March 27, 1945. A. C. SINCLAIR 2,372,523

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Original Filed May 12, 1939 3 Sheets-Sheet 1

Fig. 1.

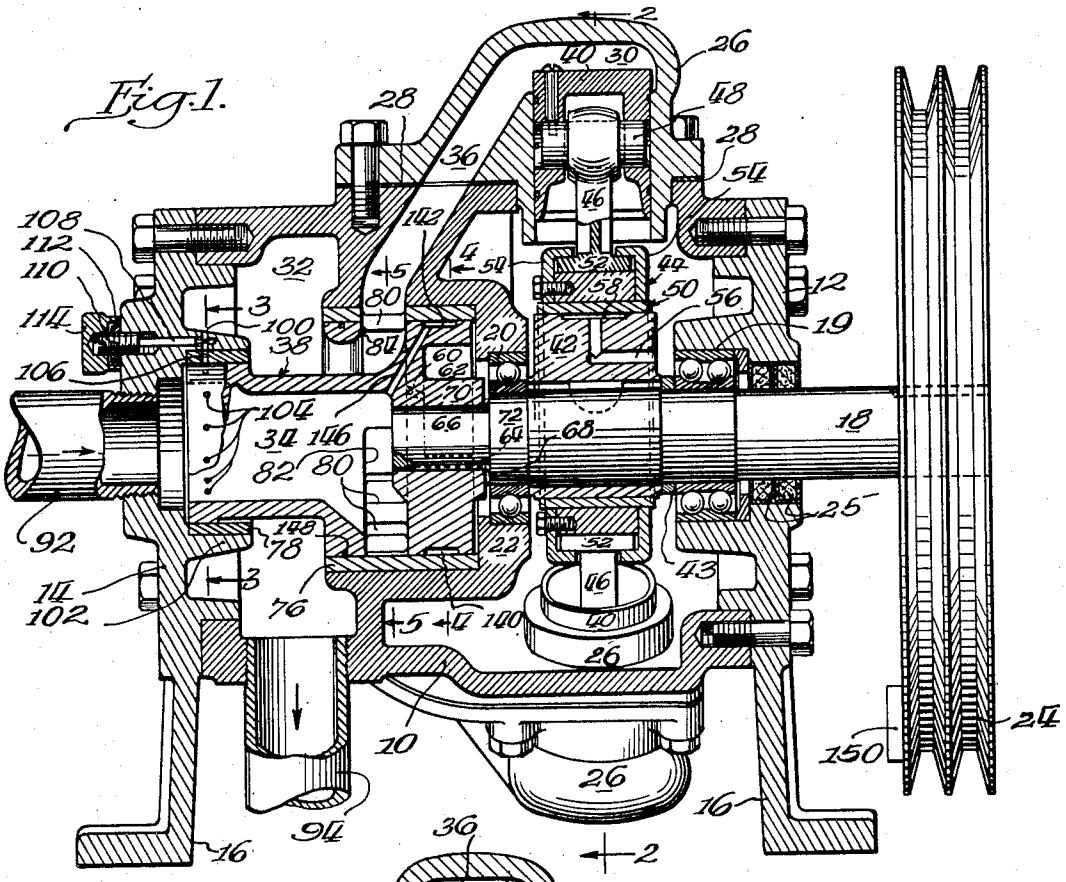
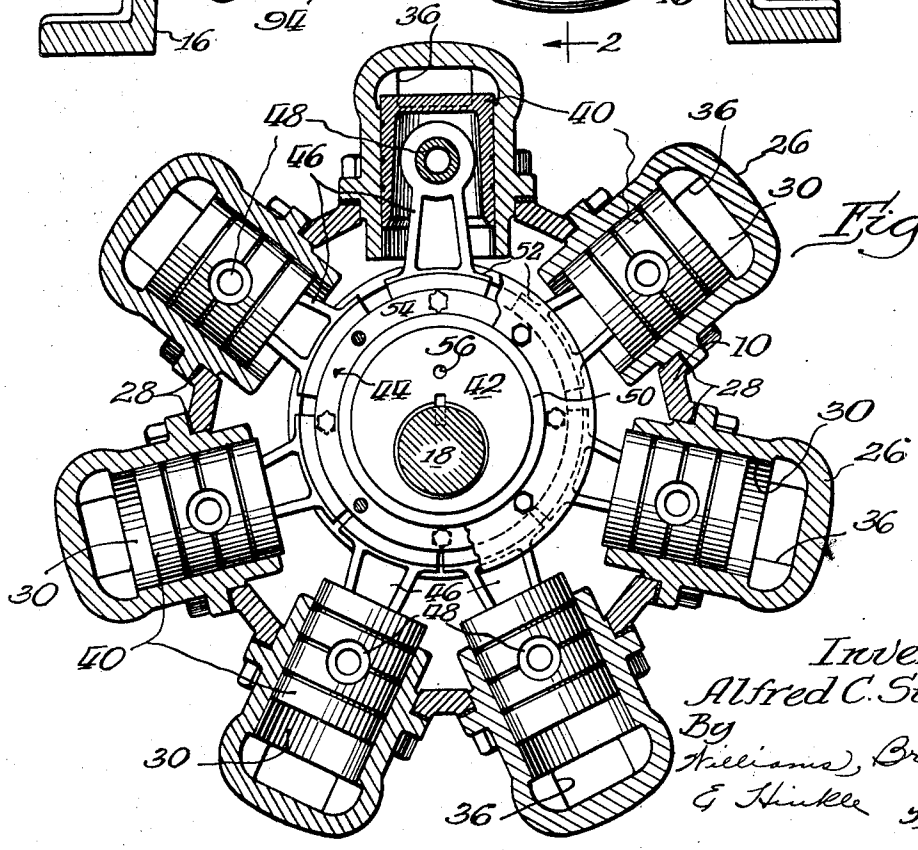


Fig. 2.



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Fig. 3.

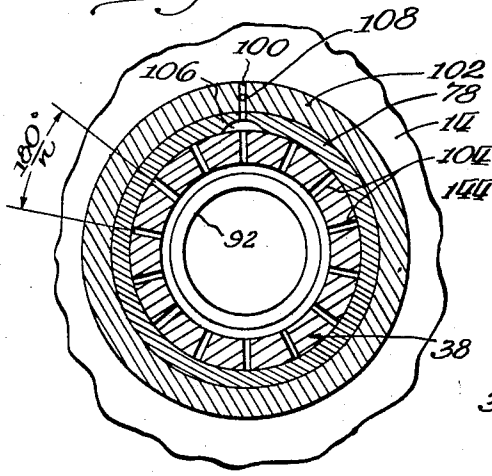


Fig. 4.

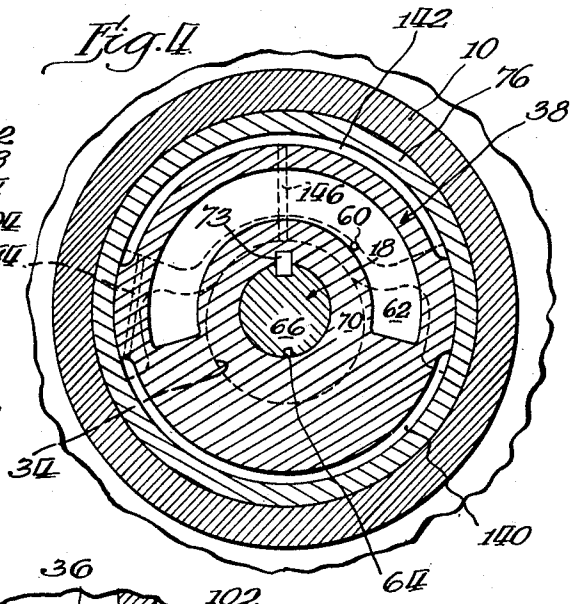


Fig. 5.

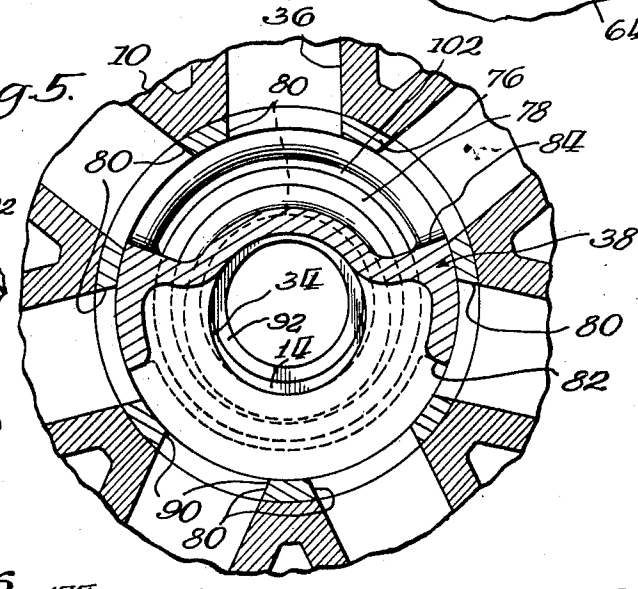


Fig. 10.

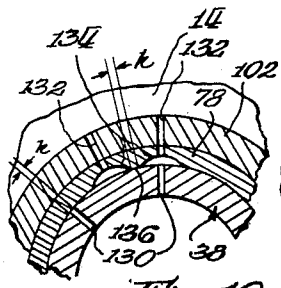
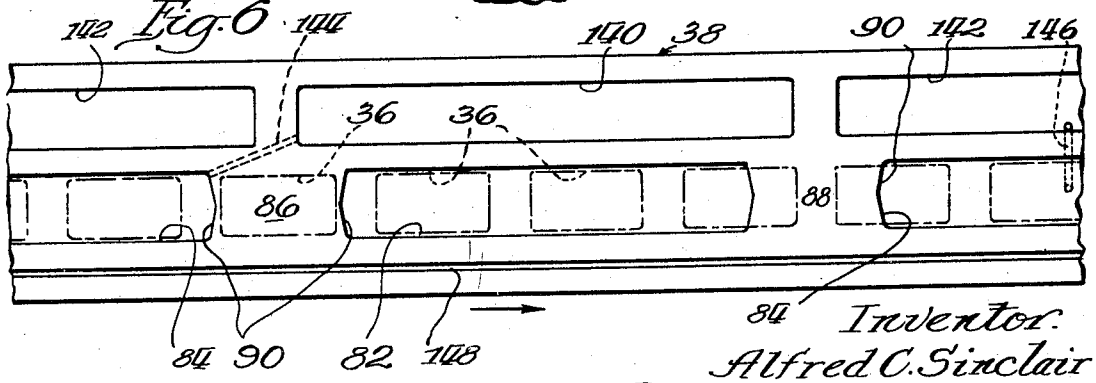


Fig. 6



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Fig. 11.

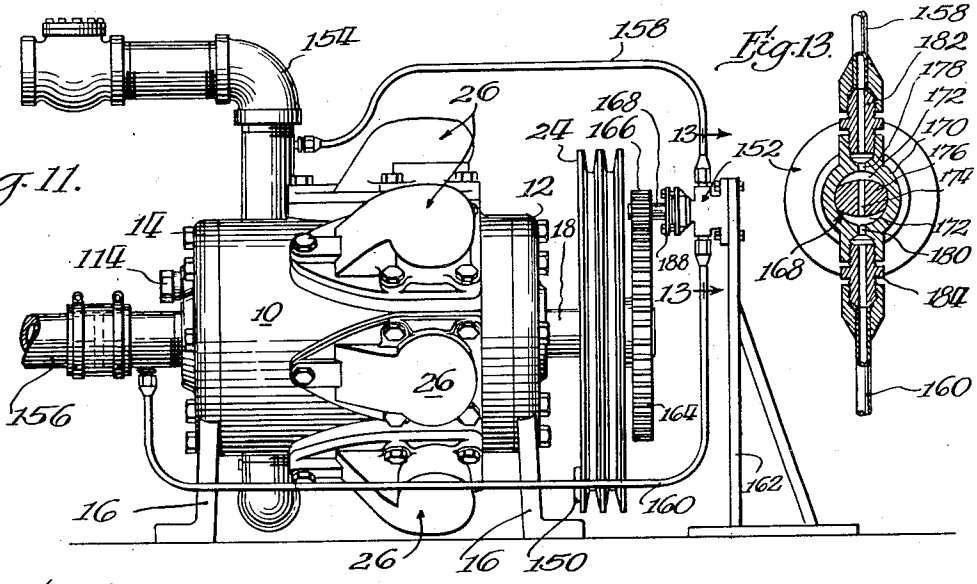


Fig. 13.

Fig. 7.

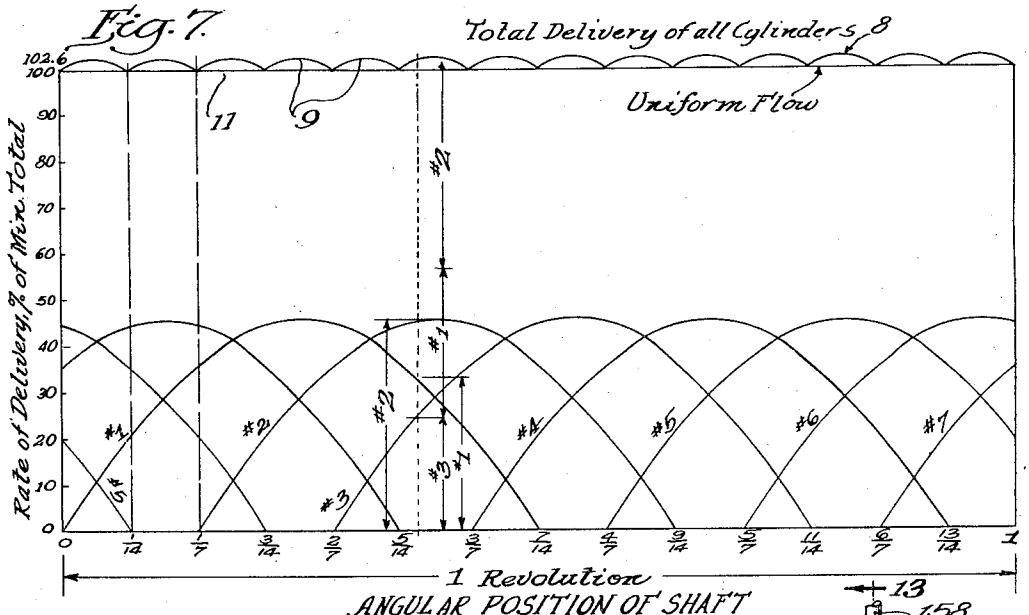


Fig. 8.

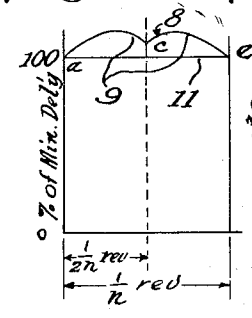


Fig. 9.

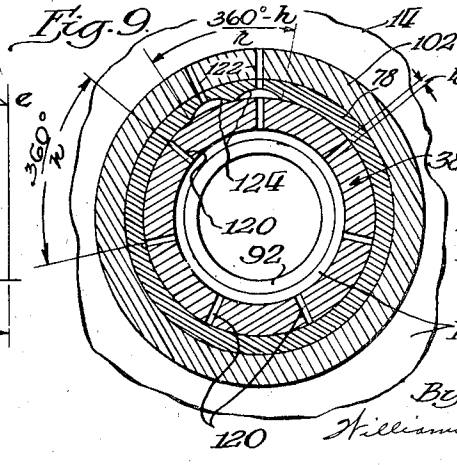
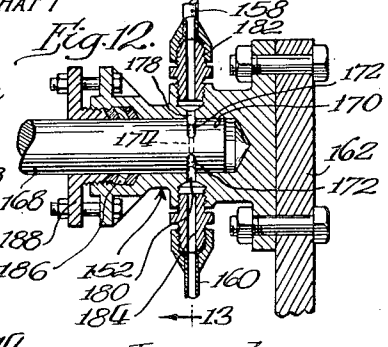


Fig. 12.



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Patented Mar. 27, 1945 2,372,523

UNITED STATES PATENT OFFICE

2,372,523

PUMP

Alfred C. Sinclair, Houston, Tex.

Original application May 12, 1939, Serial No. 273,227. Divided and this application August 9, 1941, Serial No. 406,116

17 Claims. (Cl. 103—174)

The present invention relates to pumps of the positive displacement type and has for its primary object the provision of a new and improved pump of high efficiency, a pump that may be built at relative low cost, and one capable of delivering a smooth flow of fluid which may be used substantially noiselessly at the point of application.

This application is a division of my prior copending application Serial No. 273,227, filed May 12, 1939, issued as Patent No. 2,279,645 on April 14, 1942. In it there is disclosed a pump of the positive displacement type having a substantially smooth delivery, which smooth delivery is obtained by by-passing limited quantities of the pump discharge from the discharge side to the suction side of the pump. This by-passing is controlled by a valve preferably forming part of a rotary distribution valve that alternately connects the pump cylinders to the suction and discharge passages.

An object of the present invention is to provide a rotary distribution and by-pass controlling valve that is balanced against hydraulic thrust.

A further object of the invention is to provide a rotary distribution and by-pass controlling valve that is balanced against both axial and lateral hydraulic thrust, thereby to prevent rapid wear of the bearing surfaces of the valve and the development of leakage around the valve.

A further object of the present invention is to provide a pump of the type described in which a positive pressure is maintained in the crank case.

A further object of the present invention is to provide a new and improved rotary distribution valve wherein the above-mentioned hydraulic balance and maintenance of positive pressure is provided for, in part at least, by the valve.

A further and somewhat more general object of the present invention is to provide means preventing the ingress of air into the crank case around the valve operating shaft, and thereby prevent the impairment of lubrication of the moving parts of the crank mechanism and to prevent the noisy operation of the pump which may result from permitting air to find its way into the cylinders through piston clearances.

A further object of the present invention is to provide a pump construction that permits the use of considerably lighter parts in the construction of the pump without decreasing the factor of safety. This advantage results from the fact that the crank case is maintained at positive pressure.

A further object of the invention is so to construct the distribution valve and the passages leading therefrom to the cylinders that substantially all clicking noises are avoided.

Other objects and advantages of the present invention will become apparent from the ensuing description in the course of which reference is had to the accompanying drawings, in which:

Fig. 1 is an axial vertical cross-sectional view through one embodiment of the pump of the present invention (see Fig. 11 for a side elevational view of a similar pump);

Fig. 2 is a transverse view taken along the line 2—2 of Fig. 1 (this view is partly in section better to illustrate certain details of construction);

Fig. 3 is an enlarged partial transverse cross-sectional view taken along the line 3—3 of Fig. 1, illustrating details of the pressure pulsation compensation means of the present invention;

Fig. 4 is a view similar to Fig. 3 taken along the line 4—4 of Fig. 1, illustrating certain details of construction of the hydraulic balancing means for the rotary distribution valve;

Fig. 5 is a view similar to Fig. 4 taken along the line 5—5 of Fig. 1 illustrating further details of construction of the rotary distribution valve.

Fig. 6 is a view showing the developed peripheral surface of the right half of the rotary distribution valve illustrating further details of construction thereof and particularly the means for balancing the lateral pressures;

Fig. 7 is a chart in which the angular positions of the shaft (and likewise of the rotary distribution valve) are plotted as the abscissae, and the rates of delivery are plotted as the ordinates and showing the individual output of each cylinder as well as the total output of the pump, which, in the illustrated embodiment, comprises seven cylinders. The chart also shows the output of the pump when supplied with pulsation compensating means of the present invention. The chart is for a pump having a piston connecting rod that is long as compared with the piston stroke;

Fig. 8 is a view similar to that shown in Fig. 7, illustrating the nature of the discharge of a pump having an odd number of cylinders and a connecting rod the length of which is short as compared to the piston stroke;

Fig. 9 is a view similar to Fig. 3 of a modified form of by-pass valve adapted particularly to compensate for pulsations of the type shown in Fig. 8;

Fig. 10 is a view similar to Fig. 9 illustrating how that modification may be modified to provide compensation for pumps having pulsations of the type shown in Fig. 7;

Fig. 11 is a side elevational view of a pump

similar to that shown in the previous figures provided, however, with a further modified means for compensating for the pressure pulsations. In this modification the compensation is provided by a separate valve operated in synchronism with the pump shaft and rotary distribution valve and adapted periodically to connect the suction and discharge passages of the pump;

Fig. 12 is a vertical axial cross-sectional view through the valve illustrated in Fig. 11; and

Fig. 13 is a vertical transverse cross-sectional view taken through the valve along the line 13—13 of Fig. 11.

The pump of the present invention is best illustrated, as a whole, in Figs. 1 and 2 (for a side elevation of a pump having substantially the same external appearance, attention is directed to Fig. 11). It comprises a stationary crank case 10 that is made liquid tight for a purpose set forth hereinafter. The crank case is preferably made of automobile cylinder iron and is closed at its end by centrally apertured end plates 12 and 14 attached to the crank case by suitably spaced bolts. The end plates are each provided with integral identical downwardly extending feet 16, whereby the pump may be securely attached to a suitable base plate or other supporting means.

The pump shaft 18 is rotatably supported in and held against end play by a pair of spaced apart bearings 19 and 20 of a familiar type, mounted respectively in the drive end plate 12 and a radially inwardly extending partition 22 formed centrally of and integrally of the crank case. The pump shaft may be driven by any suitable means, and in Fig. 1 the shaft is illustrated as driven by a double pulley 24 secured to the end of the shaft projecting beyond end plate 12. Leakage of fluid along the pump shaft is prevented by suitable seals 25.

The pump comprises a plurality of cylinders 26 cast and machined independently of the crank case and preferably made of the same material as the crank case. The cylinders are removably mounted upon the crank case by a plurality of bolts, and leakage of fluid around the cylinders is prevented by gaskets 28. The cylinders are equally spaced about the periphery with the cylinder bores 30 located nearest the drive end of the pump. It is preferable but not necessary that the pump have an odd number of cylinders for the reason, as will be explained more fully hereinafter, that the magnitude of the pulsations in pressure is substantially less than when an even number of cylinders is utilized.

The cylinder bores are selectively placed in communication with a delivery chamber 32, or a suction chamber 34, through cored passageways 36 under the control of a full floating, ported rotary distribution valve 38 that is rotated synchronously with the pump shaft 18. The valve forms one of the more important features of the present invention and will be described in considerable detail hereinafter.

The pump pistons 40 are reciprocated in their associated cylinders by an eccentric 42 fixedly attached to the pump shaft, as by a key. The eccentric is spaced from the end plate 12 by a spacer 43 interposed therebetween and bearing 19. The eccentric is operatively connected to the pistons by a bearing ring assembly, indicated generally by reference character 44, connecting rods 46 and pins 48. The bearing ring assembly comprises a composite eccentric ring 50 rotatable around the eccentric 42. The connecting rods have slipper end portions 52 that are held in contact with the outer surface of the eccentric ring by a pair of retainer rings 54 secured to the ring. It may be seen that the connecting rods are maintained in contact with the eccentric ring but are free to oscillate about the common center of the slipper portions and the eccentric ring.

The eccentric and its associated ring are lubricated through a conduit 56 leading from within the interior of the crank case to a flat 58, provided on the peripheral surface of the eccentric, it being contemplated that the fluid circulated by the pump of the present invention be of the type possessing lubricating properties.

In accordance with another of the important features of the present invention, the interior of the crank case is filled with the fluid at a pressure that bears a predetermined relation to the delivery pressure of the pump, thereby to enable lighter parts to be used in the construction of the pump without decreasing the factor of safety and at the same time prevent the ingress of air into the pump which would, if not prevented from entering the crank case, impair lubrication and result in noisy operation.

The fluid is supplied to the interior of the crank case from the delivery chamber through a small diameter bore 60 formed at the right-hand end of the rotary distribution valve 38 (as viewed in Fig. 1), a substantially semicircular channel 62 formed in the right-hand end of the valve, and through openings between the spaced apart balls of the bearing 20. The pressure is relieved through a variable area pressure relieving groove 64 formed at the reduced end portion 66 of the pump shaft 18, adapted to rotate the rotary distribution valve. This valve is capable of limited sliding movement with respect to the shaft in order that the valve may be balanced against axial hydraulic thrust in a manner to be described shortly. When a balance exists, the groove 64 establishes communication from the interior of the crank case to the suction chamber 34 through an annular space, indicated by the reference character 68, between a hub portion 70 of the rotary valve and a shoulder formed by portion 72 of the shaft and a portion of the bearing 20. By proportioning the areas of the hole bore, groove 64, and the valve surfaces subject to the delivery and discharge pressures, the pressure within the interior of the crank case may be designed to bear a fixed relationship to the pressure in the discharge chamber.

A decided advantage in economy of material is secured by maintaining a positive crank case pressure. During the upstroke of each piston full delivery pressure is exerted upon the piston head, thereby determining the thrust which would have to be carried by the piston pin and the connecting rod slipper. However, the crank case pressure is exerted on the underside of each piston, thereby to reduce the thrust in the same proportion that the crank case pressure bears to the delivery pressure. The pistons and rods of the cylinders performing their suction strokes at this time are, of course, subject to a tension due to the crank case pressure, whereas they would otherwise be practically without load. However, the bearing loads so created will not exceed those produced on the delivery stroke. The result is to reduce, by as much as desired up to one half, the maximum load on each piston pin and connecting rod, thereby allowing lighter parts to be used with the same degree of safety.

The rotary distribution valve 38, which serves selectively to connect the cylinders to the dis-

charge and suction chambers 32 and 34, respectively, is keyed to the reduced end portion 66 of shaft 18 by a key 73 so constructed and arranged with respect to the shaft and valve that the latter is free to move endwise of the shaft a limited extent, as previously mentioned. The valve is journaled for rotation in a pair of bushings 76 and 78, and the large end of the valve (the right end as viewed in Fig. 1) is fitted closely within the bushing 76. The latter is provided with a series of spaced apart ports 80 coinciding with the openings in the passageways 36 giving the latter access to the valve. The large end of the valve is also provided with two identical oppositely located ports 82 and 84, respectively, the suction and discharge ports (see Fig. 6), separated by solid portions 86 and 88 each of a length sufficient to cover one port in the bushing 76 with a slight overlap.

In order to prevent the too sudden admission and cut off of fluid and the resulting clicking sound, the ends of the ports 82 and 84 are preferably rounded or pointed, as indicated by reference character 90.

Fluid is supplied to the suction chamber 34 of the pump through an inlet passage comprising a pipe 92 threaded into the end plate 14 concentrically with respect to and opening directly into the suction chamber 34. The discharge passage comprises a pipe 94 threaded into a suitably located hole in the crank case. In the illustrated embodiment the discharge passage is shown on the under side of the pump, but it should be understood the pipe may be located wherever desired.

The distribution valve 38 is keyed to the pump shaft 18 in such position with respect to the shaft that as the piston in a cylinder commences its downstroke, the corresponding port 36 is just opened to the suction chamber 34 by the suction port 82 and as the piston attains its maximum velocity, the port is fully opened to the suction chamber, and as the piston reaches the bottom of its stroke the port is just closed. Further rotation of the shaft causes the piston to ascend in the cylinder and at the same time the rotation of the distribution valve 38 opens a passage from port 36 to the discharge port 84 in the valve, thereby to connect that port with the discharge chamber 32. Each cylinder is alternately connected to the suction and discharge chambers and the construction of the valve is such that at least three cylinders are open to each of the chambers at any one time.

Before proceeding further with a detailed discussion of the pump of the present invention, it is deemed advisable to consider the nature of the discharge from a pump of this type to bring out more clearly the disadvantages of the pumps now known to the art and thereafter to point out how these disadvantages are overcome.

Referring now to the chart shown in Fig. 7, it may be seen that the discharges of the seven separate cylinders of the pump described above combine to produce a total discharge flow that is nearly uniform, when the shaft is driven at a constant speed. Each cylinder discharges during one half of a revolution, and the discharges overlap in the manner shown in the chart of Fig. 7, where the discharge from each cylinder during one rotation of the pump shaft is indicated by the substantially sinusoidal curves marked numbers 1 to 7, inclusive. The summation of the output of the individual cylinders gives the total output of the pump, as indicated by reference character 8. This curve shows that there are two ripples 9 for each cylinder per revolution of the pump shaft, which is the case with pumps having an odd number of cylinders greater than one. As previously mentioned, when the number of cylinders is an even one, there are as many ripples as cylinders, but the magnitude of the ripples is about twice as great.

The minimum delivery 11, to which a value of 100% has been abitrarily assigned, is only about 2.6% less than the maximum delivery. The diagram shown in Fig. 7 is illustrative of a pump having connecting rods that are very long compared with the stroke of the pump. Where the connecting rods are relatively short compared with the stroke of the pump, the total output curve is somewhat different, as shown in Fig. 8.

For many purposes the pulsations in the discharge of the pump are not of particular importance, and the amount of variation may be reduced by increasing the number of cylinders and by using an odd number of cylinders. For certain purposes, as for operating directly actuated plunger hydraulic elevators, the pulsations in the discharge of the pump are apparent in the elevator cab as a vibration and noise which is disagreeable and oftentimes intolerable. Damping devices installed in the delivery line are found to be ineffective as a means of reducing this vibration to a desired minimum.

To provide an economical as well as thoroughly satisfactory means for avoiding the pulsations referred to and to provide a substantially uniform discharge, the pulsations are eliminated in the pump of the present invention by periodically by-passing a portion of the fluid from the discharge chamber 32 into the suction chamber 34, the period between by-passes being so arranged that they correspond to the ripples in the output of the pump, and the amount being so regulated as to correspond to the magnitude thereof. While this arrangement for smoothing the discharge of the pump entails a sacrifice of some efficiency, as a deliberate leak is introduced between the discharge and suction chambers of the pump, the loss is only a very small percentage of the total flow. In the chart shown in Fig. 7 the loss is only about 1½% of the pump output with a corresponding energy loss. However, the mechanical efficiency of the pump may still be well above that of other types of pumps which are inherently free from pulsations in the delivered flow.

Referring now more particularly to Figs. 1 and 3, it may be seen that the discharge chamber is adapted periodically to be connected to the suction chamber through a small diameter hole 100 extending through an internal boss 102 formed integrally with end plate 14 and through the valve bushing 78 (which is mounted in the boss) and through one of fourteen small diameter holes 104 through the left end of the rotary distribution valve 38, depending upon the position of the pump shaft and valve. The hole 100, extending through the boss 102 and bushing 78, opens into a groove 106 having a vertical cross-section of meniscus shape for the purpose of making the by-pass passage of a shape to produce a by-pass corresponding to the shape of the pulsation. The hole 100 and groove 106 are so located as to be entirely closed by the portions of the valve between the holes 104 of the latter when a piston is at its upper or lower dead center, that is, when the output of the pump is at one of its minimum points.

The action of the by-pass valve is to produce a leak from the discharge chamber into the suction chamber which begins gradually, builds up to a maximum, decreases gradually, and is cut off for an instant; and begins again in the manner described. This cycle is repeated fourteen times per revolution of the pump shaft 18, thereby to compensate for the increases in output which would otherwise occur. The result is a change in the pump output from the type shown by curve 8 to that shown in curve 11 which is substantially uniform.

The rate of fluid leakage from the discharge into the suction chamber may be regulated by a needle valve 108 adapted to vary the effective area of the hole 100. The needle valve is provided with a threaded head 110 that is held in locked relation to the end plate by a lock nut 112. Accidental adjustment of the needle valve is prevented by a cap 114 surrounding it and the lock nut so that access to these can be had only when the cap is unscrewed.

The exact size of the bleed holes 100 and 104 as well as the shape of the groove 106 may best be determined by trial for the particular combination of pump speed and delivery pressure of the pump. A properly selected combination will work successfully over the range of delivery pressures ordinarily required in an installation.

When the effective length of the connecting rods is short, as compared with the stroke of a cylinder, it may occur that this method of compensation for delivery pulsation is inadequate because of the distortion of the individual cylinder discharges. The resulting curve for total discharge in a pump with an odd number of cylinders may then have unsymmetrical cusps occurring alternately, the minimum occurring as each cylinder reaches the top dead center being perceptively greater than the minimum occurring as each cylinder is at bottom dead center—substantially as shown in Fig. 8. Referring now to this figure, which illustrates only a portion of a curve similar to that shown in Fig. 7 with exception that the individual cylinder discharge curves are not shown and, instead of illustrating a pump having a specific number of cylinders, the number of cylinders has been generalized. From this it may be seen that the number of ripples 9 is equal to twice the number of cylinders per revolution of the pump shaft and that the first-mentioned minimum values (*a* and *e* in Fig. 8) are spaced apart a distance equal to $1/n$ revolution as are the second-mentioned minimum values (*c* in Fig. 8), whereas the latter are spaced a distance $1/2n$ revolution from the former.

To compensate for pulsations of the type illustrated in Fig. 8, the construction illustrated in Fig. 9 is utilized. The valve 38 is provided only with a number of holes equal to the number of cylinders, in this case seven, indicated by the reference characters 120 of a diameter *h*. Two holes 122 are drilled through the bearing bushing 78 and the boss 102 formed integrally with the end plates 14. They are spaced apart a distance equal to one-half the distance between adjacent holes 120 in the bearing and open into grooves 124 having a cross-section that is meniscus shaped. The grooves 124 are so spaced that their outer ends reach from one hole to another without opening either. Thus, when the discharge from the pump is at a minimum (point *a* Fig. 8) as one piston is at its bottom dead center, there is no leakage through holes 120 and 122. As the discharge increases to a maximum, the flow through a pair of holes 120 and 122 becomes a maximum. As the flow reaches the second minimum point (point *c* Fig. 8), the grooves 124 only partly cover a hole 120 and sufficient flow occurs to neutralize the difference between the ordinates of points *a* and *c*. In this manner complete compensation may be secured in spite of the irregularities of discharge and a smooth uniform flow corresponding to the line 11 (marked 100%) in Fig. 8 is obtained.

The last-described compensation arrangement may be modified to provide compensation for pulsations occurring when the length of the connecting rod is great enough to produce a discharge similar to that shown in the chart of Fig. 7. This modification, which is shown in Fig. 10, possesses the advantage that it requires less drilling than that first described.

Referring now to Fig. 10, it may be seen that the distribution valve 38 is provided with seven equally spaced holes 130 just as in the modification of Fig. 9, and that bushing 78 and boss 102 are likewise provided with a pair of holes 132 spaced apart as described in the last-mentioned modification and opening into the grooves 134 of meniscus shaped cross-section. The grooves 134 are so spaced apart that the portion of the bearing bushing 78, indicated by reference characters 136, is wide enough to cover a hole 130. It may be seen that by this arrangement the leakage pulsations are all alike and therefore compensate for the type of ripples disclosed in the chart of Fig. 7.

Another of the primary features of the present invention is the provision of a distribution valve that is balanced against both axial and lateral (or radial) hydraulic thrust, thereby to provide a valve having less wear of the seating surfaces and less likelihood of the development of leakage at these surfaces. The means for providing the axial balance also serves to maintain the positive pressures in the crank case previously described but it may be utilized independently of this feature, if desired, as by utilizing a separate chamber in place of the crank case. This means comprises the constant area orifice or passage 60 from the discharge chamber to the interior of the crank case and the variable area orifice or passage 64—68 from the interior of the crank case to the suction chamber 34, and includes the proper dimensioning of the axial cross-section areas of the valve subject to the discharge and crank case pressures. The ratio between the areas of the orifices determines the ratio between the pressure differences from the discharge chamber to the crank case and from the crank case to the suction chamber.

Referring now to Fig. 1, it may be seen that the entire right end of the valve as viewed in this figure (the larger end) may be considered as a piston subject to crank case pressure, which pressure tends to shift the valve to the left. The left end (the smaller end) of the valve, forms a piston subject to the suction pressure, which is substantially zero. The difference between the areas of the two ends is subject to the pressure existing in the delivery chamber, which pressure tends to urge the valve to the right. It may be seen that the ratio of the area of the right end of the valve to the difference in areas of right and left ends, must be the same as the ratio between the discharge and crank case pressures in order for the valve to be in axial equilibrium.

If the valve be assumed to be in the extreme right-hand position, the hub portion 70 thereof closes the passage 68 and fluid under pressure may enter the crank case from the discharge chamber through the passage 60. Under these circumstances the pressure in the crank case tends to increase until it equals that in the discharge chamber. However, as soon as the crank case pressure becomes great enough, the thrust against the entire area of the right end of the valve overbalances the opposite thrust due to delivery pressure against the smaller area exposed thereto and the valve is moved to the left uncovering a portion of the groove 64. The valve finally takes a position such that the opposing thrusts are equal, at which time the pressure on each area multiplied by the area against which the pressure acts, are equal, and the relation between discharge and crank case pressures set forth, will exist. It may be seen that exact axial balance is obtained and that any fractional part of the discharge pressure that may be desired can be maintained within the crank case by properly proportioning the cross-section areas of the valve at its ends.

The valve is balanced against lateral (or radial) hydraulic thrust, which results from the fact that different pressures exist in the opposite ports 36 at any one time, by opposed balancing areas subjected to pressures which neutralize the thrust.

Referring now more particularly to Figs. 1, 4, 5 and 6, it may be seen that the distribution valve 38 is provided with a pair of balance recesses 140 and 142 oppositely located with respect to each other and communicating with the discharge and suction chambers respectively, by passages 144 and 146, as best shown in Figs. 4 and 6. To equalize the pressures over the bearing area nearest the discharge chamber 32, an annular groove 148 is utilized. (See Figs. 1 and 6.) The areas of the balance recesses may be made equal on opposite halves of the valve and when this equality is attained there can be no unbalanced lateral pressures tending to force the valve to one side or the other at any suction or delivery pressures.

Proper dynamic balance of the pump, in part at least, is obtained by making the large end of the valve heavier on the side opposite the high point of the eccentric. This is accomplished by cutting away a portion of the valve end to provide the substantially semicircular groove 62 which lies, as may be seen from Fig. 4 and a comparison of this figure with Fig. 1, at the same side of the shaft as the high point of the eccentric. It may be noted that the balance weight afforded by the valve construction is symmetrically located with respect to the plane of the cylinders.

In the embodiment illustrated, if the enlargement of the valve is not sufficient to compensate fully for the dynamic balance, additional compensation may be effected by mounting a balance weight 150 to the driving pulley 24.

Before proceeding with a detailed description of the operation of the pump and the modifications of the pulsation compensating means described above, it is deemed best to describe a further modification of the compensating means. In the compensating means heretofore described, the discharge chamber is connected periodically to the suction chamber, but equally good results may be obtained by connecting the suction passage to the discharge passage and utilizing a separate valve driven synchronously with the pump for the purpose of periodically connecting the two passages, thereby to eliminate the undesirable pulsations. A construction of this nature is illustrated in Figs. 11, 12, and 13.

The pulsation compensating valve, indicated generally by reference character 152, is adapted periodically to connect the discharge and suction passages 154 and 156 through a pair of small diameter tubes 158 and 160, thereby to by-pass an amount of the pumped fluid to eliminate the pulsations in the pump delivery. The valve may be, and is shown, mounted separately from the pump upon a standard 162, to which it is suitably secured, as by bolts.

The valve is rotated in synchronism with the pump shaft and distribution valve by a pair of gears 164 and 166, the former of which is mounted on the pump shaft and the latter affixed to a shaft 168 one end of which forms the moving part of the valve.

The valve proper comprises a valve chamber 170 within which rotates the aforesaid end of shaft 168. This end of the shaft is provided with opposed grooves 172 interconnected by a small diameter hole 174. The grooves 172 are of a shape to form part of the shaft with a substantially elliptical cross-section, as indicated by the reference character 176 in Fig. 13, and the hole 174 extends through the short diameter portion of the ellipse. The valve chamber is connected to the tubes 158 and 160 by small holes 178 and 180 and nipples 182 and 184, respectively.

Leakage of fluid from the valve along the shaft 168 is prevented by a gasket 186 held in place by an adjustable clamping member 188.

In order to compensate for the pulsations of the type illustrated in the chart of Fig. 7, it is necessary that the rotary portion of the valve rotate at a speed seven times that of the pump shaft because there are fourteen ripples per revolution of the shaft and the valve is so constructed that the discharge passage is connected to the suction passage twice per revolution of the rotary portion of the valve.

The shape of the valve is so designed as to provide a leakage that corresponds substantially to the shape of the ripples so that the ripples are substantially entirely eliminated to give a uniform pump output, the elliptical cross-section portion 176 providing for the gradual increase and decrease in the amount of the fluid by-passed.

Reviewing now the operation of the pump and the modifications thereof described above, it is assumed first that the pump is driven at a constant speed from a suitable prime mover connected to the pulley 24. Rotation of the pump shaft effects reciprocation of the seven pistons of the pump and the cylinders are selectively connected to the suction and discharge chambers by means of the rotary distribution valve in a manner that it is believed to be clear from the preceding description.

The pump, if not provided with the pulsation compensating means, would have an output of the character indicated by reference character 8 in Fig. 7. However, each time that a piston reaches its maximum velocity position whereat the pump output is a maximum, the discharge chamber is connected to the suction chamber through a by-pass passage. However, as the output of the pump increases from a minimum value to a maximum value, a gradually increasing amount of the output is by-passed from the discharge to the suction chamber, the amount being just enough to compensate for the increase above

the minimum value. When the output decreases from the maximum to the minimum value a gradually decreasing amount of the output is by-passed, the amount being just enough to compensate for the excess of the output over the minimum value. As fully explained above the amount of the by-pass is controllable by a manual adjustable needle valve and by the size and shape of the various component parts of the by-pass passage.

A pump of light construction and one that operates in noiseless manner is obtained by maintaining a reduced but positive pressure within the crank case by connecting the latter to the discharge and suction chambers through orifices of different cross-section areas.

The rotary distribution valve is hydraulically balanced against axial and lateral pressures. It is balanced against axial pressures by connecting the portions thereof subject to the pressures in the delivery and suction chambers 32 and 34 to the interior of the crank case through the variable area orifice 64—68 and the constant area orifice 60 and by constructing the said portions of the valve with predetermined areas. The various orifices and the balancing action have been described in the preceding portion of the specification and it is not deemed necessary that the description be repeated at this point. The lateral hydraulic balance is effected by providing balance recesses 140 and 142 connected by the passages 144 and 146 to the discharge and suction chambers of the pump, respectively. To effect further balancing, a portion of the valve is provided with the annular groove 148 to equalize pressures thereabout.

The pump is dynamically balanced by making the rotary distribution valve heavier on the side opposite the high point of the eccentric 42 and by adding a suitable balance weight 150 to the pulley 24 also as described above.

All the rotating parts of the pump are readily lubricated by the pump fluid itself which possesses lubricating qualities. The interior of the crank case is filled with the fluid and consequently all the bearings and bearing surfaces are lubricated. The eccentric ring is lubricated through the passage 56 and the flat 58 provided upon the surface of the eccentric 42.

If the pump is constructed so as to have an output of the character illustrated in the chart of Fig. 8, then the construction of the rotary distribution valve is modified to conform with the construction illustrated in Fig. 9. This construction, it will be recalled, provides for the compensation of ripples having alternate minimum values that are different from each other.

The compensating means disclosed in Fig. 10 may be utilized in place of that disclosed in Figs. 1 to 6. The modification of Fig. 10 eliminates a number of drilling operations as the rotary valve requires only as many holes as there are cylinders and is preferable for this reason.

As indicated above, it is not necessary that the rotary distribution valve be constructed to provide the necessary by-passing of liquid from the discharge to the suction side to effect compensation of the pulsations. As a matter of fact, the valve may be entirely separate from the pump and a modification of this nature has been described above in connection with Fig. 11. The valve comprises a rotary valve member having a single passage therethrough and adapted to connect the discharge to the suction passage twice for every revolution of the valve. The valve is driven in synchronism with the pump shaft at a speed such that the discharge side is connected to the suction side of the pump the necessary number of times to effect the desired compensation.

Other modifications of the present invention will suggest themselves to those skilled in the art and it is contemplated that the above described embodiments are descriptive and not limitative in character.

What I claim as new and desire to secure by United States Letters Patent is as follows:

1. In combination, a pump having a crank case and pistons and cylinders relatively movable to each other therein, suction and discharge passages, valve means for alternately connecting the cylinders to the passages, and means including a restricted passageway of fixed area permanently and continuously connecting the interior of the crank case to the discharge passage and means providing a restricted and variable area passageway from the interior of the crank case to the suction passage for maintaining a reduced positive pressure within the crank case.

2. A pump of the positive displacement type, including in combination, a crank case, pistons and cylinders mounted therein for relative movement with respect to each other, a discharge passage, a suction passage, a restricted passageway of fixed area permanently connecting the interior of the crank case to the discharge passage, and means providing restricted communication between the interior of the crank case and the suction passage whereby a reduced positive pressure is maintained in the interior of the crank case.

3. In combination, a pump having a crank case and pistons and cylinders movable relative to each other therein, suction and discharge passages, a rotary valve for alternately connecting the cylinders to the passages, and a restricted constant cross-section area passageway continuously connecting the interior of the crank case to the discharge passage and a variable area passageway continuously connecting the interior of the crank case to the suction passage when the pump is in operation, for maintaining a relatively low positive pressure within the interior of the crank case.

4. In combination, a radial pump including a crank case, relatively movable pistons and cylinders mounted therein and a centrally located rotatable driving shaft for effecting relative movement thereof, suction and discharge chambers, a rotary valve for alternately connecting the cylinders to the chambers, and restricted constant and variable area passageways both providing communication from one side of the valve to the other connecting the interior of the crank case to the discharge and suction chambers, respectively, for maintaining a relatively low but positive pressure within the crank case.

5. In combination, a pump having a crank case and pistons and cylinders relatively movable to each other therein, suction and discharge passages, a valve for alternately connecting the cylinders to the passages, said valve being axially movable and having an end portion open to a chamber formed as part of the pump, said valve also having portions subject to suction and discharge pressures, and restricted constant and variable area passageways connecting the said chamber to the discharge and suction passages, respec-

tively, whereby said valve is maintained in axial equilibrium.

6. In combination, a pump having a crank case and pistons and cylinders relatively movable to each other therein, suction and discharge passages, a valve for alternately connecting the cylinders to the passages, said valve being axially movable and having an end portion of said valve being open to a chamber formed as part of the pump, and restricted constant and variable area passageways connecting the said chamber to the discharge and suction passages, respectively, said variable area passage being constituted, in part, by the valve.

7. In combination, a pump having a crank case and pistons and cylinders relatively movable to each other therein, a rotatable shaft for effecting relative movement of the pistons and cylinders, suction and discharge chambers, a valve for alternately connecting the cylinders to the suction and discharge chambers, said valve being slidably movable between predetermined limits with respect to the shaft, and having surfaces open to both chambers and the interior of the crank case, a restricted passageway of fixed area from the discharge chamber to the interior of the crank case, and a variable area passageway from the suction chamber to the interior of the crank case, said last-mentioned passageway comprising a variable cross-section area opening defined by the slidably movable valve and shaft, the cross-section areas of the passageways and the surfaces open to the chambers and interior of the crank case being so proportioned that the valve is maintained in axial equilibrium between the limits of its movement and a positive pressure is maintained in the interior of the crank case.

8. In combination, a pump of the positive displacement type actuated by a rotatable shaft, suction and discharge chambers, a valve rotatable by and slidably movable with respect to the shaft between predetermined limits, a chamber facing and open to one end of the valve formed as part of the pump, said valve having surfaces open to all said chambers, a restricted passageway of fixed area leading from the discharge chamber to the last-mentioned chamber, and a variable area restricted passageway from the suction chamber to said last-mentioned chamber, said last-mentioned passageway comprising a variable cross-section area opening defined by the slidably mounted valve and the shaft, the cross-section areas of the passageways and the surface areas of the valve open to the chambers being so proportioned that the valve is maintained in axial equilibrium between the limits of its movement.

9. A positive displacement pump of the rotary type, including in combination, relatively movable pistons and cylinders, suction and discharge chambers, a valve for alternately connecting said cylinders to the suction and discharge chambers, said valve being axially movable between predetermined limits, and means including valve surfaces open to the discharge and suction chambers and crank case and restricted passageways interconnecting said chambers and case for maintaining a reduced positive pressure in the interior of the crank case and maintaining the valve in axial equilibrium between its limits of movement.

10. In a pump, the combination including, a plurality of relatively movable pistons and cylinders, suction and discharge passages, a rotary distribution valve having a pair of opposed ports for alternately connecting the cylinders to the suction and discharge passages, a bearing surface for the valve, opposed grooves of uniform depth and width on the surface of the valve adjacent the bearing surface, said grooves being substantially coextensive in length with said ports, and passageways from the discharge and suction passages to said grooves.

11. In a pump, the combination including a plurality of relatively movable pistons and cylinders, suction and discharge chamber, a rotary distribution valve having opposed substantially semi-circular ports for alternately connecting the cylinders to the suction and discharge chambers, a bearing surface for the valve, said valve having opposed grooves of uniform depth and width on the surface thereof adjacent the bearing surface, said grooves being substantially coextensive with the length of the ports, and passages extending through the valve from the ports to the grooves opposite thereto.

12. In a pump, the combination including, a plurality of relatively movable pistons and cylinders, suction and discharge chambers, a rotary distribution valve having limited axial movement for alternately connecting the cylinders to the suction and discharge chambers, and means including valve surfaces subject to pressures developed by the pump for maintaining said valve in axial equilibrium between the limit of its movement and balancing the radial pressures thereon.

13. A pump of the positive displacement type for fluids having lubricating qualities, including in combination, a crank case, pistons and cylinders mounted therein for relative movement with respect to each other, a shaft having an eccentric portion and a ring rotatably mounted thereon for effecting relative movement of the pistons with respect to the cylinders, said crank case having an interior portion opening to said relatively movable parts, means for supplying a continuous flow of fluid under pressure from the pump discharge to the interior of the crank case, and means including a passage from the interior of the crank case to a space between the eccentric and ring for lubricating the contact surfaces therebetween.

14. In combination, a pump having a crank case and pistons and cylinders relatively movable to each other therein, means including a shaft, for effecting relative movement between the pistons and cylinders, suction and discharge passages, a valve rotated by said shaft for alternately connecting the cylinders to said passages, and a restricted passageway of fixed area extending through the valve and permanently connecting the interior of the crank case to the discharge passage.

15. In a pump of the positive displacement type, in combination, a plurality of relatively reciprocable pumping cylinders and pistons, a crankcase, a discharge passage, an intake passage, a valve for connecting each of said cylinders with said passages alternately, said valve separating said discharge passage from the inside of said crankcase and being mounted for limited translational movement between said crankcase and said discharge passage so that the fluid pressure in said discharge passage tends to move said valve in one direction and fluid pressure in said crankcase tends to move said valve in the opposite direction, restricted passages providing for the flow of fluid from said discharge passage to said crankcase and the discharge of fluid therefrom, and means controlled by the movement of said valve for controlling the flow through said restricted passages to maintain a positive pressure

in said crankcase of a value to maintain said valve in equilibrium between its limits of movement.

16. In a pump, in combination, a crankcase, a plurality of radially disposed pumping cylinders mounted around said crankcase, a piston in each cylinder, a drive shaft passing through said crankcase for reciprocating the pistons in said cylinders, a suction chamber, a discharge chamber, a valve rotated by said shaft for connecting the cylinders with the chambers alternately, a portion of the valve being subjected to the pressure of fluid in the discharge chamber, a second portion of said valve being subjected to the pressure of fluid in the suction chamber and a third portion of the valve being subjected to the pressure of fluid in the crankcase, said pressures tending to move the valve axially, means providing for the flow of fluid from the discharge chamber through said crankcase, and means controlled by the axial movement of said valve for controlling the flow of fluid through said crankcase so that the axial force exerted upon the valve by the pressure of fluid in said crankcase, is equal and opposite to the resultant of the axial forces exerted upon the valve by the pressure of fluid in the discharge and suction chambers.

17. In a pump, the combination comprising a plurality of relatively movable pistons and cylinders, a discharge passage, an intake passage, a valve for connecting each of the cylinders with the passages alternately, a portion of the valve being subjected to the fluid pressure in the outlet passage, which pressure tends to move the valve along a line, a second portion of the valve being subjected to the fluid pressure in the inlet passage, which pressure also tends to move the valve along said line, means for subjecting a third portion of said valve to fluid pressure tending to move the valve along said line, and means controlled by the movement of said valve along said line to cause the fluid pressure on said third valve portion to exert a total force equal and opposite to the total resultant force exerted by the inlet and outlet passage pressures, so that the resultant of the fluid pressure forces tending to move the valve along said line is zero.

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