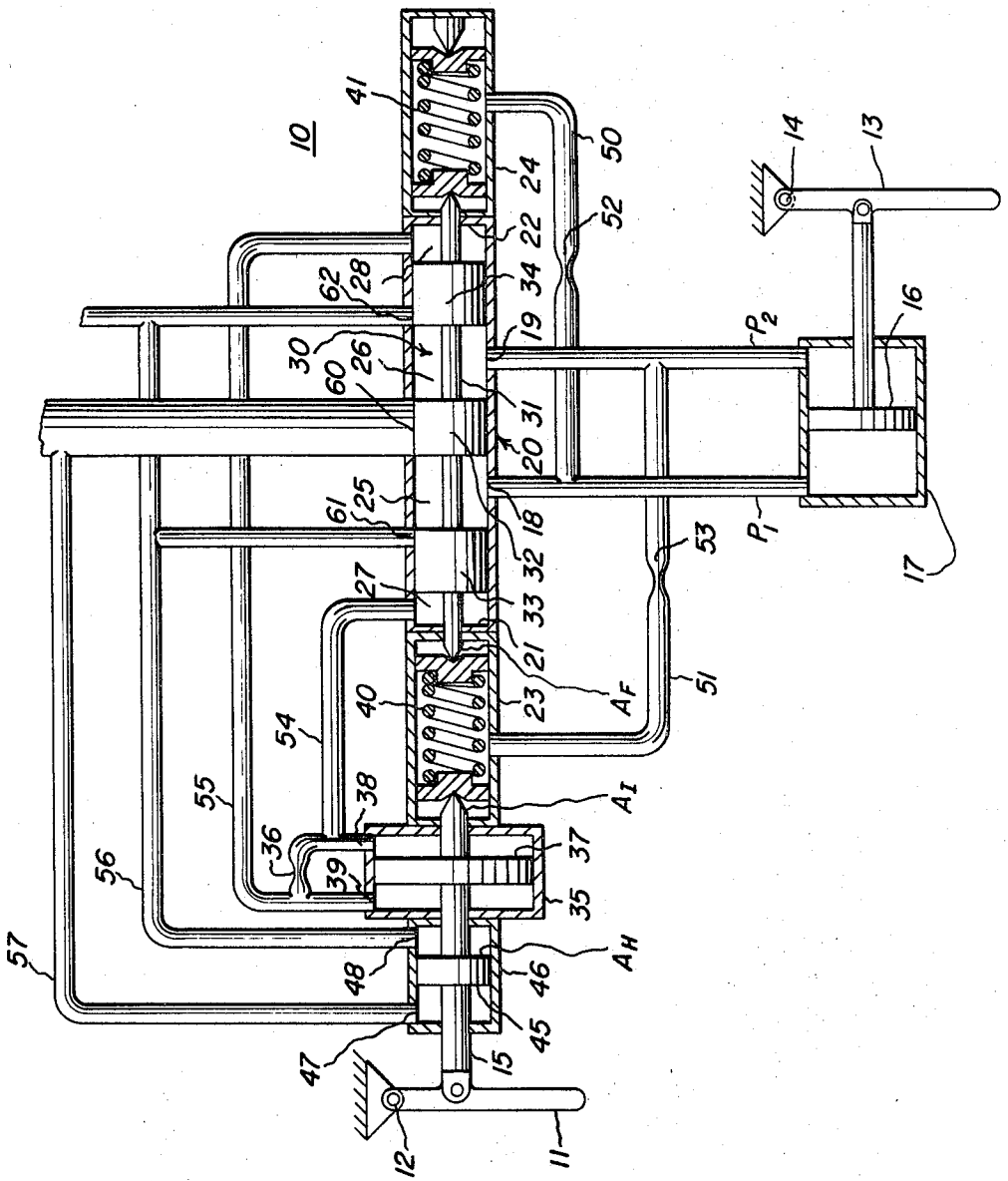
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STABILIZED HYDRO-MECHANICAL PRESSURE CONTROL VALVE

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STABILIZED HYDRO-MECHANICAL PRESSURE CONTROL VALVE

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ABSTRACT OF THE DISCLOSURE

A hydraulic network is provided in a hydro-mechanical pressure control valve including a sleeve and a spool with a plurality of lands thereon to produce close time correspondence between an input shaft and the difference in pressure at output ports thereof. The network includes a cylinder having a pair of outlets between which an orifice is connected and a piston. The piston is mechanically coupled to the input shaft of the valve and the two outlets are hydraulically coupled to chambers formed by the sleeve and lands. Also, the input shaft and the spool shaft are proportional to render the force on the input shaft constant and in addition means are provided for reducing the magnitude of such constant force.

The present invention relates, in general, to hydro-mechanical pressure valves and more particularly to hydromechanical pressure control valves which are stabilized by hydro-mechanical means.

In electrohydraulic systems, stabilization is provided therein by tachometer transducers and electrical stabilizing networks. Such stabilizing transducers and networks enable fast response and good performance to be obtained in such systems. Heretofore, hydro-mechanical servo systems either utilized electro-hydraulic stabilizing networks or did not employ stabilizing networks. When electro-hydraulic networks are utilized in such systems, a source of electrical power is required and additional complexity is introduced into the system by the added components. Accordingly, such networks are not commonly used in hydro-mechanical servo systems with a result that their performance is inferior to that of the more complex electro-hydraulic servo systems.

An object of the present invention is to provide an improved hydro-mechanical pressure control valve for use in hydro-mechanical servo systems.

Another object of the present invention is to provide a hydro-mechanical pressure control valve which is stabilized to provide close correspondence between input mechanical displacement and resultant output pressure differential corresponding to a particular displacement of the input.

Another object of the present invention is to provide a means in a hydro-mechanical pressure control valve for rendering the force required to produce a displacement substantially independent of the displacement of the input thereof.

Another object of the present invention is to provide hydro-mechanical means in a hydro-mechanical pressure control valve for reducing the force required to produce input displacement.

In carrying out the objects of my invention in one illustrative form thereof there is provided a main cylinder in which is contained a spool having an intermediate and a pair of end lands to form with the cylinder a pair of intermediate chambers and a pair of end chambers. A pair of end cylinders are secured to the main cylinder and contain spring biasing elements supporting the shaft of the spool in the axial directions. A pair of feedback passages are provided between each of the end cylinders

and a remotely positioned intermediate chamber. A source port and a pair of drain ports are located on the main cylinder. In the null position of the spool the source port registers with the intermediate land, and each of the drain ports registers with a respective end land. A pair of output ports is provided each connected to a respective intermediate chamber. An input shaft is provided within the remote wall of one of the end caps and contacts one of the bias springs. A piston is provided on the input shaft which in turn is housed in an auxiliary cylinder. One side of the cylinder is connected to an adjacent end chamber of the main cylinder and the other side of the cylinder is connected to the other end chamber in the main cylinder. An orifice is provided between the two ends of the auxiliary cylinder. Accordingly, an aiding force is provided on the spool which is a function of the rapidity of the displacement of the input shaft thereby maintaining a close time correspondence between axial displacement of the input shaft and difference in pressure in the output ports corresponding to such displacement. In addition, in accordance with another aspect of the present invention, by proportioning the cross-sectional area of the input shaft to be twice the cross-sectional area of the spool shaft, the force required to displace the input shaft is made independent of the displacement thereof.

The novel features which are believed to be characteristic of the present invention are set forth in the appended claims. The invention itself, however, together with further objects and advantages thereof may best be understood by reference to the following description taken in connection with the accompanying drawings in which:

The figure is a schematic view in cross section of an embodiment of my invention.

In the figure, there is illustrated a pressure control valve **10** in accordance with my invention incorporated in a control system in which a mechanical angular displacement of an input lever **11** about a pivot point **12** produces a certain rate of change of angular displacement of an output lever **13** about another point **14**. The lever **11** is connected to an input shaft **15**. A piston **16** is connected to the output lever **13** and is movable in a cylinder **17** in response to a difference in pressure at the output ports **18** and **19** of the valve **10**. The valve **10** includes a main cylinder **20** having a pair of end walls **21** and **22** and a pair of end cylinders **23** and **24** each including a pair of end walls, and axially aligned on each side of the main cylinder **20** in fixed relationship thereo. A spool **30** is provided in the main cylinder **20** consisting of a shaft **31**, and intermediate land **32**, and a pair of end lands **33** and **34** mounted thereon. The lands and the main cylinder form a pair of intermediate chambers **25** and **26** and a pair of end chambers **27** and **28**. A source port **60** and a pair of drain ports **61** and **62** are located on the main cylinder **20**. In the null position of the spool, the source port **60** registers with the intermediate land **32**, drain port **61** registers with end land **33**, and drain port **62** registers with end land **34**. The shaft **31** of the spool **30** is supported in the end walls of the main cylinder **20** and has one end extending into end cylinder **23** and has the other end extending into the other end cylinder **24**. An auxiliary cylinder **35** consisting of a central section and a pair of end walls is also provided and is axially aligned and secured in fixed relationship to the end cylinder **23**. A shaft **15** with a piston **37** thereon is movably mounted in the cylinder **35**. A pair of ports **38** and **39**, each connected to a respective one of the chambers formed between the piston **37** and the auxiliary cylinder **35** is provided. A pair of centering springs **40** and **41** is provided. Spring **40** is mounted in end cylinder **23** between ends of the shafts **31** and **15** extending therein. The other spring **41** is mounted between the other end of the shaft **31** and

the end wall of end cylinder **24**. The other end of the second shaft **15** is connected to input lever **11** as mentioned above. Another piston **45** is provided on the shaft between the input lever **11** and the auxiliary cylinder **35** and is contained within a second auxiliary cylinder **46** in fixed relationship to the first auxiliary cylinder **35**, having a central section and a pair of end walls through which the shaft **15** extends. A pair of outlets **47** and **48** are provided to end chambers formed by the piston **45** and the cylinder **46**.

Each of output ports **18** and **19** is connected by passageways **50** and **51** to a respective remotely situated end cylinder to provide feedback thereto. Orifices **52** and **53** may also provided to dampen oscillations in the valve, if needed. A duct **54** is provided connecting port **38** of auxiliary cylinder **35** to adjacently located end chamber **27** in the main cylinder **20** and another duct **55** is provided connecting port **39** of the auxiliary cylinder to the other end chamber **28**. An orifice **36** is provided across ports **38** and **39**. A passageway **56** is provided from port **48** of the second auxiliary cylinder **35** to the drain port **61**. Another passageway **57** is provided from port **47** of the second auxiliary cylinder to the source port **60**.

With a spool in the null position, the pressures P_1 and P_2 in outlets **18** and **19** are equal. The force tending to move the spool **30** to the left is the compression of the spring **41**. In the steady state condition, this force is counteracted by the pressure differential across the force compensation piston **45** and the input force on the shaft **15** which is transmitted through the spring **40**. Moving the input lever **11** to the right compresses spring **40** and forces the spool **30** to move to the right. The piston **37** is also moved to the right causing a compression of the fluid in the cylinder **35** and a pressure drop across the orifice **36**. The pressure differential in cylinder **35** acts on the spool **30** to cause it to move to the right. This additional force which is a function of the velocity of the input shaft **15** decreases the time for the spool to move to ultimate position thereby decreasing the phase lag between the input motion of shaft **15** and the output pressure differential P_1-P_2, i.e., it provides a time correspondence between input shaft **15** displacement and output pressure differential even for rapid displacements. Of course, when the lever **11** is released, the pretensioning springs **40** and **41** return the input shaft **15** to its starting position and the spool **30** is returned to its null position.

The proportioning of the active area of the piston **37** and the cross-sectional area of the orifice **36** in relation to the cross-sectional area of the end face of the land **33** depends upon the response desired in the servo system in which the valve is connected. The valves of the areas of the auxiliary piston **37**, the end land **33** and the cross-sectional area of the orifice **36** are selected to give the desired stable response, i.e., a design with the desired phase margin. When the area of the piston **37** is large in relation to the cross-sectional area of the orifice **36**, a larger pressure drop is produced between the outlet ports **38** and **39** of the auxiliary cylinder for a given displacement of the shaft **15**, thereby producing a close correspondence in the motion of the shaft **15** and the shaft number **31** with the result that the output pressure differential is in close time correspondence with input shaft motion. Similarly, when the area of the piston **37** is large in relation to the area of the end face of the land **33**, a stiffer or more faithful time correspondence between input motion and output pressure differential is obtained. The impedance provided by the orifices should be small in relation to the impedance to fluid flow provided by the end seals in the end cylinder **23** which seal the shafts **15** and **31**. Leakage through such end seals affects the gain of the valve. Should the valve have oscillations in certain applications, orifices **52** and **53** may be introduced into the feedback passageways **50** and **51** to dampen any such oscillations.

The force required to move the input shaft **15** depends on the pressure exerted by spring **41** and pressure feedback on spool **30** as will be apparent from the analysis below. In accordance with another aspect of the present invention, by making the cross-sectional area of shaft **15** equal to twice the cross-sectional area of the shaft **31**, the force required to move the shaft **15** is made independent of the position of the shaft **15**, that is, it is constant with input shaft displacement.

In accordance with a further aspect of the present invention, a means is provided to counterbalance the aforementioned constant force. To this end a piston **45** and cylinder **46** are provided. The piston **45** and the second auxiliary cylinder **46** when energized with fluid under pressure from the pressure supply, provides a force on shaft **15** to reduce the force required to move the spool **30**. However, as seals must be provided and frictional forces are inherently involved in such seals, the force cannot be practically reduced to zero.

The operation of the invention will be more fully appreciated by considering the equations governing the operation of the servo valve. Steady state force on shaft **15** is given by the following equation:

$$F=C_1+k_cY+A_I\left(P_0-\frac{k_cY}{2A_F}\right)-A_HP_s \qquad (1)$$

where

F=the force on the shaft **15**,
C_1 is the constant representing the pretension force on the springs **40** and **41** when the spool is in the null or neutral position,
k_c=spring gradient,
Y=displacement of shaft **15**,
A_I=cross-sectional area of the shaft **15**,
P_0 designates the cross-over pressure,
P_1 designates the pressure at outlet **18**,
P_2 designates the pressure at outlet **19**,
P_s designates supply pressure,
A_F is the cross-sectional area of the shaft **31** and is referred to as the feedback area,
A_H is equal to the effective area of the piston **45**

To simplify the derivation it will be assumed that the pretension force C_1 is zero and the force A_HP_s on the second cylinder is zero. Terms representing these factors can readily be inserted in the final equation.

In deriving the aforementioned Equation 1 the following assumptions are made:

(1) The motion of the spool **30** is neglected, that is, Y is much greated than χ, where χ is the displacement of the spool from its neutral or null position.

(2) Cross-over pressure

$$P_0=\frac{P_1+P_2}{2}$$

At null pressure differential, that is, when the spool **30** is in the neutral or null position, $P_1=P_2=P_0$, and

$$F=A_IP_0, \qquad (2)$$

neglecting pretension in the springs **40** and **41**.

The force on shaft **15** is given by the following equation:

$$F=k_c(Y-x)+A_IP_2 \qquad (3)$$

that is, the force on the shaft **15** is equal to the spring force plus the pressure P_2 acting on the cross-sectional area of the shaft **15**. Taking a force balance on the spool **30**, that is, in the displaced position summing the forces acting on the spool, yields the following equality:

$$k_c(Y-x)+P_2A_F=k_cx+P_1A_F \qquad (4)$$

and hence

$$x=\frac{Y}{2}+\frac{P_2-P_1}{2k_c}A_F$$

The pressure again is given by the relationship

$$\frac{P^2-P_1}{Y}=-\frac{k}{A_F} \quad (5)$$

Substituting Equation 5 in Equation 3 and rearranging terms yields the following relationship:

$$F=k_c(Y-x)+\frac{A_I P_2}{2}-A_I\frac{k_c}{2A_F}Y+\frac{A_I P_1}{2}$$

or as it is assumed that x is much smaller than Y and noting that the cross-over pressure is equal to

$$\frac{P_1+P_2}{2}$$

the following equation is obtained:

$$F=k_c Y+A_I\left(P_0-\frac{k_c Y}{2A_F}\right) \quad (6)$$

To take into account pretension, a constant C_1 is added to the Equation 6 and to take into account the force applied by the piston **45** to shaft **15** the pressure supply, pressure P_s times the effective cross-sectional area of the piston A_H is subtracted from the right hand side of equation to give Equation 1.

In the steady state condition no pressure differential exists across the orifice **36**. Accordingly, the input force will be independent of displacement if A_I is equal to twice A_F as Equation 1 reduces to

$$F=C_1+A_I P_0-A_H P_s \quad (7)$$

While I have shown a particular embodiment of my invention it will, of course, be understood that I do not wish to be limited thereto since many modifications may be made in the structural arrangement shown and in the instrumentalities employed. I contemplate by the appended claims to cover any such modifications as fall within the true spirit and scope of my invention.

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

1. A fluid pressure control valve comprising

a main cylinder, a pair of end cylinders, an auxiliary cylinder, each having a hollow central section and a pair of end walls,

said cylinders being aligned on an axis in fixed relationship to one another, each end cylinder being adjacent to a respective end wall of said main cylinder and one of said end cylinders being adjacent to said auxiliary cylinder,

a first shaft axially aligned in said main cylinder and having each end extending through a seal in a respective end wall thereof into a respective end cylinder, said shaft having a pair of end lands and an intermediate land mounted on said shaft to form with said main cylinder a pair of intermediate chambers and a pair of end chambers,

a second shaft axially mounted in said auxiliary cylinder having one end thereof extending through a seal in an end wall thereof adjacent to said one end cylinder thereinto and having the other end thereof extending through a seal in the other end wall to a point external to said auxiliary cylinder, a piston mounted on said second shaft in said auxiliary cylinder,

a first spring mounted in said one end cylinder between one end of said first shaft and said one end of said second shaft,

a second spring mounted between the other end of said first shaft and said other end cylinder,

a pair of passageways each connecting a respective end cylinder to a respective intermediate chamber remotely located with respect thereto,

a fluid supply port connected to said main cylinder, said port being blocked by said intermediate land when in its null position, a pair of fluid drain ports connected to said main cylinder, said ports being blocked by said end lands when in their central position, a pair of outlet ports each connected to a respective intermediate chamber,

a pair of ducts, one duct connecting an end of said auxiliary cylinder adjacent to said main cylinder to an adjacent end chamber thereof, the other of said ducts connecting the other end of said auxiliary cylinder to the other of said end chambers, a passageway having an orifice therein connected between said ducts, whereby a rapid displacement of said second shaft develops a pressure differential between said outlet ports in close time correspondence therewith.

2. The combination of claim **1** in which the active area of an end wall of said piston in relation to the cross-sectional area of said orifice is sufficiently large to produce a buildup of sufficient pressure in an end chamber in response to rapid displacement of said second shaft to maintain a close time correspondence between axial displacement of said second shaft and difference in pressure in said output ports corresponding to such axial displacement.

3. The combination of claim **1** in which the active area of an end wall of said piston is substantially greater than the active area of an end wall of an end land.

4. The combination of claim **1** in which the impedance to fluid flow of said orifice is substantially less than the impedance to fluid flow provided by shaft seals in said one end cylinder.

5. The combination of claim **1** in which is included a pressure differential utilization device connected to said outlet ports.

6. The combination of claim **1** in which the cross-sectional area of said second shaft is twice the cross-sectional area of said first shaft, whereby the force required to displace said second shaft is independent of the displacement thereof.

7. The combination of claim **1** including a second auxiliary cylinder having a hollow central cylinder and a pair of end walls axially aligned with said first axial cylinder and in fixed relationship therteo, said second shaft extending through seals in the end walls thereof, another piston mounted on said second shaft in said second auxiliary cylinder, and

a second pair of ducts, one of said ducts connecting an end of said second auxiliary cylinder adjacent said main cylinder to said drain ports, the other of said ducts connecting the other end of said second auxiliary cylinder to said supply port, the cross-sectional area of said second shaft being twice the cross-sectional area of said first shaft.

8. The combination of claim **7** in which said first auxiliary cylinder is located between said one end cylinder and said second auxiliary cylinder.

9. The combination of claim **1** in which fluid damping means are provided in each of said passageways.

10. A fluid pressure control valve comprising

a main cylinder, a pair of end cylinders, an auxiliary cylinder, each having a hollow central section and a pair of end walls,

said cylinders being aligned on an axis in fixed relationship to one another, each end cylinder being adjacent to a respective end wall of said main cylinder and one of said end cylinders being adjacent to said auxiliary cylinder,

a first shaft axially aligned in said main cylinder and having each end extending through a seal in a respective end wall thereof into a respective end cylinder, said shaft having a pair of end lands and an intermediate land mounted on said shaft to form with said main cylinder a pair of intermediate chambers and a pair of end chambers,

a second shaft axially mounted in said auxiliary cylinder having one end thereof extending through a seal in an end wall thereof adjacent to said one end cylinder thereinto and having the other end thereof extending through a seal in the other end wall to a point external to said auxiliary cylinder, a piston

mounted on said second shaft in said auxiliary cylinder,

a first spring mounted in said one end cylinder between one end of said first shaft and said one end of said second shaft,

a second spring mounted between the other end of said first shaft and said other end cylinder,

a pair of passageways each connecting a respective end cylinder to a respective intermediate chamber remotely located with respect thereto,

a fluid supply port connected to said main cylinder, said port being blocked by said intermediate land when in its null position, a pair of fluid drain ports connected to said main cylinder, said ports being blocked by said end lands when in their central position, a pair of outlet ports each connected to a respective intermediate chamber, and

a pair of ducts, one duct connecting an end of said auxiliary cylinder adjacent to said main cylinder to said drain ports, the other of said ducts connecting the other end of said auxiliary cylinder to said supply port, the cross-sectional area of said second shaft being twice the cross-sectional area of said first shaft, whereby the force required to displace said second shaft is independent of the displacement thereof.

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