

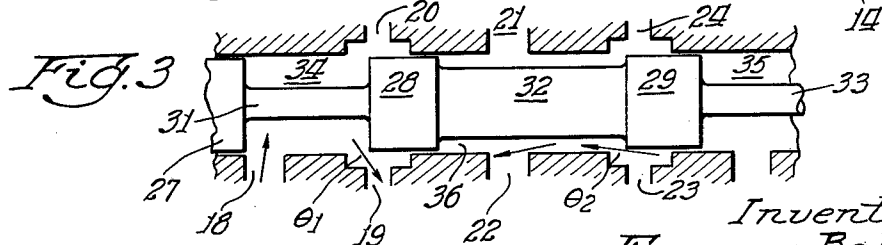
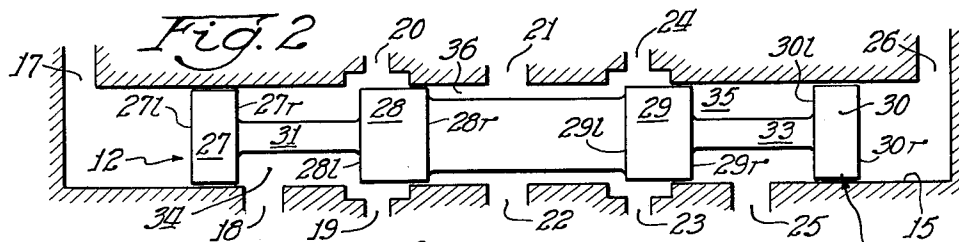
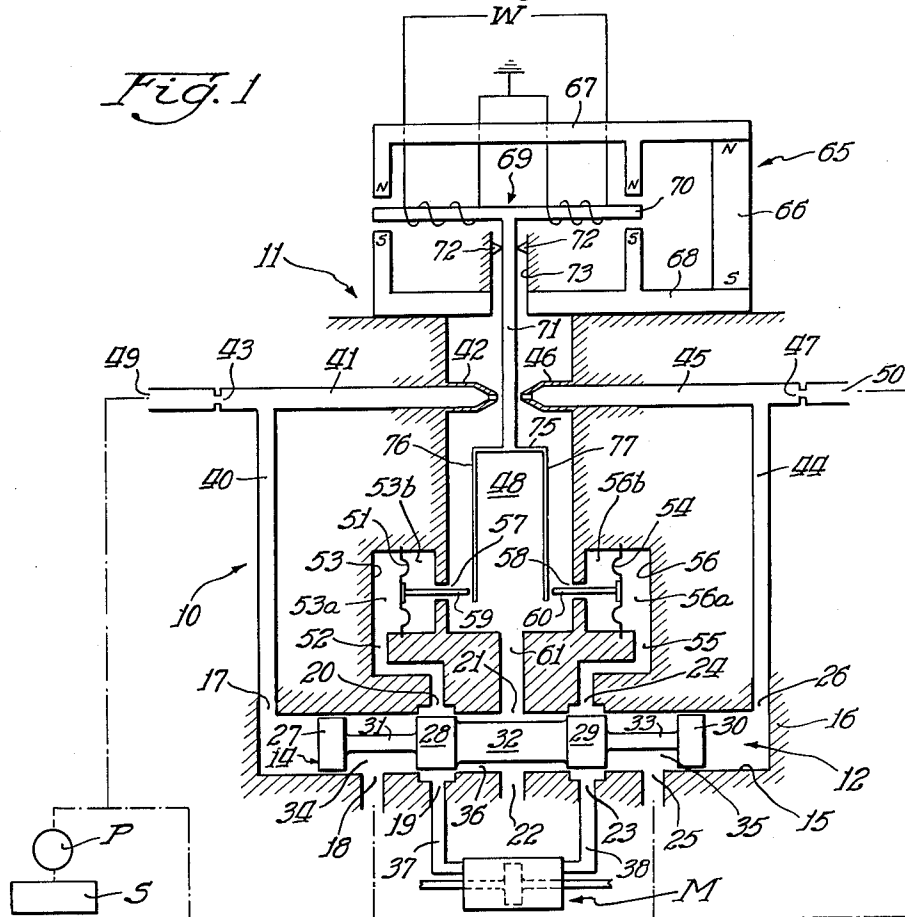
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CONTROL VALVES

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CONTROL VALVES

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This invention relates generally to hydraulic control or power valves, and more particularly to improvements in such control valves as applied in electro hydraulic servo-mechanisms.

Valves of this type are normally used to control the supply and/or pressure of hydraulic fluid to a motor or other such hydraulically operated devices. In normal operation of the embodiment herein described, the spool is moved in a certain direction thereby causing one or more fluid ports to be progressively opened for the purpose of allowing fluid from the supply source to reach the fluid motor and return.

Valves of this type are often employed in combination with an electrical device which receives an electrical signal which is interpreted in the form of a low level force. The electrical device transmits the signal to the control valve for the purpose of actuating said control valve in response to the electrical input. Ideally, the purpose of the combination herein referred to as an electro-hydraulic servo, is to increase or decrease fluid pressure supplied to the hydraulic output member in response to demands communicated to the valve by the electric signal.

It is highly desirable that the response of the control valve be linear, i.e., that the change in hydraulic pressure to the output member be linear with respect to the input signal sensed by the electrical mechanism. It is also very desirable that the amount of force required to move the spool of the control valve be as small as possible in order that the electrical input signal may be as small as possible.

However, there are certain forces which inherently occur in conventional valves of this type which act in derogation of these desirable characteristics. These forces have been the subject of extensive research, voluminous publications, and a number of patented valve mechanisms, none of which have served to satisfactorily solve the numerous attendant problems.

The forces which create the most concern are those created by fluid flow through the orifices created by the lands of the spool partially opening the valve ports. These forces may be grouped into general categories, the first category being referred to as steady state forces, and the second group being termed transient flow forces.

One of the more recent learned works relating to these forces is found in the A.S.M.E. transactions of 1952, given by Dr. Lee and J. F. Blackburn of the Massachusetts Institute of Technology. In a series of two papers, a mathematical derivation of steady state and transient flow forces was presented, and these derivations are generally accepted for the purpose of this invention. In these papers a number of possible solutions to the problems arising from the presence of these forces are set forth, among which is a new port configuration subsequently patented by Dr. Lee, Patent No. 2,747,612. It is to be noted that this configuration is particularly concerned with elimination of steady state forces.

The forces with which the present invention is most concerned are those known as transient flow forces. Basically, these transient flow forces are the summation of the reaction forces to the acceleration of the individual particles of fluid entering and leaving the valve and in conventional valve designs they are impressed on the movable valve element. This tends to cause valve instability, as well as other undesirable characteristics.

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According to the papers of Lee and Blackburn, these forces are a function of the length of the flow path and the angle of fluid flow entering and/or leaving the valve. The angle of the fluid flow leaving or entering a metering port is generally referred to as the jet flow angle. However, for the purposes of the present invention, jet flow angle is interpreted to be the flow entering or leaving the valve from both metering and nonmetering ports.

On the basis of work by Von Mises, discussed in the papers before referred to, it has been generally accepted that the jet angle must remain constant, or substantially so, at 69°. Therefore, most attempts to dampen the forces acting on the spool have been in the area of varying the length of the flow path.

Basically, it is the intent and purpose of this invention to control transient flow forces in order to prevent valve instability and oscillation which is an effect of these forces, whereby a valve may be provided which will be linearly responsive, and stable in operation.

Accordingly, it is an object of the present invention to provide a control valve having the above stated characteristics which may function in cooperation with a low input servo mechanism.

It is a further general object to provide a valve which is not subject to the effects of adverse fluid forces which would otherwise contribute to instability and nonlinearity of operation.

It is a more particular object of the present invention to substantially control the effects of steady state and transient forces resulting from fluid flow to or from the valve ports.

It is a further object of the present invention to provide a valve having the above advantages without increasing the expense thereof through use of exotic land and port configurations, but retaining the conventional square edged land in order that no sacrifice to response be made.

It is a more particular object of the present invention to accomplish the above advantages and their desirable results by changing or altering the angle of fluid flow entering and leaving the spool.

It is a specific objective of the present invention to cause the angle of ingress and egress of the fluid in the valve to be altered by modifying the relative diameter of the spool between certain lands in relation to the diameter of the valve bore.

The invention consists of the novel constructions, arrangements, and devices to be hereinafter described and claimed for carrying out the above stated objects and such other objects as will be apparent from the following description of the preferred form of the invention, illustrated with reference to the accompanying drawings, wherein:

FIG. 1 is a schematic view, partially in section, illustrating a preferred embodiment of a servo mechanism including a control valve.

FIG. 2 is an enlarged sectional view of the control valve of FIG. 1.

FIG. 3 is a scalar diagram of the fluid flow through a port of the control valve of FIG. 2.

Referring now to the figures wherein like numerals refer to like parts in the various views, an electro-hydraulic servo mechanism is indicated generally at 10. The servo 10 comprises an electro-hydraulic preamplifier generally indicated at 11 and a control or transfer valve assembly generally indicated at 12.

Referring first to the hydraulic control or transfer valve 12, a movable valve member, or spool is slidably retained in a bore 15 in a casing 16. The casing 16 is formed with a plurality of ports 17, 18, 19, 20, 21, 22, 23, 24, 25, and 26. The spool 15 is of substantially cylindrical con-

struction formed with a plurality of enlarged diameter portions, or lands 27, 28, 29 and 30. Lands 27 and 28 are separated by a valve body portion 31; lands 28 and 29 are separated by valve body portion 32 and lands 29 and 30 are separated by valve body portion 33. Each land is formed with a pair of opposed faces substantially perpendicular to the axis of the spool 14. Land 27 has faces 27l and 27r; land 28 has faces 28l and 28r; land 29 has faces 29l and 29r; and land 30 has faces 30l and 30r.

Valve body portion 32 is relatively greater in diameter than valve body portions 31 and 33, and is of a slightly smaller diameter than the lands 27, 28, 29 and 30.

Lands 27 and 28 define in conjunction with valve body portion 31, and bore 15, a fluid chamber 34. Lands 29 and 30, in a like manner, define a substantially identical fluid chamber 35 in conjunction with valve body portion 33 and bore 15. A third fluid chamber 36 is defined by lands 28 and 29, valve body portion 32 and bore 15. Fluid chamber 36, due to the relatively large diameter of valve body portion 32 is of a substantially lesser volume than chambers 34 and 35.

Port 19 is connected to conduit 37 which is connected to a fluid motor M. Port 23 is connected to conduit 38 which is in communication with the opposite side of the motor M. The valve 12 is operable to select the direction of fluid flow by connecting port 19 or 23 with the fluid pressure source.

Port 18 is in communication with a source of fluid pressure provided by a pump P, as is port 25, and port 22 is an exhaust port and is in communication with the sump S.

Port 17 is connected to conduit 40 which is in communication with branch conduit 41. Branch conduit 41 is in communication with a fixed nozzle 42 at one end thereof, and a fixed orifice 43 at the other end thereof. Similarly, port 26 is in communication with conduit 44 which in turn communicates with branch conduit 45. Branch conduit 45 is connected to nozzle 46 at one end thereof, and orifice 47 at the other end. Nozzles 42 and 46 are situated in case 16 in a position diametrically opposed to one another in a central discharge chamber 48. The orifices 43 and 47 are connected to the pressure source through ports 49 and 50 respectively.

Port 20 is in communication with a diaphragm 51 through a conduit 52. A diaphragm 50 is secured in any manner well known in the art in the casing 16, and divides an enlarged portion 53 of conduit 52 into chambers 53a and 53b. In a like manner, port 24 is in communication with diaphragm 54 by means of conduit 55 which has an enlarged portion 56. Diaphragm 54 is secured in the enlarged portion 56 of conduit 55 in any manner well known in the art and divides enlarged portion 56 into chambers 56a and 56b. Slots 57 and 58 are provided in casing 16 in order to allow communication of chambers 53b and 56b with discharge chamber 48. Rigid projections or pins 59 and 60 are secured respectively to diaphragms 51 and 54, perpendicular thereto and extend through slots 57 and 58, and into chamber 48.

Port 21 is in communication with chamber 48 through conduit 61 which is in constant communication with port 22 and thence to sump S, thereby maintaining chamber 48 and chambers 53b and 56b at sump pressure at all times.

Referring now to the electro-hydraulic preamplifier 11, there is found an electrical input sensing device 65 comprising a permanent magnet 66 having a laterally extending north pole piece 67 and south pole piece 68, and a mechanical input amplifier 69.

The amplifier 69 which is used here is generally known as a flapper valve. The flapper is of a generally T shape, having a cross bar 70 and stem portion 71. The flapper is mounted on a pair of fulcrums 72 secured to casing 16 in a channel 73 which is in communication with chamber 48.

The cross bar 70 acts as an armature, having a plurality

of winds of electrical conducting wire thereabout. The terminals of the cross bar extend between the pole pieces 67 and 68 of the permanent magnet, and are operable to assume any position between the pole pieces by pivoting about the pivots 72.

The stem 71 of the flapper extends through channel 73 and into the chamber 48. In so doing it passes between the diametrically opposed nozzles 42 and 46, which are continuously discharging fluid against the face of the stem. The discharge from both nozzles is substantially equivalent and opposite in direction thereby providing a balance of fluid forces on the stem. At the terminal of the stem 71 opposite that which joins with the cross bar 70, there is a transverse member 75 to which a pair of relatively flexible members such as cantilever springs 76 and 77 (shown by way of example in FIG. 1) are attached. The cantilever springs extend downward (as seen in FIG. 1) into chamber 48 parallel to stem 71, so as to be in a position opposite the pins 59 and 60, and in such relation so as to allow the pins 59 and 60 to effect a movement of its adjacent cantilever spring in a manner which will be described hereinafter.

In operation, an input signal in the form of an electric current is transmitted to the armature 70 of the flapper 69. The flow of current in the windings about the armature creates a magnetic field having a certain polarity depending upon the direction of current flow. The terminal points of the armatures will accordingly assume the nature of a south or north pole. The pole thus created will then be attracted to its opposite on the permanent magnet 66 and the flapper will rotate accordingly to a position where the magnetic and fluid forces, which consist of the opposing discharge from nozzles 42 and 46, are again balanced. The amount of attraction is a function of the strength of the input signal or current.

In response to the input signal the flapper 69 will be rotated about the fulcrum 72 in amount proportional to the strength of the input signal. The resultant movement will bring the stem 71 relatively closer to one of the nozzles 42 or 46.

Simultaneously, assuming the valve 12 to be initially in a neutral position, fluid pressure is continuously provided to fluid chambers 34 and 35 by means of pump P. In a like manner, fluid under pressure is supplied to ports 49 and 50. The fluid under pressure supplied to ports 49 and 50 passes respectively through orifices 43 and 47 and into conduits 41 and 45 respectively at a reduced pressure as determined by the orifice size and initial pressure of the fluid. The fluid flow in conduit 41 is then divided, a portion thereof being discharged through nozzle 42 as before mentioned while the other portion thereof is passed through conduit 40, through port 17 and into bore 15 where it is impressed upon face 27l of spool 14. In a like manner, the fluid in conduit 45 is divided, a portion thereof being discharged through nozzle 46 while the remaining portion passes through conduit 44, through conduit 46 and into bore 15 at the opposite end thereof where it is impressed upon face 30r of spool 14.

The hydraulic circuit previously described is so constructed that when the valve spool 14 is in the steady state neutral position, the pressures on the spool 14 are everywhere in balance. That is to say, the forces of the fluid on face 27l are equivalent to the forces of the fluid on face 30r, and in a like manner, the fluid forces in each chamber 34, 35 and 36 are balanced so that the valve spool 14 is static.

The fluid being discharged through nozzles 42 and 46 passes into discharge chamber 48, and through conduit 61 and is exhausted to sump.

As previously indicated, the flapper 69 is caused to rotate when an input signal is transmitted to the preamplifier 11. This rotation about the fulcrum 72 causes the stem 71 to move relatively closer to one of the nozzles 42 and 46. In so doing, the stem offers a restriction to

the flow of fluid from the nozzle to which the stem has moved.

For example, assume that an input signal has been received which causes a clockwise rotation of the flapper. The stem 71 is then moved relatively closer to nozzle 42. The result of this movement is a restriction of the fluid flow from the nozzle 42 accompanied by a decrease in restriction of fluid flow from nozzle 46. The fluid flowing in conduit 41, sensing a greater resistance to flow through the nozzle 42 will begin back up in conduit 41 building pressure in conduit 40 and subsequently the fluid force on face 271 will increase.

Simultaneously, more fluid will flow through nozzle 46 which is now relatively less restricted by the stem 71. This change in flow is accompanied by a decrease in pressure in conduit 44 and an accompanying decrease in the fluid forces on face 30r.

This adjustment of the pressures acting on the spool results in a movement of the spool 14 to the right as seen in FIG. 1.

As spool 14 moves to the right, land 28 will gradually uncover ports 19 and 20. The progressive uncovering of port 19 allows fluid under pressure from the pump to pass from chamber 34 through port 19 and into conduit 37 with increasing acceleration to the fluid motor M, which is here shown as of a differential pressure type. As before noted, as port 19 is uncovered port 20 is also uncovered, and in a like manner, fluid under pressure from fluid chamber 34 will flow through port 20, conduit 52, and into chamber 53a where it is imposed upon the diaphragm 51.

As the spool moves to the right, port 23 is simultaneously opened by the land 29, thereby allowing fluid in conduit 38, which is attached to the other side of the differential pressure motor M, to pass through port 23 into chamber 36 and thence to sump through port 22. In this manner, the pressure in conduit 38 is reduced whereby a differential pressure is created on the motor M to actuate the same. As land 28 opens ports 23 and 24 are also opened, thereby exposing chamber 56a to atmospheric pressure through chamber 36 and port 22 to sump.

The spool 14 will continue to move to the right as long as there is an unbalance of fluid forces, in that direction, on it. In order that the spool response will be proportional to the input signal, it is necessary to provide a means for stopping the spool progress when it has successfully opened port 19 the precise amount to provide the fluid pressure differential to the motor which is called for by the input signal. To this end, means is provided in the fluid circuit for sensing the fluid pressure delivered to the motor including means for causing retardation of the valve spool and feeding the information back to the operating mechanism in order that the valve spool may be stopped in a position which will allow the delivery of the desired output pressure.

The before mentioned means includes opposed diaphragms 51 and 54 having transverse pins 59 and 60 respectively mounted thereto. The pins are so positioned as to contact the cantilever springs 76 and 77, when the desired output pressure is attained, for urging the springs, and subsequently the stem 71, away from the particular nozzle which is being restricted thereby in response to the input signal.

The actuation of this "feedback" mechanism is accomplished as follows:

It is seen from FIG. 1 that as the valve spool 14 moves to the right, thereby uncovering ports 19 and 23, that ports 20 and 24, being diametrically opposed (as shown in the drawings), in casing 16, to ports 19 and 23, will also be uncovered at the same time and by the precise amount as ports 19 and 23. Therefore, as the fluid under input pressure which is in chamber 34, flows between land 28 and port 19, so too, fluid at the same pressure flows through port 20. Land 28 in conjunction with ports 19 and 20 form a sharp edged orifice which will cause a

reduction in fluid pressure proportional to the size of the orifice. Therefore, the pressure flowing to the motor M through conduit 37 will be at some pressure less than that input pressure which is in chamber 34. Since the size of the orifice at port 20 is identical to that at the port 19, the pressure in conduit 52, and that subsequently acting against diaphragm 51, will be the same pressure which is applied to the fluid motor M.

Accordingly, and in the same manner, the pressure in chamber 56a acting on diaphragm 54 will be substantially equivalent to the pressure in conduit 38, which represents the pressure on the low pressure side of the motor as a result of the rightward movement of the valve. As before mentioned, the low side pressure is normally at sump pressure due to the direct connection through port 23, chamber 36 and port 22 to sump.

In this manner, the pressures acting on diaphragms 51 and 54 which is sensed by stem 71 through the action of pins 59 and 60 on springs 76 and 77, represents the differential pressure on the fluid motor M which is actuating the same. Accordingly, each diaphragm will respond to the fluid pressure acting thereon, which in the present description will result in a rightward movement of diaphragm 51 and pin 59 against spring 76.

The rightward movement of the pin 59 exerts a force on the spring 76, in proportion to the pressure acting on diaphragm 51, to urge spring 76, and accordingly stem 71, towards the right. At the same time, spring 77 which is in contact with diaphragm 54 by means of pin 60 will be urged towards the right, the result being a tendency to urge the stem 71 in a counter clockwise direction in opposition to the force of the input signal.

The subsequent movement of the stem 71 back in the counter clockwise direction, will, of course, tend to relieve the restriction at the nozzle 42. As the restriction is relieved, the pressure in conduit 40, which is acting on face 271 to move it to the right, will decrease. This readjustment will continue until the unbalance of the pressures on the diaphragms 51 and 54 are no longer effective to cause movement of the stem 71. At this time, theoretically all the forces acting on the spool 14 will be balanced and the spool will be stationary at some position which will provide the precise output pressure called for by the input signal.

In servo systems of the general type described hereinabove, there is sometimes a tendency towards instability which may result in oscillation of the valve and other attendant difficulties. The unstable conditions in systems of this type may be founded in a variety of adverse forces, among which are transient flow forces, which in certain instances may be a determinate cause of instability. FIG. 3 is a scalar diagram of these forces showing how they effect the valve 12. Transient flow forces include the reaction forces to the acceleration and deceleration of the fluid entering and exhausting from the valve.

In the instant example, fluid under pressure passes through the orifice formed between land 28 and ports 19 and 20 at a progressively increasing velocity as spool 14 uncovers the ports. This fluid passes through the orifice at an angle θ_1 . Fluid from the motor M re-enters valve 12 through an orifice formed between land 29 and port 23 at an angle θ_2 .

Remembering that in the example, the valve spool 14 is moving to the right, a particle of fluid in chamber 34 will be accelerated to the right as the spool moves. Accordingly, there is a reaction force to the acceleration of the fluid particle which will be imposed upon the spool in opposition to its direction of motion. The magnitude of the force is a function of jet angle and the amount of fluid in the chamber which is proportional to the length of the chamber, and the rate of fluid acceleration.

In a like manner fluid entering through the orifice between port 23 and land 29 will enter at an angle greater than 90° , the fluid entering having an acceleration to the left. The reaction forces to this acceleration will be in

the same direction as the movement of the spool and therefore will complement the movement of the spool and tend to further accelerate it. This is known as negative damping.

Assuming the fluid flow to enter and leave chambers 34 and 36 at the same angle, it is seen that if the number of particles in chamber 34 is greater than the number of particles in chamber 36 the sum of the reaction forces will be greater acting in opposition to the movement of the spool than the reaction forces acting to complement the movement of the spool, thus causing a very unstable situation in which the valve spool may tend to oscillate at a high frequency and otherwise show unstable tendencies.

It is the basic purpose of this invention to provide a method of controlling the transient flow forces by means of modifying the jet flow angle of the fluid. It has been found that if the particles of fluid entering chamber 36 are allowed to enter and leave in such a manner that the greater portion of the reaction forces will not act upon the spool itself, but rather on the valve casing 16, that the negative damping forces will be reduced. To this end, valve body portion 32 has been greatly increased in diameter so as to cause a decrease in the volume of chamber 36. As seen therefore, in FIG. 3, fluid entering and leaving chamber 36 must then do so at an angle approaching 180°. This being the case, the greater portion of the negative damping forces of the fluid are on the casing 16 and conduit 38.

In chamber 34, however, the valve body portion 31 is of a conventional size, and as shown in FIG. 4, the jet angle of the fluid entering and leaving chamber 34 approaches 90°. Thus an opposite result is attained in that a great portion of the reaction forces to the acceleration of the fluid particles in chamber 34 acts on the spool in opposition to its velocity to the right. This provides a desirable positive damping of the spool with little or no negative damping, and accordingly no valve oscillation or instability.

The total result of the improved structure is the damping of the spool accompanied by the increased stability of the hydraulic valve.

While this invention has been described in connection with certain specific embodiments thereof, it is to be understood that is by way of illustration and not by way of limitation and the scope of this invention is defined solely by the appended claims which should be construed as broadly as the prior art will permit.

What is claimed is:

1. In a fluid system through which fluid under pressure is flowing, a control valve comprising a casing having a bore, a movable valve element slidably retained in said bore, said element being formed with a plurality of body

portions and a plurality of lands, said body portions and said lands defining, in co-operation with said casing, a plurality of fluid chambers, each of said chambers having at least one port to allow the intake or exhaust of fluid, said movable element being operable to allow fluid to exhaust or enter said chambers, said chambers having an axial area considerably greater than the area of said ports to provide a linear flow path in said chambers, said fluid when entering or exhausting from said chambers exerting certain flow forces on said movable element whereby said movable element is urged to move in an oscillatory manner, said movable element having relatively larger body portions between some of said lands and relatively smaller body portions between others of said lands, whereby fluid entering or exhausting from certain of said chambers will do so in a relatively axial direction while fluid entering or exhausting from others of said chambers will do so in a relatively radial direction, whereby oscillation damping forces are created to act on said movable element whereby said movable element is stabilized.

2. In a fluid system, a control valve comprising a casing having a bore, a movable valve element slidably retained in said bore, said element being formed with a plurality of body portions and a plurality of lands having a diameter substantially equivalent to that of the bore, said lands being in axial spaced relation between said body portions of said movable element, said body portions and said lands together defining, in cooperation with said casing, a plurality of fluid chambers, each chamber having a fluid port to allow the intake or exhaust of fluid, said movable element being operable to allow fluid to exhaust from certain of said chambers and to enter others of said chambers, said fluid exerting certain flow forces on said movable element by virtue of its movement whereby said movable element is urged to oscillate, said movable element having an enlarged body portion between said lands defining, in co-operation with said casing, said others of said chambers, into which fluid is flowing, and a relatively smaller body portion between said lands defining, in co-operation with said casing, said certain of said chambers from which fluid is exhausted, whereby fluid flowing in said certain of said chambers exhausts in a relatively radial direction while the fluid flowing in said others of said chambers is caused to enter in a relatively axial direction, whereby oscillation damping forces are created which stabilize said movable element.

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