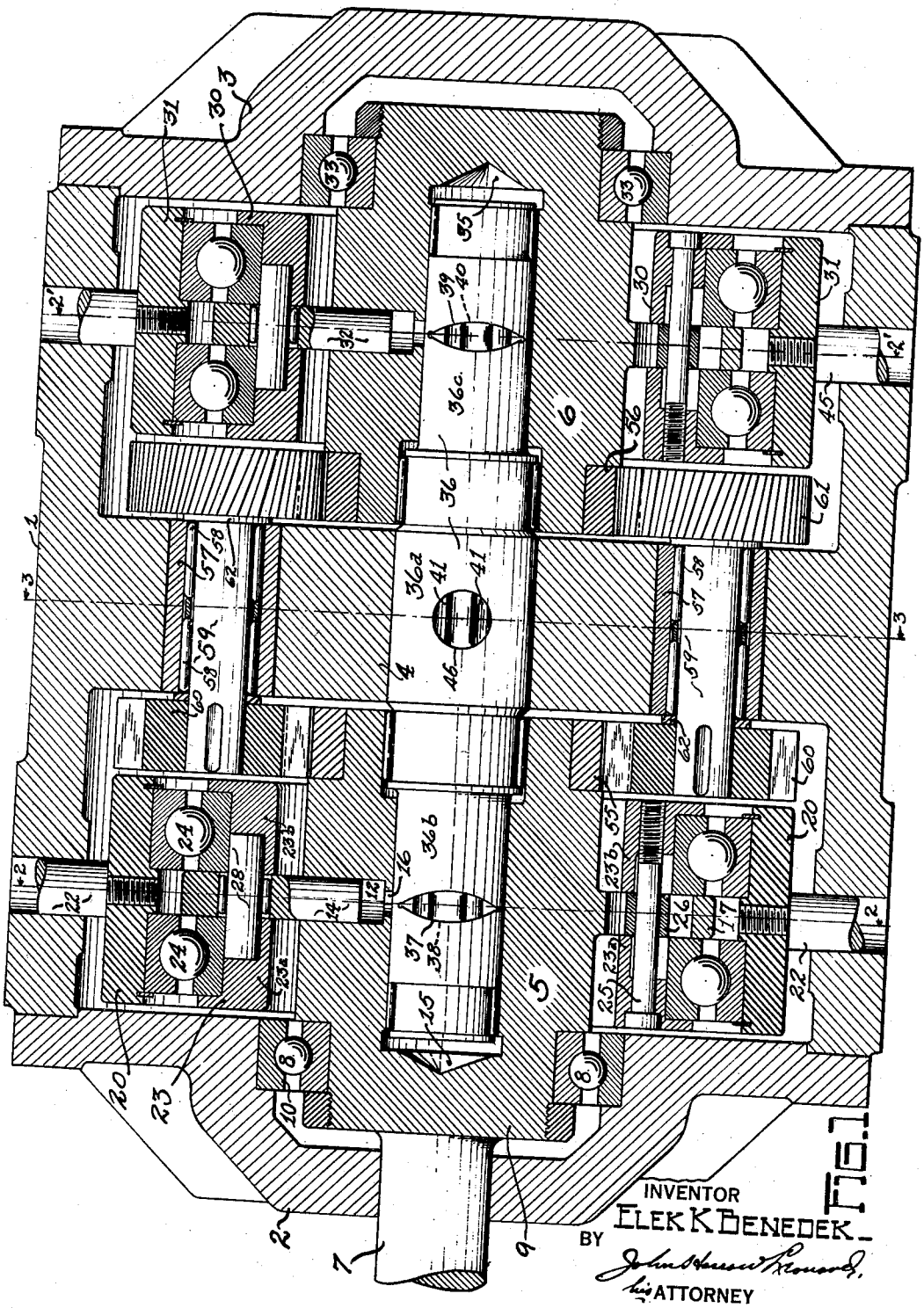
Sept. 12, 1939. E. K. BENEDEK 2,172,900

MULTISTAGE HYDRAULIC PUMP OR MOTOR

Filed Feb. 23, 1935 5 Sheets-Sheet 1



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FIG. 1

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MULTISTAGE HYDRAULIC PUMP OR MOTOR

Filed Feb. 23, 1935 5 Sheets-Sheet 2

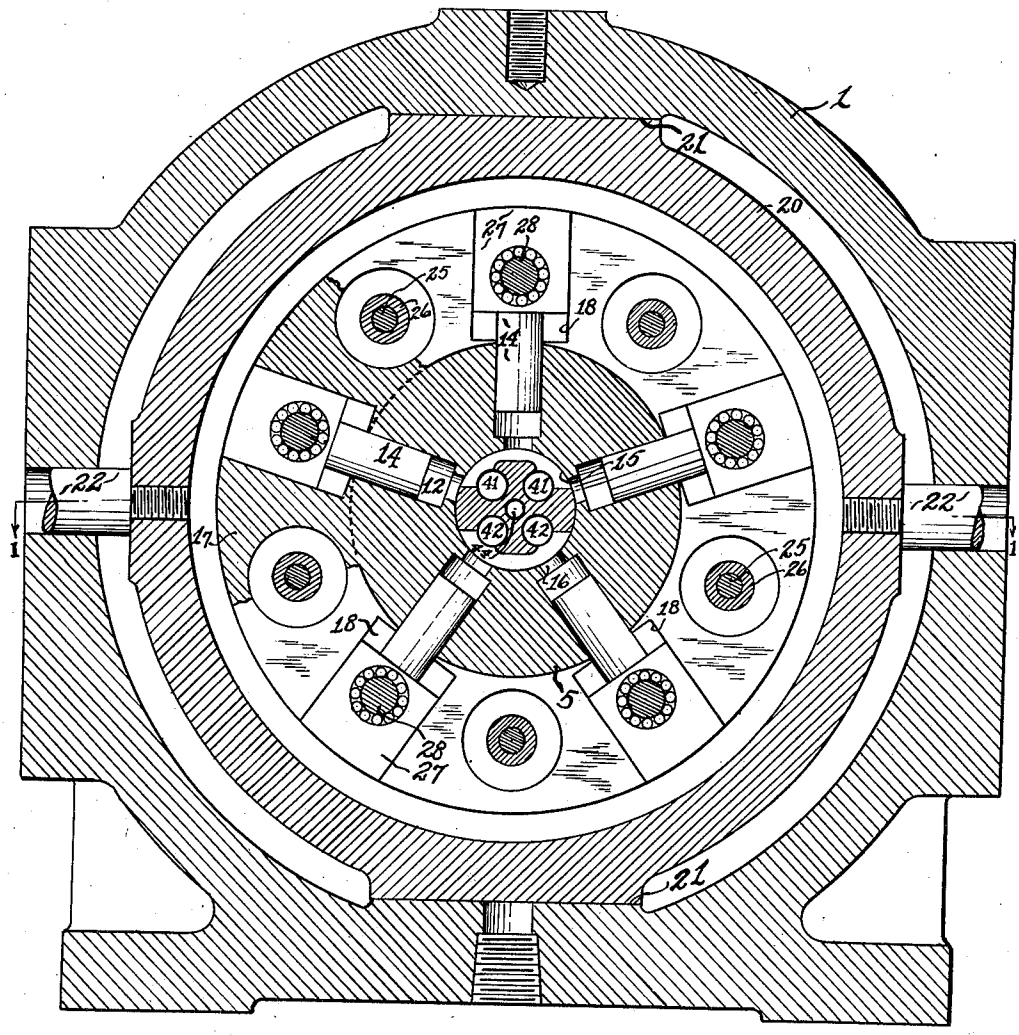


FIG. 2.

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MULTISTAGE HYDRAULIC PUMP OR MOTOR

Filed Feb. 23, 1935 5 Sheets-Sheet 3

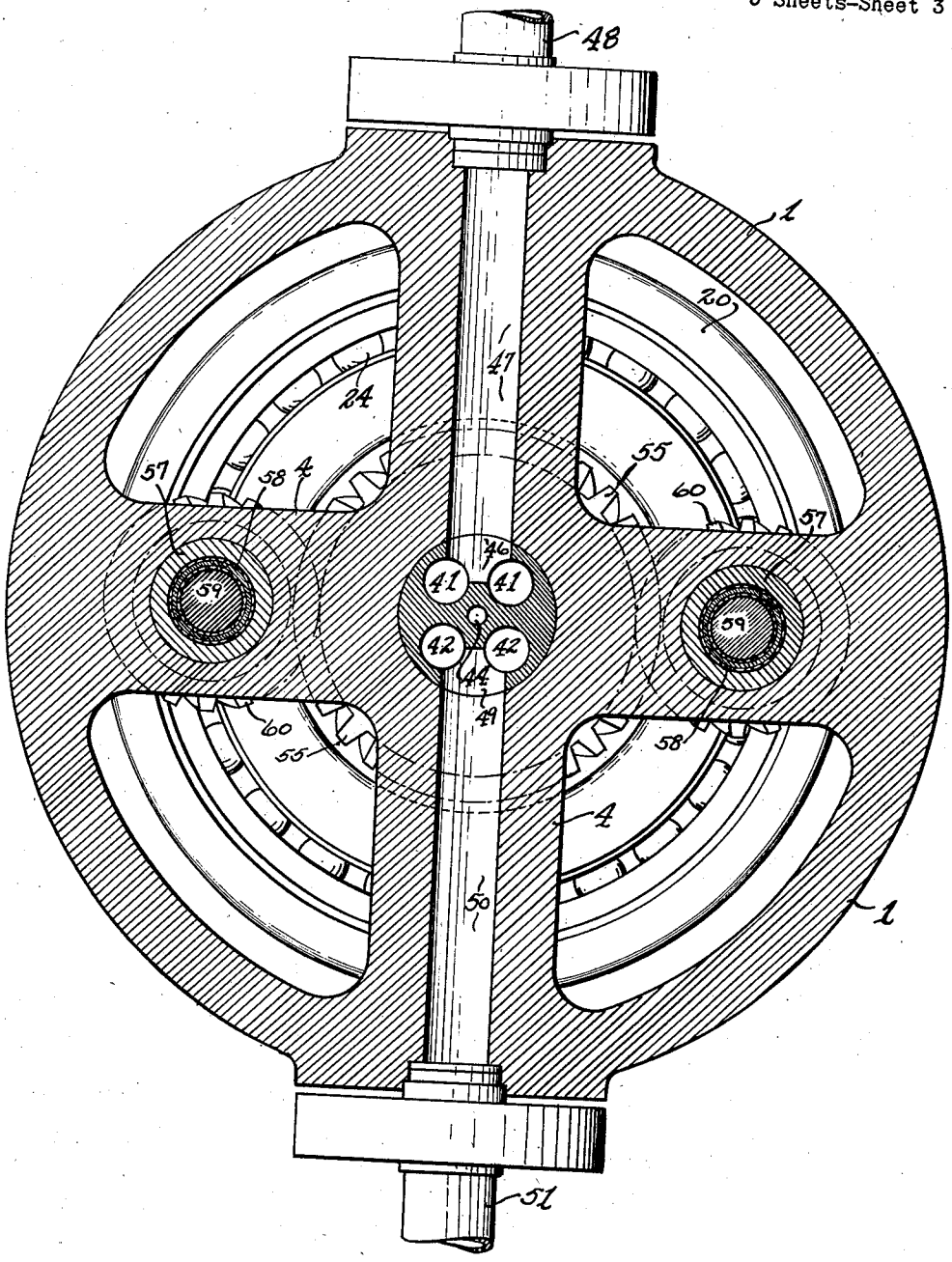


FIG. 3

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MULTISTAGE HYDRAULIC PUMP OR MOTOR

Filed Feb. 23, 1935 5 Sheets-Sheet 4

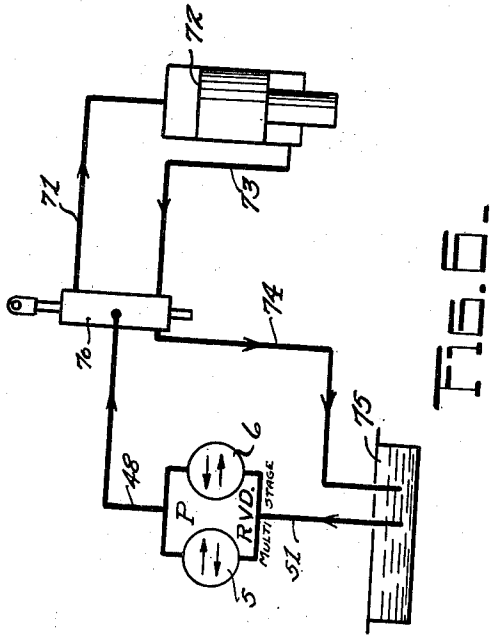


FIG. 6.

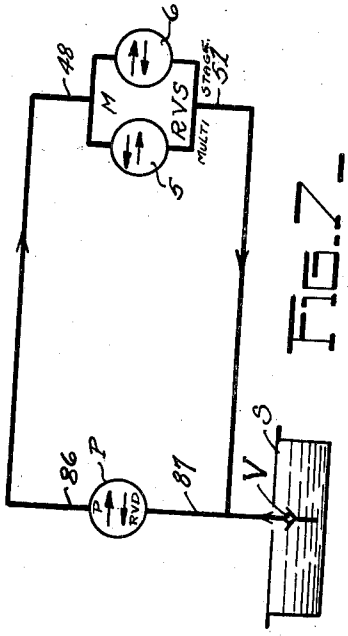


FIG. 7.

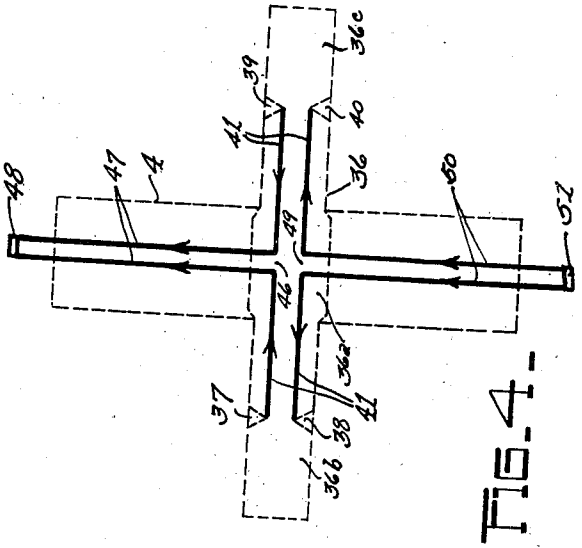


FIG. 4.

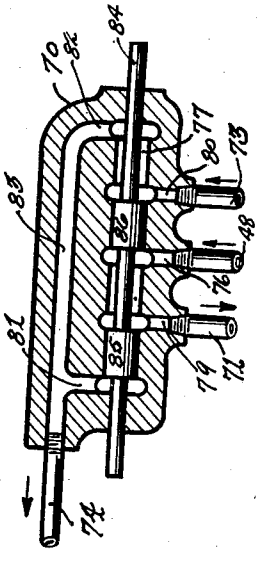


FIG. 5.

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MULTISTAGE HYDRAULIC PUMP OR MOTOR

Filed Feb. 23, 1935 5 Sheets-Sheet 5

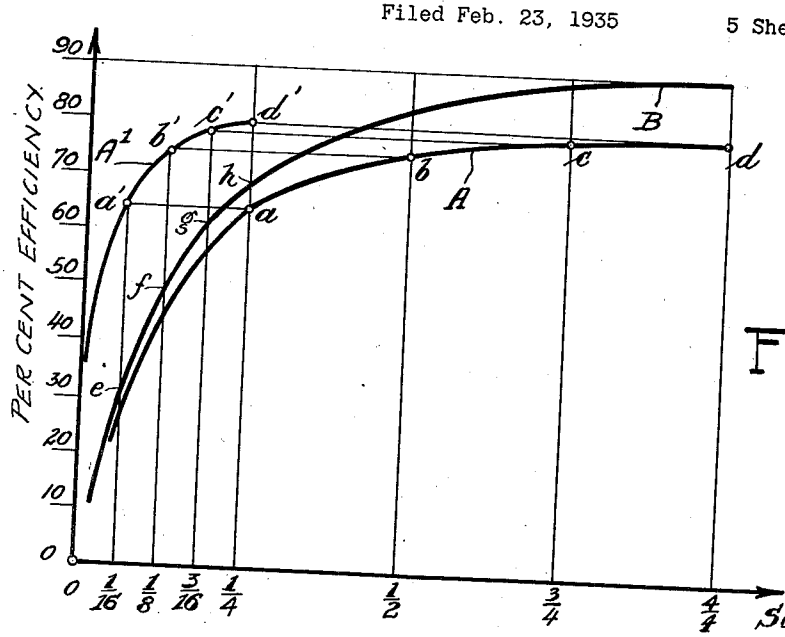


FIG. 8.

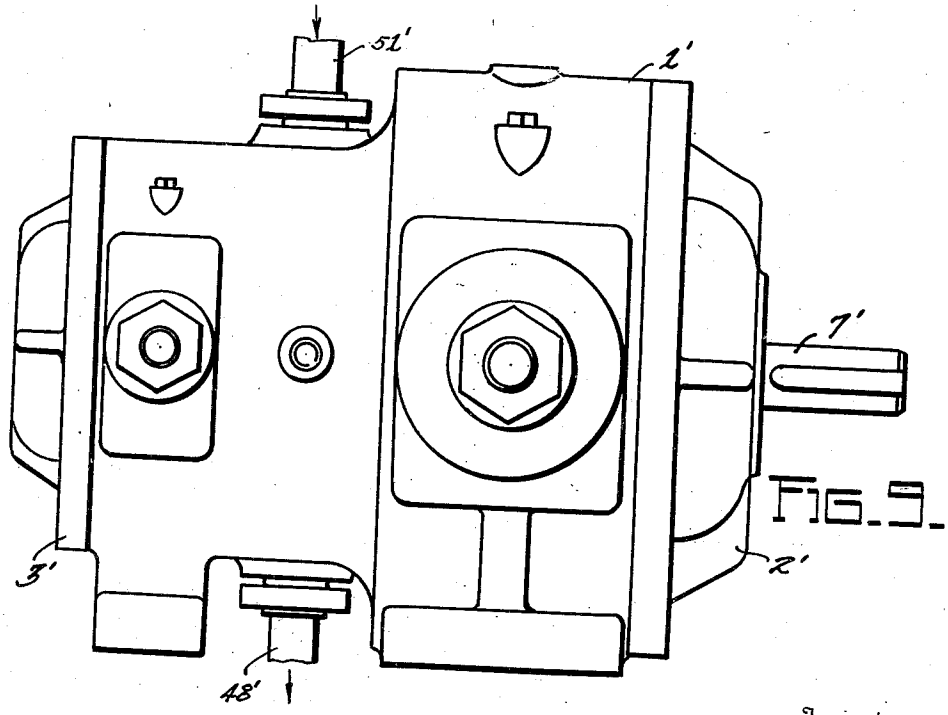


FIG. 9.

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Patented Sept. 12, 1939 2,172,900

UNITED STATES PATENT OFFICE

2,172,900

MULTISTAGE HYDRAULIC PUMP OR MOTOR

Elek K. Benedek, Bucyrus, Ohio

Application February 23, 1935, Serial No. 7,809

1 Claim. (Cl. 103—161)

This invention relates to radial piston type hydraulic pumps and motors and particularly to a compact and efficient multi-stage pump or motor unit of this character by which greater ranges of both speed and torque delivery and rapid changes in the relation therebetween may be obtained without the usual sacrifice in efficiency at the upper and lower limits of the range of operation and without appreciable loss of time between changes in the stages of operation.

The present practice in connection with hydraulic pumps and motors for comparatively wide ranges of operation is to provide two smaller pumping units instead of a single larger unit. This practice is due primarily to certain disadvantages inherent in large pumps and motors of this general type, which disadvantages are augmented in those instances in which the apparatus must operate throughout a wide range of torque and speed. Because of the extremely heavy pressures and high speeds of such pumps and motors, and the close fits required for efficiency between the various working parts any increases in size of the unit over a certain limit is accompanied by disproportionate decreases in the safety and reliability of operation and disproportionate increases in the wear of the working parts. As specific examples, the larger pumps require bearings and valve pintles of greater diameter, and heavier piston and rotor parts. Thus, with a larger diameter, not only is the peripheral speed of the bearings greatly increased for a given angular rotation, but also the centrifugal forces are disproportionately increased.

Further, due to the greater total hydrostatic load that must be resisted by the pintle, the greater overhang of the pintle that is required and the greater flow capacity which the pintle must accommodate, the pintle necessarily must be of considerably increased diameter in such larger pumps. This increase in diameter, in turn, results in much larger surfaces in frictional engagement between the wall of the rotor bore and the valve pintle. Furthermore, due to the larger diameter of the pintle, there is not only this greater frictional surface engagement between the pintle and rotor bore wall but also a greater peripheral speed between these surfaces for a given angular velocity of the rotor.

When operating at high speeds these frictional and centrifugal forces damage and wear the equipment, render proper lubrication very difficult, and requires a sacrifice in closeness of fit to compensate for greater expansion of the parts, all of which result in reduced efficiency of the pump.

If the large pumps are run at slower than their normal speed or if the displacement is lowered beyond a very limited amount, the capacity and efficiency decreases so disproportionately that much less horsepower output can be obtained for a given input by means of a larger pump than could be obtained by a smaller pump operating at a higher speed. Obviously, in the smaller pump not only are the frictional surfaces reduced but also the peripheral speed and centrifugal forces are likewise greatly reduced over those occasioned in the larger pumps operating at the same angular velocity.

Another inherent disadvantage of the larger variable delivery pumps for many applications resides in the sluggishness of the unit in unloading itself quickly. Assuming, for example, that a larger pump with necessarily less angular velocity is used and the eccentricity set at zero or greatly reduced, necessarily a longer time is required for a complete revolution. Even assuming the pump could completely unload in one revolution therefore it is apparent that a similar higher speed pump could unload more promptly. However, the eccentricity cannot be set so promptly but must be gradually changed to maintain a proper operating fluid pressure while unloading. Due to the much greater capacity even greater delay for unloading of the pump when shifting from one setting to another is occasioned in the case of the larger pumps, even though both the larger and smaller pumps are delivering the same total fluid capacity. As an example in many hydraulic press operations, a high speed is desirable for moving the press into contact with the work, and thereafter a much less speed but higher pressure is required for the forming operation. It is desirable, therefore, that a large volume of flow be provided for the idle stage of the press, followed, at the instant of engagement of the press with the work, with prompt unloading of the pump and setting of the pump at lesser eccentricity for building up and holding the higher pressure for the forming operation. This, of course, is important so as to obtain as many operations of the press per minute as possible. Otherwise, the pump cannot compete with the rapid changes in speeds and pressures provided by mechanical gearing. A smaller pump operating at the higher angular velocities obtainable therein, may be unloaded and the eccentricity changed for higher pressure with no appreciable loss of time.

The advantages of a relatively small and compact hydraulic unit of this type which can operate efficiently throughout an infinite variety of stages, thus providing a larger volume of flow at lower pressure followed almost instantaneously by a smaller flow but at very much higher pressure, as desired, are obvious, as such a unit makes possible the utilization of a much smaller driving motor, requires much less power for the complete operation, and is not subject to the disadvantages above pointed out as present in larger wide range pumps.

The present invention has as its principal objects the provision of a unit in which these disadvantages are eliminated and the provision of a pump or motor unit of this general type in which a wide range of pressures and volumes of delivery are obtainable while maintaining, at the same time, a relatively constant horsepower output and very high efficiency for both extremes of operation.

More specifically, an object is to provide a hydraulic pump mechanism capable of producing a substantially constant horespower output in the form of extremely high torque and low fluid delivery and as a lower torque and higher fluid delivery, while maintaining throughout the entire range of torque and delivery combinations even to its extreme limits, the greatest efficiencies inherent in pumps and motors of this general type.

Another object is to render possible the changeovers to and from large flow, low pressure cycles and the low flow, high pressure cycles, quickly and efficiently.

Another object is to provide a multi-rotor pump or motor mechanism combined into a single compact unitary structure in which the pintle diameter and resultant peripheral speeds between the pintle and the rotors is kept relatively small and the centrifugal forces greatly reduced.

Another object is to provide a great range of flexibility of speeds and pressures while maintaining a substantially constant horsepower output.

A further object is to provide a unitary structure including a plurality of separate pumping units cooperating with a single pintle and connected therethrough in parallel and to a common main delivery circuit.

More specific objects reside in the provision of means for maintaining the desired synchronous relation between the cooperating rotors of the unit and a means for resisting and balancing the axial thrusts on the rotors occasioned by the operation of the mechanism.

Another specific object is to maintain a cooperative relation between the cooperating rotors of the mechanism such that, at the higher speeds required for high pressure, low volume delivery, the kinetic energy of one of the rotors is utilized to eliminate vibration of the other and provide more even delivery of flow.

Other objects and advantages will become apparent from the following specification wherein reference is made to the drawings in which

Fig. 1 is a horizontal sectional view through a dual pump or motor mechanism embodying the principles of the present invention;

Fig. 2 is a cross sectional view of the mechanism taken on a plane indicated by the line 2—2 of Fig. 1 and also on a plane indicated by the line 2'—2' of Fig. 1, the rotor being slightly revolved from the position in which shown in Fig. 1;

Fig. 3 is a cross sectional view of the mechanism taken on a plane indicated by the line 3—3 of Fig. 1;

Fig. 4 is a diagrammatic illustration of the flow stream of the present mechanism;

Fig. 5 is a control valve for use in connection with the mechanism;

Fig. 6 is a diagrammatic illustration of an application of the mechanism in connection with a hydraulic press;

Fig. 7 is a diagrammatic illustration of an application of the mechanism as a motor in circuit with a single unit conventional reversible variable delivery pump;

Fig. 8 is a graphic comparison of the efficiency of the present unit and prior structures relative to the respective proportions of the full stroke of each; and

Fig. 9 is a reduced side elevation of a slightly modified form of the invention illustrating the utilization of different size rotors in the two units forming the combination.

Referring particularly to Figs. 1 to 3 inclusive, the invention is illustrated in connection with a structure which may be utilized either as a pump or motor. For convenience in reference, however, the description and claims will refer to the invention as a pump, the use as a motor being readily apparent therefrom. The word "pump" in the claim is meant to include the structure therein recited either as a pump or motor.

The pump comprises a main body comprising a rigid weight supporting casing 1, closed at the driving end of the pump by an end wall 2 and at the opposite end by a rigid weight supporting end wall 3. Within the casing 1, and preferably integral therewith, is a partition wall or spider 4 which divides the casing into two separate rotor compartments. Mounted within and coaxial with the casing and with each other are rotors 5 and 6, arranged at opposite sides of the partition wall 4. Since the rotors are the same in form and function, except, in some instances, for differences in dimensions, only the rotor 5 will be described in detail.

In the form illustrated, the rotor 5, which is driven by a main drive shaft 7 from a suitable source of power, is mounted near the driving shaft end in a suitable set of anti-friction bearings 8. The bearings 8 are arranged on an annular axially extending hub portion 9 of the rotor and cooperate therewith and with internal annular recesses 10 in the end cover 2 for resisting both radial thrusts and axial thrusts tending to move the rotor toward the end cover 2. The bearings 8 are sufficiently rigid to withstand radial rocking of the rotor.

The rotor is provided with a set of circumferentially spaced, radially extending, hydraulic cylinders 12 in each of which is carried a piston 14, reciprocable radially therein. The rotor is provided with a dead end axial bore 15 which is in communication with the cylinders 12 through rotor valve ports 16 opening from the cylinders into the bore 15. The rotor is provided, preferably midway between its ends, with a radially extending flange 17 in which are a plurality of circumferentially spaced radially extending slots 18 corresponding in number to the pistons 14, the walls of which slots are parallel to each other and to the axis of the piston with which associated.

Mounted within the housing 1, with its axis parallel to the axis of the rotor 5, is a reactance housing 20 having external guide sur-

faces, cooperable with complementary guideways in the casing 1, as indicated at 21 in Fig. 2, for supporting the housing 20 for movement thereof to different adjusted positions while maintaining the axis of the housing parallel to that of the rotor 5. Suitable control rods 22 are secured to the housing 20 and extend to the outside of the casing through appropriate bores in the casing for shifting the housing to dispose its axis coaxial with or eccentric and parallel to the axis of the rotor 5. The rods 22, in turn, may be operated by any suitable means such, for example, as a "servo-control" mechanism.

Rotatably carried within the reactance housing 20, and coaxial therewith at all times, is a secondary or reactance rotor 23 cooperable with the pistons for reciprocating the same consequent upon concurrent rotation of the rotor 5 and the reactance rotor 23. The reactance rotor 23 is mounted within the housing 20 preferably on anti-friction bearings 24, the races of which are spaced apart at each side of the midportion of the housing so as to accommodate therebetween the cooperating head portions of the piston assemblages. For convenience in manufacture, assembly and service, the reactance rotor 23 preferably is formed of two axially separable annular companion members 23a and 23b, drawn by bolts 25 against the outer ends of the inner race of the bearings 24 and the ends of suitable spacers 26 to form, in effect, a unitary structure, suitable openings, as illustrated, being provided in the flange 17 to accommodate the bolts 25 and spacers 26. For cooperating the pistons with the reactance rotor, each piston is provided with a crosshead 27 receivable in a corresponding one of the radial slots 18 in the primary rotor, which slots guide the cross-heads for radial reciprocation only and transmit thereto, along the guiding surfaces, the tangential forces resulting from operation of the mechanism. Each piston crosshead 27 is provided with a bore extending parallel to the axis of the rotor, which receives a crosspin 28, the crosspin protruding beyond the crosshead at both ends. The protruding ends of the crosspins are received in the reactance rotor 23, anti-friction capillary needle bearings being provided between each crosspin and the walls of the bore of the piston crosshead with which associated.

The rotor 6 corresponds, in form and function, to the rotor 5 and is correspondingly mounted. A secondary rotor 30 and cooperating shiftable rotor housing 31, for operating the pistons 32 of the rotor 6, are also mounted in the casing. The remote end of the rotor 6 is carried in anti-friction bearings 33 which cooperate with an internal recess in the end cover 3 for rotatably supporting the rotor 6 in a position coaxial with the rotor 5. As above mentioned, the rotor 5 is provided with a dead end axial bore 15, the open end of which bore is disposed toward the partition wall 4. The rotor 6 correspondingly is provided with a dead end bore 35 which opens toward the partition wall 4 and which is coaxial with the rotor and with the bore 15 of the rotor 5.

For effecting the proper valving relation between the two rotors and valve pintle means, a single stationary valve pintle 36 is provided, the pintle having a heavy shank portion 36a intermediate its ends by which it is fixedly secured in the partition wall 4. The pintle protrudes each way from the wall 4 coaxial with the bores 15 and 35, one end 36b of the pintle being received snugly in the bore 15 and the other end 36c being received in the bore 35. The pintle end 36b is provided with the usual valve portion cooperable with the corresponding portion of the wall of the bore 15 and having therein pintle ports 37 and 38 positioned to cooperate successively with the valve ports 16 of the cylinders upon rotation of the rotor 5. The end 36c of the pintle is provided with a valve portion cooperable with the corresponding portion of the wall of the bore 35 and carrying ports 39 and 40 cooperable successively with the valve ports of the cylinders of the rotor 6. As better illustrated in Fig. 2, the valve ports 37 and 39 of the pintle both cooperate with the same longitudinally extending ducts 41 formed in the pintle and the ports 38 and 40 communicate with the same longitudinal ducts 42 in the pintle.

As more fully described in my copending application Ser. No. 5,429 filed February 7, 1935, those of the ducts 41 and 42 which constitute the pressure ducts of the pintle in a given operation of the pump, may extend the full length of the pintle and be open at the free ends of the pintle so as to discharge into the dead ends of the respective bores of the rotors or, if desired, a balancing duct 44 may be provided in the pintle. The duct 44 extends entirely therethrough axially for maintaining equal hydrostatic pressures in the bores 15 and 35.

Each protruding portion of the pintle is provided both at its shank end and free end with anti-friction capillary needle bearings cooperating with corresponding internal bearing surfaces in the rotor bore with which the pintle is associated for resisting hydrostatic thrusts on the pintle and maintaining it in proper coaxial and spaced relation to the walls of the bore with which it is in valving cooperation. Each protruding portion of the pintle likewise terminates in slightly spaced relation to the dead end bore with which associated so that lubricant or fluid passing from the bore onto the valving surface of the pintle and the bearings is discharged thereover as an annular stream for maintaining a film of fluid on the pintle and bearings. Furthermore, each of the bores is counterbored at the open end so that a large shank portion of the pintle may protrude thereinto as close as practicable to the valve portion so as to insure the utmost rigidity of the pintle and provide as great strength as possible adjacent the valve surface to withstand hydrostatic thrusts, all as more fully described in my copending application, Ser. No. 5,429, filed February 7, 1935.

The secondary reactance housing for the rotor 6 may likewise be shifted through a medium of suitable rods 45 for varying the eccentricity of the reactance rotor 30 of the rotor 6 independently of or in predetermined fixed relation to that of the rotor 23.

The pintle shank portion 36a which is received in the wall 4 is seal fitted thereinto and is provided with a duct 46 with which both of the ducts 41 communicates. The partition wall 4 is drilled to provide a bore 47 directly therein and communicating at its inner end with the duct 46 and, at its outer end, with the main pressure line 48 of the mechanism. The shank 36a is also provided with a duct 49 which communicates with the ducts 42, the partition wall being provided with a duct 50 which communicates at its inner end with the duct 49 and at its outer end with the main inlet or feed lin of the mechanism.

It is apparent that if the rotors are operated simultaneously with their secondary reactances

disposed eccentrically in the same direction from the common rotor axis both will supply fluid under pressure to the line 48. If, however, either the secondary reactance 23 or the reactance 30 were disposed coaxial with the rotor with which associated only one of the rotors would act as a pump and the other one would be idle.

In order to drive the rotors in the predetermined timed relation with respect to each other so as to effect the particular result desired, the rotors are connected together by mechanical gears. Referring to Fig. 1, the rotor 5, at the end adjacent the wall 4 is provided with an external helical gear 55 which preferably is positioned axially of the rotor to overlie the needle bearings adjacent the shank portion of the pintle. The rotor 6 is correspondingly provided with a helical external gear 56 similarly arranged. Mounted within the partition wall 4 are circumferentially spaced bushings 57 which extend parallel to the axis of the pintle. The bushings are spaced equidistant from each other and each carries longitudinally spaced sets of capillary needle bearings 58 through the medium of which a driving shaft 59 is rotatably supported in each bushing. Each shaft 59 protrudes at each end beyond the wall 4 and is provided at its respective ends with helical gears 60 and 61 which cooperate respectively with the gears 55 and 56 of the rotors for effecting the driving relation therebetween. The helical gears are so pitched that the reactance thrust on the rotors due to the driving therethrough is in a direction opposite to the thrust on the rotors due to the hydrostatic pressure in the respective dead end bores thereof so that the rotor bearings are relieved of as much reactance stress as possible. Thrust washers 62 are provided between each gear and the wall 4 for maintaining the gears and shafts 59 and needle bearings in proper axial position. Any number of such gear connections between the rotors may be provided, two being shown herein for purposes of illustration. The sets of gears should be spaced circumferentially equidistant from each other, as stated above, so that the driving stresses imparted by the gear teeth shall be in mechanical balance with reference to the pintle and will not tend to deflect it radially due to any mechanical unbalance. This balance can readily be effected by arranging the sets of gears so that for each set there is a diametrically opposite set. By "helical" gears I means generally gears whose teeth are directed other than axially, in other words, are so directed that the reactance thrust on the gears will produce an axial thrust component.

The rotors may be driven at the same speed or at different speeds depending upon the particular gear relation provided and the particular requirements of the particular job or function to be performed by the pump mechanism. For example to drive the rotor 6 at a greater or less speed than the rotor 5, it is only necessary to provide a larger gear on one rotor and smaller gear on the corresponding end of the shaft 59. For illustrative purposes, the gears 60 and 61 and the cooperating gears 55 and 56 are shown as cooperated for driving the rotors 5 and 6 at the same angular velocity.

Having in mind that a smaller pump at high speed is generally more efficient than a larger pump of the same delivery at low speed, it is obvious that with a given power input through the medium of the shaft 7 and with the same high eccentricity of the secondary reactances 23 and 30, as large a volume of fluid can be delivered to the main pressure line 48 providing the pressure is not too great as with a larger pump. If merely the eccentricity of both rotors were decreased, a higher pressure but less volume could be obtained. If this increase were continued, the pressure could be built up but when a very low eccentricity was reached, there would be a great decrease in the efficiency of the pumps. Assuming therefore, that a large volume and low pressure were desired, both units could be set for maximum stroke and the large volume of flow at low pressure delivered. Assuming that this flow must be followed by a prompt increase in pressure, then the eccentricity of the rotor 6 could be set to zero and that of the rotor to less eccentricity than previously operated but not so low as would be necessary were both pumps operating. With a constant power input and idling of the rotor 6, it is apparent that the rotor 5 could be driven at much greater speed and with a resonably large proportion of its maximum stroke. This would provide a much higher efficiency than were both the rotors driven at extremely low eccentricity. Since both pumps are small, it is apparent that they can be operated at higher angular velocity than a single larger pump of equal capacity, and the one which is idled can readily be unloaded from its oil supply much more quickly than could a larger pump. Furthermore, since the greatest eccentricities of each of the two smaller pumps is less than would be provided in such larger pump, the time required for setting the reactance rotors to the new positions for high pressure is very much shorter, in fact, so short as to require no substantial delay.

Another feature that should be pointed out is that high speed and high pressure often create in a pump of this character a certain tendency toward vibration, due to the time elapsing between cooperation of one cylinder port and the cooperation of the next succeeding cylinder port with the same pintle port. This is due somewhat to the fact that the mass of the motor and cooperating parts is reduced to a minimum commensurate with the work to be done. Accordingly, much less kinetic energy is stored in the pump than would be the case were a large rotor used. Assuming the rotor 5, however, is operating at high speed and the rotor 6 is idling, the rotor 6 provides additional mass with resultant kinetic energy for carrying the rotor past the peaks and troughs of the pressure delivery and renders the operation more uniform. When a large and small rotor are used in combination, suitable check valves may be provided in the pintle ducts to protect the larger rotor from the high pressures developed by the smaller, and thus reduce the cost of the larger pump and insure full efficiency of the smaller.

It is apparent that many combinations can be obtained by changing the driving gears between the rotor 5 and the rotor 6. The range can readily be extended further by the utilization of a structure such as illustrated in Fig. 9, in which the rotors are of different sizes.

Referring next to Figs. 4 and 5, the flow of fluid in the unit, when operating as a pump, is illustrated diagrammatically. Assuming both rotors are driven at the same speed and are of the same size and same eccentricity for each rotation, fluid under pressure is introduced into the pintle concurrently through the ports 37 and 39 and passes through the pintle into the common feed line 48. Correspondingly, fluid is drawn

through the feed line 51 to the ports 38 and 40 concurrently and again discharged when the corresponding pistons again reach the ports 37 and 39. Accordingly, a fluid delivery of twice the capacity of either rotor may be obtained with the rotors operating at high speed. Assuming, however, that the rotor 6 is so set that its eccentricity is zero it then acts as a fly wheel for the rotor 5, and the only fluid delivered is delivered through the port 37.

It is apparent that a single rotor may be run at much higher speed with moderate eccentricity to obtain an extremely high pressure instead of both rotors being run at a lower speed or at an eccentricity so low that the efficiency of both is greatly reduced. Further, short stroke efficiency of both large and small pumps is less than full stroke efficiency. This effect can be more readily appreciated by reference to Fig. 8. In the graph therein illustrated, the abscissa represents the proportional part of the possible stroke or eccentricity, the percentage of efficiency being represented by the ordinate. In the graphical representation the curve B represents the efficiency curve of a pump or motor having a flow capacity about four times that shown by the curve A. The courve A_1 is the same as A but is brought into capacity relation to curve B. Referring to curve B, it is apparent that when the eccentricity if $\frac{1}{16}$ of the total, the efficiency is only 30% as indicated by the point *e*; for eccentricity ⅛ the efficiency is about 48% as indicated by the point *f*; for $\frac{3}{16}$ eccentricity the efficiency is about 61% indicated by the point *g* and for ¼ eccentricity the efficiency is about 68%. Not until the eccentricity is about half the total does the efficiency reach 80%. While delivering the same capacity of flow, at these lower limits, a smaller pump, indicated by the curve A_1, will reach about 63% efficiency for a capacity at which the larger pump would be operating at about 30%. It is obvious to reference to the points *a'*, *b'*, *c'* and *d'* that the efficiency of the smaller pump increases far more rapidly for capacities equal to those delivered by the larger pump when the latter is operating at less than 50% eccentricity. Therefore two smaller pumps operating at higher eccentricity would be much more efficient than a large pump of capacity equal to the two and operating at ¼ or less eccentricity. Furthermore, by idling one pump, the total input of horsepower can be applied to the other and operate it at an eccentricity commensurate with high efficiency while maintaining a low flow, high pressure delivery.

As one exemplary use of the invention, it is diagrammatically illustrated in Fig. 6 for use in connection with a hydraulic press in which a large flow and corresponding movement of the press is required preparatory to engagement with the work and a much higher pressure but less flow is required for the forming operation. It is apparent that with this structure both the rotors 5 and 6 can be operated concurrently to provide for the initial movement of the press until the press has engaged the work. Thereafter the rotor 6 can be set at zero eccentricity and the high pressure for the forming operation performed by the rotor 5. In this operation the flow from the unit passes along the main line 48 into a control valve 70 from which it passes into the feed line 71 for supplying pressure to the piston 72 of the press. Fluid in front of the piston concurrently discharges through the line 73 and through the valve 70 and line 74 to a feed sump 75 from which it is drawn through the line 51 into the unit. The control valve 70 is better illustrated in Fig. 5. The valve 70 comprises a body having a main inlet 76 connected to the main feed line 48 of the mechanism for supplying fluid under pressure thereto. Within the body is a main bore 77 with which the passage 76 communicates. Likewise communicating with the bore 77 are passages 79 and 80, these passages opening into the bore 77 at opposite sides of the opening of the passage 76 thereinto. The passage 79 communicates with the line 71 leading to the working side of the piston 72 and the passage 80 communicates with the line 73. At the ends of the bore 77 are ducts 81 and 82 which communicate with a main trunk line 83 which, in turn, communicates with the line 74. Mounted in the 70 is a valve rod 84 on which are carried sleeve valves 85 and 86 snugly received with a hydraulic fit in the bore 77 and spaced apart from each other longitudinally thereof. The sleeve valves 85 and 86 are of sufficient diameter to close that portion of the bore 77 in which received. The passages 76, 79 and 80 are spaced apart equal distances, such distance being slightly greater than the lengths of the sleeve valves 85 and 86. Obviously, when the valve is set in the position illustrated in Fig. 5, fluid delivered from the feed line 48 can flow only to the line 71 leading to the pressure side of the piston. The fluid at the opposite side of the piston discharging through the line 73 enters the valve through the passage 80 and can pass through the ducts 82 and 83 to the discharge line 74, the valves 85 and 86 preventing passage of fluid from either passage 76 or 79 into the duct 83. To reverse the piston 72 the valve stem is pulled to the right, this operation communicating the line 73 with the line 48 thus forcing the fluid to the lead side of the piston. Concurrently line 73 is prevented from communication with the duct 83 and the passage 76 is prevented from communication with the passage 79. The fluid on the force side of the piston 72 therefore is discharged through the line 71 and through the duct 81 into the main trunk 83 from which it passes into the line 74. It is apparent that by using a Servo-motor, it is a simple matter to control the eccentricities of the two pump units herein shown in response to actions of the press, a simple lever connection between the press and Servo-motor being provided to idle the rotor 6 when the press engages the work for the forming operation.

Referring next to Fig. 7, the operation of the unit as a motor is illustrated, fluid under pressure being supplied from a suitable sump S to a reversible variable delivery pump P, the pressure line of which is designated 86, the suction side being designated 87. Fluid, under pressure therefore is forced from the pump into the passage 48 of the unit and thereby drives the rotors 5 and 6 concurrently and is discharged therefrom through the pasage 51 which, when the unit was operating as a pump, was the suction passage. From the passage 51, in turn, the fluid will be conducted to the suction 87 line of the pump. A check valve V is provided in the line 87 between the sump and juncture with the line 51 to assure an adequate supply of fluid and thus replace any lost by slippage through the device. Instead of the pump P, a multi-stage reversible variable delivery pump, the same as the motor of Fig. 7, may be employed, in which combination the effective range and possible variations are infinitely increased.

It is apparent therefore that with the unit described and the variations set forth, an extremely wide range of flow and pressure and a wide range of combinations can be provided from a substantially constant horsepower input and that the total horsepower input can be translated either into large volume of flow or high pressure at a substantially constant horsepower output and high efficiency throughout, the only losses occurring being those inherent in hydraulic pumps and motors even when operating at their highest efficiency. While I have described, for purposes of illustration, the use of the present invention in connection with a reciprocatory press operated at various stages, it is readily apparent that great advantages can be obtained with the present pump or motor as a constant horsepower transmission device in numerous kinds of machinery, such as automobile transmissions, machine tools, excavators, cranes and in such apparatus as utilizes Diesel power engines as prime movers. The advantages reside largely in the fact that in these various types of apparatus the efficiency of the prime mover itself is developed to the highest if it can be operated at its rated speed constantly.

I claim:

A multi-stage hydraulic pump or motor comprising a pair of coaxial rotary radial piston stage units, each unit including a rotor having a closed end valve bore, piston and cylinder assemblies for each rotor, reactance means for the asemblies respectively for actuating the same, means rotatably mounting said rotors in spaced relation axially from each other and with the valve bores coaxial and opening toward each other, a rigid support between the rotors, a valve pintle fixedly secured in said support and extending in opposite directions therefrom into the respective bores of the rotors and being in valving relation with the respective asemblies, whereby hydrostatic reactance thrusts result in said bores urging said rotors axially apart, a rotatable shaft mounted in said support, helical gears respective to the ends of said shaft and fixedly secured thereon, cooperating gears on the rotors respectively for drivingly connecting the rotors together mechanically, and said gears being helically pitched to cause reactionary thrusts urging the rotors axially toward each other, and impeller means for one of said rotors and coaxial therewith.

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