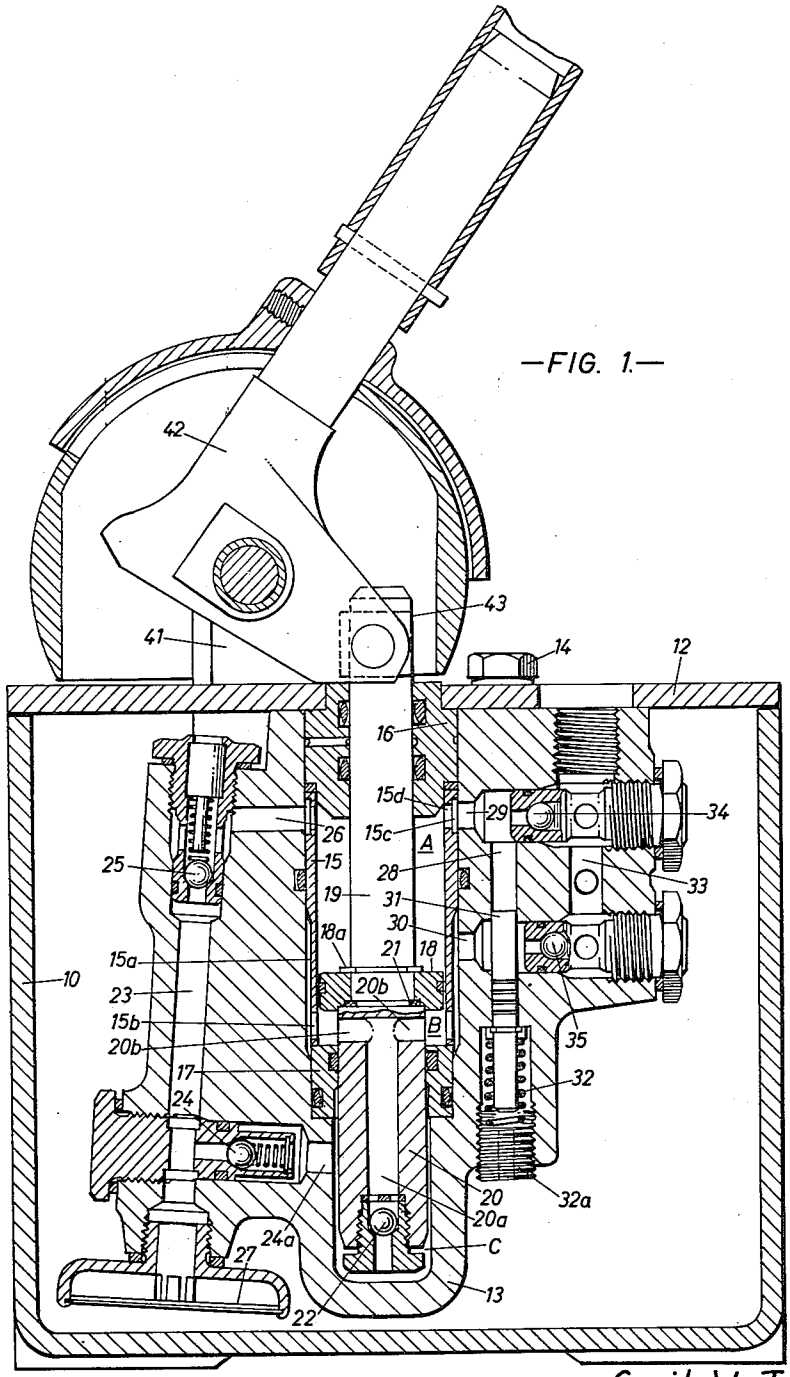
Nov. 20, 1962 C. W. TRIDGELL 3,064,581

RECIPROCATING HYDRAULIC PUMPS

Filed July 6, 1960 5 Sheets-Sheet 1

—FIG. 1.—

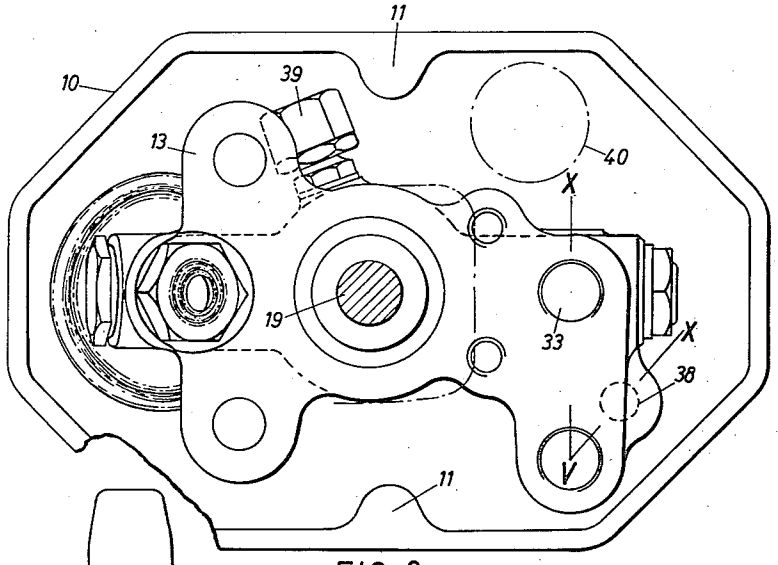



INVENTOR:
Cyril W. Tridgell
BY
Bierman & Bierman
Attorneys

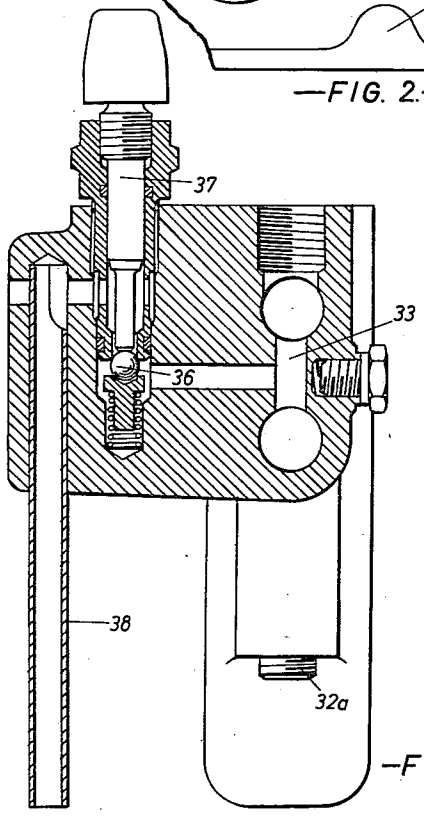
Nov. 20, 1962 C. W. TRIDGELL 3,064,581

RECIPROCATING HYDRAULIC PUMPS

Filed July 6, 1960 5 Sheets-Sheet 2




—FIG. 2.—



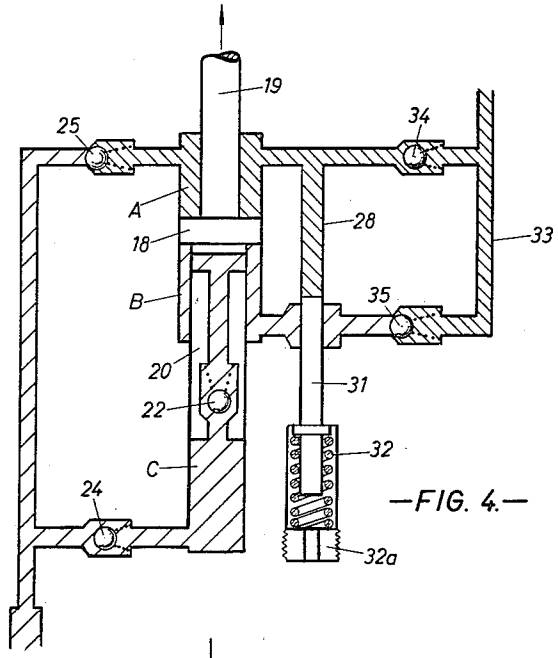

—FIG. 3.—

INVENTOR:
Cyril W. Tridgell
BY
Bierman & Bierman
Attorneys

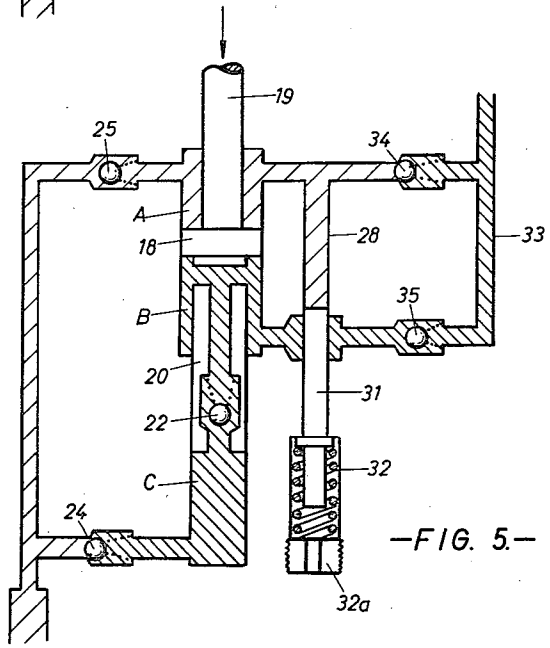
Nov. 20, 1962 C. W. TRIDGELL 3,064,581

RECIPROCATING HYDRAULIC PUMPS

Filed July 6, 1960 5 Sheets-Sheet 3




—FIG. 4.—



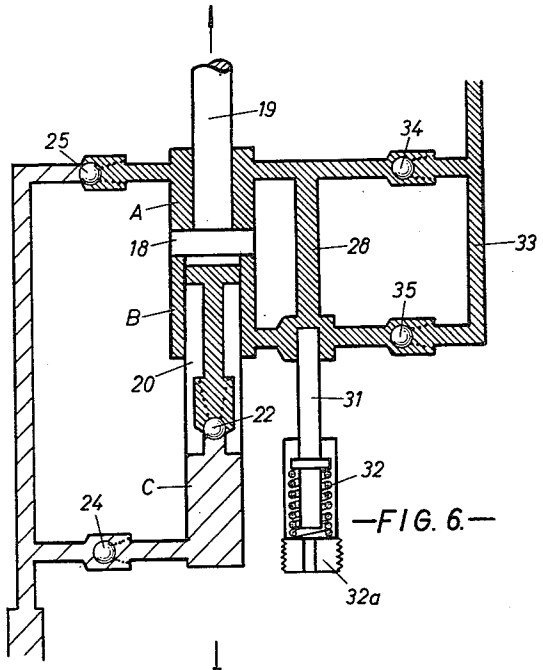

—FIG. 5.—

INVENTOR:
Cyril W. Tridgell
BY
Bierman & Bierman
Attorneys

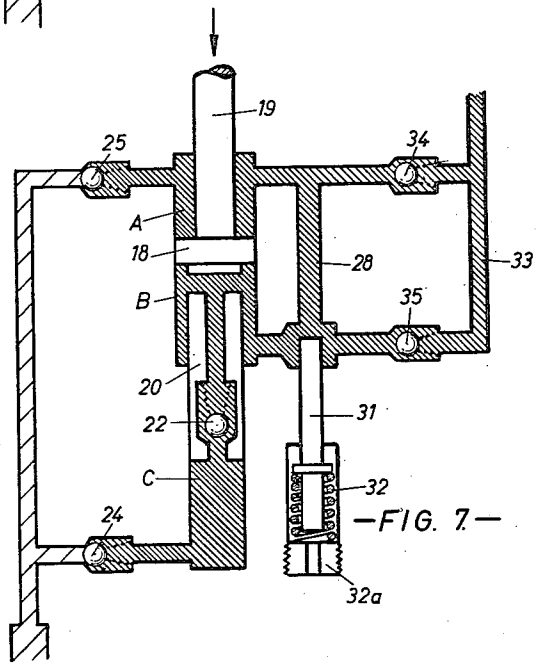
Nov. 20, 1962 C. W. TRIDGELL 3,064,581

RECIPROCATING HYDRAULIC PUMPS

Filed July 6, 1960 5 Sheets-Sheet 4




—FIG. 6.—



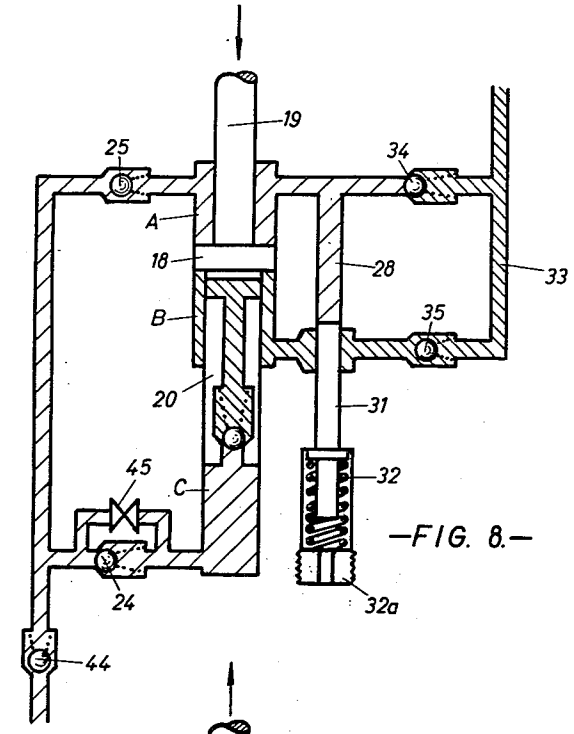

—FIG. 7.—

INVENTOR:
Cyril W. Tridgell
BY
Bierman & Bierman
Attorneys

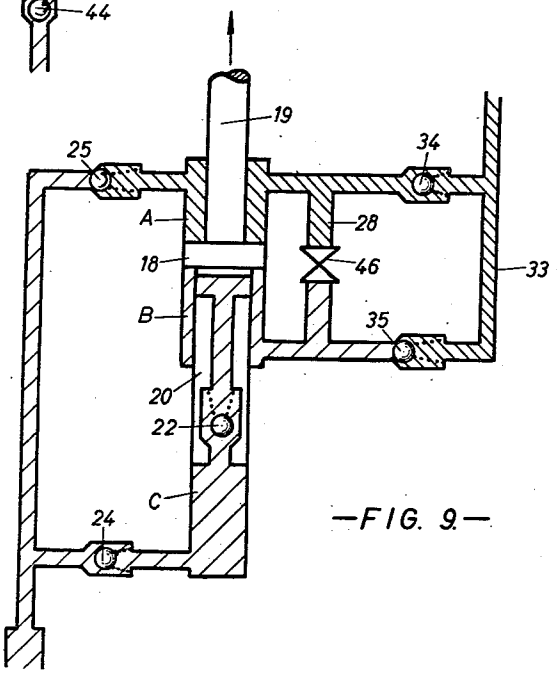
Nov. 20, 1962 C. W. TRIDGELL 3,064,581

RECIPROCATING HYDRAULIC PUMPS

Filed July 6, 1960 5 Sheets-Sheet 5




—FIG. 8.—




—FIG. 9.—

INVENTOR:
Cyril W. Tridgell
BY
Bierman & Bierman
Attorneys

United States Patent Office

3,064,581
Patented Nov. 20, 1962



3,064,581

# RECIPROCATING HYDRAULIC PUMPS

Cyril William Tridgell, Hemel Hempstead, England, assignor to Power Jacks Limited, a British company

Filed July 6, 1960, Ser. No. 41,050
Claims priority, application Great Britain July 10, 1959
4 Claims. (Cl. 103—37)

This invention relates to reciprocating hydraulic pumps and particularly, though not exclusively, to hydraulic pumps adapted to be manually actuated, i.e. by hand lever or pedal and such as are used to actuate hydraulic jacks, clamps or other hydraulically actuated devices.

In connection with hydraulically-actuated devices, it is often desired to have a dual-range pump, as distinct from a 2-stage pump in which a first low-pressure stage feeds a second high-pressure output stage; that is to say, by "dual-range pump" is meant one which is capable of delivering either a large volume at low pressure or a smaller volume at higher pressure, the former to bring the jack, clamp or other device quickly into its operative position or to operate against relatively light loads, and the latter to provide greater mechanical advantage as the load is taken up, or when dealing with heavier loads. Preferably, of course, such pumps are double-acting; that is to say, pumping occurs in both directions of movement of the actuating lever.

Double-acting dual-range pump units are already well known, but, hitherto, have consisted essentially of two individual pumps, both of which operate for low-pressure use, but one of which is "open-circuited" when the unit is required for high pressure use. Change-over from low to high pressure or vice versa is not usually automatic, and the unit assembly entails all the parts for two separate pumps mechanically coupled for simultaneous reciprocation.

One object of the present invention is a double-acting dual-range pump unit of improved construction embodying less parts than would be required for two separate pumps and having only three displacement chambers which are adapted to be hydraulically coupled in parallel or differential arrangement for low- or high-pressure operation respectively.

Another object of the invention is a double-acting dual-range hydraulic pump as aforesaid which is pressure-responsive for automatic change of range between low-pressure and high-pressure functioning.

A still further object of the invention is to enable the pump to be not only used either as a low-pressure or high-pressure pump, but alternatively, as a medium-pressure pump with medium-volume delivery.

According to the present invention, a reciprocating dual-range hydraulic pump comprises, in combination, one double-acting piston-and-cylinder assembly and one single-acting pump with non-return inlet and outlet valves for the three displacement chambers provided, in combination with control means for a connecting passage between the two chambers of the double-acting assembly such that the closing of said passage causes the pump direction to deliver alternately from one of said two chambers and from both the second of said two chambers and the single chamber of the other assembly, whilst the opening of said passage causes the pump to deliver alternately the differential volume of said two chambers and the volume of said single chamber less the negative differential volume of said two chambers.

In a preferred embodiment of the invention, the piston of the double-acting assembly has a piston rod at each end, one such rod being used for actuation of said piston and the other as the plunger of the single-acting assembly.



According to a further feature of the invention, the control means comprises a valve adapted automatically to open the associated passage when the pressure on the output side of the pump exceeds a predetermined value.

According to a still further feature of the invention, the non-return inlet valve to the chamber of the single-acting assembly is bridged by a manual valve whose opening causes the pump to operate as a medium-pressure unit delivering only from alternate chambers of the double-acting assembly.

In the accompanying drawings;

FIG. 1 is a central sectional side elevation of one example of a reciprocating dual-range hydraulic pump for manual operation and made in accordance with the present invention.

FIG. 2 is a plan view of the pump of FIG. 1 with the cover removed.

FIG. 3 is an unfolded section on the line of FIG. 2.

FIGS. 4 and 5 are diagrammatic views showing the action of the pump during up- and down-strokes respectively of the piston, when operating against low-pressure resistance.

FIGS. 6 and 7 are diagrammatic views showing the action of the pump during up- and down-strokes respectively of the piston, when operating against high-pressure resistance.

FIGS. 8 and 9 are further diagrams showing two modifications of the pump and in accordance with the present invention.

In the construction illustrated in FIGS. 1 to 3, an aluminium reservoir casting **10** has lugs **11** for the attachment of a cover **12**, to the underside of which the body **13** of the pump unit is secured by bolts **14**. Within such body a cylinder liner **15** is located between an upper gland assembly **16** and a lower gland assembly **17**, the lower part of such liner being reduced in external diameter at **15***a* and formed with through holes **15***b*. Slidable within the liner **15** is a piston **18** provided at its upper side with an actuating piston rod **19** and at its lower side with a separate piston road **20** serving as a pump plunger. The piston is secured by means of a spring ring **18***a* against a shoulder formed at the junction of the rods **18**, **20**, a sealing ring **21** being also provided at such shoulder. The lower piston rod or plunger **20** has an axial bore **20***a* and is fitted at its free end with a non-return inlet valve **22**. Adjacent the shoulder aforesaid diametrally-opposed ports **20***b* enable hydraulic fluid entering through the bore **20***a* to reach the lower end of the cylinder below the piston **18**. The upper and lower chambers of the cylinder are designated A and B respectively, and within the pump body **13** is formed a third displacement chamber C in which works the plunger **20** aforesaid, the cross-sectional areas of the several chambers being relatively important and suitable proportions thereof being given later herein.

Within the pump body **13** are formed several communicating passages including an inlet passage **23** whose lower end communicates with the displacement chamber C through a non-return inlet valve **24** and passage **24***a* and whose upper end is provided with a non-return valve **25** controlling a passage **26** which communicates with upper chambers A of the cylinder by way of an outer circumferential groove **15***c* and ports **15***d* in the wall of the liner **15**. At its lower end the inlet passage **23** is fitted with a filter **27**.

The outlet passages in the pump body **13** comprise a bore **28** whose upper end communicates with the cylinder chamber A by way of a cross passage **29** leading to the liner groove **15***c* and whose lower part communicates with the cylinder chamber B by way of a cross passage **30** leading to the externally relieved portion **15***a* of the liner. Within the lower part of the bore **28**, which thus

provides a connecting passage between the chambers A, B, is slidably mounted a close-fitting valve rod **31** urged by a spring **32** towards a position in which it prevents transfer of fluid between the chambers aforesaid, this spring being adjustably loaded by a screw plug **32***a*. Parallel with the bore **28** is the final outlet passage **33** which communicates with the cross passages **29**, **30** by way of non-return valves **34**, **35** respectively.

Also within the pump body **13** and in communication with the final outlet passage **33** (FIG. 3) is a pressure release device consisting of a spring-loaded ball **36** and a plunger **37** adapted, when screwed down, to unseat such ball and thereby permit return of hydraulic fluid into the reservoir **10** by way of the tube **38**. A safety pressure-relief valve is fitted at **39** (FIG. 2) and a filler plug is provided at **40** in the cover **12**. On top of the latter is a mounting bracket **41** for the L shaped actuating lever **42**, whose shorter arm carries slide-blocks **43** engaged with the upper end of the piston rod **19**.

During operation of the unit as a low-pressure pump, as shown in FIGS. 4 and 5, the valve rod **31** prevents communication between the chambers A, B, so long as the pressure in the connecting passage **28** is below a predetermined value determined by the strength of the spring **32** and the setting of its adjusting plug **32***a*. On the upward stroke of the piston **18** (FIG. 4), the delivery volume is represented by the capacity of the upper cylinder chamber A the valves **25**, **35** being closed and the valves **22**, **24**, **34** being open. On the downward stroke of the piston (FIG. 5), the delivery volume consists of the combined capacities of the chambers B, C, the valves **24**, **34** being closed and the valves **22**, **25**, **35** being open.

When the pressure in the connecting passage **28** becomes sufficient to push back the valve rod **31** therein, as shown in FIGS. 6 and 7, communication is established between the chambers A, B. On the upward stroke of the piston **18**, the delivery is a differential volume comprising the capacity of the chamber A minus that of the chamber B, this reduction of the delivery volume enabling the operator to work against higher pressures without having to exert greater force on the lever **42**. During this stroke, the valves **22**, **25** are closed whilst the valve **24** and either or both of the valves **34**, **35** open to pass the delivery volume to the final outlet passage **33**. On the downward stroke, the delivery volume will be equal to the displacement of the piston rod **19**, which is nearly the same as the combined capacity of the chambers B, C less that of the chamber A.

In the modification shown in FIG. 8, a further non-return valve **44** is provided in the main inlet passage **23**, as well as a manually-operable valve **45** by-passing the non-return valve **24**. If the valve **45** is opened, then the single-acting pump represented by the plunger **20** in the chamber C becomes inoperative whilst the piston **18** and cylinder **15** function as a normal double-acting pump, whether or not the delivery pressure is sufficient to affect the valve rod **31**. This valve **45** may be opened for either high- or low-pressure actuation; that is to say, regardless of the position of the valve rod **31**.

In the further modified construction shown in FIG. 9, the pressure-responsive rod **31** is replaced by a manually-operable valve **46** so that the operator can, himself, decide whether to work the pump on maximum or reduced delivery rate.

Convenient proportions for the cross-sectional areas of the upper and lower cylinder chambers A and B, the plunger **20** and the piston rod **19** are as follows:

Chamber A=.921
Chamber B=.626
Plunger **20** } =.601
Chamber C }
Piston-rod **19**=.306



On this basis the delivery volume per unit distance of piston travel will be in the following ratios—

Low pressure:

Up=.921=A } Mean volume=1.074
Down=1.227=B—C }

(The maximum force required is, therefore, determined by the down stroke.)

High pressure:

Up=.295=A—B } Mean volume =.300
Down=.306=B—C—A }

(displacement of piston-rod **16**)

Intermediate pressure (FIG. 8):

Up=.921=A } Mean volume=.774
Down=.626=B }

(The maximum force required is determined by the up stroke.)

The invention is obviously not limited to all the details above set forth and illustrated, these being given by way of example only and some of them being capable of modification without departure from the scope of the invention.

I claim:

1. A reciprocating dual-pressure hydraulic pump with pressure-sensitive valve means for automatic change-over from low to high pressure operation and vice-versa comprising, large and small cylinders, a piston in the large cylinder forming a double-acting pump, the piston areas of the two faces of said double-acting piston in said large cylinder being of different areas, a plunger in the smaller cylinder forming a single-acting pump, means for moving the piston and plunger simultaneously, automatic inlet valves for one end of each cylinder, a passage connecting the other end of the large cylinder with the smaller cylinder, a non-return valve allowing fluid to pass through said passage from the smaller cylinder, outlet passages from each end of the larger cylinder, non-return valves in each outlet passage, a transfer passage connecting the said outlet passages, said transfer passage being disposed between said cylinders and said non-return valves, a piston valve operable in said transfer passage, said piston valve adapted to enter said transfer passage from the one of said outlet passages which is connected to the said other end of the large cylinder, and spring means urging the said piston valve into said passage to close the same.

2. A reciprocating pump as claimed in claim 1 characterized by a manually controllable valve and by-pass for the inlet valve of the smaller cylinder.

3. A reciprocating pump as claimed in claim 1 characterized in that the passage connecting the end of the large cylinder with the smaller cylinder is formed in the plunger.

4. A reciprocating pump as claimed in claim 3 characterized in that the non-return outlet valves are connected to the ends of said transfer passage which is controlled by said piston valve.


**References Cited** in the file of this patent

UNITED STATES PATENTS

| | | |
|---|---|---|
| 2,410,808 | Christensen | Nov. 12, 1946 |
| 2,686,477 | Hoschle et al. | Aug. 17, 1954 |
| 2,989,001 | Wilkenloh et al. | June 20, 1961 |

FOREIGN PATENTS

| | | |
|---|---|---|
| 565,125 | France | Jan. 19, 1924 |
| 1,075,780 | France | Apr. 14, 1954 |

OTHER REFERENCES

German application (KL5C–10/01), 1,007,715, May 9, 1957.