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[54] **AUTOMATIC PRESSURE RELIEF AND SNUBBING IN HYDRAULIC ACTUATORS**

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[51] **Int. Cl. F15b 15/22,** F15b 13/042, F15b 11/08
[58] **Field of Search** 91/452, 451, 405, 91/407, 408, 420, 441, 421

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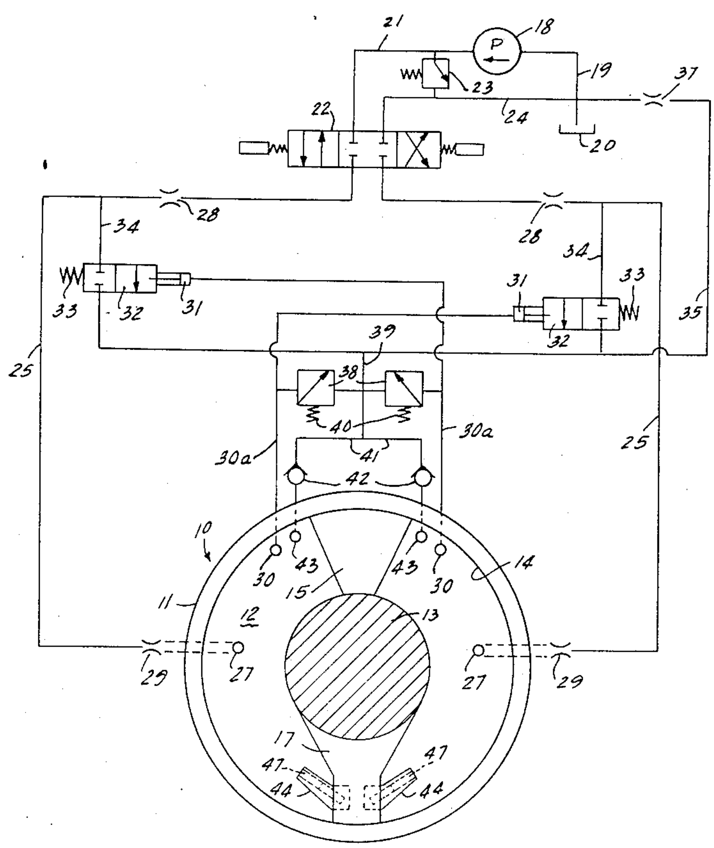
[57] **ABSTRACT**

Rapid start-up acceleration is automatically attained by pressure relief provided for the discharge chamber through a dump valve referenced to the drive chamber of the actuator. Deceleration is accomplished by snubbing hydraulic fluid discharge and by diverting pressure directly from the hydralic feed line. Negative pressure relief or anti-cavitation is provided for in the hydraulic system of the actuator. A novel, efficient snubber valve is carried by the actuator piston to control hydraulic fluid passing through the discharge port of the actuator.

18 Claims, 4 Drawing Figures



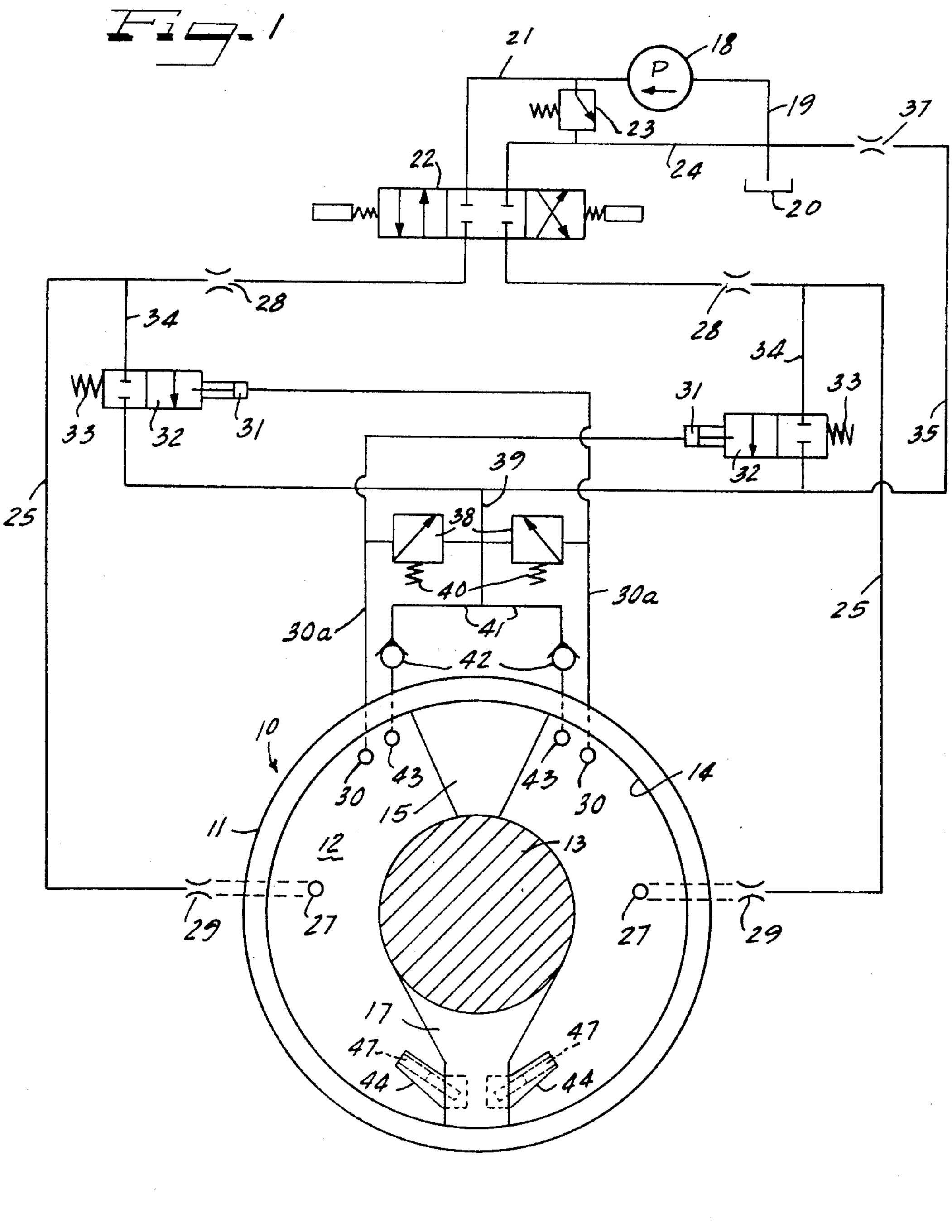


Fig. 1
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Fig. 2

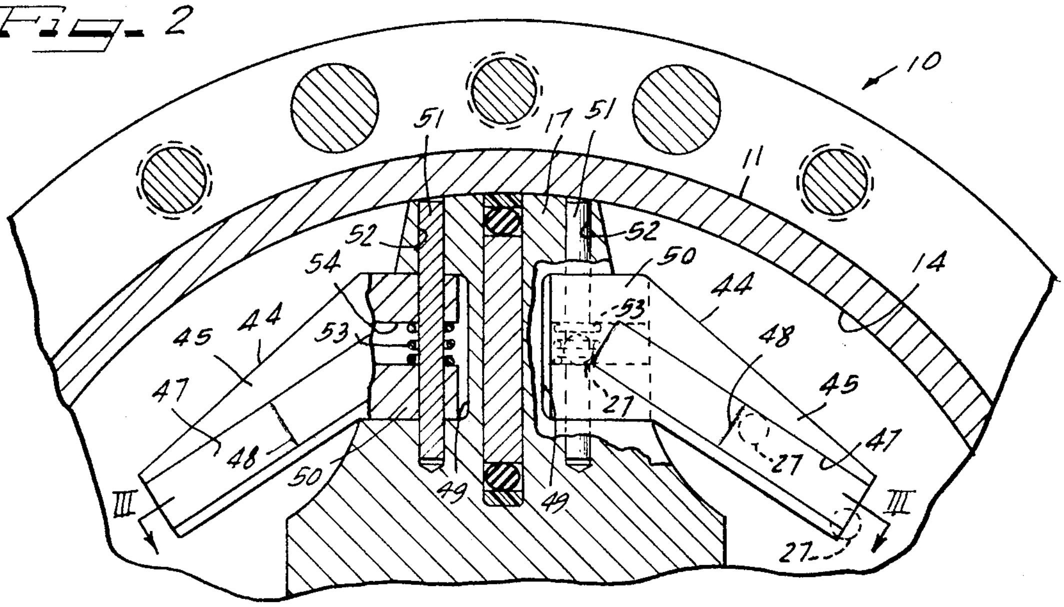
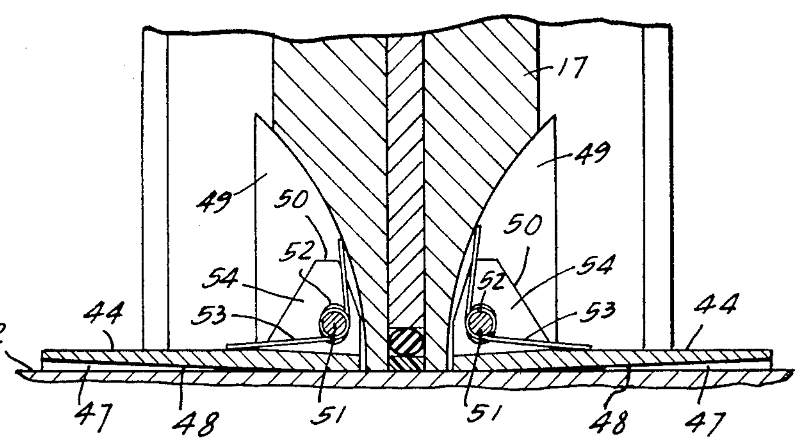


Fig. 3



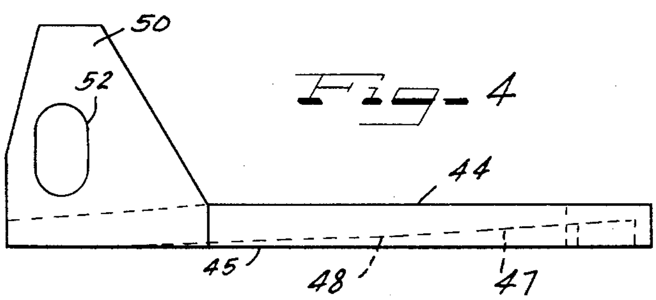


Fig. 4

AUTOMATIC PRESSURE RELIEF AND SNUBBING IN HYDRAULIC ACTUATORS

This invention relates to hydraulic actuators in general and is more particularly concerned with automatic pressure relief in a hydraulic system, and driving stroke terminal deceleration snubbing.

Several areas in hydraulic actuator operation, and more particularly in the operation of heavy-duty, large-load, high inertia actuators, have been in need of improvement. Such actuators are especially useful in numerous and varied types of industrial, load-lifting, earth-moving and similar types of equipment. High hydraulic pump pressures are needed to perform the desired work. Because of the necessity for start-up from zero velocity, abrupt stops, definite operating and return stroke limits, rapid changes of direction, and the like, fairly sophisticated hydraulic controls have been developed in the actuator operating systems. Nevertheless, some areas in the operating systems have heretofore escaped proper attention or solution, or attempts at solutions have raised additional problems.

For example, during start-up, pressure leveling, surge-preventing, anti-cavitation devices in the hydraulic systems have retarded acceleration to the point of undesirability and have reduced the efficiency of operation of the actuator and its associated equipment.

During deceleration snubbing, progressive discharge flow restriction and thus stopping back pressure on the piston causes a corresponding pressure build-up on the driving side of the piston and in the driving chamber of the cylinder, with potentially damaging consequences in the apparatus. Some attempts, lacking in efficiency have heretofore been made to relieve this condition.

Due to high inertia load and necessity for high operating pressures, but necessarily rapid and sometimes abrupt decelerations, cavitation in the operating chambers of the actuator is an ever-present hazard. Prior attempts to solve this problem have lacked efficiency.

There has been need for an effective, rugged, trouble-free and durable snubber valve construction for hydraulic actuator pistons, even though numerous prior attempts have been made in this direction.

Accordingly, it is an important object of the present invention to overcome the foregoing and other disadvantages, defects, deficiencies, shortcomings and problems in prior methods and structures and to attain important advantages, improvements and new results in and in connection with hydraulic actuators.

Another object of the invention is to attain new and improved start-up acceleration in the operation of hydraulic actuators and more particularly heavy-duty actuators.

A further object of the invention is to provide new and improved deceleration method and means in the operation of hydraulic actuators.

Still another object of the invention is to provide new and improved anti-cavitation control for hydraulic actuators.

Yet another object of the invention is to provide new and improved snubber valve control means in hydraulic actuators.

A still further object of the invention is to improve the operating efficiency and equipment life in and in association with hydraulic actuators.

Other objects, features and advantages of the invention will be readily apparent from the following description of a preferred embodiment thereof, taken in conjunction with the accompanying drawings, although variations and modifications may be effected without departing from the spirit and scope of the novel concepts embodied in the disclosure, and in which:

FIG. 1 is a schematic illustration of a representative hydraulic actuator and hydraulic operating and control system;

FIG. 2 is a fragmentary sectional plan view of the actuator and snubber control valve structure;

FIG. 3 is a fragmentary sectional detail view taken substantially along the line III—III of FIG. 2; and

FIG. 4 is a side elevational view of one of the snubber valve members.

Although in some respects it will be readily apparent that the present invention is applicable to a linear actuator, that is an actuator having a cylindrical working chamber and a rectilinearly stroking piston, especially advantageous applicability is evident in connection with rotary actuators, that is of the type having an oscillatably stroking piston in an annular working chamber structure. As represented in FIG. **1**, a rotary hydraulic actuator **10** comprises an annular housing **11** having opposite end walls **12** (only one being shown) coaxially journalling a wingshaft **13** within an annular working chamber **14** defined within the housing **11**. Suitable means (not shown) are provided, as is customary, for securing the housing **11** and the shaft **13** to respective parts of equipment which must be relatively moved by operation of the actuator. Such operation is effected by means of hydraulic fluid such as suitable oil under pressure introduced into the working chamber **14** to cause hydraulic pressure reaction between an abutment **15** carried by the chamber and a piston vane **17** carried by the shaft **13**. As is usual, the abutment **15** is fixedly mounted in the housing and in sliding engagement with the perimeter of the shaft **13** and provides a liquid barrier across the chamber **14**. By its projection from the shaft **13** into sliding engagement with the cylindrical housing wall defining the chamber **14**, the piston vane **17** divides the chamber in cooperation with the abutment **15** into respective subchambers, with the result that when hydraulic fluid is introduced under pressure into one subchamber while the opposite subchamber is permitted to drain or discharge, expansion of the pressurized chamber causes relative rotation of the shaft **13** and the housing **11** by driving force of the pressure against the piston vane **17**.

Hydraulic pressure fluid for driving the actuator **10** is supplied through a hydraulic circuit by a pump **18** controlled and powered in a suitable manner and operative to draw hydraulic fluid through a duct **19** from a tank or sump **20** and force the fluid under pressure into a supply duct **21** under a constant head of pressure of, for example, about 2,000 psi., depending upon service requirements. Normally, the supply duct **21** may be blocked by a control valve **22** which may be of the reciprocable manually operated or controlled type having a central non-demand or neutral zone. As thus blocked the hydraulic fluid may by-pass from the duct **21** through a normally closed pressure release valve **23** to return by way of a duct **24** to the intake duct **19** or the sump **20**.

When it is desired to operate the actuator **10** by driving the vane shaft **13** in either the clockwise or counterclockwise direction relative to the housing **11** as shown in FIG. **1**, the control valve **22** is shifted to a effect communication between the pressure side of the pump **18** and the actuator subchamber which is to be pressurized and between the low pressure or suction side of the pump and the discharge or low pressure subchamber of the actuator. In the present instance a symmetrical reversible hydraulic control system is provided wherein each of the working subchambers of the actuator is connectable alternatively with either side of the pump **18** through a similar respective duct **25** communicating with its subchamber through a port **27** in the selected end wall **12** of the actuator housing, such port being located in an instance near but suitably spaced from the abutment **15.** When the spool of the right, the lefthand working subchamber of the actuator as shown in FIG. **1** is pressurized by connecting its duct **25** with the pressure delivery duct **21** while the other communication duct **25** is placed in communication with the low pressure duct **24** communicating with the intake side of the pump **18.** When the valve **22** is moved towards the left as shown in FIG. **1,** the reverse communication relationship is effected, namely, the right hand working subchamber is pressurized and becomes the drive chamber and the lefthand subchamber becomes the discharge chamber. In order to limit and smoothly regulate the flow of pressure fluid from the pump through either of the ducts **25,** each such duct is provided with a flow restriction **28** near the valve **22.** In addition, pressure flows through each of the ducts **25** to its port **27** is controlled by a flow limiting restriction **29** located near the subchamber port.

Upon adjusting the valve **22** for pressurizing either selected working subchamber of the actuator **10,** drive chamber pressure will promptly reach substantially pump pressure, i.e., 2,000 psi., and the actuator wingshaft **13** will begin to turn by reason of the pressure exerted on the piston vane **17.** However, in view of the restrictions **29** and **28,** in that order, in the discharge line **25,** back pressure will tend to resist rapid acceleration. Therefore pressure is automatically relieved from the discharge line between the restrictions **28** and **29.** For this purpose, means are provided comprising a port **30** adjacent to the abutment **15** and by which pressure is referenced from the drive chamber through a duct **30*a*** to a drive piston **31** by which a dump valve plunger **32** is shifted in opposition to biasing means in the form of a spring **33** to connect a normally blocked bypass duct **34** connecting the duct **25** between the orifices **28** and **29** with a return duct **35** communicating with the suction line **19** and/or the sump **20.** Inasmuch as the flow restriction through the path provided by the duct **34** and the valve **32** is substantially less than through the restriction **28,** the pressure relief afforded results in faster acceleration of the wingshaft **13** and its attached inertia load. Although for maintaining a filled system, the return line duct **35** has a flow restriction **37** therein downstream from the point of communication of the by-pass duct **34,** the flow path through this restriction is substantially less than through the restriction **28,** thereby facilitating the rapid acceleration phase in operation.

As actuator operation attains the desired running or designed speed, the pressure line restrictions **28** and **29** reduce the drive chamber pressure to a lower value, for example to 1,000 psi., and below the bias value of the dump valve spring **33** which is thus closed to disconnect or block the by-pass duct **34** and restore the discharge line **25** to flow through the restriction **28** thereof to pump or sump. The exhausting hydraulic fluid is thereby maintained at a back pressure which nearly approximates the drive pressure to maintain smooth, constant speed in the driving stroke of the operating cycle. As a result of this back pressure, if there is any tendency to over-speed due to changing gravitational or other bias on the load being moved, higher pressure will be generated in the exhaust chamber, and according to the present invention compensating pressure relief is provided. This comprises opening the dump valve **32** connected with the pressure line **25** to by-pass and thus relieve sufficient pressure downstream from the orifice **28** to the return line duct **35** to reduce driving pressure in the drive chamber enough to maintain substantially constant running speed. Operation of the pressure line dump valve **32** is the same as already described for the dump valve **32** of the exhaust line, the excess pressure being referenced by way of the port **30** communicating with the exhaust chamber through the associated duct **30*a*** to the drive piston **31** of the dump valve **32** in opposition to its spring bias **33** to open the by-pass line **34.** It will thus be apparent, that when either of the subchambers of the working chamber **14** is in the pressurized condition the dump valve **32** referenced thereto serves to effect acceleration pressure relief for the exhaust line, while when the same subchamber is in the relationship of exhaust chamber, the dump valve **32** referenced thereto acts as an overload relief, speed leveling dump valve in control of the high pressure line.

Should the main control valve **22** be closed for any reason while the actuator is operating at running speed, as for example through the mid-portion of travel of the piston after attaining full acceleration, pressure in the exhaust chamber will rise due to operating momemtum. Since the exhaust line **25** is at this time closed, pressure relief means to the return duct **35** are provided. For this purpose, a respective crossover relief valve **38** for each of the working subchambers of the actuator communicates on the pressure responsive or upstream side with the respective referencing duct **30*a*** and on the downstream or discharge side with the return line **35,** desirably by way of a branch duct **39.** Each of the valves **38** desirably is normally biased as by means of a spring **40** to remain closed until a predetermined pressure is referenced thereto from the exhaust chamber, for example 1800 psi., which pressure is great enough to effect reasonably prompt deceleration of the actuator but low enough to avoid system over-load and damaging stress.

Further, during sudden stops by closing of the control valve **22** while the actuator is running at force of substantially full speed, cavitation tendency may develop in the pressure chamber. To alleviate that condition, make-up hydraulic fluid is supplied to the shut-off pressure chamber from the low pressure or the return line portion of the hydraulic system. For this purpose, a respective anti-cavitation make-up duct **41** communicates with the return line duct **35** through the duct **39** with a respective check valve **42** communicating with the associated working subchamber of the actuator through a respective port **43** adjacent to the abutment **15.** The check valves **42** prevent outflow through the ports **43** but permit anti-cavitation inflow as required. Anti-cavitation inflow or replenishment to the respective subchamber is implemented by the constant pressure maintained in the return duct line **35** by the restriction **37.**

New and improved means are provided for snubbing deceleration of the actuator piston at terminus of each driving stroke which, in the actuator **10** and its hydraulic control circuitry as illustrated, is in each opposite direction of travel of the piston vane **17,** since the same pump pressure is utilized in each opposite direction and the hydraulic control circuitry in the system is symmetrical as shown. To this end, the piston vane **17** has at each side thereof a respective laterally projecting snubber valve plate **44** operative to close off or throttle discharge through the port **27** of the associated working subchamber progressively in the terminal portion of each driving or working stroke, thereby attaining smooth stress-free deceleration and stopping of the actuator. As the port **27** is closed, pressure increases in the exhaust chamber between the location of the port **27** and the abutment **15.** Such pressure is referenced to and operates the dump valve **32** of the pressure line **25,** thereby diverting pump pressure to the return line **35** and relieving pressure in the drive chamber of the actuator. By dumping the hydraulic fluid directly from the pressure line **25** into the return line **35,** by-passing the control valve **22,** no check valves are required between the dump valves and the inlet line and the outlet line to prevent the fluid from flowing the wrong way through their respective dump valves and simplifies and increases the efficiency and safety of the system, especially when driving at full supply pressure during snubbing.

Construction and operation of the snubber valves **44** is such as to maintain pressure in the exhaust chamber at a value which will decelerate the load at a constant rate and prevent excessive surges or pressure peaks. In addition, the valves are constructed and arranged to enable ready, restriction-free reverse operation of the actuator by inlet flow through the associated port **27,** when the control valve **22** is reversed to reverse the driving stroke of the piston vane **17.** To this end, having reference to FIGS. **2–4,** each of the valve plates is constructed to provide a slide face **45** arranged to bear against the face of the actuator housing closure **12** through which the port **27** communicates. The face **45** is on a portion of the valve plate which is elongated to project a substantial distance laterally from the associated side of the piston vane **17** and in the direction of movement of the vane into the respective working subchamber. Thereby as the vane **17** moves toward the port **27** in the terminal portion of a working stroke the projecting valve member **44,** and more particularly the bearing face **45** progressively covers the port **27** which, of course, is properly located in the surface of the closure **12** for this purpose. By having the throttle valve member **44** constructed to provide an orifice restriction operative to effect deceleration at a constant rate and prevent surges or pressure peaks, the port **27** can be in the form of a single hole in the face of the closure **12.** For this purpose, the bearing face **45** has therein a longitudinally extending shallow orifice groove **47** which although is desirably straight to facilitate machining is of substantial width to remain at substantial registration with the port **27** as the throttle valve advances along an arc with the vane **17.** As shown schematically in FIG. **2,** as the tip of the valve member **44** comes into registration with the port **27,** the leading end of the orifice groove **47** registers with the port and registration of the groove with the port continues to the trailing end of the groove. Progressive throttling is effected by having the groove **47** progressively diminishing in depth to merge with the bearing face **45** at the trailing end of the orifice groove. Thereby a smooth, progressive throttling action is attained. In the approximately first half of its length, the orifice groove **47** diminishes in depth at a greater angle to a transition point or line **48** from which the throttling orifice groove diminishes at a shallower angle to the terminal end of the groove, both tapering angles being calculated to be complementary for smooth, progressive throttling coordinated with functioning of the pressure line dump valve **32** by reference to the pressure built up in the exhaust chamber as the throttling progresses. At the trailing end of the throttle orifice groove **47,** the sliding surface **45** fully covers the port **27** and thus brings the piston vane to a complete halt. Aiding the throttling action of the throttle valve member **45** is the pressure built up in the exhaust chamber as throttling progresses so that at the terminus of throttling the pressure firmly presses the throttle valve face **45** against the face of the closure **12.**

To make the throttle valve **44** responsive not only to chamber pressure for firm throttling bearing against the closure **12** but also to enable release of the throttle valve in response to hydraulic pressure fluid entry through the associated port **27** to effect reversal of the actuator piston, the valve member **44** is mounted in a manner to enable a limited range of movement toward and away from the face of the closure **12.** For this purpose, the side of the vane **17** from which the throttle valve member **44** projects is provided with a mounting recess **49** for the trailing or butt end of the valve member, such recess opening not only from the side of the vane but also from the edge of the vane which opposes the closure **12** in which the port **27** is located. In width, depth and length, the mounting recess **49** is complementary to and accommodates an integral mounting lug **50** on the butt end portion of the valve member, with the sides of the mounting lug and the sides defining the recess **49** being in close but slidable relation so that movement of the valve member is permitted longitudinally of the recess but the valve member will be held firmly against any significant movement laterally thereof as it is carried into throttling relation to the port **27.** For connecting the valve member in the mounting recess, means comprising a connecting pin **51** are provided which pin is secured at its opposite ends in the vane **17** and extends through a bearing opening **52** in the lug **50** and which bearing lug is sized to retain the valve member against displacement in its longitudinal direction relative to the vane **17.** To enable a limited range of movement of the valve member **44** to assure firm bearing of the face **45** against the surface of the closure **12** independently of the bearing relationship of the vane **17** to the closure member and to enable backing away of the valve member **44** under inlet pressure from the associated port **27,** the bearing hole **52** is elongated on an axis normal to the valve face **45,** as best seen in FIGS. **3** and **4.**

Desirably, the valve member **44** is normally biased toward and into bearing engagement with the closure member **12.** For this purpose, means comprising a biasing spring **53** are provided desirably in the form of a coiled torsion or sear spring engaged about the pin **51** within a clearance slot **54** provided therefor in the lug

50. One leg of the spring 53 thrusts against the valve member 44 and the other leg thrusts against the back wall defining the recess 49. Bias provided by the spring 53 need be only sufficient to positively hold the valve member against the confronting face of the closure 12. Such bias should be of only limited thrust value so as to enable quick release of the valve member 44 in response to hydraulic driving pressure through the port 27 closed by the valve, when it is desired to reverse operation of the actuator.

It will be understood that variations and modifications may be effected without departing from the scope of the novel concepts of the present invention.

I claim as my invention:

1. In combination with a hydraulic actuator including a housing defining a working chamber divided into subchambers on opposite sides of piston means relatively movable in said chamber;

a. means providing a hydraulic pressure fluid source;

b. a pressure line for conducting hydraulic pressure fluid from the source to one of said subchambers;

c. an exhaust line leading from the other of said subchambers and having a pair of spaced restrictions therein; and

d. means referenced to said one subchamber to operate in response to initial pressure build up in said one subchamber to relieve pressure in said exhaust line between said restrictions for accelerating running stroke of the piston.

2. In a combination according to claim 1, including a return line to said source, and said means for relieving pressure comprising a bypass from said exhaust line to said return line and a normally closed dump valve in said bypass referenced to and driven open by predetermined pressure in said one subchamber greater than normal running pressure.

3. In a combination according to claim 2, said return line having a restriction therein which is less than the restriction in said exhaust line located downstream from said bypass, whereby to maintain the return line in a filled hydraulic system and enabling bypass pressure relieving dumping from the exhaust line into the return line upon opening of said dump valve.

4. In a combination according to claim 1, means connected directly to said pressure line and referenced to said other subchamber for relieving pressure in the pressure line upon development of excessive pressure in said one chamber.

5. In a combination according to claim 4, said pressure line including spaced restrictions therein, and said pressure line pressure relief means communicating with said pressure line between said restrictions.

6. In a combination according to claim 5, a return line to said source, said pressure relief means for the pressure line being connected to said return line and including a dump valve normally closed but responsive to excessive pressure in said other chamber to open a passage from the pressure line to said return line.

7. In a combination according to claim 6, means providing an excess pressure crossover relief for each of said subchambers communicating with said return line, and anti-cavitation hydraulic fluid make-up means communicating with each of said subchambers and leading from said return line.

8. In combination with a hydraulic actuator including a housing defining a working chamber divided into subchambers on opposite sides of piston means relatively movable in said chamber;

means providing a hydraulic pressure fluid source;

a pressure line for conducting hydraulic pressure fluid from said source to one of said subchambers to drive said piston means;

an exhaust line leading from the other of said chambers and communicating with a low pressure means;

overload pressure relief means communicating directly with said pressure line upstream from said one subchamber and referenced for response to pressure buildup in said other subchamber to bypass pressure from the pressure line to said low pressure means;

a return line to said low pressure means in addition to said pressure and exhaust lines, and said overload pressure relief means communicating with said return line;

said overload pressure relief means comprising a duct leading from said pressure line to said return line and having a normally closed dump valve therein;

a duct leading from said other subchamber and connected in referencing relation to said dump valve to effect opening of the dump valve in response to overload pressure from said other subchamber;

and startup pressure relief means connected to said exhaust line and discharging into said return line and responsive to pressure in excess of running pressure developed in said one subchamber during startup;

said startup pressure relief means comprising a dump valve in a bypass duct from said exhaust line to said return line and normally closed, with a referencing passage leading from said one subchamber to said startup pressure dump valve.

9. In a combination according to claim 8, a control valve operative to connect or disconnect the pressure and exhaust lines simultaneously relative to said source, crossover pressure relief means connecting said subchambers with said return line, and anti-cavitation means connecting said return line with said subchambers.

10. In combination with a hydraulic actuator comprising a housing having a working chamber therein divided into subchambers by piston means movable in respectively opposite directions in response to pressure differential in the respective subchambers, and a symmetrical hydraulic control system for the actuator;

including means providing a hydraulic pressure fluid source; low pressure means;

respective pressure/exhaust lines communicating with said subchambers;

multi-position control valve means operative for selectively and alternatively effecting pressure or exhaust relation between said lines and said source and said low pressure means;

a separate return line communicating with said low pressure means;

respective pressure relief means communicating said return line with said pressure/exhaust lines upstream from the subchambers and respectively referenced to be operative in response to overload pressure sensed in the other of the subchambers in each instance;

each of said pressure/exhaust lines having spaced restrictions therein; and said respective pressure relief means connected to said pressure exhaust lines between said restrictions.

11. In a combination according to claim **10**, said return line having a restriction therein downstream from said pressure relief means and providing for less restriction than the respective restrictions in said pressure/exhaust lines which are located nearest to said control valve means.

12. In a combination according to claim **11**, said valve means being operative to disconnect said pressure/exhaust lines from said source, crossover pressure relief means connecting said subchambers with said return line, and anti-cavitation means leading from said return line to each of the respective subchambers.

13. In a combination with a hydraulic actuator comprising a housing having a working chamber therein divided into subchambers by piston means movable in respectively opposite directions in response to pressure differential in the respective subchambers, and a symmetrical hydraulic control system for the actuator;

including means providing a hydraulic pressure fluid source; low pressure means;

respective pressure/exhaust lines communicating with said subchambers;

multi-position control valve means operative for selectively and alternatively affecting pressure or exhaust relation between said lines and said source and said low pressure means;

a separate return line communicating with said low pressure means;

respective pressure relief means communicating said return line with said pressure/exhaust lines upstream from the subchambers and respectively referenced to be operative in response to overload pressure sensed in the other of the subchambers in each instance;

each of said pressure/exhaust lines communicating with its respective subchamber through a port by which pressure fluid is delivered to either respective subchamber to move the piston means into the other of the subchambers from which hydraulic fluid is discharged through the port communicating therewith;

snubber valve means carried by the piston means operative to terminate driving stroke of the piston means by closing the discharge port whereby overload pressure in the associated subchamber referenced to the overload relief means of the pressure fluid delivering line bypasses the pressure therefrom to the return line;

said ports being in respective surface defining the respective subchamber, and said snubber valve means comprising a valve member slidably engaging the respective surface and having progressively restricting orifice means operative to progressively throttle and close the associated port in the movement of the piston means toward the port; and

said throttle valve members being normally biased into snubbing relation to said respective surfaces and being movable away from said port in response to fluid pressure to enable reversal of the piston drive.

14. In combination with a hydraulic actuator comprising a housing having a working chamber therein divided into subchambers by piston means movable in respectively opposite directions in response to pressure differential in the respective subchambers, and a symmetrical hydraulic control system for the actuator;

including means providing a hydraulic pressure fluid source; low pressure means;

respective pressure/exhaust lines communicating with said subchambers;

multi-position control valve means operative for selectively and alternatively affecting pressure or exhaust relation between said lines and said source and said low pressure means;

a separate return line communicating with said low pressure means;

respective pressure relief means communicating said return line with said pressure/exhaust lines upstream from the subchambers and respectively referenced to be operative in response to overload pressure sensed in the other of the subchambers in each instance;

each of said pressure/exhaust lines communicating with its respective subchamber through a port by which pressure fluid is delivered to either respective subchamber to move the piston means into the other of the subchambers from which hydraulic fluid is discharged through the port communicating therewith;

snubber valve means carried by the piston means operative to terminate driving stroke of the piston means by closing the discharge port whereby overload pressure in the associated subchamber referenced to the overload relief means of the pressure fluid delivering line bypasses the pressure therefrom to the return line;

said actuator being of the rotary type wherein said piston means comprise a vane carried by an oscillating wing shaft with which an abutment in the working chamber cooperates as a barrier to divide the working chamber with said piston vane into said subchambers and said ports are located in an end closure of the housing providing such surfaces;

said snubber valve means comprising respective elongated plate members mounted on and projecting in respectively opposite directions from said piston vane;

means connecting said valve members operatively to the piston vane and enabling limited movement of the valve members toward and away from said closure surfaces;

means normally biasing the valve members toward said surfaces; and

each of said valve members having a face slidably engaging with said surfaces and provided with a respective orifice groove diminishing from a leading end to a trailing end for progressively throttling discharge through the associated port as the respective valve member is moved over the port by advance of the piston vane in a driving stroke.

15. A method of controlling operation of a hydraulic actuator having a housing providing a working chamber and piston means in said housing relatively movable in response to hydraulic pressure in a driving subchamber of the working chamber and discharge of hydraulic fluid from an exhaust subchamber of the working chamber, comprising:

delivering pressure fluid from a source to said driving subchamber;

returning the hydraulic fluid from said exhaust subchamber through a return line and spaced running speed stabilizing restrictions in said line to a sump; and

at the start of piston drive referencing pressure buildup in said driving subchamber to a bypass valve in a bypass passage connecting said return line between said restrictions with the sump, and thereby opening said bypass valve in response to said pressure buildup to relieve pressure in the exhaust line whereby to accelerate attainment of running stroke of said piston.

16. In combination with a hydraulic actuator including a housing providing a working chamber, piston means movably operative in said chamber in response to hydraulic pressure fluid differential, means for delivering hydraulic pressure fluid to one side of said piston, and means for exhausting hydraulic fluid from the opposite side of said piston including a surface having an exhaust port therein, the improvement comprising:

a snubber valve member carried by the piston means and comprising an elongated plate movable therewith on said surface toward said port and having a face slidably engaging said surface;

said valve member having progressive throttling orifice means comprising a groove in said face of substantially the same width throughout its length and diminishing in depth from a leading end to a trailing end operative as the valve member moves over the port and the groove advances from said leading end in alignment with the port to effect progressive throttling of exhaust flow through the port;

said groove having a first portion diminishing in depth from said leading end at a greater angle than a second portion of the groove which diminishes at a shallower angle to said trailing end, for smooth progressive throttling effect in the advance of the groove over the port.

17. A combination according to claim 16, wherein said piston includes a wing shaft having a radial vane rotatably movably operative in said chamber relative to an abutment in said chamber, said means for delivering hydraulic pressure fluid delivering the pressure fluid into the chamber between said abutment and one side of said vane and said exhaust port being located between the opposite side of said vane and said abutment, said vane having a mounting recess therein at the side of the vane which faces toward said port, said valve member comprising an elongated plate having lug mounting structure at one end received in said recess and with the remainder of the valve plate projecting from the vane toward said port, and means connecting said mounting lug to said vane within said recess enabling limited relative movement of the valve plate toward and away from said surface but retaining the valve plate against transfers relative movement.

18. A combination according to claim 17, said connecting means comprising a pin mounted in said vane across said recess, said mounting lug having a clearance hole therein elongated in the direction of permissible movement of the valve plate toward and away from said surface, said mounting lug having a slot therein across which the pin extends, and a torsion biasing spring engaged about the pin in said slot and having one end thrusting against the valve plate and an opposite end thrusting against the vane in said recess.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,771,422 Dated November 13, 1973

Inventor(s) Gordon W. Kamman

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 3, line 13, after "of", insert --the control valve 22 is moved to--.

Signed and sealed this 9th day of July 1974.

(SEAL)
Attest:

McCOY M. GIBSON, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents