Dec. 29, 1953 M. W. HUBER 2,664,048

HYDRAULIC PUMP WITH BY-PASS FLOW

Filed May 21, 1951 2 Sheets-Sheet 1

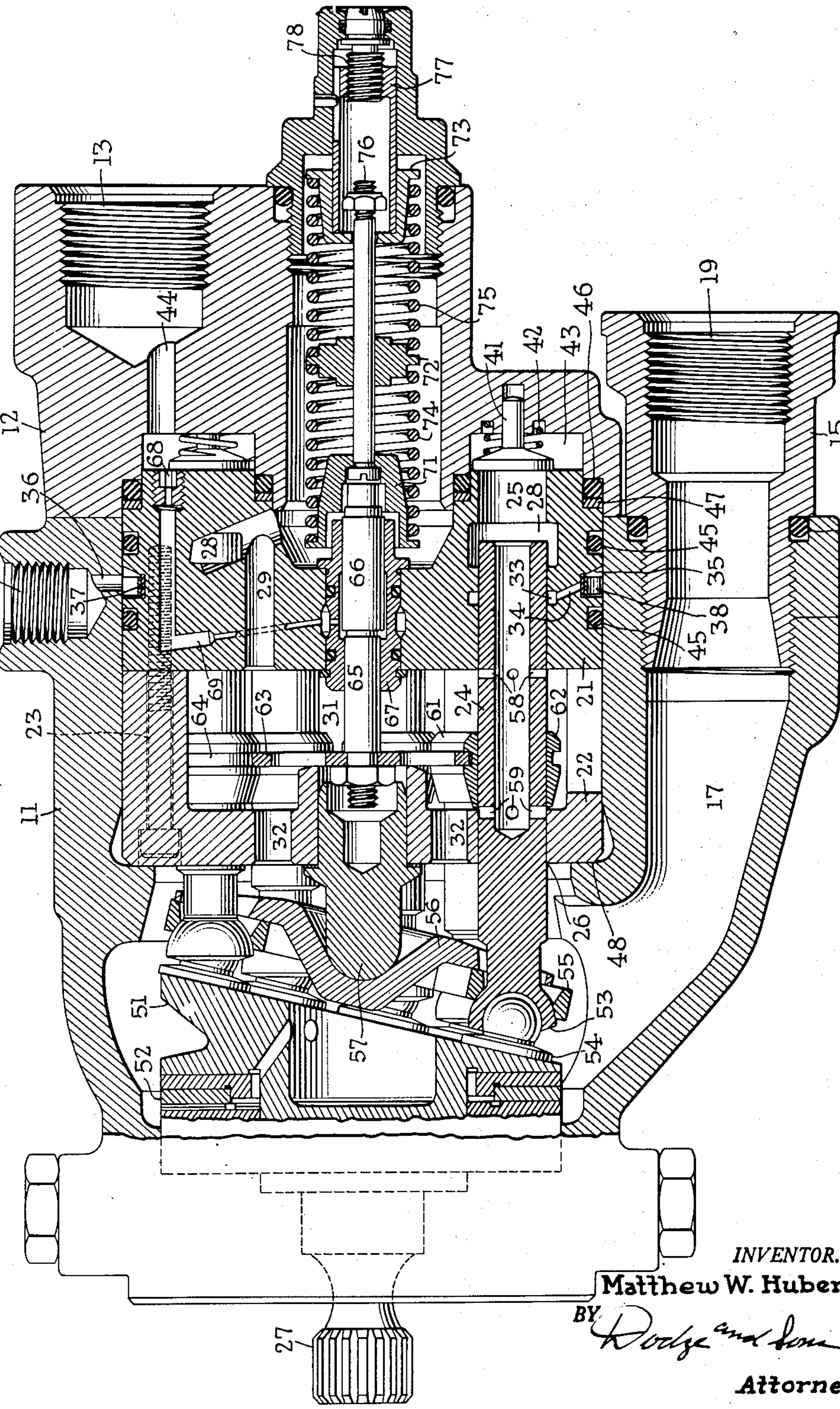

FIG. 1

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Fig. 3

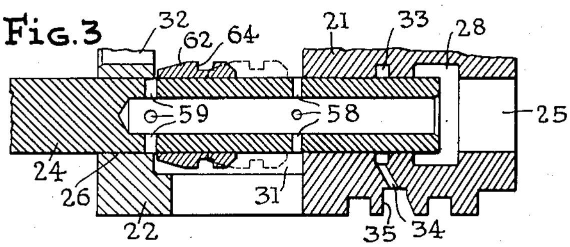


Fig. 2

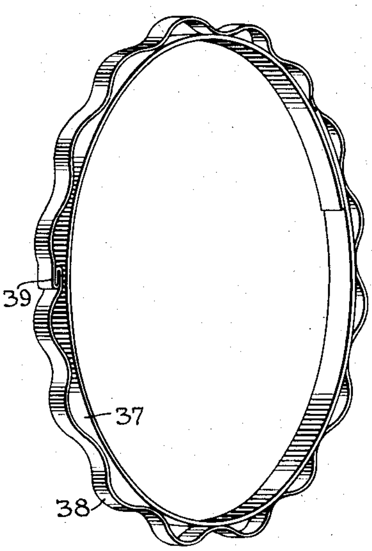


Fig. 4

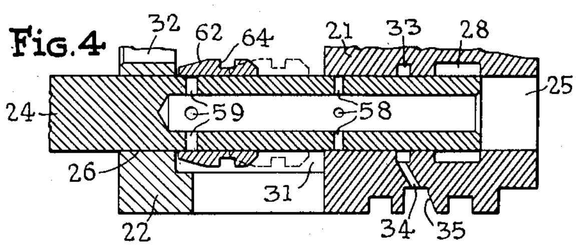


Fig. 5

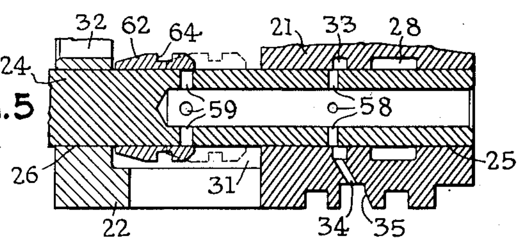


Fig. 6

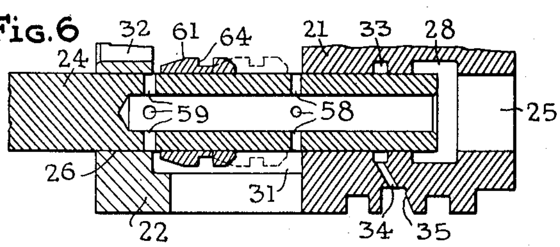
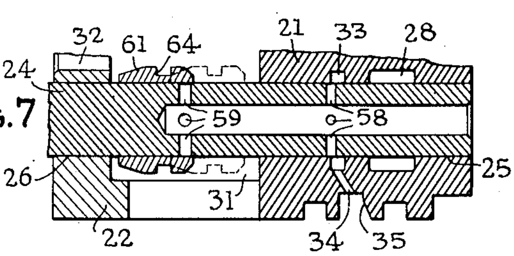


Fig. 7



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UNITED STATES PATENT OFFICE

2,664,048

HYDRAULIC PUMP WITH BY-PASS FLOW

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Application May 21, 1951, Serial No. 227,331

15 Claims. (Cl. 103—37)

This invention relates to multiple delivery hydraulic pumps, and particularly to pumps in which the main delivery is variable according to principles characteristic of the patent to M. W. Huber 2,433,222, issued December 23, 1947.

The objects of the invention are several. One is to assure a by-pass flow of liquid through the pump to provide cooling and lubrication, even when the pump is operating at or near zero delivery, and even when the by-pass flow must be against a substantial pressure head.

An ancillary feature is the construction of the pump in such a way as to permit delivery of by-passed liquid against head pressures which may be nearly as high as the head pressure against which the main pump-delivery occurs, or alternatively may be much less.

A very important object is to ensure the minimum practicable power consumption when the pump operates at, or substantially at, zero delivery.

The by-pass flow is delivered from the cylinders through ports in the cylinder wall which are exposed by the corresponding plungers near the ends of each displacement stroke. Where the by-pass flow encounters high back pressures check-valves control these ports to inhibit back flow. The need for this is particularly evident when the pump is operating at and near zero delivery, because then the spill-back ports tend to be continuously open. However at normal operative speeds there are pronounced inertia effects that cause by-pass delivery to occur against head pressures of considerable magnitude (several hundred pounds per square inch) and consequently the check-valves are not always needed. In their absence, by-pass delivery pressures can be controlled to a useful extent by selection of the size of the by-pass ports.

In cases where adequate by-pass delivery pressures can be had in the absence of check-valves, these valves will be omitted, because in such case there is protection against the generation of destructive pressures. Also the simpler construction is desirable.

If one assume that a variable delivery pump such as the patented pump, referred to above, is delivering into an absolutely tight line, one can imagine the pump moving to zero delivery position and staying there. Absolutely tight lines being rarely encountered, because of seepage past valves and similar losses, the fact is that the pump controller often hunts in a narrow range close to zero delivery position. Since it requires considerable force to start flow against a head

of 3000 p. s. i. the plungers deliver impacts near the end of each stroke as their spill-back valves close. These impacts deliver little or no oil but they do absorb considerable energy. Hence such a pump is likely to be characterized by undue energy consumption at its zero delivery.

Power consumption at and near zero delivery is minimized according to the present invention, by causing a minority of the plungers, and preferably only one plunger to continue to deliver after the pressure control has caused all the others to cease to deliver. The active plunger maintains the head by displacing oil through a short but useful effective stroke while the others are freed from the recurrent loads which consume energy at rates out of proportion to the negligible quantities of oil which they might deliver.

The result is secured by changing the dimensions of one spill-back sleeve, or the location of the port which it controls, so that in the zero delivery setting of the pump controller its spill-back port closes or at any rate nearly closes but the ports of the other plungers remain open. A somewhat similar by-pass arrangement is described and claimed in applicant's co-pending application Serial No. 159,825, filed May 3, 1950.

The present invention can be used with or without the by-pass. When a by-pass is provided it ensures that regardless of pump speed, one plunger will deliver positively to the by-pass, as well as to the pump discharge. In practice, however they all deliver to the by-pass because of inherent inertia effects.

Assured by-pass circulation for cooling and lubrication permits closer fitting of plungers. Comparative tests of adequate duration show significant gains in volumetric efficiency, and a reduction of power consumption at zero delivery to ⅔ of the amount heretofore considered minimum.

The pump so tested will now be described by reference to the accompanying drawing in which:

Fig. 1 is a longitudinal axial section through the pump on a plane intersecting the axes of the inlet and discharge connections.

Fig. 2 is a perspective view of the multiple check-valve which may be used to control the by-pass ports, or may be omitted altogether.

Figs. 3 to 7 inclusive are fragmentary sections showing plunger positions. In each, the full delivery position of the spill-back valve is shown in full lines and the zero or minimum delivery position in broken lines.

Figs. 3, 4 and 5 show the one plunger which has the atypical sleeve respectively at the start of the displacement stroke, at closing of the inlet port and at the end of the displacement stroke. Figs. 6 and 7 show one of the eight plungers having regular or typical sleeves at the start and at the end of the displacement stroke.

The pump housing comprises a main body portion 11 and a cap 12. A threaded connection 13 for the main discharge line is formed in cap 12. A smaller threaded connection 14 for the by-pass line is formed in body 11. A fitting 15 is threaded at 16 into the entrance end of inlet passage 17 and sealed by gasket 18. Passage 17 leads directly to the swash plate chamber. The fitting 15 is threaded at 19 to receive the suction line.

The by-pass connection 14 commonly would return by-passed liquid (oil) to the sump of the hydraulic system in which the pump is included, either directly or through some device to be lubricated and cooled. The connection 14 commonly would be and at any rate can be under substantial back-pressure. These details concern the system with which the pump is used and hence do not require elaboration. Whatever becomes of oil by-passed through connection 14, it assures continuous oil flow while the pump is running, so that oil continuously enters through passage 17 and flows in contact with the plunger-actuating mechanism.

The head 12 and body 11 are held together by threaded connections, not visible in the drawing and confine between them a generally cylindrical cylinder block 21 and a generally cylindrical guide block 22, which are fixed in assembled relation by conventional means including screws, one of which appears in broken lines at 23.

The illustrated pump has nine plungers 24. Consequently the block 21 has nine cylinders 25 and the guide block 22 has nine guide ways 26, each axially alined with a corresponding cylinder. The axes of the plungers are parallel and are uniformly spaced in circular arrangement around the axes of the cylindrical blocks 21 and 22 and also around the axis of the drive shaft 27.

Each cylinder 25 has an annular inlet port 28 fed by a passage 29 bored in cylinder block 21. These lead from the spill-back valve-space 31 enclosed between the cylinder block 21 and the guide block 22 which is cup shaped. Passages 32 connect space 31 with passage 17.

To the left of port 28 each plunger 24 is guided in block 21 for a considerable distance, and formed in this guide portion is an annular by-pass port 33, one for each plunger. A passage 34 leads from each port 33 to the bottom of an annular channel 35, flared outward in cross section as shown. Channel 35 communicates through a drilled port 36 with connection 14.

In channel 35 is a valve comprising a thin flexible strip 37 curled to circular form with its ends overlapping. The valve is contracted by an encircling garter spring 38 formed of a sinuous metal strip whose ends are hooked together at 39. The strip 37 functions as a plurality of check valves individually controlling passages 34. The use of the parts 37, 38 is optional and depends on the performance desired. The size of ports 34 is also determined by the performance desired, particularly when parts 37, 38 are omitted.

The end of each cylinder 25 is closed by a discharge valve 41 which is urged to its seat on block 21 by a coil compression spring 42. The valves are housed in an annular space 43 between block 21 and cap 12, and connected with discharge connection 13 by passage 44. The cap 12 is formed with seats for springs 42 and with a guide way for the axial stem of each valve as clearly shown in Fig. 1.

It is contemplated that space 43 will be maintained at high pressure, pressures of 3000 p. s. i. being everyday practice and pressures of at least 5000 p. s. i. being attainable. They will be used whenever apparatus to use them becomes available.

The block 21 is sealed to body 11 by toric gaskets 45 on opposite sides of channel 35 and to cap 11 by toric gasket 46 and filler ring 47. Pressure in space 43 forces the blocks 21, 22 against shoulder 48, so that they are positively positioned.

The means for positively reciprocating the plungers 24 forms the subject of Patent 2,620,738, issued December 9, 1952, subsequently to the filing of the present application and requires only brief description.

Shaft 27 drives swash plate 51 which is sustained by thrust bearing 52. Each plunger 24 has a hemispherical enlarged end 53 in which is socketed a universally tiltable slipper 54 and against which is seated a universally tiltable thrust ring 55. The slippers 54 engage the face of swash plate 51 and the rings 55 engage the plane face of a nutating plate 56. Plate 56 tilts on a hemispherical thrust journal 57 and has notches at its periphery to receive the necks of plungers 24 which pass freely therethrough.

The axes of tilt of each slipper and its related ring are coincident and the plane face of plate 56 engaged by the rings 55 passes through the axis of spherical journal 57. Any suitable geometrically similar reciprocating mechanism might be substituted, so far as the present invention is concerned.

Each plunger 24 has several radial by-pass ports 58 and several radial spill-back ports 59. Four of each have proved satisfactory and are illustrated.

Ports 58 are so located that they open into space 31 when their plungers are fully retracted (see Figs. 3 and 6) and are blanked by block 21 before the end of the plunger closes the inlet port 28 (see Fig. 4). Near and at the end of the displacement stroke they communicate with their corresponding by-pass ports 33 (see Figs. 5 and 7).

Ports 59 are so located that in the fully retracted position of their plungers they communicate with space 31 (see Figs. 3 and 6). On their displacement strokes they are controlled by either the typical spill-back sleeve valve 61 of Figs. 6 and 7 or the atypical sleeve valve 62 of Figs. 3, 4 and 5. All the sleeve valves 61, 62 are positioned by a spider 63 which engages the annular groove 64 in each valve.

The difference between typical valves and the atypical valve is simply that when the valves 61, 62 of Figs. 3 to 7 are in their dotted line positions (zero delivery setting) the typical valves 61 will close their ports 59 later in the stroke than does the atypical valve 62. The easiest way to attain this result (but not the only way) is to make the valve 62 longer than the valves 61 the extra length being at the left hand end of the valve as viewed in Figs. 3, 4 and 5.

As a consequence plungers having typical valves in zero delivery position will have their spill-back ports open until the by-pass ports open but any plunger having an atypical valve in its corresponding position will have its spill-back valve closed before its by-pass ports open, and so will deliver a small quantity through its discharge valve 41.

Spider 63 is carried on the end of piston rod 65 which has a coaxial cylindrical enlargement 66. Parts 65, 66 extend through the cylinder bushing 67 sealed in block 21 by toric gaskets, as shown. Pressure head in chamber 43 is admitted by choke 68 and passage 69 to the space within cylinder bushing 67 where it reacts to the right (as seen in Fig. 1) on the differential area between parts 65 and 66. The spring seats 71, 72, 73, springs 74, 75, tie bolt 76, adjustable slide 77 and adjusting screw 78 are components of a known adjustable spring loading assembly for biasing spider 63 to the left, as viewed in Fig. 1.

In any reciprocating positive displacement pump operating at 3000 R. P. M. and higher against head pressures of 3000 p. s. i. and higher, inertia effects on the liquid are substantial and stresses developed by static pressures are cyclically augmented by kinetic reactions about which little can be known except that they are severe.

The thickness of the housing is limited by practical considerations, and while the structure is rigid so far as is visible there must be some rather substantial deflections. Moreover dimensional tolerances are a manufacturing necessity and some of these are cumulative, so that it is next to impossible to determine valve timing even from the actual pump.

As a consequence, figures such as Figs. 3 to 7 must be considered as a basis of explanation of what apparently happens, rather than as a demonstration of the precise timing of ports.

A striking confirmation of this is the circumstance that removal of valve 37 and its spring 38 reduces the maximum head against which bypass flow can be delivered, but does not greatly reduce flow against heads as high as four or five hundred pounds per square inch. It does inhibit generation of destructive pressures in case the by-pass path is closed.

Referring to Figs. 5 and 7 it is convenient for purpose of explanation to say that in the zero-displacement (broken line) position of a typical sleeve 61 flow to 33 starts at least as soon as flow through 59 ends whereas in the corresponding position of the atypical sleeve 62 flow through 59 ends before flow to 33 can start. However there are inertia effects, and just as a little "lead" is beneficial in a distributing valve, so is "lead" beneficial here. For any pump an optimum setting can be determined on a trial and error basis, but exactly what it is dimensionally has so far defied attempts at measurement.

Figs. 3 to 7 should not be too literally interpreted for they illustrate a principle rather than strict dimensional relations. They are offered and should be considered on that basis.

They demonstrate better than any verbal description can, the fact that at and near the zero displacement setting the atypical sleeve 62 alone regulates, allowing the typical sleeves 61 to open wide their spill-back ports. This takes the load off eight of the nine plungers and saves energy because one can supply the minute flow more economically than can nine.

The views also demonstrate that all nine plungers, because of inertia effects deliver to the by-pass, but the one having the atypical sleeve delivers positively because its spill-back port closes.

The pump illustrated is typical and exemplary. Modifications obviously are possible within the scope of the invention.

What is claimed:

1. In a pump, the combination of a plurality of cylinders; a plurality of plungers, one reciprocable in each cylinder, each plunger having a spill-back port; means for reciprocating said plungers; a discharge connection common to said cylinders and in which a discharge pressure is maintained by operation of the pump; a pressure motor subject to said pressure; means for loading said motor in opposition to said pressure; valves controlling said spill-back ports and shiftable in the direction of plunger reciprocation to vary the effective stroke of the corresponding plungers; and connections through which the motor shifts said valves in ranges of position so differentiated that as to the majority of plungers the effective stroke is varied in inverse relation to discharge pressure between a maximum and zero, and as to the minority, not less than one, the effective stroke is simultaneously varied between a maximum and an effective stroke greater than zero, whereby at least one cylinder remains effective to maintain discharge pressure against leakage when the remaining cylinders have ceased delivery.

2. The combination defined in claim 1 in which said minority is one plunger, and the differentiation between the effective strokes of the majority and the effective stroke of said one is sufficient to afford a precise control range near zero delivery in which control is effected by varying the effective stroke of said one plunger.

3. The combination defined in claim 1 in which the spill-back valves are sleeves through which the spill-back ported portion of the respective plungers reciprocate, each sleeve having a control edge which times the closure of its ports, said differentiation being attained by variation of the positional relation between the motor and said control edge.

4. The combination of a hydraulic displacement pump comprising a cylinder having a suction connection, 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 reciprocating said plunger; means for varying the effective stroke of said plunger between a maximum stroke and a minimum stroke in all of which strokes positive displacement occurs; means affording a secondary discharge port controlled by said plunger and opened thereby in each stroke at a time when positive displacement is occurring; and means for causing at least the liquid passing through said secondary discharge port to flow in contact with said plunger-reciprocating means.

5. The combination defined in claim 4 in which the means which varies the effective plunger stroke is of the type which delays in varying degree the commencement of positive displacement, and the secondary discharge port is arranged to be opened by the plunger near the end of its displacing stroke.

6. A multi-cylinder pump comprising in combination a plurality of cylinders having inlet ports and a suction connection common thereto; plungers, one reciprocable in each cylinder; discharge valves one for each cylinder; a main discharge connection into which all said discharge valves deliver, the pressure in said connection serving to bias said valves in their closing directions; means for reciprocating said plungers; means common to all said plungers and adjustable to vary the effective stroke thereof between full stroke and minimum effective stroke, in all of which adjustments positive displacement by a

minority comprising at least one plunger occurs; means affording secondary discharge ports for the various cylinders, controlled by respective plungers and so arranged that each is open during part of the minimum effective stroke of its plunger; and means for passing at least that liquid which flows through the secondary discharge ports in contact with said plunger reciprocating means.

7. The combination defined in claim 6 in which the means adjustable to vary the effective strokes of the plungers are so contrived that the majority of said plungers have their strokes simultaneously varied between full and zero effective stroke and said minority between full and a minimum effective stroke greater than zero.

8. The combination defined in claim 6 in which the means adjustable to vary the effective strokes of the plungers are so contrived that the majority of said plungers have their strokes simultaneously varied between full and zero effective stroke and said minority between full and a minimum effective stroke greater than zero within a part of which minimum said secondary discharge ports are open.

9. The combination with the structure defined in claim 6 of check valve means serving to inhibit back flow through said secondary discharge ports.

10. In a hydraulic displacement pump unit the combination of means enclosing a working space having an inlet port adapted to be controlled by a plunger, a discharge port and a guide way coaxial with said working space; a plunger reciprocable in and filling said guide way with its end so positioned in said working space that when moved in a displacing direction it first closes said inlet port and then displaces hydraulic liquid through said discharge port; a spring-loaded pressure actuated valve controlling said discharge port; secondary discharge means comprising ports respectively in said plunger and in said guide way positioned to communicate as the plunger approaches the end of its displacing stroke; spill-back means including a port leading through the plunger from said working space and an adjustable spill-back valve controlling said port to close the same through variable portions of the displacement stroke; means for adjusting the spill-back valve; and means for reciprocating said plunger.

11. The combination defined in claim 10 in which the secondary discharge means, and the spill-back means are so coordinated that when the secondary discharge means are open the spill-back means are closed.

12. A pump comprising a plurality of units as defined in claim 10 with single means for adjusting the spill-back valves and single means for reciprocating the plungers each common to all units, the majority of said units having their spill-back and secondary discharge means so coordinated that in zero discharge position both are open simultaneously; the minority of said

units, not less than one being characterized by a different coordination such that when the secondary discharge means are open the spill-back means are closed even when adjusted for minimum delivery.

13. In a hydraulic displacement pump unit the combination of means enclosing a working space having an inlet port adapted to be controlled by a plunger, a discharge port and a guide way coaxial with said working space; a plunger reciprocable in and filling said guideway with its end so positioned in said working space that when moved in a displacing direction it first closes said inlet port and then displaces hydraulic liquid through said discharge port; a spring-loaded pressure-actuated valve controlling said discharge port; secondary discharge means comprising ports respectively in said plunger and in said guide way positioned to communicate as the plunger approaches the end of its displacing stroke; check valve means inhibiting back flow through said secondary discharge means; spill-back means including a port leading through the plunger from said working space and an adjustable spill-back valve controlling said port to close the same through variable portions of the displacement stroke; means for adjusting the spill-back valve; and means for reciprocating said plunger.

14. In a hydraulic pump, the combination of a cylinder having an inlet port, a main discharge port and a secondary discharge port; a plunger reciprocable in said cylinder and controlling said inlet port, said plunger obstructing said secondary discharge port through a first and major portion of its displacement stroke and having a port which thereafter connects the cylinder and secondary discharge port through a minor portion of the displacement stroke to afford brief periodic discharges through said secondary port; a pressure-actuated valve controlling flow through the main discharge port, said valve being biased in a closing direction by the pressure against which the pump discharges and being capable of being opened by pressure in said working space when said pressure is dominant; and adjustable means associated with the plunger for neutralizing the displacing action of said plunger through variable portions of its displacement stroke within the aforesaid major portion thereof.

15. The combination of the structure defined in claim 14 with one way flow means controlling said secondary discharge port and inhibiting back flow therethrough.

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References Cited in the file of this patent

UNITED STATES PATENTS

Number	Name	Date
2,223,759	Dillstrom	Dec. 3, 1940
2,436,797	Deschamps et al.	Mar. 2, 1948
2,512,799	Huber	June 27, 1950