United States Patent

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[54] **HYDRAULIC CYLINDER WITH PRESSURE CONTROL**
10 Claims, 5 Drawing Figs.

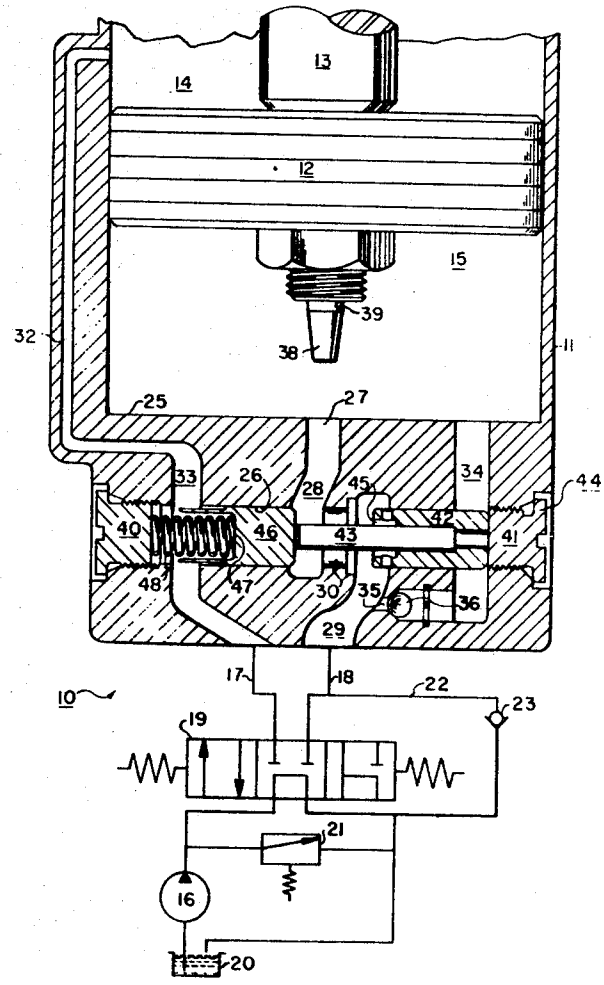
[52] **U.S. Cl.** **91/26,** 91/396, 91/413, 91/420, 91/421
[51] **Int. Cl.** **F15b 15/22,** F15b 13/07
[50] **Field of Search** 91/420, 394, 396, 421, 25, 26, 413

[56] **References Cited**
UNITED STATES PATENTS

3,523,490	8/1970	Bianchetta	91/420
3,054,384	9/1962	Darling	91/26
3,129,720	4/1964	Allen et al.	91/420 X
3,272,085	9/1966	Hajma	91/420
3,398,650	8/1968	Garnjost	91/421

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ABSTRACT: A pressure-actuated control valve which is responsive to an increase in pressure in the contracting or discharge chamber of a hydraulic actuator to restrict discharge flow from said chamber thereby reducing the velocity of the movable barrier therein. The operation of the valve is conditional on inlet pressure as it is held in a non-restricting position by pressure fluid being admitted to the intake or expanding chamber and moves to a flow-restricting position only when a predetermined loss of pressure occurs in the intake chamber. An additional valve may be included in the system which places the actuator discharge and intake supply passages in communication to shunt fluid therebetween when discharge chamber pressure increases to a predetermined value.



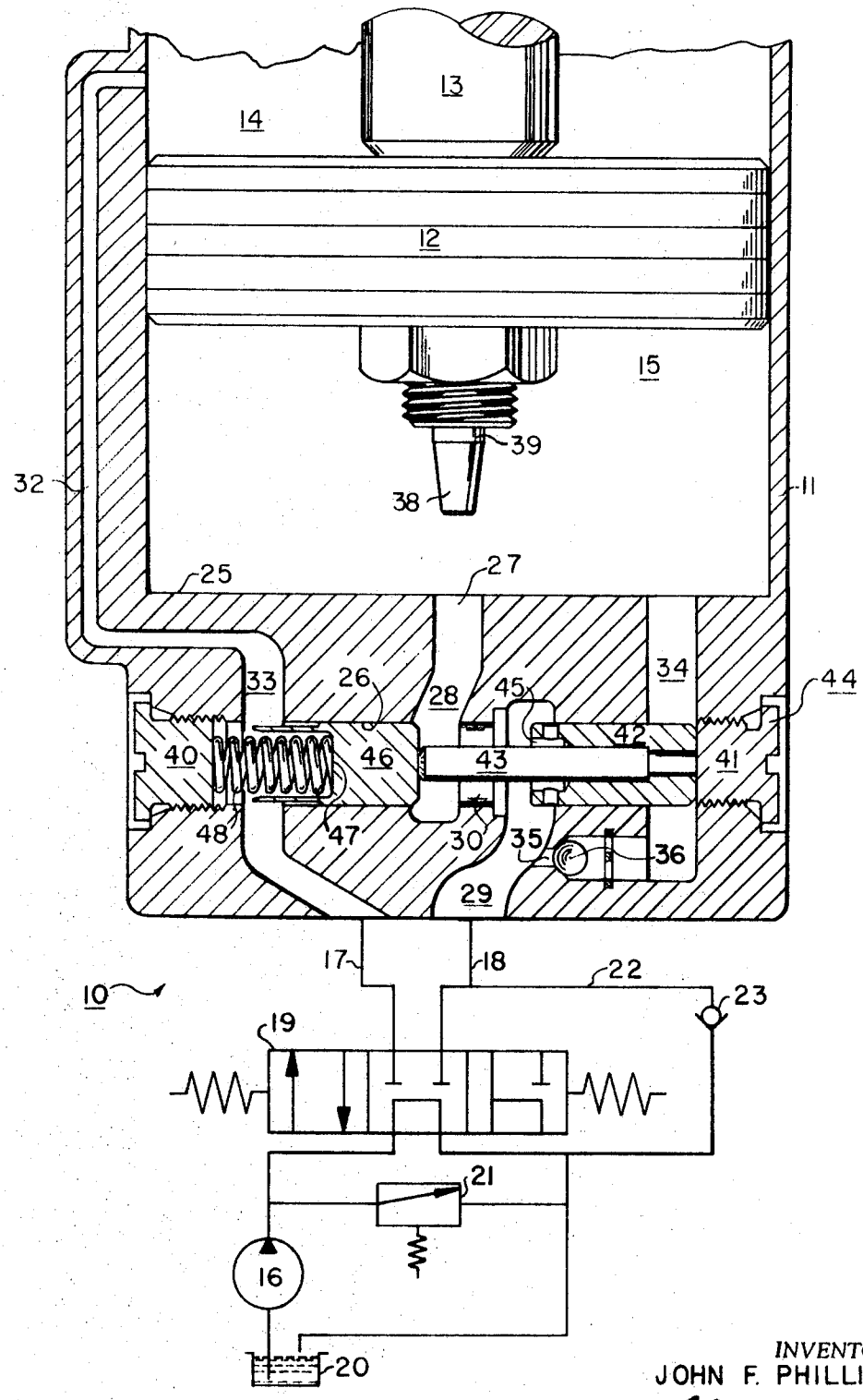


FIG. 1

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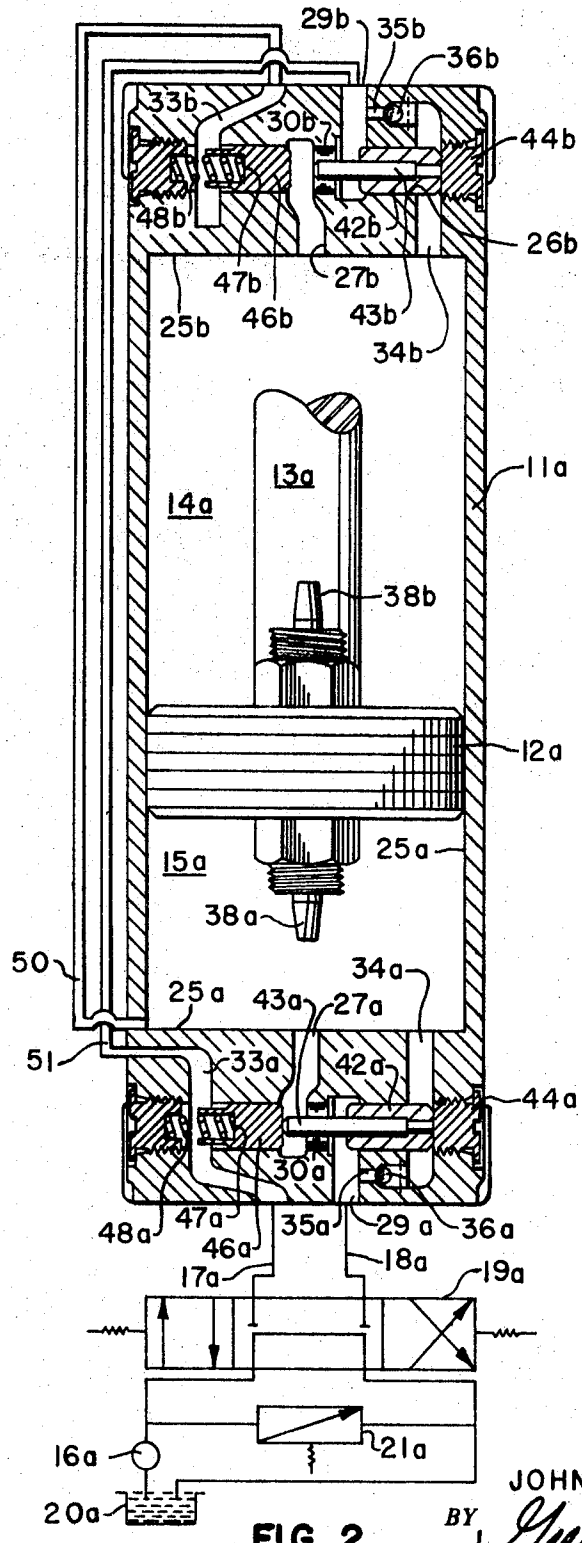


FIG. 2

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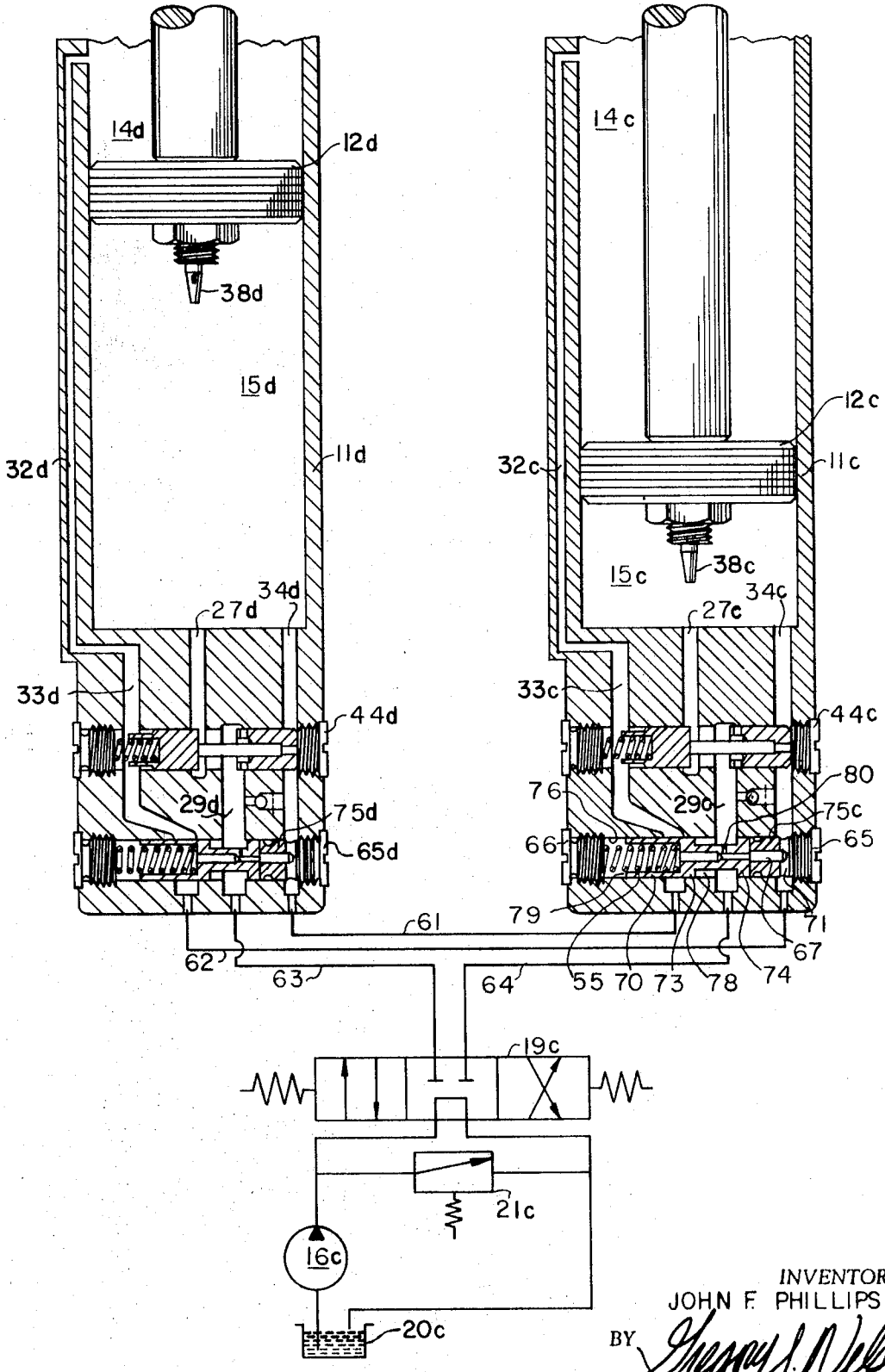


FIG 3

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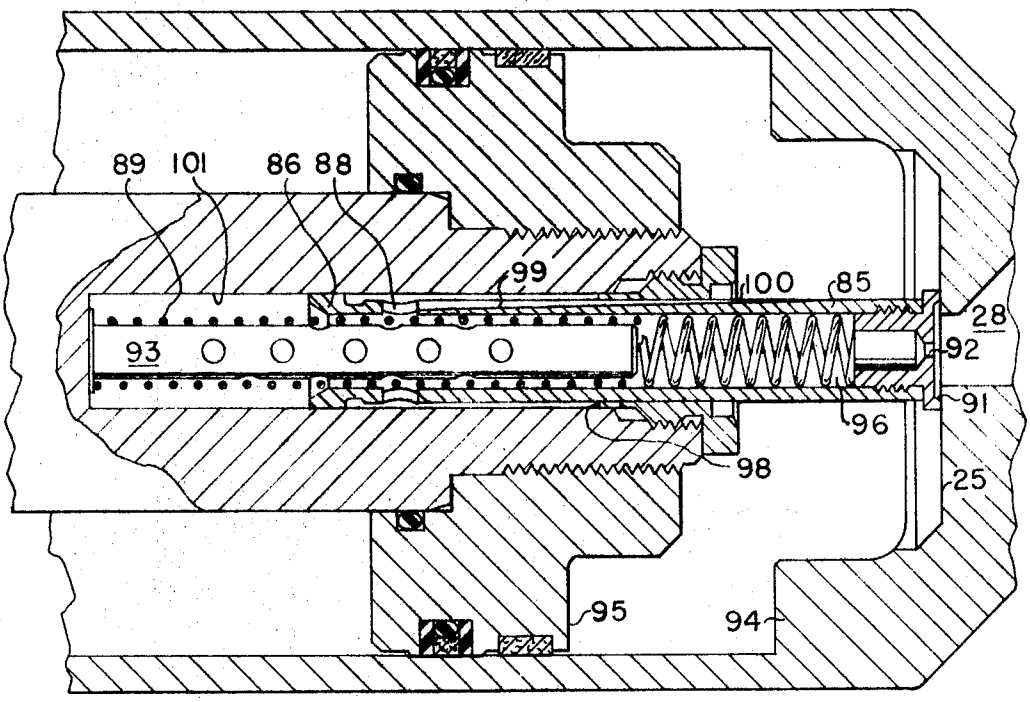


FIG. 4

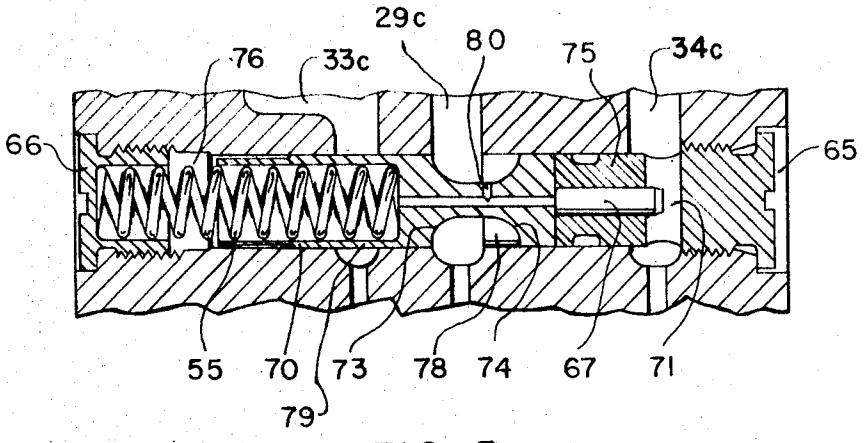


FIG. 5

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HYDRAULIC CYLINDER WITH PRESSURE CONTROL

This invention relates to hydraulically actuated motors of the linear or rotary cylinder type. More particularly, this invention relates to a pressure-actuated valving system, for use with both single or tandem cylinder systems, for controlling the velocity at which the piston may travel during reciprocation, the velocity regulation being conditional upon inlet pressure. Further, my invention provides, in several embodiments, a valve actuated by discharge chamber pressure for shunting fluid between the inlet and discharge passages to assist in velocity control and cushioning.

Economy and efficiency of operation demands that hydraulically operated mechanisms such as booms, backhoes, and the like, perform their work functions at maximum velocity. However, excessive velocity of the movable piston or vane can be harmful causing a loss of working pressure and cavitation in the expanding chamber. Further, in high-speed operation at the end of the stroke, the piston or vane must be brought to a controlled stop. Inertia loads add kinetic energy in the system which must be absorbed in a way to prevent shock loads and sudden pressure buildups that can damage the cylinder and hydraulic system.

Several schemes for governing cylinder piston velocity can be found in the prior art. For example, it is known to provide a biased floating-spool-type velocity-limiting member positioned in the cylinder inlet-outlet passage. The floating spool is movable to vary the permitted volume flow of fluid outward through the inlet-outlet passage in response to a pressure differential arising due to flow velocity out of the contracting cylinder chamber or in response to a predetermined rise in discharge chamber pressure.

One disadvantage of a simple pressure-controlled flow restrictor is that it serves to limit the velocity of the piston even if no pressure drop occurs in the expanding cylinder chamber. Although the fluid-supply pump of the system is keeping up with the load and is able to meet the volume requirements of the cylinder, the control may act to restrict piston travel. As mentioned above, it is desirable to achieve maximum piston velocity limited only by the ability of the system to maintain an adequate supply of pressure fluid to the cylinder.

It is, therefore, a broad object of the present invention to provide a velocity control for hydraulic actuators which serves to restrict discharge flow and which is operatively conditional upon the fluid inlet pressure to the actuator.

Another broad object of this invention is to provide a linear tandem cylinder system which is provided with means for governing the velocity of piston travel within the cylinders.

Another object of this invention is to provide an improved means of cushioning deceleration in a hydraulic actuator as the movable barrier approaches the extreme end of its travel.

A more specific object is the provision of an improved progressively restricting metering member which acts to restrict the outward volume of flow from the contracting cylinder chamber to provide a cushioned piston deceleration.

These and other objects of the present invention will become clear when taken in conjunction with the specification and drawings in which:

FIG. 1 is a partial cross-sectional view of a linear cylinder provided with piston velocity control means and its associated hydraulic circuit;

FIG. 2 shows, partially in pictorial and partially schematic, a double-acting cylinder including the velocity control system;

FIG. 3 is a combined pictorial and schematic of a tandem hydraulic cylinder system, each cylinder being provided with the automatic piston velocity control valve and a second shunt valve to assist in velocity control and cushioning;

FIG. 4 illustrates a cylinder having a piston equipped with an alternate embodiment of means for progressively restricting discharge flow; and

FIG. 5 is an enlarged broken-out view of the velocity control spool shown in FIG. 3

Generally, according to the present invention, a floating restricting valve member is provided in the actuator in a passage transverse to and intersecting the cylinder inlet-outlet passage which when actuated retards discharge flow through the inlet-outlet passage. One end of the floating restrictor is subject to the pressure existing in the contracting cylinder chamber. The floating valve is biased in a fully nonrestricting position by inlet or supply pressure fluid being admitted to the expanding piston chamber. When inlet or supply pressure drops below a certain predetermined value and expanding chamber pressure increases to a predetermined value, the float member is responsive to the differential pressure acting at its ends to move into the inlet-outlet passage to restrict outlet flow from the cylinder, causing a pressure increase in the contracting cylinder chamber which acts to retard the movement of the piston. The floating spool flow restrictor of the present invention is particularly adaptable for use with a tandem cylinder system.

An additional feature of my invention includes a flow-restricting or metering member which is movable with the piston of a linear cylinder to progressively restrict flow from the cylinder as the piston nears the end of the stroke providing a gradual, cushioned stop. Also a shunt valve may be included which, when actuated by pressure in the discharge chamber, places the inlet and discharge passages in communication to assist in the velocity control and cushioning phases.

Turning now to the drawings in which the same numerals throughout the views identify similar elements, FIG. 1 shows a hydraulic circuit generally indicated as **10**, including linear cylinder **11** having a movable piston **12** and rod **13** connected to an external work load, not shown.

Cylinder **11** is divided by piston **12** into chambers **14** and **15**. Hydraulic motive fluid is supplied by pump **16** through lines **17** and **18**. Valve **19** controls the direction of movement of piston **12** by admitting fluid through either line **17** or **18** to the cylinder. For example, if valve **19** is moved to the right, fluid from pump **16** enters chamber **14** through line **17** causing inward travel of piston **12** and retraction of rod **13**. Fluid is exhausted to reservoir **20** from chamber **15** through line **18**. Relief valve **21** protects the system for overpressurization bypassing fluid, when necessary, to reservoir **20**.

The novel velocity control feature is contained within end wall **25** of cylinder **11**. Within the end wall is transverse passage **26** which is intersected by a passage designated as **27**. Passage **27** connects fluid conduit **18** with chamber **15** and is made up of upper duct **28** and lower duct **29** which communicate through a portion of transverse passage **26** designated as **30**. It will be seen that section **30** acts as an orifice to restrict flow through the passage and cause a corresponding drop in the fluid pressure through the passage. Fluid line **17** is in communication with cylinder chamber **14** through transfer duct **32** and passage **33** which is located in end wall **25** and is in communication with the left end of transverse passage **26**.

Another short duct **34** also intersects passage **26** and places the right end of this passage in communication with cylinder chamber **15**. Ducts **29** and **34** are in hydraulic communication through bypass **35** and across one-way check valve **36** which permits flow from duct **29** into duct **34** but prevents reverse flow.

Axially aligned with duct **28** is tapered pin **38** integrally mounted on the chamber **15** side of piston **12**. It will be seen that as piston **12** nears the end of its downward stroke approaching end wall **25**, pin **38** will enter duct **28**. Due to the taper of pin **38**, flow through duct **27** will progressively be snubbed or metered until land **39** engages duct **28** to completely restrict flow therethrough. FIG. 4 discloses an alternate embodiment of the progressively restrictive orifice which will be described in greater detail hereafter.

Transverse passage **26** is closed at both ends by plugs **40** and **41** in threaded engagement in cylinder head **25**. The restricting valve, generally designated by numeral **44**, is slidably positioned in passage **26** and includes valve spool **42** which in a normal nonrestrictive position has its right end subject to pressure existing in duct **34** and its left end subject to pressure downstream of restriction **30**. The positioning of spool **42** is

controlled by biasing spool **46** acting through pin **43**. Biasing spool **46** abuts the left end of pin **43** and is provided with a recessed seat **47** at its opposite end which receives spring **48** acting against plug **40** and spool **46**. As will be explained in more detail hereinafter, the force of spring **48** alone is insufficient to urge spool **46**, pin **43**, and valve spool **42**, to their extreme rightward positions against the pressure force acting against the spool. However, the pressure exerted by fluid in duct **33** acting against seat **47** of spool **46** assists the force of spring **48**, and only when this pressure drops below a certain predetermined value will the spool mechanism **44** be allowed to move to the left to restrict flow from the cylinder. Biasing spool **46**, pin **43**, and spool **42**, may be affixed to each other, making spool **44** integral, or spool **46** may simply abut pin **43**, in either case operation of valve **44** is essentially the same.

The left end of spool **42** is provided with a slotted axial opening at **45**. Lateral openings extend into slot **45** and serve as metering orifices to stabilize operation as spool **42** moves to the left and begins to restrict flow through passage **30** into duct **29**.

OPERATION - FIGURE 1

When directional control valve **19** is moved to the right (straight through position), pressure fluid from pump **16** is admitted to cylinder chamber **14** through line **17** and passages **33** and **32**, causing piston **12** to retract with its external work load.

Fluid pressure delivered by pump **16** through passage **33**, along with the force exerted by spring **48**, holds spool assembly **44** in the right nonrestricting position. As piston **12** moves inwardly, fluid is exhausted through passage **27**, return line **18**, to reservoir **20**. Fluid discharge through orifice **30** in passage **27** will result in a pressure drop and a differential pressure between chamber **15** and duct **29**. This same differential will also exist, since **34** is at chamber pressure, across spool **42**. Although the right end of spool **42** is subject to the pressure existing in chamber **15** through communication therewith by passage **34**, spool **42** is normally held against right hand plug **41** by spring **48** and the inlet pressure existing in chamber **33**, and does not restrict flow through passage **27**.

If the momentum of the external load causes piston **12** to accelerate, the fluid flow velocity through passage **27** will increase, increasing chamber pressure and the pressure differential across spool **42**. If the piston travel velocity becomes excessive, that is the volume of chamber **14** is expanding more rapidly than the flow capacity of pump **16**, a pressure loss occurs in chamber **14**, duct **32**, passage **33**, and line **17**. This pressure loss, since spring **48** is by itself insufficient to rightwardly bias spool **42** against the pressure force applied to its right end allows biasing spool **46**, pin **43**, and valve spool **42** to move to the left. Restriction of flow through orifice **30** will immediately begin as the left end of valve **42** enters section **30** of the transverse bore **26**.

As the discharge flow is restricted by spool **42**, the pressure in duct **28**, chamber **15**, and duct **34** will increase, thus stopping leftward movement of valve **44** and resisting the inward movement of piston **12** and retarding its travel. As piston **12** is decelerated, the discharge volume of pump **16** again becomes adequate to meet the system requirements and pressure increases in chamber **14** and accordingly in passage **33**. The increase in pressure in passage **33** acting against the left end of spool **46** at seat **47**, along with the force exerted by spring **48**, will return spool **46** to the right, which through pin **43** urges spool **42** to the position shown in FIG. **1**, allowing unrestricted flow through discharge passage **27**.

As the piston travel continues toward end **25**, taper pin **38** will enter the upper end of passage **27** progressively reducing the size of the discharge passage opening. The reduction causes a pressure increase in chamber **15** which quickly decelerates the piston velocity. When land section **39** of the pin enters in passage **27**, all flow through the passage is blocked.

It should be noted that the piston velocity and cushioning controls are independent of each other and need not be used conjointly. The operation of the velocity control does not depend upon progressively restrictive orificing near the end of the piston stroke.

To return the cylinder to an extended position, directional control valve **19** is placed in its right-hand position. The weight of the load or some mechanical force acts to extend rod **13**, and piston **12** moves upward. Oil in chamber **14** is released to reservoir **21** through passages **32** and **33** and line **17**. Pump oil is recirculated to the reservoir. To prevent cavitation in fluid-filled chamber **15**, bypass line **22** with one-way check valve **23** allows supplemental flow into chamber **15** through passage **27**. During initial movement of piston **12** away from head **25**, flow into chamber **15** will be bypassed from passage **29** across check valve **36**. Once passage **27** is not restricted by land **39** of taper pin **38**, supplemental fluid will pass through passage **27** directly into chamber **15**.

DESCRIPTION - FIGURE 2

This FIG. is similar to FIG. **1**, but includes the piston velocity control in conjunction with a double acting cylinder. In the description of this and the subsequent FIGS., the same reference numerals indicate similar elements but are distinguished in different parts of the FIG. with an appended letter. In the design of cylinder shown in this FIG., both chambers **14*a*** and **15*a*** can be supplied with pressure fluid, whereas the cylinder shown in FIG. **1** is single acting and can be operated hydraulically in one direction only. The valving providing the piston velocity control in both the rod end **25*b*** and the head end **25*a*** of cylinder **11*a***, is the same as that shown and described with reference to FIG. **1**. The upper end of passage **33*a*** is in hydraulic communication with duct **29*b*** of passage **27*b*** through conduit **51**. Cylinder chamber **15*a*** is in hydraulic communication with passage **33*b*** through conduit **50**. Passage **33*b*** intersects transverse bore **26*b*** but is closed at its inner end and does not communicate with cylinder chamber **14*a***.

Chamber **14*a*** is also provided with metering pin **38*b*** which is aligned to enter passage **27*b*** to progressively restrict discharge flow to decelerate and cushion the piston smoothly at the end of its stroke. Because of the central axial piston rod **13*a*** which extends through end **25*b*** of the cylinder, both the bore **26*b*** and cooperating valving spool assembly **44*b*** in head **25*b*** as well as the metering pin **38*b*** and passages **33*b***, **27*b***, and **34*b***, must be radially displaced from a center position in the cylinder so as not to interfere with the piston rod.

OPERATION - FIGURE 2

The operation of the differential pressure-controlled flow-restricting valve spools **44*a*** and **44*b*** is identical to that described with reference to the FIG. **1** embodiment. The spool mechanisms serve to limit the velocity of the piston in either directions of piston travel, when the flow requirements to the expanding cylinder chamber are not being met by the system as indicated by a drop in the pump supply pressure to the expanding chamber.

If control valve **19*a*** is actuated to the right (straight through position), fluid pressure is delivered to cylinder chamber **14*a*** through line **17*a***, passage **33*a***, conduit **51** and passage **27*b***. Discharge of fluid from chamber **15*a*** is through passage **27*a*** in head end **25*a*** and line **18*a*** to reservoir **20*a***. If the velocity of piston **12*a*** becomes excessive and a loss of supply pressure occurs, it is also reflected in a pressure drop passage **33*a***. This allows the pressure impressed against the right end of **42*a*** to overcome spring **48*a*** to move spool **42*a*** to a flow-obstructing position in orifice passage **30*a***. The obstructing of discharge flow causes a buildup of pressure in chamber **15*a***, decelerating the piston **12*a***. Piston deceleration is reflected by a pressure increase in chamber **14*a*** and accordingly passage **33*a*** and against spool seat **47*a*** which acts to reposition spool **42** to its right-hand, nonrestricting position.

Similarly to extend the piston rod 13*a*, valve 19*a* is moved to the left (crisscross position), pressurizing chamber 15*a* through line 18*a* and passage 27*a*. Initially, if the rod is fully retracted, pressure fluid will be bypassed through duct 35*a* and check valve 36*a* until metering pin 38*a* moves out of passage 27*a* and fluid is allowed to flow therethrough. Pump discharge pressure in chamber 27*a* is sensed in passage 33*b* through conduit 50. Discharge fluid from chamber 14*a* flows through passage 27*b* and by orifice 30*b*, through conduit 51, passage 33*a* and line 17*a* to reservoir 20*a*. If the velocity of piston 12*a* becomes excessive and a pressure loss occurs in chamber 15*a*, spool 46*b* will be allowed to move to the left as a result of the corresponding pressure drop in passage 33*b*.

The movement of spool 46*b* then allows the pressure differential that is imposed across spool 42*b* then allows the pressure differential that is imposed across spool 42*b* to act to immediately move spool 42*b* to obstruct flow at orifice 30*b*, resulting in a pressure increase in chamber 14*a* to oppose the piston travel. When pressure in chamber 15*a* is restored, spool 46*b* is forced to the right, also forcing pin 43*b* and metering plug 42*b* to the right, again allowing unrestricted discharge flow through 30*b*.

As piston 12*a* travels toward end wall 25*b*, metering pin 38*b* also continues its travel and begins to enter chamber 27*b*. As the piston movement continues, pin 38*b* acts to progressively restrict flow out of chamber 14*a*, cushioning the deceleration of the piston.

FIGURE 3

FIG. 3 shows the piston velocity control in a tandem cylinder system. In addition to the velocity control spool assembly, each cylinder is also provided with another valve spool which assists the variably orificed discharge in cushioning the piston at the end of its stroke as well as providing a flow-diverting or shunting function when a substantial pressure exists in the discharge chamber of the cylinder. Identical elements of the two cylinders of this FIG. and of other FIGS., are identified by the same number, but are distinguished by the appended small letters "*c*" and "*d*."

In FIG. 3, two identical cylinders 11*c* and 11*d* are connected to operate in tandem and receive fluid pressures from pump 16*c* through directional control valve 19*c*. In tandem the cylinders operate synchronously so that one of the cylinders is retracting at the same rate the other is extending. Tandem arrangement is particularly suited for laterally swinging a boom of a backhoe or the like about a pivot point. Proper control of the swing velocity of backhoes is extremely important to prevent high-inertia loads from seriously damaging equipment.

In the present system, in addition to valve spools 44*c* and 44*d*, which as previously described in detail function to control piston velocity, cylinders 11*c* and 11*d* are provided with a second, separate valve spool assembly 75*c* and 75*d* respectively. Looking at right-hand cylinder 11*c* and FIG. 5 which shows the spool in greater detail, with this description applying equally well to cylinder 11*d*, spool bore 76 transversely intersects the lower extension of passages 33*c*, 29*c* and 34*c*. The left end of the bore is closed by threaded plug 66. The right end is closed by plug 65, also threaded into the cylinder. Plug 65 extends into bore 76 to a point midway between passages 34*c* and 29*c*, and is provided with a cross bore 71 in through communication with passage 34*c*.

A slidable pin 67 is centrally aligned within plug 65 and in a normal position extends from within cross bore 71 to the left end of the plug. Spool element 70, in a nonactuated position as shown, is held in abutment, partially by spring 55 of predetermined size, against the left end 2c, plug 65. The left end of spool 70 is bored at 79 to seat the spring 55 which extends between the seat and plug 66. Annular groove 78 axially defined by lands 73 and 74, is provided in the body of spool 70. Seat 79 of spool 70 is in communication with the pressure existing in passage 29*c* through hole 80 opening into groove 78. In a normal unobstructing position, the annular groove 78 permits unobstructed flow through passage 29*c*, and spool 70 as constructed does not block flow through passage 33*c*. When spool 70 is leftwardly actuated so that spool land 73 moves at least into passage 33*c*, passage 33*c* is placed in communication through annular groove 78 with passage 29*c*, while flow is still permitted unobstructed through passage 33*c*.

OPERATION - FIGURE 3

As mentioned above, cylinders 11*c* and 11*d* are arranged in a system to operate synchronously in tandem. When the manual control valve 19*c* is actuated to the right (straight through position), fluid from pump 16*c* is introduced into chamber 15*d* of cylinder 11*d* through line 63 and passage 27*d* to move piston 12*d* upward in the cylinder and extend piston 12*d*. Pressure fluid also flows out of chamber 15*d*, through passage 34*d* and line 61 into passage 33*c*, and through line 32*c* into chamber 14*c* of cylinder 11*c*, imparting an inward travel to piston 12*c*. Discharge fluid from chamber 14*d* flows through line 32*d*, passage 33*d*, line 62, and passage 34*c* to chamber 15*c*. The discharge from chamber 15*c* flows to reservoir 20*c* through passage 27*c* and line 64.

If a condition arises under which there is a loss pressure in chamber 14*c* and pistons 12*c* and 12*d* are speeding, valve spool 44*c* in retracting cylinder 11*c* will act to orifice flow at 30*c*, as heretofore described, creating a back pressure in chamber 15*c*, thereby decelerating piston 12*c*. The back pressure will also be sensed through the circuit in chamber 14*d* of cylinder 11*d* to decelerate piston 12*d*.

Valve spool assembly 75*c* in the retracting cylinder functions to cushion the piston stop at the completion of retraction, as well as assists to decelerate piston velocity during the stroke. When, for example, during operation the inertia of the load accelerates piston 12*c* to a velocity which exceeds the capacity of pump 16*c*, a pressure drop occurs in chamber 14*c* and, accordingly, passage 33*c*. The right end of pin 67 is subject to the high pressure in chamber 15*c* and the pressure differential, acting through pin 67, causes leftward actuation of spool 70 until land 73 moves into passage 33*c* and land 74 is in passage 29*c*, placing passage 33*c* and 29*c* in hydraulic communication through groove 78. Discharge flow from chamber 15*c* through passage 29*c* and pump flow through 33*c* will follow the path of least flow resistance. If a reduced or partial vacuum exists in chamber 14*c*, discharge flow from chamber 15*c* will be diverted through groove 78 into passage 33*c* and conduit 32*c* into chamber 14*c*, as well as to chamber 15*d* through line 61. When pressure builds in chamber 14*c* and passage 33*c*, fluid from passage 33*c* will pass through groove 78 to line 64 to tank, thereby relieving pressure in chamber 14*c* to reduce the driving force and the velocity of piston 12*c*. This also reduces pressure in chamber 15*d* to reduce the speed of piston 12*d* as well as piston 12*c*.

Once piston speed beings slowing, the pressure force exerted against pin 67 diminishes enough to be overcome by the force of spring 55, causing spool 70 to move to the right isolating passages 33*c* and 29*c*. The increase in pressure in passage 33*c* moves valve 44*c* to the right to allow efficient, unobstructed discharge flow. In this manner, piston velocity is limited to a predetermined maximum. Thus, during the velocity control phase of the control operation, valve spool assembly 75*c* assists by preventing cavitation and relieving pressure from the expanding interconnected chambers 14*c* and 15*d*. Valve 75*c* is not conditional on supply pressure and responds only to a predetermined increase in pressure in the cylinder discharge chamber to be moved to a shunting position.

As the piston 12*c* approaches the end of its stroke, metering pin 38*c* will enter the upper end of 29*c* and begin to orifice flow of discharge fluid from chamber 15*c*. A pressure rise occurs in chambers 15*c* and 34*c* which causes pin 67 to actuate spool 70 to the left placing passages 33*c* and 29*c* in communication through groove 78. Pressure fluid flow from pump 16*c*

to chamber 14c is diverted through groove 78 to passageway 29c and through return line 64 to reservoir 20. Drop of pressure in chamber 14c allows the back pressure in chamber 15c to better decelerate and cushion piston movement and permits valve 44c to move into a flow-restricting position to further assist in cushioning.

Piston 12c comes to a stop fully retracted and the system becomes static with the pressure in passages 27c and 34c, and chamber 15 going to zero. This allows spool 70 to return to the right in a nondiverting position.

By moving control valve 19c to the left (crisscross position), passage 27c becomes the inlet to chamber 15c and pistons 12c and 12d reverse their direction of travel. Since piston 12c is fully retracted against head 25c, inlet flow will be introduced through passage 29c, across bypassage 35c and the check valve and into chamber 15c. Inlet flow to cylinder 11d is from passage 34c, line 62, passage 33d, duct 32d to chamber 14d.

Discharge from chamber 14c will flow through 32c, 33c, 61 and 34d into chamber 15d of cylinder 11d. Discharge flow from 15d will be through passage 27d of cylinder 11d to line 63 and to reservoir 20c. Valve 75d in retracing cylinder 11d, will now function in the manner described with reference to 75c to divert flow in the cushioning and acceleration control phases. Valve 44d will operate in the circuit as described with reference to cylinder 11c, performing the piston velocity control by selectively orificing discharge flow from cylinder chamber 15c.

FIGURE 4

FIG. 4 depicts an alternate form of the progressive metering device which acts to restrict flow out of the cylinder at the end of the stroke to provide a cushioned cylinder stop. With the present embodiment, the necessity for the metering pin 38, bypass 35 and one-way check 36 is eliminated.

In this embodiment, instead of the tapered metering pin, the piston and piston rod are provided with an axial bore 101 in alignment with upper duct 28 of inlet-discharge passage 27 in cylinder end wall 25. Slidably located within bore 101 is cylindrical plunger 85 having central axial flow passage 96 with orificed inlet 88 near the inner end. A compression spring 89 extending between the end of plunger 85 and the bottom of the axial bore will extend the plunger as the piston extends. Spring guide 93 extends from the bottom of bore 101 within the spring. Guide 93 is hollow and open-ended and provided with openings in the sidewall so that fluid may flow through passage 96 with a minimum of obstruction. Skirt 86 at extreme inner end of plunger 85 serves to guide the relative travel of the plunger within the bore and also contacts a mating stop 98 on the bore sidewall which limits the extension of the plunger. An orifice 92 in the recessed face 91 of the plunger affords communication between passage 96 and the inlet-outlet passage 28 of the cylinder.

The piston 95 contacts stop 94 to limit the inward travel of the piston so that plunger 85 is not fully retracted when the piston is fully retracted. A tapered notch 99 extends axially along the plunger wall and cooperates with bore 101 at 100 to provide a progressive orifice for discharge flow.

OPERATION - FIGURE 4

The operation of the plunger-type flow-restricting member is as follows: With the piston 95 fully retracted, fluid is admitted to inlet passage 28. The fluid pressure acting against the face 91 of the plunger 85 forces the plunger inward compressing spring 89. This admits fluid to the head end of the cylinder chamber actuating the piston. As the piston extends, spring 89 causes plunger 85 to extend until skirt 86 engages mating stop 98. When the piston travel is reversed and the piston approaches end 25 of the cylinder, the end of plunger 85 contacts the cylinder head end 25 and shuts off direct flow from the cylinder chamber to discharge 28. The discharge flow must now pass along taper groove 99, through orifices 88, into passage 96 and through orifice 92 to discharge passage 28. As the piston moves inward, plunger 85 further retracts to progressively orifice discharge flow as the volume of fluid flowing into passage 28 is limited by the opening defined by groove 99 and cooperating wall 100. From this description it will be seen that the extension of the piston from a fully retracted position can be accomplished with the use of the present plunger-type progressive orifice, eliminating the need for a separate bypassage. This mechanism could be substituted for the tapered metering pin or similar snubbing device for cushioning the piston stop.

Although I have described the novel velocity and cushioning control devices herein with reference primarily to linear actuators, it will be obvious that the invention herein can be modified and is equally adaptable for use with rotary actuators. To the extent that such modifications and changes do not depart from the spirit of the invention, they are intended to be included in the scope thereof, which is not limited to the embodiments specifically illustrated in the drawings, but rather only by a just and fair interpretation of the claims.

Having clearly described the invention and thereby enabling those skilled in the art to construct and use its principles, I claim:

1. In a hydraulic actuator including a cylinder defining a working chamber and having an end wall, a piston and rod reciprocable in said chamber and dividing it into a first variable volume chamber in the rod end of the cylinder and a second variable volume chamber adjacent the end wall, a first passage in said end wall connected to said first chamber which serves as an inlet for hydraulic pressure fluid when said first chamber is expanding in volume, a second passage in said end wall connected to said second chamber and which serves as a discharge passage when said second chamber is contracting in volume, a piston velocity control which comprises:
   - a transverse passage in the cylinder end wall communicating with said first and second passages and intersecting said second passage to define a flow restriction in the path of flow through said second passage;
   - a duct communicating said second chamber with said transverse passage;
   - valve means slidable in said transverse passage having one end exposed to the reduced pressure in the discharge passage at the discharge side of the flow restriction and the other end subject to chamber pressure existing in said duct; said valve means having an open position which allows unobstructed flow through said second passage and a closed position which restricts flow through said second passage;
   - valve-controlling means including reciprocable plunger means in said transverse passage which bias said valve means in said normally open position, said valve-controlling means being held in a biasing position by a predetermined inlet pressure in said first passage whereby a loss in inlet pressure to a predetermined level attendant excessive piston velocity allows the valve means to override said bias to move to a closed position and restrict flow through said discharge chamber while permitting unrestricted flow through said discharge chamber while permitting unrestricted flow through said first passage; and
   - said valve-controlling means including a spring exerting a force to assist in holding said valve spool in a normally open unobstructing flow position.

2. The apparatus of claim 1 provided with means movable by and with the piston which cooperate with said second passage to additionally and progressively further restrict the flow of fluid from the cylinder outward through said second passage to decelerate and cushion the stop of said piston as the piston approaches said end wall.

3. The apparatus described in claim 1 further provided with a bypassage in the cylinder end wall communicating at one end with said second passage and at the other with said second chamber and having a check valve associated therewith to permit flow only from said second passage into the second chamber.

4. The apparatus of claim 1 wherein said cylinder is further provided with a second transverse passage communicating with said first and second passages,

a cannelured valve spool slidably disposed in said second transverse passage, said spool in a first position blocking flow between said first and second passages and in a second position establishing communication between said first and second passages, said spool being normally biased to said first position and being pressure responsive to a predetermined pressure in said second chamber to overcome said bias to move to said second flow position and permitting flow through said first passage in both positions.

5. The apparatus of claim 1 wherein said valve means is provided with metering orifices which permit a predetermined discharge flow through said second passage when said valve means is in a closed position.

6. A hydraulic system comprising:

a first and second hydraulic cylinder each having an end wall;

a piston and rod reciprocable in each cylinder dividing each into a first variable volume chamber in the cylinder rod end and a second variable volume chamber adjacent said end wall;

each cylinder including a first inlet-discharge passage in direct communication with said first cylinder chamber, a second inlet-discharge passage in communication with said second chamber, and a third recirculation passage in said end wall in communication with said second chamber;

a source of fluid pressure having a low-pressure supply reservoir connected to a control valve;

conduit means interconnecting said first and third passages of said first cylinder with the third and first passages of said second cylinder respectively;

said second inlet-discharge passages of said first and second cylinders being selectively connectable to said pressure source and reservoir through said control valve for tandem operation whereby one piston rod is extending while the other is retracting;

a transverse passage in each cylinder end wall communicating with said first, second, and third passages;

valve means slidably positioned in said transverse passages, said valve means having a first nonobstructing position and second position effective to restrict discharge flow through said second inlet-discharge passage of the retracting cylinder, said valve means being normally biased to said first position by pressure fluid in said first passage in said retracting cylinder, and being pressure responsive whereby a predetermined reduction in pressure in the retracting cylinder chamber allows said valve means to overcome the bias to move into said second position to restrict discharge flow thereby controlling the velocity of the pistons.

7. A hydraulic system comprising:

a first and second hydraulic cylinder each having an end wall;

a piston and rod reciprocable in each cylinder dividing each into a first variable volume chamber in the rod end and a second variable volume chamber adjacent said end wall;

each cylinder including a first inlet discharge passage in direct communication with said first cylinder chamber, a second inlet-discharge passage in communication with said second chamber, and a third recirculation passage in said end wall in communication with said second chamber;

means interconnecting said first, second, and third passages of said first and second cylinder for tandem operation whereby one piston rod is extending while the other is retracting, said means interconnecting the expanding cylinder chambers to a common supply of pressure fluid and the contracting chamber to a common discharge;

a transverse passage in the cylinder end walls communicating with said first, second, and third passages;

pressure responsive valve means slidable in said transverse passages having a first nonobstructing position and a second position effective to restrict discharge flow through said second passage, said valve means operative in the retracting cylinder and being normally biased to said first position by pressure fluid in said first passage in the retracting cylinder and being pressure responsive whereby a predetermined reduction in pressure in said retracting cylinder first passage allows said valve means to overcome the bias to move into said second position.

8. The apparatus of claim 7 provided with means movable by and with the piston of said first and second cylinder which cooperate with said second passages in said cylinders as said pistons retract to additionally and progressively further restrict the flow of fluid from the cylinders outward through said second passages to decelerate and cushion the stop of said pistons.

9. The apparatus of claim 7 wherein each of said cylinders is further provided with a bypassage in the cylinder end wall communicating at one end with said second passage and having a check valve associated therewith to permit flow only from said second passage into the cylinder.

10. The apparatus of claim 7 wherein each of said cylinders is further provided with a second transverse passage communicating with said first and second passages,

a cannelured valve spool slidable disposed in said second transverse passage, said spool in a first position blocking flow between said first and second passages and in a second position establishing communication between said first and second passages, said spool being normally biased to said first position and being pressure responsive to a predetermined second chamber pressure to overcome the bias to move to said second flow position and permitting flow through said first passage in both positions.