

Dec. 29, 1953

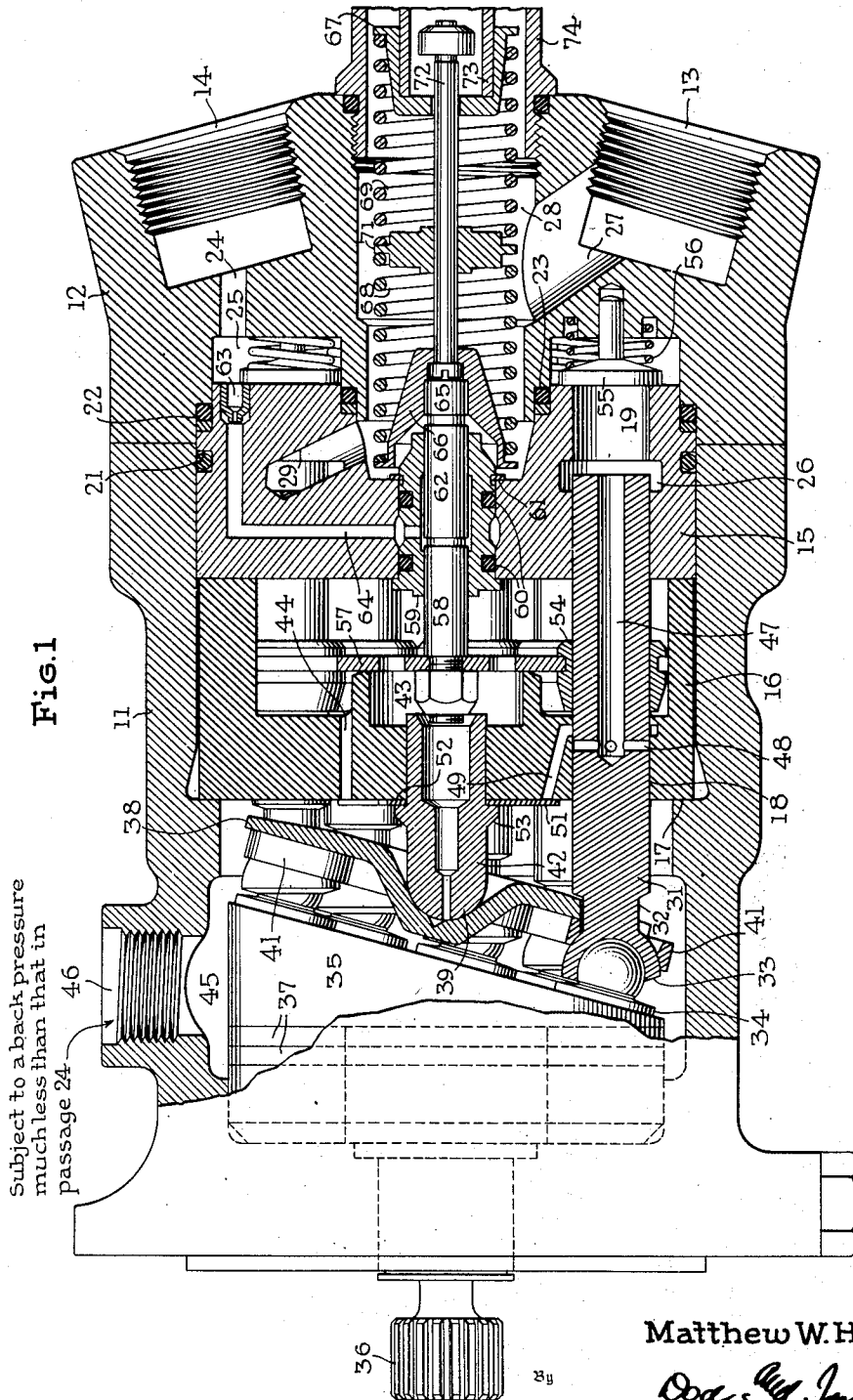
M. W. HUBER

2,664,047

VARIABLE DELIVERY PUMP

Filed May 3, 1950

2 Sheets-Sheet 1



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2 Sheets-Sheet 2

FIG. 2

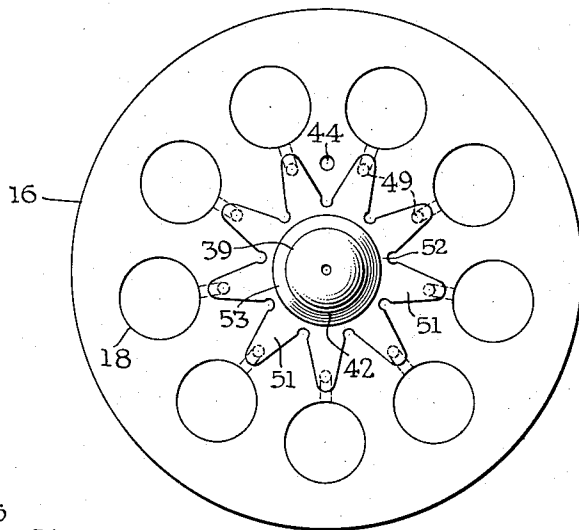


FIG. 3

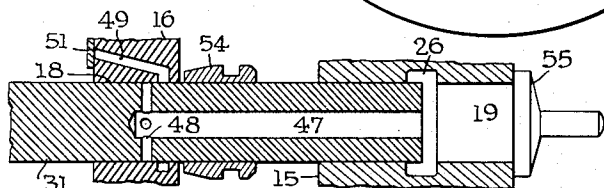


FIG. 4

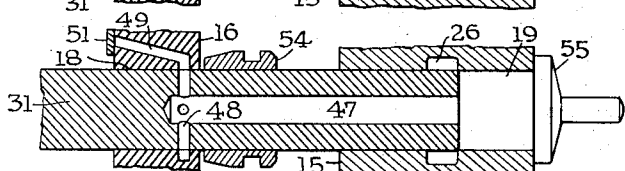


FIG. 5

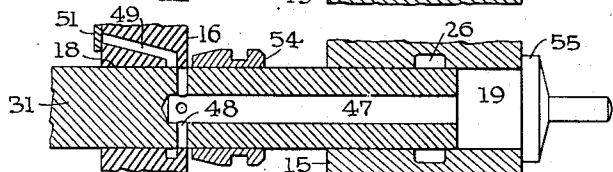


FIG. 6

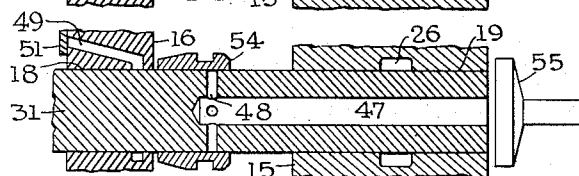
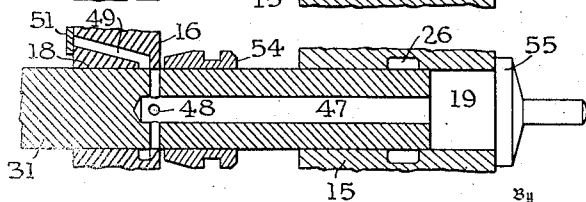


FIG. 7



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2,664,047

VARIABLE DELIVERY PUMP

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Application May 3, 1950, Serial No. 159,825

7 Claims. (Cl. 103—2)

1

2

This invention relates to multiple delivery hydraulic pumps, and applies the multiple delivery principle to variable displacement pumps of the type shown in the patent to Huber 2,433,222, December 23, 1947, in such a way as to contribute new functions and greatly extend the utility of that pump, and of other pumps controlled by varying the effective stroke of the plunger.

The pump of the Huber patent has spill-back valves which control the effective stroke of the plungers. At zero pump delivery hydraulic liquid (oil) flows back and forth between cylinders but the plungers draw in practically no new liquid from the pump intake. As a consequence the pump would not be well cooled or even well lubricated, when operating at zero delivery, except for the addition to the plunger-actuating swash plate of a simple centrifugal pump which draws a by-pass flow of hydraulic liquid from the intake and causes it to pass in contact with the plunger and swash plate mechanism.

The airplane industry is demanding for pumps, higher rotary speeds and operation against higher head pressures. The pressures against which the by-passed circulation must be delivered are also rising beyond values conveniently attainable with a centrifugal pump. A separate displacement pump to propel the by-passed liquid would be expensive to build and maintain and would consume more power than the scheme offered by the present invention. This scheme so modifies the patented Huber pump that, at least within the useful speed range of the pump, a definite predetermined fraction (say $\frac{1}{10}$) of the maximum pump displacement is delivered each stroke to the by-pass circuit against whatever head exists.

The delivery just stated occurs during a definite portion of each stroke of each plunger and the fraction so delivered is determined by port location. The remainder of the stroke of each plunger is controlled by the capacity control mechanism at settings ranging from zero delivery to full delivery. Since a spill-back mechanism normally has, in zero-delivery setting, the effect of suppressing pump suction, an important feature of the invention is means for avoiding this characteristic. This is accomplished by so locating the ports that the by-pass delivery occurs at a time when the spill-back path is closed for any delivery setting.

Mechanically, all that is added to the patented pump is one specially located port for each plunger, and a check valve, one for each such port. The illustrated construction comprises a

single star-shaped plate valve in which there is a separate valve-finger for each of said specially located ports. The centrifugal impeller is omitted.

The plunger-actuating mechanism shown in the drawings differs from that shown in the Huber patent in that it operates the plungers positively in both directions, a characteristic demanded by extreme high speed operation. It forms the subject matter of application Serial No. 156,275, filed April 17, 1950, now Patent 2,620,733, December 9, 1952.

The invention will now be described as applied to the pump of the Huber patent above identified but it should be understood that it is applicable to other pumps having similar control characteristics so that the mechanism here described in detail should be taken as illustrative and not limiting.

In the drawings:

Fig. 1 is a longitudinal axial section of a pump incorporating the invention.

Fig. 2 is an elevation of the rear face of the plunger guide showing a preferred form of multiple check valve used to control the various by-pass ports.

Figs. 3 to 7 inclusive are longitudinal axial sections of one plunger showing the various functionally significant positions which it assumes in one cycle. In all views the sleeve valve which controls the effective stroke is set for maximum plunger displacement.

Fig. 3 shows the plunger at the moment of reversal, i. e. at the start of its displacement stroke before the inlet port has been closed, and Fig. 4 shows it at the moment of closure of the inlet port. In the latter view the by-pass port is wide open.

Fig. 5 shows the conditions as the by-pass port closes. Fig. 6 shows the end of the displacement stroke.

Fig. 7 shows the by-pass port about to open and illustrates the function of the check valve on the by-pass port. During motion from the position of Fig. 7 to the position of Fig. 3, the suction developed in the cylinder draws in a new charge of liquid through the inlet port, the by-pass port being closed by its check valve against the entrance of fluid.

Refer first to Fig. 1. The pump housing is constructed in two parts, a body 11 and a cap 12. The cap has an inlet connection 13 and a discharge connection 14. These take the form of threaded openings into which the suction and discharge pipes are screwed.

An assembly made up of a cylinder block 15 and a guide block 16 is mounted within the housing. It is positioned by seating directly on the shoulder 17. The parts 15 and 16 are held together as usual by screws and dowels not visible in the drawings, so that they form in effect a single unit in which there are aligned plunger guideways 18 and cylinders 19. The cylinder block is sealed to the housing 17 by a ring gasket 21 mounted in a groove as shown, and is sealed to the cap by two annular gasket assemblies 22 and 23 which are confined between shoulders on the cylinder block and opposed shoulders on the cap.

Each of the assemblies 22 and 23 is centered on the axis of the housing, and each comprises a ring of rubber-like material round in cross-section and an adjacent filler ring. The precise form of these assemblies is not a feature of the invention and hence need not be elaborated.

The discharge connection 14 is connected by a passage 24 with the annular discharge chamber 25. The discharge pressure in the chamber 25, which is commonly above 3500 p. s. i., urges the cylinder block and plunger-guide assembly toward the shoulder 17 when the pump is in operation. The threaded connectors which hold the parts 11 and 12 together are conventional and no attempt to illustrate them in the drawings has been made.

Each cylinder 19 has an annular inlet port 26 and these ports 26 are all fed from the inlet connection 13 by a passage 27, the axial bore 28 formed in the cap 12 and cylinder block 15 and a series of passages 29 each of which communicates at one end with the bore 28 and at its other end with the corresponding one of the annular ports 26. In the illustrated pump there are nine plungers 31 each of which is reduced to form a neck 32 beyond which is a hemispherical head 33 which is formed with a hemispherical socket to receive the spherical head of the corresponding slipper 34. The slippers 34 engage a swash plate 35 which is driven through a rotary shaft by a spline connection 36. Portions of thrust plates for the swash plate are illustrated at 37.

The various slippers 34 are held against the swash plate 35 by a nutating plate 38 whose lefthand face, as viewed in Fig. 1, is a plane surface in which is located the geometric center of the spherical thrust head 39 on which the plate 38 may tilt. The plate 38 is peripherally notched, and the lefthand (plane) surface supports a plurality of sliding rings 41 which, as best shown in Fig. 1, have spherical concave surfaces which engage the spherical convex surfaces of the heads 33 on respective plungers.

Rotation of the swash plate 35 entails positive reciprocation of all nine plungers. So far as the invention is concerned, it is immaterial how the plungers are driven. At speeds in excess of 3000 R. P. M. positive actuation is desirable. The spherical support 39 for the nutating plate is formed on the end of a stud 42 pressed into an aperture at the center of the plunger guide 16. This stud is bored axially so as to transmit to the spherical bearing 39 hydraulic liquid (oil) which will always be present in the chamber 43. Chamber 43, as best shown in Fig. 1, is formed as a recess within the guide block 16. There is a passage 44 which allows hydraulic liquid to flow freely from the space 43 to the space 45 in which the swash plate 35 and the related plunger actuating mechanism are housed.

An outlet connection 46 is provided for out-

flow of oil which is supplied to the space 45 through the operation of the pump, as will hereinafter be described.

It seems advisable to emphasize at this point that the liquid which flows from chamber 45 through the connection 46 is supplied at a fixed rate, whether the pump be operating at full displacement or zero displacement or any displacement between these, provided the pump speed does not exceed the limit for which the particular pump is designed.

Each plunger 31 has an axial bore 47 which leads from the end of the plunger, and consequently from the working space in the cylinder 19, to a number of cross ports 48. In the most retracted position of the plungers 31 (see the lowermost plunger in Fig. 1) the cross ports 48 are beyond a by-pass flow passage 49 formed in guide block 16. There is such a passage 49 for each one of the plungers 31. Each passage 49 leads from a groove which encircles the plunger in the guideway to a point on the rear or lefthand face of the guide member 16. There each passage 49 is controlled by a plate check valve 51.

As best shown in Fig. 2 these check valves are preferably constructed as fingers radiating from a ring 52 which is clamped under the flange 53 formed on stud 42. The function of each valve 51 is to allow liquid to flow outward through the passage 49 and inhibit inward flow therethrough. The valves 51 are designed to seal tightly.

Slidable on each of the plungers 31 within the chamber 49 is a sleeve valve 54. These valves encircle their corresponding plungers closely and determine by their position the point in the displacement stroke of the plunger 31 at which the cross ports 48 will be closed by the sleeve valve 54. At that point the plunger starts displacement past the main discharge valve 55 for that cylinder. These valves are of conventional form, seat against the face of the cylinder block and are biased closed by coil compression springs 56.

It should be observed, however, that the main seating force on the valves 55 is the discharge pressure in the connection 14. The pressure in the connection 14 will be 3500 p. s. i. or higher, whereas the back pressure in the overflow connection 46 is low although it might in particular cases amount to several hundred pounds per square inch. Substantial back pressure on the connection 46 is not necessary, so far as the operation of the pump is concerned, but in certain installations of these pumps it is advisable to maintain a substantial positive back pressure. The pump is not adversely affected by such conditions.

The valves 54 are circumferentially grooved and all are engaged by a spider 57 which is fixed on a stem 58 slidable in the bushing 59. This bushing is mounted in an aperture in the center of the cylinder block 15 and at its lefthand end, as viewed in Fig. 1, it has a retaining flange. At its righthand end it is retained by a snap ring 61 which engages a groove in the bushing. The bushing 59 serves as the cylinder for the stroke adjusting motor.

The stem 58 functions as the piston rod and the slightly larger portion 62 functions as the piston. Discharge pressure communicated by the passage 24, chamber 25, choke 63 and passage 64 acts in this cylinder within the bushing 59 to urge the piston 62 to the right. The bushing 59 is sealed in the opening in the cylinder block by two toric gaskets 60 of rubber-like material, one at each side of the passage 64.

5

The effective annular area of the piston 62 is so small as to be hardly visible in the drawings. It is made small to permit the use of loading springs which are relatively light and which have a relatively low scale. The stem 53 is extended to the right beyond the piston 62 and terminates in a head 65. On this is mounted a spring seat 66. Between this and a companion spring seat 67 are mounted in tandem two coil compression springs 68 and 69. These react through an intermediate spring seat 71 which is slidable on the rod 72. The rod 72 is used as a guide for the seat 71 and also as a convenient means for limiting the separation of the spring seats 66 and 67.

The spring seat 67 is sustained by a member 73 which is adjustably mounted on a closure member 74, the latter being shown only partially in the drawings. The details of this construction are of minor present importance and have been omitted to permit the pump as a whole to be shown on a large scale. So far as here material it should be understood that member 74 is closed at its right hand end and that the member 73 is adjustable in the direction of the axis of piston 62, to vary the stress on the springs 68, 69.

The operation of the pump can be explained most simply by referring to Figs. 3 to 7 inclusive. In Fig. 3 the plunger 31 is in its extreme lefthand position and is about to start its movement to the right. The cross ports 48 are slightly to the left of the by-pass port 49. The end of the plunger exposes the inlet port 26 so that the plunger can move to the right without developing pressure in the cylinder 19.

Fig. 4 shows the next significant stage. Here the end of the plunger has just closed the inlet port 26. The cross ports 48 are in full register with the by-pass port 49. Since the valve 55 is heavily loaded hydraulically it remains closed and the pressure development in the cylinder 19 is relieved by flow through the passage 47, cross ports 48, by-pass 49 and check valve 51 which opens against a relatively low back pressure, and remains open until the position of Fig. 5 is reached.

The oil discharging past the valve 51 in the motion from the position of Fig. 4 to the position of Fig. 5 enters the space 45 and ultimately discharges through the connection 46.

Fig. 5 shows the conditions when the cross ports 48 have moved out of register with the by-pass port 49. The sleeve valves 54 are shown in their lefthand position and as soon as the cross ports 48 have entered the sleeve valve 54, displacement past the main discharge valve 55 commences. Thus, in Fig. 5 the valve 55 is closed but is about to open. If the sleeve valves 54 are in a position to the right of that illustrated, displacement past the valve 55 will start later and the total displacement past the valve 55 will consequently be smaller in volume.

Displacement ends when the plunger reaches its righthand limiting position shown in Fig. 6. Observe that the cross ports 48 are still within the sleeve valve 54.

The ensuing lefthand motion of plunger 31 develops suction in the cylinder 19 but this suction will be dissipated when the cross ports 48 pass out of the control of the sleeve valve 54. This occurs before the plunger exposes the inlet port 26. However, from the position shown in Fig. 7 in which the cross ports 48 are wholly within the guide member 16, to the final lefthand position shown in Fig. 3, suction is developed in the cylin-

6

der 19 because the valve 51 closes against back flow through the port 49.

I claim:

1. In a hydraulic pump, the combination of a cylinder having a main discharge port in communication with the cylinder at one end thereof, and a side port which serves as an inlet port; a piston reciprocable in said cylinder between positions in which it obstructs and exposes said side port said piston and cylinder enclosing a working space whose volume is varied by the reciprocation of the piston; a pressure-actuated valve controlling flow through said main discharge port and subject in an opening direction to pressure in said working space; means providing a secondary discharge port leading from said working space, and so obstructed and exposed alternatively by the piston as to be exposed for a minor portion of the piston stroke at a time when the piston closes said inlet port; and one-way flow means controlling said secondary port and serving to permit outward and inhibit inward flow there-through.

2. In a hydraulic pump, the combination of a cylinder having an inlet port and a main discharge port; a piston reciprocable in said cylinder between positions in which it opens and closes said inlet port, said piston enclosing within the cylinder a working space whose volume is varied by the reciprocation of the piston; a pressure-actuated discharge valve controlling flow through said main discharge port and subject in an opening direction to pressures in said working space; means providing a secondary discharge port leading from the working space and so obstructed and exposed alternatively by the piston as to be open for a minor portion of the piston stroke at a time when the piston closes the inlet port, the ports being so arranged that the piston starts its displacement stroke with the secondary discharge port closed and the inlet port open, and as it moves first opens the secondary discharge port, then closes the inlet port and then closes the secondary discharge port; and one-way flow means controlling said secondary port and serving to permit outward and inhibit inward flow therethrough.

3. In a hydraulic pump, the combination of a cylinder having an inlet port and a main discharge port; a piston reciprocable in said cylinder between positions in which it opens and closes said inlet port, said piston enclosing within the cylinder a working space whose volume is varied by the reciprocation of the piston; a pressure-actuated discharge valve controlling flow through said main discharge port and subject in an opening direction to pressures in said working space; means providing a secondary discharge port leading from the working space and so obstructed and exposed alternatively by the piston as to be open for a minor portion of the piston stroke at a time when the piston closes the inlet port, the ports being so arranged that the piston starts its displacement stroke with the inlet port open, and the piston while exposing the secondary discharge port closes the inlet port, and thereafter closes the secondary discharge port; and one-way flow means controlling said secondary port and serving to permit outward and inhibit inward flow there-through.

4. The combination of a hydraulic displacement pump comprising a cylinder, a plunger reciprocable therein and a main discharge valve loaded in a closing direction by the hydraulic pressure against which the pump discharges;

adjustable spill-back valve means for controlling the effective stroke of the plunger by venting the cylinder throughout portions of its displacement stroke and varying the extent of such portion between zero and a maximum less than the full stroke, whereby there is a remaining portion of the stroke within which venting does not occur; means affording a secondary discharge port normally obstructed by the plunger but exposed thereby within said remaining portion; and flow-controlling means controlling said secondary discharge port to permit outward and inhibit inward flow therethrough.

5. In a hydraulic displacement pump the combination of a cylinder having an inlet port and a discharge port; outward opening check valve means controlling flow through said discharge port; a ported plunger guide spaced from said cylinder; a reciprocable plunger extending through said guide with its end extending into said cylinder and controlling said inlet port, the plunger and cylinder enclosing a working space whose volume varies when the plunger reciprocates, and the plunger having a port which extends from the working space longitudinally through the plunger and terminates on the side of the guide and communicates with the port in the guide when the plunger is retracted, and is between the guide and the cylinder at other times; a longitudinally shiftable sleeve valve encircling the plunger between the guide and cylinder and adapted to control the plunger port; a check valve controlling the port in the guide and serving to inhibit inward flow therethrough; means for shifting said sleeve valve to vary the effective

displacement stroke of said plunger; and means for reciprocating said plunger.

6. The combination with the structure defined in claim 5 of a housing which encloses the means for reciprocating the plunger, is arranged to receive the discharge from the port in the guide and has an outflow passage so arranged as to maintain the housing liquid-filled.

7. The combination of a hydraulic displacement pump comprising a cylinder, a plunger reciprocable therein and a main discharge valve loaded in a closing direction by the hydraulic pressure against which the pump discharges; means for varying the effective reciprocating stroke of said plunger, between full stroke and a minimum effective stroke in which positive displacement always occurs; means affording a secondary discharge port controlled by the plunger and always opened thereby during said minimum stroke regardless of the total effective stroke; flow-controlling means controlling said secondary discharge port and serving to permit out-flow and inhibit in-flow therethrough; plunger actuating mechanism; and a housing enclosing said mechanism and into which the secondary discharge port discharges, said housing having an outflow connection so arranged as to maintain the housing liquid-filled.

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