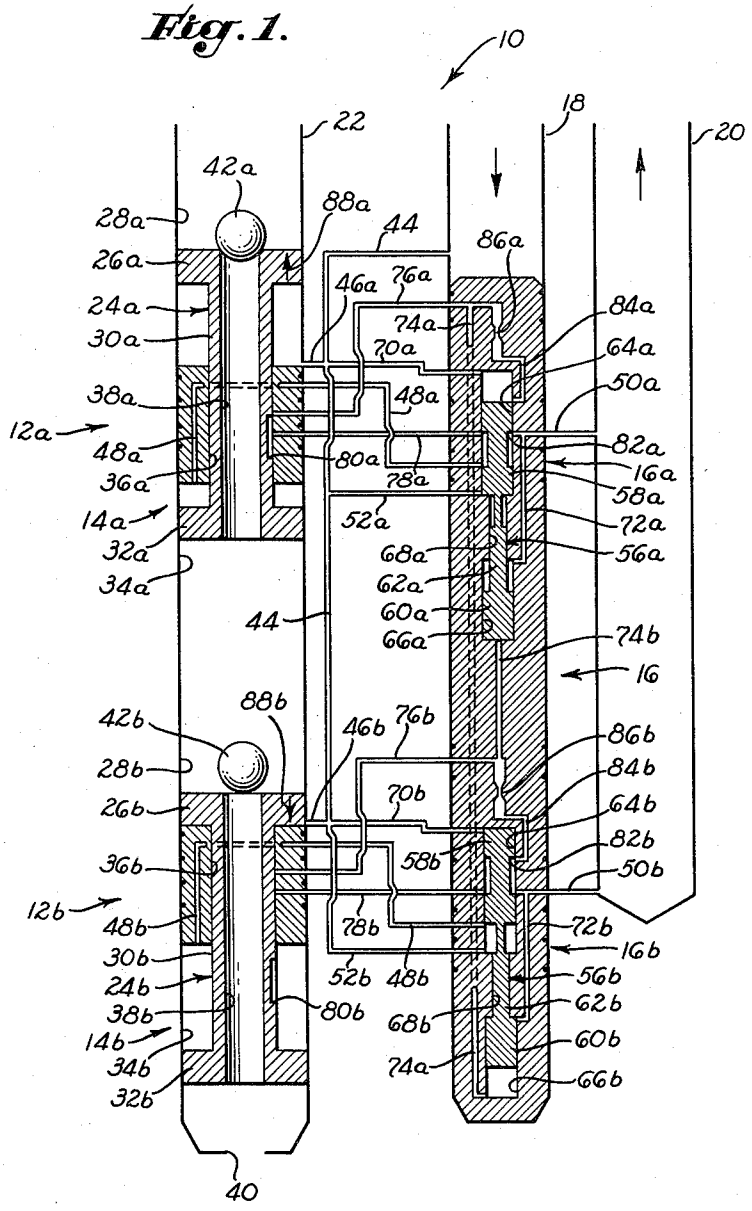
June 7, 1960 C. J. COBERLY 2,939,397

FLUID OPERATED MULTIPLEX PUMP

Filed Dec. 10, 1956 4 Sheets-Sheet 1

Fig. 1.



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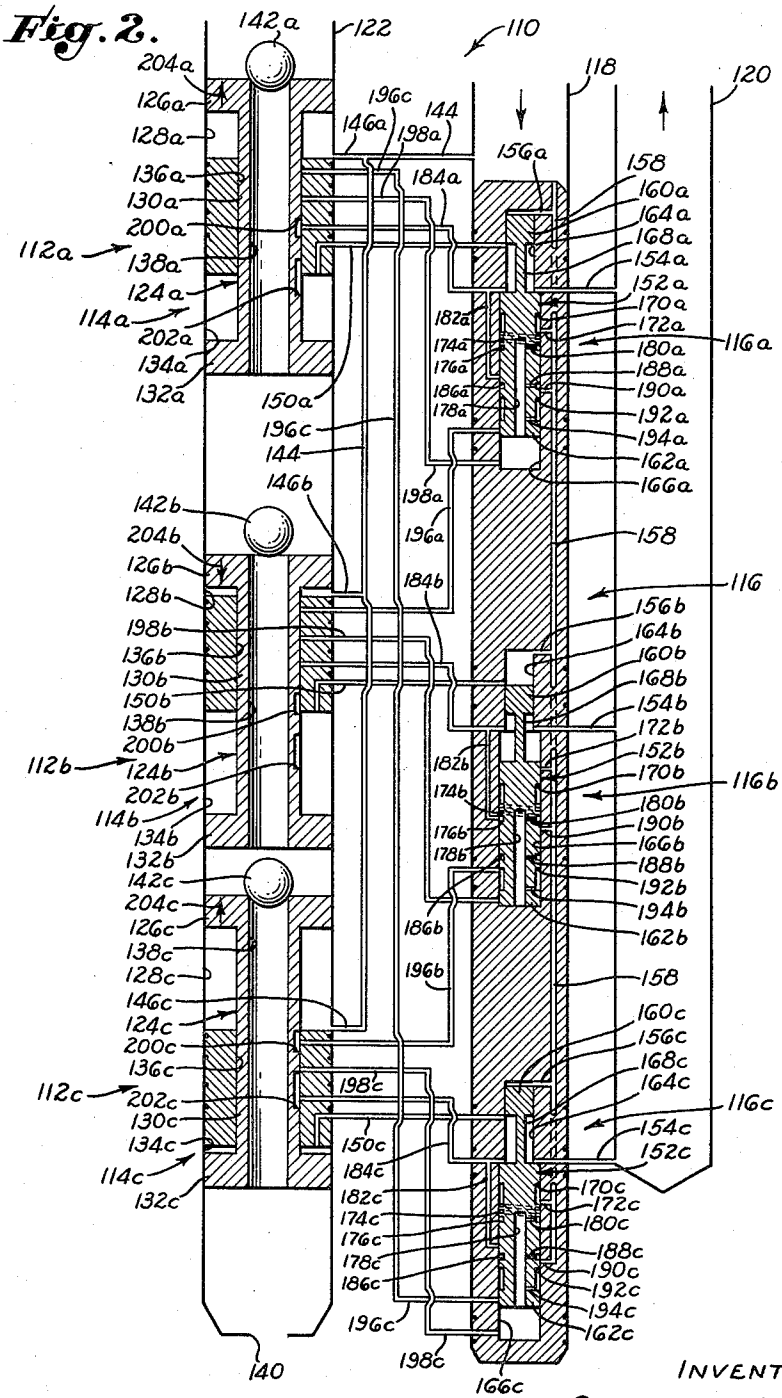
BY HIS ATTORNEYS.
HARRIS, KIECH, FOSTER & HARRIS

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Fig. 2.



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Fig. 3.

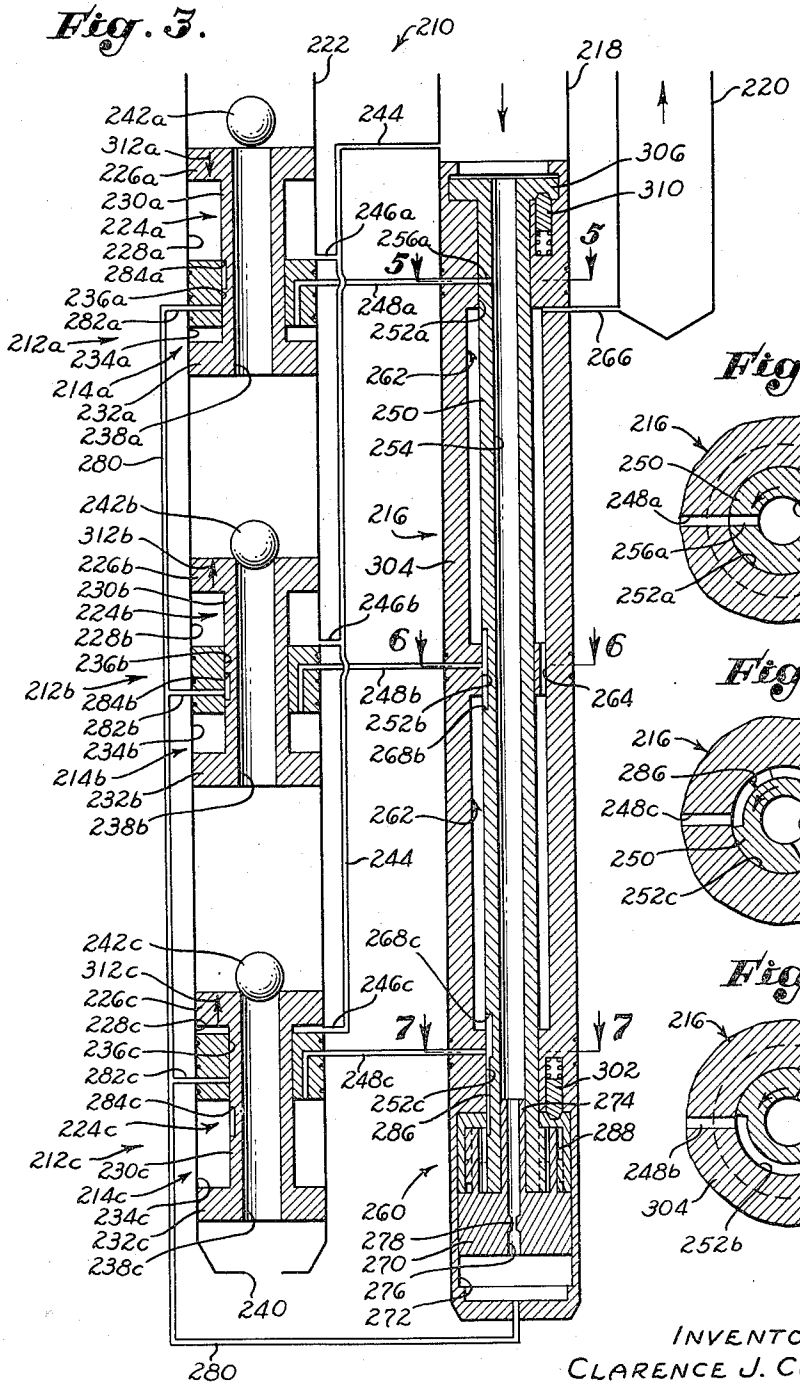


Fig. 5.

Fig. 7.

Fig. 6.

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June 7, 1960 C. J. COBERLY 2,939,397

FLUID OPERATED MULTIPLEX PUMP

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Fig. 4.

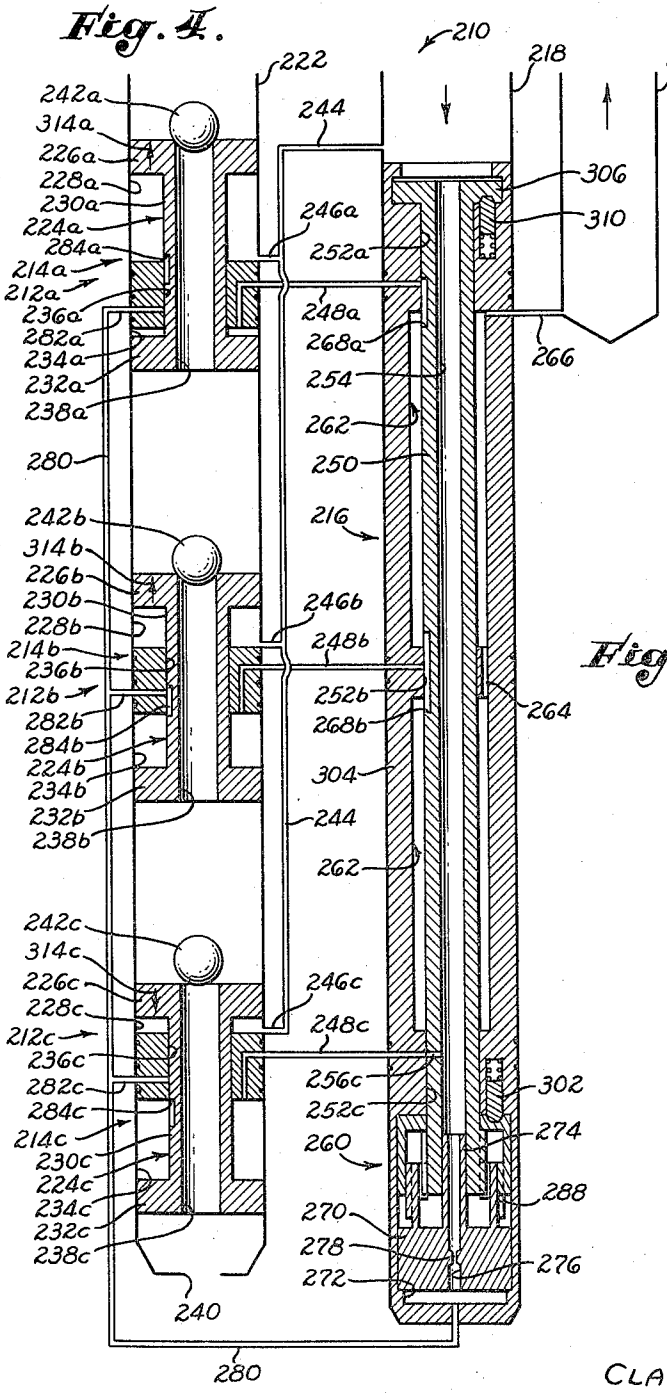
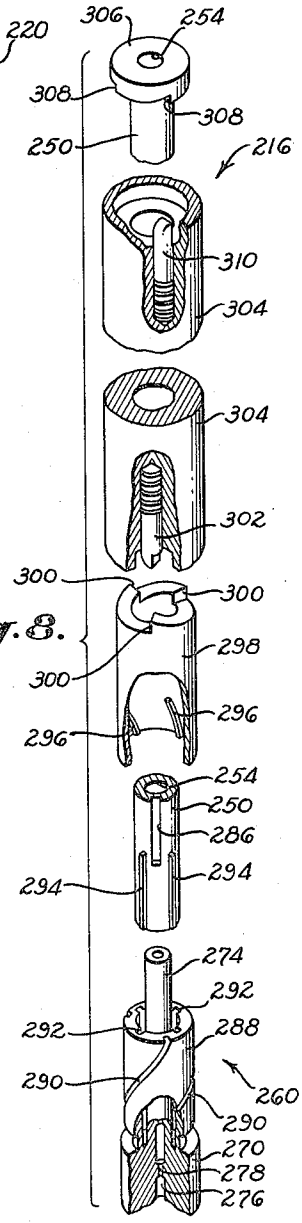


Fig. 8.



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United States Patent Office 2,939,397 Patented June 7, 1960

2,939,397

FLUID OPERATED MULTIPLEX PUMP

Clarence J. Coberly, San Marino, Calif., assignor to Kobe, Inc., Huntington Park, Calif., a corporation of California

Filed Dec. 10, 1956, Ser. No. 627,298

15 Claims. (Cl. 103—5)

The present invention relates in general to fluid operated or hydraulic bottom hole pumps for wells, and particularly oil wells, and a general object of the invention is to provide a pump of this type which achieves a high pumping capacity with a minimum of hydraulic shock.

Basically, a fluid operated oil well pump of the reciprocating type includes interconnected engine and pump piston means and includes engine valve means for applying pressure forces to the engine piston means alternately in opposite directions so as to reciprocate the two piston means, whereby the pump piston means pumps fluid from the well upwardly through a production tubing to the surface. The operating fluid for actuating the pump, which is usually clean crude oil, is conveyed downwardly thereto at a relatively high pressure through a supply tubing set in the well. The spent operating fluid discharged by the pump at a lower pressure is either mixed with the production fluid and returned to the surface through the production tubing, or is returned to the surface through a separate, return tubing set in the well.

The engine and pump piston means of a fluid operated pump of the foregoing character are incorporated in a pump unit and the engine valve means thereof is incorporated in a valve unit which may be combined with the structure of the pump unit, or which may be a separate structure, as disclosed in my copending application Serial No. 598,785, filed July 19, 1956. In the first instance, the pump and valve units are installed in and removed from the well in unison, while in the second the pump and valve units may be installed in and removed from the well independently of each other. The present invention is applicable to either case.

Fluid operated pumps are also of either the free type or the set type, a pump of the former type being movable between its operating position in the well and the surface through one of the tubings of the tubing system associated therewith so that removal of the tubing system in order to remove the pump is unnecessary. With a pump of the set type, on the other hand, removal of at least part of the tubing system is necessary to permit removal of the pump since the pump is attached to one of the tubings. Again, the present invention is applicable to pumps of either of these types.

Fluid operated oil well pumps may be of either the single acting or double acting type, the former being the more common since it is simpler and since it permits the use of larger engine and pump pistons or piston means, the pistons of a double acting pump necessarily being smaller than those of a single acting one because of the additional passages required for the operating fluid and the pumped fluid. Any single acting pump has the limitation that only one stroke is the working stroke so that a high piston speed must be used to obtain a reasonably adequate pumping capacity. Also, a single acting pump has the disadvantage that the flows of both the operating fluid and the pumped fluid are intermittent,

which tends to introduce hydraulic shock, this being particularly true when the pump is run at high speed. Hydraulic shock, of course, is responsible for many of the repairs required by fluid operated pumps, particularly those utilized in deep wells.

Double acting pumps have a considerable advantage over the single acting type in that both strokes are working strokes. Thus, a double acting pump gets the benefit of the full piston speed, instead of one-half of the piston speed as in a single acting pump. However, the displacement is usually not double that of a corresponding single acting pump since the engine and pump piston means must be smaller for the reasons previously outlined. Double acting pumps do permit more uniform flows of the operating and pumped fluids and thus are less likely to produce hydraulic shock than a single acting pump. However, the piston means of double acting pumps must stop at each end of their stroke, which results in momentary interruptions of the flows of the operating and pumped fluids and tends to produce some hydraulic shock. Also, all single acting and double acting pumps are subject to the tendency to race when the pump cylinder does not completely fill with solid fluid. In prior fluid operated pumps, it is necessary to provide some form of governing means to prevent such racing, which makes for complexity. While various features of the invention may be embodied in either a single acting pump or a double acting pump, the invention is preferably applied to a pump of the single acting type since the basic objects of the invention may be achieved with this type and the more complex structure of the double acting type is thus avoided.

With the foregoing as background, a primary object of the invention is, as hereinbefore suggested, to provide a fluid operated bottom hole pump having a much higher pumping capacity than pumps heretofore available and having substantially no tendency to introduce hydraulic shock into the system, the latter effect in particular being achieved by producing substantially continuous flow of both the operating and pumped fluids.

More particularly, a basic object of the invention is to provide a fluid operated pump which may conveniently be termed a "multiplex" pump and which includes two or more fluid operated pump units having parallel-connected engine means and series connected pump means, this multiplex pump including control means for operating such pump units in timed relation, with predetermined phase relations therebetween, in such a manner that at least one of the pump units is on its working stroke at all times. Another object in this connection is to provide a multiplex pump comprising two or more fluid operated pump units wherein only one of the pump units is on its return stroke at any instant, the remaining pump unit or units being on its or their working stroke or strokes.

A further object of the invention is to provide a fluid operated multiplex pump in which the control means overlaps the working strokes of the pump units in such a way that the pumping load is progressively shifted from one pump unit to another as each pump unit approaches the end of its working stroke. In other words, a further object of the invention is to provide a multiplex pump having control means for initiating the working stroke of the engine and pump piston means of the next pump unit in the series as the engine and pump piston means of the preceding pump unit therein approach the end of their working stroke.

With the foregoing construction, a pumping capacity very much higher than anything heretofore attainable under comparable conditions is achieved through the use of two or more fluid operated pump units, which is an

important feature of the invention. Also, since the pump units are operated in timed, overlapping relation with the engine means thereof in parallel and the pump means thereof in series, the flows of operating fluid and pumped fluid through the tubing system with which the multiplex pump is associated are substantially constant to minimize hydraulic shock, which is another important feature of the invention. Further, since the pump means of the pump units are connected in series, and since the pumping load is divided among the pump means of two or more pump units in any embodiment which includes three or more pump units, either the operating fluid pressure, or the areas of the engine means of the pump units on which the operating fluid acts, may be reduced, which is still another important feature.

Another object of the invention is to provide a multiplex pump wherein the control means for maintaining the proper phasing between or among the pump units is also fluid operated and is actuated by the pump units as the piston means thereof approach the ends of their working strokes. Thus, as the piston means of one pump unit approach the end of their pumping stroke, they actuate the control means to cause the latter to initiate the working stroke of the next pump unit in the series.

An object in connection with one embodiment of the invention is to provide a multiplex pump wherein the control means includes a number of valve units corresponding to the number of pump units and wherein the valve units control the respective pump units. A related object is to provide such a control means wherein each pump unit in turn either directly or indirectly actuates the valve unit corresponding thereto, and also actuates the valve unit corresponding to another of the pump units. Thus, the operations of the various pump and valve units are interrelated so as to achieve the proper phasing of the pump units.

Another object is to provide a control means which includes a single valve unit controlling all of the pump units and controlled thereby in turn to achieve the desired phasing of the pump units. More particularly, an object is to provide a control means comprising a single valve unit which includes a rotary valve and fluid operated means actuable by the various pump units for rotating the valve from one of its operating positions to the next as the piston means of each pump unit approach the end of its working stroke.

Another object of importance is to provide a multiplex pump having two or more pump units provided with pump piston means and working valves related in such a manner in structure and operation that no standing valve or valves are required. This result is achieved by so phasing the operation of the pump units that at least one of the pump piston means is on its working stroke at all times so that the corresponding working valve is closed, thereby obviating any necessity for one or more standing valves.

Another important object of the invention is to provide a multiplex pump wherein the pump units are axially aligned and wherein the pumped fluid flows through axial passages in the engine and pump piston means of the pump units seriatim, thereby minimizing flow resistance. A related object is to provide each engine and pump piston means with a working valve preventing flow through the corresponding axial passage in the reverse direction and to so phase the working strokes of the various piston means that at least one is always on its working stroke so that the corresponding working valve is closed, whereby no standing valve or valves are required.

A further object of the invention is to provide a multiplex fluid operated pump including two or more fluid operated pump units which are independent, but interrelated in operation, and which have some or all of the features hereinbefore outlined, each pump unit having its own fluid operated engine means and its own pump means.

Another object is to provide a fluid operated multiplex pump wherein the engine and pump piston means of each pump unit move through their return stroke at a speed such that they are ready for their working stroke before the engine and pump piston means of another of the pump units reach the end of their working stroke. In the case of a triplex pump embodying the invention, this means a return stroke speed at least twice the working stroke speed, and another object of the invention is to provide such a triplex pump.

The foregoing objects, advantages, features and results of the present invention, together with various other objects, advantages, features and results thereof which are presented in the original claims of this application, or which will be apparent to those skilled in the fluid operated pump art in the light of this disclosure, may be attained with the exemplary embodiments of the invention described in detail hereinafter and illustrated in the accompanying drawings, in which:

Fig. 1 is a diagrammatic view illustrating a fluid operated duplex pump which embodies the invention;

Fig. 2 is a diagrammatic view of a fluid operated triplex pump embodying the invention;

Fig. 3 is a diagrammatic view which illustrates another embodiment of a fluid operated triplex pump of the invention;

Fig. 4 is a view similar to Fig. 3, but illustrating various components in different operating positions;

Figs. 5, 6 and 7 are enlarged, fragmentary sectional views respectively taken along the arrowed lines 5—5, 6—6 and 7—7 of Fig. 3; and

Fig. 8 is an exploded perspective view illustrating fragmentarily a pump-unit control means of the invention incorporated in the triplex pump of Fig. 3 of the drawings.

Duplex pump 10

Considering Fig. 1 of the drawings, illustrated therein is a duplex pump **10** of the invention which includes two independent but interrelated fluid operated pumping devices **12***a* and **12***b* respectively including fluid operated pump units **14***a* and **14***b*. As will become apparent, each of these pump units has its own fluid operated engine means and its own pump means, the two engine means being connected in parallel and the two pump means in series. The two pump units **14***a* and **14***b* are controlled by a control means **16** which includes independent but interrelated engine valve units **16***a* and **16***b*, these valve units forming parts of the pumping devices **12***a* and **12***b*, respectively, and controlling the pump units **14***a* and **14***b*, respectively.

The foregoing components of the pump **10** are actuated by operating fluid under pressure delivered to the pump through a supply tubing **18**. In the particular construction illustrated, the spent operating fluid is returned to the surface through a return tubing **20**. A production tubing **22** serves to convey the production fluid pumped by the pump units **14***a* and **14***b* to the surface.

The pump units **14***a* and **14***b* are similar to those disclosed in my aforementioned copending application Serial No. 598,785 and include combined engine and pump pistons or piston means **24***a* and **24***b*, respectively. The pistons **24***a* and **24***b* respectively include plungers **26***a* and **26***b* reciprocable in cylinders **28***a* and **28***b*, and include rods **30***a* and **30***b* connecting the plungers **26***a* and **26***b* to plungers **32***a* and **32***b* reciprocable in cylinders **34***a* and **34***b*. The rods **30***a* and **30***b* extend through bores **36***a* and **36***b* in fluid-tight, sliding engagement with the walls thereof.

The pistons **24***a* and **24***b* are provided therethrough with axial passages **38***a* and **38***b* for fluid being pumped from an inlet **40** into the production tubing **22**, the passages **38***a* and **38***b* being connected in series and reverse flow therethrough being prevented by working valves **42***a* and **42***b*, respectively. As will be apparent, as long as the working stroke of each piston **24***a* and **24***b*, the

working stroke thereof being its upward stroke, is initiated before the end of the working stroke of the other so that one of the pistons is always on its working stroke, one of the working valves 42*a* and 42*b* is always closed to prevent reverse flow from the production tubing 22 to the inlet 40. Thus no standing valve or valves are required, which is an important feature. The manner in which the desired phasing of the working strokes of the pistons 24*a* and 24*b* is maintained will be considered hereinafter, this being accomplished by the control means 16.

As hereinbefore suggested, during the working stroke of each piston 24*a* and 24*b*, the corresponding one of the working valves 42*a* and 42*b* is seated so that the pumped fluid is displaced upwardly through the production tubing 22. During the return stroke of each piston 24*a* and 24*b*, which is the downward stroke thereof, the corresponding one of the working valves 42*a* and 42*b* is open to permit bypassing of pumped fluid from below such piston through the corresponding one of the passages 38*a* and 38*b* into the corresponding one of the cylinders 28*a* and 28*b* thereabove.

The working stroke of the piston 24*a* is affected by applying the operating fluid pressure in the supply tubing 18 to the lower surface of the plunger 26*a* and the spent operating fluid pressure or exhaust pressure in the return tubing 20 to the upper surface of the plunger 32*a*, whereupon the piston 24*a* moves upwardly to displace pumped fluid upwardly through the production tubing 22 and to draw pumped fluid upwardly through the axial passage 38*b* through the piston 24*b*. The working stroke of the piston 24*b* is effected in the same way and, as this piston moves upwardly through its working stroke, it displaces pumped fluid upwardly through the axial passage 38*a* in the piston 24*a* and draws fluid upwardly through the inlet 40. The return stroke of the piston 24*a* is produced by applying the operating fluid pressure to both the lower surface of the plunger 26*a* and the upper surface of the plunger 32*a*, the area of the latter surface slightly exceeding that of the former so that the piston 24*a* moves downwardly through its return stroke when equal pressures are applied to such surfaces. Since, in order to effect the return stroke of the piston 24*a*, it is only necessary to overcome mechanical friction between the plungers 26*a* and 32*a* and the cylinders 28*a* and 34*a* and fluid friction resulting from the flow of pumped fluid through the axial passage 38*a* past the working valve 42*a*, the area of the upper surface of the plunger 32*a* needs be only slightly greater than the area of the lower surface of the plunger 26*a*. This may be accomplished by making the diameter of the cylinder 34*a* a few thousandths of an inch greater than the diameter of the cylinder 28*a*. The return stroke of the piston 24*b* is effected in exactly the same way so that a detailed description is unnecessary.

Considering the manner in which the operating fluid and exhaust pressures are applied to the pistons 24*a* and 24*b* to accomplish the foregoing, the operating fluid pressure is constantly applied to the lower surfaces of the plungers 26*a* and 26*b* through a main passage 44 and branch passages 46*a* and 46*b*, the main passage communicating with the supply tubing 18 and the branch passages communicating with the cylinders 28*a* and 28*b*. Communicating with the cylinders 34*a* and 34*b* to apply either operating fluid pressure or exhaust pressure to the upper surfaces of the plungers 32*a* and 32*b* are passages 48*a* and 48*b*, these passages being controlled by the valve units 16*a* and 16*b*, respectively. When the exhaust pressure is applied to the upper surfaces of the plungers 32*a* and 32*b* to produce the working strokes of the pistons 24*a* and 24*b*, the valve units 16*a* and 16*b* connect the passages 48*a* and 48*b* to the return tubing 20 through passages 50*a* and 50*b*, respectively. Conversely, when the operating fluid pressure is applied to these areas to produce the return strokes of the pistons 24*a* and 24*b*, the valves 16*a* and 16*b* connect the passages 48*a* and 48*b* to branch passages 52*a* and 52*b*, respectively, these branch passages communicating with the main passage 44 from the supply tubing 18.

Considering the control means 16 in more detail, the valve units 16*a* and 16*b* respectively include differential area engine valves 56*a* and 56*b* respectively having plungers 58*a* and 58*b* connected to plungers 60*a* and 60*b* by rods 62*a* and 62*b*, these plungers and rods being slidable in suitable bores or cylinders in fluid-tight engagement with the walls thereof, as will be apparent from Fig. 1 of the drawings. The bores for the plungers 58*a* and 58*b* are identified by the reference characters 64*a* and 64*b*, the bores for the plungers 60*a* and 56*b* are identified by the reference characters 66*a* and 66*b*, and the bores for the rods 62*a* and 62*b* are identified by the reference characters 68*a* and 68*b*.

The operating fluid pressure is constantly applied to the upper ends of the valves 56*a* and 56*b* through branch passages 70*a* and 70*b* leading from the main passage 44 to the upper ends of the bores 64*a* and 64*b*. The passages 52*a* and 52*b* communicate with the lower ends of the bores 64*a* and 64*b* so as to constantly apply the operating fluid pressure to the annular areas at the lower ends of the plungers 58*a* and 58*b*. The annular areas at the upper ends of the plungers 60*a* and 60*b* are constantly exposed to the exhaust pressure through branch passages 72*a* and 72*b* communicating with the passages 50*a* and 50*b* leading from the return tubing 20, the passages 50*a* and 50*b* also communicating with the bores 64*a* and 64*b* intermediate the ends thereof. With the foregoing pressures constantly applied to the upper ends of the plungers 58*a* and 58*b*, the annular areas at the lower ends of these plungers and the annular areas at the upper ends of the plungers 60*a* and 60*b*, the valves 56*a* and 56*b* will move upwardly when the operating fluid pressure is applied to the lower ends of the plungers 60*a* and 60*b* and will move downwardly when the exhaust pressure is applied thereto. When the lower ends of the valves 56*a* and 56*b* are exposed to exhaust pressure in the bores 66*a* and 66*b*, the net downward force applied to each valve is equal to the product of the cross sectional area of the corresponding one of the rods 62*a* and 62*b* and the difference between the operating fluid and spent operating fluid pressures. When the lower ends of the plungers 60*a* and 60*b* are exposed to operating fluid pressure in the bores 66*a* and 66*b*, the net upward force on each of the valves 56*a* and 56*b* is equal to the product of the difference between the operating fluid and exhaust pressures and the difference between the cross sectional area of the corresponding one of the rods 62*a* and 62*b* and the cross sectional area of the corresponding plungers, assuming that both plungers of each valve have the same cross sectional area.

Considering the manner in which either the operating fluid pressure or the exhaust pressure is applied to the lower ends of the valves 56*a* and 56*b*, passages 74*a* and 74*b* respectively communicate with the lower ends of the bores 66*b* and 66*a* and thus communicate with the lower ends of the plungers 60*b* and 60*a*. The passages 74*a* and 74*b* communicate with passages 76*a* and 76*b* leading to the bores 36*a* and 36*b* in which the rods 30*a* and 30*b* of the pump-unit pistons 24*a* and 24*b* are disposed, these passages being referred to hereinafter as control passages. The passages 76*a* and 76*b* are adapted to be connected to passages 78*a* and 78*b* by control ports 80*a* and 80*b* in the rods 30*a* and 30*b* as the pistons 24*a* and 24*b* approach the ends of their working strokes, but before these pistons reach the ends of their working strokes. The passages 78*a* and 78*b*, which are also control passages, communicate with the bores 64*a* and 64*b* and constantly register with annular channels 82*a* and 82*b* in the plungers 58*a* and 58*b*. The channels 82*a* and 82*b* also constantly communicate with the passages 50*a* and 50*b* leading to the return tubing 20 so that the spent operating fluid pressure

is constantly present in these channels. When the valves 56*a* and 56*b* are in their lower positions, the channels 82*a* and 82*b* also communicate with the passages 48*a* and 48*b* leading to the pump-unit cylinders 34*a* and 34*b*. However, when the valves 56*a* and 56*b* are in their upper positions, the channels 82*a* and 82*b* also communicate with passages 84*a* and 84*b* leading from the bores 64*a* and 64*b* to the passages 74*a* and 74*b* through restrictors 86*a* and 86*b*.

The foregoing interconnections between the pump and valve units 14*a*, 14*b*, 16*a* and 16*b* interrelate the operations of these units in such a way that the flow of pumped fluid through the production tubing 22 and the flows of operating fluid under pressure and spent operating fluid through the supply and return tubings 18 and 20 are substantially constant to minimize hydraulic shock. This interrelation will become apparent from the following description of the operation of the duplex pump 10.

Operation of pump 10

It is thought that the operation of the duplex pump 10 may be understood more readily if it is kept in mind that, insofar as the operation of the pump units 14*a* and 14*b* is concerned, the valves 56*a* and 56*b* control the working and return strokes of the pistons 24*a* and 24*b*, these pistons moving upwardly through their working strokes when the valves are in their lower positions and moving downwardly through their return strokes when the valves are in their upper positions. As regards upward movement of the valves 56*a* and 56*b*, the valve 56*b* controls the valve 56*a* and the valve 56*a* controls the valve 56*b*. With respect to downward movement of the valves 56*a* and 56*b*, the piston 24*a* controls the valve 56*b* and the piston 24*b* controls the valve 56*a*. With the foregoing in mind, the operation of the duplex pump 10 will now be considered in detail.

It will be assumed that the pistons 24*a* and 24*b* and the valves 56*a* and 56*b* are in the positions shown in Fig. 1 of the drawings. This being the case, the piston 24*a* is moving upwardly through its working stroke, as indicated by the arrow 88*a*, and the piston 24*b* is moving downwardly through its return stroke, as indicated by the arrow 88*b*, or has just reached the end of its return stroke. The working stroke of the piston 24*a* results from the application of exhaust pressure to the upper surface of the plunger 32*a* through the passage 48*a*, the annular channel 82*a* in the valve 56*a*, and the passage 50*a*. The return stroke of the piston 24*b* results from the application of the operating fluid pressure to the upper surface of the plunger 32*b* through the passage 48*b*, the lower end of the bore 54*b*, the passage 52*b* and the passage 44. During the working stroke of the piston 24*a*, the working valve 42*a* is closed so that pumped fluid is displaced upwardly through the production tubing 22. At the same time, the working valve 42*b* is open as the piston 24*b* negotiates its return stroke so that pumped fluid flows from the inlet 40 through the piston 24*b* into the region between the two pistons.

As the piston 24*a* approaches the end of its working stroke, but before it reaches the end of such stroke, the control port 80*a* bridges the control passages 76*a* and 78*a*. When this occurs, the lower end of the valve 56*b* is exposed to the exhaust pressure through the passage 74*a*, the control passage 76*a*, the control port 80*a*, the control passage 78*a*, the annular channel 82*a* in the valve 56*a*, and the passage 50*a*. Consequently, the valve 56*b* moves downwardly into its lower position, with two results.

First, it disconnects the upper surface of the plunger 32*b* of the piston 24*b* from the supply tubing 18 and connects it to the return tubing 20 through the passage 48*b*, the annular channel 82*b* in the valve 56*b*, and the passage 50*b*. Consequently, the working stroke of the piston 24*b* is initiated to displace pumped fluid upwardly, the working stroke of this piston being initiated before the working stroke of the piston 24*a* is terminated so that the pumping load is gradually shifted from the piston 24*a* to the piston 24*b* to avoid hydraulic shock, which is an important feature of the invention.

The second thing that happens when the valve 56*b* moves downwardly into its lower postiion is that it applies the operating fluid pressure to the lower end of the valve 56*a* through the passage 44, the passage 70*b*, the bore 64*b* above the valve 56*b*, the passage 84*b*, the restrictor 86*b* and the passage 74*b*, the passage 76*b* being closed by the rod 30*b* under such conditions. This application of the operating fluid pressure to the lower end of the valve 56*a* through the restrictor 86*b* produces upward movement of the valve 56*a* toward its upper position at a rate controlled by the restrictor 86*b*, which rate is such as to delay termination of the working stroke of the piston 24*a* until the piston 24*b* attains full speed on its working stroke. When the valve 56*a* reaches its upper position, it disconnects the upper surface of the plunger 32*a* of the piston 24*a* from the return tubing 20 and connects it to the supply tubing 18 through the passage 44, the passage 52*a*, the bore 64*a* below the plunger 58*a* of the valve and the passage 48*a*. Consequently, the return stroke of the piston 24*a* is initiated. The piston 24*a* moves downwardly at a substantially uniform rate which is relatively high due to the fact that only mechanical friction and fluid friction through the axial passage 38*a* past the working valve 42*a* must be overcome. Consequently, the piston 24*a* reaches the end of its return stroke before the piston 24*b* completes its working stroke, and remains in its lower position until the piston 24*b* approaches the end of its working stroke. The desired speed of the piston 24*a* during its return stroke is readily achieved by making the area of the upper surface of the plunger 32*a* sufficiently larger than the area of the lower surface of the plunger 26*a*.

The valve 56*b* is held in its lower position during the working stroke of the piston 24*b* in the following manner. When the valve 56*a* is in its upper position, as hereinbefore described, the passage 84*a* communicates with the return tubing 20 through the annular channel 82*a* in the valve 56*a* and the passage 50*a*. Consequently, the exhaust pressure in the return tubing 20 is applied to the lower end of the valve 56*b* through the passage 50*a*, the annular channel 82*a* in the valve 56*a*, the passage 84*a*, the restrictor 86*a* and the passage 74*a*, the passage 76*a* being closed by the rod 30*a*. Thus, a hydraulic locking means for maintaining the valve 56*b* in its lower position throughout the working stroke of the piston 24*b* is provided.

Summarizing the foregoing, as the piston 24*a* approaches the end of its working stroke, it moves the valve 56*b* downwardly to start the working stroke of the piston 24*b* before the piston 24*a* completes its working stroke. The valve 56*b* moves the valve 56*a* upwardly to initiate the return stroke of the piston 24*a*. In a similar manner, the piston 24*b* controls the valve 56*a*, which, in turn, controls the valve 56*b*. In other words, as the piston 24*b* approaches the end of its working stroke, it moves the valve 56*a* downwardly to initiate the working stroke of the piston 24*a*. The valve 56*a* also moves the valve 56*b* upwardly to initiate the return stroke of the piston 24*b*.

Considering how the foregoing is accomplished, as the piston 24*b* approaches the end of its working stroke, but before it completes such stroke, the control port 80*b* bridges the control passages 76*b* and 78*b* to apply exhaust pressure to the lower end of the valve 56*a* through the passage 50*b*, the annular channel 82*b* in the valve 56*b*, the control passage 78*b*, the control port 80*b*, the control passage 76*b*, and the passage 74*b*. Consequently, the valve 56*a* moves downwardly, the flow resistance offered by the restrictor 86*b* being sufficiently high that, even though it is open to the operating fluid pressure in the supply tubing 18 through the passage 44, the passage 70*b*, the bore 64*b* above the valve 56*b*, and the passage 84*b*

under such conditions, it does not appreciably increase the pressure applied to the lower end of the valve **56***a* above the exhaust pressure. When the valve **56***a* moves downwardly into its lower position in the foregoing manner, two events take place.

First, the upper surface of the plunger **32***a* of the piston **24***a* is disconnected from the supply tubing **18** and is connected to the return tubing **20** through the passage **50***a*, the annular channel **82***a* in the valve **56***a*, and the passage **48***a*. Thus, the working stroke of the piston **24***a* is initiated, this taking place before the completion of the working stroke of the piston **24***b* so that again the pumping load is gradually transferred from the piston **24***b* to the piston **24***a* to prevent hydraulic shock.

The second thing that occurs in response to downward movement of the valve **56***a* to its lower position is that it connects the lower end of the valve **56***b* to the supply tubing **18** through the passage **44**, the passage **70***a*, the bore **64***a* above the plunger **58***a*, the passage **84***a*, the restrictor **86***a*, and the passage **74***a*. Consequently, the valve **56***b* is moved upwardly into its upper position to disconnect the upper surface of the plunger **32***b* of the piston **24***b* from the return tubing **20** and to connect it to the supply tubing **18** through the passage **44**, the passage **52***b*, the bore **64***b* below the plunger **58***b*, and the passage **48***b*. This produces the downward or return stroke of the piston **24***b*, the latter moving downwardly at a sufficiently high speed that it reaches the end of its return stroke before the piston **24***a* reaches the end of its working stroke, as hereinbefore discussed in connection with the piston **24***a*. The desired, relatively high return speed for the piston **24***b* is achieved by making the area of the upper surface of the plunger **32***b* sufficiently larger than the area of the lower surface of the plunger **26***b*.

As previously described, the valve **56***b* is locked in its lower position when the valve **56***a* is in its upper position by applying the exhaust pressure to the lower end of the valve **56***b* through the passage **50***a*, the annular channel **82***a*, the passage **84***a*, the restrictor **86***a* and the passage **74***a*. The valve **56***a* is locked in its lower position when the valve **56***b* is in its upper position by applying exhaust pressure to the lower end of the valve **56***a* through the passage **50***b*, the annular channel **82***b*, the passage **84***b*, the restrictor **86***b* and the passage **74***b*. The valve **56***b* is locked in its upper position when the valve **56***a* is in its lower position by applying the operating fluid pressure to the lower end of the valve **56***b* through the passage **44**, the passage **70***a*, the upper end of the bore **64***a*, the passage **84***a*, the restrictor **86***a* and the passage **74***a*. The valve **56***a* is locked in its upper position when the valve **56***b* is in its lower position by applying the operating fluid pressure to the lower end of the valve **56***a* through the passage **44**, the passage **70***b*, the upper end of the bore **64***b*, the passage **84***b*, the restrictor **86***b*, and the passage **74***b*.

From the foregoing description of the operation of the pump **10**, it will be apparent that as each of the pistons **24***a* and **24***b* approaches the end of the working stroke, it initiates the working stroke of the other to effect a gradual transfer of the pumping load from one piston to the other. Subsequently, the first piston completes its working stroke and thereafter executes its return stroke before the second piston reaches the end of its working stroke. Thus, the second of the pump units **14***a* and **14***b* always starts up before the first completes its working stroke, the resulting gradual transfer of the pumping load resulting in continuous flow of the pumped fluid so that no hydraulic shock in the production tubing **22** results. Also, since the engine portions or means of both pump units **14***a* and **14***b* are connected in parallel and thus are always receiving operating fluid under pressure and discharging spent operating fluid, the flows of operating fluid under pressure and of spent operating fluid are substantially continuous so that no hydraulic shock results in either the supply tubing **18** or the return tubing **20**.

The amount by which each of the pump units **14***a* and **14***b* leads the other with respect to initiation of the working stroke of its piston **24***a* or **24***b* depends upon the locations of the control ports **80***a* and **80***b* in the rods **30***a* and **30***b*. These ports are so located as to permit the second piston to attain normal full speed before the first piston completes its working stroke. The amount of overlap of the working strokes of the two pistons **24***a* and **24***b* is also controlled partly by the restrictors **86***a* and **86***b* since these restrictors control the rates of upward movement of the valves **56***a* and **56***b* and thus control the times at which the working strokes of the pistons **24***a* and **24***b* are completed and the return strokes thereof initiated. Thus, the phasing of the working strokes of the two pistons **24***a* and **24***b* is determined by both the locations of the control ports **80***a* and **80***b* and the flow resistances of the restrictors **86***a* and **86***b*.

Since the pumping portions or pump means of the two pump units **14***a* and **14***b* are connected in series to produce a substantially continuous flow of pumped fluid, it will be apparent that the pumping capacity is greatly increased for the same over-all pump diameter, which is an important feature. Conversely, the over-all pump diameter may be greatly reduced to provide the same pump capacity as conventional pumps of larger diameter. This is sometimes of great economic advantage, as the use of a pump of a smaller over-all diameter permits the use of a smaller tubing for the pump and thus permits the use of a smaller well casing and a smaller well bore, thereby reducing drilling costs, casing costs and tubing costs. The net result is a very substantial saving in capital costs.

Triplex pump 110

The triplex pump of the invention which is illustrated in Fig. 2 of the drawings is designated generally by the numeral **110** and includes three pumping devices **112***a*, **112***b* and **112***c*, respectively, including axially aligned pump units **114***a*, **114***b* and **114***c* having pump portions or means connected in series and engine portions or means connected in parallel, as will become apparent. The pump units **114***a*, **114***b* and **114***c* are regulated by a control means **116** which, in the embodiment presently under consideration, comprises axially aligned valve units **116***a*, **116***b* and **116***c*, respectively, controlling the pump units **114***a*, **114***b* and **114***c*, the valve units being spaced from and located alongside the pump units.

The triplex pump **110** is supplied with operating fluid under pressure through a supply tubing **118** and, in the particular construction illustrated, which is a closed system, the spent operating fluid is returned to the surface through a return tubing **120**. The fluid discharged by the pump units **114***a*, **114***b* and **114***c* is pumped to the surface through a production tubing **122**.

The pump units **114***a*, **114***b* and **114***c* are all identical and are substantially identical to the pump units **14***a* and **14***b* of the duplex pump **10**, differing therefrom solely in substituting two control ports, which will be described hereinafter, for the single control ports **80***a* and **80***b* of the pump units **14***a* and **14***b*. Consequently, those components of the pump units **114***a* and **114***b* which are identical to corresponding components of the pump units **14***a* and **14***b* will be designated by reference characters differing from the reference characters utilized in connection with the pump units **14***a* and **14***b* by the addition of the number "100" thereto. Also, since the pump unit **114***c* is identical to the pump units **114***a* and **114***b*, the corresponding components thereof will be designated by reference characters differing from the reference characters associated with the pump units **114***a* and **114***b* by the substitution of the suffix "*c*" for the suffix "*a*" or the suffix "*b*." The inlet corresponding to the inlet **40**

of the duplex pump 10 will be designated by the reference character 140.

As in the case of the pistons 24a and 24b of the pump units 14a and 14b, the lower surfaces of the plungers 126a, 126b and 126c of the pistons 124a, 124b and 124c are constantly exposed to the operating fluid pressure in the supply tubing 118 through a main passage 144 and branch passages 146a, 146b and 146c. The upper surfaces of the plungers 132a, 132b and 132c are alternately exposed to the operating fluid pressure in the supply tubing 118 and the spent operating fluid pressure or exhaust pressure in the return tubing 120 through passages 150a, 150b and 150c controlled by the valve units 116a, 116b and 116c. The valve units 116a, 116b and 116c respectively include engine valves 152a, 152b and 152c which are movable between upper and lower positions, these valves connecting the passages 150a, 150b and 150c to passages 154a, 154b and 154c leading to the return tubing 120 when in their upper positions, and connecting the passages 150a, 150b and 150c to branch passages 156a, 156b and 156c of a main passage 158 leading to the supply tubing 118 when in their lower positions. Thus, the valves 152a, 152b and 152c are in their upper positions to produce the upward or working strokes of the pistons 124a, 124b and 124c and are in their lower positions to produce the downward or return strokes of the pistons. Thus, the relation of valve position to piston stroke in the pump 110 is the reverse of that in the pump 10.

The valves 152a 152b and 152c are similar in construction and operation to those disclosed in my aforementioned copending application Serial No. 598,785 so that an extremely detailed description thereof herein will not be necessary.

The valves 152a, 152b and 152c are differential area valves each having a three-speed action during its upward movement, at which time it is controlling the acceleration of the corresponding piston 124a, 124b, or 124c at the beginning of its upward or working stroke, and each having a two-speed action during its downward movement, at which time it controls the initiation of the downward or return stroke of the corresponding piston. The valves 152a, 152b and 152c include relatively smaller plungers 160a, 160b and 160c and relatively larger plungers 162a, 162b and 162c, the smaller plungers being reciprocable in bores 164a, 164b and 164c, and the larger plungers being reciprocable in bores 166a, 166b and 166c and being connected to the smaller plungers by reduced-diameter rods or stems 168a, 168b and 168c. When the valves 152a, 152b and 152c are in their upper positions, the passages 150a, 150b and 150c are connected to the passages 154a, 154b and 154c through the annular spaces around the rods 168a, 168b and 168c to produce the working strokes of the pistons 124a, 124b and 124c. When the valves are in their lower positions, the passages 150a, 150b and 150c are connected to the passages 156a, 156b and 156c through the bores 164a, 164b and 164c above the plungers 160a, 160b and 160c to produce the return strokes of the pistons 124a, 124b and 124c.

The plungers 162a, 162b and 162c of the valves 152a, 152b and 152c are provided adjacent the upper ends thereof with external annular channels 170a, 170b and 170c, respectively. These channels communicate with branch passages 172a, 172b and 172c of the main passage 158 when the valves 152a, 152b and 152c are in their upper positions and as they move upwardly from their lower positions to their upper positions, the main passage 158 leading to the supply tubing 118, as hereinbefore described, and thus supplying operating fluid under pressure to the channels 170a, 170b and 170c under the conditions specified. The lower ends of the channels 170a, 170b and 170c communicate with external helical speed control grooves or threads 174a, 174b and 174c in the plungers 162a, 162b and 162c, these grooves communicating at their lower ends with external annular grooves 176a, 176b and 176c which, in turn, communicate with axial passages 178a, 178b and 178c in the plungers 162a, 162b and 162c through radial ports 180a, 180b and 180c therein. These axial passages extend to the lower ends of the valves 152a, 152b and 152c. When the valves 152a, 152b and 152c are in their lower positions, the annular grooves 176a, 176b and 176c communicate with branch passages 182a, 182b and 182c of control passages 184a, 184b and 184c which extend from the pump-unit bores 136a, 136b and 136c to the lower ends of the valve-unit bores 164a, 164b and 164c. As will be apparent, the branch and control passages just mentioned always communicate with the return tubing 120 through the annular spaces around the valve rods 168a, 168b and 168c and the passages 154a, 154b and 154c so that the exhaust pressure is always present in these passages.

The plunger 162a, 162b and 162c of the valves 152a, 152b and 152c are provided below the grooves 176a, 176b and 176c with external annular grooves 186a, 186b and 186c which communicate with the axial passages 178a, 178b and 178c through radial ports 188a, 188b and 188c, and which register with branch passages 190a, 190b and 190c of the main passage 158 in intermediate positions of the valves 152a, 152b and 152c, the main passage 158, as hereinbefore stated, communicating with the supply tubing 118 so that it always contains operating fluid under pressure.

Below the annular grooves 186a, 186b and 186c the plungers 162a, 162b and 162c are provided with relatively wide, external annular channels 192a, 192b and 192c which communicate with the axial passages 178a, 178b and 178c through radial ports 194a 194b and 194c, and which register with control passages 196a, 196b and 196c when the valves 152a, 152b and 152c are in their lower positions and when they are in intermediate positions. The control passage 196a leads to the bore 136b of the pump unit 114b to provide for control of the valve 152a by the piston 124b. Similarly, the control passage 196b leads to the bore 136c of the pump unit 114c so that the piston 124c exercises control over the valve 152b. The control passage 196c leads to the bore 136a of the pump unit 114a, which means that the piston 124a exercises control over the valve 152c. Thus, the uppermost pump unit 114a exercises control over the lowermost valve unit 116c while the intermediate and lowermost pump units 114b and 114c exercise control over the uppermost and intermediate valve units 116a and 116b to provide the desired timed relation or phasing between the pump units, as will be discussed in more detail hereinafter.

Additional control passages 198a, 198b and 198c extend from the bores 136a, 136b and 136c of the pump units 114a, 114b and 114c to the lower ends of the major bores 166a, 166b and 166c of the respective valve units 116a, 116b and 116c. As will be described in detail hereinafter, the control passages 198a, 198b and 198c permit the pump units 114a, 114b and 114c to exercise control over the respective valve units 116a, 116b and 116c. Thus, control over the valve units 116a, 116 b and 116c is exercised not only by pump units other than the pump units 114a, 114b and 114c which the respective valve units 116a, 116b and 116c control, but also by the pump units 114a, 114b and 114c which the respective valve units 116a, 116b and 116c control.

The rods 130a, 130b and 130c of the pistons 124a, 124b and 124c are provided with control ports 200a, 200b and 200c, respectively, and control ports 202a, 202b and 202c, respectively. As the pistons 124a, 124b and 124c approach the ends of their working strokes, but before they complete their working strokes, the control ports 200a, 200b and 200c connect the control passages 196c, 196a and 196b, respectively, to the cylinders 128a, 128b and 128c, respectively. These cylinders always contain operating fluid under pressure because of their connec-

tions to the supply tubing 118 through the branch passages 146a, 146b and 146c, respectively, and the main passage 144. At the same time as or shortly after the control ports 200a, 200b and 200c accomplish the foregoing, but again before the pistons 124a, 124b and 124c reach the ends of their working strokes, the control port 202a bridges the control passages 184a and 198a, the control port 202b bridges the control passages 184b and 198b, and the control port 202c bridges the control passages 184c and 198c. As will become apparent, the control ports 200a, 200b and 200c thus cause the pump units 114a, 114b and 114c to exercise control over certain of the valve units 116a, 116b and 116c other than the valve units by which the respective pump units are controlled, while the control ports 202a, 202b and 202c cause the pump units 114a, 114b and 114c to exercise control over the respective valve units 116a, 116b and 116c by which they are controlled.

Operation of pump 110

For convenience in considering the operation of the triplex pump 110, the detailed operation of the valves 152a, 152b and 152c will be considered first. Thereafter, the over-all operation of the pump 110, including the interrelations between the pump units 114a, 114b and 114c and the valve units 116a, 116b and 116c, will be considered. Since the valves 152a, 152b and 152c are identical, they all operate in exactly the same way and only the operation of the valve 152a will be discussed in detail.

As will be apparent, the upper end of the valve 152a is always exposed to the operating fluid pressure in the supply tubing 118 through the main passage 158 and the branch passage 156a. Similarly, the annular areas at the lower end of the plunger 160a and the upper end of the plunger 162a are always exposed to the exhaust pressure in the return tubing 120 through the passage 154a. Thus, the operating fluid pressure constantly acts downwardly on the area of the plunger 160a and the exhaust pressure constantly acts downwardly on the difference between the areas of the plungers 160a and 162a. Therefore, when the exhaust pressure is applied to the lower end of the valve 152a, this valve moves downwardly, the downward force producing this movement being the product of the area of the plunger 160a and the difference between the operating fluid and exhaust pressures. Conversely, the valve 152a moves upwardly when the lower end thereof is exposed to the operating fluid pressure, the net upward force being the result of the application of the difference between the operating fluid and exhaust pressures to the difference between the areas of the plungers 162a and 160a.

Considering first upward movement of the valve member 152a from its lower position, such movement is initiated by the application of the operating fluid pressure to the lower end thereof through the control passage 196a, the annular channel 192a, the radial port 194a and the axial passage 178a. All of these passages offer relatively little restriction to operating fluid flow so that the valve 152a moves upwardly at a relatively high first speed. Eventually, the annular channel 192a moves upwardly out of register with the control passage 196a, which marks the end of upward movement of the valve 152a at its first speed. At this point, the channel 170a registers with the branch passage 172a of the main passage 158 leading to the supply tubing 118, whereupon the pressure of the operating fluid in the supply tubing is applied to the lower end of the valve 152a through the main passage 158, the branch passage 172a, the channel 170a, the helical groove or thread 174a, the annular groove 176a, the radial port 180a and the axial passage 178a. Under these condition, the valve 152a continues its upward movement at a relatively slow second speed, determined by the flow capacity of the helical thread 174a. As the valve 152a moves upwardly at its relatively slow second speed, the plunger 160a begins to uncover the passage 150a to connect the cylinder 134a above the plunger 132a to the return tubing 120 through the annular space around the rod 168a and the passage 154a. This connection of the cylinder 134a above the plunger 132a to the return tubing 120 is thus initiated slowly so as to initiate the working stroke of the piston 124a slowly to avoid hydraulic shock and to prevent racing of the piston 124a in the event that it is not acting on solid fluid.

After the plunger 160a has begun to uncover the passage 150a to initiate the working stroke of the piston 124a in the foregoing manner, the annular groove 186a registers with the branch passage 190a leading to the supply tubing 118 through the main passage 158, whereupon operating fluid under pressure from the supply tubing may flow through the main passage 158, the branch passage 190a, the annular groove 186a, the radial port 188a and the axial passage 178a to the lower end of the valve 152a. The port 188a offers materially less resistance to flow than the helical thread 174a so that the valve 152a now moves upwardly at a relatively high third speed. Normally, the size of the radial port 188a is such that it will supply sufficient operating fluid to cause an upward speed of the valve 152a approximately three times the speed attainable with the helical thread 174a alone. Thus, the relatively high third speed of the valve 152a is approximately four times the relatively slow second speed thereof. The relatively high third speed of the valve 152a permits the establishment of full communication between the return tubing 120 and cylinder 134a above the plunger 132a at a relatively high rate, after initial communication has been established slowly in the manner hereinbefore described, to permit the piston 124a to achieve its normal working-stroke speed as quickly as possible, such slow establishment of initial communication preventing hydraulic shock as the piston 124a starts its working stroke.

As the valve 152a approaches its upper position, the annular groove 186a moves upwardly out of register with the passage 190a so that the final upward movement of the valve is again controlled by the helical thread 174a. Consequently, this helical thread also acts to lock the valve 152a in its upper position since, under the condition existing at this stage, the lower end of the valve 152a is exposed to the operating fluid pressure in the supply tubing 118 through the passage 158, the passage 172a, the annular channel 170a, the helical thread 174a, the annular groove 176a, the radial port 180a and the axial passage 178a.

Considering now the downward movement of the valve 152a, as soon as the exhaust pressure is applied to the lower end of this valve through the passage 198a, the valve starts downwardly at a relatively high first speed. The passage 198a communicates with the bore 166a somewhat above the lower end thereof so that the plunger 162a the the valve 152a eventually covers the passage 198a, the plunger 162a beginning to cover the passage 198a when the plunger 160a begins to uncover the passage 150a. Thereafter, the valve 152a moves downwardly at a relatively slow second speed to establish communication between the supply tubing 118 and the cylinder 134a above the plunger 132a slowly, thereby initiating the return stroke of the piston 124a slowly to eliminate hydraulic shock. The relatively slow second downward speed of the valve 152a is produced by the radial port 180a, the lower end of the valve 152a being connected to the return tubing 120 by way of the passage 154a, the annular space around the rod 168a, a portion of the control passage 184a, the branch passage 182a, the annular groove 176a, the port 180a, and the axial passage 178a. Preferably, the radial port 180a, determining the relatively slow second downward speed of the valve 152a is approximately the same size as the radial port 188a controlling the third upward speed of the valve. Once the valve 152a reaches its lower position, the lower end

thereof continues to be exposed to the exhaust pressure in the return tubing 120 through the passage 154a, the annular space around the rod 168a, a portion of the control passage 184a, the branch passage 182a, the annular groove 176a, the radial port 180a and the axial passage 178a. Thus, the port 180a acts as a hydraulic lock for the valve 152a when it is in its lower position.

As hereinbefore stated, the valves 152b and 152c operate in exactly the same way as the valve 152a in moving from their lower positions to their upper positions and from their upper positions to their lower positions. Consequently, no description of the operation of the valves 152b and 152c is necessary.

Consider now the over-all operation of the triplex pump 110, the valves 152a and 152c are shown in their upper positions and the valve 152b is shown in its lower position so that the pistons 124a and 124c are on their working strokes, as indicated by the arrows 204a and 204c, and the piston 124b is on its return stroke, as indicated by the arrow 204b, or has reached the end of its return stroke. The piston 124c is leading the piston 124a by approximately 120°, the piston 124a is leading the piston 124b by approximately 120°, and the piston 124b is leading the piston 124c by approximately 120°, considering the working and return strokes of one of the pistons as constituting a 360° operating cycle. This timing or phasing of the movements of the pistons 124a, 124b and 124c is achieved in a manner that will become apparent in the light of the description which follows.

Considering the piston 124c, it will be noted that it is approaching the end of its working stroke and that the control port 200c is about to connect the cylinder 128c to the control passage 196b. When this occurs, operating fluid under pressure flows from the supply tubing 118 through the main passage 144, the branch passage 146c, the cylinder 128c, the control port 200c, the control passage 196b, the annular channel 192b in the valve 152b, the radial port 194b in this valve, and the axial passage 178b therein into the bore 166b below the valve 152b. The operating fluid pressure is thus applied to the lower end of the valve 152b to move it upwardly in the manner hereinbefore described, whereupon the valve 152b disconnects the upper surface of the plunger 132b from the supply tubing 118 and connects it to the return tubing 120, through the passage 150b, the annular space around the rod 168b and the passage 154b, to initiate the working stroke of the piston 124b. Thus, the piston 124c, as it approaches the end of its working stroke, moves the valve 152b from its lower position to its upper position to initiate the working stroke of the piston 124b, thereby effecting a gradual pumping load transfer from the piston 124c to the piston 124b. Thus, the piston 124c exercises control over a valve, viz., the valve 152b, other than the valve, viz., the valve 152c, by which the piston 124c itself is controlled.

The piston 124c also exercises control over the valve, viz., the valve 152c, by which it is controlled. More specifically, at about the time the control port 200c connects the control passage 196b to the cylinder 128c, or immediately thereafter, the control port 202c bridges the control passage 184c and 198c. When this occurs, the exhaust pressure is communicated to the lower end of the valve 152c through the passage 154c, the annular space around the rod 168c, the passage 184c, the control port 202c and the passage 198c. The application of exhaust pressure to the lower end of the valve 152c results in downward movement of this valve into its lower position, as hereinbefore described. Downward movement of the valve 152c into its lower position terminates communication between the upper surface of the plunger 132c and the return tubing 120 and establishes communication between this surface and the supply tubing 118 through the passage 158, the passage 156c, the bore 164c above the valve plunger 160c, and the passage 150c. Consequently, the return stroke of the piston 124c is initiated, this occurring after initiation of the working stroke of the piston 124b to insure the hereinbefore-described pumping load transfer from the piston 124c to the piston 124b.

Similarly, when the piston 124a has substantially reached the end of its working stroke, the control port 200a acts through the control passage 196c to move the valve 152c upwardly to initiate the working stroke of the piston 124c, and the control port 202a acts through the control passage 198a to move the valve 152a downwardly into its lower position to initiate the return stroke of the piston 124a. Thus, a gradual pumping load transfer from the piston 124a to the piston 124c occurs. Again in a similar manner, as the piston 124b approaches the end of its working stroke, the control port 200b acts through the control passage 196a to move the valve 152a upwardly to initiate the working stroke of the piston 124a, and the control port 202b acts through the control passage 198b to cause the valve 152b to move downwardly to initiate the return stroke of the piston 124b, thereby gradually transferring the pumping load from the piston 124b to the piston 124a. The operating cycle is completed when, as hereinbefore described in detail, the piston 124c moves the valve 152b upwardly to initiate the working stroke of the piston 124b and moves the valve 152c downwardly to initiate its own return stroke. The piston 124a controls the valves 152c and 152a and the piston 124b controls the valves 152a and 152b in exactly the same manner as the piston 124c controls the valves 152b and 152c so that a detailed description is unnecessary.

As will be apparent from the foregoing, the phasing or timing of the working and return strokes of the pistons 124a, 124b and 124c is such that two of these pistons are on their working strokes at all times, since the pumping load carried by each of the pistons as it approaches the end of its working stroke is transferred to a piston which has completed its return stroke.

As two of the pistons 124a, 124b and 124c move upwardly through their working strokes in phased relation, with one leading the other by approximately 120°, the third piston is moved downwardly through its return stroke, or has completed its return stroke and is standing at the lower end of its travel waiting for the next piston in the series to approach the end of its working stroke. In order to achieve this, it will be apparent that the speed of each piston during its return stroke must be at least twice its speed during its working stroke. This is accomplished readily by making the area of the upper surface of each plunger 132a, 132b and 132c sufficiently larger than the area of the lower surface of the corresponding one of the plungers 126a, 126b and 126c. Since only mechanical friction and fluid friction through the axial passages 138a, 138b and 138c must be overcome during the return strokes of the pistons 124a, 124b and 124c, the desired high return stroke speed may be achieved by making the diameter of the cylinders 134a, 134b and 134c a few thousandths of an inch larger than the diameter of the cylinders 128a, 128b and 128c.

Considering some of the advantages of the triplex pump 110, since the pumping load is always transferred from the piston 124a, 124b or 124c which is nearing the end of its working stroke to the piston which is standing at the lower end of its travel before the first piston mentioned reaches the end of its working stroke, there is substantially no tendency to induce hydraulic shock in the pumped fluid. Also, since the engine portions or means of the pump units 114a, 114b and 114c are connected in parallel, the flows of operating fluid under pressure from the supply tubing 118 and of spent operating fluid to the return tubing 120 are substantially constant to avoid inducing hydraulic shock in either the operating fluid column or the spent operating fluid column.

Since the pumping load is always carried by two of the pistons 124a, 124b and 124c, and since these pistons

are preferably identical, it will be apparent that the pumping load is always divided equally between two of the pistons. Consequently, the operating fluid pressure required for a given pumping capacity is only half that which would required if only one of the pistons were on its working stroke at any instant. Conversely, if the same operating fluid pressure is used, the engine areas, i.e., the areas of the upper surfaces of the plungers **132***a*, **132***b* and **132***c* and the areas of the lower surfaces of the plungers **126***a*, **126***b* and **126***c* may be reduced by a factor of two for the same pumping capacity.

Reducing the operating fluid pressure by a factor of two has the advantage of reducing the strength requirements of the entire structure of the pumping system, while reducing the engine areas by a factor of two with the same operating fluid pressure has the advantage of a reduction in pump size without any reduction in pumping capacity. Such a reduction in pump size is, as hereinbefore pointed out, occasionally of great economic advantage, since it permits the use of a smaller pump tubing, a smaller well casing and a smaller well bore, thereby reducing tubing, casing and drilling costs.

On the other hand, if maximum operating fluid pressure and maximum pump size are utilized with the triplex pump **110** of the invention, very large increases in pumping capacity, as compared to conventional pumps, may be realized. For example, a triplex pump **110** of the invention operating at the same pressure as and of the same over-all diameter as a comparable prior art pump will have a pumping capacity of nearly five times the pumping capacity of the prior art pump.

The triplex pump **110** also has the advantages hereinbefore ascribed to the duplex pump **10**. For example, the axially aligned flow passages **138***a*, **138***b* and **138***c* for the pumped fluid greatly reduce fluid friction. Also, since two of the pistons **124***a*, **124***b* and **124***c* are always on their working strokes, while the third is on its return stroke, or standing at the lower end of its travel, no standing valve or valves are required, the working valves **142***a*, **142***b* and **142***c* carried by the respective pistons making a standing valve or valves unnecessary. The straight axial flow through the pistons **124***a*, **124***b* **124***c* and the absence of standing valves contribute materially to the increased capacity of the pump **110** by minimizing resistance to flow of the pumped fluid.

While the invention has thus far been considered as applied to duplex and triplex pumping, it will be apparent to those skilled in the art that the invention may very readily be extended to quadruplex pumping, quintuplex operation, and so forth. This may be accomplished very readily by inserting additional pumping devices, similar to the pumping devices **112***a*, **112***b* and **112***c*, between any two of the pumping devices **112***a*, **112***b* and **112***c*. For example, the triplex pump **110** may be converted to a quadruplex pump by inserting a pumping device similar to the pumping devices **112***a*, **112***b* and **112***c* between the pumping devices **112***a* and **112***b*, this additional pumping device being related to the pumping device **112***b* in the same manner as the pumping device **112***a* is related to the pumping device **112***b*, and being related to the pumping device **112***a* in the same manner as the pumping device **112***b* is related to the pumping device **112***a*. A quintuplex pump may be produced by inserting two such pumping devices in the manner indicated. In any such multiplex pump, all but one of the pistons would be on their working strokes at all times. Thus, assuming a multiplex pump having n pumping devices, the pumping load is always carried by $n-1$ pistons. Therefore, either the operating fluid pressure, or the engine areas, may be reduced by a factor of $n-1$, or, alternatively, by maintaining the operating fluid pressure and the engine areas at maximum values, the pumping capacity may be correspondingly increased. As will be apparent, by increasing the value of n, the operating fluid pressure, or the engine areas, may be reduced to any extent desired, or the pumping capacity may be increased as desired.

Triplex pump 210

Referring now to Figs. 3 to 8 of the drawings, and particularly to Figs. 3 and 4 thereof for the time being, illustrated therein is another triplex pump of the invention which is designated generally by the numeral **210** and which includes pumping devices **212***a*, **212***b* and **212***c*. These pumping devices include pump units **214***a*, **214***b* and **214***c* the operation of which is regulated by a control means **216** which consists of a single valve unit common to the three pump units.

The pump **210** is supplied with operating fluid under pressure through a supply tubing **218** and spent operating fluid is returned to the surface through a return tubing **220**. Pumped fluid enters the pump **210** through an inlet **240** and is conveyed to the surface through a production tubing **222**.

The pump units **214***a*, **214***b* and **214***c* are substantially identical to the pump units **114***a*, **114***b* and **114***c*, the only difference being that control ports to be described are substituted for the control ports **200***a*, **200***b*, **200***c*, **202***a*, **202***b* and **202***c*. Consequently, components of the pump units **214***a*, **214***b* and **214***c* which are identical to corresponding components of the pump units **114***a*, **114***b* and **114***c* are identified by reference characters higher by "100" than the reference characters associated with the pump units **114***a*, **114***b* and **114***c*. Thus, detailed descriptions of the pump units **214***a*, **214***b* and **214***c* will be unnecessary.

As in the previous embodiments, the lower surfaces of the plungers **226***a*, **226***b* and **226***c* are constantly exposed to the operating fluid pressure through a main passage **244** and branch passages **246***a*, **246***b* and **246***c*. The upper surfaces of the plungers **232***a*, **232***b* and **232***c* are alternately exposed to the exhaust pressure in the return tubing **220** and the operating fluid pressure in the supply tubing **218** through passages **248***a*, **248***b* and **248***c* controlled by the valve unit **216**, exposure of the upper surfaces of the plungers **232***a*, **232***b* and **232***c* to the exhaust pressure resulting in the working strokes of the pistons **224***a*, **224***b* and **224***c*, and exposure of these surfaces to the operating fluid pressure resulting in the return strokes of the pistons.

Considering the valve unit **216** in more detail, it is located alongside the pump units **214***a*, **214***b* and **214***c* and includes a tubular rotary valve **250** rotatable about an axis parallel to the axis of the pump units. The valve **250** is rotatable in axially aligned bores **252***a*, **252***b* and **252***c* with which the passages **248***a*, **248***b* and **248***c* communicate respectively, the valve being in fluid-tight engagement with the walls of such bores.

The valve **250** is provided with an axial passage **254** therethrough which communicates at its upper end with the supply tubing **218** so that operating fluid under pressure is always present in such passage. The valve **250** is provided with radial pressure ports **256***a*, **256***b* and **256***c*, as shown in Figs. 5, 6 and 7, respectively, which are adapted to register with the passages **248***a*, **248***b* and **248***c*, respectively, to produce the return strokes of the pistons **224***a*, **224***b* and **224***c*, respectively, by communicating the operating fluid pressure to the upper surfaces of the plungers **232***a*, **232***b* and **232***c*, respectively. The ports **256***a*, **256***b* and **256***c* are spaced apart circumferentially of the valve **250** by 120° so that, as the valve is rotated through three operating positions in sequence by actuating means **260** to be described, the pistons **224***a*, **224***b* and **224***c* are caused to execute their return strokes in sequence with phase angles of 120° between the return strokes.

The valve **250** is surrounded by an annular space **262** which is divided into two axially spaced parts at the bore **252***b*, these axially spaced parts being connected by a passage **264**. The annular space **262** communicates with

the return tubing 220 through a passage 266. Thus, spent operating fluid at the exhaust pressure in the return tubing 220 is always present in both parts of the annular space 262.

The valve 250 is provided therein with external exhaust recesses or ports 268a, 268b and 268c which communicate with the annular space 262 and which are adapted to communicate with the passages 248a, 248b and 248c, respectively, as the valve 250 is rotated by the actuating means 260, thereby applying the exhaust pressure to the upper surfaces of the plungers 232a, 232b and 232c to effect the working strokes of the pistons 224a, 224b and 224c. The ports 268a, 268b and 268c are spaced apart circumferentially by angles of 120° so that they produce the working strokes of the pistons 224a, 224b and 224c in sequence with phase angles of 120° between such working strokes. The ports 268a, 268b and 268c are located diametrically opposite the ports 256a, 256b and 256c, respectively, and have angular extents in excess of 120°, as best shown in Figs. 5, 6 and 7. Consequently, whenever one of the ports 256a, 256b and 256c is in register with the corresponding one of the passages 248a, 248b and 248c to produce the return stroke of the corresponding one of the pistons 224a, 224b and 224c, two of the exhaust ports 268a, 268b and 268c are in register with the corresponding pair of the passages 248a, 248b and 248c to produce the working strokes of the corresponding pair of the pistons 224a, 224b and 224c. Thus, two of the pistons are on their working strokes at all times, while the third is on its return stroke, or is standing at the lower end of its travel. The necessary high return speeds for the pistons 224a, 224b and 224c are attained in the same manner as hereinbefore described in connection with the triplex pump 110. The leading edges of the exhaust ports 268a, 268b and 268c are so positioned circumferentially relative to the pressure ports 256a, 256b and 256c that, as the valve 250 rotates from one of its three positions to the next, one of the exhaust ports 268a, 268b and 268c registers with the corresponding one of the passages 248a, 248b and 248c before one of the pressure ports 256a, 256b and 256c registers with the corresponding one of the passages 248a, 248b and 248c. This starts one of the pistons 224a, 224b and 224c on its working stroke before another of the pistons is started on its return stroke, thereby providing a gradual load transfer to minimize hydraulic shock.

Considering the actuating means 260, it includes a piston 270 reciprocable axially of the rotary valve 250 in a cylinder 272, the piston having a stem 274 which makes a sliding fit in the lower end of the passage 254 through the valve. The piston 270 and its stem 274 are provided with an axial passage 276 therethrough which communicates at its upper end with the passage 254 and at its lower end with the cylinder 272, there being a restrictor 278 between the ends of the passage 276. The cylinder 272 below the piston 270 communicates with a main passage 280 having branch passages 282a, 282b and 282c respectively communicating with the bores 236a, 236b and 236c of the pump units 214a, 214b and 214c. The passages 282a, 282b and 282c are adapted to be connected to the cylinders 234a, 234b and 234c, respectively, by control ports 284a, 284b and 284c, respectively, in the rods 230a, 230b and 230c, respectively, of the pistons 224a, 224b and 224c, respectively. These control ports regulate the pressure applied to the lower end of the piston 270, in a manner which will be described hereinafter. The upper end of the stem 274 is always exposed to the operating fluid pressure in the passage 254, and an area of the upper surface of the piston 270 equal to the difference between the cross sectional area of the piston and the external cross sectional area of the stem 274 is constantly exposed to the exhaust pressure through a passage 286 in the valve 250, the passage 286 communicating with the annular space 262 through the exhaust port 268c. Thus, as will be discussed in more detail hereinafter, the piston 270 is a differential area device, which is caused to move upwardly when the operating fluid pressure is applied to the lower end thereof through the passage 276, and which is caused to move downwardly when the exhaust pressure is applied to the lower end of the piston through the passage 280, the downward movement of the piston under such conditions being the result of the constant application of operating fluid pressure to the upper end of the stem 274.

As best shown in Fig. 8, the piston 270 is provided with an upwardly extending annular flange or skirt 288 provided externally with helical ribs or splines 290 and internally with straight ribs or splines 292. The valve 250 is provided with straight external splines 294 which key the valve to the piston 270 for simultaneous rotation while permitting reciprocation of the piston relative to the valve. The external helical splines 290 on the skirt 288 engage internal helical splines 296 on a ratchet member 298 having three teeth 300 spaced 120° apart and engageable by a spring biased pawl 302 carried by the body, 304, of the valve unit 216. The teeth 300 are so oriented that the pawl 302 permits counterclockwise rotation of the ratchet member 298 as viewed from above, but prevents clockwise rotation thereof. Thus, when the piston 270 moves upwardly, the helical splines 290 and 296 try to rotate the ratchet member 298, but this is prevented by the pawl 302. Therefore, as the piston 270 moves upwardly, it rotates in the counterclockwise direction and causes the valve 250 to rotate with it due to the action of the straight splines 292 and 294.

A second ratchet member 306 is carried by the upper end of the valve 250 and is provided with teeth 308 engageable by a spring biased pawl 310 carried by the valve body 304. The ratchet teeth 308 are also spaced 120° apart and are so oriented that the pawl 310 permits counterclockwise rotation of the valve 250, but prevents clockwise rotation thereof. Therefore, when the piston 270 moves upwardly, it causes the valve 250 to rotate counterclockwise through an angle of 120°, but when the piston 270 moves downwardly, the valve 250 is held against rotation and the ratchet member 298 rotates counterclockwise instead. As may be concluded from the foregoing, the angles of the helical splines 290 and 296 and the stroke of the piston 270 are so related that one stroke of the piston produces 120° of rotation of the valve 250.

Operation of pump 210

As shown in Fig. 3 of the drawings, the piston 224a is adjacent the upper end of its travel and is moving downwardly through its return stroke, as designated by the arrow 312a, the return stroke resulting from the fact that the valve 250 is in a position to connect the pressure port 256a to the passage 248a to apply operating fluid pressure to the upper surface of the plunger 232a. At the same time, the pistons 224b and 224c are moving upwardly through their working strokes, as indicated by the arrows 312b and 312c, this being due to the fact that the exhaust ports 268b and 268c are in communication with the passages 248b and 248c to apply exhaust pressure to the upper surfaces of the plungers 232b and 232c. The piston 224b is leading the piston 224c by 120° due to the fact that the exhaust port 268b was in communication with the passage 248b in the angular position of the valve 250 preceding that shown in Fig. 3 of the drawings, whereas the port 268c was not in communication with the passage 248c in the preceding valve position. As previously explained, the exhaust ports 268a, 268b and 268c register with the corresponding passages 248a, 248b and 248c for two angular positions of the valve 250 spaced 120° apart due to the fact that these exhaust ports are more than 120° in angular extent.

Under the foregoing conditions, after the piston 224a has reached the end of its return stroke and as the piston

224*b* reaches the end of its working stroke, the control port 284*c* of the piston 224*c* connects the cylinder 234*c* to the passage 282*c*. Since exhaust pressure is present in the cylinder 234*c* under these conditions, the exhaust pressure is applied to the lower end of the piston 270 through the passage 282*c* and the passage 280. When this occurs, the piston 270 moves downwardly, Fig. 4, rapidly at a rate determined by the sizes of the passages connecting the cylinder 272 to the return tubing 220 and the difference between the operating fluid and exhaust pressures. As the piston 270 moves downwardly, it rotates the ratchet member 298 through an angle of 120° and the pawl 302 engages the next ratchet tooth 300 carried by this ratchet member. As this occurs, the piston 224*c* continues to move upwardly through its working stroke to cause the control port 284*c* to cut off communication between the cylinder 234*c* and the passage 282*c*. Consequently, the cylinder 272 below the piston 270 is cut off from its source of exhaust pressure and, when this occurs, the pressure in the cylinder 272 below the piston 270 builds up to the operating fluid pressure as operating fluid flows through the restrictor 278, this restrictor being sufficiently small to prevent any pressure buildup during the preceding application of exhaust pressure to the lower end of the piston. The operating fluid pressure now applied to the lower end of the piston 270 through the restrictor 278 causes the piston 270 to move upwardly again and, since the ratchet member 298 is now prevented from rotating by the pawl 302, the valve 250 rotates in the counterclockwise direction, having previously been prevented from rotating in the clockwise direction during downward movement of the piston 270 by the pawl 310 and ratchet member 306.

As previously explained, the foregoing downward and upward movement of the piston 270 rotates the valve 250 through an angle of 120° in the counterclockwise direction to bring the exhaust port 268*a* into register with the passage 248*a* and to bring the pressure port 256*b* into register with the passage 248*b*. The initiates the working stroke of the piston 224*a*, and initiates the return stroke of the piston 224*b*, the piston 224*c* meanwhile continuing on its working stroke. Thus, two of the pistons are always on their working strokes, while the third is on its return stroke, or is standing at the lower end of its travel.

When the control ports 284*a* and 284*b* connect the cylinders 234*a* and 234*b* to the passages 282*a* and 282*b*, the valve 250 is rotated through angles of 120° in the same manner as hereinbefore described in connection with bridging of the cylinder 234*c* and the passage 282*c* by the control port 284*c*. Thus, whenever one of the control ports 284*a*, 284*b* and 284*c* bridges the corresponding one of the cylinders 234*a*, 234*b* and 234*c* and the corresponding of the passages 282*a*, 282*b* and 282*c*, the actuating means 260 is energized to advance the valve 250 through 120° to start one of the pistons 224*a*, 224*b* and 224*c* on its working stroke and to start another thereof on its return stroke.

It will be noted that when the pistons 224*a*, 224*b* and 224*c* move downwardly, the control ports 284*a*, 284*b* and 284*c* respectively bridge the cylinder 234*a* and the passage 282*a*, the cylinder 234*b* and the passage 282*b*, and the cylinder 234*c* and the passage 282*c*. However, when the pistons are moving downwardly, the cylinders 234*a*, 234*b* and 234*c* are open to the supply tubing 218 for the reasons hereinbefore described so that they contain fluid at the operating fluid pressure. Since the operating fluid pressure is normally applied to the lower end of the piston 270 through the restrictor 278 to hold it in its upper position, such application of operating fluid pressure by the control ports 284*a*, 284*b* and 284*c* during the return strokes of the pistons 224*a*, 224*b* and 224*c* has no effect. Thus, the control ports cause rotation of the valve 250 during the working strokes of the pistons only.

While it is thought that the operation of the triplex pump 210 will be clear from the foregoing, Fig. 4 of the drawings shows more clearly the action of one of the control ports, viz., the control port 284*b*, in moving the piston 270 downwardly. In Fig. 4, the pistons 224*a* and 224*b* are moving upwardly, as indicated by the arrows 314*a* and 314*b*, while the piston 224*c* is moving downwardly through its return stroke, as indicated by the arrow 314*c*. The piston 224*b* is in a position to cause the control port 284*b* to bridge the cylinder 234*b* and the passage 282*b*, whereby to apply exhaust pressure to the lower end of the piston 270 through the passage 280 to produce movement of the piston downwardly into its lower position, as shown. As soon as the piston 224*b* moves upwardly to cut off communication between the cylinder 234*b* and the control port 284*b*, the piston 270 will move upwardly due to pressure built up therebeneath through the restrictor 278.

The triplex pump 210 has all of the advantages of the triplex pump 110 so that a detailed discussion thereof is unnecessary, it being sufficient to point out that the pump 210 may readily be converted to a duplex pump, or to a quadruplex or quintuplex pump, or the like. For example, the triplex pump 210 may be converted to a quadruplex pump readily by adding another pump unit identical to the pump units 214*a*, 214*b* and 214*c* and by modifying the structure of the actuating means 260 and the porting of the valve 250 to provide for four operating positions of the valve spaced apart angularly by 90°. Conversion to duplex pumping, quintuplex pumping, or the like, may be effected in similar fashion.

Another important feature of the invention common to the embodiments hereinbefore disclosed is that such embodiments may be assembled from parts presently available, such as the parts disclosed in my aforementioned copending application Serial No. 598,785, irrespective of whether the resulting pump is a duplex pump, a triplex pump, a quadruplex pump, a quintuplex pump, or the like. This interchangeability tends to lower costs appreciably since a few basic components may be utilized for a wide variety of pumps.

Although various exemplary embodiments of the present invention have been disclosed herein for purposes of illustration, it will be understood that numerous changes, modifications and substitutions may be incorporated in such embodiments without departing from the spirit of the invention as defined by the following claims.

I claim:

1. In a fluid operated multiplex well pump, the combination of: at least two axially aligned fluid operated pump units connected in tandem and each adapted to sustain a pumping load; and control means located alongside and spaced from said pump units and operable by said pump units for applying the pumping load to said pump units seriatim, said control means including means for removing the pumping load from each of said pump units after the pumping load has been applied to another of said pump units.

2. In a fluid operated multiplex well pump, the combination of: two or more axially aligned fluid operated pump units connected in tandem and each having a fluid operated engine means and a pump means connected to and actuable by said engine means; and fluid operated control means actuable by said pump units for operating said engine means of said pump units in timed relation, said control means including axially aligned fluid operated valve units corresponding in number to the number of said pump units and respectively controlling said engine means of said pump units, said axially aligned valve units being connected in tandem and being lo-

cated alongside and spaced from said axially aligned pump units.

3. In a fluid operated multiplex well pump, the combination of: two or more axially aligned fluid operated pump units connected in tandem and each having a fluid operated engine means and a pump means connected to and actuable by said engine means; and fluid operated control means actuable by said pump units for operating said engine means of said pump units in timed relation, said control means including a single, rotary, fluid operated valve unit controlling said engine means of all of said pump units and located alongside and spaced from said axially aligned pump units.

4. In a fluid operated multiplex pump, the combination of: two or more fluid operated pump units each having a fluid operated engine means and a pump means connected to and actuable by said engine means; and fluid operated control means actuable by said pump units for operating said engine means of said pump units in timed relation, said control means including a single valve unit controlling said engine means of all of said pump units, said valve unit including a rotary valve and including fluid operated means actuable by said pump units for rotating said valve.

5. In a fluid operated multiplex pump, the combination of: two or more fluid operated pump units each having a double acting fluid operated engine means and a single acting pump means connected to and actuable by said engine means; fluid operated control means actuable by said pump units for operating said engine means of said pump units in timed relation; means connecting said engine means of said pump units in parallel; and means connecting said pump means of said pump units in series.

6. In a fluid operated multiplex well pump, the combination of: at least two axially aligned fluid operated pump units connected in tandem and each having piston means movable through working and return strokes, each of said piston means having therethrough an axial passage, said axial passages being connected in series; a single working valve in each of said passages; and fluid operated control means connected in fluid communication with said pump units and controlled by said piston means thereof for initiating the working stroke of one of said piston means as another of said piston means approaches the end of its working stroke.

7. In a fluid operated multiplex pump, the combination of: at least two fluid operated pump units each having working and return strokes; and valve units equal in number to the number of pump units and respectively controlling said pump units, each of said pump units having means for operating a valve unit other than the valve unit by which it is controlled, and each of said valve units having means for operating another of said valve units.

8. In a fluid operated multiplex pump, the combination of: at least two fluid operated pump units each having piston means movable through working and return strokes and each having a passage therethrough for pumped fluid, said passages being in series; working valves in said passages, respectively, and preventing flow therethrough in one direction; and fluid operated control means connected in fluid communication with said pump units and controlled by said piston means thereof for maintaining at least one of said piston means on its working stroke at all times.

9. In a fluid operated multiplex well pump, the combination of: at least two axially aligned fluid operated pump units connected in tandem and each having piston means movable through working and return strokes, each of said piston means having an axial passage therethrough for pumped fluid, said passages being in series; working valves carried by said piston means, respectively, and preventing flow through said passages, respectively, in one direction; and fluid operated control means connected in fluid communication with said pump units and controlled by said piston means thereof for maintaining at least one of said piston means on its working stroke at all times.

10. In a fluid operated multiplex well pump, the combination of: at least two axially aligned fluid operated pump units connected in tandem and each having piston means movable through working and return strokes, each of said piston means having an axial passage therethrough for pumped fluid, said passages being in series; working valves carried by said piston means, respectively, and preventing flow through said passages, respectively, in one direction; and fluid operated control means located alongside and spaced from said pump units and actuable by said piston means of said pump units for maintaining at least one of said piston means on its working stroke at all times.

11. In a fluid operated multiplex well pump, the combination of: at least two axially aligned fluid operated pump units connected in tandem and each including fluid operated double acting engine means and single acting pump means connected to and operable by said engine means; means connecting said engine means of said pump units in parallel; and means connecting said pump means of said pump units in series.

12. In a fluid operated triplex well pump, the combination of: three axially aligned fluid operated pump units connected in tandem and each including fluid operated double acting engine means and single acting pump means connected to and operable by said engine means; means connecting said engine means of said pump units in parallel; means connecting said pump means of said pump units in series; and fluid operated control means actuable by said pump units for operating said engine means of said pump units in timed, overlapping relation.

13. In a fluid operated triplex well pump, the combination of: three axially aligned fluid operated pump units connected in tandem and each including fluid operated engine means and pump means connected to and operable by said engine means; means connecting said engine means of said pump units in parallel; means connecting said pump means of said pump units in series; and fluid operated control means actuable by said pump units for operating said engine means of said pump units in timed, overlapping relation, said control means including a single valve unit located alongside and spaced from said pump units and controlling said engine means of all of said pump units.

14. In a fluid operated multiplex pump, the combination of: at least two fluid operated pump units each having working and return strokes; and valve units equal in number to the number of pump units and respectively controlling said pump units, each of said valve units including a reciprocable valve, each of said pump units having means for moving the valve of the valve unit by which it is controlled in one direction and for moving the valve of a valve unit other than the valve unit by which it is controlled in the opposite direction.

15. In a fluid operated multiplex pump, the combination of: at least two fluid operated pump units each having working and return strokes; and valve units equal in number to the number of pump units and respectively controlling said pump units, each of said valve units including a reciprocable valve, each of said pump units having means for moving in one direction the valve of a valve unit other than the valve unit by which it is controlled, and each of said valve units having means for moving in the opposite direction the valve of another of said valve units.

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CERTIFICATE OF CORRECTION

Patent No. 2,939,397 June 7, 1960

Clarence J. Coberly

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 3, line 73, after "each" insert -- such --; column 7, line 51, for "bore 54b" read -- bore 64b --; column 14, lines 43 and 44, for "condition" read -- conditions --; column 15, line 14, for "Consider" read -- Considering --; column 17, line 5, after "would" insert -- be --; column 21, line 39, for "The" read -- This --; line 53, before "of the passages" insert -- one --.

Signed and sealed this 8th day of November 1960.

(SEAL)
Attest:

KARL H. AXLINE
Attesting Officer

ROBERT C. WATSON
Commissioner of Patents