March 22, 1938. E. K. BENEDEK 2,111,659

HYDRAULIC PUMP OR MOTOR

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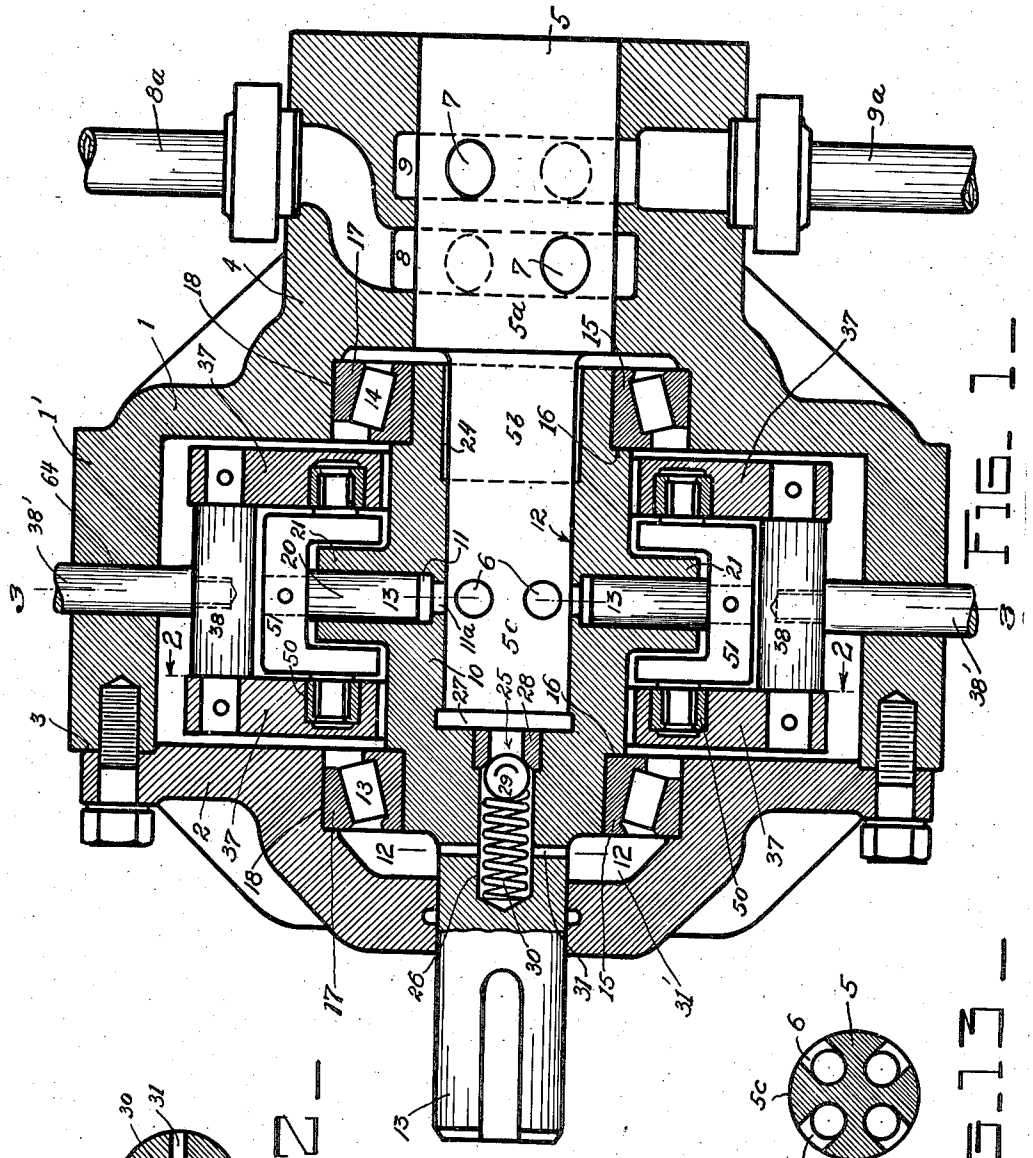


FIG. 1

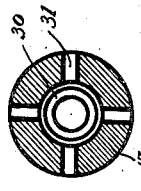


FIG. 12

FIG. 13

Inventor

ELEK K BENEDEK

By Soule & Leonard

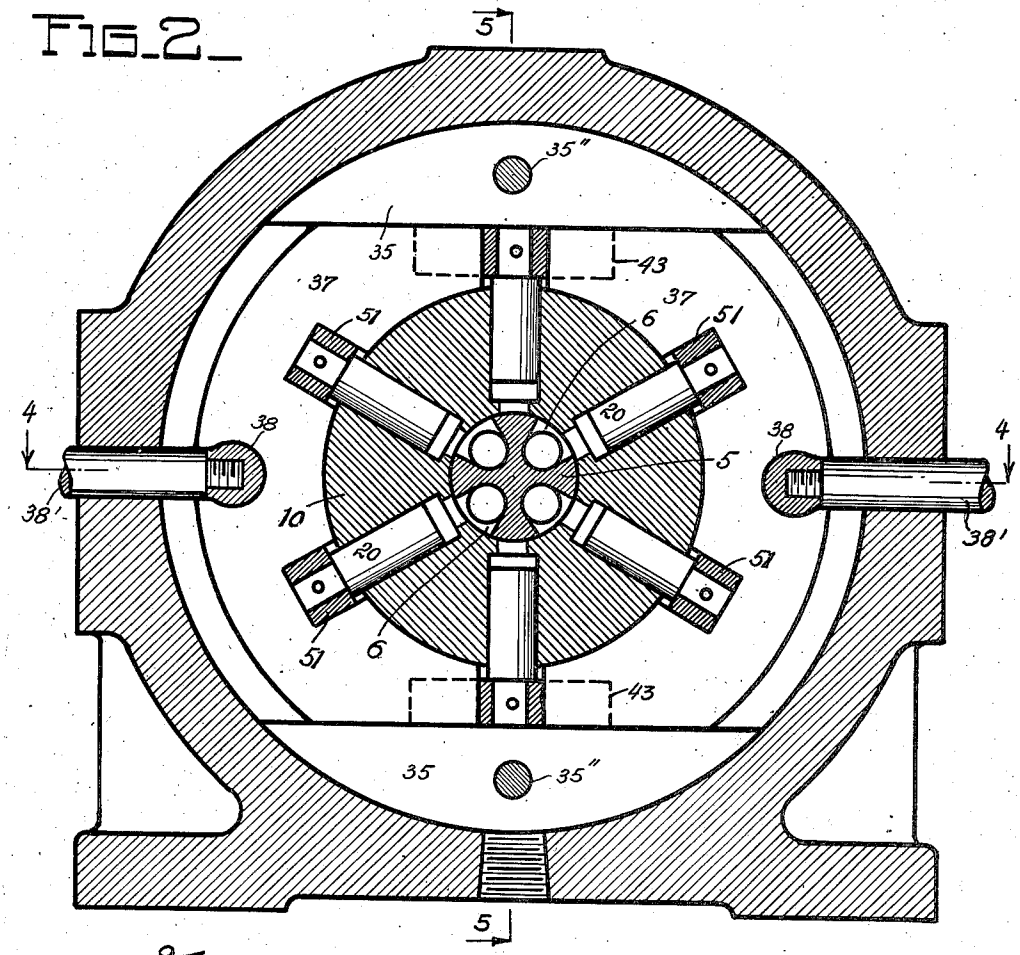
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FIG. 2.



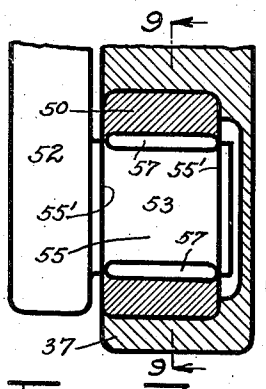


FIG. 7.

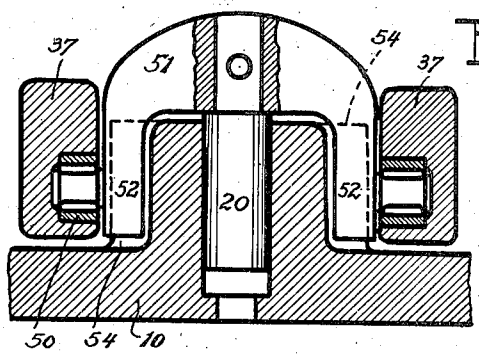
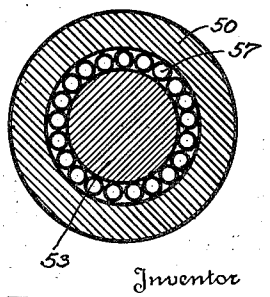


FIG. 6.

FIG. 9.



Inventor

ELEK K. BENEDEK.

By Soule & Leonard

Attorneys.

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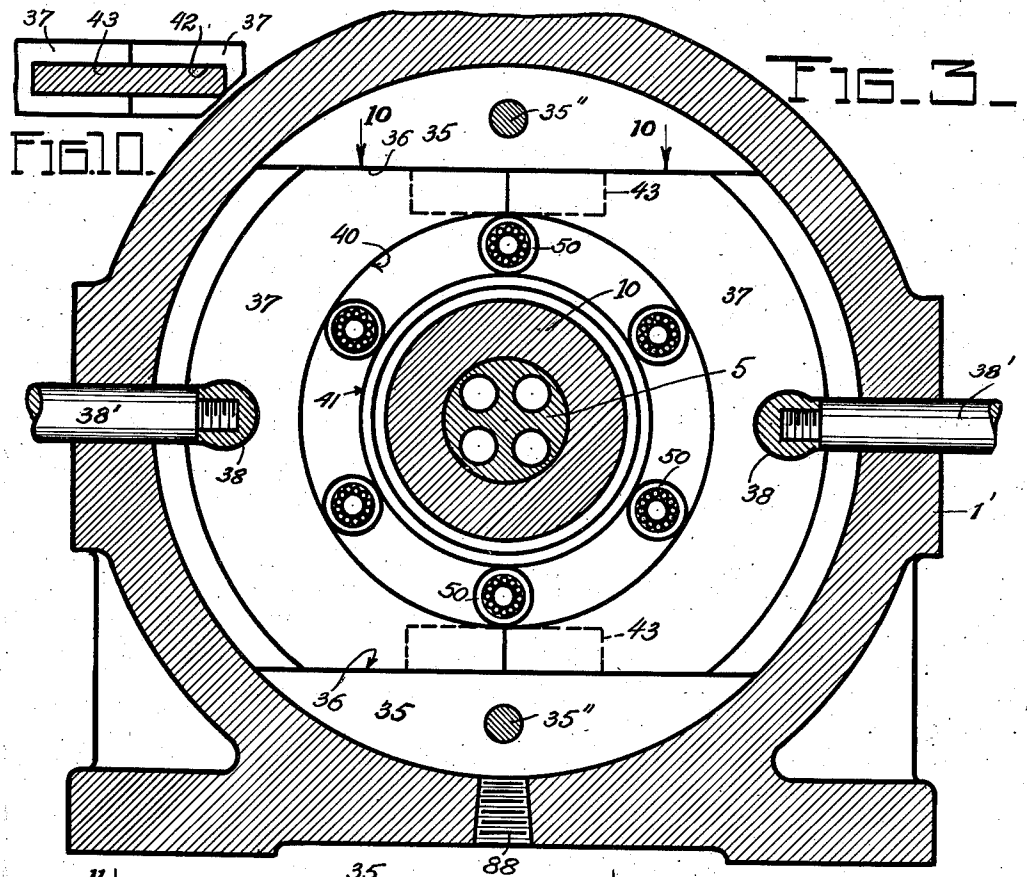


FIG. 3.

FIG. 10.

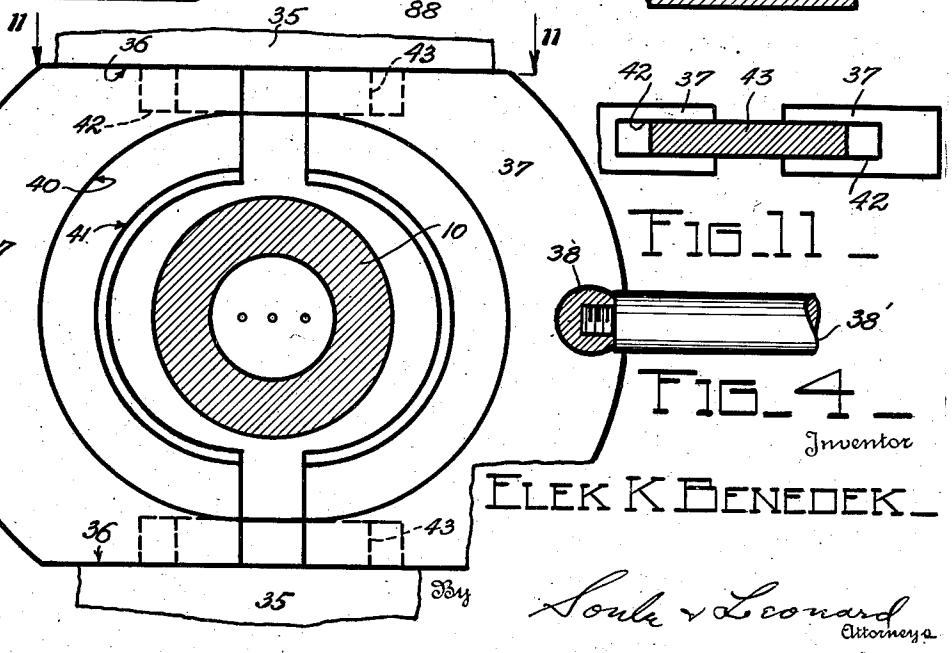


FIG. 11.

FIG. 4.

Inventor

ELEK K BENEDEK

By Soule & Leonard

Attorneys

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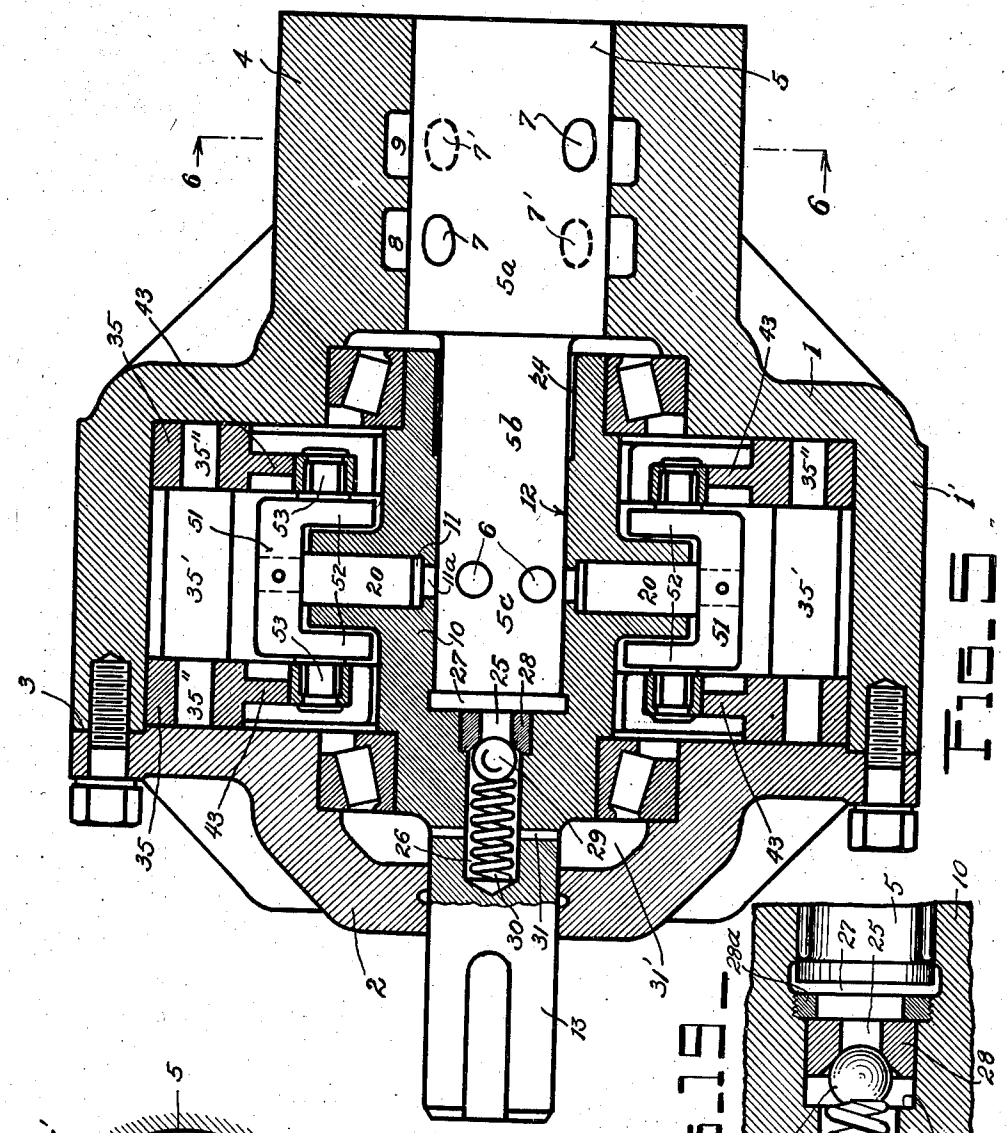


FIG. 5

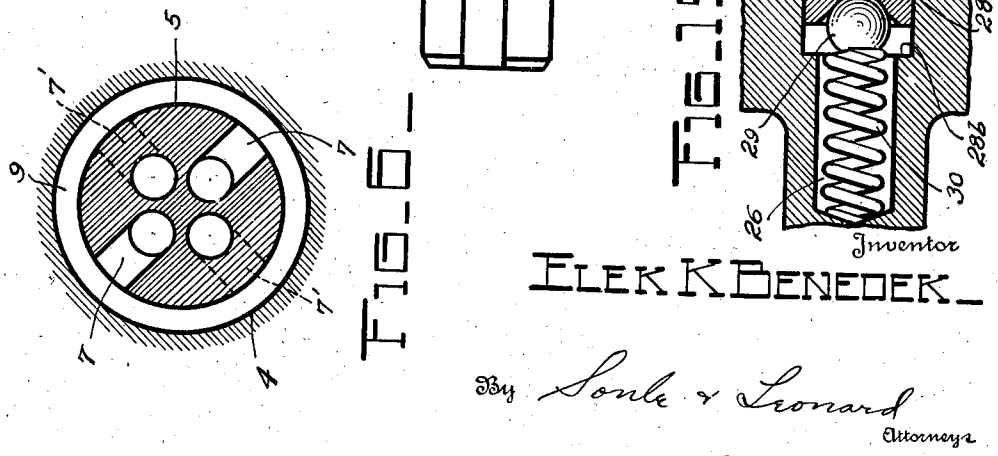


FIG. 15

FIG. 6

Inventor

ELEK K BENEDEK

By Soule & Leonard

Attorneys

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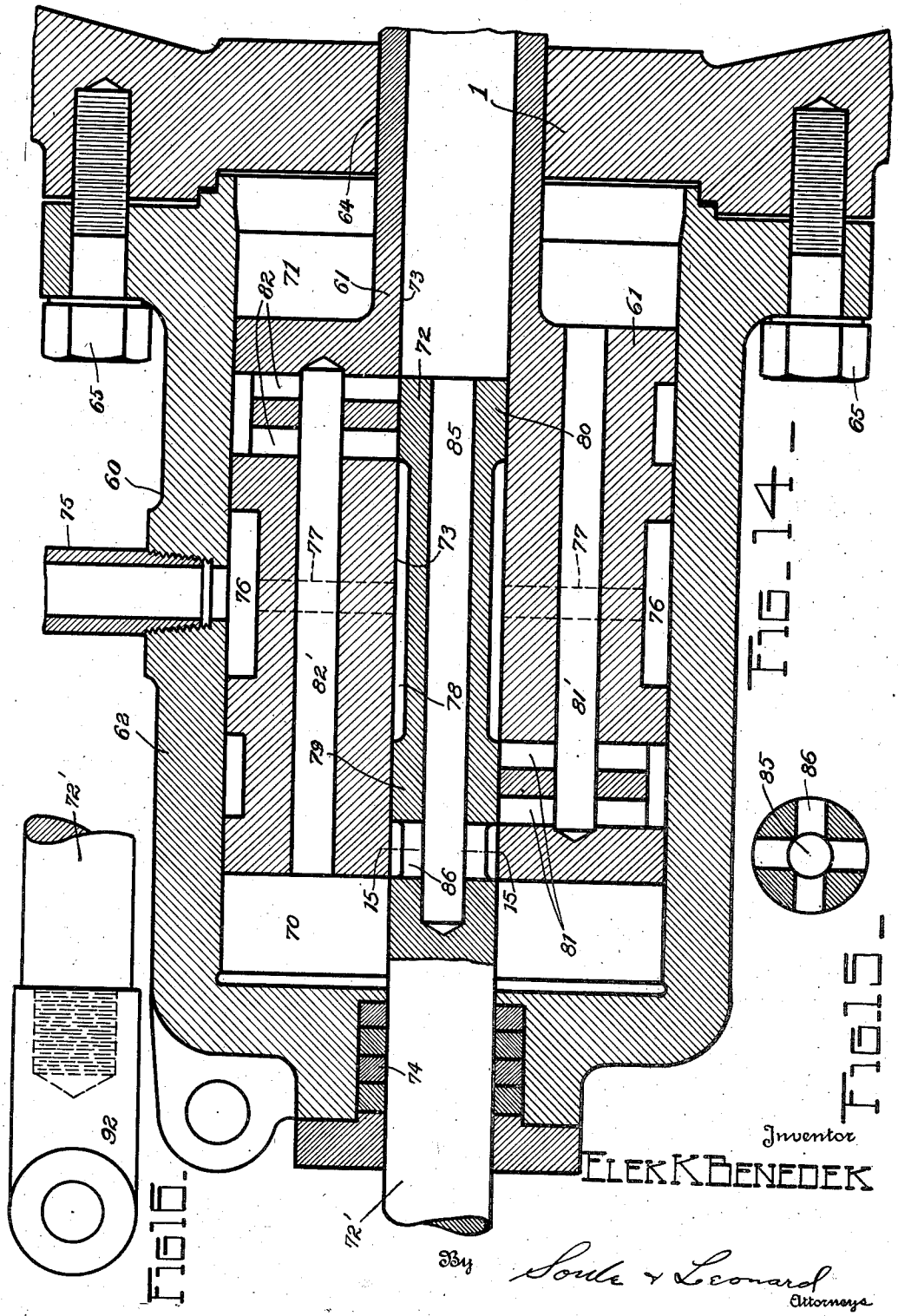


Fig 14

Fig 15

Fig 16

Inventor
ELEK K BENEDEK

By Soule & Leonard
Attorneys

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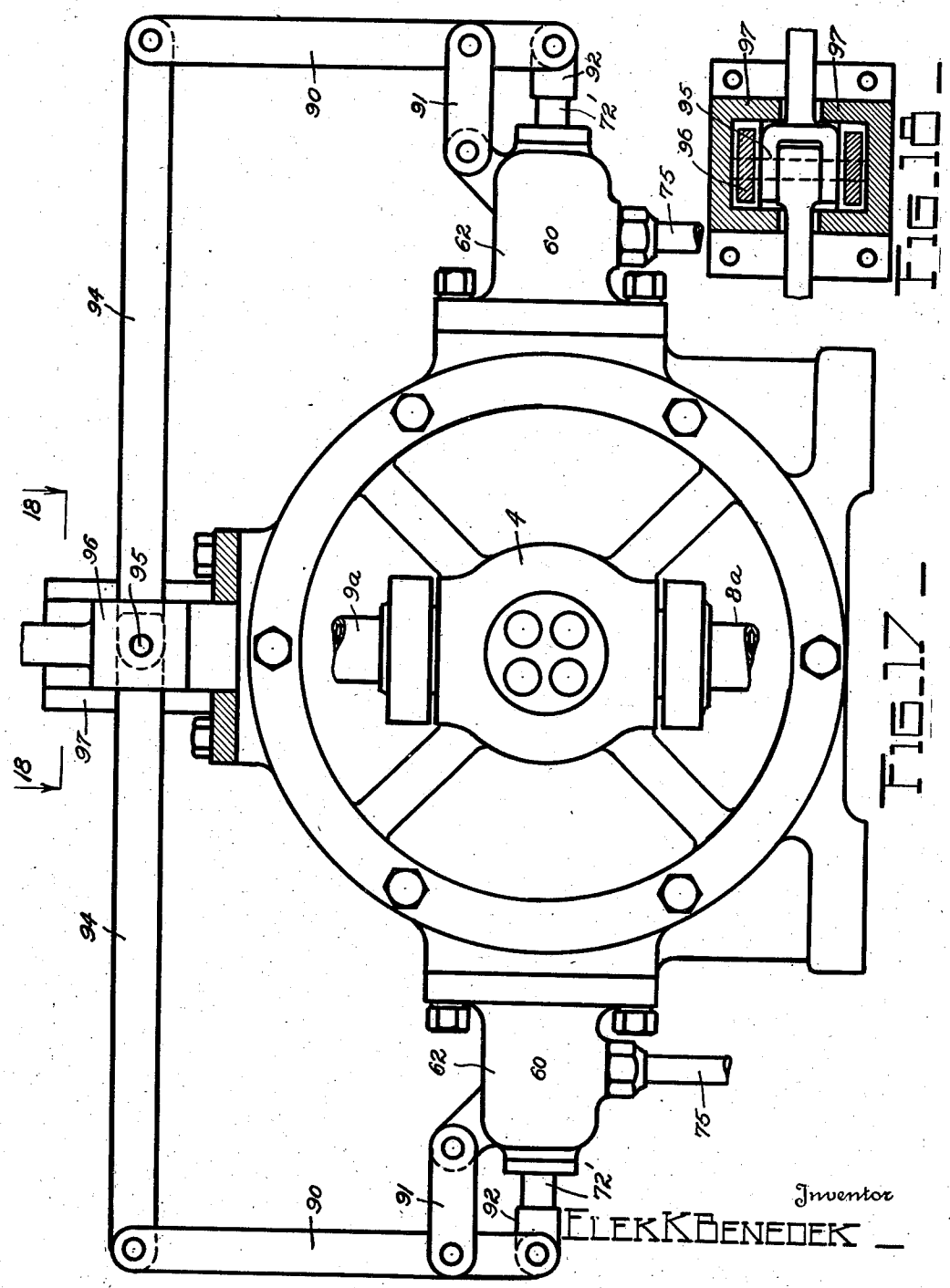


FIG. 16

FIG. 17

Inventor
ELEK K. BENEDEK

By Soule & Leonard
Attorneys

Patented Mar. 22, 1938 2,111,659

UNITED STATES PATENT OFFICE

2,111,659

HYDRAULIC PUMP OR MOTOR

Elek K. Benedek, Bucyrus, Ohio

Application October 24, 1934, Serial No. 749,746

6 Claims. (Cl. 103—161)

This invention relates to variable displacement, multi-plunger type pumps or motors and has for its primary object the provision of a more efficient and more durable pump or motor than heretofore designed.

The machine illustrated herewith is of the known type comprising a stationary valve pintle and a rotary cylinder barrel (primary rotor) mounted on the valve pintle. Ordinarily, in this type of machine, the cylinder barrel is surrounded by a secondary rotor which actuates the pistons when disposed eccentrically of the barrel. These pumps have the disadvantage of imposing great hydrostatic load on one side of the valve pintle because the eccentric effects pressure strokes of the pistons at one side of the barrel only, in other words, is single acting; each piston having one pressure and one suction stroke during each revolution of the rotor. This imposes a great unbalanced hydrostatic load on the hydraulically acting surfaces of the cylinder barrel and great unbalanced mechanical load on the supporting bearings. Another disadvantage of such secondary rotor arises from the necessity for large supporting bearings for the eccentric mechanism, which bearings cannot operate properly at the necessary high speeds for high volume delivery at comparatively high pressures, due to the great centrifugal forces imposed on the rolling elements and associated cages.

The principal object of this invention is the provision of a hydrostatically balanced variable displacement pump wherein the hydrostatic pressure load is balanced about the valve pintle, and wherein the load imposed on the bearings of the piston barrel or primary rotor are likewise balanced.

Another object of the invention is the elimination of a secondary rotor and necessary large anti-friction bearings therefor to provide a cheaper and more practical construction for large pumps and motors.

A further object is to reduce the number of moving parts and rotating mass in pumps or motors of the type above referred to.

In pumps of prior design, pintle lubrication is ordinarily effected by the working pressure of the pump, consequently when the pump is running idle, or at low pressure, the pintle is not lubricated except at low pressure.

A further object of the invention therefore is in the provision of pressure lubricating means for the hydrostatically balanced valve in order to lubricate the pintle at all times with fluid under high pressure.

A further object is the provision of adjustable fully balanced stationary cam or reactance means to control the stroke of the pistons and the delivery of the pump.

A further object is the provision of a novel hydraulic control to operate the adjustable cam or reactance means of a pump of the class mentioned, to thereby change the stroke of the pistons and the delivery of the pump.

Further objects and advantages will appear from the following description relating to the accompanying drawings, showing a preferred form.

Referring briefly to the drawings:

Fig. 1 is a longitudinal horizontal central sectional view of a pump constructed according to the spirit of this invention;

Fig. 2 is a transverse sectional view of the pump taken on line 2—2 in Fig. 1, showing the pistons and the adjustable cam mechanism in its zero stroke position whereby the pistons revolve bodily in a circle but do not reciprocate;

Fig. 3 is a transverse sectional view similar to that of Fig. 2 with the parts in the same position, taken on line 3—3 in Fig. 1;

Fig. 4 is a sectional view similar to Fig. 3 with the piston operating rollers omitted and showing the adjustable cam mechanism in its maximum stroke position;

Fig. 5 is a longitudinal central sectional view at right angles to Fig. 1, taken on line 5—5 in Fig. 2, showing the pistons in their radially inward-most position;

Fig. 6 is a transverse sectional view of one of the main pump connections taken on line 6—6 in Fig. 5;

Fig. 7 is an enlarged detail sectional view of the piston operating yoke and roller assembly according to Figs. 1 and 5;

Fig. 8 is a fragmentary sectional view of a piston unit assembly showing the connection of one piston with the piston operating rollers and also showing the cooperative relationship between the rollers and respective cams;

Fig. 9 is a partial transverse sectional view taken on line 9—9 in Fig. 7;

Fig. 10 is partial sectional view taken on line 10—10 in Fig. 3, showing the bridging relationship of the movable and stationary parts of the piston operating cam mechanism when the cam parts are in idle position, according to Figs. 2 and 3;

Fig. 11 is a view similar to Fig. 10, showing the cam parts in maximum stroke position thereof;

Figs. 12 and 13 are detail sectional views taken on lines 12—12 and 13—13 respectively in Fig. 1;

Fig. 14 is a longitudinal central sectional view of one of two pilot valves of a hydraulic servomotor designed to actuate the aforesaid cam mechanism of the pump;

Fig. 15 is a sectional view thereof taken on line 15—15 in Fig. 14;

Fig. 16 shows the clevis connection between one of the pilot valves and associated linkage;

Fig. 17 shows a front elevation of the pump

equipped with the hydraulic servo-motor of Fig. 14, together with the operating toggle mechanism thereof designed to effect simultaneous adjustment of the piston actuating cams;

Fig. 18 shows in sectional plan the lever connections of the toggle mechanism, taken on line 18—18 in Fig. 17; and

Fig. 19 is a detail section of a modified relief valve.

Referring first to Figs. 1, 2 and 5, the general casing of the machine comprises heavy casing sections 1 and 2, the section 1 having a central cylindrical body portion 1' and a suitable end wall integral therewith, the end wall carrying a heavy integral hub 4 for supporting a valve pintle to be hereinafter described. The casing section 2 is appropriately secured as by a shoulder joint 3 to the cylindrical portion 1' of the casing member 1 completing the casing.

Secured in the hub 4 is a valve pintle 5 having an enlarged cylindrical portion 5a adapted to be securely fastened as by press fit into the central bore of the hub and a reduced portion 5b extending within the casing and having a working portion 5c hydraulically fitted into the central bore of a cylinder block or barrel 10, above referred to as the primary rotor. The rotor 10 is preferably a one piece forging and is provided with radial cylinders 11 communicating at their inner ends with the central portion 12 of the rotor which, as above stated, is in hydraulic fit relation to the working section or portion 5c of the pintle. The complementary working surfaces may be slightly tapered and the clearance between the working surfaces is very slight. The rotor also has an impeller shaft portion 13 projecting from one end through a suitable central opening in the casing member 2.

The support for the rotor comprises preferably spaced antifriction bearings 13 and 14 of equal diameters. Preferably, these bearings are of the double end thrust type (Timken taper roller bearings as shown). The inner race members 15 of the bearings are preferably pressed onto cylindrical reduced ends of the rotor against appropriate shoulders 16. The outer races 17 are preferably in slip fit relation to cylindrical shouldered recesses 18 in the respective casing sections 1 and 2.

The piston and cylinder assemblies comprise pistons proper indicated at 20 in the radial cylinders 11, the cylinders being preferably formed in a radially extending flange portion 21 of the rotor extending continuously about the rotor in the zone of the pistons. The cylinders 11 at their inner ends have reduced inlets 11a adapted for communication with four equally spaced ports 6 in the hydraulically fitted portion 5c of the pintle. The ports 6 communicate through respective axial passages in the pintle (not shown) with the radial main inlet and discharge ports 7 in the relatively enlarged portion 5a of the pintle. These main ports 7 respectively communicate with circular passages 8 and 9 in the casing hub 4 which, in turn, communicate with main fluid feed and discharge pipes 8a and 9a through suitable connections. The mains are used interchangeably for admitting and discharging the operating fluid depending upon the direction of rotation of the rotor (when used as a pump e. g.). To further insure the free running of the rotor at all times, there is preferably a relief space 24 provided between the pintle and rotor in the radial zone of bearing 14, to thus avoid binding of the pintle upon unequal expansion and contraction of the rotor and pintle due to heating of the bearing.

It is important to retain the slip fluid which passes the working surfaces of the pintle and rotor from the ports 6 rather than to allow the escaped fluid to pass freely into the outer surrounding casing. For this purpose I have provided a strong relief valve located in a reduced extension of the central rotor bore. This extension provides a fluid reception chamber 27 communicating with a central opening 25 in a valve seat member 28. A ball check 29 is pressed tightly against the central opening 25 by a suitable spring 30 bearing at one end in the end of the bore. The bore portion 26 which contains the spring is provided with radial passages 31 leading to the casing space 31' adjacent the bearing 13 as shown in Figs. 1 and 12. When the pump e. g. is operated at high pressure the slip fluid escaping from the working surfaces of the rotor and pintle accumulate in the spaces 25 and 26 and when the pressure becomes excessive the excess fluid escapes by lifting the ball 29. The fluid is retained under pressure in the said reception spaces and, whenever the working pressures of the pump become less, then fluid from the reception spaces travels in the reverse direction toward the supported end of the pintle, maintaining lubrication for the working surfaces during such low pressure periods or cycles of the pump. The bearing 13 is supplied with lubricating fluid from the passages 31 and the bearing 14 by fluid escaping from the working surfaces through relief space 24.

Fig. 19 shows a modified relief valve which may be used instead of the construction shown in Figs. 1 and 5. The parts that correspond to the relief valve parts of Fig. 5 e. g. are similarly numbered.

In Fig. 19 the ring 28 forming the valve seat member is closely fitted into its counterbore in the rotor but may slide in said counterbore (lap fit e. g.). The ring 28 at one end abuts a retaining member such as a threaded ring 28a against which the ring 28 is normally held by the spring 30 in its operation of holding the ball 29 in its seat. When pressure builds up in the effective fluid reception chamber 25—27 the fluid, rather than being permitted to escape by lifting the ball, will act on the relatively larger exposed transverse area of the ring 28 forcing the ring and ball to move bodily to the left until the ring abuts the surface 28b of the counterbore, whereupon should the pressure in the now enlarged but closed reception spaces continue to build up the ball will finally unseat to relieve the pressure into the chamber surrounding the spring 30. The released fluid may, of course, be suitably drained, say as shown in Fig. 5.

Now when the operating pressure of the machine is reduced to such a point that fluid is no longer forced between the hydraulically fitted working surfaces of the pintle and rotor toward the impeller shaft end of the rotor the spring returns the ball and ring 28 as a piston, effecting forced lubrication for the said working surfaces during such short periods as the machine is usually operated at greatly reduced pressures.

The piston actuating means, though variable in its action to reciprocate the pistons, is always bi-symmetrical on opposite sides of the rotor in any piston stroke setting of the machine, and said means effects two pressure strokes and two suction strokes of each piston on each complete rotation of the rotor. Thus the hydraulic load

on the pistons is always balanced on opposite sides of the pintle axis, and the hydraulic load between the pintle and barrel bore is likewise balanced on opposite sides of said axis. This also obviously balances the load on the main bearings 13 and 14 of the rotor.

The piston actuating means will now be described.

As shown in Figs. 2, 3 and 5, the cylindrical inner peripheral surfaces of the casing support segmental guide blocks 35, two at each side of rotor in spaced relation, one adjacent each flat inner end wall of the casing at respective sides of the rotor. The chordal surfaces 36 provide four parallel guides. The adjustable cam members 37 ride on these guide blocks, which latter are suitably secured rigidly to the casing by any convenient means (not shown), the blocks being spaced apart and connected by suitable bars 35' having reduced ends 35'' entering openings in respective blocks. The individual cam members 37 are C-shaped and each has outer and inner track forming surfaces 40 and 41 respectively forming guides for piston rollers e. g. 50 carried on suitable cross head members 51 of the pistons, to be later described.

The tracks afforded by the cam members 37 when the latter are in abutting relation, as shown in Fig. 3 form a complete true circle coaxial with the rotor and hence, in this position, impart no reciprocation to the pistons.

The cam members 37 are, as shown, four in number, being in pairs on opposite sides of the rotor between the guide blocks 35, the members of each pair being connected by suitable cross head bars 38 to which oppositely extending adjustment bars 38' are connected for shifting the cam members 37 equal distances from and toward the axis of the rotor as will be later described. The arm portions of the C-shaped cam members are slotted as at 42—see Figs. 10 and 11—to straddle straight tongues or ribs 43 rigid on the respective blocks 35, the tongues thus serving as rigid right line guides for the cam members 37 during adjustment. Additionally, the inner surfaces of the tongues are made exactly tangential to the outer roller track surfaces 40 so as to bridge these track surfaces as piston roller supports when the C-shaped cam members are spread apart (as in Fig. 4).

The rollers are, of course, long enough to extend into contact with the tongues 43 as well as the track surfaces 40 as shown particularly in Fig. 5. Thus adequate continuous outer roller guiding tracks are provided for the piston rollers in all adjustments of the cam members 37; no inner bridging tracks being necessary because the function of the inner tracks is solely to force the pistons outwardly on their suction strokes, and these strokes always begin at the position of the rollers shown in Fig. 3, i. e. aligned with both the inner and outer track surfaces.

Referring now to the piston roller supports 51 on the pistons, earlier mentioned, these, in order to reduce the diameter of the cam tracks and thus the peripheral speed of the rollers at any given rotor speed, comprise yokes pinned or otherwise securely fastened to the outer ends of the pistons, each yoke having a pair of arms 52 extending inwardly toward the rotor axis at opposite sides of the central rotor flange, the arms carrying the rollers 50 on laterally extending stub shaft portions 53 of the respective arms. To reduce torsional strain on the pistons and yokes the arms of the yokes may be slidably supported in radial ways 54 (see Fig. 8) formed in the central web or flange 21 of the rotor.

An important novel feature of the invention is the provision of needle roller supports for the rollers 50 on the cross head pins such as the stub shafts 52 of the yokes. The needles are very small and inexpensive and roll directly on undercut surfaces 55 which afford endwise retaining shoulders 55' for the needle elements. The needles, as shown, roll directly on the inner substantially cylindrical surfaces of the piston actuating rollers 50 and the rollers may be confined by the adjacent surfaces of the cam members 37 and by the yoke arms 52 against excessive endwise movement on the stub shafts. The needle elements 57 are entirely free for individual rotation and this may best be assured by taking out one needle roller 57 from each assembly after the races are full.

A further novel feature of the invention is the provision of means whereby the cam members 37 may be accurately and easily adjusted toward and away from the working axis of the machine, equal distances at each side thereof at all times in keeping with the provision of a truly hydrostatically and mechanically balanced pump or motor mechanism. To this end I provide servo-motors 60 and actuating toggle mechanism shown in Figs. 14 to 18 inclusive.

One of the servo-motors 60 is shown in section in Fig. 14, the servo-motors comprising double acting piston heads 61 (one being shown) in suitable cylinders 62, the operating rods of the heads extending into the main pump or motor housing through respective aligned openings 64 in the cylindrical central portion of the casing part 1, the rods being, in fact, the control rods 38', earlier described, which connect through suitable crossheads with the cam members 37.

The cylinders 62 are, as shown, one piece castings appropriately secured as by cap screws 65 to the pump or motor casing in sealed relation thereto. The chambers 70 and 71 at the opposite ends of the piston head 61 are fluid tight. A pilot valve 72 slides in a central bore 73 of each piston head and the operating stems 72' of the pilot valves extend from respective outer cylinder ends through suitable packing gland assemblies 74 (Fig. 14) to the operating toggle mechanism as shown in Fig. 17.

Centrally of the cylinder 62 operating fluid is fed to the cylinder under pressure through appropriate pipes 75 from any convenient source. Such fluid enters a peripheral channel 76 formed in the piston head as shown in Fig. 14 and from thence the fluid passes through passages indicated diagrammatically at 77 to a long peripheral channel 78 formed on the pilot valve. The cylindrical portions 79 and 80 of the pilot valve normally close radial passages 81 and 82 in the piston head which respectively communicate with axial bores 81' and 82' in the head leading to respective chambers 70 and 71 so as to convey the operating fluid to one or the other of said chamber depending on the setting of the pilot valve.

It will be seen that whenever the pilot valve is shifted toward the right (Fig. 14) fluid will be passed to the chamber 70 from 78 to 82' through passages 82, and the head will be moved to the right whatever distance the pilot valve has been so moved, and then such passage of fluid into the chamber 70 will be cut off by reason of the portions 72 of the valve blocking the passages 82. During the time the head has been so moved to the right the fluid remaining in the chamber

71 is vented to a central bore 85 of the valve by reason of cross-passages 86 (see Fig. 15) which, during the time the chamber 70 is being supplied, register with one of the passages 81 of the head. The servo-motor operates in exactly the reverse manner when the pilot valve is moved to the left (Fig. 14) except that the cross-passage 86 are then communicated directly with the chamber 70 to relieve the same as will be apparent from the drawings. The fluid discharged from the bore 85 may be conducted into the general pump casing 1—2 etc. from which it may be suitably drained as through an opening 88, Fig. 3, in the bottom of the casing.

To operate the pilot valves from a common point and thus assure the accurate setting of the cams 37 equal distances from the rotor axis, I provide levers 90—Fig. 17—pivoted on swingable links 91, the lower ends of which levers are attached as by suitable clevis-pivots 92, see also Fig. 16, to the outer ends of the pilot valve stems 72'. The levers 90 extend upwardly to toggle links 94 pivotally secured to the levers and which are pivotally secured to each other as by a pin 95 (Figs. 17 and 18) and to an operating head 96 in the nature of a fork slidably mounted in a vertical guide block 97 surmounting the machine.

The position of the levers and toggle links illustrated in Fig. 17 corresponds to the neutral setting of the machine wherein the pistons revolve idly but do not reciprocate. When the operating head is shifted from the illustrated position either up or down the upper ends of the levers 90 will be brought toward each other and the pilot valves withdrawn with respect to the cylinders 62, thus causing the servo-motors to correspondingly move the cam members 37 away from the rotor axis.

The great advantage of hydrostatically balancing the valve and cylinder mechanism whereby the valve is under no strain and consequently can be mounted floatingly in the casing, so as to find its own ways in the cylinder block bore without any strain, will be appreciated from the above description. In pumps of the hydrostatically unbalanced type the valve is always under a strain and one sided pressure, therefore a very rigid mounting in the casing is necessary for such valves and even with such very rigid mounting, operation is very often accompanied by rubbing, due either to mis-alignment or deformation of the parts under load. These faults are entirely eliminated from the present design and consequently greater safety factors and longer useful life are accomplished and obtained. Another main advantage of this structure is the elimination of large antifriction bearings and the substitution of capillary needle bearings for the rolling elements which are operated by the stationary cams.

It will be seen that by a 90° rotation of the cam mechanism, in either direction relative to the pintle, the delivery of the machine acting as a pump may be reversed, or the rotation as a motor may be reversed. Thus all the advantages of a single acting unbalanced type variable delivery pump are not only available in this design but, due to the fact that great hydrostatic pressures are balanced and thus great mechanical loads are eliminated for heavy duty service, and efficient service and durability are assured.

I claim:

1. In a hydraulic pump or motor of the rotary radial plunger type, including a casing, a rotatable barrel, and a plurality of plungers carried thereby, and adjustable reactance means therefor providing a continuous reactance track for the plungers in all adjusted positions of said means, said means including a pair of reactance elements having arcuate symmetrical reactance track portions and being arranged diametrically opposite to each other and being movable toward and away from the axis of rotation of the barrel, said reactance track portions defining an arcuate continuous track when the elements are adjusted to their innermost positions, and means to move said elements concurrently toward and away from each other while maintaining the elements in symmetrical relation with respect to the axis of rotation and for retaining the elements in said adjusted positions, said means comprising a pair of hydraulic mechanisms, said mechanisms being directly rigidly connected to the casing and each having an operating plunger, said plungers being diametrically opposite to each other and axially aligned with each other and connected directly one to each of said elements, and said mechanisms being connected in parallel relation to each other in a fluid pressure circuit, a common control mechanism for effecting simultaneous connection and disconnection of said mechanisms in said circuit, whereby the movement of each of said elements and the reactance resistance afforded by each is equalized.

2. In a hydraulic pump or motor of the rotary radial plunger type, including a casing, a rotatable barrel, and a plurality of plungers carried thereby, adjustable reactance means for the plungers and providing a continuous reactance track for the plungers in all adjusted positions of said means, said means including a pair of reactance elements having arcuate symmetrical reactance track portions and being arranged diametrically opposite to each other and movable toward and away from the axis of rotation of the barrel, said reactance track portions defining an arcuate continuous track coaxial with the barrel when the elements are adjusted to their innermost position, means to move said elements concurrently toward and away from each other while maintaining the elements with the track portions in symmetrical relation with respect to the axis of rotation and for retaining the elements in said adjusted positions, said means including a pair of equal size pilot controlled follow-up hydraulic servo-motor mechanisms, said mechanisms being directly rigidly connected to the casing diametrically opposite to each other and each having an operating plunger, said plungers being diametrically opposite to each other and axially aligned to each other and being directly connected one to each of said elements, and said mechanisms being connected in parallel in a pressure fluid circuit, a common control means for the said mechanisms for effecting corresponding equal operations thereof concurrently in either direction whereby the reactance forces on said elements are hydraulically balanced.

3. In a hydraulic pump or motor of the rotary radial plunger type including a casing, a rotor mounted therein, plunger and cylinder assemblies carried by the rotor, valve means for the assemblies, reactance means for the assemblies within said casing, and including arcuate reactance track elements arranged diametrically opposite each other with respect to the axis of rotation of the rotor and symmetrical with respect to each other, said reactance elements being adjustable relatively toward and away from

each other to different positions for varying the stroke of the assemblies, means in the casing supporting said reactance elements with their axes parallel to the axis of rotation of the rotor and for movement parallel to a fixed plane through the axis of rotation of the rotor into different adjusted positions, means to move said reactance elements to said adjusted positions, said last named means including a pair of pilot plunger controlled follow-up hydraulic servo-motor mechanisms, each mechanism including a main operating reciprocable piston, said mechanisms being rigidly connected to diametrically opposite sides of the casing with the main pistons thereof axially aligned with respect to each other and the axes of the pistons lying in said fixed plane and extending at right angles to the axis of rotation of the rotor, rigid rods coaxial with the said main pistons of said mechanisms respectively and directly connected to said reactance elements respectively at diametrically opposite sides of the axis of rotation, said mechanisms being connected in a common fluid circuit and a common control means for the pilots of said mechanisms for effecting concurrent operations thereof to effect, in turn, concurrent operation of the main pistons thereof.

4. In a hydraulic pump or motor of the rotary radial plunger type including a casing, a rotor mounted therein, plunger and cylinder assemblies carried by the rotor, valve means for the assemblies, reactance means for the assemblies within said casing, and including arcuate reactance track elements arranged diametrically opposite each other with respect to the axis of rotation of the rotor and symmetrical with respect to each other, said reactance elements being adjustable relatively toward and away from each other to different positions for varying the stroke of the assemblies, means in the casing supporting said reactance elements with their axes parallel to the axis of rotation of the rotor and for movement parallel to a fixed plane through the axis of rotation of the rotor into different adjusted positions, means to move said reactance elements to said adjusted positions, said last named means including a pair of pilot plunger controlled follow-up hydraulic servo-motor mechanisms, each mechanism including a main operating reciprocable piston, said mechanisms being rigidly connected to diametrically opposite sides of the casing with the main pistons thereof axially aligned with respect to each other and the axes of the pistons lying in said fixed plane and extending at right angles to the axis of rotation of the rotor, rigid rods coaxial with the said main pistons of said mechanisms respectively and directly connected to said reactance elements respectively at diametrically opposite sides of the axis of rotation, said mechanisms being connected in a common fluid circuit, a common control means for the pilots of said mechanisms for effecting concurrent operations thereof to effect, in turn, concurrent operation of the main pistons thereof, said common control means including a main control rod and balanced toggle connections between the main control rod and the pilots of said mechanisms for causing operation of said pilots in unison with each other.

5. In a hydraulic pump or motor of the rotary radial plunger type including a rigid casing, a rotor mounted therein, plunger and cylinder assemblies carried by the rotor, valve means for the assemblies, adjustable reactance means for the assemblies including a pair of separable elements having curved operating track portions, each track portion having an axis parallel to the axis of rotation of the rotor and said track portions being normally symmetrically arranged and diametrically opposite to each other with respect to the axis of rotation, bearing means in the casing mounting said elements for movement to different adjusted positions while constraining the elements to positions wherein their axes lie in a fixed plane through the axis of rotation of the rotor and parallel to said axis of rotation, means to move said elements toward and away from each other to different adjusted positions while maintaining the same with their track portions in said symmetrical relation, said means including a pair of equal sized pilot controlled, follow-up, hydraulic, double-acting servo-motor mechanisms, each mechanism including a main operating reciprocating piston, said mechanisms being connected to the casing with their respective pistons axially aligned and with the axis of each piston lying in said fixed plane and extending at right angles to the axis of rotation, rigid rods coaxial with the pistons of said mechanisms respectively and directly connected to said elements respectively for transmitting forces directly between said elements and associated pistons, said pistons being connected to the elements at diametrically opposite sides of the axis of rotation, said mechanisms being connected in parallel in a fluid circuit, and a common control means for said mechanisms for effecting equal operations thereof concurrently in opposite directions, whereby the reactance forces on said reactance means are balanced by the pistons of said mechanisms at all times and are equal and opposite with respect to each other.

6. In a pump or motor of the character described, a casing; a rotor rotatably mounted in the casing; radiating, co-planar piston and cylinder assemblies carried by the rotor; valve means therefor; reactance means surrounding the rotor and supported in the casing, said reactance means including a pair of adjustable separable load track members movable toward and away from each other, each load track member having an arcuate groove opening in a plane substantially transverse to the axis of rotation of said rotor and axially offset from the plane of said assemblies, said grooves providing inner and outer trackways, the inner and outer trackways of the respective load track members being symmetrically disposed with respect to the axis of rotation of the rotor; connecting outer load track means associated with the outer trackways of said members and forming therewith a continuous outer working contour in all adjusted positions of said members; and thrust-transmitting members respectively projecting laterally from the outer ends of said assemblies, and into the grooves in said load track members, and being in operative association with both the outer and the inner trackways of said members.

ELEK K. BENEDEK.