

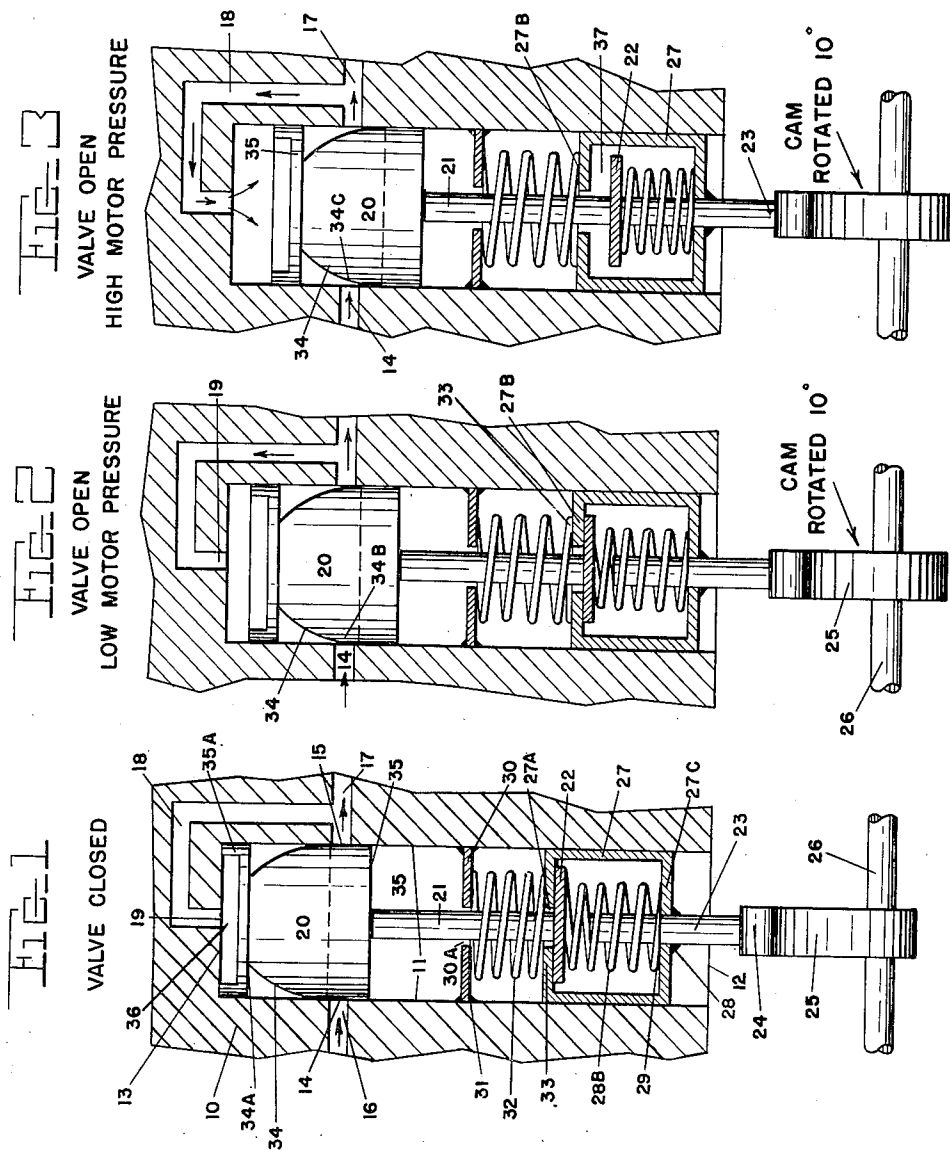
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A. R. STONE

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PISTON VALVE

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ALBERT RIVINGTON STONE INVENTOR.

BY *John M. Joynt*
HIS ATTORNEY

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PISTON VALVE

Albert Rivington Stone, Anneslie, Md., assignor
to Gerotor May Corporation, a corporation of
Maryland

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1

My invention constitutes a continuation in part of my co-pending application for Letters Patent of the United States, Serial No. 594,279, filed May 17, 1945, and entitled Hydraulic system, now Patent Number 2,441,339, granted May 11, 1948. It relates generally to an apparatus for metering fluid systems and has particular application to a novel metering valve and method of operating the same for the accurate control of variable load hydraulic circuits.

One object of my invention is to provide metering apparatus of the general type disclosed which is simple, reliable, and highly efficient in operation and which, while of low first cost because of its comparatively small number of parts and easy assembly, nevertheless displays low wear and long life over intensive service.

Another object is to provide a metering valve of the general type described having the characteristic of leakage compensation under heavy-duty, variable-load hydraulic service, being fully load-responsive with high sensitivity and close regulation in entirely self-adjustable and automatic manner and in the substantial absence of supervision, and this with substantially the same high degree of sensitivity throughout a wide control range.

Yet another object is to provide a metering valve of this general type whereby, from a constant volume source of hydraulic energizing fluid, variable loads ranging above a predetermined minimum load for which the system has been preliminarily regulated, can be serviced in efficient and automatic manner through a mechanism brought into play by leakage pressure occasioned by increase in system load.

Other objects and advantages will in part be obvious and in part pointed out hereinafter during the course of the following description.

My invention accordingly resides in the several elements, features of construction and combination of parts, and in the several operational steps, as well as in the relation of each of the same to one or more of the others, the scope of the application of all of which is fully set forth in the claims at the end of this specification.

In the drawings, wherein solely by way of illustration and not in a limitative sense I disclose one embodiment of my invention which I prefer at present, Figures 1, 2, and 3 are like sectional views in which in Figure 1 the valve is shown as closed, in Figure 2 the valve has been cracked slightly under manual regulation and is operating under minimum load conditions for that degree of basic regulation, while in Figure 3 the

2

main control adjustment remains as in Figure 2 but the valve has been further cracked in robot-controlled manner due to leakage pressure which is attendant upon temporary increase of the load on the system.

So that more thorough appreciation may be had of the precise nature and scope of my present invention, it is interesting to note at this point that extremely important impetus was given to the field of hydraulic powering equipment when multi-chambered rotary hydraulic pumps and motors were first introduced on the market. It had long been recognized theretofore that basically, fluid systems were fundamentally capable of a characteristic flexibility and adaptability for widely varying load conditions which probably surpassed by substantial margins practically all other types of powering equipment, even including electric motors. Nevertheless, the acceptance of such equipment in commerce and industries necessarily was drastically curtailed due to the substantial absence of satisfactory apparatus or mechanism for carrying these advantageous characteristics into use.

To illustrate, the familiar reciprocating pump, perhaps constituting the prototype of all present-day hydraulic equipment and itself still in service for many specialized operating conditions, possessed many fundamental deficiencies which effectively precluded it from representing the epitome of hydraulic service. It was bulky, cumbersome, costly and uncertain in operation. It required close supervision. Its pulsating, surge-like delivery even where costly equalizing domes were employed, detracted from its desirability.

While, for the first time, surging pulsations were effectively eliminated upon the introduction into the field of the now widely-accepted single-chambered rotary pump, this pump was available only for low-pressure service. It was quickly established that it could not develop sufficient force to work against pressure heads of appreciable magnitude. Leakage and slippage combined effectively to prevent high-pressure duty.

Upon the advent of the multi-chambered rotary pump, however, wherein the separate effects of the several component pumping chambers were combined in additive relation, the picture was appreciably changed. For the first time high-pressure loads could be serviced in continuous non-pulsating manner. Full advantage could be taken of the characteristic flexibility inherent in hydraulic systems.

Flexible as is the hydraulic equipment which has been described, nevertheless it is subject to a

diminished extent, to a limiting factor which is inherent in all powering equipment, that is, no type of prime mover, dynamo, or other powering equipment, either driving or driven, is capable of operating at its maximum efficiency throughout its entire operating range. Explained in other terms in order to emphasize this point, the entire operating range of practically all powering equipment is substantially greater than the relatively small portion thereof in which peak efficiencies are obtainable.

The foregoing phenomenon is of no particular moment when loads are being serviced which are either constant or else are of limited variability in but a small number of definitely defined steps. In such instances powering equipment is selected, operable at maximum efficiency for the particular load being serviced. Where a small number of definite increments in load are involved, then two or more powering units may be connected in tandem or parallel, so that each unit operates at its maximum efficiency, the total number of units being selected in response to the immediate load demands, so that at all times highly satisfactory system performance is achieved.

The situation is entirely different, however, where highly-variable loads are being serviced over a wide load range. Ordinarily where such loads are powered directly from a single powering unit, it is necessary to design the motive equipment such that it will adequately supply the maximum load demand. Should the powering equipment be designed for service at maximum efficiency at or about this peak load, however, then the operation of such unit for normal load demand, below peak performance, will be at reduced efficiency, and hence with increased operational costs. On the other hand, should such powering equipment be designed for maximum efficiency at normal loads, substantially below peak load demand, then the risk exists that should such peak load demand persist for any unusual duration, damage may result to the powering unit from continued operation in excess of its rated capacity. More practically, slippage and leakage would occur so that the powering equipment would refuse the load. All in all, this presented a very practical disadvantage which, prior to the introduction of the system according to my said parent application was resolved, if at all, only by the addition of an auxiliary and somewhat parasitical regulatory adjunct to the powering equipment. Measured in the light of practical tests, equipment provided with such auxiliary controls were unsatisfactory in performance. They did not satisfy the demands of the industries.

In accordance with the teaching of my said parent application, a radical departure was had from the modes of control theretofore suggested in that, while these prior systems uniformly undertook to vary the output of the particular hydraulic powering unit in accordance with the instantaneous load demands; on the contrary, and following the teaching of my invention it was proposed to employ a constant output power unit, operating always at maximum efficiency, and then to meter the quantity of fluid output of such constant-delivery unit which was directed to the load, in sensitive response to the instantaneous load demands. In short, instead of using a variable output powering unit with complicated controls, a constant output installation was employed with simple metering means. Results were achieved superior to any which had theretofore been considered possible. Not only were first costs low,

but maintenance and upkeep were materially reduced, and appreciably less supervision was required.

Despite the sensitive operation of such new system, it nevertheless required the services of an intelligent and highly observant operator to regulate the metering system in accordance with the observed variations in the load. Not only did this mean that there had to be constant supervision, but there was necessarily some time lag between first variation in the load and subsequent adjustments in conformity therewith, enduring for the normal reaction time of the operator. Such lag became appreciable upon wide and rapid fluctuations in the load conditions. To a certain extent at least, this rendered the system somewhat impractical for certain uses, as for example in the operation of a rolling mill and other equipment having repeated wide variations in load demands.

The improvements according to my present invention are directed to the removal in large measure of the aforementioned disadvantages characterizing the art as heretofore known, and to provide a fully automatic, somewhat robot-like control which quickly and sensitively compensates for all variations in load, introducing so much of the output of the constant output hydraulic powering equipment as is required to satisfy the immediate load demands for any particular normal regulation of the control mechanism, and being superbly adapted for rapid adaptation to and supply of loads variable in rapid, irregular and unpredictable manner.

With the foregoing generalized picture of the objectives of my invention kept prominently in mind, reference is now had to the three views of the drawings for readily understanding the following, more detailed description.

It will be assumed that the metering valve shown is interposed in a hydraulic line between a source of hydraulic power of suitable conventional type, not shown, and which typically may comprise a Gerotor make, multi-chambered rotary pump on the one hand; and means for powering a variable load, such as a hydraulic motor which similarly conveniently may be a Gerotor make, multi-chambered hydraulic motor on the other hand. This motor likewise is not shown, inasmuch as the details thereof do not per se form any part of this invention.

The present invention and therefore, the present disclosure, is directed primarily to the control means, taking the form of a metering valve, which is interposed between the several units of powering equipment. As has been stated, the characteristic feature of the hydraulic pump is that it is a constant speed, constant output hydraulic unit operating always at rated maximum efficiency, and with sufficient capacity to service properly the maximum load demand expected on the line. This pump operates through a closed system, returning directly to the inlet side of the pump all of the output fluid except that part thereof which at any instant is bled off to supply the instantaneous load demands. It is the metering valve now to be described which controls the part of the energizing fluid which is transmitted to the load, and determines the unit rate of flow.

This metering valve comprises a casing through which extends a vertically-disposed cylindrical bore of substantial diameter and opening to the exterior at its bottom surface at

12. It terminates short of the top surface of the casing 10 in a machined wall 13.

Casing 10 is laterally ported as at 14 and 15, these ports being let through the casing into the bore 11 at calibrated points along the extent of the bore near the upper end thereof, and preferably being diametrically opposed to each other. In the instance undergoing description, port 14 comprises the admission or inlet port, while port 15 comprises the outlet port. In the preferred embodiment, these ports are disposed horizontally and radially. It is entirely feasible, however, to arrange them in tangential manner and at any desired angle to the horizontal.

Channel 16 leads from the constant delivery source of fluid power through the casing 10 up to the admission port 14. Similarly, channel 17 leads from outlet port 15 through casing 10 to the load, not shown. Within the casing 10, a branch channel 18 leads from a generally T connection with channel 17 to a port 19 letting into the top of bore 11 through machined wall 13. Preferably channel 18 extends vertically and centrally through wall 13.

A skirted piston indicated generally at 20 is disposed for reciprocation in the bore 11, motion being imparted to the piston 20 by a centrally-disposed and downwardly-extending piston rod 21. The skirted portion of the valve 20 serves to control the degree of opening of ports 14 and 15. Its primary function, however, may be said to be the control of the admission or inlet port 14. Its simultaneous control of outlet port 15 is entirely incidental to its control of the inlet port.

Piston rod 21 has an extent substantially less than that of the bore 11, the construction being such that at no time does the outer end of rod 21 protrude beyond the confines of bore 11. At its lower or outer end, rod 21 terminates in a pressure plate 22 which, while of moderate thickness, is of substantial diameter, only slightly less than that of bore 11.

A cam follower rod 23 is disposed in the prolongation of the axis of piston rod 21, so that the two elements are coaxial. This rod 23 extends into the bottom of bore 11 from the lower exterior, and normally terminates short of the pressure plate 22. A cam follower 24 serves as a head for the lower end of follower rod 23 and may be of any suitable configuration. In the present instance, this conveniently is cylindrical, to conform with the surface contours of the co-operating control cam. This control cam 25 preferably is of the disc type and operates off of a manually-rotated control shaft 26. Shaft 26 is operated through a manual crank or other suitable control means, conventional in the art, and not shown.

The function of the manual control means, including shaft 26, cam 25, follower 24, etc., is to regulate the piston valve initially so as to provide desired calibrated cracking of the admission port 14 for typical mean load conditions prevailing for the operation undergoing energization.

It is desired to provide in some conventional manner that the cam setting be self-locking in nature, so that variation in load conditions will not produce corresponding variation in the cam positioning until the operator intentionally manually readjusts the same. Either a shaft lock is provided for shaft 26, or the design of the cam profile allows for this requirement, while at the same time providing for calibrated cracking of the admission port 14. Either such expedient is

conventional in the art and needs no further description here.

Snugly received in the bore 11 near the lower end thereof is a cylindrical box-like housing 27 which is machined and finished for ready reciprocation within the bore 11. It is made fast to follower rod 23, which extends centrally there-through, in any suitable manner at 28, as by welding, soldering, spinning or the like.

Pressure plate 22 is disposed interiorly of this box-like hollow cage member 27 through an opening 27A provided centrally in the top of this latter member. The pressure plate 22, which is attached to the piston rod 21 in any desired and suitable manner itself entirely conventional, is inserted into cage 27 by the use of any suitable manufacturing technique, as by radially slitting the top surface 27B to cage 27 to form wing portions, bending these wings backwardly to provide an opening, inserting the plate 22 within the hollow cage, and thereupon folding down the wing portions, and if desired, sealing them as by seam welding, soldering or the like.

Interposed between the pressure plate 22 and the interior bottom 27C of cage 27 is a coiled spring 28B. Spring 28B encircles follower rod 23 within cage 27 and is secured to the bottom interior of such cage, on bottom plate 27C, in any suitable manner at 29 as by soldering, welding, or the like. This spring 28B rests freely against the under side of follower plate 22 and serves as a resilient buffer therefor.

I provide what I term an annular stub 30 within bore 11, nearly centrally thereof and about piston rod 21 which extends freely there-through a central bore 30A. This stub 30 is secured to the walls at 31 of the casing 10 in any suitable manner at its outer periphery, as by welding, soldering, or the like. I dispose stub 30 at a point within bore 11 which is appreciably above the maximum upward travel of hollow cage 27 under the impetus of cam 25. Figure 1 discloses the cage 27 at its maximum upward position, being so positioned by cam 25, corresponding to full closure of inner port 14.

Stub 30 serves as a reaction plate against which a coiled spring 32 reacts in its downward thrust against the top surface 27A of cage 27. Spring 32 is disposed concentrically about piston rod 21, and is secured at 33 to the top side 27B of cage 27 in any suitable manner as by soldering, welding, or the like.

In profile or section, the generally cylindrical skirted piston valve 20 has what I term a generally curved side wall or generatrix 34 which is of calibrated contour, and which may have a surface of hyperbolic, parabolic, or other configuration developed either in accordance with natural or empirical equations, the prerequisite being satisfied, however, that calibrated and desired cracking of the admission port 14 is accomplished upon movement of piston valve 20. To illustrate, were the piston valve 20 to be fully cylindrical, then the admission port 14 would remain fully closed until at such time as the extreme top surface of the valve 20 began to clear the port 14. At that time, and not before that point is reached, port 14 would begin to open quickly, with a rapid surge of power and with a minimum of wire drawing. Sensitive control, however, would be absolutely lacking.

By properly contouring the walls of the valve 20, however, and upon calibrated departure from the straight line generator of the true cylinder, and by employing the curved generatrix

hereinbefore described, it is possible to shape valve 20 so that the cracking of the port 14 is sensitively in conformance with the instantaneous load requirements. That is, when the valve 20 is moved but a slight distance down from its closed position shown in Figure 1, generatrix 34 still remains closely adjacent port 14 and cracks the same but a slight amount. Only a small amount of energizing fluid courses across the bore 11 to the outlet port 15, which likewise is cracked but a small extent. Only minimum loads can be serviced.

A further like increment of vertical movement, however, presents a portion of generatrix 34 opposite admission port 14 which is more remote from the latter. An appreciably greater increment of energizing fluid thereupon courses port 14 and bore 11 to port 15 and thence to the load. Upon still further downward movement of piston valve 20 a like increment, the corresponding cracking of port 14 becomes much greater, in a manner which will be readily understood. Due to the departure of generatrix 34 from proximity to port 14, a progressively greater quantity of hydraulic fluid will course the hydraulic load circuit. It will thus readily be seen that the opening of admission port 14 is a power function of the increment of vertical displacement of valve 20. It is unnecessary, however, that this power exponent be an integral number, and its value determines the precise shape of the generatrix 34.

It is desirable to repeat at this point that as the fluid from the constant output power source, not shown, is diverted through admission port 14 to the variable load, the quantity of fluid returned directly to the low pressure, inlet side of the power source is correspondingly diminished. This is because the return line from the load is returned in closed hydraulic circuit to the system reservoir. The total quantity of fluid per unit of time by the main, constant delivery pump, however, remains always substantially the same. It is in this closed hydraulic circuit that my new metering valve is inserted in series connection.

To understand the operation of my new metering valve under manually-operated cam control, let us assume that no-load conditions exist, corresponding to the showing in Figure 1. Admission port 14 is entirely closed. The shaft 26 is rotated so that cam 25, in its topmost position, forces follower 24, follower rod 23, and cage 27 to their uppermost extent relative to bore 11. Spring 28B bears resiliently against pressure plate 22 just to the under side of piston rod 21, and forces this pressure plate up against the under side of top plate 27B of cage 27. Against the compression of spring 23 thrusting off of annular stub 30, the plate 22, piston rod 21, and piston 20 are forced by spring 28B, carried in cage 27, to the top of bore 11, the substantially cylindrical lower part 35 of the skirt of piston 20 then being opposite admission port 14.

During rest conditions, spring 28B ensures that piston 20 is retained at or near the upper limit of its travel in bore 11, effectively closing admission port 14.

What has been described to this point comprises simply the control of the piston valve 20 relative to the admission port 14 for any given manual setting of the piston valve 20 through the cam 25 in a manner fully detailed in the foregoing. It is of course apparent that the widest and most sensitive latitude of adjustment is available through the manual control. That is, in continu-

ous as distinguished from step by step control, the piston valve can be adjusted from one extreme position wherein the admission port 14 is fully closed, substantially to its other extreme position, wherein the admission port 14 is nearly opened.

Throughout the range of intermediate positions and in dependence upon the momentary valve position a corresponding fraction of the constant output delivery of the hydraulic powering unit, not shown, is passed through admission port 14 to the hydraulic load. For constant load corresponding to the particular cam setting, no more regulation is required. If load variation is infrequent and slow, then rather satisfactory results can be achieved simply by manually varying the setting of cam 25, and hence of piston 20, to correspond with the variations in load demands.

If, however, the load variation be frequent, unpredictable, and somewhat rapid, then not only is the required supervision of the control so great as to render its use for such purpose impractical, but even if continuous supervision is provided, the control is not sufficiently responsive or sensitive to accommodate for the rapid fluctuations which occur. A robot or automatic control is indicated.

To provide for such automatic regulation, I construct the top surface of the piston valve 20 at the break 34A at the top of generatrix 34, with a substantially plane-surfaced header 35, the outer periphery of which is substantially of the same radius of curvature as, and conforms with the periphery of, the bore 11 and is smoothly slidable therein. At its periphery, an annular upstanding flange 35A is provided, which forms, together with bore 11 and end wall 13, a circular recess 36.

This top plate 35 serves to transmit leakage pressure which occurs in the conduit 17, and is transmitted to the topside of such plate, to the pressure plate 22 at the lower extremity of piston rod 21. There, this excess pressure is algebraically compounded on spring 28B and overcoming the same, serves to urge it downwardly, against the upward force exerted by the spring itself. Pressure plate 22 thereby forces the spring 28B downwardly to a certain extent, the maximum extent of which movement is determined by the contact of plate 22 with the upper end of cam follower rod 23.

It has been stated that pressure on the top of valve plate 35 and hence in recess 36, which fluid pressure is exerted through port 14 from branch conduit 18 and conduit 17, is the result of pressure build-up in this last-mentioned conduit. This pressure, which may accurately be considered here as back pressure or leakage pressure, depending upon the point of view, results from increase of the load for any particular manual setting of the cam 25. Outlet conduit 17 from outlet port 15 services any particular load-energizing device such as a hydraulic motor, not shown, all as has been more fully set forth hereinbefore. Upon momentary increase of the torque requirements of the output shaft of such hydraulic motor, for example, a back pressure builds up which causes increased fluid leakage about the moving parts of this motor. In response thereto, a back pressure builds up in the conduit 17, and there being no channels for pressure relief other than through the branch conduit 18, this increased pressure is imposed on the top side of valve 20, at plate 35. The latter is moved downwardly against the force of the calibrated spring 28B in the manner narrated hereinbefore, to crack further the admission port 14 in graduated manner in

dependence upon the momentary increase in load. Thus, leakage compensation, and hence load compensation, is provided for in highly sensitive and accurate manner, within substantial limits, for any particular manual setting of the valve 20 through cam 25. The manner in which this operation is brought about is clearly illustrated in Figures 2 and 3, a comparison of which will make clear both the purpose and operation of my new invention.

In Figure 2, cam shaft 26 has been manually rotated to move cam 25 through a ten degree angle. Inasmuch as the control of piston valve 20 at this time is solely from the manually-controlled cam, this valve cracks admission port 14 only to the extent provided for by the cam control means. This corresponds to minimum load conditions for such valve setting, on the hydraulic motor. Only enough fluid courses port 14 from the hydraulic pump as is required to service such load.

Let us assume, however, that this load on the hydraulic motor serviced through this metering valve momentarily increases, and that such load increase is to be accommodated by robot control through my automatic metering valve, without further adjustment of the cam 25. This automatic action results, as has been disclosed, from leakage pressure build-up in channels 17 and 18. When this pressure is exerted on the top side 35 of piston valve 20 (Figure 3), it moves this latter downwardly to a graduated extent, the maximum limit of which is determined by the contact of pressure plate 22 at the end of piston rod 21 with the top end of cam follower rod 23. This limit position is clearly shown in Figure 3, wherein it is also to be noted that a space 37 here separates follower plate 22 from the top surface 27B of hollow cage 27. Moreover, it is to be noted that in this position it is the increment 34C of the curved generatrix 34 which is opposite admission port 14 rather than the portion 34B thereof, as is true in the case of Figure 2. Accordingly, substantially greater unit volume of fluid courses port 14, corresponding to increased load demands on the hydraulic motor.

My new construction responds admirably to the accurate and sensitive control of widely varying loads, and this with operation at maximum efficiency of a constant delivery hydraulic pump. As is clearly disclosed in my said parent application, it is entirely feasible, where desired, that in addition to being of constant output, this energizing source may have constant direction characteristics, in which case valves may be employed, for determining the direction of energization of the load-energizing motor.

In the instance that the load demand on the system increases beyond the robot control compensation permitted by the movement of piston 20 through the distance 37, then further compensation to a new normal or mean value can be achieved through further throw of the cam 25 into a new position. After such further adjustment, the cage 27 is lowered to a new point, so that contact is restored between pressure plate 22 and top surface 27B of cage 27. Compression is removed from spring 28B. Spring 32 insures that the cage 27 closely follows the cam 25, and tends to overcome any frictional resistance of the cage 27 relative to the bore 11 and to assist in centering the cage therein. Further robot control can now be achieved, in a manner and mode fully corresponding to that described.

The rugged and simple nature of my new construction is readily apparent from the foregoing. It may be further readily observed from my disclosure that in direct, simple, and reliable manner the several objects of my invention, including among them automatic and sensitive leakage- and load-responsive regulation and high sensitivity with wide control range, are accomplished, together with many thoroughly practical advantages.

Inasmuch as many embodiments may be made of my invention, and since many modifications may be made to the embodiment herein disclosed, it is to be understood that the foregoing disclosure is merely illustrative, and not in any sense to be construed as limitative.

I claim as my invention:

1. A hydraulic metering valve for supplying energizing fluid to a variable load, comprising in combination, a valve casing having a central bore and having inlet and outlet ports therein, a rodless piston valve displaceable in said bore, and having a working surface of calibrated contour for calibrated passage across said inlet port, said piston valve having an enlarged head portion and said casing having a channel leading from the outlet side of said bore to the top end of said bore, said piston rod terminating in a pressure plate, a manually-operable cam for initially setting said piston valve to open the inlet port a predetermined amount, a cam follower rod carrying a hollow cage member, slidable in said bore, and within which member the pressure plate of the piston rod is received, a spring extending between the top side of said hollow member and a stud provided within said casing, and a second spring provided between said pressure plate and the bottom interior of said hollow member, normally separating said two rods and yieldable to back pressure on said outlet in excess of a certain value transmitted to the top side of said piston thus displacing the latter and providing further admittance through said inlet port.

2. A hydraulic metering valve for supplying energizing fluid to a load in accurately metered quantities, comprising in combination, a valve casing having inlet and outlet ports therein, a valve piston displaceable in said casing for modifying the effective opening of at least one of said ports and having an end susceptible to a displacing pressure of fluid, and means for moving said piston to desired controlling relation with at least one said port and including adjacent members spring urged one toward the other for movement in unison to initially set the piston and susceptible to relative movement by pressure of fluid in excess of a certain value exerted on said piston end, for modifying the specific initial piston setting.

3. A hydraulic metering valve for supplying energizing fluid to a load in accurately metered quantities, comprising in combination, a valve casing having inlet and outlet ports therein, a valve piston displaceable in said casing for modifying the effective opening of at least one of said ports and leaving an end susceptible to a displacing pressure of fluid, means for moving said piston to desired controlling relation with at least one said port and including adjacent members spring urged one toward the other for movement in unison to initially set the piston and susceptible to relative movement by pressure of fluid in excess of a certain value exerted on said piston end, for modifying the spe-

11

cific initial setting, and spring means biasing both of said overlapping members in favor of one direction of movement of the piston to initial setting.

4. A hydraulic metering valve for supplying energizing fluid in accurately metered quantities, comprising in combination, a valve casing having a piston bore and inlet and outlet ports through the wall thereof, a rod valve piston fitting within said bore for modifying the effective opening of at least one of said ports and having an end susceptible to pressure of said energizing fluid, and means for variably setting said piston to desired controlling relation with at least one said port and including a member receiving the valve piston rod and slidable along the casing bore for adjusting the desired position of the piston, said piston rod having a biased

12

plate overlapping the inside surface of said receiving member and being separable from the same by pressure in excess of a certain value on said piston end in a given setting of the piston.

ALBERT RIVINGTON STONE.

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