Aug. 31, 1954 J. A. MORROW 2,687,615

RADIAL TYPE HYDRAULIC PUMP

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Fig. 1.

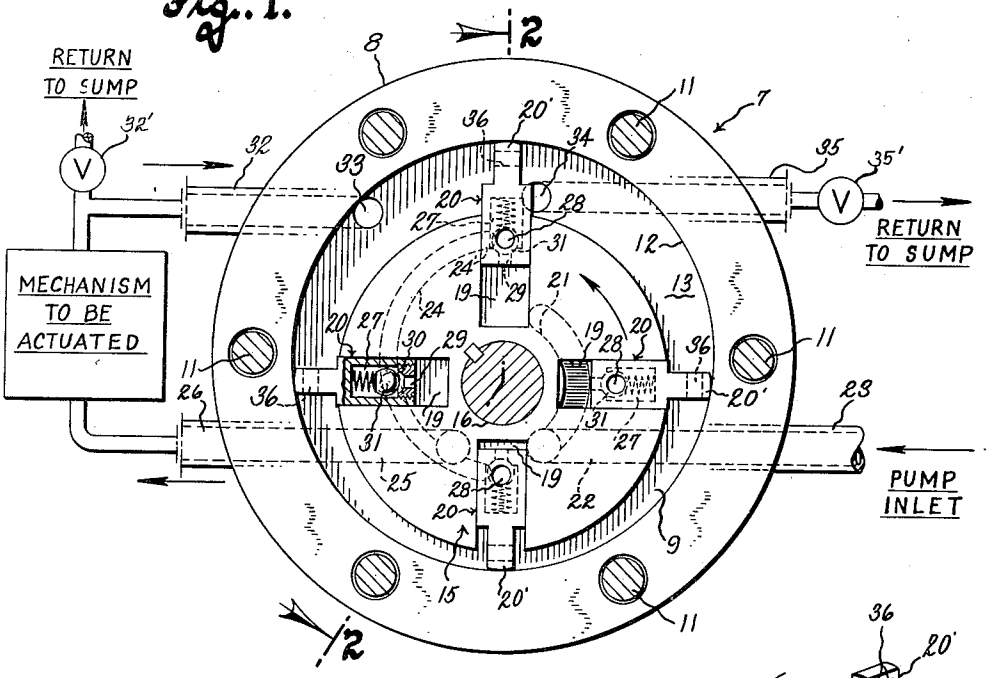


Fig. 2.

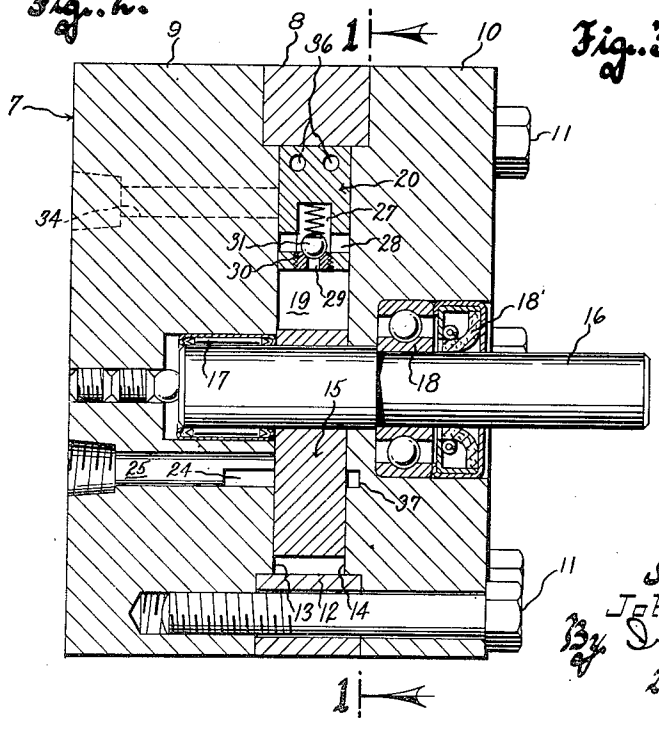
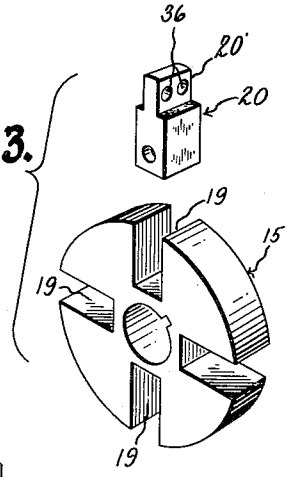


Fig. 3.



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Fig. 4.

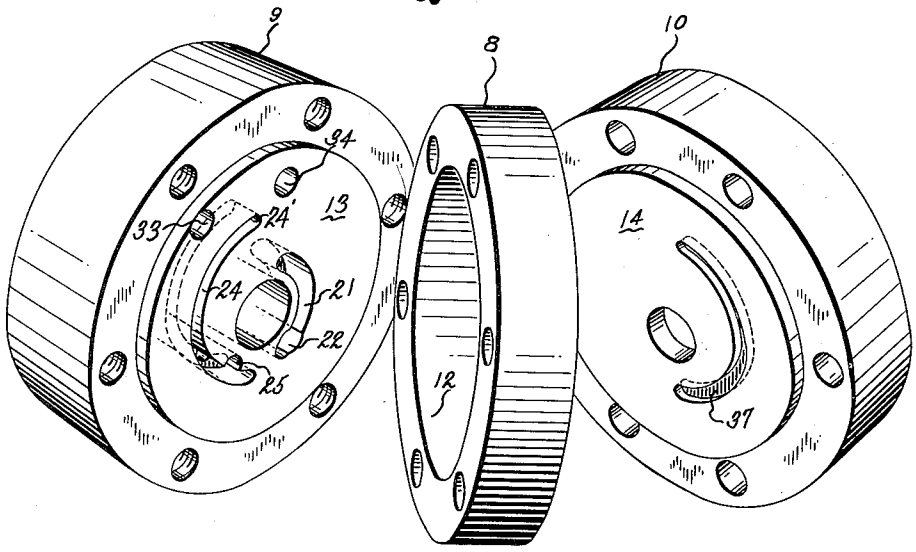
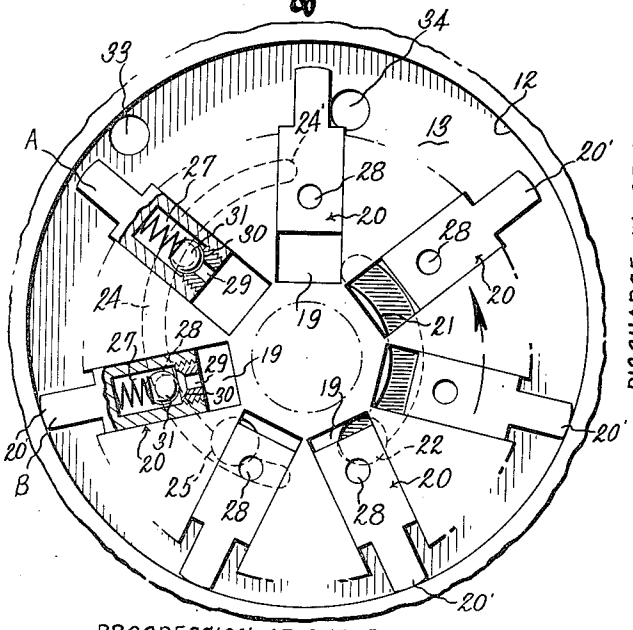


Fig. 5.



PROGRESSION OF ONE PLUNGER IN ITS CYCLE

Fig. 6.



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RADIAL TYPE HYDRAULIC PUMP

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Application February 21, 1952, Serial No. 272,841

8 Claims. (Cl. 60—52)

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This invention relates to hydraulic pumps and refers more particularly to what might be characterized broadly as a radial type pump, but wherein the pumping action is derived not from the rotary movement of vanes along with the rotor but rather from the successive radially inward depression of a series of radially disposed pistons or plungers carried by the rotor.

In pumps of this type the pistons or plungers are slidably received in pockets formed in the rotor and project from the outer ends of these pockets to bear against the inner cylindrical wall of a pumping chamber in which the rotor revolves eccentrically to the chamber. Radially outward projection of the plungers draws the fluid to be pumped into the plunger pockets through an appropriately located inlet port and inward depression of the plungers expells the fluid from the pockets into a discharge port.

The patent of I. T. Dyer No. 559,534 illustrates a pump operating upon this general principle.

Pumps which operate upon this principle have the advantage of compactness and comparative simplicity, but heretofore were not suitable for service which requires a constant volumetric output despite substantial changes in the speed of the pump rotor. This condition is met, for instance, in hydraulic boosters for the steering gears of automobiles and other vehicles where the pump which supplies the hydraulic pressure, being driven from the engine, varies in speed and runs at low speed when the need for the booster is greatest and at high speed when little help is required from the booster.

This invention has as its purpose to adapt the so-called radial pump to service such as that demanded by hydraulic steering boosters and thus has one of its objects to provide a pump of the chaarcter described which has an effective output at a relatively low rotor speed and in which the output remains substantially constant over a wide range of rotor speeds.

This result is achieved with the present invention by so balancing the forces acting upon the plungers or pistons that for a relatively wide range of rotor speeds the quantity of fluid drawn into the plunger pockets diminishes proportionately as the speed of the rotor rises so that the pump has substantially the same volumetric output per unit of time throughout the given range of rotor speeds.

With reduction in the size of the charges taken into the plunger pockets, the plungers, during part of their travel about their circular orbit, leave the cylindrical wall of the chamber

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against which they would otherwise bear. Re-engagement of the plungers with the circular chamber wall if it were allowed to be an abrupt impact could be a source of highly objectionable noise. In recognition of this undesirable possibility this invention has as another of its objects to provide means for assuring gradual and gentle re-engagement between the plungers and the cylindrical chamber wall under all operating conditions.

With the above and other objects in view, which will appear as the description proceeds, this invention resides in the novel construction, combination and arrangement of parts substantially as hereinafter described and more particularly defined by the appended claims, it being understood that such changes in the precise embodiment of the hereindisclosed invention may be made as come within the scope of the claims.

The accompanying drawings illustrate one complete example of the physical embodiment of the invention constructed according to the best mode so far devised for the practical application of the principles thereof, and in which:

Figure 1 is a cross sectional view through a pump embodying this invention, said view being taken on the plane of the line 1—1 of Figure 2;

Figure 2 is a longitudinal sectional view through the pump taken on the plane of the line 2—2 in Figure 1;

Figure 3 is a perspective view of the pump rotor and one of its plungers;

Figure 4 is a perspective view of the elements of the pump casing shown separated but in their proper order of assembly;

Figure 5 is an enlarged view more or less diagrammatically illustrating the progression of motion of one plunger in its pumping cycle at a rotor speed at which the plunger is not projected its full distance; and

Figure 6 is a chart generally illustrating the volumetric output of the pump at different rotor speeds.

Referring now particularly to the accompanying drawings in which like numerals indicate like parts, the numeral 7 designates the casing of the pump which may, of course, be constructed in any suitable manner but in the present instance comprises an annular midsection 8 and two disc-like end sections 9 and 10, the three sections being held together by cap screws 11. This casing defines a cylindrical chamber having a cylindrical wall 12 and flat end walls 13 and 14.

Within the cylindrical chamber is a rotor 15

keyed to a drive shaft 16. The shaft 16 is journalled in coaxial bearings 17 and 18 in the end sections 9 and 10, respectively, and so disposed with respect to the axis of the cylindrical chamber that the rotor revolves eccentrically thereto. As a result the radial distance between the rotor axis and the cylindrical wall 12 increases for half a rotor revolution and decreases during the other half.

If the pump is used to supply hydraulic pressure to a steering gear booster on an automobile the drive shaft 16 which protrudes from the casing through an appropriate oil seal 18' is drivingly connected to a suitable power takeoff shaft on the automobile engine (not shown).

The sides of the rotor 15 have a close running fit with the flat end walls 13 and 14 and a plurality of equiangularly spaced plunger pockets 19, four in the present case, are formed in the rotor. These pockets open to the opposite sides of the rotor as well as to its outer periphery and within each is a plunger 20 of a size and shape to have a snug sliding fit not only with the opposite side walls of the pocket but also with the flat end walls 13 and 14.

Attention is directed to the fact that there are no springs to urge the plungers 20 one way or the other. Hence, the plungers are free to respond to centrifugal force as the rotor revolves and by it to be thrown radially outward to bear against the cylindrical wall 12. Except for the restraining forces to be later discussed the plungers would, therefore, retain contact with the wall 12 as long as the rotor was revolving and by virtue of the eccentricity of the rotor and chamber, would move radially outward during half a revolution of the rotor and radially inward during the other half.

During that half of the rotary orbit of a plunger when it is moving radially outward oil or other fluid being pumped is drawn into the inner end portion of the plunger pocket by virtue of the communication between the pocket and an arcuate feed port 21 in one end wall of the chamber, in this instance in the wall 13. This arcuate feed port, which is merely a groove in the wall 13, has an inlet passage 22 leading thereto and an inlet pipe 23 connects the passage 22 with the source of oil or other fluid to be pumped (not shown).

It is, of course, necessary that the arcuate feed port 21 be so located with respect to the circular orbit of the plungers as to be swept by the inner end portions of the plunger pockets during that half of the plunger orbit, or at least a substantial portion thereof, when the plunger is moving radially outward. Hence, the arcuate feed port 21 is concentric to the rotor axis.

The fluid drawn into the plunger pockets in consequence to the radially outward projection of the plungers as they sweep along the feed port 21 is discharged from the pockets into an arcuate discharge port 24 formed in one or the other of the two flat side walls of the chamber (the wall 13 in this case) and connected with an outlet passage 25. A pipe 26 leads from the outlet passage 25 to the mechanism to be actuated, diagrammatically illustrated in the drawings by an appropriate legend.

The manner in which the plunger pockets are communicated with the discharge port 24 which, like the port 21, is an arcuate groove in the wall 13, is of importance particularly from the standpoint of assuring quiet operation. It will be noted

that each plunger is provided with a cavity 27 which opens through a port 28 to at least that side of the plunger which is contiguous to the wall 13 in which the arcuate feed and discharge ports are located, though for convenience in production the port 28 may pass entirely through the plunger. The cavity 27 also opens to the inner end of the plunger through a port 29. Between the port 29 and the interior of the cavity is a valve seat 30 upon which a ball check valve 31 normally seats, being spring urged into engagement therewith.

The check valve 31 thus closes radially inward and opens radially outward. Outward radial projection of the plunger to draw in a charge of fluid from the feed port 21, therefore, does not affect the seated condition of the valve but radially inward depression of the plunger to expel the fluid from under it opens the valve to allow the fluid under the plunger to escape through the open valve and the port 28 into the groove 24 and thence to the mechanism to be actuated.

This manner of expelling the fluid from the plunger pockets, of course, requires the discharge port 28 of the plungers to align with the arcuate port 24 throughout the entire time that a plunger is being depressed. Accordingly the arcuate port 24 is concentric to the cylindrical wall 12 of the pump chamber, so that if the plunger is fuily projected as is the top plunger of Figure 1, its port 28 aligns with the outermost end 24' of the arcuate groove 24 as soon as the plunger comes opposite it and remains in alignment with the arcuate groove throughout the entire depression of the plunger.

On the other hand, if the plunger is not fully projected the operation is as illustrated in Figure 5 which shows the progression of positions occupied by one plunger during one complete cycle. As here shown the port 28 is radially inward of the outermost end 24' of the arcuate groove or port 24 and does not begin to communicate with the port 24 until a position A is reached approximately fifty degrees to the left of vertical. However, although the port 28 aligns with the arcuate groove 24 at this point in the cycle there is no communication between the groove or port 24 and the plunger pocket beneath the plunger until the ball check valve is unseated and the valve will not unseat until radial inward depression of the plunger begins.

Consequently, as distinguished from what would be the case without the ball check valve, the plunger remains in its partially projected position and does not fly out to impinge the cylindrical chamber wall 12 the instant back pressure in the port is manifested at the plunger port 28. As a result the plunger gradually and gently contacts the cylindrical chamber wall 12 between its positions A and B and upon such contact radial inward depression of the plunger begins. As soon as such inward depression is begun the ball check valve is unseated and the fluid under the plunger is forced out of the plunger pocket into the discharge port 24 and hence to the mechanism to be actuated.

The spent fluid leaving the mechanism to be actuated could, of course, be conducted directly to the sump from which the pump draws its supply but instead the pressure remaining in the returning fluid is put to useful purpose by returning at least a part thereof through the pump chamber by way of a shunt line 32, the amount so by-passed depending upon the setting of a control valve 32' in the return to the sump. The

pipe line 32 connects with a passage 33 formed in one of the two end sections of the housing and opening into the hollow pump chamber at a point near its periphery.

From the pump chamber a passage 34 and a pipe line 35 carries the returning fluid back to the sump. This return line is preferably equipped with a throttling valve 35' so that by coordinated adjustment of the valves 32' and 35' practically any desired pressure condition can be established in the pump chamber. The pressure in the pump chamber is, of course, also a function of the speed of the rotor—the higher the speed the greater the pressure in the chamber and vice versa. Hence, the extent to which the pressure in the pump chamber resists the outward projection of the plungers by centrifugal force is self compensating.

To augment the hydraulic pressure within the pump chamber resulting from its connection to the return line for the spent fluid, the points of communication between the passages 33 and 34 with the chamber are so located that at high speeds when the plungers do not make their full stroke the path of the plunger past the mouth of the passage 33 is radially inward therefrom so as not to interfere with the inflow of fluid whereas the location of the mouth of the passage 34 being close to the periphery of the rotor is always in the path of the plungers regardless of their positions in the pockets. The plungers sweeping across the mouth of the passage 34 with increasing rapidity as the speed of the rotor rises, thus more and more interfere with the discharge of the fluid from the pump chamber and thus produce a desirable shuttering effect which augments the hydraulic pressure within the chamber.

Pressurizing of the pump chamber supplies one of the forces resisting the centrifugal force acting upon the plungers. The other force which acts against centrifugal force and thus combines with the pressure in the pump chamber is suction under the plungers which results from the orifice effect of the feed port 21.

The outward projection of the plungers in response to centrifugal force draws fluid into the plunger pockets but the size of each charge or, in other words, the quantity of fluid thus drawn into the plunger pockets depends upon the relative magnitude of all of the forces acting on the plungers and, as has been shown, all of these forces are functions of the speed of the rotor. It follows, of course, that the volume of fluid discharged by the pump can be no greater than that taken into the plunger pockets, hence if these various contending forces are predetermined to decrease the size of the charge drawn into the plunger pockets as the rotor speed rises, a substantially uniform discharge volume can be maintained over a relatively wide speed-range. The desired balance between the forces acting on the plungers is readily effected by controlling the volume of fluid returned through the pump chamber and regulating the constricting effect of the feed port 21 according to the weight of the plungers.

With these control factors properly adjusted or predetermined almost any performance requirements can be met. Thus, as illustrated in the curve of Figure 6, the volumetric output of the pump can be made to rise relatively abruptly during an initial rise in rotor speed, and then remain substantially constant for a substantial range of increasing speeds. The final upturn of the curve results from the centrifugal force overbalancing the opposing forces.

Attention is directed to the specific shape of the plungers. Their narrow outer ends 20' not only reduce the weight of the plungers and hence diminish the effect of centrifugal force thereon, but in addition have the advantage of reducing the throttling effect upon the inflow of the hydraulic fluid into the pump chamber at the time when the plungers are fully projected and thus sweep across the mouth of the inlet passage 33. At least the throttling or metering effect upon the inflow is not as great as it is upon the discharge of the fluid from the chamber. The holes 36 through the outer thin ends of the plungers are for the purpose of minimizing the turbulence within the pump chamber.

By virtue of the fact that the sides of the rotor have a close running fit with the flat end walls 13 and 14, the loss of pressure by leakage out of the plunger pockets is extremely slight but some fluid inevitably will enter the clearance between the sides of the rotor and the end walls 13 and 14 and to equalize the pressure at both sides of the rotor and thus assure smooth operation the wall 14 preferably has an arcuate oil groove 37 formed therein.

From the foregoing description taken in connection with the accompanying drawings, it will be readily apparent to those skilled in this art that this invention, though it deals with a relatively old form of pump, has many advantages especially since it adapts the so-called radial type pump to service which requires substantially uniform volumetric output over a relatively wide range of rotor speed.

What I claim as my invention is:

1. A radial type pump of the character described wherein a rotor revolves eccentrically within a bore, the wall of which is fixed with respect to the rotor axis and wherein the rotor has plungers slidable in radial pockets in the rotor so that by their orbital travel around the wall of the bore, the plungers alternately move outwardly to draw fluid through the pump inlet and into their pockets and then move inwardly to expel such fluid from their pockets and through the pump outlet: characterized by the provision of means to increasingly resist outward movement of the plungers as the speed of the rotor increases and thereby limit their outward movement to progressively less then their full stroke as defined by the eccentricity of the bore with respect to the rotor axis, so that the amount of fluid drawn into each plunger pocket decreases as the speed of the rotor increases and substantially uniform volumetric output for the pump is maintained over a relatively wide range of rotor speeds, said means comprising walls embracing the rotor and cooperating with the wall of the bore in which the rotor revolves to define a closed chamber into which the outer ends of the plungers project, and means to subject the interior of said chamber to a fluid pressure which increases as the speed of the rotor increases.

2. The radial pump set forth in claim 1 further characterized by the fact that the means for subjecting said chamber to pressure includes control means for regulating the pressure in said chamber and for determining the minimum rotor speed at which such pressure becomes effective to limit outward movement of the plungers.

3. The radial pump set forth in claim 1 further characterized by the fact that the fluid pressure manifested in said chamber is derived from the

pump outlet and thus increases substantially proportionately to the speed of the rotor.

4. The radial pump set forth in claim 3 further characterized by the provision of duct means connecting the pump outlet with said chamber through a mechanism to be actuated by the fluid pressure developed in the pump so that the pressure manifested in said chamber is that obtaining at the outlet side of said mechanism to be actuated.

5. The radial pump set forth in claim 4 further characterized by the provision of exhaust duct means leading from the fluid outlet of the mechanism to be actuated; and control means comprising a restriction in said exhaust duct means downstream from the point of communication between said chamber and the fluid outlet of the mechanism to be actuated, said control means regulating the pressure in said chamber and determining the minimum rotor speed at which such pressure becomes effective to limit outward movement of the plungers.

6. The pump defined in claim 1 further characterized by the provision of means for preventing sudden and forcible impingement of partially projected plungers against the wall of the bore by back pressure at the outlet port of the pump greater than the pressure in said closed chamber into which the outer ends of the plungers project and manifested in the pockets of said plungers as they are brought into communication with the outlet port, said means comprising check valve means controlling communication between the plunger pockets and the outlet port arranged to be closed by back pressure at the outlet port.

7. The pump as defined in claim 6 further characterized by the fact that communication between each plunger pocket and the outlet port of the pump is through a passage in its plunger; and further by the fact that said check valve means comprises a check valve located in said passage of each of the plungers.

8. A radial type pump of the character described wherein a rotor revolves eccentrically within a bore, and wherein the rotor has plungers slidable in radial pockets in the rotor so that by movement around the wall of the bore each plunger alternately moves outwardly to draw fluid through the pump inlet and into its pocket and then moves inwardly to expel such fluid from the pocket and through the outlet port of the pump; characterized by the fact that communication between each pocket and the outlet port of the pump is through a passage in its plunger; and further characterized by the provision of means for preventing sudden and forcible impingement of a partially projected plunger against the wall of the bore by back pressure at the outlet port manifested in the pocket of such partially projected plunger as said pocket is brought into communication with the outlet port, said means comprising a check valve located in the passage of each plunger and arranged to be closed by back pressure at the outlet port.

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