

April 15, 1941.

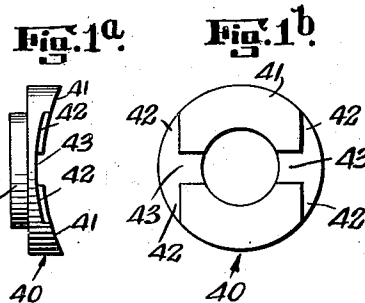
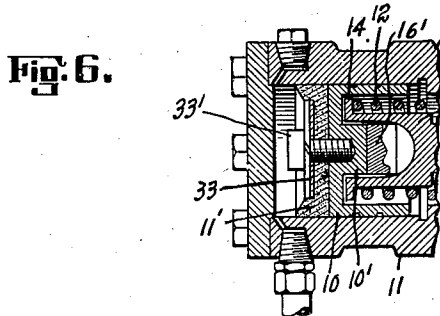
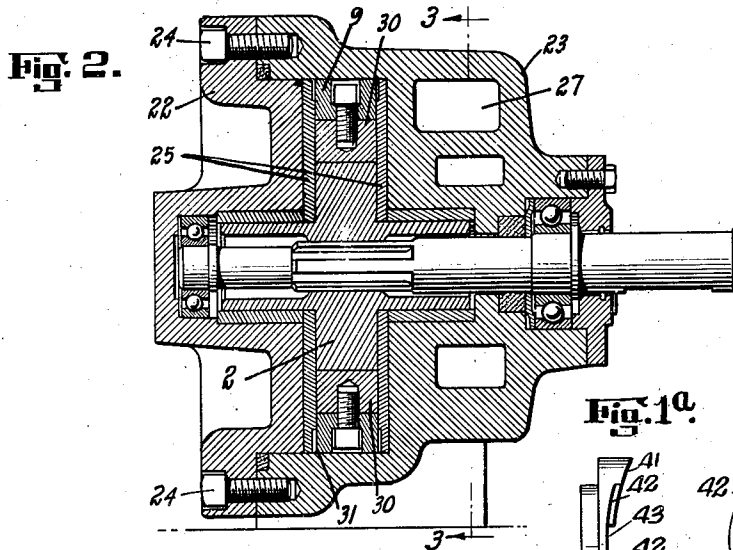
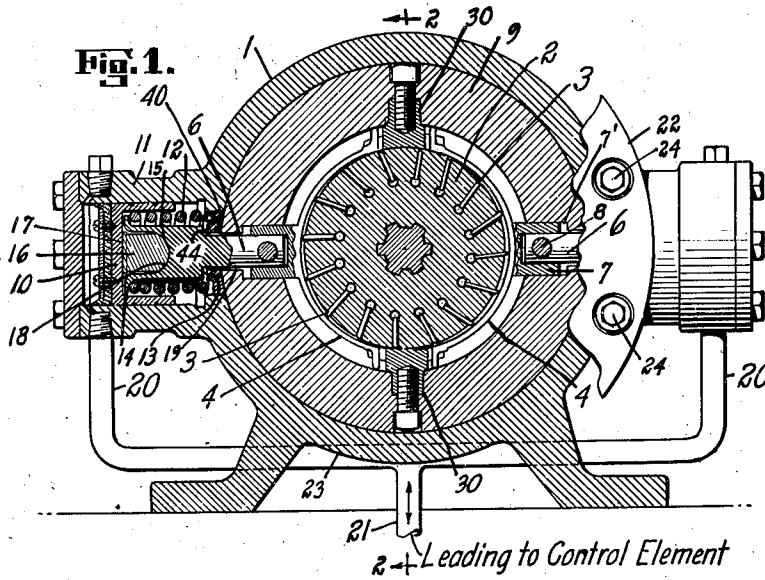
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2,238,062

VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

Filed Jan. 14, 1939

6 Sheets-Sheet 1



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VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

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6 Sheets-Sheet 2

Fig. 3.

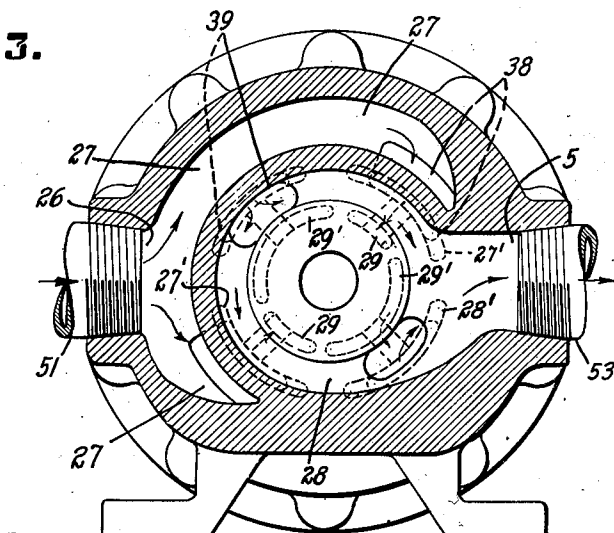


Fig. 4.

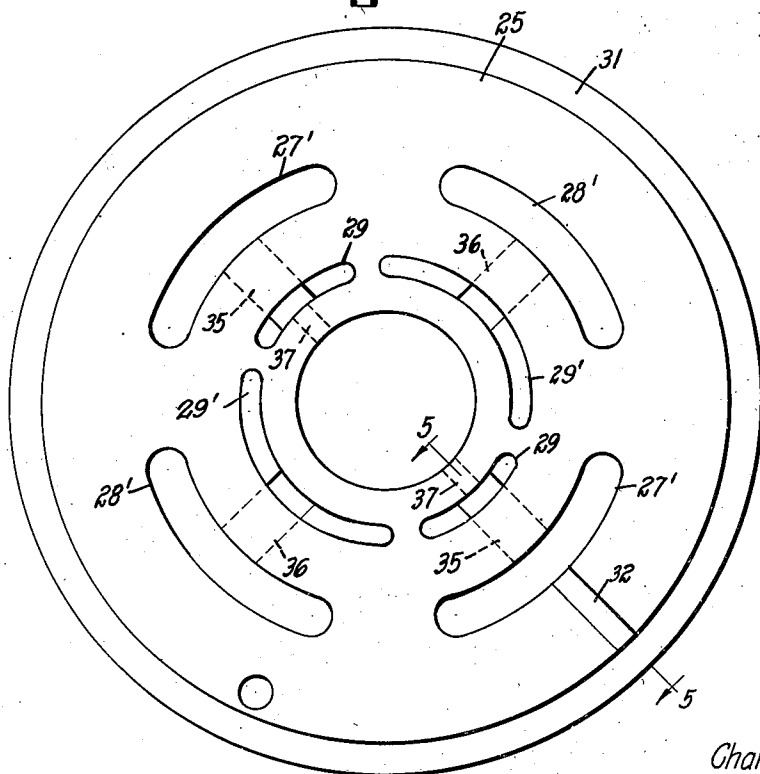
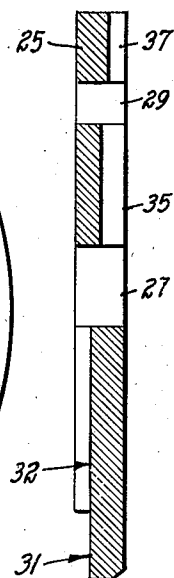


Fig. 5.



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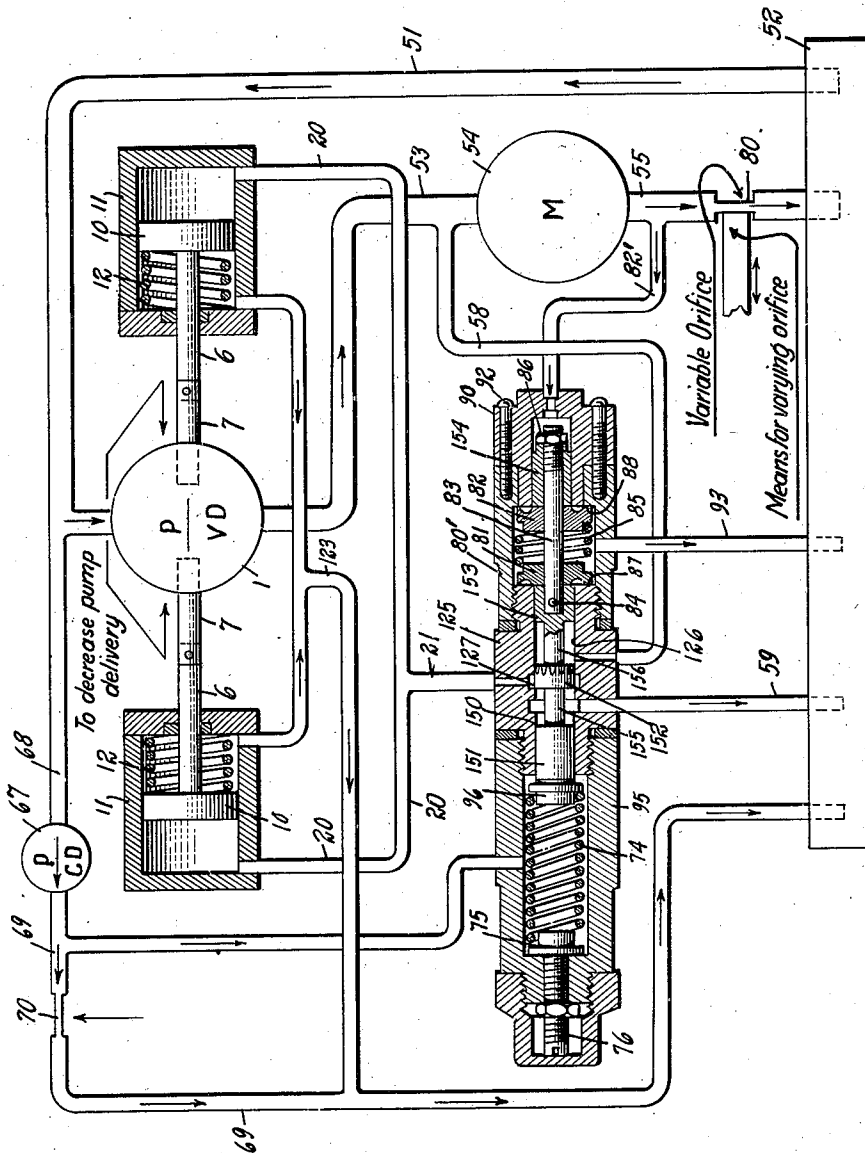
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VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

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Fig. 7.



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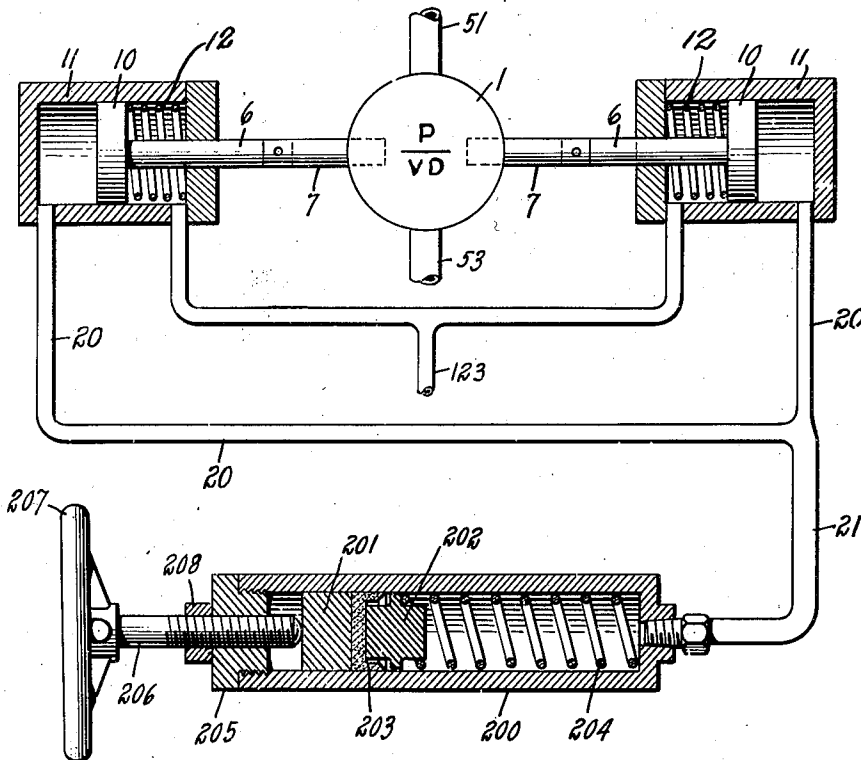
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VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

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6 Sheets-Sheet 4

Fig. 8.



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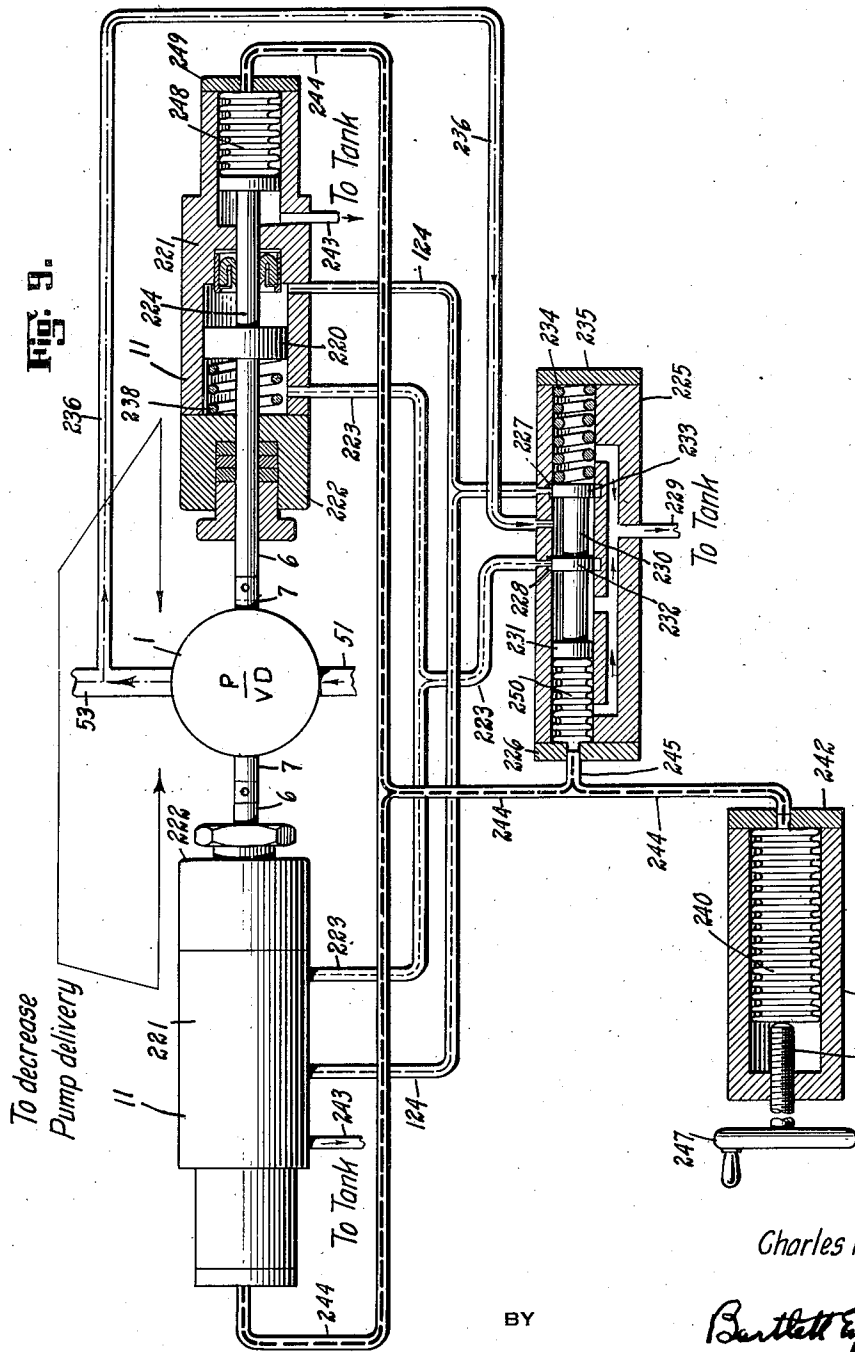
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VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

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6 Sheets-Sheet 5.



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VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

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Fig. 10.

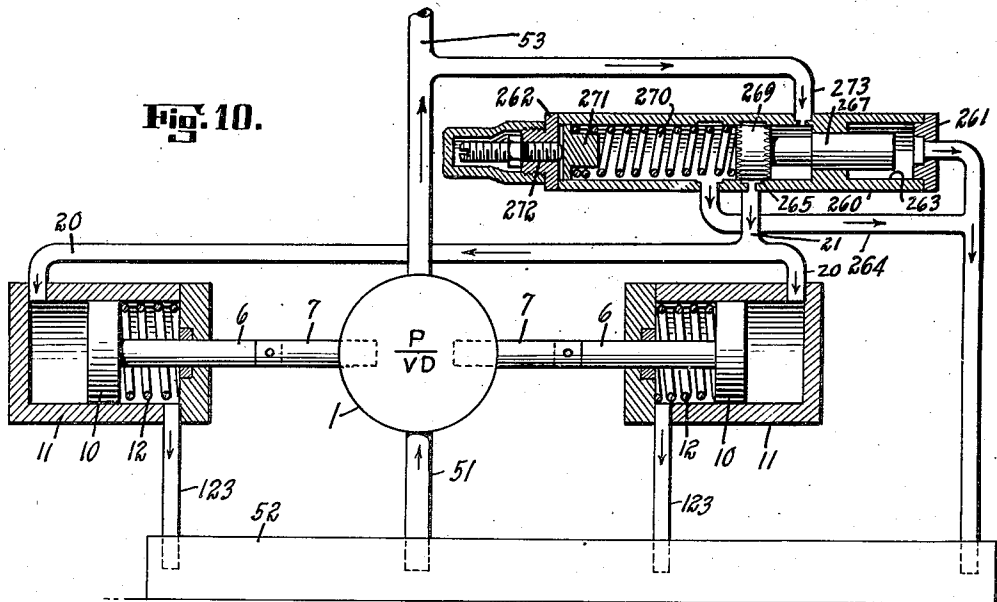
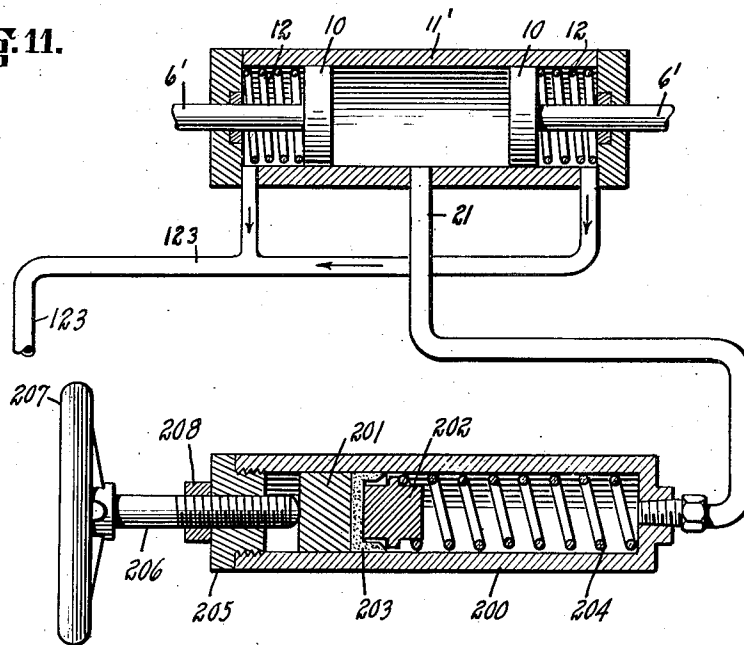


Fig. 11.



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UNITED STATES PATENT OFFICE

2,238,062

VARIABLE CAPACITY PUMP AND CONTROL THEREFOR

Charles M. Kendrick, New York, N. Y., assignor to Manly Corporation, Washington, D. C., a corporation of Delaware

Application January 14, 1939, Serial No. 250,864

28 Claims. (Cl. 103—38)

This application is a continuation in part of my co-pending application filed May 12, 1938, Serial No. 207,512.

That application for Letters Patent relates to systems for the transmission of power by means of fluid pressure, wherein fluid pressure operated devices are supplied with fluid under pressure by variable delivery pumps. The present invention in one aspect relates to variable delivery pumps or pump units and the control thereof, and more particularly to rotary piston pumps of the vane type having two or more sections or parts delivering fluid to a common discharge and whose capacity or output may be infinitely varied from minimum to maximum.

One object of the invention is a novel and improved variable delivery pump unit of the character indicated and control therefor, whereby a more certain, accurate and efficient control and operation of the unit is obtained.

A further object of the invention is a pump unit and control therefor of the character indicated which is characterized by its efficiency and flexibility in operation and its simplicity in construction and assembly.

A further object of the invention is a novel and improved pump and control of the character indicated including fluid pressure means for operating and flexibly interconnecting the separate control elements of the different pump unit sections and a single control means for said fluid pressure means.

Another object is to provide means for controlling the total fluid volume delivered by multiple pump sections or parts into their common discharge conduit without respect to the division of this total output between the individual pumping sections.

A further object is to provide a control of this character in which each section is provided with its individual fluid motor for actuating the delivery-determining element thereof to vary the output of its section, the operation of all of said fluid motors being under joint control of a common controlling element active to regulate the combined operation of all of said fluid motors independent of the extent of operation of any one of them.

A further object of the invention is a novel and improved fluid pressure system including a variable delivery pump unit and control therefor of the character indicated.

A further object of the invention is a pump unit and control of the above indicated character including a novel and improved fluid pressure follow-up control.

Other objects of the invention will hereinafter appear.

For a better understanding of the invention reference may be had to the accompanying draw-

ings forming a part of this application wherein:

Fig. 1 is a view, partly in section, of a pump unit and control embodying the invention;

Figs. 1a and 1b are views of an element of Fig. 1;

Fig. 2 is a sectional view taken at right angles to Fig. 1 along the line 2—2 thereof;

Fig. 3 is a sectional view indicating the pump unit chambers and circuits taken along the line 3—3 of Fig. 2;

Fig. 4 is an enlarged view of one of the plate elements of the pump unit showing the face thereof adjacent the rotor;

Fig. 5 is a sectional view to larger scale than Fig. 4 along the line 5—5 of Fig. 4;

Fig. 6 is a sectional view of a modified pump unit;

Fig. 7 is a diagrammatic view illustrating a fluid pressure system embodying the pump unit and control of the invention together with novel means for controlling the same;

Fig. 8 is a view of a manual control means for use with the pump unit and control of Fig. 1;

Fig. 9 illustrates a further modification embodying a novel, improved hydraulic follow-up;

Fig. 10 is a diagrammatic view illustrating another embodiment of the invention; and

Fig. 11 shows a modified form of pump unit and control.

Referring to Figs. 1 to 4 of the drawings, I have illustrated my invention as embodied in a rotary piston multi-section or part pump of the vane type. The pump, which is indicated generally by the numeral 1, includes a rotor 2 which is provided with a multiplicity of vanes 3. The vanes are movable inwardly and outwardly as the rotor rotates with their outer edges following adjustable tracks indicated generally at 4. In the particular embodiment shown the pump is a double pump comprising two pumping sections or halves which discharge into a common discharge 5. The particular structure of the rotor 2, the vanes 3 and the track 4 forms no part of the present application but a part of my co-pending application, original filed December 24, 1938, Serial No. 247,586. It is understood, however, that the structure of these pump parts may assume any suitable form of double or multiple pump of the general character indicated, such, for example, as that of either of the structures disclosed in United States Patents 2,141,270 and 2,141,171, issued December 27, 1938.

The capacity or output per revolution of either pumping section may be varied from minimum to maximum by the lateral or radial movement of its delivery determining element or adjusting rod 6, which is suitably attached to a radially slidable member 7 carrying the adjustable track 4 as, for example, by means of a pivot pin 8. The member 7 is slidably mounted in a recess or opening 7' formed in a spacer ring 9 on the interior of the

casing. In the embodiment shown the output of either pumping section is increased by movement of its delivery determining element 6 in an outward direction with respect to the axis of the rotor and the output is decreased by the inward movement of this element.

Each of the delivery determining elements 6 is actuated or moved in an inward direction by a fluid pressure operated piston 10 moving in a cylinder 11 which is formed, in the particular embodiment shown, as a part of the casing of the pump. The fluid pressure tending to move the piston 10 inwardly is opposed by the fluid pressure inside of the pump tending to move the track 4 and the delivery-determining element 6 outwardly, and is also opposed by a coil spring 12, the inner end of which rests upon a seat on a stop ring 40 and the outer end of which engages a flange 14 formed on the enlarged outer end 15 of the delivery-determining member 6. The enlarged part 15 of the delivery-determining element and the coil spring 12 extend up into the sleeve of the piston 10 and a bearing member 16 is provided between the piston and the delivery-determining element having a flat outer surface 17 in engagement with the piston and an inner spherical surface 18 in engagement with a correspondingly curved surface at the bottom of a recess formed centrally of the enlarged part 15. A clearance is provided between the side walls of the bearing member 16 and the side walls of the recess, as indicated. In the particular embodiment shown a clearance 19 is provided between the delivery element 6 and the walls of the opening through the spacer ring 9 and the stop ring 40 to avoid binding in the operation of the delivery-determining element, and further to maintain communication between the inner part of the cylinder 11 and the recess 7' in which the member 7 slides.

The outer ends of the cylinders 11 are connected through a communicating passage or pipe system 20 containing an incompressible fluid such, for example, as oil, etc. In order to control the pump delivery per revolution incompressible fluid is admitted to or withdrawn from the communicating passage 20 and the cylinders 11. This admission and exhaust of fluid for this purpose is effected through a branch pipe or passage 21 which leads to either a manual or other control, as will hereinafter be described more particularly.

The piston 10 and corresponding delivery-determining element 6 of each of the pumping sections is subjected on one side, through the hydraulic passage 20 and the other piston, to the fluid pressure generated in the other pump section, and on the other side to the pressure being generated by its pump section and the spring 12, the piston and the delivery determining element being balanced and floating between these forces.

Thus, by the construction and arrangement shown, each of the pistons 10 and its corresponding delivery-determining element 6 is at all times floatingly balanced between the pressures generated in the two halves or sections of the pump and each is independently movable to assume the balanced condition or position, without binding or undue strains upon any of the parts. The delivery per revolution may be infinitely and accurately varied and controlled through the simple mechanism shown, merely by admitting fluid pressure to or exhausting it from the common passage 20 leading to the outer ends of the cylinders 11 disposed at the diametrically opposed

points of the casing. This admission and exhaust is under the control of a single common mechanism which, as indicated, may be either manually or automatically manipulated.

In the particular embodiment of pump structure shown (Figs. 1, 2, 3, 4 and 5) the pump casing is in two axially separable parts 22 and 23, being suitably fastened together as by screws 24, and a pair of mating end plates 25 are disposed on the opposite sides of the rotor and between the rotor and the casing parts, the end plates being axially positioned by the spacer ring 9. The intake conduit 51 is suitably connected with the pump's intake 26 which branches to form an intake channel or chamber 27 in the pump casing for each half or pumping section of the pump. Each intake channel 27 is connected by an appropriate passage and port, as indicated partly by the dotted lines at 38 in Fig. 3, with the corresponding arcuate intake port 27' formed in the end plate 25 adjacent the side wall of the casing 23. The space between adjacent vanes is filled with fluid as said vanes sweep across the intake ports 27' and this fluid is discharged as the vanes move past the arcuate outlet port 28' of each pumping section; the work of pumping is accomplished as the vanes move across the "pumping arc" on the radially inner end of each of the slidable members 7. Fluid thus discharged into the arcuate ports 28' passes through appropriate passages and ports in the casing, which are likewise indicated in Fig. 3 partly in dotted lines at 39, into the common annular discharge channel 28 which in turn is suitably connected with the pump outlet 5 into the discharge conduit 53. Each half or pumping section of the pump thus comprises a separate pumping unit having its own intake and discharge areas, but both halves discharge their pumped fluid into a common discharge conduit and preferably take their supply of fluid from a common inlet conduit as shown. It will be understood that the volume of fluid pumped by each pumping section is dependent upon the distance between the periphery of the rotor and the "pumping arc" on the corresponding slidable member 7, so that the output of each half or pumping section may be altered at will. It will also be understood that the total output of the pump comprises the combined outputs of its two halves or pumping sections.

The end plates 25 are also provided with two pairs of arcuate vane slot ports 29 and 29' respectively which are adapted to register successively with the inner ends of the vane slots as the rotor revolves. The vane slot ports 29 are connected with the arcuate ports 27' of the corresponding end plate 25, as by radial grooves 35 formed on the outer faces of said end plates, and similarly the vane slot ports 29' are connected as by radial grooves 36 with the outlet ports 28' of said end plates 25. The arrangement is such that the inner ends of the vanes are supplied with fluid having the same pressure as that acting on the exposed outer ends thereof while said vanes are passing across the intake ports 27' and the outlet ports 28' of the two halves or pumping sections, so that said vanes are hydraulically balanced. The vane slot ports 29' are preferably made of such length, however, that the fluid from the outlet ports 28' is supplied to the inner ends of the vanes during the time said vanes are moving in contact with the "pumping arcs" and the vane track surfaces on the radially inner ends of the fixed abutments 30. The vane slot ports 29

are also connected as by radial passages 37 with the holes at the center of said end plates through which the rotor hubs pass to provide return passages for leakage. Diametrically opposed abutments 30 separate the two halves of the pump from each other in the conventional manner. At least one of the end plates 25 is rabbeted on its inner side around its peripheral edge to form an annular chamber as indicated at 31, and a radially disposed groove or channel 32 is formed to connect the annular chamber thus formed with one of the suction ports 27'. The chamber 31 overlaps slightly the recesses 7' formed in the lining member 9. Each of the cylinders 11 is thus in communication at its inner ends at all times with the suction port for the purpose of keeping the pressure of the fluid at the inner ends of the cylinders at a relatively low value substantially equal to the pressure of the fluid in the intake ports 27' and also for the purpose of providing means by which fluid leaking past the slidable members 7 is returned to the intake port.

As above indicated, each stop ring 40 is provided with a clearance hole at its center (Figs. 1a and 1b) through which passes the corresponding delivery-determining element 6. To facilitate manufacture of the parts, the stop ring 40 is also preferably formed with a cylindrical outer contour and is located in an appropriate bore formed in the casing member 23 as shown in Fig. 1. Each stop ring 40 is adapted to bear against and be positioned upon the outer cylindrical surface of the spacer ring 9 and is accordingly provided with an arcuate surface 41 (Fig. 1b) having the same curvature as that of the periphery of said spacer ring. This arcuate surface 41 is preferably of the same width as that of the spacer ring 9 in order that the stop ring 40 may be positioned solely thereon and for the added purpose of preventing any contact between said stop ring and the end plates 25 which might distort or displace the latter; outer areas, adjacent the arcuate surface 41, are accordingly relieved as indicated at 42 in Fig. 1b. A transverse groove or passage 43 is also provided to connect the relieved areas 42 with the clearance hole at the center of the stop ring in order to assure fluid connection at all times between said clearance hole and the recesses 31 of the end plates 25. As already stated, the stop ring 40 forms a seat for the inner end of its corresponding coil spring 12, and the spring 12 thus holds the stop ring 40 in proper alignment with the other parts, which alignment is facilitated by the ability of the stop ring to adjust itself circumferentially with respect to the outer surface of the spacer ring 9.

The stop rings 40 perform still another important function, from which they take their name, that is to say, each stop ring limits the maximum inward movement of its corresponding delivery-determining element 6 and hence limits the extreme inward position of the corresponding slidable member 7, thereby limiting and determining the minimum output of the corresponding pumping section. Each stop ring 40 is accordingly provided with a surface 44 (Fig. 1a) adapted to contact the shoulder 13 on the enlarged end of its delivery-determining element 6 to thus limit the maximum inward movement thereof.

The construction and co-operating arrangement of the adjusting pistons 10, bearing members 16, delivery-determining elements or adjusting rods 6, stop rings 40, springs 12 and 75

spacer ring 9 have outstanding advantages. As already stated, the stop ring is free to adjust itself upon the spacer ring 9 in alignment with the other parts, thereby decreasing any tendency toward "cocking action" or binding. Use of the stop ring to determine maximum inward position of the corresponding delivery-determining element 6 not only provides a simple yet accurate means of accomplishing this result but further makes it possible to alter the maximum possible inward position of said delivery-determining element 6 (thereby correspondingly changing the minimum output of the pumping section) by merely substituting another stop ring of different thickness. The free-floating or self-aligning relation between each piston 10 and its corresponding delivery-determining element 6 provides greatly reduced cost as compared with rigid or non-self-adjusting constructions or relationships; with the construction as described, errors in manufacture do not cause binding of the parts which are moved to alter the pump's output so that it is not necessary to work to precise dimensions, whereas without this construction and arrangement all internal parts of the pump must be held to very precise dimensions and the slideways 7' of the spacer ring 9, delivery-determining elements 6, vane tracks 4, etc. must be precisely aligned in every plane with respect to the bore of the adjusting cylinders 11. In fact, this construction and arrangement, or its equivalent, are regarded as of great commercial importance in a pump of this type as they eliminate the cost of the extremely precise manufacture otherwise required.

Fig. 6 shows a modification of the fluid pressure cylinder construction wherein the piston 10 is provided with a center projection 10' extending down into the sleeve a short distance, against which abuts the bearing members 16'. The packing 11' is secured to the end of the piston by means of a washer 33 and a single centrally disposed bolt 33'.

As already indicated, the simple structure previously described enables control of the pump's output to be effected by regulating the volume of fluid admitted to or withdrawn from the branch pipe or passage 21. This admits of the use of a wide variety of controls for regulating the admission of fluid through or withdrawal of fluid from the passage 21, and in order that this may be fully understood the invention is illustrated in connection with controls in which such admission or withdrawal are effected entirely by manual means, and in connection with controls in which such admission or withdrawal are effected by difference between the pressure of the fluid in the passage 20 and in other parts of the fluid system and in which such withdrawal or admission are regulated either by manual means alone, or by hydraulically-actuated mechanism in turn under control of manual means, or by hydraulically-actuated mechanism in turn under control of automatically-acting means.

In Fig. 7, for example, I have shown my invention as embodied in a fluid pressure system including a fluid pressure operated device M, a multiple or double section pump 1, fluid pressure adjusting cylinders 11 for adjusting the outputs of the pumping sections, a single element 152 hydraulically-actuated to regulate the withdrawal or admission of fluid to or from the passage 21 under influence of pressure differences in the passage 20 and elsewhere in the fluid system, a variable orifice 80 which may be opened or closed, either manually or otherwise, to cause regulating

movement of the element 152, and a fixed orifice 70 for maintaining a predetermined fixed output corrected for changes in viscosity.

The pump 1 of Fig. 7 is similar to that of Figs. 1, 2, 3, 4 and 5 but it is somewhat schematically illustrated for convenience. A portion of its discharge conduit 53 leads to the fluid motor M and fluid discharged by said fluid motor passes into the reservoir 52 through the portion 55 of said discharge conduit. With this arrangement, motive fluid delivered by the pump 1 operates the fluid motor and the operation of the latter is controlled by varying the output of said pump 1.

The capacity or output of each pumping section may be infinitely varied from minimum to maximum by lateral movement of its delivery determining element or adjusting rod 6 as described above. Each adjusting rod 6 is independently movable to regulate and control a portion of the total output of the pump 1 that is to control the output of its pumping section. For example, if the output of the pump 1 is 20 gallons per minute at a particular speed, each adjusting rod 6 would be independently movable to control an output of 10 gallons per minute if the total maximum output of the pump 1 is equally divided between its two pumping sections.

The valve mechanism illustrated in Fig. 7 includes a valve piston 150 slidably fitted within the valve bore 126 of the valve housing 125 and having three heads 151, 152 and 153 respectively, separated by the reduced portions 155 and 156. The head 152 covers the cylinder port 127 when the valve piston 150 is in its neutral position in which it is shown in Fig. 7. Movement of the valve piston 150 to the right of its neutral position connects the cylinder port 127 with the portion of the valve bore surrounding the valve piston's reduced portion 155, so that fluid may be exhausted from the passage 20 and the outer ends of the adjusting cylinders 11, such exhausted fluid returning to the reservoir 52 through the passage 59. The adjusting pistons 10 move outward under the influence of the springs 12 and pump pressure when the outer ends of the adjusting cylinders 11 and passage 20 are thus connected with the exhaust, as already explained.

Movement of the valve piston 150 to the left of its neutral position connects the cylinder port 127 with the portion of the valve bore 126 surrounding the valve piston's reduced portion 156; pressure fluid supplied to this portion of the valve bore 126, as through the passage 58, may then pass around the reduced portion 156, through the cylinder port 127 and the branched passage 21 to the common passage 20 and outer ends of the adjusting cylinders 11, causing the adjusting pistons 10 to move in an inward or delivery decreasing direction. In this connection it may be pointed out that pressure of fluid in the passage 20 will always be less than the pressure of the fluid in the discharge conduit 53 except at extremely low pressures; this is due to the fact that the areas of the adjusting pistons 10 exposed to pressure of fluid in the adjusting cylinders 11 are greater than the areas of the inner ends of the slidable members 7 which are exposed to pump pressure. The valve head 152 is shown as provided with a number of conventional V-notches on its end adjacent the reduced portion 156, these notches providing a relatively gradual connection between the reduced portion 156 and the cylinder port 127 when the valve piston 150 moves to the left of its neutral position, thus re-

ducing any tendency toward abrupt movement of the adjusting pistons 10.

In some instances it has been found desirable to provide a centering mechanism for the valve piston to assure its proper location with respect to the other parts when said valve piston is in its neutral position and for the further purpose of reducing or eliminating any tendency toward "hunting" or "surging" that may be present. A centering mechanism suitable for this purpose and of conventional type is illustrated in Fig. 7.

As here shown, the centering mechanism is located in the bore of a housing 80' which is suitably attached to the right hand end of the valve housing 125, as by threaded engagement therewith. The outer end of the housing 80' is in turn closed by an auxiliary valve housing 90, the purpose of which will be later explained, and the two parts are appropriately fastened together, as by the screws 92. It is preferable that there be substantially no pressure of any fluid that may enter the bore of the housing 80, as through leakage, and a passage 93 leading to the reservoir 52 is accordingly connected with said bore.

The centering mechanism includes a pair of washers or annular members 81 and 82 freely slidable upon a rod 83 which is of smaller diameter than that of the valve piston 150 to which it is securely attached as by the pin 84. A relatively light compression spring 85 is positioned between the two washers 81 and 82 and urges them toward the stops or seats 87 and 88 formed respectively by the end of the valve housing 125 and the reduced end portion of the bore of the housing 80'. The arrangement is such that when the valve piston 150 is in its neutral position the washer 81 is in contact both with its seat 87 and with the adjacent end of the valve piston 150; and similarly, the washer 82 is then in contact with its seat 88 and is also in contact with the adjacent end of the auxiliary valve piston 154 which is carried by the rod 83 and is positioned with respect thereto by the nut 86 on the threaded end of said rod 83.

The washers 81 and 82 thus move toward each other, upon further compression of the spring 85, and one or the other of them is so moved whenever the valve piston 150 moves out of its neutral position. In other words, when the valve piston 150 moves toward the right of its neutral position the end of said valve piston 150 will move the washer 81 toward the right and away from its seat 88, the rod 83 sliding through the hole at the center of said washer 82. Similarly, when the valve piston 150 moves to the left of its neutral position, the end of the auxiliary valve piston 154 moves the washer 82 toward the left and away from its seat 88, the washer 81 remaining against its seat 87 as the rod 83 slides through the hole at its center. In this manner the centering mechanism tends to maintain the valve piston 150 in its neutral position and presents a relatively slight increased resistance to displacement of said valve piston in either direction.

The valve mechanism of Fig. 7 also includes an auxiliary valve bore formed in the auxiliary valve housing 90 and of the same diameter as the valve bore 126, into which the auxiliary valve piston 154 is slidably fitted. The auxiliary valve bore is thus equivalent to a continuation of the valve bore 126 and the auxiliary valve piston 154 is likewise equivalent to an additional head on the valve piston 150. The valve piston 150, rod 83 and auxiliary valve piston 154 may thus be said to comprise the valve piston assembly.

The left hand end of the valve piston 150 projects beyond the end of the valve housing 125 and into the bore of a housing 95 which is attached to the left hand end of the valve housing 125 by threaded engagement therewith. Also positioned in the bore of the housing 95 is a spring 74 having abutment pieces 75 and 96 respectively on each of its ends. The abutment piece 96 bears against the adjacent end of the valve piston 150 and the abutment piece 75 bears against the end of the screw 76 which extends through the closed end of the housing 95 and provides means for adjusting the compression of the spring 74. As will be readily understood, the spring 74 exerts a force upon the valve piston 150 tending to move the valve piston assembly toward the right as viewed in Fig. 7 and the abutment piece 96, which is of larger diameter than that of the valve bore 126, limits the maximum distance in this direction to which said valve piston assembly can be moved by said spring 74.

The position and movement of the valve piston assembly, by which the output of the pump 1 is regulated, are determined and effected by the actual drop existing across the metering orifice 80 relative to a predetermined pressure drop thereacross. The orifice 80 is here shown as positioned in the portion 55 of the discharge conduit and the end of the auxiliary valve housing 90 is accordingly connected, as by the passage 82' with said portion 55 of said discharge conduit at a point on the inlet side of said orifice 80. It has been found that when the metering orifice 80 is positioned at a point in the discharge conduit intermediate the driven member 54 and the reservoir 52 the pressure of the fluid on the outlet side of the orifice 80 is so small and subject to such minor variations that in practice it may frequently be neglected. In other words, the actual amount of the pressure existing on the inlet side of the orifice 80 may alone be practically employed in many instances as the measure of the pressure drop across said orifice and hence may be alone employed in measuring the rate of fluid flow therethrough. The outlet side of the orifice 80 is therefore not connected with the valve mechanism in the arrangement of Fig. 7. The force exerted upon the valve piston assembly by the action of the pressure fluid from the inlet side of the orifice 80 upon the end of the auxiliary valve piston 154 is thus opposed only by the force of the spring 74 when the valve mechanism is employed without viscosity compensating means. This arrangement has the advantage of simplifying the structure and of reducing the necessary fluid connections.

With the pump 1 continuously driven, the valve mechanism of Fig. 7 functions to jointly control movement of the two adjusting pistons 10 whereby a definite output of said pump 1 is maintained for each adjustment or extent of opening of the variable metering orifice 80. This is accomplished through use of the pressure drop across the orifice 80 to measure the rate of fluid flow therethrough, the pressure existing on the inlet side of said orifice 80 being alone employed for this purpose in the present instance as hereinbefore stated. In Fig. 7 the valve piston assembly is shown in its neutral position in which the head 152 closes the cylinder port 127, thereby completely cutting off communication between the passage 20 connecting the outer ends of the adjusting cylinders 11 and the valve bore 126. The parts will remain in the position as

shown whenever the output of the pump 1 is such that it produces a pressure on the inlet side of the orifice 80 of such an amount that the force exerted thereby upon the auxiliary valve piston 154 equals the opposing force exerted by the spring 74. With the viscosity of the fluid considered constant or neglected, it will be understood that the pressure existing on the inlet side of the orifice 80 (for any setting or adjustment of said orifice 80 and of the compression of the spring 74) will depend entirely upon the rate of fluid flow therethrough and hence will depend entirely upon the output of the pump 1. It will also be understood that the valve piston assembly will remain in its neutral position as long as the output of the pump 1 remains constant at an amount producing the pressure on the inlet side of the orifice 80 required to balance the opposing force exerted by the spring 74.

Increase in the output of the pump 1, as because of increase in its speed or for any other reason, will of course produce an increase in the pressure existing on the inlet side of the orifice 80. This increase in pressure will be immediately communicated to the end of the bore in the auxiliary valve housing 90 where it will act upon the auxiliary valve piston 154, causing the valve piston assembly to move toward the left as viewed in Fig. 7. This movement of the valve piston assembly connects the cylinder port 127 with the portion of the valve bore 126 surrounding the reduced portion 156 of the valve piston; pressure fluid from the passage 58 is thus admitted to the outer ends of the adjusting cylinders 11, forcing the adjusting pistons 10 inward or in a delivery-decreasing direction and thereby reducing the output of the pump 1.

The pressure existing on the inlet side of the orifice 80 will, of course, decrease conformably with and immediately upon decrease in the output of the pump 1 responsive to the corrective action just described. Responsive to this decrease in pressure, the spring 74 will correspondingly and simultaneously move the valve piston assembly toward the right as viewed in Fig. 7. Decrease in the output of the pump 1 accompanied by corresponding movement of the valve piston assembly toward the right, will continue until the delivered volume is reduced to the exact amount producing the pressure on the inlet side of the orifice 80 as established by the spring 74 when the valve piston assembly will be restored to its neutral position and will again stabilize the adjusting pistons 10.

The operation of the mechanism is, of course, the reverse of that above explained when the output of the pump 1 is for any reason decreased below the amount at which the corresponding pressure existing on the inlet side of the orifice 80 balances the force exerted upon the valve piston assembly in its neutral position by the spring 74. Upon such decrease, the valve piston assembly will be displaced from its neutral position and moved toward the right by the spring 74. This movement of the valve piston assembly will connect the cylinder port 127 with the portion of the valve bore 126 surrounding the reduced portion 155 of the valve piston, and hence will permit emission of fluid from the passage 20 and the outer ends of the adjusting cylinders 11. The adjusting pistons 10 will immediately be moved outward or in a delivery-increasing direction, under influence of the pump pressure and the springs 12 as far and as rapidly as permitted by exhaust of fluid. This

movement of the adjusting pistons 10 will continue until the output of the pump 1 has been restored to the amount producing the pressure on the inlet side of the orifice 80 as determined by the spring 74, when the valve piston assembly will again be restored to its neutral position.

These corrective changes in the pump's output and restoring movements of valve piston assembly and the adjusting pistons 10 take place almost instantaneously, and the adjustments are such as to set the corrective mechanisms into operation upon slight departures from the output to be maintained.

From the foregoing it will be understood that there is only one rate of fluid flow through the orifice 80 (viscosity being neglected or considered as constant) that will produce a predetermined amount of pressure on the inlet side of the orifice 80 for any adjustment or extent of opening thereof, and hence there is only one output of the pump 1 which meets this requirement; that is to say, there is only one volume of fluid delivered by the pump 1 into the discharge conduit 53, regardless of pressure in the portion of said conduit 53 intermediate the pump 1 and the motor M. It will therefore be seen that the control mechanism of Fig. 7 functions to so control joint operation of the two adjusting pistons 10 that a predetermined output of the pump 1 is maintained for each particular setting or adjustment of the spring 74 and of the orifice 80, the actual output of the pump 1 being instantaneously altered to correct for any variations from this predetermined output. The output of the pump 1 is thus held substantially constant regardless of variations in operating conditions as long as the setting or adjustment of the spring 74 and of the orifice 80 remain unchanged. Moreover, this holds true for any adjustment of the orifice 80 to control the motor M, and there is an instant response to any such adjustment to change the pump's delivery accordingly which changed delivery is then maintained constant until a further variation in the size or extent of opening of the orifice 80 is made.

In the foregoing explanation the viscosity of the circulated fluid has been neglected or considered as constant. In practice, however, the viscosity of the fluid (usually oil) will change over a relatively wide range upon change in the temperature of the fluid and will noticeably affect the pressure on the inlet side of the orifice 80 resulting from a constant rate of fluid flow therethrough for any particular adjustment of extent of opening of the orifice 80. This change in pressure on the inlet side of the orifice 80, due to viscosity change, will in turn affect the output of the pump 1 unless compensation therefor is provided. Fig. 7, therefore, also illustrates viscosity compensating mechanism for this purpose.

The viscosity compensating mechanism includes a constant capacity pump 67 receiving its fluid supply through an inlet conduit 68 connected with the inlet conduit 51 which leads to the pump 1, so that fluid of the same viscosity is supplied to both of these pumps. The pump 67 is also provided with a discharge conduit 69 leading to the reservoir 52 and having a metering orifice 70. Fluid from the inlet side of the orifice 70 is admitted to the bore of the housing 95, which also serves as a compensating cylinder, where it acts, through the abutment piece 96, upon the exposed end of the valve piston 150 and thus supplements the force exerted by the spring 74. As in the case of the fluid pressure on the outlet side of

the orifice 80 of Fig. 7, it has likewise been found that the pressure on the outlet side of the orifice 70 may be neglected for most practical purposes. The pressure existing on the inlet side of the orifice 70 is accordingly alone employed as the measure of the pressure drop thereacross and the valve piston assembly is therefore not acted upon by fluid from the outlet side of said orifice 70.

Pressure existing on the inlet side of the orifice 70 will vary conformably with change in the viscosity of the circulated fluid. Compensation for viscosity change is thus effected by variation in the supplementary force upon the valve piston 150 exerted by the pressure fluid in the bore of the housing 95, the pressure of this fluid varying substantially with the pressure drop across the orifice 70 and substantially with the change in viscosity of the fluid as already stated. In this manner the amount of pressure drop to be maintained across the orifice 80 (which is determined by the spring 74 alone when viscosity compensating mechanism is not employed) is modified substantially in accordance with the effect of the change in viscosity of the fluid upon the pressure drop across the orifice 80 caused by a constant rate of fluid flow therethrough, so that the output of the pump 1 is unaffected by change in the viscosity of the circulated fluid.

The output of the pump 1 is preferably regulated by varying the adjustment or extent of opening of the variable orifice 80. In this manner the pump's output may be infinitely varied from maximum, when the orifice 80 is fully open, to a minimum such as zero when the orifice 80 is fully closed.

Maximum output of the pump 1 can, as hereinbefore stated, be obtained only when each of its two pumping sections delivers its maximum capacity or output, in which case both adjusting pistons 10 and their attached adjusting rods 6 occupy their extreme outward positions. The adjusting pistons 10 under influence of pump pressure and the spring 12, immediately move to these extreme outward positions immediately upon full opening of the variable orifice 80. Similarly, the minimum output of the pump 1 (which is preferably substantially zero) can be obtained only when both of the two pumping sections deliver their minimum outputs or capacities, in which case the adjusting pistons 10 and their attached adjusting rods 6 are in their extreme inward positions. Upon full closure of the variable orifice 80 the pressure fluid admitted to the outer ends of the adjusting cylinders 11 immediately moves both adjusting pistons 10 and their attached adjusting rods 6 into their extreme inward positions, as for example, against suitable stops, not shown, thus reducing the output of both pumping sections to minimum.

Since each adjusting piston 10 is independently movable, it will be seen that there are an almost infinite number of combinations of relative positions of said adjusting pistons 10 and the adjusting rods 6 capable of producing each output of the pump 1 intermediate its maximum and its minimum outputs. In other words, each intermediate output of the pump 1 may be divided between its two pumping sections in numerous proportions. In practice it has been found that, when maximum outputs of both pumping sections are substantially equal, this division between the two pumping sections is substantially equal and that both adjusting pistons 10 occupy substantially identical positions in their paths of inward and outward movement. Relative division of the

pump's output between its two pumping sections is not important, however, as the combined outputs will be held at the proper total amount irrespective of the proportion of this total contributed by each pumping section. For example, if the orifice 80 is adjusted to provide an output of 10 gallons per minute by the pump 1, this output will be maintained regardless of the amount contributed to it by each pumping section. It is immaterial therefore, whether this output of 10 gallons per minute comprises equal outputs of 5 gallons per minute by each pumping section, or whether it is comprised of 6 gallons per minute from one pumping section and 4 gallons per minute from the other, or whether it is comprised of any other combination of outputs which total 10 gallons per minute.

The output of the pump 1, comprising the combined outputs of its two pumping sections, may thus be accurately controlled throughout the entire range from maximum to minimum. This control, which is effected independent of definite control of the output of either of the two pumping sections and without control of the division of the combined outputs between the two sections (except, of course, at maximum and minimum outputs), is accomplished by a single control valve mechanism common to both adjusting cylinders 11 and which regulates the joint operation of both adjusting pistons 10 by regulating the fluid volume admitted jointly to or exhausted jointly from the passage 20 which connects the adjusting cylinders 11.

In Fig. 8 I have shown a manual means for controlling the admission and exhaust of the fluid to and from the passage 20 and cylinders 11. The passage 21 leads to and connects with the bore of a control master cylinder 200 within which is a piston 201 slidably fitted. The piston 201 is provided with a suitable packing 203 held in place by a member 202 abutting against one end of the compression spring 204. The other end of the spring 204 abuts against the closed end of the control master cylinder 200 and the arrangement is such that the spring 204 exerts force upon the member 203, packing 202 and piston 201 throughout the entire range of movement thereof in the bore of the control master cylinder. The outer ends of the adjusting cylinders 11, the branched passage 20, 21 and the connected portion of the bore of the master cylinder 200 thus form a closed fluid circuit which for convenience is termed the "adjusting circuit".

The piston 201 is adapted to be moved toward the right, as viewed in Fig. 7, by any suitable manually actuated means. In the present instance there is provided a screw 206 which threadedly engages and extends through the plug 205 which closes the open end of the master cylinder 200, and the inner end of said screw bears against the adjacent end of the piston 201. The outer end of the screw 206 is provided with a handwheel 207 by which it may be rotated and with a lock-nut 208 by which it may be fastened in any desired position.

The adjusting circuit is filled with a substantially non-compressible fluid, such for example as oil or glycerine. Rotation of the handwheel 207 in one direction will cause movement of the piston 201 toward the right, as viewed in Fig. 8, thereby decreasing the volumetric capacity of the portion of the bore of the master cylinder 200 which is connected with the branched passage 21. Fluid is thus displaced from the bore of the master cylinder 200 and enters the pas-

sage 20 and the outer ends of the adjusting cylinders 11, forcing the adjusting pistons 10 in an inward direction and thereby reducing the output of the pump 110. It will be seen that movement of the piston 201 through any given distance toward the right will cause displacement of a definite amount of fluid from the bore of the master cylinder 200. This fluid of necessity must enter the passage 20 and the outer ends of the adjusting cylinders 11 where it will cause joint or combined movement of the two adjusting pistons 10 in an inward or delivery-decreasing direction, sufficient to increase the combined volumetric capacities of the connected outer ends of said adjusting cylinders 11 to the exact extent of the decrease in the volumetric capacity of the connected portion of the bore of the master cylinder 200. Joint or combined inward movement of the two adjusting pistons 10 through a definite distance therefore results from each amount of decrease in the volumetric capacity of the connected portion of the master cylinder's bore, and hence the output of the pump 1 is likewise decreased by a definite and corresponding amount. Since the adjusting pistons 10 are of equal size and each pumping section is of equal capacity, it will be seen that division of this joint or combined movement between the two adjusting pistons 10 thus produced is immaterial. For example, if the two adjusting pistons 10 move substantially equal distances, as will usually be the case, it will be evident that the output of each pumping section will be approximately equally diminished. If, however, one adjusting piston 10 moves through a greater distance than the other, the output of its pumping section will be reduced below that of the other pumping section but the combined outputs of the two pumping sections will be the same irrespective of this unequal division.

Upon rotation of the handwheel 207 in the opposite direction the piston 201 will be moved toward the left as far as permitted by the inner end of the screw 206, this movement of the piston 201 taking place under influence of the spring 204 and of any fluid pressure that may exist in the bore of the master cylinder 200. An increase is thus brought about in the volumetric capacity of the portion of the bore of the master cylinder 200 which is connected with the branched passage 21. Immediately upon such increase the adjusting pistons 10 are moved outward or in a delivery-increasing direction by the pump pressures and by the springs 12 and the output of the pump 1 is thus increased. The joint or combined movement of the two adjusting pistons 10 is an amount sufficient to displace from the outer ends of the adjusting cylinders 11 fluid equal in volume to the increase in the volumetric capacity of the master cylinder 200 and a definite and corresponding increase in the output of the pump 1 results. It is immaterial whether the joint or combined movement of the two adjusting pistons 10 is equally divided between them as will be understood from the explanation previously given.

It will thus be seen that the output of the pump 1, comprising the combined outputs of its two pumping sections, may be infinitely varied from minimum to maximum by rotation of the handwheel 207. This control is effected without definite control of the division of the combined outputs between the two pumping sections (except, of course, at maximum and minimum outputs) and is accomplished by a single common

master cylinder which regulates the joint or combined movements of the two adjusting pistons 10 by regulating the amount of fluid admitted jointly to or expelled jointly from the two adjusting cylinders 11 in accordance with change in said master cylinder's volumetric capacity. It will further be understood that there is a definite combined output of the two pumping sections for each position of the piston 201 and hence for each position of the handwheel 207 although this combined output may be divided between the two pumping sections in numerous proportions.

In the foregoing explanation of the arrangement of Fig. 8 it has been presumed that there is a "straight line" relationship between movement of each adjusting piston 10 and the output of the pumping section under its control; that is to say, it has been presumed that movement of an adjusting piston 10 through a given distance will produce the same variation in the output of its pumping section irrespective of the particular portion of the range of inward and outward travel of the adjusting piston 10 in which said movement occurs, and that variation in output of each pumping section is proportional to movement of its adjusting piston 10 and adjusting rod 8. The "straight line" relationship usually exists or is closely approximated in "double" pumps of the general character under consideration. The control hereinbefore described will, however, function satisfactorily when this "straight line" relationship does not exist provided the existing relationship is continuous throughout the range of inward and outward movement of the adjusting piston 10 and is the same for both pumping sections. With the latter arrangement a compensating relationship exists between the relative movements of the two adjusting pistons 120 and the corresponding outputs of the two pumping sections, and the combined output of the two pumping sections is substantially unaffected thereby.

The advantages of the invention are many, whether controlled solely by manual effort, as in Fig. 8, or otherwise. No linkage of the elements 6 to a fixed point or coordinating structure is required between the two adjusting rods or their attached adjusting pistons 10; this greatly simplifies the structure and materially reduces cost and also makes it unnecessary to provide exact alignment between the two adjusting rods 6 or adjusting pistons 10 which in turn facilitates manufacture, eliminates the necessity of holding all parts to extremely close tolerances and thus contributes to cost reduction. One control valve mechanism serves to regulate the operation of both adjusting cylinders, with the advantage of cost reduction and simplification. It becomes unnecessary to make both pumping sections of exactly the same capacity as minor variations in their capacities are unimportant; the combined outputs of the two pumping sections are held to the proper amount irrespective of variation in the output of one pumping section with respect to the output of the other.

It is desirable to provide for the escape of any fluid that may leak by the adjusting pistons 10 into the inner ends of the adjusting cylinders 11. As explained in connection with Figs. 1-5, such leakage fluid may be conveniently conducted back to a suction port of the pump 1 through the passages 31 and 32 which are formed in the end plates 25. In the schematic arrangement of Figs. 7, 8, 10, and 11, however, the inner ends of the adjusting cylinders 11 are shown as connected with the reservoir 52, as through passages 123

which are branched in Figs. 7, 8 and 11 and which lead either directly to the reservoir or to other passages connected therewith.

The adjusting piston and adjusting rod arrangement shown in Fig. 1 and schematically illustrated in Figs. 7, 8 and 10, in which the delivery-determining elements are moved outward in a delivery-increasing direction by springs in co-operation with pump pressure acting on the "pumping arcs" of the slides 7, has many practical advantages. One such advantage is the simplicity of the fluid control circuit and of the mechanism which may be used for regulating the withdrawal of fluid from or admission of fluid to the branch passage 21 in order to vary and control the output of the pump. Another advantage is that there is no lost motion in the adjusting mechanism, due to the co-operating relationships of the parts and of the forces acting upon them. For these and other reasons I prefer to use the adjusting rod and piston arrangement as already described in connection with Figs. 1, 7 and 8, but other adjusting piston arrangements and other methods of outward actuation of the delivery-determining elements may be employed and certain of the advantages of the invention still retained. In order that this may be clearly understood, in Fig. 9 I have shown an arrangement in which the adjusting rods 6 are adapted to be moved outward by their corresponding adjusting pistons 220 by admission of pressure fluid to the inner ends of the adjusting cylinders 221 (i. e. the ends of said cylinders adjacent the pump 1) while simultaneously permitting withdrawal of fluid from the outer ends of said cylinders, and vice versa. The outer ends of the cylinders 221 are connected by a branched passage 124 and the inner ends thereof are connected by another branched passage 223, and said cylinders and passages are filled with an incompressible fluid such, for example, as oil. The output of the pump 1 is varied and controlled by regulating the admission of fluid to one of these branched passages and simultaneously regulating the withdrawal of fluid from the other. In Fig. 9 this admission and withdrawal is under control of my novel and improved hydraulic follow-up which may be readily modified for use in regulating the admission and withdrawal of fluid to and from the branch passage 21 of Fig. 1, as will be understood from the description which follows.

As in the embodiments previously described, the pump 1 (Fig. 9) has a pair of laterally movable adjusting rods 6 which control the outputs of its two pumping sections. Each adjusting rod 6 is attached to its adjusting piston 220 by which it is moved and its position controlled. Each adjusting piston 220 is slidably fitted within the bore of its adjusting cylinder 221 which is closed on its inner end (that is, its end adjacent the pump 1) by an end cover 222. The piston rod of each adjusting piston passes through a suitable opening in the end cover 222 of its adjusting cylinder 221, appropriate packing being provided to prevent leakage. Each adjusting piston 220 is also modified to provide a rod 224 extending outwardly therefrom, the purpose of which will be presently explained, the rod 224 passing through an opening in the closed outer end of its adjusting cylinder 221 which is suitably packed to prevent leakage. While only one adjusting cylinder 221 is shown in section together with its adjusting piston 220 and associated parts, it will be understood that both adjusting cylinders 221 are of identical construction.

The branched passage 124, which connects the

outer ends of the adjusting cylinders 221, leads to and connects with an annular cylinder port 227 in the bore of the valve housing 225 and similarly the branched passage 223, which connects the inner ends of said adjusting cylinders, leads to and connects with an annular cylinder port 228 in the bore of the valve housing 225.

A valve piston 230 is slidably fitted within the bore of the valve housing 225 and is formed with three heads, 231, 232 and 233 respectively. The proportions are such that the heads 232 and 233 cover the cylinder ports 228 and 227 respectively when the valve piston 230 occupies its neutral position in which it is shown in Fig. 9, thus cutting off communication between the valve bore of the housing 225 and both ends of the adjusting cylinders 221. Movement of the valve piston 230 to the right of its neutral position connects the cylinder port 227 with the supply of pressure fluid, which is admitted through the passage 236 from any suitable source such as the discharge conduit 53, and simultaneously connects the cylinder port 228 with the exhaust passage 229. Pressure fluid is then admitted to the outer ends of the adjusting cylinders 221 and the adjusting pistons 220 are moved in an inward or delivery-decreasing direction. In the same manner, movement of the valve piston 230 to the left of its neutral position admits pressure fluid to the cylinder port 228 and the inner ends of the adjusting cylinders 221 so that the adjusting pistons 220 are moved in an outward or delivery-increasing direction.

The valve piston 230 is moved responsive to action of my novel and improved hydraulic follow-up which will now be described. It includes a manually adjustable element here shown as an expansible Sylphon bellows 240 mounted in the bore of a containing member 241 in such manner that the parts thereof may move freely as the axial length of said bellows 240 is extended and contracted. The right hand end of the bellows 240 abuts against the end cover 242 of the container 241 and is provided with a suitable connection by which the interior of said bellows 240 is connected with a branched passage 244. The other end of the bellows 240 terminates in a rigid end piece adapted to bear against the end of a screw 246 which extends through a threaded opening in the closed end of the container 241 and which is provided with a handwheel 247 by which it may be rotated. Rotation of the handwheel 247 in one direction advances the screw 246 into the bore of the container 241, causing the bellows 240 to contract in axial length so that the volumetric capacity of its interior is reduced. Rotation of the handwheel 247 in the other direction withdraws the screw 246 from the bore of the container 241 and permits the bellows 240 to extend its axial length and thus increase the volumetric capacity of its interior, this extending movement of the bellows 240 taking place under the influence of its spring-like property and the force exerted by any fluid pressure that may exist therein.

The branched ends of the passage 244 lead to and connect with the interiors of a pair of smaller expansible Sylphon bellows 248, one of which is located in a suitable bore formed in the extended outer end of each adjusting cylinder 221. The outer end of each bellows 248 is held against outward movement by the cylinder end cover 249 and the other end thereof terminates in a rigid end piece adapted to abut against the end of the rod 224 of the adjusting piston 220. The rigid

end piece of each bellows 248 moves with and is moved by its rod 224 so that the position of the corresponding adjusting piston 220 determines the axial length of said bellows 248 and hence the volumetric capacity of its interior. The interiors of the two bellows 248 are connected with the ends of the branched passage 244 and their sizes are such that changes in their combined interior volumetric capacities incident to movement of the adjusting pistons 220 between their extreme inward and outward position is equal to or less than the maximum change which can be made in the volumetric capacity of the interior of the bellows 240 by rotation of the handwheel 247.

A passage 245 leads from the branched passage 244, and connects with the interior of another expansible Sylphon bellows 250 positioned in the left hand end of the bore of the valve housing 225. The left end of the bellows 250 abuts against the end cover 226 of the valve housing 225 and the right end thereof terminates in a rigid end piece abutting against the end of the head 231 of the valve piston 230. The right hand end of the valve bore contains a compression spring 234, one end of which abuts against the valve housing's end cover 235 and the other end of which abuts against the adjacent end of the valve piston 230.

The interiors of the bellows 240, the two bellows 248, the bellows 250 and their connecting fluid passages 244 and 245 are filled with a substantially non-compressible fluid, such as oil or glycerine, and comprise the closed hydraulic follow-up circuit.

Upon rotation of the handwheel 247 in the direction which decreases the axial length of the bellows 240, and the volumetric capacity of the interior thereof, some of the fluid therein will be forced out into the branched passage 244. The rods 224 present greater resistance to movement than can be overcome by the force thus exerted thereon through the bellows 248 and hence the volumetric capacities of said bellows 248 will not change in initial response to movement of the handwheel 247. The spring 234 presents relatively small resistance, however, so that fluid displaced from the bellows 240 by rotation of the handwheel will enter the bellows 250, increasing its axial length and moving the valve piston 230 out of its neutral position and toward the right as viewed in Fig. 9.

As already explained, movement of the valve piston to the right of its neutral position causes movement of the adjusting pistons 220 in an inward or delivery-decreasing direction, thus reducing the output of the pump 1. The bellows 248 will expand simultaneously with this inward movement of the adjusting pistons 220 and their interior capacities will be increased conformably with the extent of movement of said adjusting pistons 220 and the rods 224, permitting the entry of an amount of fluid equal to the increase thus produced in their combined volumetric capacities. This inward movement of the adjusting pistons 220 and simultaneous expansion of the bellows 248 will continue as long as the handwheel 247 is rotated in the direction which contracts the axial length of the bellows 240 until, of course, the adjusting pistons 220 reach their extreme inward positions and the output of the pump 1 is reduced to its minimum. When rotation of the handwheel is stopped, the adjusting pistons 220 will continue to move for a very slight distance and the valve piston 230, under in-

fluence of the spring 234, will move toward the left, contracting the bellows 250 and forcing fluid therefrom into the interior of the two bellows 248. This movement of the valve piston 230 toward the left continues until said valve piston is restored to its neutral position and the adjusting pistons 220 are again rendered inoperative. While described progressively, these movements take place almost instantaneously and there is no "lag" except the extremely small amount incident to movement of the valve piston 230.

Rotation of the handwheel 247 in the opposite direction permits expansion in the axial length of the bellows 240 and increase in its volumetric capacity. The spring 234 then moves the valve piston 230 away from its neutral position and toward the left as the axial length of the bellows 250 is slightly contracted and a small amount of fluid is emitted therefrom. As already explained, this movement of the valve piston 230 causes movement of the adjusting pistons 220 in an outward or delivery-increasing direction so that the output of the pump 1 is increased. Restoring movement of the valve piston 230 to its neutral position takes place when the handwheel 247 is stopped and fluid from the bellows 248 is forced into the bellows 250.

From the foregoing it will be clear that the valve piston 230 is moved out of its neutral position, in one direction or the other, responsive to movement of the handwheel 247, which movement of the valve piston causes movement of the adjusting pistons 220, and that said valve piston 230 is restored to its neutral position by responsive movement of said adjusting pistons 220. It will also be clear that the volume of fluid admitted jointly to or exhausted jointly from the two bellows 248 is always exactly equal to the change in volumetric capacity of the bellows 240 so that a definite combined movement of the two adjusting pistons 220 results from each extent of movement of the handwheel 247, this resulting movement of the adjusting piston 220 producing a definite change in the combined outputs of the two pumping sections of the pump 1 when the operating conditions of the latter remain unchanged. Division of this movement between the two adjusting pistons 220 is immaterial as the combined outputs of the two pumping sections is unaffected thereby when said two pumping sections are of equal maximum capacities or outputs as is here presumed for simplicity of explanation. Restoration of the valve piston 230 to its neutral position is likewise unaffected by unequal movement of the adjusting pistons 220, this restoration taking place whenever the changes in the combined volumetric capacities of the two bellows 248 equals the change in the volumetric capacity of the bellows 240, regardless of whether the volumetric capacities of the two bellows 248 are changed in equal or unequal amounts.

In a similar manner the follow-up acts to correct for any displacement of the adjusting pistons 220 which could alter the output of the pump 1 from the output corresponding to any particular position in which the handwheel is held stationary. For example, with the handwheel 247 stationary, if one of the adjusting pistons 220 moves outwardly, as because of leakage or otherwise, and the other adjusting piston likewise moves outwardly or remains stationary, an increase in the output of the pump 1 would result unless corrective action is set in motion. Such movement of one or both adjusting pistons 220 will, however, cause corresponding contraction of

the axial length and volumetric capacity of the bellows 248, and the fluid thus forced out of the bellows 248 will enter the bellows 250, moving the valve piston 230 out of its neutral position and toward the right as viewed in Fig. 9. This movement of the valve piston 230 will cause the adjusting pistons 220 to move in an inward or delivery-decreasing direction until the combined volumetric capacities of the two bellows 248 again equal the proper amount as determined by the bellows 240 and the handwheel 247, when the valve piston 230 will again be restored to its neutral position, this corrective action also having reduced the output of the pump 1 to the amount as determined by the position of the handwheel 247. Corrective movement of the adjusting pistons 220 is thus brought about whenever the combined capacities of the two bellows 248 is either increased or decreased by movement of one or both of the adjusting pistons which does not occur responsive to movement of the handwheel 247. The follow-up therefore functions not only to produce a definite output of the pump 1 (when operating conditions of said pump 1 remain unchanged, as is presumed throughout the explanation of the embodiment of Fig. 9) for each position of the handwheel 247, but also functions to continuously maintain such definite output.

The handwheel 247 may thus be rotated to cause any desired change in the capacity of the bellows 240 and hence to cause any desired output of the pump 1 so that the output of said pump 1 may be infinitely varied from minimum to maximum. The output of the pump 1, comprising the combined outputs of its two pumping sections, may thus be accurately controlled throughout the entire range of its adjustment. This control, which is effected independent of definite control of the output of either of the two pumping sections and without control of the division of the combined outputs between the two sections (except, of course, at maximum and minimum outputs), is accomplished by a single control valve common to both adjusting cylinders 221 and which regulates the joint operation of both adjusting pistons 220 by regulating the fluid volume admitted jointly to and expelled or exhausted jointly from both of said adjusting cylinders 221.

Springs 238 are provided in the inner ends of the adjusting cylinders 221 to furnish means for moving the adjusting pistons 220 outward when the output of the pump 1 is reduced to a minimum which is zero. These springs 238 are compressed as the adjusting pistons 220 move into positions corresponding to zero output of their pumping sections and the energy thus stored is available to move the adjusting pistons to cause the pump 1 to again deliver fluid.

My improved hydraulic follow-up has many advantages. One of these is that it is free from lost motion as the spring 234 takes up all "play" between the parts and continuously maintains a substantially constant pressure in the follow-up circuit. The parts are inexpensive and may be made of relatively light material as they are subject to no load except that due to the pressure in the follow-up circuit, which pressure is never high and cannot, in fact, exceed the amount determined by the spring 234; the parts are protected against the application of any external fluid pressure by the drain or exhaust connections 243 in the bores housing the two bellows 248 and the exhaust connection adjacent the bel-

lows 250 in the bore of the valve housing 225. Another and important advantage of my hydraulic follow-up is that the manually movable control member (that is the bellows 240 and the handwheel 241) and the control valve may be separated from the pump 1 or from each other to any desired extent, thus facilitating convenient location of the manual control member and making the follow-up well suited for use as a remote control.

When it is desired to use my improved hydraulic follow-up in connection with the adjusting cylinder and spring-loaded adjusting piston arrangement of Fig. 1, the branch passage 21 is connected with the annular cylinder port 227 in the bore of the valve housing 225 and the annular cylinder port 228 is suitably closed or may be omitted. With this arrangement the head 231 and the reduced portion of the valve piston intermediate the heads 231 and 232 may also be omitted, thus simplifying the structure and reducing its size, as the end of the bellows 250 will then be adapted to bear against the adjacent end of the head 232. The operation of this modified arrangement will be understood from the explanation already given.

A further modification of the mechanism for regulating the admission of fluid to and withdrawal of fluid from the branch passage 21 is illustrated in Fig. 10. The operation of this regulating mechanism is responsive to pressure of the fluid in the discharge conduit 53 and it functions to so regulate the admission and withdrawal of fluid from the branch passage 21 that the output of the pump 1 is altered to maintain a substantially constant predetermined pressure in said discharge conduit.

The branch passage 21 of Fig. 10 connects with an annular cylinder port 265 in the bore of the valve housing. Admission and withdrawal of fluid to and from the branch passage 21 are controlled by the head 269 of the valve piston which is slidably fitted within the valve bore, so that, as in previously described arrangements, a single element serves to regulate the volume of fluid admitted jointly to the outer ends of the adjusting cylinders 11 or withdrawn jointly therefrom. The arrangement is such that the head 269 closes the cylinder port 265 and completely cuts off communication between the branch passage 21 and the valve bore when the valve piston occupies its neutral position in which it is shown in Fig. 10. Movement of the valve piston to the left of its neutral position admits pressure fluid from the discharge conduit 53 to the branch passage 21, thus causing the adjusting pistons 10 to move inward to reduce the pump's output; pressure fluid from the discharge conduit is supplied to the valve bore as through a passage 273 which connects with said valve bore at a point intermediate the heads 267 and 269 of the valve piston. When the valve piston is moved to the right of its neutral position, fluid may pass from the branch passage 21 into the left end of the valve bore, through the exhaust passage 264 and to the reservoir 52, so that the adjusting pistons 10 are moved outward to increase the pump's output, this piston movement taking place under influence of the springs 12 and of pump pressure acting upon the inner ends of the slides 7, as already explained. The head 269 is shown as provided with conventional V-notches on each end thereof to provide gradual connection of the cylinder port 265 with the supply of pressure fluid and with the exhaust passage respectively when

the valve piston moves to the left and to the right of its neutral position. The right hand end 263 of the valve bore is also connected with the exhaust passage 264 to prevent building up of pressure therein, this connection being here shown as through the cover 261 which closes the right hand end of said valve bore.

Movement of the valve piston takes place responsive to change in pressure of the fluid in the discharge conduit 53 and its position in the valve bore is determined by the pressure in said discharge conduit relative to a pre-determined pressure value therein. This is accomplished by making the diameter of the head 269 greater than the diameter of the head 267 which is slidably fitted in the reduced portion of the valve bore, so that pressure of the fluid from the discharge conduit which is admitted to the valve bore intermediate these two heads exerts a force upon the valve piston tending to move said valve piston toward the left in opposition to the force exerted upon the valve piston by the spring 270 which tends to move said valve piston toward the right. It will thus be seen that the valve piston occupies its neutral position, in which the head 269 closes the cylinder port 265, only when the fluid pressure in the discharge conduit 53 has a definite value as determined by the spring 270, the compression of which may be adjusted by means of the screw 272 which extends through the end cover 262 of the valve housing and the inner end of which bears against the abutment piece 271 intermediate said screw's inner end and the spring 270. It will also be understood that increase in pressure of the fluid in the discharge conduit 53 above the value determined by the spring 270 will increase the fluid-exerted force urging the valve piston toward the left, so that said valve piston will be displaced to the left of its neutral position, causing the adjusting pistons 10 to move inward to decrease the pump's output. Conversely, decrease in pressure of the fluid in the discharge conduit 53 below the value determined by the spring 270 will decrease the fluid-exerted force active in opposition to the force exerted by said spring and said spring 270 will immediately displace said valve piston to the right of its neutral position, permitting withdrawal or exhaust of fluid from the branch passage 21, whereby the adjusting pistons 10 will be moved outward or in a delivery-increasing direction. In either case, the valve piston will be restored to its neutral position and movement of the adjusting pistons will cease upon re-establishment of the pre-determined pressure value in the discharge conduit 53, as will in many instances result from corrective change in the pump's output.

In this manner the valve piston is moved, responsive to change in pressure of the fluid in the discharge conduit, to so regulate admission and withdrawal of fluid to and from the branch passage 21 that the output of the pump is automatically altered, within its capacity range, to provide a delivered volume at which the pressure of the fluid in the discharge conduit is held substantially constant at a pre-determined value.

As in the embodiments previously described, each adjusting piston 10 is independently movable to control the output of its pumping section and there are an almost infinite number of combinations of relative positions of the adjusting pistons 10 and their attached adjusting rods 6 which these parts may occupy at any intermediate output of the pump 1. As in the previously

described embodiments, relative division of the pump's output between its two pumping sections is not controlled and does not affect the control of the pump's output as the combined outputs will be held at the proper total amount required to produce the pre-determined pressure in the discharge conduit irrespective of the proportion of this total contributed by each pumping section. This control of output is therefore effected independent of definite control of the output of either of the two pumping sections (except, of course, at maximum and minimum outputs) and is accomplished by a single control valve mechanism common to both adjusting cylinders 11 and which regulates the joint operation of both adjusting pistons 10 by regulating the fluid volume admitted jointly to or expelled jointly from the passage 20 and said adjusting cylinders 11.

In the embodiment of Fig. 11 the floating pistons 10 are disposed in the same cylinder 11', with the springs 12 disposed between the ends of the cylinder and the respective pistons and the piston rods 6' extending out from the pistons. The piston rods 6' may be connected in any suitable manner with the delivery-determining members 6 of the pump sections or units, these connections being omitted for convenience in illustration. The piston and cylinder arrangement of Fig. 11 may be employed with any of the previously described output control means but for convenience manually adjustable means similar to that illustrated in Fig. 8 is shown. These means comprise the master cylinder 200 and the handwheel 207 for either introducing more fluid to or exhausting fluid from the passage 21 and the double cylinder 11'. The ends of the double cylinder are connected by a passage 123 leading to the exhaust. The pistons 10, as described above in connection with the other modifications, float between the pressures generated by the pump sections and each is free to assume a position in balanced relation through the flexible hydraulic transmission between the pistons exteriorly of the pumping sections. By rotating the manually adjustable wheel 207 in opposite directions more of the incompressible fluid may be supplied to the passage 21 and the double cylinder or fluid may be exhausted therefrom, which operation causes corresponding movements of the adjusting pistons 10.

It will be understood that the several embodiments of my invention have been described for the purpose of illustrating the operation and construction of the apparatus of my present invention and that changes may be made without departing from the spirit of the invention.

I claim:

1. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein diametrically opposite delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease their deliveries, and a control thereof, in combination, diametrically opposed fluid pressure cylinders in alignment with said elements, pistons in said cylinders operatively connected with said elements, a fluid pressure passage connecting the outer ends of said cylinders, said passage and outer ends of said cylinders containing an incompressible fluid and means for forcing additional fluid into said passage and said cylinders against the pump section pressures and for permitting the exhaust of the fluid therefrom.

2. In a rotary pump comprising two separate

pump sections discharging into a common chamber wherein radially movable elements vary and determine the outputs per revolution of said sections, and a control thereof, in combination, a pair of separate fluid pressure pistons operatively connected with said elements, said elements and said pistons being urged in the delivery increasing direction by the internal pressures of the pump sections, an adjustable incompressible fluid circuit connecting said pistons exteriorly of the pumping sections, said incompressible fluid exerting forces on said pistons in opposition to the pressures generated by the pump sections, and means for forcing fluid into said circuit against the pump pressure and to permit the exhaust of fluid therefrom responsively to movements of the pistons by the pump pressures.

3. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein diametrically opposite delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease the delivery, and a control thereof, in combination, diametrically opposed fluid pressure cylinders in alignment with said elements, pistons in said cylinders operatively connected with said elements, springs behind said pistons urging them in the direction of the force exerted by the pump internal pressures, a fluid pressure passage connecting the outer ends of said cylinders, said passage and outer ends of said cylinders containing an incompressible fluid, and means for forcing additional fluid into said passage and said cylinders against the pump pressures and the springs and for permitting the exhaust of the fluid therefrom.

4. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein each section is provided with a delivery varying element which is urged in one direction by the pressure generated in the corresponding pump section, in combination, fluid pressure pistons operatively connected with said elements to move in unison therewith, an incompressible fluid circuit connecting said pistons, said incompressible fluid exerting forces on said pistons in opposition to the pressures generated by the pump sections and means for forcing fluid into said circuit against the pump section pressures and to permit the exhaust of fluid therefrom responsively to movements of the pistons by the pump pressures.

5. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein each section is provided with a delivery varying element which is urged in one direction by the pressure generated in the corresponding pump section, in combination, a pair of separate fluid pressure pistons operatively connected with said elements to move in unison therewith, an incompressible fluid pressure circuit connecting said pistons exteriorly of the pump with each piston being thereby floatingly balanced between the pressures generated in the separate pump sections, and means for varying at will the length of said incompressible fluid circuit.

6. In a rotary piston pump unit of the multi-section type discharging into a common outlet wherein diametrically opposite delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease their deliveries, and delivery control means therefor, in combination, diametrically opposed fluid pressure cylinders in

alignment with said elements, pistons in said cylinders operatively connected with said elements, a fluid pressure passage leading to the outer end of each cylinder and containing a fluid adapted to hold the piston of the corresponding cylinder against the pressure of the corresponding pump section and means for forcing additional fluid into the passage leading to each cylinder and for permitting the exhaust of fluid therefrom to cause the piston of each cylinder to assume different positions.

7. In a rotary piston pump unit of the multi-section type discharging into a common outlet wherein each section is provided with a delivery varying element which is urged in one direction by the pressure generated in the corresponding pump section, in combination, a pair of separate fluid pressure pistons operatively connected with said elements to move in unison therewith, each of said pistons being operated by fluid pressure against the pressure generated in the corresponding pump section to vary the positions of the pistons and the corresponding delivery varying elements and means for varying the fluid pressure opposed to pressures generated by the pump sections.

8. A rotary piston pump unit of the multi-section type discharging into a common outlet wherein diametrically opposed delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease their deliveries, comprising diametrically opposed fluid pressure cylinders in alignment with said elements, piston elements in said cylinders for operating said delivery determining elements, and self-aligning operative connections between said elements and said pistons, said self-aligning operative connections each including a member positioned intermediate the piston element and the corresponding delivery determining element, said member having on one of its ends a flat surface engaging one of said elements and said member having on its other end a spherical member engaging the other of said elements.

9. A rotary piston pump unit of the multi-section type discharging into a common outlet wherein diametrically opposed delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease their deliveries, comprising diametrically opposed fluid pressure cylinders in alignment with said elements, a piston element in each cylinder for operating the corresponding delivery determining elements, and a self-aligning operative connection between each delivery determining element and its corresponding piston element, said operative connection comprising a bearing member having a flat face in engagement with one of said elements and an engaging surface of arcuate form with the other of said elements.

10. A rotary piston pump unit of the multi-section type discharging into a common outlet wherein diametrically opposed delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease their deliveries, comprising diametrically opposed fluid pressure cylinders in alignment with said elements, pistons in said cylinders for operating said delivery determining elements, and self-aligning operative connections between said elements and said pistons, each of said pistons having a guiding skirt and a spring

surrounding the delivery determining element and urging the delivery determining element and the piston radially outward.

11. A rotary piston pump unit of the multi-section type wherein diametrically opposed delivery determining elements are movable radially inwardly against the pressure or pressures of the pump sections to decrease their deliveries, comprising an intake for each section, diametrically opposed fluid pressure cylinders in alignment with said elements, pistons in said cylinders exposed to pressure fluid only in the outer ends of said cylinders and operatively connected with said elements, and means for maintaining at the inner end of each cylinder and piston a relatively low pressure substantially equal to the pressure of the fluid in said intakes comprising connections between the inner ends of said cylinders and at least one of said intakes.

12. In mechanism for controlling the volume of fluid delivered to a common discharge conduit by a multi-section pump, a separate output-determining element for each section independently movable to vary the output thereof independent of the outputs of the other sections, a separate fluid motor for each of said sections adapted to actuate the output-determining element thereof, the operation of each of said fluid motors in one direction of its movement being responsive to the admission of pressure fluid thereto and in the other direction of its movement being responsive to the emission of fluid therefrom, and valve mechanism common to all of said fluid motors for regulating the volume of pressure fluid admitted jointly to all of said fluid motors and the volume of fluid expelled jointly therefrom, whereby the volume of fluid delivered to the common discharge conduit by all of said pumping sections is controlled irrespective of the portion thereof contributed by each of said sections at outputs intermediate the maximum and the minimum combined outputs of all of said sections.

13. In mechanism for controlling the volume of fluid delivered to a common discharge conduit by a multi-section pump, a separate output-determining element for each section independently movable to vary the output thereof independent of the outputs of the other sections, separate independently operating power means for each of said sections adapted to actuate the output-determining element thereof, and control means common to all of said power means for regulating the extent of the combined operation of all of said power means without predetermined control of the extent of operation of any one of them, whereby the volume of fluid delivered to the common discharge conduit by all of said sections is controlled irrespective of the division thereof between said pumping sections at outputs intermediate the maximum and the minimum outputs of all of said sections.

14. In mechanism for controlling the volume of fluid delivered to a common discharge conduit by a plurality of variable output pumping units, a separate output-determining element for each of said units independently movable to vary the output thereof independent of the output of the others of said units, a separate fluid motor for each of said units adapted to actuate the output-determining element thereof, the operation of each of said fluid motors in one direction of its movement being responsive to the admission of pressure fluid thereto and in the other direction of its movement being responsive to the emission

of fluid therefrom, and valve mechanism common to all of said fluid motors for regulating the volume of pressure fluid admitted jointly to all of said fluid motors and the volume of fluid expelled jointly therefrom, whereby the volume of fluid delivered to the common discharge conduit by all of said pumping units is controlled irrespective of the portion thereof contributed by each of said pumping units at outputs intermediate the maximum and the minimum combined outputs of all of said units, said valve mechanism including a valve element active to control admission of pressure fluid to and emission of fluid from said fluid motors and movable responsive to fluid pressure existing in the discharge conduit relative to a predetermined pressure therein, said element occupying a neutral position at which the combined outputs of said pumping units is unchanged when said predetermined pressure exists in said discharge conduit, said element moving in one direction away from its neutral position to cause operation of said fluid motors in a direction whereby the combined outputs of said pumping units is decreased responsive to pressure in said discharge conduit in excess of said predetermined pressure and said element moving in the other direction away from its neutral position to cause operation of said fluid motors in a direction whereby the combined outputs of said pumping units is increased responsive to pressure in said discharge conduit below said predetermined pressure.

15. In mechanism for controlling the output of a double pump comprising two pumping sections, said pumping sections having a common discharge conduit and the combined outputs of said pumping sections constituting the output of said double pump, a separate output-determining element for each of said pumping sections independently movable against the fluid pressure generated in the corresponding pumping section to vary the output of its pumping section independent of the output of the other pumping section, a separate fluid motor for each of said pumping sections adapted to actuate the output-determining element thereof, the operation of each of said fluid motors being responsive to the admission of pressure fluid thereto and the emission of fluid therefrom, and valve mechanism common to all of said fluid motors for regulating the volume of pressure fluid admitted jointly to and the volume of fluid expelled jointly from all of said motors, whereby the output of said double pump is controlled irrespective of the portion thereof contributed thereto by each of said pumping sections at outputs intermediate the maximum and minimum output of said double pump.

16. In mechanism for controlling the output of a double pump comprising two pumping sections, said pumping sections having a common discharge conduit and the combined outputs of said pumping sections constituting the output of said double pump, a separate output-determining element for each of said pumping sections independently movable to vary the output of its pumping section independent of the output of the other pumping section, a hydraulically-actuated piston for each of said pumping sections movable against the fluid pressure generated in the corresponding pumping section to actuate the output-determining element thereof, movement of each of said pistons being responsive to admission of fluid to and emission of fluid from the adjusting cylinder thereof, and means common

to the cylinders of both of said pistons for regulating the volume of fluid admitted jointly to and emitted jointly from one side of each of said pistons, whereby the extent of the combined movement of both of said pistons is varied without individual control of the extent of movement of either of them, thereby controlling the output of said double pump irrespective of the division thereof between said pumping sections at outputs intermediate the maximum and the minimum outputs of said double pump.

17. In mechanism for controlling the volume of fluid delivered to a common discharge conduit by a plurality of variable output pumping units, a separate output-determining element for each of said units independently movable to vary the output thereof independent of the output of the others of said units, a separate fluid motor for each of said units adapted to actuate the output-determining element thereof, the operation of each of said fluid motors being responsive to the admission of pressure fluid thereto and the emission of fluid therefrom, and valve mechanism common to all of said fluid motors for regulating the volume of fluid admitted jointly thereto and the volume of fluid expelled jointly therefrom, whereby the volume of fluid delivered to the common discharge conduit by all of said pumping units is controlled irrespective of the portion thereof contributed by each of said pumping units at outputs intermediate the maximum and the minimum combined outputs of all of said units, said valve mechanism including a control valve having a piston movable under control of a hydraulic follow-up to regulate admission of fluid to and emission of fluid from said fluid motors, said hydraulic follow-up comprising a closed fluid circuit having a first fluid chamber the volumetric capacity of which is variable to cause operation of said fluid motors, a separate fluid chamber for each of said units the volumetric capacity of which varies simultaneously with and responsive to operation of the fluid motor thereof and a fluid chamber the volumetric capacity of which varies responsive to changes in the volumetric capacities of said other chambers and active thereupon to cause movement of said valve piston, and means for varying the volumetric capacity of said first named chamber.

18. In mechanism for controlling a plurality of variable output pumping units, a separate fluid motor for each of said units for varying the output thereof, each of said fluid motors comprising a piston which is operated by the admission and exhaust of fluid pressure from the opposite sides thereof, a valve for controlling the exhaust and admission of fluid pressure, an adjustable hydraulic circuit to which said valve is operatively responsive, operative connections between the pistons and said circuit and means for adjusting the said circuit at will to alter the combined outputs of the said pumping units.

19. In a control mechanism for the combined outputs of a plurality of variable-output pumping units having a common discharge conduit, a separate output-determining element for each of said units movable to vary the output thereof independent of the output of said other units, a separate fluid motor for each of said units adapted to actuate the output-determining element thereof, the operation of each of said fluid motors being responsive to admission of fluid thereto and emission of fluid therefrom, and means common to all of said fluid motors for regulating the

volume of fluid admitted jointly to all of said fluid motors and emitted jointly therefrom, whereby the combined operation of all of said fluid motors is controlled independent of the extent of their relative operation.

20. In a rotary piston pump of the multi-section type wherein each section is provided with a delivery varying element, in combination, means for actuating said delivery varying elements and controlling the total fluid volume delivered by the multiple pump sections but without definite control of the division of the total output between the individual pumping sections and means for equalizing the fluid pressure generated in all of said pumping sections irrespective of the relative outputs of each of said sections.

21. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein each section is provided with an element which is movable to vary the output of that particular section, in combination, means for actuating said delivery varying elements to produce a total combined movement thereof, but without control of the division of said total movement between them.

22. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein each section is provided with an element which is movable to vary the output of its particular section and is urged in one direction by the internal pressure of its section, in combination, means linking said delivery varying elements together so as to be floatingly balanced between the internal pressures of the pump sections whereby said elements may compensatingly change their relative positions without change in adjustment of said link means and means for adjusting said link means to alter the combined outputs of all of said sections but without definite control of the output of any section.

23. In a rotary piston pump unit of the multi-section type discharging into a common chamber wherein each section is provided with a delivery varying element which is urged in one direction by the pressure generated in the corresponding pump section, in combination, a cylinder, fluid pressure pistons disposed in said cylinder and operatively connected with said elements to move in unison therewith, an incompressible fluid circuit connecting said pistons, said incompressible fluid exerting forces on said pistons in opposition to the pressures generated by the pump sections and manual means for forcing fluid from a reservoir into said circuit against the pump section pressures and to permit the withdrawal of fluid therefrom responsively to movements of the pistons by the pump pressures.

24. In a variable delivery pump and a control therefor a delivery determining element, a fluid pressure cylinder in alinement with said element, a piston in said cylinder for operating said element and self-aligning operating connections between said element and said piston, said connections comprising a member operatively connected with said delivery determining element and passing through an opening in a spacer ring, a bearing member having a flat face in engagement with the piston and an engaging surface of spherical form in engagement with said member, a stop ring through which said member passes and having a cylindrical surface corresponding

to the surface of the spacer ring and a spring surrounding said member with one end engaging said stop ring and the other end engaging said member.

25. In mechanism for controlling the volume of fluid delivered to a common discharge conduit by a plurality of variable output pumping units, a separate output-determining element for each of said units independently movable to vary the output thereof independent of the output of the others of said units, a separate fluid motor for each of said units adapted to actuate the output-determining element thereof, the operation of each of said fluid motors being responsive to the admission of pressure fluid thereto and the emission of fluid therefrom, and valve mechanism common to all of said fluid motors for regulating the volume of fluid admitted jointly thereto and the volume of fluid expelled jointly therefrom, whereby the volume of fluid delivered to the common discharge conduit by all of said pumping units is controlled irrespective of the portion thereof contributed by each of said pumping units at outputs intermediate the maximum and the minimum combined outputs of all of said units, said valve mechanism including a control valve having a piston movable under control of a hydraulic follow-up to regulate admission of fluid to and emission of fluid from said fluid motors.

26. In mechanism for controlling the volume of fluid delivered to a common discharge conduit by a multi-section pump, a separate output-determining element for each section independently movable to vary the output thereof independent of the outputs of the other sections, separate power means for each of said units adapted to actuate the output-determining element thereof, and follow-up control means common to all of said power means for regulating the extent of the combined operation of all of said power means without predetermined control of the extent of operation of any one of them, whereby the volume of fluid delivered to the common discharge conduit by all of said sections is controlled irrespective of the division thereof between said pumping sections at outputs intermediate the maximum and the minimum outputs of all of said sections.

27. In a variable delivery pump and a control therefor, a delivery determining element, a fluid pressure cylinder, a piston element in said cylinder for operating said delivery determining element and a self-aligning operating connection between said delivery determining element and said piston element including a member relatively laterally slidable with respect to one of said elements to compensate for lateral misalignment between said elements and universally movable with respect to the other of said elements to compensate for angularity of one element with respect to the other.

28. A variable delivery rotary piston pump unit of the multi-section type wherein a separate variable delivery means is provided for each section, a fluid pressure cylinder for each section, a piston in each cylinder for operating its corresponding delivery varying means, a self-aligning operative connection between each piston and its delivery varying means, and a spring opposing the movements of the piston in one direction.

CHARLES M. KENDRICK.

CERTIFICATE OF CORRECTION.

Patent No. 2,238,062.

April 15, 1941.

CHARLES M. KENDRICK.

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction as follows: Page 1, second column, line 50, for the patent number "2,141,270" read --2,141,170--; page 8, first column, line 27, for the reference numeral "8" read --6--; and that the said Letters Patent should be read with this correction therein that the same may conform to the record of the case in the Patent Office.

Signed and sealed this 20th day of May, A. D. 1941.

(Seal)

Henry Van Arsdale,
Acting Commissioner of Patents.