July 21, 1936. E. J. SVENSON 2,048,524

PLUNGER PUMP

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Fig. 1

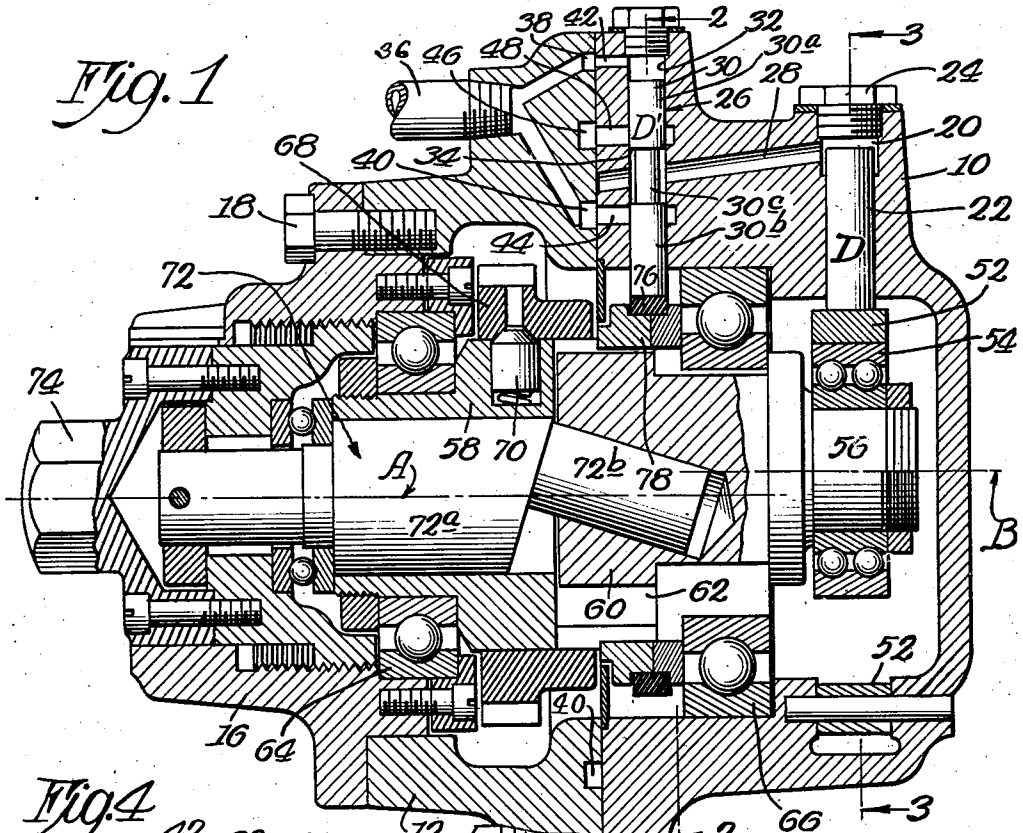
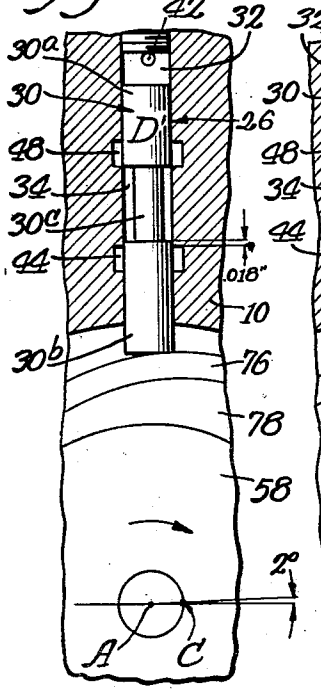


Fig. 4



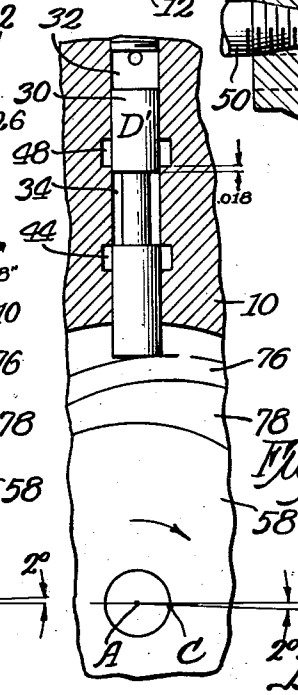
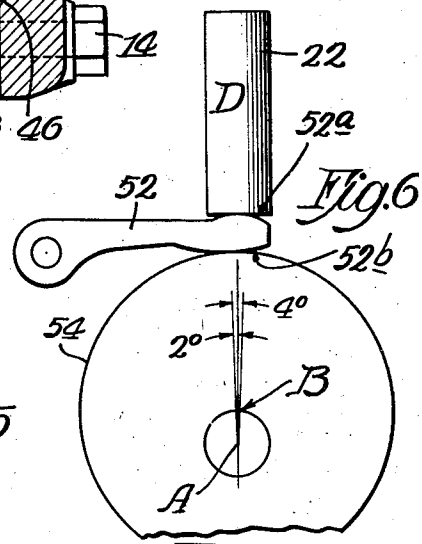


Fig. 5

Fig. 6



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Fig. 2

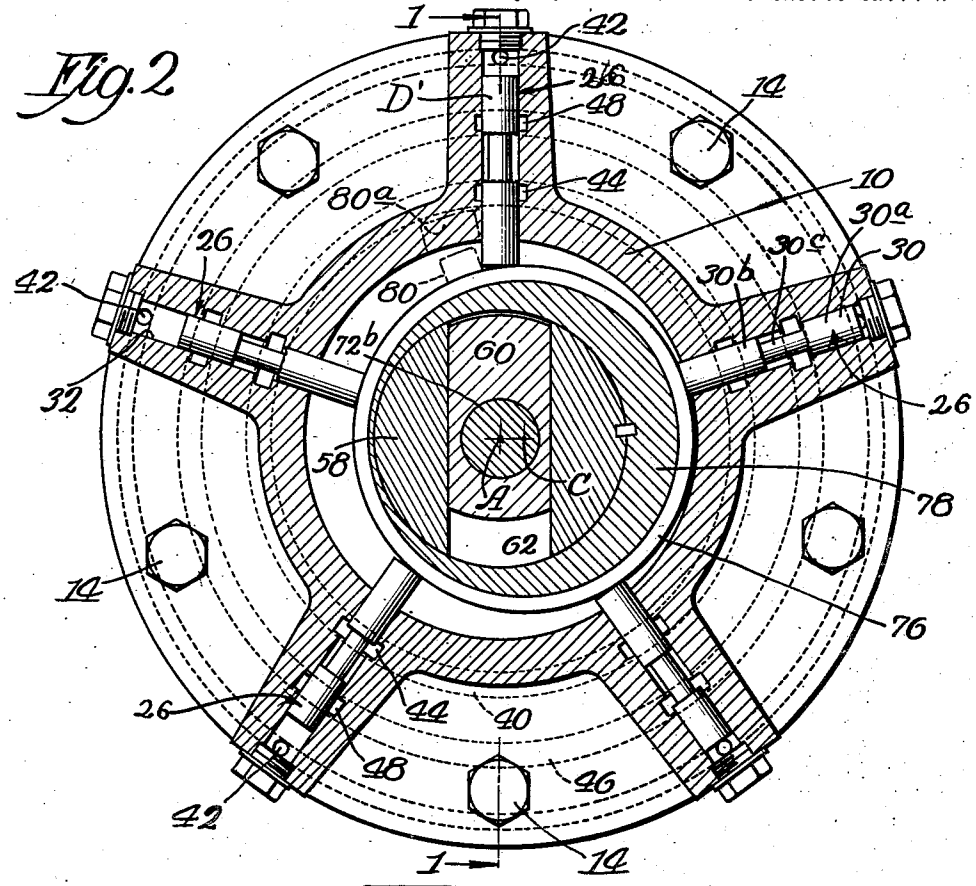
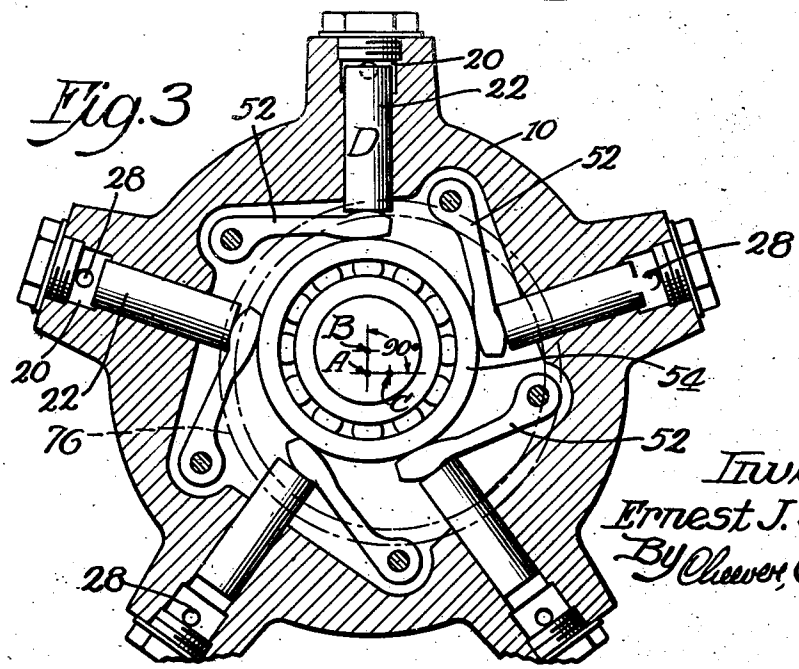


Fig. 3



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PLUNGER PUMP

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FIG. 7.

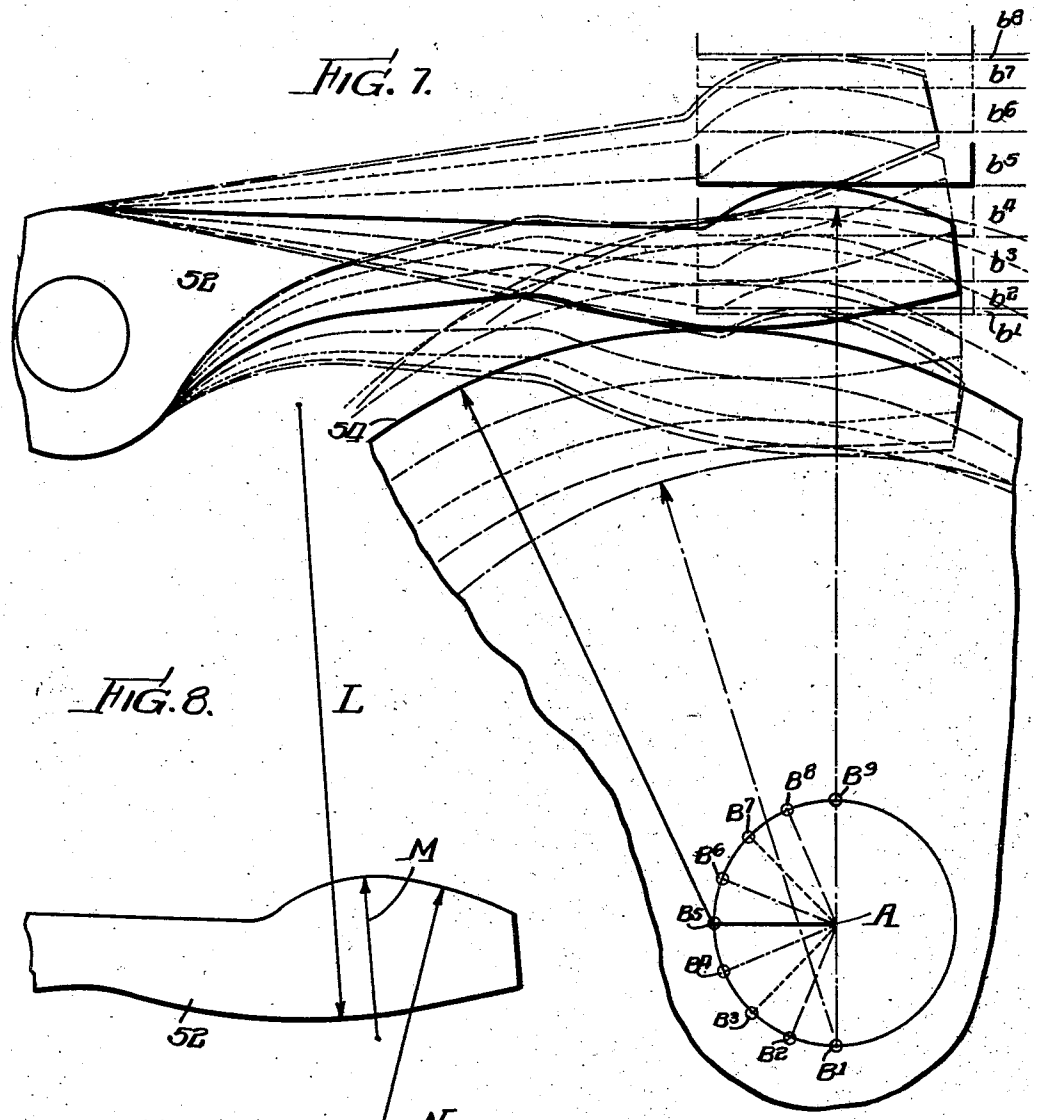
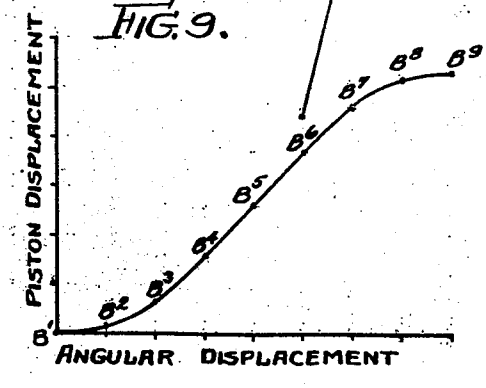


FIG. 8.

FIG. 9.



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UNITED STATES PATENT OFFICE

2,048,524

PLUNGER PUMP

Ernest J. Svenson, Rockford, Ill.

Application July 3, 1931, Serial No. 548,569

11 Claims. (Cl. 103—174)

My invention relates generally to plunger pumps and particularly to plunger pumps adapted to be driven at relatively high speeds.

In certain types of plunger pumps fluid under relatively low pressure is directed from a control valve to one end of a piston chamber during the intake stroke of the piston within said chamber, and said fluid is then subjected to pressure within the chamber during the compressing stroke of the piston and is forced in a reverse direction through said valve. In other words, a volume of low pressure fluid is directed through a passageway in a given direction, and this same fluid, after being subjected to an increase in pressure, must be suddenly urged in a reverse direction through said passageway. Therefore, it is very desirable to have the valve which controls the intake and discharge of the low and high pressure fluids accurately timed so as to enable the pump mechanism to be driven at a relatively high speed without experiencing difficulties which might arise as a result of the sudden reversal in direction of flow of the low and high pressure fluids. If the valve which controls the flow of fluid toward and away from the pistons or plungers of the pump is timed and operated so as to suddenly cut off communication between the piston chamber and the high pressure side of the circuit and contemporaneously open said chamber to the low pressure or intake side of the circuit during the interval when the piston remains stationary at the end of its stroke, I have found that the pump may be efficiently operated at very high speeds. Heretofore it has been found necessary to employ check valves and other pressure relieving devices in combination with high speed plunger pumps in order to overcome difficulties which otherwise result from the frequent, successive changes from low to high pressure fluid conditions, or vice versa. Employing these auxiliary pressure relieving devices not only complicates the mechanical structure of the pump, but also introduces elements which are constantly subjected to wear, and hence must be repaired or replaced at frequent intervals in order to maintain the pump in running order.

It is one of the primary objects of my present invention to overcome the above mentioned difficulties by providing a plunger pump equipped with valve mechanisms which are operated in timed relation with respect to the reciprocation of the pump pistons in such a manner that the opening and closing of said valves with respect to the low and high pressure fluids will take place during the interval when the pump piston companion to each valve mechanism is substantially motionless.

A further object of my invention is to provide a fluid pump of the reciprocable plunger type, which is adapted to receive fluid under a relatively low pressure, and discharge said fluid under a relatively high pressure through the agency of an improved valve mechanism while said pump is operating at a relatively high speed.

More specifically, my invention contemplates the provision, in combination with a plunger pump of the type having a plurality of radial pistons actuated by a centrally disposed eccentric mechanism, of a plurality of valves, each valve being companion to one of the pistons and adapted to control the flow of low pressure fluid thereto and the high pressure fluid therefrom, the functioning of each valve being timed so as to effect the sudden shifting from low to high fluid pressure conditions in the companion piston chamber, or vice versa, when the eccentric piston actuating mechanism is substantially on dead center with respect to the companion piston, that is, when the piston companion to the valve is substantially motionless.

Still more specifically, my invention contemplates the provision, with the combination of elements recited above, of a plurality of reciprocable control valves, each of which is companion to one of the piston chambers and an improved eccentric driving mechanism for said valves, which is angularly disposed with respect to the eccentric driving mechanism for the pistons in such a manner as to properly control the interval during which each valve functions to effect the shifting between low and high pressure fluid conditions within the piston chamber.

A further object of my invention is to provide means in combination with the pump pistons and eccentric driving means therefor for causing said pistons to experience acceleration which is symmetrical with the deceleration during at least the compression stroke thereof, and to this end I propose to provide shiftable members having surfaces shaped and disposed so as to cause the pistons to experience the above mentioned symmetrical acceleration and deceleration.

More specifically, my invention contemplates the provision of a plurality of pivoted fingers, one companion to each pump piston with respect to an eccentric driving means as to make a rolling, driving contact with the pistons for the purpose set forth above, and for transmitting a driving

load to a localized central portion of the piston companion thereto, whereby to maintain the pistons in a balanced condition free from laterally acting forces.

The foregoing and numerous other objects and advantages will be more apparent from the following detail description when considered in connection with the accompanying drawings, wherein—

Figure 1 is a central sectional view taken longitudinally of a pump mechanism embodying my invention, said view being taken substantially along the line 1—1 of Figure 2;

Figure 2 is a transverse sectional view of said pump taken substantially along the line 2—2 of Figure 1;

Figure 3 is also a transverse sectional view thereof taken substantially along the line 3—3 of Figure 1;

Figure 4 is an enlarged fragmentary view of one of the valves disclosing the position thereof just as it cuts off communication between the piston chamber and the high pressure side of the pump;

Figure 5 is a similar view disclosing the position occupied by the valve just as it is about to establish communication between the piston chamber and the low pressure or intake side of the pump;

Figure 6 diagrammatically discloses one of the positions occupied by the eccentric piston driving mechanism when the valve member companion to said piston suddenly functions to cut off the piston chamber from the high pressure fluid and establish communication with the low pressure fluid;

Figure 7 illustrates various positions occupied by each finger as it swings from the low to the high position during 180° rotation of the eccentric driving member;

Figure 8 is a fragmentary disclosure of one of the fingers to more clearly illustrate the curvatures of the oppositely disposed surfaces of the free extremities of the fingers; and

Figure 9 is a graphic representation of the symmetry in piston displacement during the first and second half of the compressive stroke thereof.

Referring now to the drawings more in detail wherein like numerals have been employed to designate similar parts throughout the various figures, it will be seen that one embodiment of my invention contemplates a plunger pump which includes a stationary cylinder block or support 10, an intermediate casing 12 secured to the block 10 by bolts 14, and an end casing section 16 which is secured to the intermediate casing 12 by bolts or screws 18. The support or block 10 is provided with a plurality of radial piston chambers 20, which house companion pistons or plungers 22. The outer ends of the chambers 20 are closed by means of threaded caps 24, and each chamber 20 communicates at its outer end with a valve mechanism 26 through a passageway 28.

Each valve mechanism 26 includes a cylindrical valve member 30 which is reciprocable within a radial valve chamber 32 formed within the support or block 10. The valve member 30 comprises an end cylindrical section 30*a* an oppositely disposed end section 30*b*, and a central reduced section 30c. The central or reduced section 30c presents an annular valve chamber or port 34, which is in continuous communication with the passageway 28.

Low pressure fluid for charging the piston chambers 20 is adapted to be introduced from a source of supply (not shown) through a pipe line 36, which is in constant communication with an outer annular passage 38 and an inner annular passage 40 formed within the intermediate casing section 12. The annular passageway 38 communicates with the outer end of each valve chamber 32 through a port 42, and the annular passageway 40 communicates with the other end of said valve chamber through a port 44. A third annular passageway 46 communicates with the central portion of the valve chamber 32 through a port 48. This annular passageway is adapted to communicate with a point externally of the pump, as, for example, with the high pressure side of a hydraulic actuator (not shown) through a pipe line 50. Fluid within the annular passageways 38 and 40 is under a relatively low pressure, while fluid within the passageway 46 is under a relatively high pressure. The mechanism for imparting movement to the pistons or plungers 22 includes a plurality of pivoted fingers 52 which are arranged tangentially with respect to and rest upon the peripheral surface of an anti-friction driving ring 54. The driving ring 54 is mounted upon the cylindrical section of a driving member 56, which is adapted to be eccentrically positioned with respect to a driving sleeve 58. The pistons 22 and the valve members 30 radiate from a horizontal axis which is coincident with the axis of the sleeve 58. The driving member 56 is formed with a section 60, which is slidable transversely of the sleeve 58 within a slot 62, Figures 1 and 2. The sleeve 58 is mounted within suitable anti-friction bearings 64 and 66, the bearing being mounted within the end casing section 16 and the bearing 66 within the block or section 10, as clearly shown in Figure 1. The sleeve 58 is driven by an annular gear 68 which is coupled to the sleeve by means of a depressible key 70. The gear 68 is adapted to be driven from an external source of supply (not shown).

The driving member 56 is adapted to be laterally adjusted within the sleeve 58 through the agency of an adjusting member 72, which is formed with an intermediate cylindrical section 72*a* mounted within the sleeve 58 and an angularly disposed projection 72*b* which extends within a companion angularly disposed recess provided within the driving member 56. By applying a turning member, such as a wrench, to a nut 74, movement longitudinally of the sleeve 58 is imparted to the adjusting member 72.

To more clearly illustrate the symmetrical manner in which the fingers 52 function to accelerate and decelerate the plungers during the compression stroke thereof, I have included Figures 7, 8, and 9. Figure 7 illustrates various positions occupied by each finger as the eccentric driving member 54 travels through 180°, beginning with the finger in its lower position and continuing by eight successive equal movements of the driving member until the finger reaches its upper position. The center about which the driving member 54 rotates is designated by the letter A. The center of the driving member 54 is designated by the letter B, beginning at the lower point with B^1 and terminating at the upper position with B^9. Thus as the center B moves from the point B^1 to B^2, it will have experienced one-eighth of the first 180° of movement, and the piston will have traveled the distance b^1, as shown in Figure 7. Thus during the next one-eighth movement of the center B, namely from B^2 to B^3, the piston will shift through the distance b^2. In the same manner, the distance B^3—B^4 corresponds with the distance b^3, etc. Particular attention is directed to the distances b^1, b^2, b^3, b^4 and b^8, b^7, b^6, b^5, respectively,

which are equal. In other words, there is symmetry in relationship between the distances on opposite sides of the solid line in Figure 7, which indicates the position occupied by the piston 22 when it has completed half of its stroke. Stating it in another way, the increase in rate of displacement of each piston or plunger 22 during the first half of its compression stroke is symmetrical with the rate of displacement during the second half of the compression stroke. With the view of still more clearly illustrating the symmetry in piston movement, I have disclosed in Figure 9, a graph or chart in which I have plotted the angular displacement of the driving member 54 against the displacement of the piston. The points have been given letter designations corresponding with letter B^1 to B^9 inclusive in Figure 7. From Figure 9, the symmetrical functioning during the first and second half of the piston stroke will be more apparent. This movement should be distinguished from the conventional piston and connecting rod, wherein the displacement of the piston during one-half of the crank shaft rotation is not equal to the displacement during the second half of the crank shaft rotation.

The structural arrangement of the finger 52 is such as to cause the uniformity or symmetry in movement as mentioned above. To more clearly disclose the various positions occupied by the fingers 52, the piston 22, and the driving member or ring 54, I have shown corresponding positions, corresponding lines, such as dotted lines, dot and dash lines, etc.

Attention is directed to the fact that the fingers 52 are provided with oppositely disposed arcuate surfaces 52*a* and 52*b* which are so arranged as to impart symmetrical motion to the pistons 22 in response to the movement of the eccentric driving ring 54 through its orbit. By having this motion, the pistons accelerate and decelerate at the same rate, and as a result a uniform flow of fluid under pressure from the chambers 20 is obtained.

Outward movement is imparted to the valve members 30 by a ring or annulus 76, which encircles and is mounted upon an eccentric member 78. This member 78 is in turn keyed to the sleeve member 58, as clearly shown in Figure 2. The mounting of the ring 76 is such that the frictional engagement of the valve members 30 therewith is normally sufficient to prevent rotation of the member 78 during the rotation of the eccentric 78. Thus, by providing the ring 76, no side thrust is imparted to the valve members 30 during the actuation of the eccentric 78. If in certain instances it may be found necessary to further prevent rotation of the ring 72, this may be done by merely welding or otherwise forming a projection 80 on the peripheral surface thereof, as indicated by the dot-and-dash lines in Figure 2. This would of course necessitate the provision of a recess 80*a* within the block 10.

From the foregoing it will be apparent that as the driving ring 54 is swept through its orbit in response to the rotation of the eccentrically positioned driving member 56, the pistons or plungers 22 will be successively urged outwardly to compress fluid present within the piston chambers 20. The inward movement of the pistons 22 is occasioned in response to the introduction of fluid under relatively low pressure, and it is the functioning of the valve member 30 in connection with controlling the low and high pressure fluids which forms an important feature of my present invention. In this connection attention is directed to the fact that the eccentric valve driving ring 76 is substantially ninety degrees out of phase with the plunger driving ring 54. This can best be observed by referring to Figure 3. The fixed axis about which the pistons 22 and valve members 30 are radially disposed is indicated by the letter A, while the axis of the driving member 56 and ring 54 is indicated by the letter B, and the axis of the eccentric driving ring 76 is indicated by the letter C. These notations are uniform through the various figures. In Figure 3 it will be noted that the axis B of the ring 54 is ninety degrees out of phase with the axis C of the ring 76, the ring 76 being shown by dot-and-dash lines in Figure 3. Bearing the phased relationship of these driving rings in mind, it will be seen that, when the driving ring 54 occupies its dead center with respect to the piston which I have designated by the letter D, said piston will, during a fractional moment, be substantially motionless. In fact, when the high point of the ring 54 is positioned on dead center, as shown in Figure 3, or within two degrees of either side thereof, substantially no movement is imparted to the piston 22. I have determined that the degree of movement experienced by the piston D within this four-degree range of angular movement of the driving ring with a one-half inch stroke, is substantially .00012 inch. Referring now to the valve member which is companion to the piston designated by the letter D, it will be seen that this valve member, which I have designated by the letter D', Figures 1 and 2, is experiencing its maximum speed of travel, while the piston D is substantially stationary or motionless. It will also be apparent that the valve member D' is not only experiencing its maximum movement at this time, but is also closing the port 48 and opening the port 44. In other words, at the time the piston D is inactive, that is to say, at the time the fluid within the companion passageway 28 is motionless, the valve member D' is positioned as shown in Figure 4. The valve member D' in this position has just closed the port 48. The valve member will continue to move inwardly a distance of .018 inch to the position designated in Figure 5, in which position it is about to open communication between the port 44 and the annular passage or port 34. During this .018 inch movement of the valve the companion piston D has only traveled a distance of .00012 inch. Thus during the four-degree travel of the driving ring 54, as clearly indicated in Figure 6, the piston D experiences practically no movement, while the companion valve member moves at its maximum speed a sufficient distance to close the high pressure port 48 and open the low pressure or intake port 44. The piston designated by the letter D and the companion valve designated by the letter D' have been selected merely to illustrate the functioning of all of the pistons and valve members. Thus it will be apparent that when each of the plungers or pistons 22 occupies its "substantially motionless" position, the valve companion thereto is experiencing its maximum speed of travel and is closing one port and opening another.

Thus far I have described the timed relationship between the valve members and their companion pistons in connection with the change from high pressure to low pressure fluid conditions. It will be obvious that the same timed relationship holds true when the valve members operate to change from low pressure to high pressure conditions. In other words, when the valve member moves outwardly so as to close the port

44, as shown in Figure 5, and subsequently open the high pressure port 48 so as to enable fluid under pressure to be discharged from the companion piston chamber 20, the piston within said chamber will be initiating its compressing stroke.

In the operation of my improved pump or motor mechanism rotation is imparted to the sleeve 58 and its associated driving member 56 through the agency of the gear 68, and this rotation causes the driving ring 54 to successively move the pistons 22 outwardly, and causes the driving ring 76 to successively move the valve members 30 outwardly. A portion of the fluid under relatively low pressure from the pipe line 36 enters the outer ends of the valve chambers 32 so as to constantly urge the valve members into engagement with the peripheral surface of the driving ring 76 and the remaining portion of said fluid is directed into the port 44. Assuming that the valve designated by the letter D' is just reaching the position shown in Figure 4, it will be clear that communication between the passageway 28 and the high pressure discharge port 48 is being closed, and at this moment the piston D has reached the outer limit of its stroke and is substantially motionless. In fact, during a two-degree angular displacement of the driving ring 54 on either side of the dead center line shown in Figure 6, the piston D experiences substantially no movement. During this four-degree displacement the valve member D' is experiencing its maximum movement and shifts from the position shown in Figure 4 to the position shown in Figure 5. In Figure 5 the port 48 is sealed or closed, while the port 44 is about to be opened. When the port 44 opens the piston D will begin its inward stroke and is in readiness to receive fluid under low pressure from the port 44. Upon reaching the limit of its inward stroke, the piston D again experiences substantially no movement, and during this interval the valve member D' serves to close the port 44 and open the port 48 so that as the piston begins its outward stroke, fluid under pressure will pass from the piston chamber 20 through the passageway 28, the valve passage or port 44, the port 48, and thence through the annular passageway 46, into the pipe line 50. Each of the valve members 30 experiences the cycle of operation just described, and it will be clear, under such operating conditions, that the pump may be driven at a relatively high speed without experiencing any pulsating or other disturbing effects within the fluid medium.

From the foregoing description it will be understood that each of the valve members 30 functions to change from high pressure to low pressure conditions, or vice versa, during the interval when the piston companion thereto is substantially motionless. This means that the fluid body within the passageway 28 companion to said valve and piston, is also substantially motionless, and is therefore in readiness to be moved in either direction without experiencing the slightest degree of pulsation. Should the change from low to high pressure conditions, or vice versa, be made at a time when the fluid within the passageways 28 and piston chambers 20 companion thereto is being moved in either direction, these pulsating and other disturbing effects would be experienced with the result that the pump could not be operated at its maximum efficiency nor at any appreciable speed. By having the eccentric for controlling the functioning of the valve members and the eccentric for actuating the pistons driven by the same mechanism, I am able to materially simplify the pump construction and to obtain accurate timed relation between the functioning of the companion valves and pistons. It will be seen that the valve members 30 are continually urged inwardly through the action of the low pressure fluid from the pipe line 36, which enters the outer ends of the valve chambers 32 through the annular passagway 38 and the ports 42 communicating therewith. By employing my simplified radial valve arrangement, the high pressure fluid at no time communicates with the low pressure fluid, or vice versa, and this is a condition which materially expedites the functioning of the pump. Obviously my improved valve construction and mechanism associated therewith are not limited for use in connection with pumps, but are adapted for use in various types of fluid motors wherein it is desirable to maintain the low pressure and high pressure fluids operable independently of each other, and where a change from one pressure condition to another is to be made when the fluid within the motor or pump is substantially inactive or motionless.

From the foregoing description it will also be apparent that the free extremities of the pivoted fingers are shaped and disposed with respect to the eccentric driving means so as to effect a rolling, driving contact therewith, and further, to transmit a driving load to a localized central portion of a companion piston in the direction in which the piston moves. In other words, the pivoted fingers, in addition to causing the pistons to experience symmetrical acceleration and deceleration during at least the compression stroke thereof, also serve to preclude the transmission of load to said pistons which would have a tendency to cock the piston and thereby subject the piston to uneven wear. Thus it might be said that my improved finger construction serves to maintain the pistons in balance, as distinguished from driving mechanisms wherein the force which serves to shift the piston does not act continuously in the direction of movement of the piston, and in a manner to preclude the tendency for the piston to tilt or cock within its cylinder. My fingers are so shaped and disposed as to continuously apply the driving force at a central localized area at the inner ends of the pistons, thereby maintaining the piston "in balance", that is to say, free from any forces tending to shift the piston laterally.

Having thus described my invention, what I claim as new and desire to secure by Letters Patent is:

1. A pump including a piston support, a plurality of radially reciprocable pistons in said support, rotary driving means operatively associated with said pistons, and a pivoted finger associated with each piston and having a surface engaging the piston and an oppositely disposed surface engaging the periphery of the rotary driving means, at least one of said surfaces being curved, the shape and disposition of said surfaces with respect to said pistons and rotary driving means being such as to cause each piston to experience acceleration which is symmetrical with the deceleration during the compression stroke thereof.

2. A pump including a piston support, a plurality of radially reciprocable pistons in said support, rotary driving means operatively associated with said pistons, and shiftable means interposed between the piston and rotary driving means, means for supporting said shiftable means independently of said rotary driving means, a curved surface on one side of said shift-

able means adapted to engage the inner end of a companion piston, and a surface having a different curvature adapted to engage the periphery of said rotary driving means, said surfaces being shaped and disposed with respect to said pistons and rotary driving means for causing the piston companion thereto to experience acceleration which is symmetrical with the deceleration during the compression stroke thereof.

3. A plunger pump of the class described including a cylinder block, a plurality of radially disposed pistons reciprocable within said block, a driving mechanism for actuating said pistons, said mechanism causing relative dwells of the pistons at the ends of the movement thereof, a plurality of valves, one companion to each piston, a passageway connecting each of said valves with its companion piston, intake and discharge ports adapted to be opened and closed by said valves, each valve in one shifted position being adapted to direct fluid from the intake port through said passageway to said piston and in another shifted position adapted to receive fluid under pressure from said passageway and direct said fluid to the discharge port, each valve arranged to overlap companion intake and discharge ports during the relative dwells of the piston communicating therewith, means for actuating each valve to close one of said ports and open the other during the dwell of the companion piston, whereby to enable the pump to be efficiently driven at relatively high speeds, and shiftable means interposed between said driving mechanism and pistons and supported independently thereof, said shiftable means being shaped and disposed with respect to both said driving mechanism and the inner ends of said pistons so as to cause said pistons to experience movement during the first half of the compression stroke which is symmetrical with the movement during the second half of the compression stroke.

4. A plunger pump of the class described including a cylinder block, a plurality of radially disposed pistons reciprocable within said block, a driving mechanism for actuating said pistons, said mechanism causing relative dwells of the pistons at the ends of the movement thereof, a plurality of valves, one companion to each piston, a passageway connecting each of said valves with its companion piston, intake and discharge ports adapted to be opened and closed by said valves, each valve in one shifted position being adapted to direct fluid from the intake port through said passageway to said piston and in another shifted position adapted to receive fluid under pressure from said passageway and direct said fluid to the discharge port, each valve arranged to overlap companion intake and discharge ports during the relative dwells of the piston communicating therewith, rotary driving mechanism including a member rotatable about a fixed axis, a driving member eccentrically adjustable with respect to and driven by said driving member for imparting movement to said pistons, a second eccentric driving means carried by the first mentioned driving member for actuating each valve to close one of said ports and open the other when the fluid within the companion passageway is substantially motionless, and shiftable means interposed between said driving mechanism and pistons and supported independently thereof, said shiftable means being shaped and disposed with respect to said driving mechanism and the inner ends of said pistons so as to cause said pistons to experience movement during the first half of the compression stroke which is symmetrical with the movement during the second half of the compression stroke.

5. A plunger pump of the class described including a cylinder block, a plurality of radially disposed pistons reciprocable within said block, a driving mechanism for actuating said pistons, said mechanism causing relative dwells of the pistons at the ends of the movement thereof, a plurality of valves, one companion to each piston, a passageway connecting each of said valves with its companion piston, intake and discharge ports adapted to be opened and closed by said valves, each valve in one shifted position being adapted to direct fluid from the intake port through said passageway to said piston and in another shifted position adapted to receive fluid under pressure from said passageway and direct said fluid to the discharge port, eccentric driving means for actuating each valve, each valve being adapted to overlap said ports as the companion piston reaches the limit of its compression stroke, thereby enabling one port to close before opening the other port, said valve actuating means functioning to open and close said ports through the agency of the valve when the companion piston is positioned at one extremity of its stroke and the valve is experiencing its maximum speed of travel, and shiftable means interposed between said driving mechanism and pistons and supported independently thereof, said shiftable means being shaped and disposed with respect to said driving mechanism and the inner ends of said pistons so as to cause said pistons to experience movement during the first half of the compression stroke which is symmetrical with the movement during the second half of the compression stroke.

6. A plunger pump of the class described including a cylinder block, a plurality of radially disposed pistons reciprocable within said block, a driving mechanism for actuating said pistons including an eccentric driving member and a shiftable member having oppositely disposed arcuate surfaces interposed between said driving member and each piston and supported independently thereof, said shiftable member being shaped and disposed with respect to both the periphery of said driving member and the inner ends of said pistons so as to cause said pistons to experience symmetry in acceleration and deceleration during the outward movement thereof and cooperating with the eccentric driving member in causing a relative dwell of the pistons upon the completion of the compression stroke thereof, a plurality of valves, one companion to each piston, a passageway connecting each of said valves with its companion piston, intake and discharge ports adapted to be opened and closed by said valves, each valve in one shifted position being adapted to direct fluid from the intake port through said passageway to said piston and in another shifted position adapted to receive fluid under pressure from said passageway and direct said fluid to the discharge port, and means for actuating each valve to close one of said ports and open the other during the above mentioned dwell of the companion piston, whereby to enable the pump to be efficiently driven at relatively high speeds.

7. A plunger pump of the class described including a cylinder block, a plurality of radially disposed pistons reciprocable within said block, a driving mechanism for actuating said pistons including an eccentric driving member and a piv-

otal finger interposed between said driving member and each piston, said finger being shaped and disposed with respect to both the periphery of said driving member and the inner ends of said pistons so as to cause said pistons to experience symmetry in acceleration and deceleration during the outward movement thereof and cooperating with the eccentric driving member in causing a relative dwell of the pistons upon the completion of the compression stroke thereof, a plurality of valves, one companion to each piston, a passageway connecting each of said valves with its companion piston, intake and discharge ports adapted to be opened and closed by said valves, each valve in one shifted position being adapted to direct fluid from the intake port through said passageway to said pistons and in another shifted position adapted to receive fluid under pressure from said passageway and direct said fluid to the discharge port, and means for actuating each valve to close one of said ports and open the other during the above mentioned dwell of the companion piston, whereby to enable the pump to be efficiently driven at relatively high speeds.

8. A pump including a piston support, a plurality of radially reciprocable pistons in said support, rotary driving means operatively associated with one end of the pistons, and shiftable means interposed between the driven end of each piston and the periphery of said rotary driving means, said shiftable means being shaped and disposed with respect to both the pistons and the driving means so as to make a rolling contact therewith for causing each piston to experience acceleration which is symmetrical with the deceleration during the compression stroke thereof, and supporting means independent of said driving means for mounting said shiftable means within the pump.

9. A pump including a piston support, a plurality of radially reciprocable pistons in said support, rotary driving means operatively associated with one end of the pistons, and shiftable means interposed between the driven end of each piston and the periphery of said rotary driving means, said shiftable means being shaped and disposed with respect to said rotary driving means and said pistons for transmitting a driving load to a localized central portion of each piston in the direction of movement of said piston and for causing each piston to experience acceleration which is symmetrical with the deceleration during the compression stroke, and supporting means independent of said driving means for mounting said shiftable means within the pump.

10. A pump including a piston support, a plurality of radially reciprocable pistons in said support, rotary driving means operatively associated with said pistons, and shiftable means comprising a pivoted finger interposed between the inner end of each piston and the periphery of said rotary driving means, said finger being shaped and disposed with respect to said rotary driving means and the inner ends of said piston so as to cause each piston to experience acceleration which is symmetrical with the deceleration during the compression stroke thereof.

11. A pump including a piston support, a plurality of radially reciprocable pistons in said support, eccentrically adjustable rotary driving means operatively associated with said pistons, a pivoted finger associated with each piston and having a surface engaging the piston and an oppositely disposed surface engaging the periphery of the rotary driving means, at least one of said surfaces being curved, the shape and disposition of said surfaces with respect to said pistons and rotary driving means being such as to cause each piston to experience acceleration which is symmetrical with the deceleration during the compression stroke thereof, and means for eccentrically adjusting said rotary driving means.

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