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Thayer

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[54] **MULTIPLE TOW-STAGE ELECTROHYDRAULIC SERVOVALVE APPARATUS**

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[52] **U.S. Cl.... 137/596.16,** 137/625.62, 137/625.64

[51] **Int. Cl.** **F16k 11/00**

[58] **Field of Search**..... 137/596.16, 625.64, 625.5, 137/596.15, 596.17, 625.63, 625.62, 85, 625.6; 91/48, 336 A, 444, 411 A

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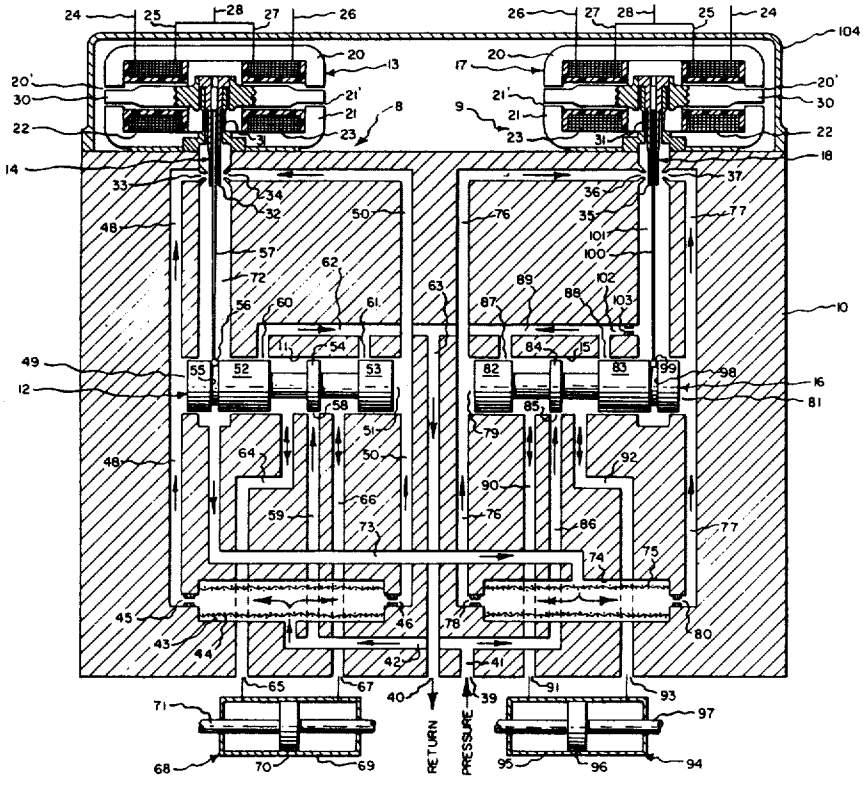
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[57] **ABSTRACT**

Two or more independently controllable two-stage electrohydraulic servovalves are collectively serviced by a single fluid supply and a single fluid return although each servovalve is associated with its own load actuator, by providing each servovalve with a first stage symmetrical hydraulic amplifier and with a second stage valve spool assocaited with metering ports, the hydraulic amplifiers of the various servovalves being connected in series fluid flow communication, so as to maintain quiescent flow essentially the same as for one such servovalve independently, and the metering ports of the various servovalves being in parallel fluid flow communication with the fluid supply and fluid return so as to be capable of developing full differential pressure across the load for each servovalve.

7 Claims, 2 Drawing Figures



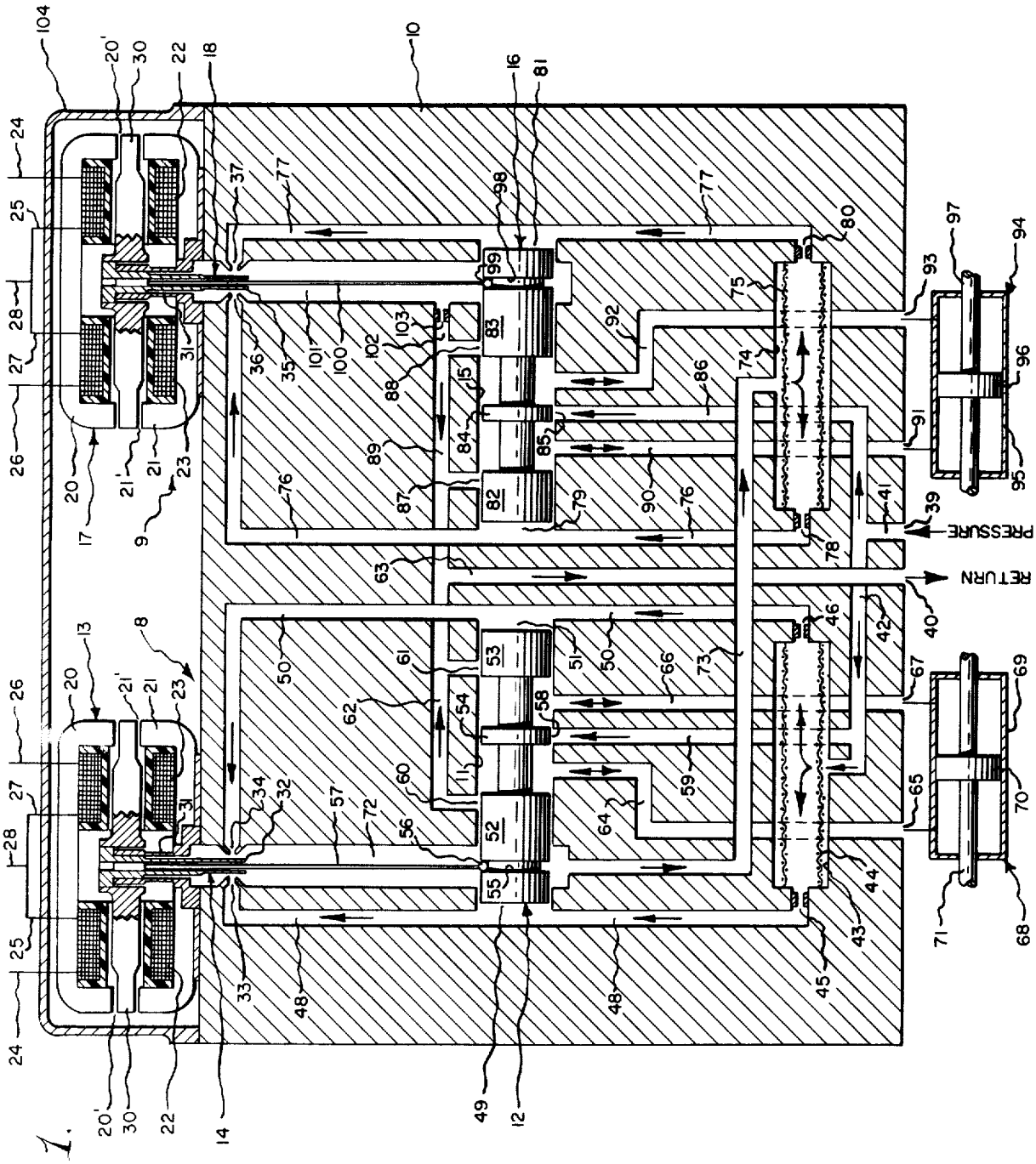


Fig. 1.

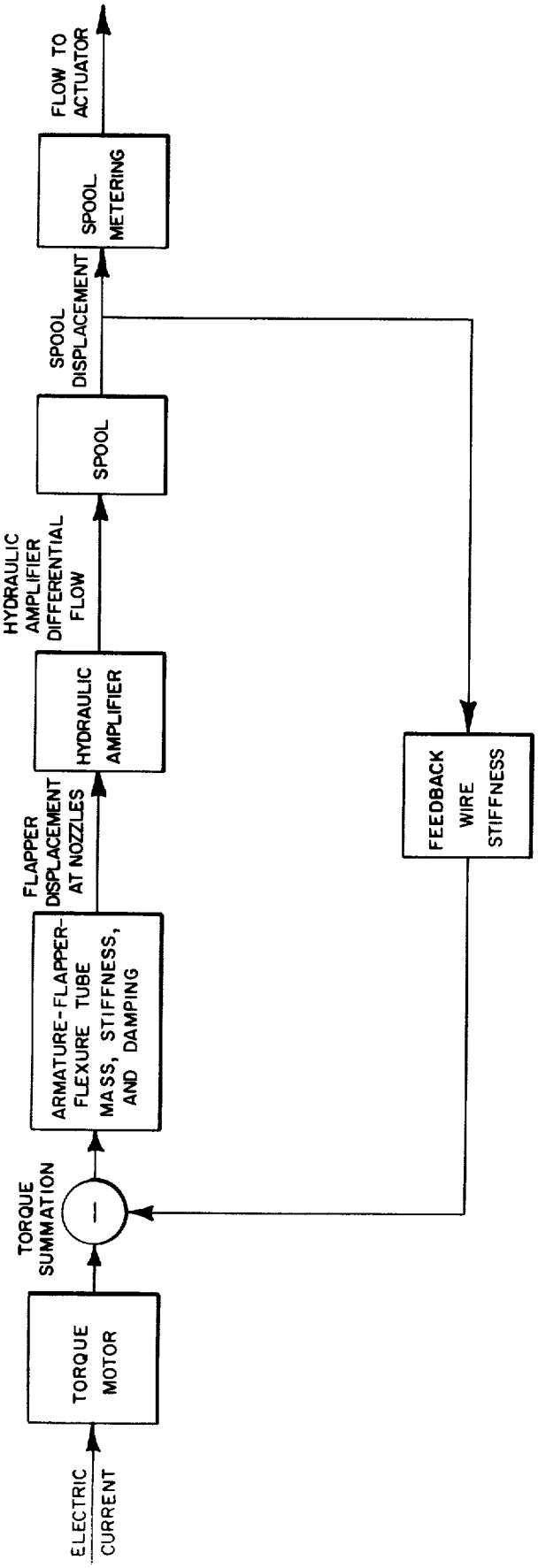

ELECTRIC CURRENT
TORQUE MOTOR
TORQUE SUMMATION
ARMATURE-FLAPPER-FLEXURE TUBE MASS, STIFFNESS, AND DAMPING
FLAPPER DISPLACEMENT AT NOZZLES
HYDRAULIC AMPLIFIER
HYDRAULIC AMPLIFIER DIFFERENTIAL FLOW
SPOOL
SPOOL DISPLACEMENT
SPOOL METERING
FLOW TO ACTUATOR
FEEDBACK WIRE STIFFNESS

Fig. 2.

MULTIPLE TOW-STAGE ELECTROHYDRAULIC SERVOVALVE APPARATUS

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to multiple two-stage electrohydraulic servovalve apparatus, and more particularly to such apparatus which comprises two or more servovalves independently controllable and collectively serviceable by a single fluid supply and a single fluid return.

2. Description of Prior Art

Many applications of electrohydraulic servovalves require the use of two or more servovalves for multiple function control within the same vehicle or equipment. Examples include missile fin control such as pitch, yaw and roll, radar antenna control such as azimuth and elevation, nozzle gimbaling control, laser beam control, jet tab control, and others.

In order to obtain high performance of the control system, two-stage electrohydraulic servovalves having a frictionless hydraulic amplifier are generally preferred. Examples of such servovalves include two-stage double nozzle-flapper servovalves such as disclosed in U.S. Pat. No. 3,023,782, jet pipe servovalves such as disclosed in U.S. Pat. No. 2,884,907, and deflector jet servovalves such as disclosed in U.S. Pat. No. 3,612,103.

A critical requirement for many applications is that power consumption be minimized. This requirement arises as many such systems are carried by missiles or other vehicles or equipment that include a self-contained energy source. Needless power consumption results in increased size and weight of the energy source. For airborne vehicles, this sacrifice in size and weight may penalize range. For hand-held equipment, the increased size and weight of the energy source may decrease the effective payload of the equipment.

Also, in some applications, if the total energy consumption of the control system can be sufficiently low, then a blow-down system may be utilized, such a system employing, for example, a stored cold gas or hot gas propellant charge, together with a one-shot hydraulic accumulator. Such a blow-down system offers simplicity, higher reliability, and lower costs than a conventional battery, electric motor and hydraulic pump configuration.

In many applications of the type mentioned, the duty cycle of the control system consists of an initial flurry of control activity, such as to correct for launch errors, followed by a relatively long period of coast that requires only small, vernier-type course corrections. The energy consumed in the initial control transients is unavoidable, as well as that required for vernier corrections. Superimposed on this necessary control activity, however, is the quiescent power drain of the servovalve hydraulic amplifiers. Often times this quiescent power drain, which continues throughout use of the control system, accounts for more total energy than does the necessary control activity.

A previous approach to solving this problem in some systems heretofore used has been to employ servovalves that have a spool type first-stage hydraulic amplifier. In systems that require low control flow a single-stage servovalve in which an electromagnetic positioning means drives directly a sliding spool valve may suffice. For higher flow systems a two-stage, double spool valve has been used. In either case, the servovalve can be made to have very low quiescent leakage by appropriate overlap of the spool null edges. However, these spool type servovalves suffer from the presence of friction acting directly upon the electromagnetic drive means. This friction seriously compromises valve performance, and only through use of complex means for adding dither, are such valves made workable. The dither requires quiescent power loss, and also adds considerably to the cost of the control system. Also, even with dither, the performance achievable with a servovalve having a sliding spool driven directly by the electromagnetic input means is far less than that obtainable with a servovalve having a frictionless first-stage hydraulic amplifier. In particular, the threshold, hystersis and null shift of single-stage spool type servovalves, or double spool two-stage servovalves, is generally inferior to that obtainable with servovalves having a frictionless first-stage hydraulic amplifier. These performance deficiencies detract from system accuracy and effectiveness.

It would be desirable in many instances to replace both spool type single-stage servovalves, and double spool type two-stage servovalves, with the more advanced two-stage servovalve of the frictionless hydraulic amplifier type, in order to realize the improved system performance. However, this change has not been practicable heretofore without a corresponding change in the stored energy portion of the vehicle or equipment in order to provide the increased energy necessary to sustain the quiescent power drain associated with the continuous flow through the frictionless hydraulic amplifier. Most often this requirement for increased capacity of the stored energy source necessitated change to a battery, motor and pump apparatus with the attendant increase in complexity, weight and cost.

SUMMARY OF THE INVENTION

The solution to the problem afforded by the present invention is based upon recognition of the fact that the quiescent flow in the first-stage of a symmetrical hydraulic amplifier of the nozzle-flapper, jet pipe, or deflector jet types, is essentially constant throughout all levels of signal input, i.e. from 0 to either plus or minus 100 percent signal input.

Further, the solution to this problem afforded by the present invention is based upon recognition of the fact that in the two-stage servovalve that utilizes feedback of spool position, whether achieved through mechanical feedback to the armature of the torque motor or electrical feedback or otherwise, the differential pressure necessary to move the second-stage valve spool, or to maintain the spool in any displaced position, is but a small portion of normal supply pressure.

Still further, the solution afforded to this problem by the present invention is based upon a recognition of the fact that dynamic response of such a two-stage servovalve incorporating spool position feedback is related primarily to the flow gain of the hydraulic amplifier, rather than the pressure gain of the hydraulic amplifier.

Accordingly, the primary object of the present invention is to relate two or more independently controllable two-stage electrohydraulic servovalves of the high performance type severally having: (a) a first-stage hydraulic amplifier which is frictionless but which requires quiescent flow; and (b) a second-stage power

valve, to one another and to a single fluid supply and a single fluid return, such that the quiescent flow for all the first-stage hydraulic amplifiers does not exceed that essentially for one such servovalve if operated by itself, and the full differential pressure between said supply and return is available for load control by the second-stage power valve of each servovalve.

This is achieved, in accordance with the inventive concept, by providing two or more servovalves each including first-stage hydraulic amplifier means capable of producing a differential pressure responsive to an independent electrical command signal, and second-stage fluid flow control valve means responsive to such differential pressure, each of the hydraulic amplifier means during operation requiring input operating fluid and producing output operating fluid, each of the fluid flow control valve means including an actuating port alternately communicable with a pressure metering port and a return metering port determined by the position of a movable valve element arranged to be moved by the differential pressure, the operating fluid for the hydraulic amplifier means of the various servovalves being in series fluid flow communication between the fluid supply and fluid return, the pressure and return metering ports of the various servovalves being in parallel fluid flow communication with the fluid supply and fluid return, respectively, and the actuating ports of the various servovalves being operatively associatable with separate load actuators.

Other objects and advantages of the present invention will be apparent from the following detailed description of a preferred embodiment illustrated in the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a vertical, central, longitudinal, sectional view through multiple independently controllable two-stage electrohydraulic servovalve apparatus embodying the present invention and illustrating the internal construction of two servovalves combined in a more or less diagramatic manner, the valve spool of each servovalve being illustrated in elevation and in a null or centered position, and each servovalve being further depicted schematically in operative association with a separate load actuator.

FIG. **2** is a block diagram indicating a relationship of factors involved between electrical current input to one servovalve of the apparatus and output from such servovalve in the form of flow to the actuator controlled thereby.

DESCRIPTION OF PREFERRED EMBODIMENT

The multiple two-stage electrohydraulic servovalve apparatus of the present invention is shown generally as comprising two two-stage electrohydraulic servovalves severally of the double nozzle-flapper, mechanical feedback, flow control type illustrated and described in U.S. Pat. No. 3,023,782 but consolidated into a single body and related to each other so as to be collectively serviced by a single fluid supply port and a single fluid return port and further arranged so that their first-stage hydraulic amplifiers are connected in series fluid flow communication and their second-stage valve spools are connected in parallel across the supply-to-return differential pressure available.

More specifically and referring to FIG. **1**, the apparatus is shown as comprising left and right electrohydraulic servovalves **8** and **9**, respectively, having a common valve body **10**. This body is internally formed to provide a left cylindrical chamber **11** in which is slidably arranged the second-stage valve spool **12** of servovalve **8** which also includes a torque motor **13** and a hydraulic amplifier **14**. Body **10** is also internally formed with a right cylindrical chamber **15** in which is slidably arranged the second-stage valve spool **16** of servovalve **9** which also includes a torque motor **17** and a hydraulic amplifier **18**.

Each of the electromagnetic torque motors **13** and **17** is suitably mounted on body **10** and is specifically in the form of a polarized force motor including an upper pole piece **20**, a lower pole piece **21** to provide therebetween a pair of air gaps **20'** and **21'** and a pair of coils **22** and **23**. Two permanent magnets (not shown), one on each side of the torque motor between these pole pieces, provide polarization of the air gaps. The coils of each torque motor may be connected in any suitable manner so as to provide differential coils, series coils, parallel coils or single coils. As shown, the leads of coils **22** and **23** are connected in a differential coil configuration. Specifically, coil **22** has two leads **24** and **25**, and coil **23** has two leads **26** and **27**. Leads **25** and **27** are connected together to provide common lead **28**. The tips of an armature **30** is shown arranged in air gaps **20'**, **21'**. This armature is shown as rigidly supported on the upper end of a flexure tube **31** the lower end of which is suitably secured to body **10**.

Hydraulic amplifier **14** of left servovalve **8** is shown as including a flapper **32** rigidly mounted at its upper end on the corresponding flexure member **31** and armature **30** and having its lower end movably arranged between a pair of oppositely facing, spaced apart, fixed nozzles **33** and **34**. In the hydraulic amplifier **18** for the right servovalve **9**, the corresponding flapper is designated **35** and the two corresponding nozzles are designated **36** and **37**.

Valve body **10** is shown as having a pressurized fluid supply port **39** adapted to be connected to any suitable source of pressurized hydraulic fluid (not shown), and also a fluid return port **40** adapted to be connected to any sump or receptacle for collecting return hydraulic fluid (not shown). Pressure port **39** is shown as connected by passage **41** to a branch passage **42**, in turn leading to an enlarged left compartment **43**. This compartment houses a filter screen **44**, schematically illustrated as being of cylindrical configuration. Entrance of the branch passage **42** to compartment **43** is shown as being midway of the length of this compartment. Fluid entering into this compartment flows through the interstices of the filter and thence divides, some of the fluid flowing to the left through a fixed restrictor or orifice **45** and some flowing to the right through a similar fixed restrictor or orifice **46**.

Restrictor **45** is shown as arranged in the lower end of a passage **48** provided in body **10** communicating with compartment **43**, the opposite end of this passage leading to left nozzle **33**. Intermediate its ends, passage **48** intercepts the left end of valve spool compartment **11** to provide a left spool end chamber indicated at **49**. The other restrictor **46** is shown as arranged in one end of a passage **50** provided in body **10** the other end of which leads to right nozzle **34**. Intermediate its ends, this passage **50** intercepts the right end of spool compartment **11** so as to provide a right spool end chamber indicated at **51**.

Left valve spool **12** is shown as having a left end lobe **52**, a right end lobe **53** and an intermediate lobe **54**, these lobes being connected by intermediate stems of reduced diameter so that the valve spool moves as an unit. Left lobe **52** is shown as provided with an annular groove **55** which receives in a substantially frictionless rolling contact manner a ball **56** formed on the lower end of a feedback spring member **57**. The upper end of this member **57** is cantilever-mounted on flapper **32**.

Valve spool **12** is shown in a null or centered position. In this position, center lobe **54** closes a pressure metering port **58** which is connected in fluid flow communication via passage **59** with passage **42**. In this null position of valve spool **12**, left lobe **52** is shown as closing a left return metering port **60**. A similar but right fluid metering port **61** is shown as closed by right spool lobe **53**. These return metering ports **60** and **61** are shown as connected by branch passage **62** with main fluid return passage **63** which leads to fluid return port **40**.

The space between lobes **52** and **54** in compartment **11** is shown as communicating via passage **64** with an actuating port **65**. The space between lobes **53** and **54** is shown as communicating via passage **66** with a second actuating port **67**. Actuating ports **65** and **67** are shown as operatively associated with opposite ends of an actuator **68** including a cylinder **69** housing a movable piston **70** mounted on a piston rod **71** which projects from opposite end walls of this cylinder and is suitably connected to a load (not shown) to be serviced by servovalve **8**.

Nozzles **33** and **34** discharge fluid into an outlet chamber **72** which surrounds left lobe **52** of valve spool **12** and is connected via passage **73** with a right compartment **74** which is similar to left compartment **43** but separate therefrom. Compartment **74** also houses a cylindrical filter screen **75** and has its ends communicating with passages **76** and **77**. Left passage **76** adjacent its lower end has arranged therein a fixed restrictor or orifice **78**. The opposite end of this passage **76** leads to left nozzle **36** of right servovalve **9**. Intermediate its ends, this passage **76** intercepts the left end of compartment **15** so as to provide left spool end chamber **79** for the right valve spool **16**. The other or right passage **77** has therein a similar fixed restrictor or orifice **80** adjacent its lower end. The upper end of this passage **77** communicates with right nozzle **37** of right servovalve **9**. Intermediate its ends, this passage **77** intercepts the right end of compartment **15** to provide right spool end chamber **81**.

Valve spool **16** is shown as having a left end lobe **82**, a right end lobe **83** and a center lobe **84**, these lobes being spaced apart and connected together by intermediate stems of reduced diameter so as to provide an unitary valve spool. This spool is also shown in its null or centered position in which center lobe **84** closes a pressure metering port **85** connected via passage **86** to passage **41**. Left lobe **82** in this position of the spool closes a left return metering port **87**, and right lobe **83** closes a right fluid metering port **88**. These return metering ports **87** and **88** are shown as connected via passage **89** to main fluid return passage **63**.

The space between lobes **82** and **84** of right valve spool **16** is shown as communicating with a passage **90** leading from compartment **15** to a left actuating port **91**. The space between lobes **83** and **84** is shown as connected by a passage **92** leading from compartment **15** to a right actuating port **93**. Actuating ports **91** and **93** are operatively associated with a separate hydraulic actuator indicated generally at **94** comprising a cylinder **95** housing a piston **96** fast to a piston rod **97** which is shown as extending outwardly through the end walls of this cylinder and connected to any suitable load (not shown).

Right lobe **83** of right valve spool **16** is shown as provided with an annular groove **98** on the walls of which a ball **99** has a substantially frictionless rolling contact. This ball **99** is mounted on the lower end of a feedback spring wire member **100** the upper end of which is cantilever-mounted on flapper **35** for right servovalve **9**.

Nozzles **36** and **37** of this right servovalve **9** discharge fluid into an outlet chamber **101** which is shown as having a fluid flow connection with passage **89** via a short passage **102** in which a restrictor or fixed orifice **103** is arranged.

From the foregoing arrangement, it will be seen that each hydraulic amplifier **14** or **18** during operation requires input operating fluid and produces output operating fluid. In the case of left hydraulic amplifier **14**, the operating fluid therefor is derived from fluid supply port **39** flowing through connected passages **41** and **42** into compartment **43**, thence dividing to flow through a left course including restrictor **45**, passage **48** and nozzle **33** into chamber **72** and a right course including restrictor **46**, passage **50** and nozzle **34** into chamber **72**, thence through passage **73** into compartment **74**. The output operating fluid from left hydraulic amplifier **14** entering compartment **74** via passage **73** becomes the input operating fluid for right hydraulic amplifier **18**. From compartment **74** this fluid divides and flows through left and right courses. The left flow course includes restrictor **78**, passage **76** and nozzle **36** into chamber **101**. The right flow course includes restrictor **80**, passage **77** and nozzle **37** into chamber **101**. From this chamber **101** flow passes through restrictor **103** into passage **102** and connected passages **89** and **63** to fluid return port **40**. In this manner the operating fluid flow through these hydraulic amplifiers is in series.

On the other hand, the pressure and return metering ports **58**, **60** and **61** of left spool **12**, and the pressure and return metering ports **85**, **87** and **88** of right spool **16**, are connected in parallel fluid flow communication with fluid supply port **39** and fluid return port **40**, respectively. Thus, pressurized fluid flows from pressure supply port **39** through passage **41** and branch passages **59** and **86** to pressure metering ports **58** and **85**, respectively. Depending upon the position of the valve spools **12** and **16**, these pressure metering ports **58** and **85** can be placed in fluid flow communication with one of their respective actuating port passages such as **64**, **66** in the case of port **58** or passages **90**, **92** in the port **85**, while the other of such actuating port passages is connected with the appropriate return metering port such as **60** or **61** for left servovalve **8**, and **87** or **88** for right servovalve **9**. Thus, fluid which flows from supply port **39** through pressure metering port **58** of left valve spool **12** (while in a displaced position) to one side of actuator piston **70**, in turn displacing fluid from the opposite side of said piston which flows past the appropriate return metering port of left spool **12** to fluid return port **40**, is in parallel with fluid that may similarly flow through right spool **16** for displacement of piston **96**.

Each of the two servovalves operates independently of the other, except that the output operating fluid

from the hydraulic amplifier of left servovalve is utilized as the input operating fluid for the right servovalve. If more than two servovalves are employed, as there can be if desired, the output operating fluid from the second servovalve would be utilized as the input operating fluid for the hydraulic amplifier of the third servovalve, and so on. Insofar as command signals for the various servovalves are concerned, these are applied independently to each valve as electrical current inputs through corresponding leads **24, 26, 28.** Each servovalve is operatively associated with its own load actuator and the second stage of each servovalve has access to the full hydraulic power available, i.e. the pressure differential between supply and return across ports **39** and **40.**

The flexure tube **31** of each servovalve provides complete isolation of the electromagnetic torque motor **13** or **17** and the associated air gaps from the hydraulic fluid, thereby eliminating problems from magnetic contamination. Inasmuch as the exterior of such flexure tube is exposed to atmosphere, both torque motors **13** and **17** can be covered by a single removable cover or cap **104** mounted on body member **10.**

As is well known to those skilled in the art, the air gaps **20'**, **21'** formed by the upper and lower pole pieces **20** and **21,** respectively, of each torque motor are polarized by permanent magnets (not shown) arranged between these pole pieces. These magnets are charged in parallel so that the entire upper pole piece is at the same magnetic potential. Electrical current in the torque motor coils **22** and **23** causes polarization of the armature **30.** The resulting unbalance of flux at the air gaps produces a torque on the armature. This torque is then balanced by torque from the spool feedback wire, **57** for the left servovalve **8** and **100** for the right servovalve **9.**

As previously indicated, the two identical torque motor coils **22** and **23** can be wired with equal ease for series, or parallel, or differential electrical inputs. The differential connection shown is generally used with a push-pull electrical circuit having a nominal quiescent current in each coil.

Each electrohydraulic servovalve operates independently of the other, except for the series fluid flow connection between the hydraulic amplifiers of the valves. In this arrangement, quiescent flow through the hydraulic amplifier **14** of the left servovalve **8,** passes also through the hydraulic amplifier **18** of the right servovalve **9,** and in any succeeding servovalve. The second-stage spools **12** and **16,** however, remain connected in parallel across the total system supply, so are capable of developing full flow and/or full differential pressure across their respective actuators.

Should two servovalves be hydraulically connected in the manner described, then in the typical **3,000** psi hydraulic system, the pressure drop across each first-stage hydraulic amplifier will be 1,500 psi nominally (neglecting the relatively small pressure drop across restrictor **103**). Similarly, with three servovalves, each hydraulic amplifier will have one-third of supply or 1,000 psi; and with four servovalves, 750 psi.

In each case, the quiescent hydraulic amplifier flow gain can be maintained the same by appropriate selection of sizing of the first-stage hydraulic orifices such as **45, 46** and **78** and **80.** Thus, the quiescent flow with two, three, four or more servovalves can be essentially the same as obtainable with a single servovalve having the same dynamic performance. Each individual servovalve in such a combination can operate independently and with essentially no sympathetic response from adjoining servovalves connected in the same control system configuration.

Fixed restrictor or orifice **103** in return passage **102** minimizes the effects of back surges in the return line and improves the discharge flow characteristics of nozzles **36** and **37** of right servovalve **9** by maintaining a back pressure in the outlet chamber **101.** The flow through this restrictor **103** produces a pressure drop in the range of from 100 to 200 psi typically.

The action of a given servovalve of the multiple servovalve apparatus of the present invention is illustrated schematically in FIG. **2.** Referring to that figure, electric current is fed to the torque motor which produces torque on the armature and flapper assembly.

The cantilever feedback spring member of each servovalve likewise produces torque on the armature and flapper assembly corresponding to displacement of the servovalve spool. Any difference between the input torque caused by electrical current, and feedback torque caused by spool displacement, acts to move or displace the flapper between the nozzles. This difference in torques is represented in FIG. **2** by the torque summation point. The actual displacement of the flapper in response to a torque difference will be determined by the combined mass, stiffness, and damping of the armature, flapper and flexure tube assembly. It is known to those skilled in the art that the stiffness of this assembly is determined by the centering spring rate of the flexure tube, together with the decentering spring rate associated with permanent magnet flux in the air gaps of the torque motor.

Displacement of the flapper between the nozzles causes an unbalance in the orifice relationship between the left and right sides of the hydraulic amplifier. The intermediate chambers on each side of the hydraulic amplifiers connect to their respective spool end chambers as previously described. Thus, the spool will be responsive to unbalance of the hydraulic amplifier.

The spool during all normal operations will move whenever very small pressure differentials exist between its spool ends. However, the velocity of spool motion will be determined by the unbalance in hydraulic amplifier flow that results from flapper displacement. Thus, spool displacement will be the integral of the spool velocity.

Therefore, it is seen that the spool displacement in a two-stage servovalve of the spool feedback type (where no spool centering springs are necessary) is related to the differential flow characteristics of the hydraulic amplifier, rather than to the differential pressure characteristics of this amplifier. There is a small, but negligible, centering spring effect associated with displacement of the cantilever feedback spring caused by spool displacement.

Thus, performance of such a two-stage servovalve of the spool feedback type is influenced primarily by the flow characteristics of the hydraulic amplifier.

Further, with the basic symmetry inherent in a double nozzle servovalve of the type described, or the summetry also inherent in a jet pipe or deflector jet servovalve, the total quiescent flow through the hydraulic amplifier will be essentially constant throughout all positions of the first-stage moving valve element, or for all positions of the second-stage moving valve element.

From the foregoing, it will be seen that quiescent flow through the series-coupled hydraulic amplifiers is essentially constant at all times whether or not an electrical current input is applied to the torque motor coils. Moreover, the differential pressure developed by a given hydraulic amplifier to move or hold its valve spool is a small portion of the pressurized fluid supply. Further, the dynamic performance of a servovalve having spool position feedback is related to the flow gain of its hydraulic amplifier and flow gain is related to hydraulic amplifier quiescent flow. Thus the multiple two-stage electrohydraulic servovalve apparatus of the present invention provides an alternate valve configuration possessing significantly improved performance over the prior art type of apparatus which employed servovalves having a spool type first-stage hydraulic amplifier.

What is claimed is:

1. Multiple two-stage electrohydraulic servovalve apparatus collectively serviceable by a single fluid supply and a single fluid return, comprising at least two servovalves each including first-stage hydraulic amplifier means capable of producing a differential pressure responsive to an electrical command signal and second-stage fluid flow control valve means responsive to said differential pressure, each of said hydraulic amplifier means utilizing input operating fluid and producing output operating fluid, each of said fluid flow control valve means including an actuating port alternately communicable with a pressure metering port and a return metering port determined by the position of a movable valve element arranged to be moved by said differential pressure, the operating fluid for the hydraulic amplifier means of the various servovalves being in series fluid flow communication between said fluid supply and said fluid return, the pressure and return metering ports of the flow control valve means of the various servovalves being operatively connected with said fluid supply and said fluid return, respectively, and the actuating ports of the flow control valve means of the servovalves being operatively associatable with separate load actuators.

2. Apparatus according to claim **1** which further comprises position feedback means operatively interposed between said flow control valve means and said hydraulic amplifier means of each of said servovalves.

3. Apparatus according to claim **2** wherein each of said hydraulic amplifiers includes a pivotal element and each of said servovalves further comprises a flexure tube isolating the hydraulic section from the electrical section of the servovalve and also serving as a frictionless pivotal support for said pivotal element.

4. Apparatus according to claim **2** wherein each of said hydraulic amplifiers is of the double nozzle-flapper type.

5. Apparatus according to claim **4** wherein said position feedback means comprises a spring member operatively interposed between said flow control valve means and the corresponding hydraulic amplifier means.

6. Apparatus according to claim **4** wherein said position feedback means comprises a spring member cantilever-mounted at one end on the flapper of the corresponding hydraulic amplifier and having its other end constrained to move with the corresponding valve element.

7. Apparatus according to claim **6** wherein each of said servovalves further comprises a flexure tube isolating the hydraulic section from the electrical section of the servovalve and also serving as a frictionless pivotal support for said flapper.

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