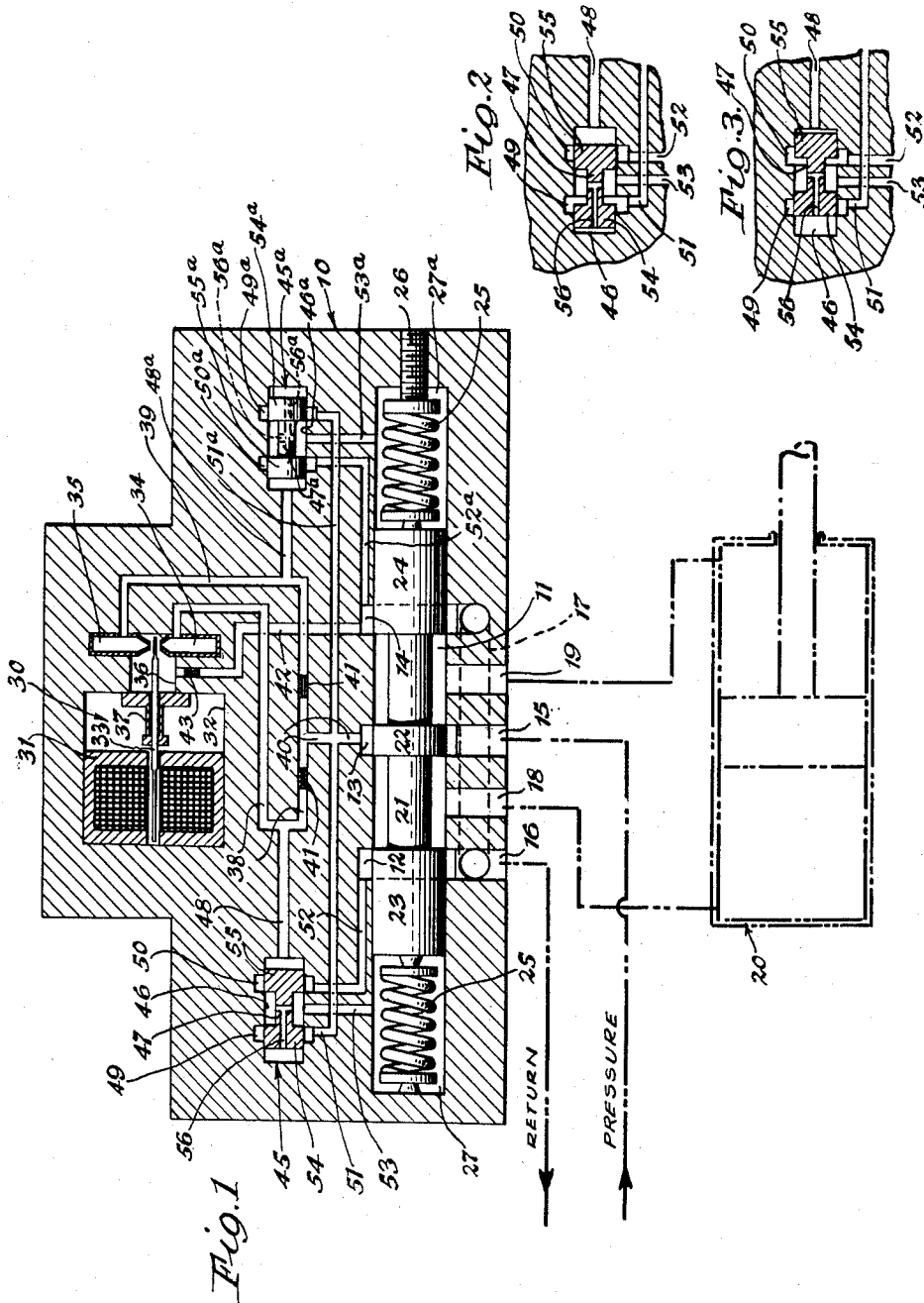


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ELECTRO-HYDRAULIC SERVO VALVE WITH PRESSURE
REPEATING POWER AMPLIFICATION
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ELECTRO-HYDRAULIC SERVO VALVE WITH PRESSURE REPEATING POWER AMPLIFICATION

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8 Claims. (Cl. 121—46.5)

This invention relates to improvements in electro-hydraulic servo valves particularly of the type shown in the patent applications of William C. Moog, Jr., namely, Serial No. 560,573, filed January 23, 1956, and entitled Fluid Control Valve, and Serial No. 356,631, filed May 22, 1953, and entitled Electro-Hydraulic Servo Mechanism, now Patent No. 2,767,689 dated October 23, 1956.

Such electro-hydraulic servo valves have a hydraulic amplifier first stage which provides a high force level output push-pull hydraulic drive with completely frictionless operation and excellent dynamic performance while requiring very little input signal. This hydraulic amplifier stage produces an output pressure differential proportionate to the electrical signal input. The output pressure differential of this first stage is applied to the ends of a valve spool for driving the same and this constitutes another stage of hydraulic amplification. For higher power requirements of the valve, i.e., higher rated output power, larger spool diameters may be utilized so as to provide metering ports of larger area, but as the spool diameter increases so does the volume of the fluid drive chambers at the ends of the valve spool and hence a greater amount of spool drive fluid must be handled. The speed of response of the valve spool is dependent upon the drive fluid flow rate between the hydraulic amplifier first stage and the spool end chambers. Where this drive fluid flow rate is limited and the spool diameter is increased, it will be seen that an increase in rated output power of the valve is obtained but at a sacrifice of dynamic response of the valve to signal input since the fluid handling ability of the hydraulic amplifier first stage is limited.

Accordingly, it is the primary object of the present invention to provide an electro-hydraulic servo valve which has high dynamic response to an electrical input signal and high power amplification. Such a high flow valve with high speed response is provided by operatively interposing pressure repeating hydraulic power amplifier means between the hydraulic amplifier first stage and the fluid drive chambers at the ends of the valve spool.

Another important object is to provide such a valve which has a small size and low weight.

Other objects and advantages of the present invention will be apparent from the following detailed description taken in conjunction with the accompanying drawings wherein:

Fig. 1 is a vertical central sectional view through an electro-hydraulic servo valve embodying the present invention and illustrating the internal construction of the valve in a more or less diagrammatic manner and with the pressure followers or floating pistons of the intermediate stage pressure repeating hydraulic power amplifier means and the main valve spool in neutral positions.

Fig. 2 is a fragmentary sectional view of the left hand pressure repeating hydraulic power amplifier means shown in Fig. 1 and showing its pressure follower or floating piston in a position to the left of neutral.

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Fig. 3 is a similar view thereof but showing its pressure follower or floating piston in a position to the right of neutral.

Referring to the drawing, the valve is shown as having a body 10 provided with an elongated cylindrical chamber 11. In its central region the body wall of the chamber is shown as provided with three annular grooves 12, 13 and 14 at axially spaced intervals therealong. The central groove 13 is shown as communicating with a pressure port 15. The groove 12 is shown as communicating with a drain or return port 16 and the groove 14 via a fluid channel 17 is also shown as communicating with the return port 16. The portion of the chamber 11 between the grooves 12 and 13 is shown as being in communication with an actuating port 18. The portion of the chamber 11 between the grooves 13 and 14 is shown as being in communication with a second actuating port 19. The pressure port 15 and return port 16 are connected with any suitable hydraulic system (not shown). The actuating ports 18 and 19 are connected with external hydraulic machinery indicated schematically and represented by the numeral 20. Thus, the servo valve has inlet, outlet and actuating ports for supplying and receiving fluid to and from the external hydraulic machinery 20 to be actuated.

A valve spool 21 is slidably arranged in the chamber 11 and is shown as having a central lobe 22 and two end lobes 23 and 24. When the valve spool 21 is in its centered or neutral position as shown in Fig. 1, the central lobe 22 covers the central annular groove 13 thereby closing off the pressure port 15, and the end lobes 23 and 24 cover the end grooves 12 and 14 thereby closing off the return or drain port 16. In this centered or neutral position of the valve spool, neither actuating port 18 or 19 has communication with either the pressure port 15 or the drain port 16.

In order to so center the valve spool 21, a spring 25 is operatively interposed between each end of this valve spool and the adjacent end wall of the body chamber 11. The springs 25 are preloaded and can be adjusted by suitable means, such as the adjusting screw 26, so as to position the valve spool in the centered or neutral position illustrated in Fig. 1 when there is no hydraulic drive on the valve spool. The end portion of the body chamber 11 which houses each spring 25 provides a spool end chamber, one being designated at 27 and the other at 27a.

Means are provided for producing a pressure differential in the spool end chambers 27 and 27a so as to drive the valve spool 21 hydraulically. Such means are shown as including a hydraulic amplifier first stage represented generally by the numeral 30 which produces an output pressure differential proportionate to an electrical signal input.

The preferred electro-hydraulic amplifier 30 is shown as having a solenoid or torque motor 31, arranged in a compartment 32 in the valve body, and adapted to move a pressure regulator member 33. A pair of nozzles 34 and 35 are arranged in a separate compartment 36 which provides a sump chamber. The end portion of the pressure regulator member 33 remote from the torque motor 31 extends between the discharge openings of the nozzles 34 and 35 and in a variable spaced relation thereto. The pressure regulator member 33 intermediate its ends is mounted on a flexure tube 37. The position of the flapper portion of the pressure regulator with respect to the discharge openings of the nozzles 34 and 35 provides variable annular orifices which develop a pressure differential within the nozzle chambers.

It will be observed that the solenoid or torque motor 31 is isolated from the sump chamber 36 into which the nozzles 34 and 35 discharge. Such a dry solenoid

type of electro-hydraulic amplifier is more fully described as to construction and operation in the said Moog application, Serial No. 560,573. Instead of the preferred dry solenoid type of first stage amplifier shown, an immersed solenoid type may be employed such as is fully described in the other of said Moog applications, Serial No. 356,631, now Patent No. 2,767,689. Both such types of electro-hydraulic amplifiers are of the balanced nozzle design and, regardless of which type is employed in the practice of the present invention, provide an output pressure differential proportionate to the electrical signal input to the solenoid or torque motor 31.

Means are provided for supplying the electro-hydraulic amplifier 30 with fluid derived from the hydraulic system connected to the pressure and return ports 15 and 16, respectively. The fluid feed is by means of independent branch fluid feed channels 38 and 39 severally communicating at one end with the chambers or interiors of the nozzles 34 and 35. At their other ends the channels 38 and 39 communicate with a main fluid feed channel 40 which leads to the annular pressure groove 13, in turn connected to the pressure port 15. Each of the channels 38 and 39 is shown as having a restriction 41 therein so that fluid supplied to the nozzles is at a lower pressure than the hydraulic system supply pressure applied to the port 15. A fluid drain channel 42 having a restriction 43 therein is shown as placing the sump chamber 36 in communication with the annular drain groove 14, in turn connected via the channel 17 to the return port 16.

As conventionally constructed, servo valves of this type have the branch feed channels 38 and 39 on the downstream side of the restrictions therein connected directly to the spool end drive chambers 27 and 27a so as to apply to the ends of the spool the pressure differential developed by the nozzles in response to an electrical input signal. Such is the arrangement shown in the two aforementioned Moog patent applications. It will be seen that with such a direct connection the speed of response of the valve spool is dependent upon the flow rate through the restrictions 41 in the nozzle branch fluid feed channels 38 and 39. This flow rate is limited so that when the spool diameter is increased to provide an output flow rate through the actuating ports 18 and 19 higher than a predetermined amount, the dynamic response or speed of response of the valve suffers. This is due to the fact that greater volumetric changes must be made in the spool end drive chambers 27 and 27a for a given displacement of the valve spool. In other words, a greater amount of drive fluid must be introduced and withdrawn from the spool end drive chambers.

It is the fundamental purpose of the present invention to overcome this loss in dynamic performance so that a servo valve with a high output power rate can also have a high frequency response. In accordance with the present invention, hydraulic power amplifier means responsive to the output pressure differential of the hydraulic amplifier first stage 30 are arranged to supply drive fluid having a high flow rate to the spool end drive chambers 27 and 27a but under such conditions that the drive fluid is applied to the ends of the valve spool 21 at the same pressure differential. This is shown as accomplished by operatively interposing a pressure repeating power amplifier 45 between the nozzle 34 and the spool end chamber 27, and a similar pressure repeating power amplifier 45a between the nozzle 35 and the spool end chamber 27a. Since the two amplifiers 45 and 45a are identical in construction a detailed description of one will suffice for both. The same reference characters employed in the description of the amplifier 45 indicate like parts for the amplifier 45a, except as distinguished by the suffix *a*.

The valve body 10 is shown as formed with a chamber to provide a cylinder 46 in which a floating piston or pressure follower 47 is slidably arranged. The interior of the cylinder 46 at the end thereof toward the nozzle

34 is shown as communicating with the chamber of this nozzle via the fluid channel 48 which leads to the branch fluid feed channel 38 on the downstream side of the restriction 41 therein. In the case of the other nozzle 35, the corresponding channel 48a leads from the cylinder 46a to the branch nozzle feed channel 39. Thus any pressure differential in the nozzle chambers is also established in the output fluid channels 48 and 48a.

The wall of the cylinder 46 is also shown as having two axially spaced annular grooves 49 and 50. The groove 49 is shown as communicating with the main fluid feed channel 40 via an auxiliary pressure channel 51. The groove 50 is shown as communicating with the drain groove 12 via an auxiliary drain channel 52. In the case of the amplifier 45a, the corresponding drain channel 52a connects with the drain groove 14. Thus, the groove 49 being connected to the supply of pressurized fluid becomes an auxiliary pressure port; and the groove 50 being connected to the hydraulic system return line becomes an auxiliary drain port. A fluid channel 53 establishes communication between the spool end drive chamber 27 and the interior of the cylinder 46, such channel being shown as ported to this cylinder centrally between the auxiliary pressure and drain ports 49 and 50 respectively.

The floating piston or pressure follower 47 is shown as having an annular peripheral recess in its central part so as to form two end lobes 54 and 55. It is to be noted that the spacing between the opposing inner end faces of the lobes 54 and 55 is the same as the axial spacing between the adjacent inner edges of the annular grooves 49 and 50. The lobes 54 and 55 are individually longer in an axial direction than the widths of the respective grooves 49 and 50 with which they are associated. Thus, with the floating piston or pressure follower 47 in its centered or neutral position as shown in Fig. 1, it will be seen that both grooves 49 and 50 are fully covered by the lobes 54 and 55 respectively, but one of these grooves is adapted to be uncovered while the other remains covered when the floating piston moves from its neutral position, as shown in Fig. 2 for a movement to the left and as shown in Fig. 3 for a movement to the right.

The floating piston or pressure follower 47 is shown as provided with a by-pass fluid channel 56 leading to opposite sides of the lobe 54. This by-pass channel 56 provides a passage for fluid to and from the end portion of the cylinder 46 to the left of the lobe 54, as may be occasioned by movement of the piston 47, and also assures that the pressure in this left end cylinder portion is the same as that which obtains in the annular space or intermediate chamber between the piston lobes and hence the same as the pressure in the spool end chamber 27.

In this manner the effective pressure on the left side of the floating piston or pressure follower 47 is static will at all times be the same as the pressure in the spool end chamber 27, and therefore the pressure on the right side of the floating piston or pressure follower 47 will at all times be the same as the pressure in the nozzle chamber 34. In like fashion, the opposite sides of the floating piston or pressure follower 47a of the other amplifier 45a will at all times be exposed severally to the pressures in the nozzle chamber 35 and spool end chamber 27a.

Thus, the hydraulic power amplifiers 45 and 45a can service the spool end drive chambers 27 and 27a respectively with high flow rate hydraulic drive fluid and in doing so repeat in these chambers the pressure differential required and established by the hydraulic amplifier first stage 30.

Assuming that there is no electrical input signal to the hydraulic amplifier first stage 30, the output pressures in the nozzle chambers 34 and 35 will be the same and the floating pistons or pressure followers 47 and 47a will take their centered or neutral position in which the respective auxiliary ports 49, 50 and 49a, 50a will be fully closed, as shown in Fig. 1. The pressure in the

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nozzle chamber 34 and spool end drive chamber 27 is the same, and the pressure in the nozzle chamber 35 and spool end drive chamber 27a is the same. The main valve spool 21 is in its centered or neutral position as shown in Fig. 1.

Assume now that the electrical input signal is such as to induce a pressure differential in the nozzle chambers 34 and 35 with the pressure in chamber 34 being higher than that in chamber 35. This drives the floating piston 47 to the left, as shown in Fig. 2, and the floating piston 47a also to the left. The piston 47 establishes communication between the spool end chamber 27 and the auxiliary pressure port 49, thereby admitting high flow rate hydraulic fluid into this chamber. At the same time, the piston 47a establishes communication between the spool end chamber 27a and the auxiliary drain port 50a. As a result, the main valve spool 21 is hydraulically driven to the right until a force balance is established with the springs 25. Keeping in mind that the system supply pressure in the pressure port 15 exceeds at all times any pressure developed in either of the nozzle chambers 34 and 35, the floating pistons 47 and 47a return to their neutral positions after the flow of hydraulic drive fluid stops which is determined by the final displaced position of the main spool valve 21. Even though the pistons 47 and 47a have so re-centered themselves the pressure differential in the nozzle chambers 34 and 35 is repeated in the corresponding spool end chambers 27 and 27a to maintain the force balance with the springs 25. For given characteristics of the springs 25, the extent of displacement of the main valve spool 21 is proportionate to the strength of the electrical signal fed into the servo valve. This controls actuation of the external hydraulic machinery 20.

Now assuming that the electrical input signal is so varied that the pressure differential in the nozzle chambers 34 and 35 changes with the result that the pressure in the nozzle chamber 34 decreases and the pressure in the nozzle chamber 35 increases, the following occurs. The floating piston 47 shifts to the right, as shown in Fig. 3, following the decrease in pressure in the nozzle chamber 34 and establishes communication between the spool end chamber 27 and the auxiliary drain port 50. The floating piston 47a shifts to the right, following the increase in pressure in the nozzle chamber 35 and establishes communication between the spool end chamber 27a and the auxiliary pressure port 49a. The result is to admit high flow rate hydraulic drive fluid to the right end of the main valve spool 21 and withdraw drive fluid from the left end of this spool, thereby driving the spool to the left until a force balance is obtained with the springs 25. When the main valve spool returns to a static condition, although displaced from a neutral position, the floating pistons 47 and 47a return to their respective centered or neutral positions as previously explained.

It is to be noted that under static, i.e., non-flow, conditions, the floating pistons 47 and 47a will remain in their respective centered or neutral positions in which their end lobes close or lap grooves 49, 50 and 49a, 50a, respectively. For example, if floating piston 47 tends to move to the left auxiliary pressure port 49 will be uncovered thereby admitting high supply pressure fluid into the space between the end lobes 54, 55. Inasmuch as this space is in communication with the left end face of the floating piston 47 through the by-pass channel 56, this high supply pressure will be effective against such left end face to drive the piston to the right so as again to lap the groove or auxiliary pressure port 49. On the other hand, if floating piston 47 tends to move to the right the groove or auxiliary drain port 50 will be uncovered thereby connecting the left end face of the piston to drain. Inasmuch as the pressure on the right end face of the floating piston 47 is then subjected to a pressure greater than drain pressure, the predominant force against the right end face of the piston will drive it to

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the left so as again to lap the auxiliary drain port 50. The piston 47 will always seek a position in which the opposing forces acting on it will be balanced and, under static conditions, this will be only when the pressure in the nozzle chamber 34 is equal to that in the spool end chamber 27. What has been described for the floating piston 47 applies equally to the other floating piston 47a. Thus it will be seen that the floating pistons 47 and 47a are not freely movable in their respective chambers under static conditions.

Of course, suitable filtering means may be incorporated in the hydraulic circuit of the present servo valve at various places in order to maintain the hydraulic fluid free of foreign material. Such filtering means have not been illustrated herein in order to confine the disclosure to the essentials of the valve.

From the foregoing, it will be seen that the present invention provides a three stage electro-hydraulic servo valve having a high dynamic response and power amplification.

I claim:

1. In a flow-control electro-hydraulic servo valve, a chamber having inlet, outlet and actuating ports for supplying and receiving fluid to and from external hydraulic machinery to be actuated, a valve spool slidably arranged in said chamber for controlling fluid flow through said ports, spring means yieldably opposing movement of said spool, and means for hydraulically driving said spool against the urging of said spring means, including hydraulic amplifier means arranged to produce an output pressure differential proportionate to an electrical signal input to the valve and hydraulic power amplifier means responsive to said output pressure differential for applying fluid at a proportionate pressure differential to the ends of said spool.

2. In a flow control electro-hydraulic servo valve, a chamber having inlet, outlet and actuating ports for supplying and receiving fluid to and from external hydraulic machinery to be actuated, a valve spool slidably arranged in said chamber for controlling fluid flow through said ports, spring means yieldably opposing movement of said spool, and means for hydraulically driving said spool against the urging of said spring means, including electrically actuated hydraulic means arranged to produce two output pressures having a difference in value proportionate to an electrical input signal, a first hydraulic power amplifier means responsive to one of said output pressures for applying fluid at a proportionate pressure to one end of said spool and a second hydraulic power amplifier means responsive to the other of said output pressures for applying fluid at a proportionate pressure to the other end of said spool.

3. The combination set forth in claim 2 wherein the supply of fluid to said inlet, electrically actuated hydraulic means and first and second hydraulic power amplifier means, are connected to a common source.

4. In a flow control electro-hydraulic servo valve, a chamber having inlet, outlet and actuating ports for supplying and receiving fluid to and from external hydraulic machinery to be actuated, a valve spool slidably arranged in said chamber for controlling fluid flow through said ports, spring means yieldably opposing movement of said spool, and means for hydraulically driving said spool against the urging of said spring means, including electrically actuated hydraulic means arranged to produce two output pressures having a difference in value proportionate to an electrical input signal, a first hydraulic power amplifier means responsive to one of said output pressures for applying fluid at a proportionate pressure to one end of said spool and a second hydraulic power amplifier means responsive to the other of said output pressures for applying fluid at a proportionate pressure to the other end of said spool, at least one of said hydraulic power amplifier means comprising a cylinder having auxiliary pressure and drain ports, a floating piston slidably

arranged in said chamber and having a recess in its periphery intermediate its ends so as to leave portions adapted to cover both of said auxiliary ports when in a neutral position and also adapted when said piston is moved in either direction from said neutral position to uncover one of said auxiliary ports and thereby establish communication between it and the intermediate chamber formed by said recess while maintaining the other auxiliary port covered, one of the sides of said piston being exposed at all times to the corresponding one of said output pressures and the other side of said piston having fluid communication at all times with said intermediate chamber and also with the corresponding end of said spool.

5. The combination as set forth in claim 4 wherein the fluid communication between said other side of said piston and intermediate chamber is provided by a by-pass channel in said floating piston.

6. In a flow control electro-hydraulic servo valve, a chamber having inlet, outlet and actuating ports for supplying and receiving fluid to and from external hydraulic machinery to be actuated, a valve spool slidably arranged in said chamber for controlling fluid flow through said ports, a portion of said chamber at each end of said spool providing a spool end chamber, spring means yieldably opposing movement of said spool, and means for hydraulically driving said spool against the urging of said spring means by establishing a fluid pressure differential in said spool end chambers, such driving means including electrically actuated first stage hydraulic amplifier means arranged to produce two output pressures in separate output channels and having a difference in value proportionate to an electrical input signal and independent second stage hydraulic power amplifier means operatively interposed between the corresponding pair of said output channels and spool end chambers, each of said second stage amplifier means comprising a cylinder communicating at one end with the corresponding one of said output channels and also having axially spaced auxiliary pressure and drain ports, a floating piston slidably arranged in said cylinder and having spaced lobes, the spacing between the opposing inner end faces of said lobes being the same as the axial spacing between the adjacent edges

of said auxiliary ports whereby said lobes are adapted severally to cover said auxiliary ports when said piston is in a neutral position, each of said lobes being longer in an axial direction than the width of that one of said auxiliary ports with which it is associated whereby when said piston is moved in either direction from said neutral position one of said auxiliary ports is uncovered while the other is maintained covered, one of the sides of said piston being exposed at all times to the pressure in said corresponding one of said output channels, and fluid channel means at all times communicating the corresponding one of said spool end chambers with the other side of said piston and also with the space between the said lobes thereof, whereby a pressure differential is obtained between said spool end chambers which is proportionate to that between said output channels.

7. The combination as set forth in claim 6 wherein the supply of fluid to said inlet, electrically actuated hydraulic means and both auxiliary pressure ports, are connected to a common source.

8. The combination as set forth in claim 6 wherein the supply of fluid to said inlet, electrically actuated hydraulic means and both auxiliary pressure ports are connected to a common source, and said outlet and both auxiliary drain ports are connected to a common return.

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