. 20, 1960 4 Sheets-Sheet 2

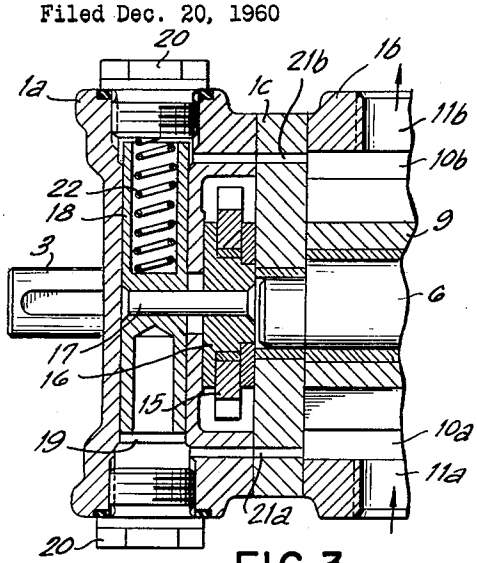


FIG. 3

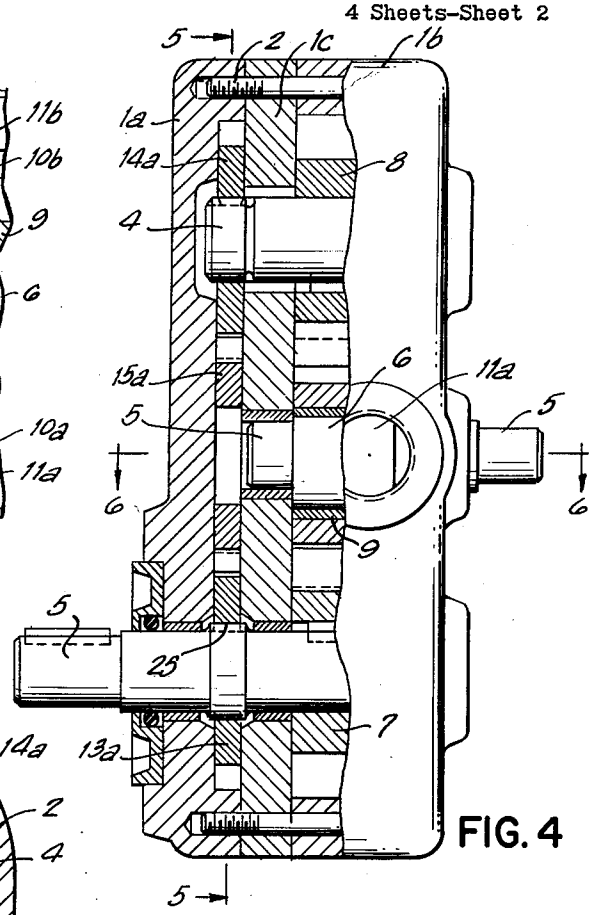


FIG. 4

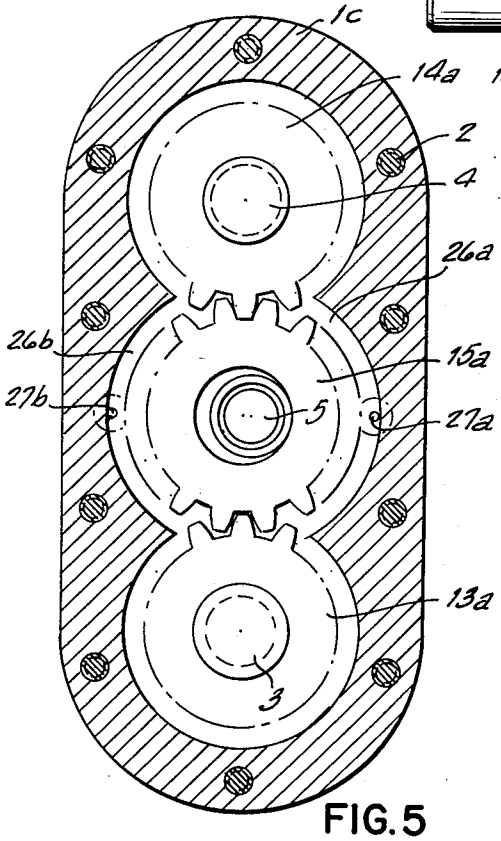


FIG. 5

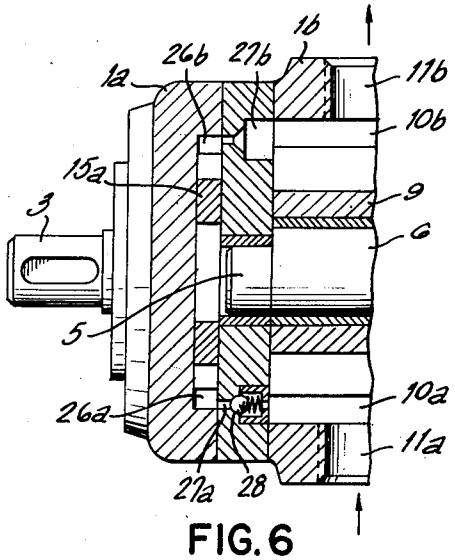


FIG. 6

Dec. 11, 1962 G. WIGGERMANN 3,067,691

CONTROLLABLE HYDRAULIC MACHINE OF THE TRIPLE-GEAR TYPE

Filed Dec. 20, 1960 4 Sheets-Sheet 3

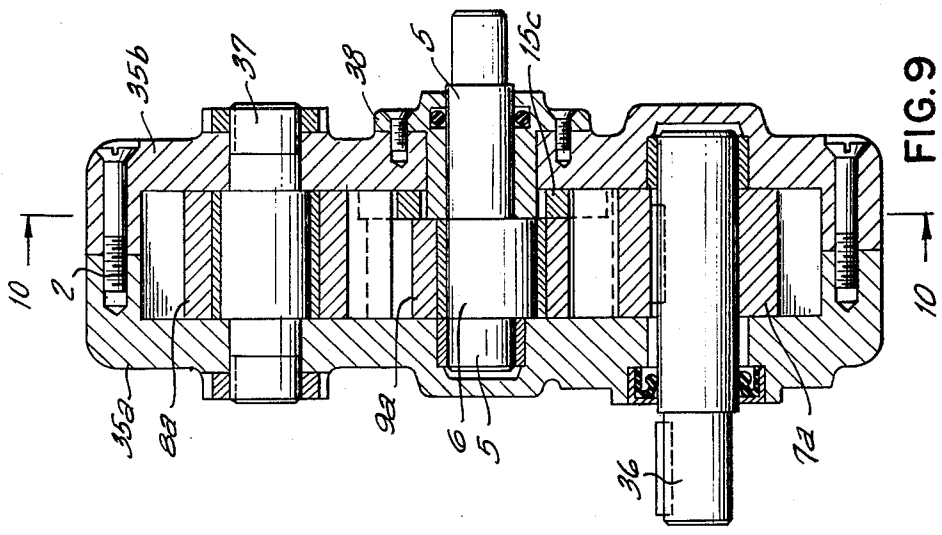


FIG. 9

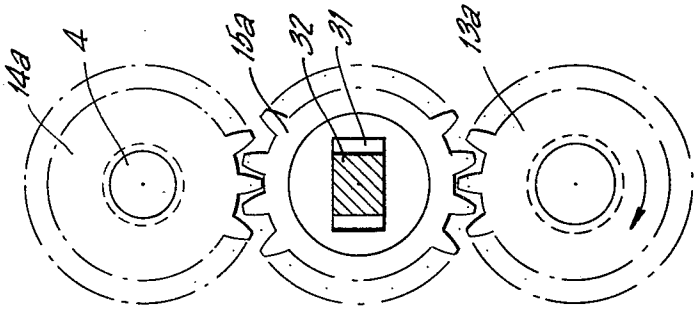


FIG. 8

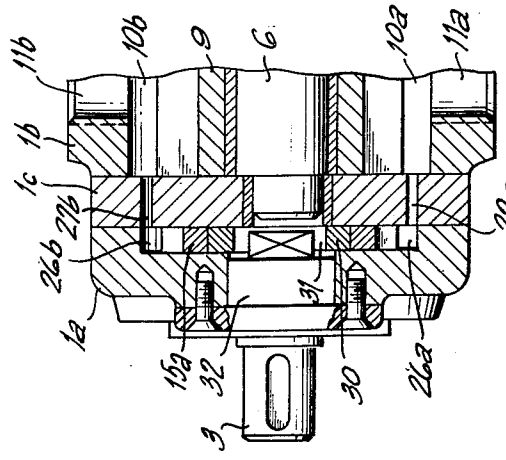


FIG. 7

Dec. 11, 1962 G. WIGGERMANN 3,067,691

CONTROLLABLE HYDRAULIC MACHINE OF THE TRIPLE-GEAR TYPE

Filed Dec. 20, 1960 4 Sheets-Sheet 4

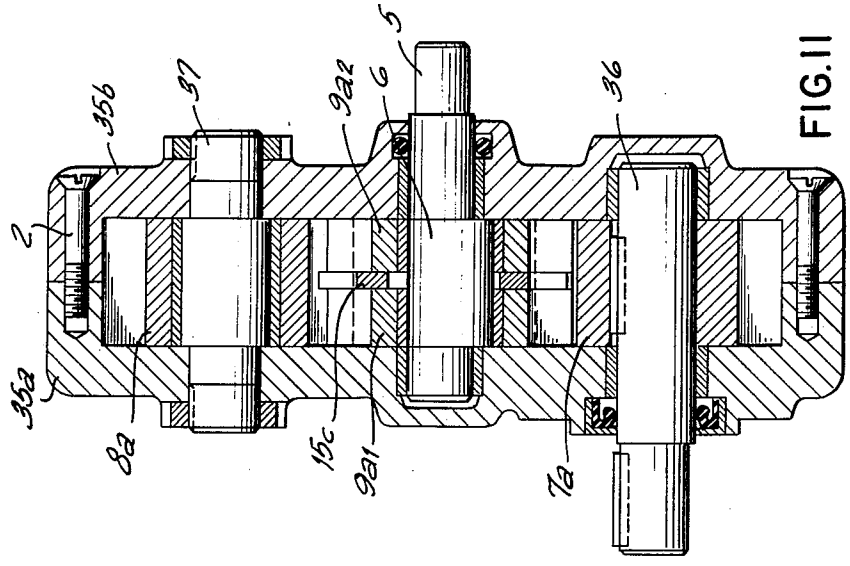


FIG. 11

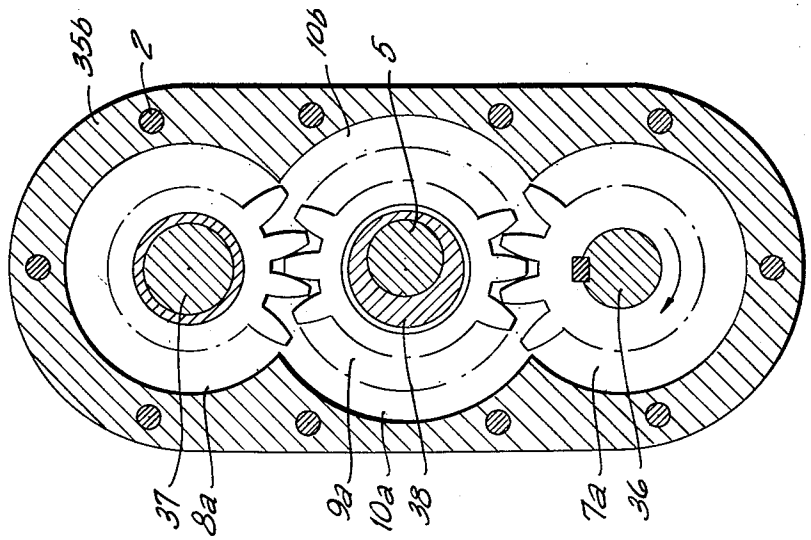


FIG. 10

United States Patent Office 3,067,691 Patented Dec. 11, 1962

3,067,691

CONTROLLABLE HYDRAULIC MACHINE OF THE TRIPLE-GEAR TYPE

Georg Wiggermann, Kressbronn am Bodensee, Germany, assignor to Reiners & Wiggermann Getriebe- und Maschinenbau, Kressbronn am Bodensee, Germany, a German corporation

Filed Dec. 20, 1960, Ser. No. 77,249
Claims priority, application Germany Dec. 21, 1959
14 Claims. (Cl. 103—120)

My invention relates to hydraulic positive-displacement machines suitable as pumps or motors, and more particularly to rotary hydraulic machines in which at least three spur gears form a gear train in a housing and are so journalled that the radial spacing between the axis of the intermediate gear and the respective axes of the two outer gears can be changed within given limits so that an increase in one spacing is accompanied by a decrease in the other. Such change in spacing permits varying the delivery of the machine as to direction and magnitude and can be effected gradually and continuously by means of a control device while maintaining a constant speed of gear rotation.

The invention will be hereinafter described in terms of a pump, although it is also applicable to hydraulic motors.

In a pump of the type here concerned, the volumetric delivery is zero when the axis of the intermediate gear is equally spaced from the two outer gears. This setting is hereinafter called "zero position." Starting from the zero position, the control device of the pump can be shifted in two directions, each direction determining the delivering direction of the pump for a given direction of gear rotation. As a rule, the suction space of the pump is the one where the teeth of the gear with the smaller radial spacing from the intermediate gear are located. Conversely, the other hollow space in the housing where the teeth of the gear having the greater radial spacing are moving, is active as the pressure space. The total control range of the control device, starting from the zero position, extends continuously from zero delivery up to a maximum determined by the kinematics of the meshing engagement, and also extends from zero delivery up to a maximum in the other delivering direction of the pump.

Unfortunately, however, the utilization of this inherently ideal control possibility encounters difficulties of such severity that the practical use of this type of pump has heretofore not been feasible. This is because the delivering pressure p (p=difference between the pressures p_1 and p_2 in the two pump spaces) subjects the three gears of the train to hydrostatic counter torques which reduce the meshing pressure between the teeth of the intermediate gear and those of the outer gear having the smaller radial spacing. This reduction in pressure goes beyond the minimum required for hydraulic separation of the suction space from the pressure space. Consequently, the counter torque causes a separation of the mutually engaging tooth flanks between the intermediate gear and the outer gear of smaller radial spacing, thus resulting in a hydraulic short-circuit between pressure and suction space. Since gear pumps of this type require a relatively great clearance between the teeth of each two intermeshed gears, the short-circuit connection can assume such a large cross section that the delivering pressure may instantaneously drop almost to zero. When this occurs, the above-mentioned detrimental hydrostatic torque is eliminated, the original engagement between the intermeshing tooth flanks is re-established, and the delivering pressure is built up anew. This, however, again causes the disturbing hydrostatic torque to build up until the intermeshing gear teeth are again separated from each other. This repeats itself in a rapid sequence, with the above-mentioned result that heretofore such pumps have been unsuitable in practice.

Investigation of the above-described phenomena has shown that the disturbing flank separation always occurs at the one outer gear whose teeth, meshing with the intermediate gear, are traveling toward the pressure space at a given direction of gear rotation. It follows that the detrimental flank separation for a uniform direction of gear rotation, takes place at one or the other outer gears depending upon the delivering direction of the pump.

It is an object of my invention to eliminate the above-described shortcomings and to thereby improve such controllable triple-gear type machines to such an extent that their inherently favorable control and regulating properties can advantageously be utilized in practice.

To this end, and in accordance with a feature of my invention, the two outer gears of the triple-gear pump are coupled with each other by an auxiliary coupling mechanism so designed that the instantaneous delivering pressure p ($=p_2-p_1$) automatically superimposes upon the two outer gears a torque which is proportional to the delivery pressure and whose rotational sense is different at the respective two outer gears, namely so that an auxiliary torque always acts upon the one outer gear whose meshing teeth travel toward the pressure space.

According to another, more specific feature of my invention, the auxiliary mechanism consists of a second train of three spur gears whose two outer gears are joined with the two outer gears of the triple-gear pump proper to rotate together therewith, whereas the intermediate gear of the auxiliary mechanism is guided in the pump housing so as to be displaceable radially and approximately in the direction of the bisecting line of the centering angle formed relative to its rotation axis by the meshing points of the teeth.

According to still another feature of my invention, the auxiliary torques are produced by having the delivering pressure p act upon a piston which is coupled through mechanical means with the displaceable bearing of the intermediate gear in the auxiliary mechanism.

According to another feature of my invention, however, the delivery pressure may also be used to act directly upon the auxiliary intermediate gear in the desired radial direction. For this purpose, the auxiliary intermediate gear is guided between planar parallel surfaces with a tight running fit, and the two partial peripheral surfaces of the auxiliary intermediate gear extending from one to the other tooth-meshing location are subjected to the pressure p_1 and to the pressure p_2 respectively. As a result, the displaceable intermediate gear of the auxiliary mechanism is radially subjected to a force proportional to the product defined by the delivery pressure p times the area defined by the pitch-circle diameter and tooth width of the auxiliary intermediate gear.

According to more specific features of my invention, subsidiary to those last mentioned, the three spur gears of the auxiliary mechanism are guided outside of the pumping space proper between planar surfaces of a further hollow space provided within the pump housing especially for the auxiliary mechanism. However, the auxiliary mechanism may also be located in the pump space proper. For this purpose, the two outer gears of the main train are preferably made wider by the width of the auxiliary gears, and the auxiliary intermediate gear, being in continuous meshing engagement with the widened portions of the respective outer gears, is guided for radial displacement without any particular bearing means, exclusively by the fact that its teeth mesh with the outer gears of the train, and is guided axially between the two planar and parallel faces of the inner housing wall and the ad-

jacent front face of the main intermediate gear that forms part of the main train.

The foregoing and more specific objects and features of my invention will be described in the following with reference to the embodiments of controllable and reversible triple-gear pumps according to the invention illustrated by way of example on the accompanying drawings in which:

FIG. 1 is a longitudinal section through a pump.

FIG. 2 is a cross section of the same pump along the line A—B in FIG. 1.

FIG. 3 is a horizontal cross section along the line C—D in FIG. 1.

FIG. 4 is a longitudinal section through another pump.

FIG. 5 is a vertical cross section along the line A—H in FIG. 4.

FIG. 6 is a horizontal cross section along the line E—F in FIG. 4.

FIG. 7 is a horizontal longitudinal section taken at the height of the eccentric shaft on which the intermediate main gear is mounted in a pump as shown in the preceding illustrations.

FIG. 8 shows separately a front view of the triple-gear auxiliary mechanism in accordance with the embodiment of FIG. 7.

FIG. 9 is a longitudinal section through still another pump.

FIG. 10 is a vertical section through the pump according to FIG. 9.

FIG. 11 is a longitudinal section through a further pump.

In all illustrations functionally similar components are denoted by the same respective reference numerals.

The machine illustrated in FIGS. 1, 2 and 3 has a housing composed of two outer portions 1*a*, 1*b* and an intermediate partitioning portion 1*c*. The three housing components are held together by screw bolts 2. Journalled in the housing are a drive shaft 3, a counter shaft 4, and a control shaft 5 which carries an eccentric 6. Located in the pumping space between housing portion 1*b* and partition 1*c* is a gear 7 mounted on the drive shaft 3 and rotating together therewith. A corresponding gear 8 is mounted on the counter shaft 4 and keyed together therewith. An intermediate spur gear 9 is rotatably seated on the eccentric 6. The three gears 7, 8 and 9 form together a triple-gear train. The outer gears 7, 8 are closely surrounded by the housing portion 1*b* along part of the gear periphery with a tight running fit. The intermediate gear 9 is radially surrounded by hollow spaces 10*a* and 10*b*. All three gears fit accurately between the planar surface of the partition 1*c* and the inner planar surface of the housing portion 1*b*. This, together with the sealing engagement between the flanks of the intermeshing teeth between the intermediate gear and the respective outer gears, has the effect of hydraulically separating the two hollow spaces 10*a* and 10*b* from each other to serve as pressure and suction space of the pump. The housing portion 1*b* is provided with threaded bores 11*a* and 11*b* which communicate with the hollow spaces 10*a* and 10*b* respectively and serve for connection of conduits or other hydraulic lines to the pump.

The gears 7, 8 and 9 are provided with involute teeth which afford a large variation in radial spacing between the respective axes of each two intermeshing gears. The change in mutual spacing, required for varying the delivering quantity and the delivering direction, is effected by turning the control shaft 5. To this end, a control arm 12 is clamped fast on the end of shaft 5. In the midposition of the eccentric 6 and of the control arm 12, the two outer gears have their respective axes equally spaced from the rotational axis of the intermediate gear. This constitutes the zero setting of the pump in which no delivery takes place. When the control shaft 5 is turned clockwise (FIG. 2), the intermediate gear 9 is displaced approximately along the vertical center line indicated in FIG. 2 in the downward direction. When the control shaft 5 is turned counterclockwise, the intermediate gear 9 moves substantially along the vertical center line in the upward direction. In the former case, the radial spacing of intermediate gear 9 from the axis of the outer gear 7 is reduced, whereas the corresponding spacing of outer gear 8 is increased. When turning the control shaft 5, starting from the zero position, in the counterclockwise sense, the respective radial spacings are changed in the inverse sense.

When the driven spur gear 7 rotates counterclockwise as is indicated by an arrow in FIG. 2, the clockwise turning of the control shaft 5 results in the delivering direction indicated by arrows at the inlet and outlet bores 11*a* and 11*b* (FIG. 2). When the control shaft 5 is turned counterclockwise, the delivering direction is reversed. In both cases, the delivering quantity increases with increasing angular displacement of the control shaft 5 and reaches a maximum when the radial displacement of the intermediate gear reaches the limit value determined by the geometry of the gear teeth.

Located in the hollow space between the housing portion 1*a* and the partition 1*c* is a spur gear 13 keyed to the drive shaft 3, another spur gear 14 keyed to the counter shaft 4, and an intermediate spur gear 15 meshing with respective gears 13 and 14. The intermediate gear 15 is radially and axially guided by the two meshing engagements in conjunction with a displaceable bearing 16. The bearing 16 has two planar and parallel faces in sliding fit with corresponding planar faces of the housing portion 1*a* and the partition 1c respectively. Thus, the bearing 16 is displaceable in a direction parallel to the plane of the intermediate gearing 9. The displaceable bearing 16 is connected with a piston 18 by a transverse pin 17. The piston 18 is displaceable in a cylinder bore 19 (FIG. 3) of the housing portion 1 extending transverse to the axis of the pin 17. The two ends of the cylinder bore 19 are closed by screw stoppers 20 and communicate hydraulically with the hollow spaces 10*a* and 10*b* of the housing portion 1*b* through respective ducts 21*a*, 21*b* in housing portion 1*a* and partition 1*c*.

The spur gears 13, 14 constitute the respective outer gears, and the spur gear 15 the intermediate gear of a gear train which, in conjunction with the cylinder and piston, constitutes an auxiliary mechanism for the purpose of exerting the above-mentioned auxiliary torques upon the respective main outer gears 7 and 8 of the pump proper. Depending upon the operating condition of the pump, the auxiliary torque is imposed upon the one main outer gear 7 or 8 whose meshing teeth travel toward that hollow space 10*a* or 10*b* which is under the higher pressure p_2. This auxiliary torque has the magnitude and rotational sense required for compensating the hydrostatic counter torque simultaneously acting upon this particular one outer gear. As a result, a lifting of the tooth flanks of the outer gear from those of the intermediate gear 9 is prevented, thus securing satisfactory performance of the pump.

As mentioned, with the rotational direction of the drive shaft 3 indicated in FIGS. 1 and 2, the pump delivery is in the direction indicated in FIG. 2 by arrows near the inlet 11*a* and outlet 11*b*. Accordingly, under these conditions the low pressure p_1 obtains in the hollow space 10*a*, and the high pressure p_2 in the space 10*b*. As explained above, under these conditions, the detrimental hydrostatic driving torque occurs at the outer gear 8 and tends to drive this gear in the prevailing direction of rotation, whereby the teeth of gear 8 lose the illustrated meshing engagement with the intermediate gear so that a hydraulic short circuit tends to occur between the pressure space 10*a* and the suction space 10*b*. However, the pressures p_1 and p_2 are transmitted through the ducts 21*a*, 21*b* to the respective sides of the control piston 18. The piston thus becomes loaded with the resultant delivering pressure $p=p_2-p_1$ and is displaced downwardly (relative

to FIG. 3) by a force $P=p\cdot Fk$ (F denotes the area of the piston, k denotes a constant). The displaceable bearing **16** connected with the piston **18** by the cross pin **17** then causes the auxiliary intermediate gear **15** to likewise shift downwardly. Consequently, the teeth of gear **15** brace themselves against the teeth of the auxiliary outer gears **13**, **14** and transmit to them mutually opposed auxiliary torques which are proportional to the instantaneous delivery pressure p. As a consequence, the auxiliary outer gear **14** is driven clockwise with the effect that the detrimental hydrostatic torque, acting in the direction of the arrow, is compensated, provided the diameter of piston **18** is properly chosen. The hydrostatic torque, which under these conditions (p_2 in space **10***b*), also acts upon the outer gear **7**, is not detrimental because under the operating conditions here in view this torque is opposed to the driving torque. The auxiliary torque transmitted to gear **7** through the auxiliary gear mechanism is likewise not detrimental for the same reason.

When in the embodiment of FIGS. 1 to 3, while maintaining the same direction of rotation, the eccentric **6** (FIG. 2) is placed into the midposition (zero position) and is then further turned upwardly, the delivering quantity at first declines down to zero and then increases in the direction opposed to that indicated by the arrows. In this case the pressures p_1 and p_2 obtain in the respective spaces **10***b* and **10***a*. Under such conditions the piston **18** is shifted upwardly (FIG. 3), and auxiliary torques are transmitted to the gears **7**, **8** with the effect of compensating the detrimental hydrostatic torque occurring at the gear **7**.

It will be recognized that the compensating effect to be performed by the auxiliary mechanism takes place for any direction of rotation and any direction of delivery. Relative to this effect it is of interest that in pumps according to the invention, namely the one described above as well as those described hereinafter, the intermeshing gear teeth, for a given direction of rotation, always engage each other at those flanks which also transmit driving torque when the pump runs idle.

According to another feature embodied in the pump according to FIGS. 1 to 3, the piston **18** has a bore extending from one of its ends axially inwardly and accommodating a helical compression spring **22** (FIG. 3) which abuts against the screw stopper **20** and loads the piston **18** so that a certain degree of loading is imposed upon the intermeshing tooth flanks already when operating without counter pressure. The spring loading of piston **18** may also serve to compensate certain unbalances between the hydrostatic torque that tends to disengage the intermeshing teeth and the auxiliary torque serving to compensate the detrimental torque. For this purpose, the tension of the spring **22** is preferably made adjustable, for example by inserting shim plates between the stopper **20** and the spring, or by providing an adjusting screw (not illustrated).

The pump portion of the embodiment illustrated in FIGS. 4, 5 and 6 is identical with the one described above with reference to FIGS. 1 to 3, with the only exception of the auxiliary mechanism. In addition to the pump mechanism proper, the machine is provided with a triple-gear train located in a recess of housing portion **1***a*. The additional gear train comprises outer gears **13***a*, **14***a* and an intermediate gear **15***a* meshing with the respective outer gears. The outer gear **13***a* is joined with the drive shaft **3** by means of key teeth **25**, and the outer gear **14***a* is joined by teeth with the counter shaft **4**. The intermediate or central gear **15***a* is guided in the radial direction by the meshing engagement of its teeth with the respective outer gears **13***a* and **14***a*. All three gears of the auxiliary mechanism are guided axially at their respective planar front faces by corresponding planar faces of the housing portion **1***a* and the partition **1***c* with a slight running fit. The two outer gears **13***a* and **14***a* are surrounded with a slight radial running clearance by the housing portion **1***a* over a portion of the gear periphery. The intermediate gear **15***a* is surrounded by hollow spaces **26***a* and **26***b* which correspond to the respective hollow spaces **10***a* and **10***b* of the housing portion **1***b*. The hollow spaces **26***a* and **26***b* communicate hydraulically through ducts **27***a*, **27***b* in partition **1***c* with the adjacent respective spaces **10***a* and **10***b* of the pump portion.

Due to the tight enclosure around the outer gears **14***a*, **13***a* and the double meshing engagement of the intermediate gear, the two hollow spaces **26***a* and **26***b* are hydraulically separated from each other. The pressures p_1 and p_2 of the medium in the respective spaces **10***a* and **10***b* are imposed upon one or the other side of the intermediate gear **15***a*. Consequently, the intermediate gear is subjected to a force corresponding to the product $p\cdot Dt\cdot B$ which acts in the radial plane of the intermediate gear **15***a* and transverse to its axis. As a result, the intermediate gear produces the auxiliary torque required for compensating the detrimental hydrostatic torque at outer gear **7** or **8**, this effect being identical with the one more fully explained above with reference to FIGS. 1 to 3. The duct **27***a* is provided with a check valve **28** (FIG. 6) whose purpose will be explained below.

FIGS. 7 and 8 illustrate the possibility, relating to the embodiment of FIGS. 4 to 6, to provide the intermediate gear **15***a* of the auxiliary mechanism with a guide in the vertical direction. According to another feature of the invention, the centerbore of the auxiliary intermediate gear **15***a* is filled by a bearing plate **30** which is rotatable relative to the bore. The bearing plate **30** has a rectangular opening **31** engaged by a rectangular piece **32** which is rigidly joined with the housing portion **1***a* by threaded engagement. The rectangular piece **32** prevents vertical motion as well as rotation of the bearing plate **30** but permits a transverse displacement of the bearing plate and thus also of the intermediate gear **15***a*. In all other respects the operation of the auxiliary mechanism is identical with the one illustrated in FIGS. 1 to 6 and described in the foregoing.

Instead of fastening the rectangular piece **32** in the housing portion **1***a*, it may be made a component of another eccentric control shaft, journalled beneath the horizontal center line of the auxiliary intermediate gear **15***a*, so that turning of the shaft causes the rectangular piece **32** to become laterally displaced in the rectangular hole **31**, thus displacing the bearing plate **30** in the transverse direction. Such setting of the rectangular piece **32** to a desired position makes it possible to somewhat modify the action of the auxiliary mechanism if necessary or desirable. The latter modification according to the invention is not illustrated because it is readily understandable from the foregoing explanations relating to the embodiment of FIGS. 4 to 8.

The pump illustrated in FIGS. 9 and 10 has a housing **35** composed of two portions **35***a*, **35***b* fastened together by bolts **2**. Journalled in bores of the housing is the drive shaft **36**, the control shaft **5** with the eccentric **6**. Firmly joined with the housing by screw connections is a bearing shaft **37**. Mounted on the drive shaft **36** is a gear **7***a* which is keyed to the shaft to rotate together therewith. Rotatably journalled on the eccentric and on the fixed shaft **37** are respective spur gears **9***a* and **8***a*. The three gears **7***a*, **8***a* and **9***a* form a triple-gear train whose intermediate gear **9***a* is in continuous meshing engagement with the two outer gears **7***a*, **8***a*. By turning the control shaft **5** the intermediate gear is caused to shift radially in a direction approximately along the vertical center line (FIG. 10). The effect of such displacement corresponds to that described above with reference to the embodiments of the preceding illustrations. The housing extends with slight running fit around the two outer gears **7***a*, **8***a* along part of their respective peripheries. All three gears of the train have their respective planar front faces adjacent to corresponding planar inner surfaces of the housing, the clearance being just sufficient for a running fit. The intermediate gear **9***a* is surrounded by two hollow spaces **10***a*, **10***b*, which are hydraulically sealed from each other by the

double meshing engagement of the intermediate gear 9*a*, provided its tooth flanks are in pressure engagement with the respective tooth flanks of the outer gears.

The auxiliary mechanism provided according to the invention for securing a continuous flank engagement of the gear teeth is greatly simplified in this embodiment by being included within the pumping space proper. The two outer gears of the auxiliary mechanism are simply formed by laterally enlarged portions of the two outer gears 7*a*, 8*a*. The intermediate gear 15*c* of the auxiliary mechanism, however, consists of a separate part. It is located directly beside the main intermediate gear 9*a* but is not prevented from radial displacement except by its two meshing engagements with the outer gears 7*a*, 8*a* which, as explained, have fixed axes of rotation. The center bore of the auxiliary intermediate gear 15*c* surrounds a flange bearing 33 with some radial clearance. The number of teeth on the intermediate gear 15*c* is identical with that of the main intermediate gear 9*a*. However, the gear teeth of the auxiliary intermediate gear 15*c* mesh with the teeth of the outer gear 7*a*, 8*a* substantially without clearance in the radial direction, this being readily obtainable under the journalling conditions illustrated in FIGS. 9, 10 and described above. The housing 35*a* of the pump is divided along a plane of division passing through the center of the gear width.

The performance of this pump is identical with that described above with reference to FIGS. 4, 5 and 6. This will readily be understood if one imagines in FIGS. 4, 5 and 6 the partition 1*c* to be removed and the spaces 26*a*, 26*b* substituted by the respective spaces 10*a*, 10*b*. Under such conditions, the pressures p_1 and p_2 in respective spaces 10*a* and 10*b* are directly imposed upon the intermediate gear 15*c* in the transverse direction on both sides of its two meshing engagements. As a result, the intermediate gear 15*c* is loaded by the difference pressure $p=p_2-p_1$, whereby the direct meshing engagement of the intermediate gear with the two outer gears imposes upon these outer gears the auxiliary torque required for compensating the detrimental hydrostatic torque.

In a pump according to FIGS. 9 and 10, it may occur, particularly at high rotating speeds, that the lateral escape of oil from the meshing range of the gears becomes troubled. This may increase the pinching pressure imposed upon the oil thereby increasing the operating noise and reducing the efficiency.

This troubling effect comes about, in the first place, by the fact that the intermediate gear 15*c* of the auxiliary mechanism engages the teeth of the outer gears 7*a*, 8*a* with only slight clearance and thereby offers a considerably reduced flow cross section, in comparison with the meshing engagement of the main intermediate gear 9*a*, to the flow of oil laterally squeezed out of the meshing space.

Furthermore, the virtually clearance-free meshing engagement of the auxiliary intermediate gear 15*c*, in conjunction with the fact that the meshing engagement of the main intermediate gear 9*a* possesses considerable clearance, causes the occurrence of phase displacements between the teeth of the respective intermediate gears 9*a* and 15*c*. Such phase displacement varies in dependence upon the particular position of the eccentric 6 and likewise promotes a throttling of the lateral escape of oil from the meshing range of the intermediate gear 9*a*.

In order to prevent the auxiliary intermediate gear 15*c* from thus impeding the escape of squeezed oil from the housing-adjacent front faces of the pump gears, the pump is preferably provided with remedial means, according to another feature of my invention described presently.

According to FIG. 11, the main intermediate gear of the pumping train is divided into two component gears $9a_1$, $9a_2$, and the auxiliary intermediate gear 15*c* is located between the two component main gears $9a_1$ and $9a_2$. The axial width of the three gears, $9a_1$, 15*c*, $9a_2$ is so chosen that the entire set of gears is readily rotatable between the two planar and parallel inner surfaces of the housing but is still sufficiently sealed at the front faces. In order to make the position of the auxiliary intermediate gear 15*c* independent of the turning motion of the eccentric 6, the middle bore of gear 15*c* is so large that its inner periphery never enters into contact with the eccentric.

The performance of the above-described embodiments does not depend upon which of the three gears of the pump is driven from the external source of power. It is, however, preferable, as exemplified by the embodiments, to always apply the driving power to one of the two outer gears of the triple-gear train. The reason for this preference is in accordance with the concepts more fully explained in my copending application Serial No. 57,797, filed September 22, 1960, entitled Hydraulic Gear-Type Machine of Controllable Displacement. As explained in the copending application, when the pump is driven at one of the two outer gears, the tooth-flank separating effect can occur only at the other outer gear, namely the one that is driven only by its meshing engagement with the intermediate gear. Furthermore, such flank separation can only take place when the delivering direction of the pump is such that the radial spacing of the intermediate gear from the axis of the externally driven outer gear is smaller than from the axis of the other outer gear. By applying this recognition to the novel pumps according to my present invention as described above, it will be understood that when one of the two outer gears is driven and the delivering direction is such that the driving outer gear has a smaller radial spacing from the intermediate gear axis than the other outer gear, the stabilizing action of the auxiliary mechanism can be dispensed with.

Consequently, according to further features of the present invention, the auxiliary mechanism is made ineffective for one given direction of delivery and is then preferably also relieved of its load. This reduces the wear of the mechanism and somewhat increases the mechanical efficiency of the pump for the proper delivery direction. In practice, the pump can then be operated in many cases so that it preferably operates with the favorable direction of delivery.

In this sense, and in accordance with a more specific feature of my invention, the transverse load imposed upon the auxiliary intermediate gear 15 at the one proper delivering direction is blocked by mechanical or hydraulic means. For example, in the pump according to FIG. 3 the piston 18 has such a length on its side adjacent to the spring 22 that the piston will already abut against the stop 20 when in a position at which the intermediate gear 15 of the auxiliary mechanism still engages the teeth of the outer gears 13, 14 with some amount of clearance (such as the clearance apparent from FIG. 8). This provides for the above-mentioned mechanical blocking.

An example of a hydraulic blocking mechanism is apparent from the section shown in FIG. 6 along the line VI—VI of FIG. 4. In this pump, one of the ducts 27 is provided with a check valve which closes automatically when medium flows from the space 10*a* to space 26*a*. The intermediate gear 15*a* can be effectively loaded only when the delivering direction coincides with the one indicated by arrows at the inlet and outlet 11*a*, 11*b* (FIG. 2).

The above-described devices for blocking the effect of the auxiliary mechanism relative to one given direction of delivery (at which the driving outer gear 7 has a greater radial spacing from the axis of the intermediate gear than the other outer gear), limit the pumps to only one given direction of rotation. However, according to another feature of my invention, the pumps can be operated in both directions without foregoing the desired blocking action. Accordingly, the blocking means are so designed that they can readily be inserted selectively to effect blocking in the other direction. Thus, for example, the piston 18 (FIG. 3) can be inserted into the cylinder bore 19 in the inverse position, with the result that the user of the pump can adjust it to one or the other direction of delivery. Analogously, the check valve 28 according to

FIG. 6 can also be inserted into the duct 27*b* whose design is suitable for this purpose, as shown in FIG. 6.

In the gear pumps described above, the teeth of each gear abut against those of the adjacent gear only along one flank. Therefore the cost of manufacturing such pumps can be decreased by subjecting only the active flank of the teeth to precision-finishing, while leaving the other flank in more or less unfinished condition.

Pumps according to the invention satisfy all practical requirements to such a satisfactory extent that they eliminate the obstacles which heretofore prevented the introduction of triple-gear hydraulic machines into actual practice.

Machines according to the invention are also suitable for operation as a hydraulic motor in one or both directions of rotation. It will also be obvious to those skilled in the art that my invention permits of various modifications other than particularly illustrated and described herein, without departing from the essential features of my invention and within the scope of the claims annexed hereto.

I claim:

1. A hydraulic machine comprising a housing having inlet and outlet ducts for hydraulic medium, a train of spur gears comprising an intermediate gear and two outer gears all mounted for rotation in said housing, said two outer gears meshing with said intermediate gear, said intermediate gear and said housing forming together two spaces located on opposite sides of said intermediate gear and hydraulically separated from each other by the meshing engagement of said respective outer gears with said intermediate gear and communicating with said respective ducts, control means for radially displacing said intermediate gear relative to said outer gears so that increase in radial spacing of the intermediate-gear axis from one outer gear is accompanied by reduction in radial spacing from the other outer gear, an auxiliary transmission intercoupling said two outer gears for imposing upon said outer gears mutually opposed auxiliary torques respectively, said transmission having torque control means responsive to the pressure difference of said two spaces for controlling said auxiliary torques approximately in proportion to the delivery pressure of the machine, the controlling sense of said torque control means being the one required for the auxiliary torque to be opposed to the prevailing driving direction at the one outer gear whose teeth travel toward the spaces having higher pressure than the other.

2. A hydraulic machine according to claim 1, comprising blocking means responsive to the delivery direction of the machine and operatively connected with said auxiliary transmission to prevent its torque-producing operation for a given delivery direction.

3. A hydraulic machine comprising a housing having inlet and outlet ducts for hydraulic medium, a main train of spur gears comprising an intermediate gear and two outer gears all mounted for rotation in said housing, said two outer gears meshing with said intermediate gear, said intermediate gear and said housing forming together two spaces located on opposite sides of said intermediate gear and hydraulically separated from each other by the meshing engagement of said respective outer gears with said intermediate gear and communicating with said respective ducts, control means for radially displacing said intermediate gear relative to said outer gears so that increase in radial spacing of the intermediate-gear axis from one outer gear is accompanied by reduction in radial spacing from the other outer gear, an auxiliary gear train comprising two auxiliary gears joined with said respective outer gears of said main train to rotate together therewith, said auxiliary train further comprising a central gear meshing with the other two auxiliary gears and displaceable relative to said housing in a radial direction approximately perpendicular to a line through the centers of said respective auxiliary gears, and displacement control means responsive to the pressure difference of said two spaces for displacing said central gear substantially proportionally to the delivery pressure of the machine in the direction required for imposing a torque opposed to the prevailing driving direction upon the one outer gear whose teeth travel toward the space having higher pressure than the other, whereby the tendency of said one outer gear to lift its teeth from those of said intermediate gear is eliminated.

4. In a hydraulic machine according to claim 3, said housing having a partitioning structure to form two enclosed machine spaces, said two gear trains being mounted in said two machine spaces respectively, each of said machine spaces having planar and parallel inner walls, and the three gears of each train having planar faces in sealing engagement with the two planar walls of the respective machine spaces.

5. In a hydraulic machine according to claim 3, said housing having an enclosed undivided machine space, and both of said two trains of gear being mounted in said machine space.

6. A hydraulic machine according to claim 3, comprising a bearing plate on which said central gear of said auxiliary gear train is rotatable, said bearing plate having a rectangular opening, and a rectangular guide structure engaging said opening and fastened to said housing for permitting said central gear to be displaced only in said radial direction perpendicular to the center line of said auxiliary gear train.

7. In a hydraulic machine according to claim 3, said housing having an enclosed undivided machine space, and both of said two trains of gears being mounted in said machine space, said auxiliary gears consisting of laterally extended portions of said respective outer gears to form integral gear units together therewith, said central gear having two planar lateral faces, said housing and said intermediate gear having respective planar faces in sealing engagement with said respective planar faces of said central gear, whereby said central gear is guided radially by its meshing engagement with said respective gear units and axially by said respective planar faces of said housing and said intermediate gear.

8. In a hydraulic machine according to claim 3, said intermediate gear of said main gear train comprising two component gears axially spaced from each other, and said central gear of said auxiliary gear train being mounted between said two component gears.

9. A hydraulic machine according to claim 3, comprising check means operatively associated with said displacement control means and responsive to a given delivery direction of the machine to then limit the radial displacement of said central gear.

10. In a hydraulic machine according to claim 3, said displacement control means comprising a cylinder and a piston movably disposed in said cylinder and dividing it into two cylinder chambers, said two chambers communicating with said respective two spaces so as to be subjected to the respective pressure of said spaces, said central gear having a bearing, and said piston being mechanically connected with said bearing for displacing it in said radial drection.

11. In a hydraulic machine according to claim 10, said piston having a central bore in one axial end portion, a compressive spring seated in said bore and abutting against said cylinder for loading said piston to subject said gear teeth to meshing force under no-load conditions of the machine.

12. In a hydraulic machine according to claim 11, said piston on the side of said spring being in abutting engagement with said housing in a position where said central gear is still in meshing engagement with a given clearance relative to said two other gears of said auxiliary gear train.

13. A hydraulic machine comprising a housing having inlet and outlet ducts for hydraulic medium, a main train

of spur gears comprising an intermediate gear and two outer gears all mounted for rotation in said housing, said two outer gears meshing with said intermediate gear, said intermediate gear and said housing forming together two spaces located on opposite sides of said intermediate gear and hydraulically separated from each other by the meshing engagement of said respective outer gears with said intermediate gear and communicating with said respective ducts, control means for radially displacing said intermediate gear relative to said outer gears so that increase in radial spacing of the intermediate gear axis from one outer gear is accompanied by reduction in radial spacing from the other outer gear, an auxiliary gear train comprising two auxiliary gears joined with said respective outer gears of said main train to rotate together therewith and further comprising a central auxiliary gear meshing with the other two auxiliary gears and being displaceable relative to said housing in a radial direction approximately perpendicular to a line through the centers of said respective auxiliary gears, said central gear of said auxiliary gear train having two peripheral ranges extending between the two meshing engagements of said auxiliary gears on the two sides of said central gear respectively, said two regions being in communication with said

respective two spaces for displacing said central gear substantially proportionally to the delivery pressure of the machine in the direction required for imposing a torque opposed to the prevailing driving direction upon the one outer gear whose teeth travel toward the space having higher pressure than the other, whereby the tendency of said one outer gear to lift its teeth from those of said intermediate gear is eliminated.

14. A hydraulic machine according to claim 13, comprising two passages through which said two regions adjacent to the periphery of said central gear communicate with said two spaces respectively, and a check valve disposed in one of said two passages so as to close said passage in response to return flow of hydraulic medium from one of said spaces to one of said regions.

References Cited in the file of this patent

UNITED STATES PATENTS

2,457,465	Grosser	Dec. 28, 1948
2,549,241	Rorive	Apr. 17, 1951
2,754,765	Joy	July 17, 1956

FOREIGN PATENTS

1,195,288	France	May 19, 1959