Jan. 17, 1961 G. WIGGERMANN 2,968,286

HYDRAULIC AXIAL-PISTON MACHINE

Filed May 23, 1957 4 Sheets-Sheet 1

FIG. 1

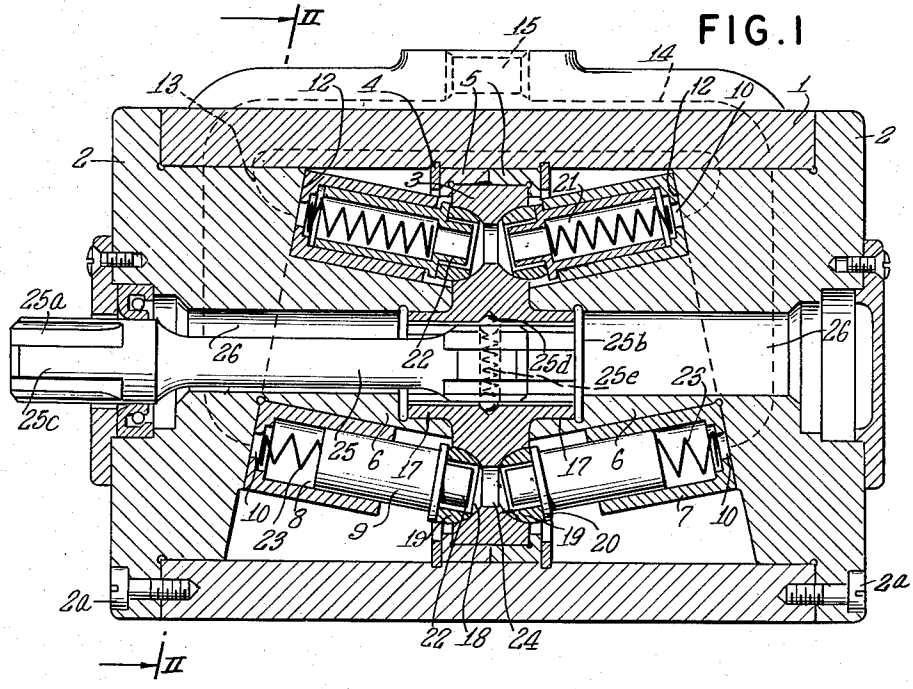
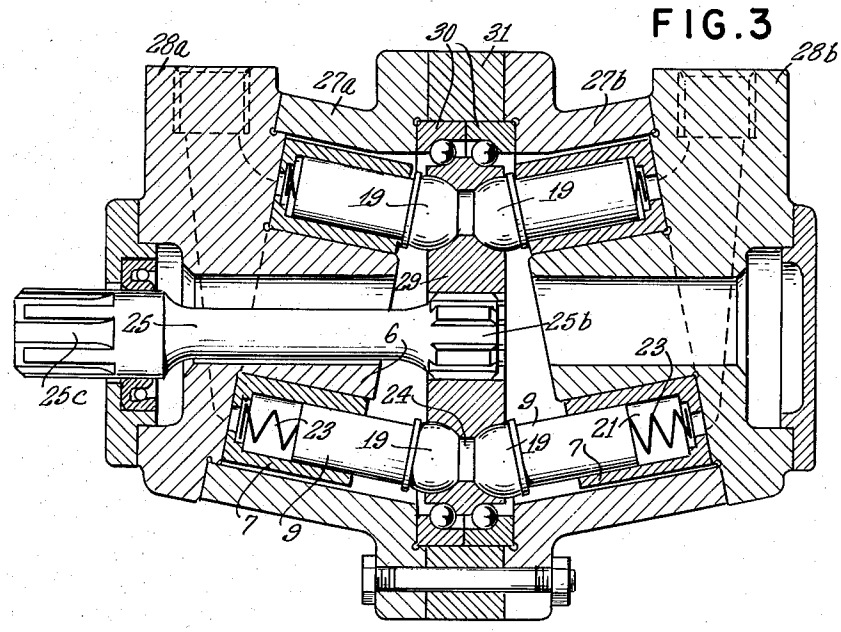


FIG. 3



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FIG. 2

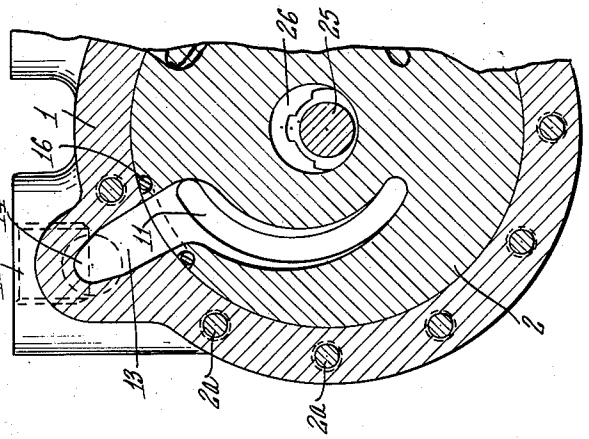


FIG. 7

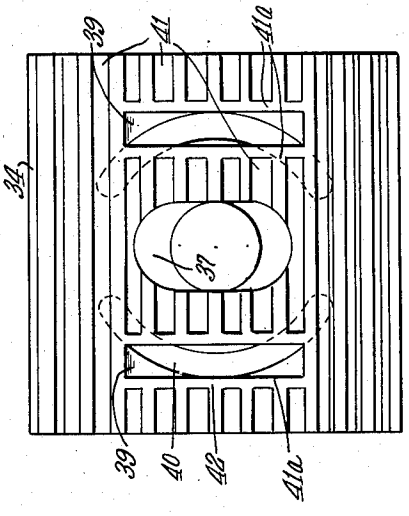
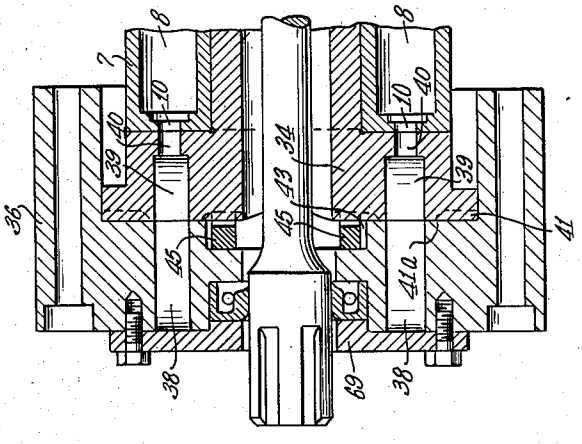


FIG. 6



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FIG. 5

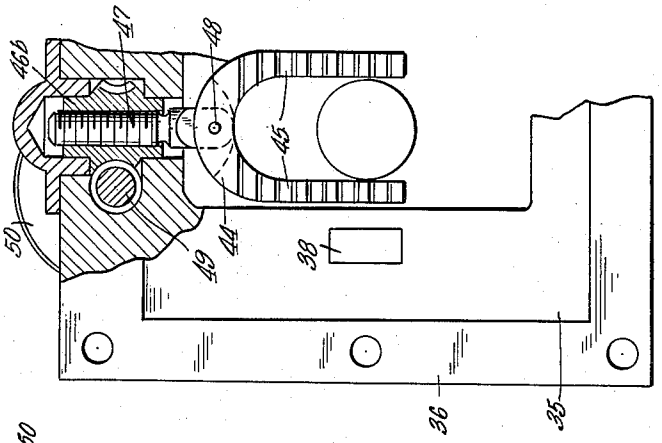
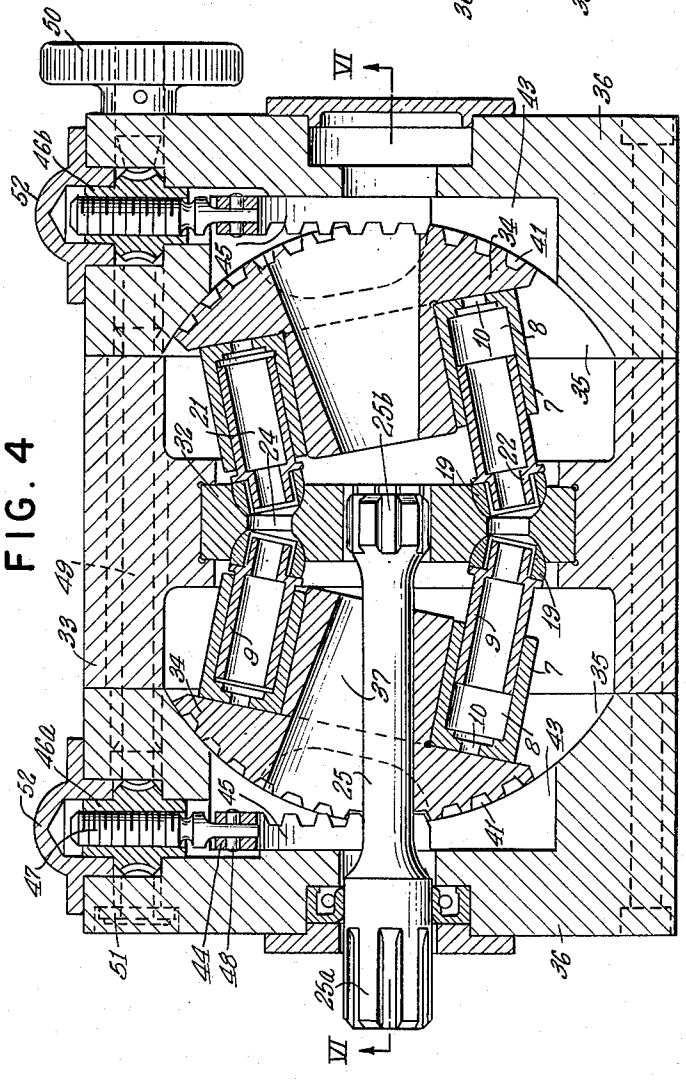


FIG. 4



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FIG. 8

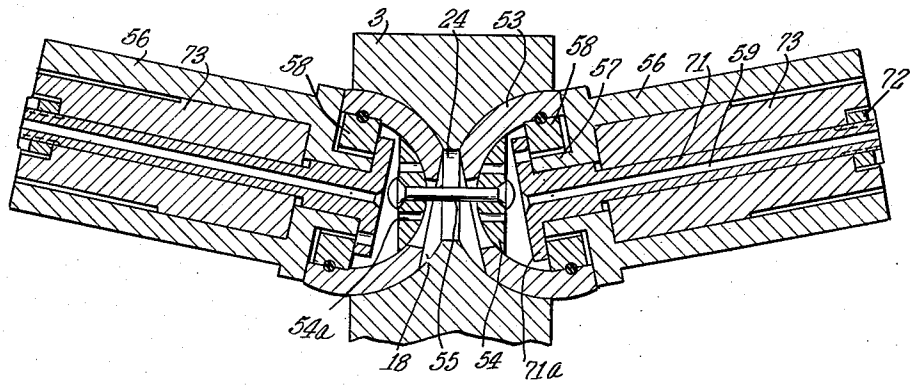


FIG. 9

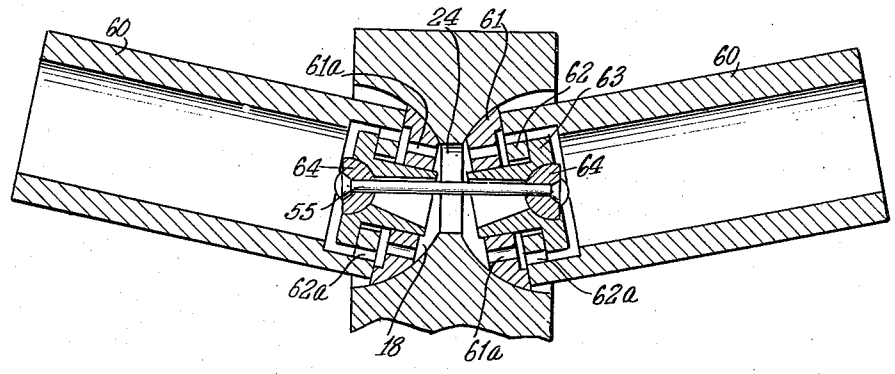
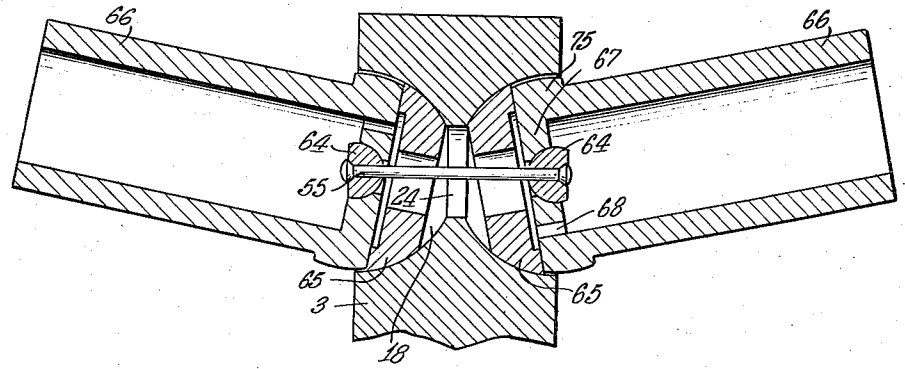


FIG. 10



United States Patent Office 2,968,286
Patented Jan. 17, 1961

2,968,286

HYDRAULIC AXIAL-PISTON MACHINE

Georg Wiggermann, Spitzgartenweg 10, Kressbronn, Germany, assignor of one-half to Walter Reiners, M. Gladbach, Germany

Filed May 23, 1957, Ser. No. 661,128

Claims priority, application Germany May 29, 1956

2 Claims. (Cl. 121—62)

My invention relates to hydraulic machines of the axial-piston type, operable as pumps or motors, in which the reciprocable pistons are located in two multi-cylinder drums so arranged that the axial components of the piston forces compensate each other. The individual pistons are connected through respective ball joints with a driving disc mounted on the machine shaft which extends through the two cylinder drums and their respective valve faces.

Hydraulic machines of this type afford a simple design of the journalling means for the rotating parts, particularly for the driving disc; and since the machine shaft passes through the cylinder drums to the outside, it is readily available for directly coupling the hydraulic machine with the driving or driven machinery.

It is an object of my invention to further improve axial piston machines of the above-mentioned type with respect to construction and operation. More specifically, it is an object of the invention to simplify the journalling of the driving disc so as to reduce the manufacturing cost of the hydraulic machine as a whole and to minimize its space requirements.

To this end, and in accordance with a feature of my invention, I provide for the two groups of pistons of both cylinder drums a single driving disc, preferably formed essentially of a single piece of material, and I journal this driving disc, together with the appertaining drive shaft of the hydraulic machine, by placing the disc directly into journal engagement with the stationary enclosure assembly of the machine. More specifically, the twin driving disc, common to both groups of pistons, has either its periphery in journalling engagement with the housing structure of the machine, or the driving disc has two coaxial hubs journalled in bearing bores of the valve-face bodies or head members of the stationary assembly. According to another feature of my invention, both journalling means just mentioned are provided simultaneously.

According to a further feature of the invention, the peripheral surface of the driving disc forms the inner race of a twin-type antifriction bearing whose outer races consist of two identical rings inserted and fixed in the stationary machine housing.

The housing structure of machines according to the invention can be given a subdivided design without detriment to the operating qualities of the rotating parts, thus further simplifying the manufacture and assembling of the machine. According to another feature of my invention, therefore, I compose the machine housing of two identical half-portions which are flanged together in the vicinity of the driving disc, preferably by means of an inserted annular bearing member for journalling the driving disc.

According to still another feature, the two identical half-portions of the housing, each having a cylindrical bore into which a cylinder-head member is inserted, have the axes of their respective cylinder bores extend at an obtuse angle to each other; and each head member has

a planar valve face perpendicular to the cylinder-bore axis and is provided with a shaft bore which extends in an inclined direction relative to that axis and is traversed by the machine shaft.

According to another feature of the invention, the housing structure of the axial-piston machine comprises a middle portion which has axially aligned cylinder bores covered by the two flanged-on head members and into which, as the case may be, a stationary bearing member for journalling the driving disc is inserted.

Further features of the invention relate to a design of the driving shaft as a torque-transmitting rod having reduced cross section along part of its length to act as a torsion spring, also to an improved articulate connection of the pistons with the driving disc particularly advantageous in machines according to the invention, and to improved means for passing the liquid operating medium through the machine.

The above-mentionad features of the invention and the favorable effects achieved thereby apply to axial-piston machines of constant delivery and uniform flow direction of the liquid operating medium, as well as, analogously, to hydraulic machines of controllable volumetric delivery and reversible flow direction.

The foregoing and other objects, advantages and features of my invention, the novel features being set forth with particularity in the claims annexed hereto, will be apparent from the embodiments illustrated on the drawings and described in the followings. On the drawings:

Fig. 1 shows an embodiment of an axial-piston machine according to the invention in longitudinal section.

Fig. 2 illustrates a section along the line II—II in Fig. 1.

Fig. 3 shows an embodiment of an axial-piston machine with a subdivided housing, also in longitudinal section.

Fig. 4 shows a longitudinal section of another embodiment of an axial-piston machine of controllable delivery.

Fig. 5 illustrates one of the head members which form the housing covers of the machine according to Fig. 4, seen from within the machine.

Fig. 6 is a fragmentary longitudinal section through the left-hand end of a modification of the axial-piston machine according to Fig. 4 and generally corresponding to a view along line VI—VI of Fig. 4.

Fig. 7 is a view onto the back of a tiltable control member appertaining to one of the head members of the same machine.

Figs. 8, 9 and 10 illustrate different modifications respectively of ball-type connections joining the pistons to the driving disc.

Functionally similar components are denoted by the same respective reference characters in all illustrations.

The machine illustrated in Figs. 1 and 2 has a stationary enclosure assembly composed of a substantially cylindrical housing structure 1 and two cylinder-head members 2 of identical design. The two head members 2 are inserted from opposite sides into the cylindrical bore centrally traversing the housing structure 1 and are flanged to the housing structure 1 and fastened thereto by screw bolts 2*a*. A twin driving disc 3 is peripherally journalled in two bearing rings 5 which are stationarily mounted in the cylinder bore of housing structure 1 and are securely held in position by expansion rings 4 so as to be prevented from rotating. Each head member 2 forms a pintle 6 upon which a multi-cylinder drum 7 is rotatably mounted. Each cylinder drum 7 has a number of cylinder bores 8 uniformly distributed about the periphery and extending parallel to the drum axis. Respective pistons 9 are reciprocably guided in the cylinder bores and are sealed against the cylinder walls. Each cylinder bore has a control port 10 which, in the proper rotational posi-

tion of the cylinder bore, communicates with one of two control grooves 11 (Fig. 2) in a planar valve surface 12 of the adjacent head member 2.

The two cylinder drums 7 are hydraulically parallel connected. That is, each of the two control grooves 11 in each head member 2 is connected with the one axially opposite groove 11 of the other head member 2 through a radial channel 13 (Fig. 2) and a duct 14 which extends between the radial channels 13 in the longitudinal direction of the housing structure. The two ducts 14 are provided with respective openings 15 for connection to the pipes that supply and discharge the hydraulic operating medium. The gap in each radial channel 13 caused by the central bore of the housing structure can be sufficiently sealed by accurate fitting of the flange-connected head member 2 into the housing bore. As a matter of precaution, however, an annular groove is milled into each head member 2 around the channel 13, and an elastic gasket ring 16 of rubber or rubber-like material is inserted into the annular groove under slight compression. The pressure of the hydraulic medium obtaining in the channels then takes care of providing for a reliable seal at this location.

For minimizing hydraulic losses, each channel 13 merges with the control groove 11 in such a manner that the hydraulic medium flowing in the groove has a direction and speed corresponding to the peripheral velocity of the cylinder ports 10 at the valve surface.

For relieving the above-described peripheral bearing of the driving disc 3, the driving disc is provided with two hubs 17 on opposite axial sides. The hubs 17 are rotatably guided in bearing bores of the respective head members 2. The bores are coaxially aligned with the disc 3 and the center bore of housing structure 1.

The twin driving disc 3 is provided on both sides with a number of semispherical cavities or "pans" 18 uniformly distributed on a circle concentric to the disc. Fitted into each pan 18 is a spherical ring 19 capable of universal motion relative to the driving disc. The spherical rings serve as axial abutments for the respective reciprocable pistons 9.

The head members 2 are so designed and mounted within the housing structure 1 that the pintle axis 6 of each cylinder drum 7 is inclined to the common axis of housing structure and twin driving disc. The two pintle axes and the disc axis are located in the same geometric plane, the bisector of the angle between the two pintle axes being perpendicular to the disc axis. The inclined position of the pintle axis on each head member 2 causes the pistons to perform the desired reciprocating stroke during rotation of the cylinder drums. When the pistons are located in the upper position shown in Fig. 1, they are fully moved into the respective cylinder bores; and when the pistons, during 180° rotary motion of the cylinder drums, reach the illustrated lowermost position, the pistons have performed a full outward stroke. Assuming operation of the machine as a pump, the cylinder port 10 of any one piston, moving from the illustrated upper position to the lower position, is in communication with the groove 11 (Fig. 2) connected with the inlet or suction duct 14 for the hydraulic medium; whereas when the piston continues its movement and returns to the stroke-minimum position, the same cylinder port 10 is in communication with the groove 11 communicating with the outlet or pressure duct of the machine.

There are equal numbers of spherical pans 18 on the two sides of the driving disc 3, and each pair of sphere-center points defines a line parallel to the axis of driving disc 3 and the appertaining machine shaft.

During operation, the liquid medium, supplied to the cylinders through the valve grooves 11, produces in each piston 9 an axial force which is directed toward the twin driving disc 3 and which may be thought of as being composed of a transverse component and an axial component acting upon the spherical joint. The axial components of each pair of adjacent spherical joints are always located on the above-mentioned line parallel to the shaft axis; they have the same line of action but mutually opposed directions and hence cancel each other. Consequently only the transverse component of each piston force remains effective. With respect to each pair of pistons, these transverse components have the same direction and the same magnitude and thus produce in the twin driving disc 3 the desired torque. The absolute magnitude of the torque is dependent upon the diameter and the number of the pistons as well as upon the inclination of the cylinder-drum axes.

Each piston 9 has an annular shoulder abutting, in intimate area contact, against a flat and plane front face 20 of the adjacent spherical ring 19. The annular abutment face of the spherical ring has an outer diameter equal or larger than the piston diameter, whereas the inner diameter of the abutment face is smaller than the piston diameter. Such dimensioning of the abutment diameters permits relieving the abutment from hydraulic load by giving the piston 9 a longitudinal bore 21 through which the pressure of the hydraulic medium can become active up to the spherical ring 19. As a result, the piston force to be taken up by the annular abutment face 20 corresponds only to the product formed by the difference of piston area minus abutment area multiplied by the hydraulic pressure. This force upon abutment face 20 can be increased, by increasing the inner diameter of the annular abutment face, to the extent permissible by the required self-sealing pressure engagement between piston and spherical ring.

The contour-constrained, universally movable joint between the pistons and the twin driving disc secures strictly synchronous rotation of the two cylinder drums 7 together with the twin driving discs, but the spherical joints are also designed to act as a positive power-transmitting coupling. To this end, each piston has a cylindrical and tubular extension 22 which protrudes, with some diametrical clearance, into the cylindrical center bore of the spherical ring 19. The slight diametrical clearance is necessary for permitting the piston a slight transverse motion required as a compensation between the elliptic travel path of the piston and the circular travel path of the spherical rings. The diametrical clearance takes care of maintaining continuous pressure transmission between the abutment face 20 of the spherical ring and the adjacent piston. While during the pressure stroke, the abutting engagement of the pistons relative to the spherical rings and the driving disc is always maintained by the hydraulic pressure, the desired reliable abutting engagement during the suction stroke is secured by helical compression springs 23 mounted in the cylinder bores inside the hollow piston. The springs also maintain intimate contact between the cylinder drum 7 and the valve surface 12 of the adjacent head member 2 in no-load condition of the machine.

Instead of return spring 23, or together with such a spring, the pistons may be secured to the disc 3 by a device as illustrated in Figs. 8, 9 or 10 and described below.

The axial components of the piston forces produced in the two cylinder drums must completely balance and cancel each other under all operating conditions if the journalling of the driving disc is to be kept free of any undesired forces or tilting moments that are not directed transversely in the direction of the rotation to be produced or not transmitted by the driving disc, such elimination of undesired forces being one of the main advantages of the novel machine design. For further improving the reliability of such pressure and force equalization in each pair of mutually opposite cylinder bores of the respective cylinder drums, each two adjacent spherical pans 18 are hydraulically interconnected through a bore 24 of the

driving disc 3. Any residual pressure differences can thus equalize themselves.

The delivering capacity of the axial-piston machine depends, among other things, upon the angle of inclination of the cylinder pintle axes relative to the rotation axis of the driving disc. To permit increasing this angle of inclination, the drive shaft 25 is made only as thick in the range of the cylinder drums as is sufficient for transmission of the required torque. That is, the diameter of the driving shaft, where it passes through the cylinder drums, is smaller than would be needed if not only torque transmission but also the great bending strength necessary in the known machines of this general type were required. Such reduction of the shaft diameter in machines according to the invention is made possible by virtue of the fact that the twin driving disc 3 is independently journalled and supported against the stationary envelope structure of the machine, rather than being mounted on, and journalled by means of, the driving shaft. For connecting the driving shaft to the twin driving disc 3 on the one hand, and to the driving or driven machinery outside of the hydraulic machine on the other hand, the two ends 25*a* and 25*b* of the shaft have larger diameter than the intermediate shaft portion and are preferably profiled to form keyways, such as the one shown at 25*c* in Fig. 1, for acommodating torque-transmitting keys or wedges. The shaft end 25*b* may also have milled-in teeth, comparable to those of a spur gear, which slidably engage mating teeth in the bore of the disc 3.

For axially fixing the shaft 25 in position, the shaft end 25*b* located within the driving disc 3 is provided with a catch device. This device comprises two balls 25*d* and an intermediate helical compression spring 25*e* located in a transverse bore of shaft end 25*b*. The balls 25*d* engage an annular groove in driving disc 3.

To permit inserting and removing the drive shaft 25, each head member 2 has a bore 26 in eccentric relation to the axis of shaft 25. The diameter of bore 26 is made just large enough to permit passage of the inner shaft end 25*b* and into engagement with the driving disc 3.

Since the two head members 2 are of identical design, the hydraulic machine can readily be adapted to various operating conditions by having the driving shaft 25 enter from one or the other side as may be desired, or by using a longer driving shaft which passes through both head members 2 and protrudes on both sides out of the machine. This offers various possibilities of connecting the axial-piston machine to driving or driven machinery.

The machine illustrated in Fig. 3 is to a large extent similar to the embodiment described above with reference to Fig. 1, particularly with respect to the design and arrangement of the driving disc, the pistons, their universal joint connections with the driving disc, and the cylinder drums. One of the essential differences, however, is the design of the stationary enclosure. According to Fig. 3, the housing structure is composed of two identical half-portions 27 and 27*b* whose respective axes form an obtuse angle with each other and which are flanged together in the vicinity of the centrally located twin driving disc 30. The head members 28*a* and 28*b*, serving as end covers for the housing structure, are mounted on, and flanged together with, the cylindrical ends of respective housing portions 27*a* and 27*b*, the planar valve face of each head member being concentric and coaxial relative to the adjacent cylindrical housing portion. In the illustrated embodiment, each cylinder drum 7 is journalled on a pintle 6 of the head member. However, this design also permits journalling the cylinder drums at their periphery in sliding or other journalling engagement with the bores of the respective housing half-portions 27*a* and 27*b* respectively.

As explained above, the axial-piston machine according to Fig. 1 has its twin driving disc provided with hubs 17 in journalling engagement with bearing bores of the respective head members 2. A prerequisite for such a

design is an accurate alignment of the two bearing bores. Such alignment can readily be secured in a machine of the type shown in Fig. 1 because the bearing bores of the two head members 2 are concentric to the respective mounting flanges of the head members thus causing no difficulty in manufacture. In contrast thereto, the twin driving disc 29 in the machine according to Fig. 3 is journalled only at its periphery and, for this reason, is provided with an antifriction bearing of extremely small space requirements. The antifriction bearing consists of a twin-type ball bearing. The inner races for the bearing balls are ground into the periphery of the twin driving disc 29. The outer races are formed by two bearing rings 30 centered and fastened by means of a supporting ring 31 which is rigidly clamped between the respective flanges of the two housing portions 27*a* and 27*b*. Additional journalling of the driving disc by means of hubs guided in bores of the head members is not resorted to because the inclined position of the centering and flange faces of the head members 2 would make it difficult in manufacture to obtain the accurate concentricity and alignment of the hub bearings required for satisfactory operation. The machine according to Fig. 3, compared with that of Fig. 1, affords smaller housing dimensions and reduced weight and hence is preferable for many applications.

The axial-piston machine shown in Fig. 4 incorporates further improvements over the machines described above with reference to Figs. 1 to 3 and affords stepless variation of volumetric delivery and delivering direction. The twin driving disc 32 is peripherally journalled for coaxial rotation in the stationary machine housing structure 33. The housing structure is covered by respective head members which, in this embodiment, are of composite construction. Each head member comprises a rigidly mounted cover member 36 flanged to the housing structure 33, and a displaceable control body 34 seated in cylinder-shaped trough 35 of the cover member 36. The control body 34 is rotationally displaceable from a mid-position toward both sides in a geometric plane determined by the center points of the spherical rings 19 and hence of the ball joints between the pistons and the driving disc, such displacing motion being about the cylinder axis of the trough 35 which extends perpendicular to the plane of illustration in Fig. 4. To permit passage of the driving shaft 25, each control body 34 has a conical opening 37 whose smallest diameter is just sufficient to permit passage of the thick shaft end 25*b*. The supply and discharge of oil or other hydraulic medium is effected at each of the two head members from two channels 38 (Fig. 6) in the stationary portion 36 of the head member through two slots 39 (Figs. 6, 7) of the displaceable control body 34 and thence through two approximately semicircular control grooves 40 in the valve face to the parts 10 of the cylinder bores 8.

In practice, advantage can be taken of the possibility of operating the machine according to Fig. 4 by supplying and discharging the hydraulic medium for both cylinder drums through only one of the two head members and the appertaining one displaceable control body 34. The slots 39 then serve only for relieving the inactive control body 34 of hydraulic loading, and the channels 38 of the adjacent cover portion 36 are then closed by means of a cover plate 69 (Fig. 6).

In a twin machine of the type exemplified by Fig. 4, the two pintles for the cylinder drums must always occupy the same angular displacement relative to the machine axis regardless of the adjusted amount of inclination. Another requirement to be met by such a design is the fact that the hydraulic equilibrium of forces at the cylindrical gliding surface on the back of control body 34 must remain as little affected as possible by the displacing forces. The embodiment illustrated in Figs. 4 and 7 satisfies both requirements as follows.

The cylinder-shaped back of each control body 34

is provided with load-relieving grooves **41** (Figs. 6, 7) which extend in the transverse direction and, in totality, form a spur gearing. Some of the grooves or tooth profiles **41** are only partly milled into the control body **34** so that the slots **39** remain framed within a continuously closed frame surface **41***a* which maintains a tight sealing engagement with the cylindrical trough surface in the stationary cover portion of the head member (Figs. 4, 6, 7). Two racks **45** (Fig. 5, 6) are guided in respective grooves **43** machined into the cylindrical trough surface of each head member. The two racks **45** are rigidly interconnected by a yoke portion **44** (Fig. 5) and are in meshing engagement with the grooves **41** of the control body **34**.

Two worm gears **46***a* and **46***b* are rotatably mounted in recesses of the respective two head members **36**. Each worm gear has a bore in threaded engagement with a screw spindle **47** (Figs. 4, 5) which is articulately linked with the yoke **44** of racks **45** by means of a pin **48** so as to be prevented from rotating about the spindle axis. Rotation of worm gears **46***a* and **46***b* causes axial displacement of the respective screw spindles **47**, thus transmitting the required displacing force to the pairs of racks **45**. The two racks **45** of each pair act upon the adjacent control body **34** at locations symmetrical to the shaft axis of the machine. The worm gears **46***a* and **46***b* are constrainedly coupled with each other for synchronous operation. To this end, a worm shaft **49**, stationarily journalled on head members **36**, is in meshing engagement with both worm gears **46***a* and **46***b* and can be turned by means of a hand wheel **50** or other control device. When assembling such axial-piston machines of controllable delivery, the two displacement control devices must be adjusted for equal inclination of the respective cylinder drums. In the embodiment of Fig. 4, such adjustment is readily possible by removing from the worm shaft **49** a shoulder ring **51** which normally fixes the shaft axially. Then the worm shaft can be pulled toward the right a slight amount out of the machine. As a result, the worm gear **46***a* is placed out of meshing engagement and, after removing a cover **52** (Fig. 4), can be individually turned the required equalizing amount. Subsequent axial shifting of the worm shaft, caused simply by turning this shaft while worm gear **46***b* is kept arrested, again places the worm gear **46***a* into threaded engagement with the worm shaft. After the worm shaft has reached its normal position it is again axially fixed by attaching the shoulder ring **51**.

Fig. 8 illustrates a constrainedly operating pull-back device for the pistons and their universal junction with the twin driving disc, such a device being applicable with any of the embodiments previously described in this specification. Only one pair of pistons cooperating with mutually aligned, adjacent spherical pans of the twin driving disc is illustrated. A spherical shell **53** is in gliding spherical-area engagement with each pan **18**. Lifting of the two shells **53** is prevented by respective calotte-shaped holder plates **54** likewise in spherical area engagement with the shells **53**. The holder plates **54** of the two adjacent ball joints are connected and held together by means of a pin **55** which traverses the connecting bore **24** of the driving disc **3**. Each spherical shell **53** has a flat annular rim surface in face-to-face engagement with an annular shoulder of the adjacent piston **56**. Each piston **56** has at its front a hollow cylindrical extension **57**. A tubular rod **71** passes through the piston and through a filler piece **73** of light-weight material, for instance aluminum or other light metal. The rod member **71** has a flange portion **71***a* which rests against the end of the cylindrical extension **57** and forms together therewith a peripheral groove into which a ring-shaped member **58**, firmly joined with the spherical shell **53**, is inserted. Sufficient clearance is provided between the cylindrical extension **57** and the ring member **58** to permit the kinematically necessary minimum displacement of the piston structure relative to the appertaining spherical joint while nevertheless always maintaining at least a line engagement between extension **57** and ring **58** for the required driving connection between the cylinder drum and the driving disc **3**. The tubular rod **71** carries on its opposite end a screw nut **72** for fastening the filler piece **73** to the piston. The filler piece **73** serves to reduce the dead cylinder space, which is particularly desirable for high-presure pumping operation.

The working pressure of the hydraulic medium to be delivered can propagate itself through the bore **59** of the tubular rod **71** into the interior of the spherical shell **53** and effects lubrication and hydraulic relief of all gliding surfaces of the spherical piston joint. This also secures hydraulic equalization of the pressure within the two cylinder spaces communicating with each other through the bore **24** of the driving disc **3** and through a number of bores **54***a* in plates **54**.

Another modification of a universal-joint connection of two mutually opposed pistons with the common driving disc is illustrated in Fig. 9. The two pistons **60** have respective annular front faces abutting against corresponding ring-shaped counter faces of spherical rings **61** which are glidably seated in the spherical pans **18** of the twin driving disc **3**. Each piston **60** has a bottom portion **62**. A bushing **63** is fitted into a mating bore of the spherical ring **61** and has a flange portion tightly abutting against the bottom portion **62** of piston **60**. The center bore of bottom portion **62** is slightly larger than the cylindrical portion of bushing **63** to provide the kinematically necessary clearance. During operation of the machine the bushing **63** and the shoulder formed by the cylinder-bottom portion **62** enter into radial engagement for the purpose of transmitting torque from the driving disc to the pistons, or vice versa. The bushing **63** thus holds the piston transversely against the spherical ring **61**. A ball member **64**, glidably seated in a spherical bore of bushing **63**, is connected with the corresponding ball **64** of the other bushing by a connecting rod **55** extending through the connecting bore **24** of the twin driving disc. This connection secures a reliable articulate attachment of all piston and joint components to the twin driving disc under any operating condition of the machine.

For lubricating and hydraulically relieving all gliding faces of the spherical rings and for providing a hydraulic equalizing communication between each two adjacent cylinders, the bottom portion **62** of the pistons and the spherical ring **61** are each provided with several bores **62***a* and **61***a* through which the pressure of the hydraulic working medium in one cylinder can equalize itself toward the adjacent cylinder whenever necessary.

A further piston joint suitable in machines according to the invention is illustrated in Fig. 10. Spherical rings **65** are seated in the spherical pans **18** of the driving disc in intimate area engagement therewith. The spherical rings **65** have planar front faces of annular shape abutting against the respective pistons **66**. Each piston **66** has a bottom portion **67** whose inner side is provided with a centrally located spherical pan. Seated in each of these pans is a ball member **64** which is connected with the corresponding ball member of the other piston by means of a connecting rod **55** that traverses the bore **24** of the driving member, thus securing the pistons and spherical rings in sealed sliding engagement with the twin driving disc. A slight degree of transverse movability of the pistons must be provided for the reasons explained with reference to the modifications already described. For this purpose, the bottom portion **67** of each piston **66** has a spherical peripheral surface **75** which projects into the spherical pan **18** but has a slightly smaller diameter than the pan. During operation, the peripheral piston surface **75** enters into radial line engagement with the spherical pan **18** and is thus constrainedly entrained by the pan to transmit rotation.

If desired, each piston **66** may be provided with a cylin-

drical extension similar to the one denoted by 22 in Fig. 1, such extension protruding into a cylindrical center bore of ring 65 with slight diametrical clearance. Compared with such a design, however, the modification according to Fig. 10 has the advantage that the spherical rings 65 are not affected by the tangential forces that serve for driving the cylinder drum, because the spherical portion 75 of the piston is directly entrained and driven by the spherical pan 18 of the driving member 3.

The required hydraulic equalizing communication of neighboring cylinder spaces, as well as the lubrication and hydraulic relief of the gliding surfaces on spherical rings 65, is obtained by means of a plurality of bores 68 in the bottom portion 67 of each piston.

It may be mentioned that the driving torques of the cylinder drums, even with highest delivery pressures, are relatively slight and for that reason do not represent an additional load of appreciable or wear-promoting magnitude. The connecting rods 55 described with reference to Figs. 8, 9 and 10, may be made of a very thin high-quality steel wire which, when assembling the junction, is electrically heated at both ends and is then widened by hot upsetting.

It will be obvious to those skilled in the art, upon study of this disclosure, that machines according to my invention may be modified in various respects and may be given embodiments other than those particularly illustrated and described, without departing from the essential features of my invention and within the scope of the claims annexed hereto.

I claim:

1. A hydraulic machine of the axial-piston type, comprising a shaft, a stationary housing structure around said shaft, two axially spaced head members mounted on said housing structure, said head members having respective planar valve surfaces facing each other, two multi-cylinder drums surrounding the axis of said shaft and rotatable on said respective valve surfaces, each drum having peripherally distributed cylinder bores parallel to the drum axis, said bores in at least one of said drums having respective ports at the adjacent valve surface to coact therewith, pistons reciprocable in said respective bores, a rotatable disc coaxially connected with said shaft and located in said housing structure between said two cylinder drums, hollow ball-joint means joining said respective pistons of both said drums with said disc, said disc having a hole between each two axially adjacent ones of said ball-joint means, and said cylinders communicating with each other in pairs through said respective holes and hollow joint means, said head member adjacent to said one drum having two arcuate control grooves angularly spaced from each other in said valve surface for hydraulic communication with said cylinder ports, and hydraulic circulation ducts in communication with said respective grooves to pass hydraulic medium through the machine, whereby the cylinder bores of said one drum are directly supplied with hydraulic medium through said grooves, whereas the cylinder bores of the other drum receive said medium through said hollow joint means and through said holes in said disc.

2. A hydraulic machine of the axial-piston type, comprising a shaft, a stationary housing structure around said shaft, two axially spaced head members mounted on said housing structure, said head members having respective planar valve surfaces facing each other, two multi-cylinder drums surrounding the axis of said shaft and rotatable on said respective valve surfaces, each drum having peripherally distributed cylinder bores parallel to the drum axis, said bores in at least one of said drums having respective ports at the adjacent valve surface to coact therewith, pistons reciprocable in said respective bores, a rotatable disc coaxially connected with said shaft and located in said housing structure between said two cylinder drums, hollow ball-joint means joining said respective pistons of both said drums with said disc, said disc having a hole between each two axially adjacent ones of said ball-joint means, and said cylinders communicating with each other in pairs through said respective holes and hollow joint means, at least one of said head members having two arcuate control grooves angularly spaced from each other in said valve surface for hydraulic communication with said cylinder ports, and hydraulic circulation ducts in communication with said respective grooves to pass hydraulic medium through the machine, whereby the cylinder bores of said one drum are directly supplied with hydraulic medium through said grooves, whereas the cylinder bores of the other drum receive said medium through said hollow joint means and through said holes in said disc.

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