

[54] **METHOD AND APPARATUS FOR CONTROLLING DISPLACEMENT OF A VARIABLE VOLUME PUMP OR MOTOR**

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[58] Field of Search 417/222; 91/505, 91/506, 485; 92/12.2

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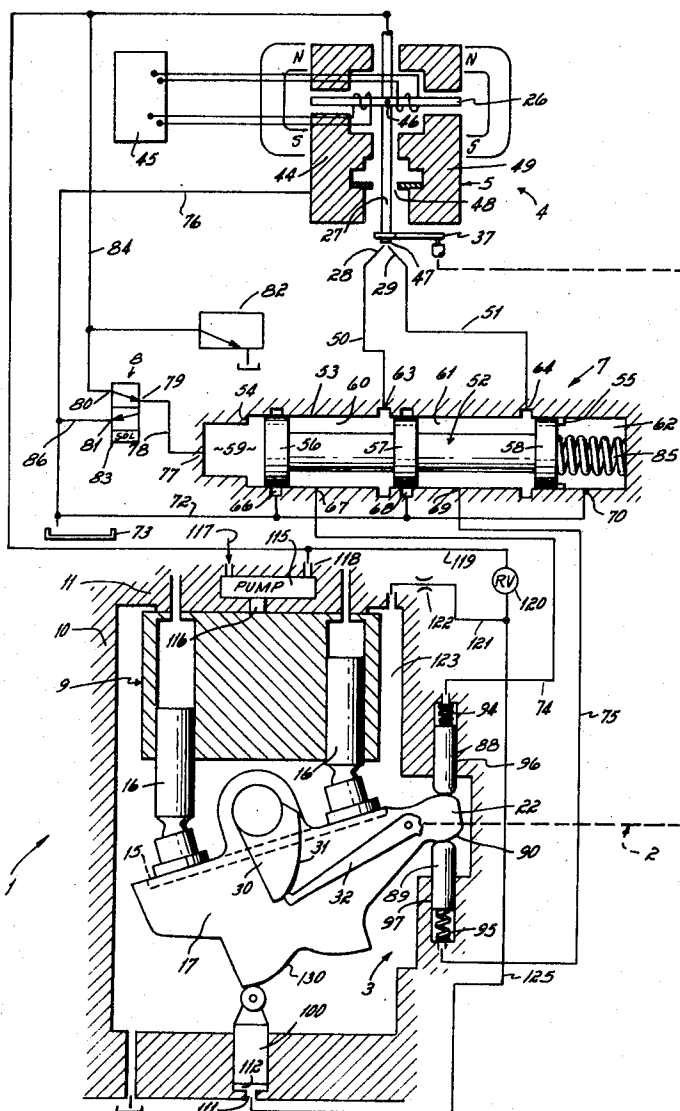
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[57] **ABSTRACT**

The displacement of a servoactuator, such as a variable volume piston pump or motor, is controlled by a servovalve of the jet discharge type having a movable jet tube which is responsive to an electrical input to establish a pressure differential between a pair of fluid lines in accordance with the magnitude of an electrical signal. The pressure differential is applied to operate opposed hydraulic motors which position the displacement changing means or hanger of the pump or motor. Hanger torque applied to the hanger by the inertia of the pump or motor pistons limits the capability of the servovalve to move the hanger under some performance conditions. To offset and neutralize this inertia torque, an auxiliary piston is operative to apply an opposing torque to the hanger of the pump or motor.

36 Claims, 4 Drawing Figures



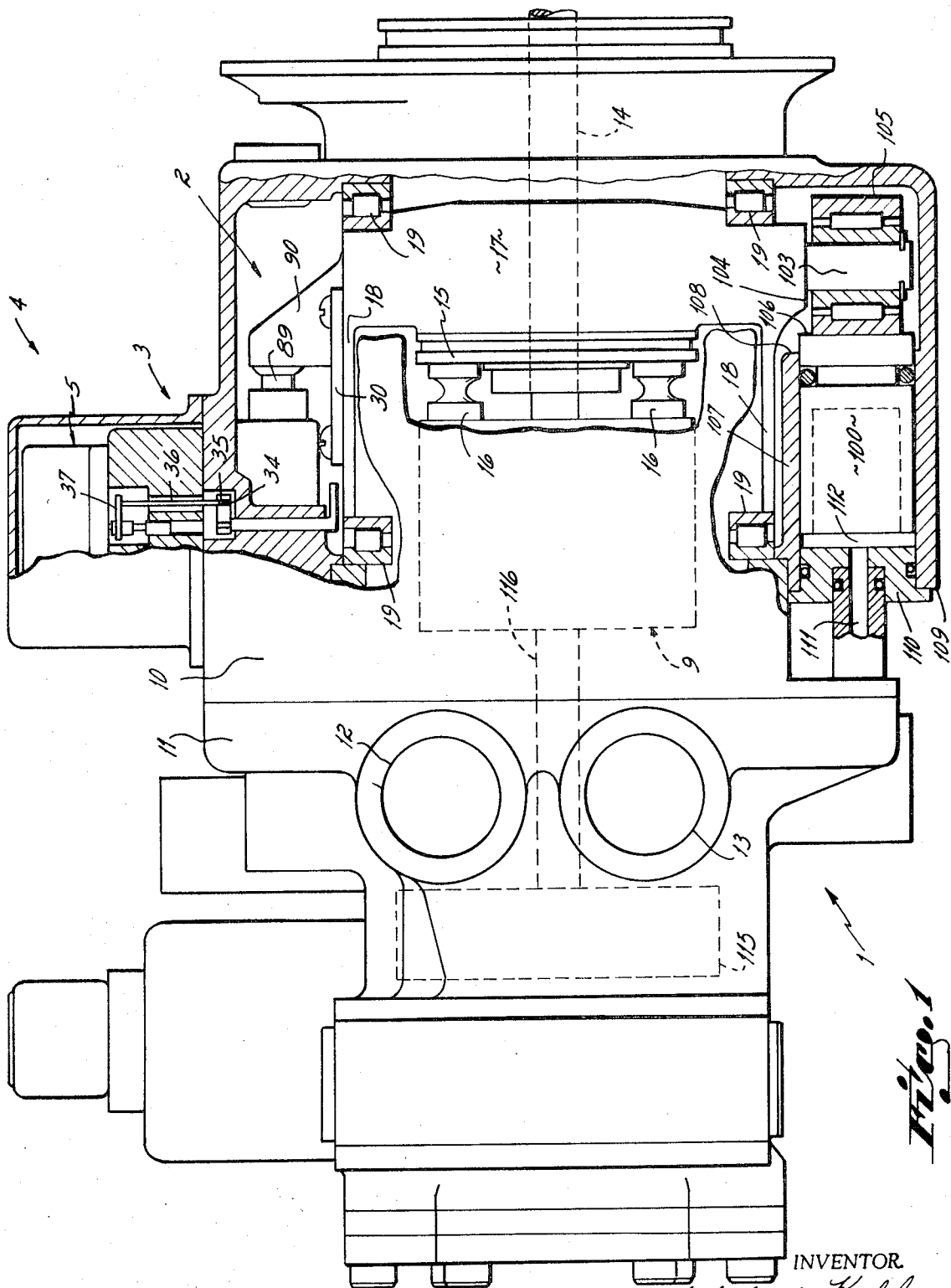
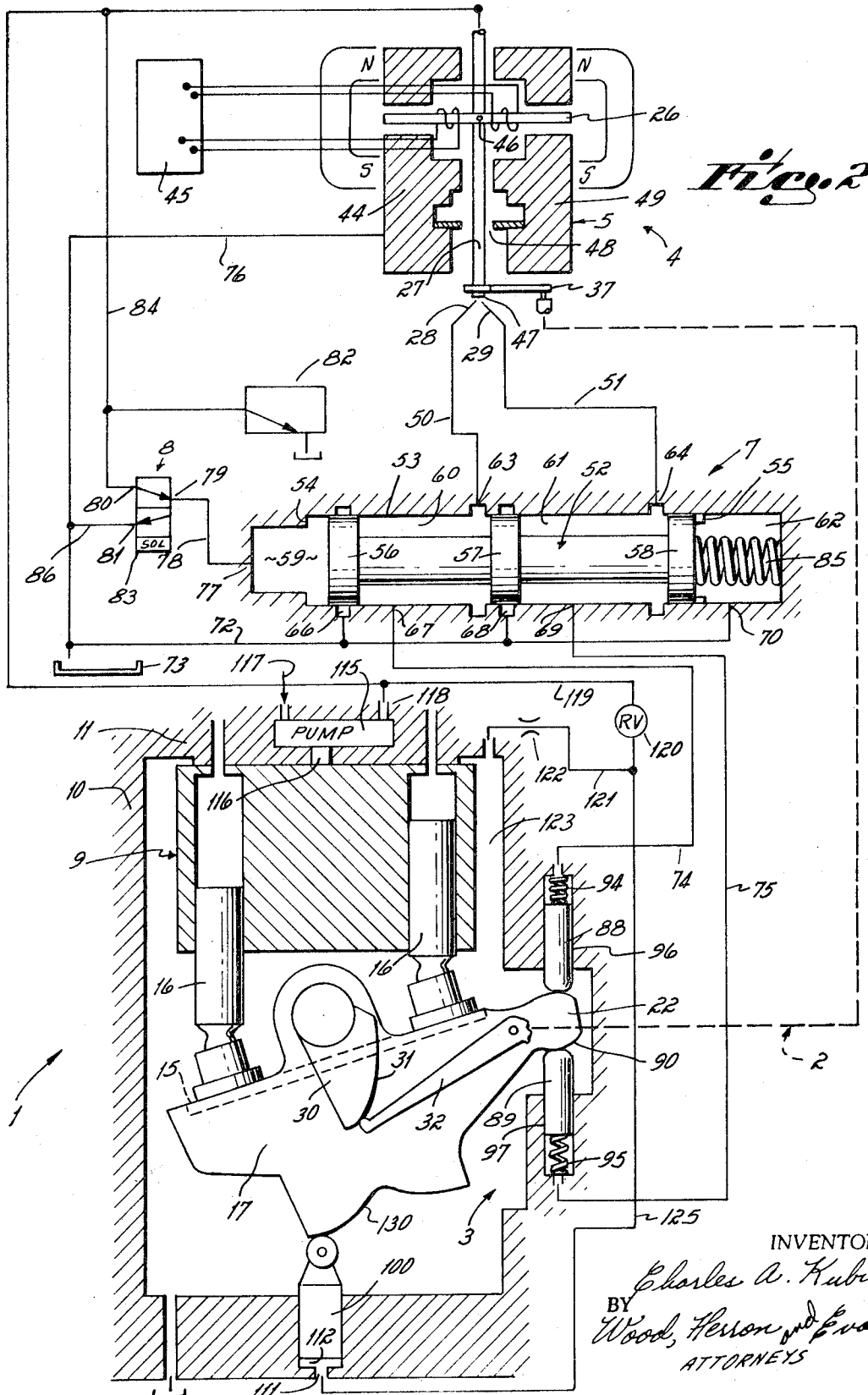


Fig. 1

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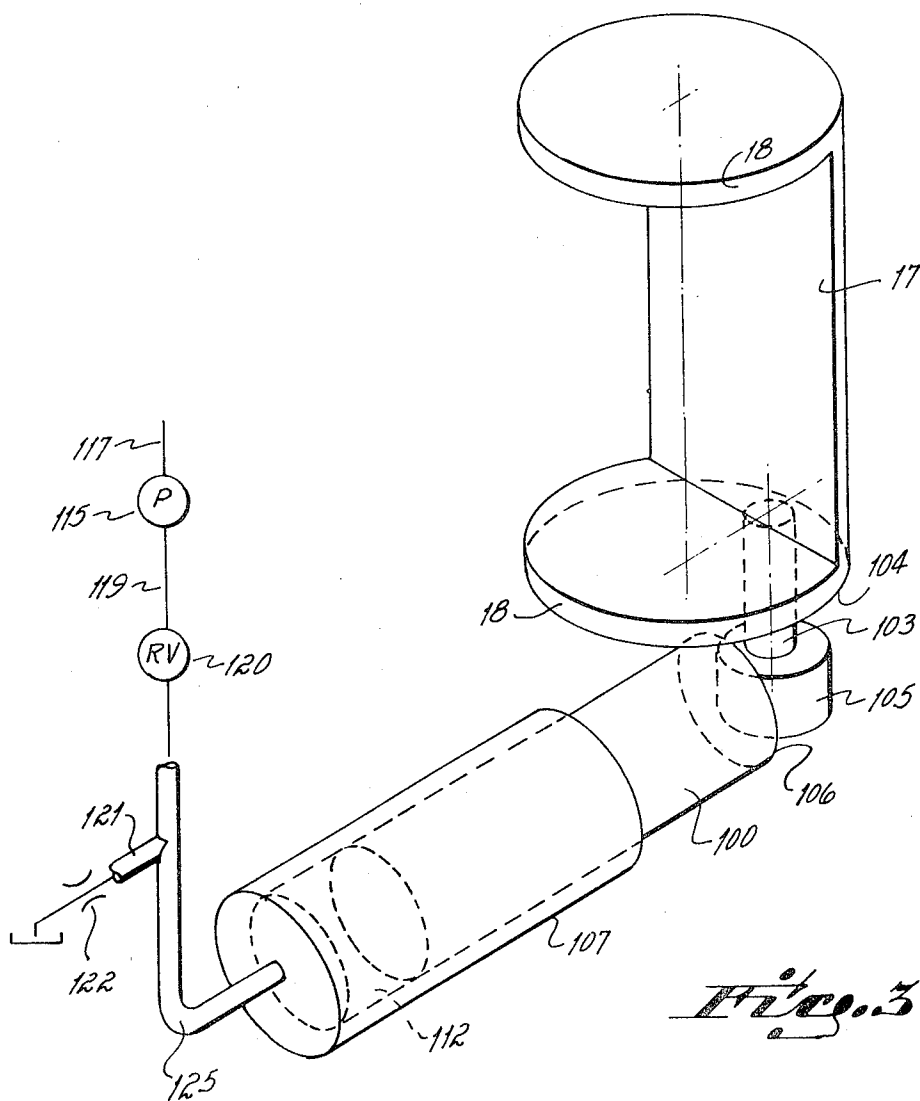


Fig. 3

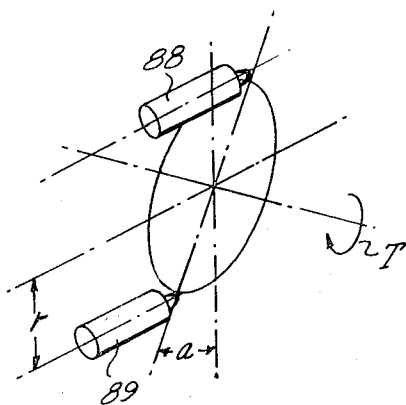


Fig. 4

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METHOD AND APPARATUS FOR CONTROLLING DISPLACEMENT OF A VARIABLE VOLUME PUMP OR MOTOR

BACKGROUND OF THE INVENTION

This invention relates to variable volume piston pumps or motors and, more particularly, to the mechanism for positioning the displacement changing means of a variable volume pump or motor. In the preferred embodiment of the invention, this mechanism for changing the position of the displacement changing means is a servomechanism which is operative to position the displacement changing means in accordance with the magnitude of an electrical command control signal.

The displacement or the volume of fluid pumped through a variable volume piston pump or motor is controlled by the angular position of a piston control plate or so-called "hanger". The angular position of this hanger determines the stroke of the pistons in the pump or motor during reciprocation of the pistons in a barrel which rotates relative to the non-rotating angulated plate or "hanger". When the plate or hanger is normal to the axis of the barrel, the pistons do not reciprocate in the barrel during rotation of the barrel. But, as the plate or hanger is angulated more and more to a greater and greater angle relative to the axis of the barrel, the stroke or the distance that the pistons are reciprocated during each revolution of the barrel is increased with the result that the volume of fluid pumped through the unit increases proportionately.

The angular position of the hanger of variable volume piston pumps may be controlled manually or mechanically. Generally, the angular position is controlled by a pair of opposed pistons acting upon the hanger so as to cause it to pivot or rotate relative to the axis of the piston-containing barrel.

This invention is particularly concerned with the application of torque to the hanger of a variable volume piston pump and motor to effect movement of the hanger. In some applications the force or torque available to effect movement of the hanger is limited and is a determining factor in the availability of the unit to a particular application. For example, many variable volume piston pumps are controlled by servovalves. These servovalves control the movement of the hanger through actuation of appropriate control motors in accordance with the command of a control signal. That signal may be either electrical or hydraulic. The signal in turn generally effects movement of a valve which controls the flow of fluid or power to the hanger positioning motors. If the power available to effect movement of the valve, or the power available to effect movement of the control motor is limited, then the size or volume of the pump available for the particular application is also limited since larger volume pumps require greater forces or torques to effect movement of the hanger.

It has been a primary objective of this invention to provide a new and improved control method and apparatus for reducing the power or torque required to effect movement of a hanger of a variable volume piston pump or motor. Thereby this invention enables smaller motors and valves of less power or torque output to control the movement of the hanger of variable volume piston pumps or motors.

One example of a situation where the available torque to effect movement of a pump or motor hanger is critical and limits the applicability of the pump or motor to a particular application involves aircraft servo controlled variable volume piston pumps or motors. These pumps generally utilize an electrically actuated jet pipe servovalve to control displacement or movement of the hanger control motors. Since the power or torque available from a jet pipe type servovalve, though, is very limited, the jet pipe valve is generally used in combination with a second stage valve to effect position changes of the pump or motor hanger. The second stage servovalve enables a greater flow and pressure to be applied to the piston motors which control the hanger displacement. The addition of the second stage, though, decreases the reliability of the system. In aircraft applications, reliability is, of course, critical; and, therefore, it is desirable, if possible, to eliminate the second stage valve. In many cases, though, that has never been possible because the power available from the jet pipe valve was just simply not sufficient to effect movement with the required response to control displacement of the hanger of the pump or motor.

It has therefore been another objective of this invention to increase the capacity or size variable volume piston pump or motor which may be controlled by a particular size and capacity servovalve by reducing the power required to effect position changes of the volume or displacement control of the pump or motor.

Expressed another way, it has been an objective of this invention to reduce the torque required to effect movement of a hanger or displacement changing means of a variable volume piston pump or motor. If the force or torque required to effect that movement is reduced, then the size, capacity, and complexity of a motor and motor control system required to effect that movement may be accordingly reduced.

In general, this invention is predicated upon the determination that one of the factors which materially effects the torque required to effect displacement of a piston pump or motor hanger is attributable to the inertia of the pistons of the pump or motor. Those pistons apply a torque to the hanger which increases as the angulation of the hanger increases relative to the axis of the pump or motor barrel. Thus, as the angulation of the hanger increases, the torque applied to the hanger as a result of acceleration and inertia of the pistons increases proportionally, further tending to increase the angle of the hanger. This piston applied torque or so-called "inertia" torque may be offset by applying a bucking or opposing torque to the hanger. When an equal and opposite bucking torque or force is applied to the hanger, the total torque required to effect movement of the hanger is reduced.

The torque applied to a hanger by piston inertia increases in direct proportion to the angulation of the hanger and the square of the rotational speed of the pistons or the barrels in which they are contained. This torque may be mathematically computed for a variable volume piston pump according to the formula $T = \frac{1}{2}R^2\omega^2M \tan a$ where R = the bore radius, a = the hanger angle, M = the piston mass, ω = the pump angular speed. In other words, the torque which is applied to the pump hanger of a variable volume piston pump as a result of piston acceleration is directly proportional to the square of the angular rotational speed of

the pistons relative to the hanger. This force always acts as a spring tending to displace and increase the angle of the hanger relative to the rotational axis of the barrel.

One aspect of this invention is therefore predicated upon the concept of applying an inertia or acceleration force opposing torque to the hanger of a piston pump or motor, which force is proportional to the rotational speed of the pump or motor barrel squared.

Another aspect of this invention is predicated upon the concept of attaching an auxiliary fixed displacement pump directly to the barrel of the pump or motor in order to derive a force which is proportional to the square of the rotational speed of the barrel of the pump or motor. This pump then has an output flow which is a function or is proportional to the rotational speed of the barrel. That proportional flow is then routed through a fixed orifice so that the pressure generated by the flow is the square of the flow and thus the square of the rotational speed of the pump barrel. That force or pressure is then used to buck or oppose piston inertia torque on the pump hanger. This is accomplished by applying the pressure derived from the auxiliary pump, after passing it through a fixed orifice, to a piston, which piston acts upon a crank arm secured to the hanger to resist or oppose angular movement of the hanger away from a position in which it is normal to the barrel of the pump. By properly sizing the orifice and by selecting an auxiliary fixed displacement pump of the requisite flow or capacity, the torque applied by the inertia piston to the hanger to oppose or buck the torque applied to the hanger by acceleration or inertia of the pistons may be exactly matched and thereby cancelled. Consequently, the total torque required to effect movement of the hanger throughout all of its angular positions may be thereby reduced.

It is common practice in piston pumps and motors to apply a spring to the hanger so as to bias the hanger to a centered position in which there is no fluid flow through the pump. This is particularly common practice in the case of pressure compensated pumps or motors. One patent which discloses this type of arrangement is Misulis Pat. No. 3,416,452. That patent also discloses an arrangement for utilizing hydraulic pressure on the intake port of the unit when it is acting as a motor to bias the hanger back to a centered or no displacement position. In each instance, though, whether the biasing force is applied by a piston or by a spring, the biasing force is not proportional and bears no relationship to the speed of the rotor or pistons of the unit. It is therefore not proportional to the inertia or acceleration of the pistons and consequently does not compensate for and oppose piston inertia torque on the hanger. Consequently, it does not effect the results which the invention of this application produces, i.e., a reduced input torque requirement to effect movement and response of the displacement control.

The primary advantage of this invention is that it reduces the power required of a control system to effect movement of a hanger or a displacement changing element of a variable volume piston pump or motor. By reducing the power required to effect that movement, the reliability and the frequency response of the control system may be increased.

These and other objects and advantages of this invention will be more readily apparent from the following description of the drawings in which:

FIG. 1 is a side elevational view partially broken away of a pump incorporating the invention of this application.

FIG. 2 is a schematic illustration of a pump hanger control system incorporating the invention of this application.

FIG. 3 is a diagrammatic illustration of the pump hanger and pump hanger inertia piston of FIG. 2.

FIG. 4 is a vector analysis diagram of the torque applied to a hanger of a piston pump or motor.

Referring to the drawings and particularly to FIGS. 1 and 2, the numeral 1 designates a variable volume axial piston pump or motor of the type to which this invention is applicable. The pump 1 is shown fragmentarily and diagrammatically in FIG. 1 and schematically in FIG. 2. It has adjustable displacement changing means 2 which is operated by parallel aligned fluid motor means 3. In the preferred embodiment of the invention, the displacement control 4 is of the jet tube servovalve type. It includes a jet discharge type single stage servovalve 5 which directs pressure fluid from a source of pressure or auxiliary pump to the fluid motor means 3, through a by-pass or shut-off valve 7 controlled by a solenoid operated valve 8. The illustrated embodiment of the invention includes the valves 7 and 8 but the system is operable without these valves so that they may be omitted without impairing the operation of the jet tube servovalve.

More specifically, as shown in FIG. 1, pump 1 has a cylinder barrel 9 located within a housing 10, a port surface 11, inlet and output ports 12 and 13 and an operating shaft 14. The barrel contains a plurality of parallel reciprocable pistons 16. These pistons 16 are constrained to run upon a conventional swash plate 15 carried on a hanger 17 which can be tilted or swung about trunnions 18 journaled in roller bearings 19 in the pump housing 10 to permit the angle or inclination of swash plate 15 with respect to pistons 16 to be varied.

The hanger 17 has an integral arm or extension 22 which is engaged by the fluid motor means or stroke control mechanism 3. The stroke control mechanism 3 operates to set the inclination angle of hanger 17, and hence of swash plate 15, with respect to pistons 16, thereby determining the stroke of these pistons 16 and the rate and direction of fluid flow through pump 1.

Pump 1 is of the cross-center type, in which hanger arm 22 can be swung from one side of the centered, or zero stroke position shown in FIG. 1, to the other side thereof, causing the direction of fluid flow through the pump to be reversed for a given direction of rotation of shaft 14. However, it should be understood that the invention can also be used with single side pumps and motors, which can be adjusted as to magnitude but not direction of flow.

The jet discharge type electrohydraulic servovalve 5 which operates the fluid motor means 3 through the shut-off valve 7, has a rotationally displaceable armature 26 to which a torque of adjustable magnitude can be applied. This armature 26 urges a jet tube 27 selectively toward one or the other or two receiver or receptor ports 28 and 29 from its normal centered or null position between the ports. Jet tube 27 at its lower or wet side end is mechanically connected through a feedback spring with the hanger 17 (FIG. 2). This mechanical connection consists of a cam 30 fixedly secured to the outer or upper end of the hanger 17. This cam has a cam surface 31 engageable by one end of a cam fol-

lower arm 32, the opposite end 33 of which is provided with a socket 34 (FIG. 1) for the reception of a ball 35 secured to the lower end of a force transmitting spring rod 36. The upper end of this spring 36 is fixedly secured to the lower or wet end of the jet tube 27 by a bracket 37. Another spring (not shown) is secured to a section of the case or pump housing so as to bias the follower 32 into engagement with the cam surface 31. This mechanical feedback system is completely described in detail in my copending application Ser. No. 858,604, filed Sept. 17, 1969, and assigned to the assignee of this application.

The jet discharge electrohydraulic servovalve 5 may suitably be of the type described in U.S. Pat. No. 3,017,864 to Atchley. As shown in FIG. 2, the servovalve 5 includes a polarized torque motor (generally at 44) receptive to an electrical current of controlled magnitude from electrical source 45 which may be conventional. The torque motor armature 26 is rotatable about an axis 46 and is connected to jet tube 27 to displace the latter through small excursions relative to the two receiver ports 28 and 29.

The spring rod 36 applies a mechanical feedback force to the jet tube varying with the position of the displacement changing means 2, for restoring the jet tube to a centered position between the two receiver ports 28 and 29 when the hanger has reached a position correlated to the electric current signal from electric source 45.

Jet tube 27 has an inlet to which pressure fluid is supplied from a pressure source or auxiliary pump. At its other end, jet tube 27 terminates in a jet nozzle 47 which passes through an aperture in a plate 48 mounted on the servovalve body 49. The jet nozzle 47 is closely adjacent receptor ports 28 and 29 and, when armature 26 is displaced from the centered position between the receptor ports in response to an electric signal, the jet of fluid issuing from nozzle 47 is divided unequally between ports 28 and 29, thereby creating a difference in the fluid pressures established at those ports.

Receptor ports 28 and 29 are connected by passageways 50 and 51 to the fluid pressure operated valve 7 which is of the spool type. Valve 7 controls the application of fluid from passageways 50 and 51 to fluid rams of the fluid motor means 3. The valve 7 includes a spool 52 slidable in a bore 53 between end stops 54 and 55. Spool 52 has circumferential lands 56, 57 and 58 spaced along it which cooperate with spool bore 53 to define chambers 59, 60, 61, and 62. Bore 53 is provided with axially spaced inlet grooves or ports 63 and 64 from passageways 50 and 51, respectively, and also with axially spaced outlet grooves 66, 67, 68, 69, and 70. Outlet grooves 66, 68 and port 70 are connected through a passageway 72 to a fluid reservoir or tank 73, to which the casing 49 of servovalve 5 is also connected via line 76. Ports 67 and 69 are connected via passageways 74 and 75, respectively, to two opposed equal area fluid rams which comprise the opposite stroke control mechanism 3. Stop 54 is dimensioned so that grooves 63 and 64 are blocked when grooves 66 and 68 are open.

The lands 56, 57, and 58 of spool 52 are positioned to direct pressure fluid from the jet nozzle 47 through the spool chambers 60 and 61 to stroke control mechanism 3 via passageways 74 and 75, or to close off ports

63 and 64 and dump the control pressure fluid to tank 73 via passage 72.

Valve 7 is also provided with an end port 77 in chamber 59, and this port 77 communicates through a passageway 78 with the solenoid operated valve 8. Valve 8 has three ports, indicated at 79, 80 and 81. When its solenoid 83 is energized, valve 8 applies pressure fluid from port 80, to which a pump to be described is connected via line 84 to port 79 and into valve chamber 59. Application of pressure in chamber 59 on the end surface of spool land 56 holds the spool 52 against stop 55 in the position shown in FIG. 2, against the biasing action of a spring 85.

When solenoid 83 is de-energized, the spool of valve 8 is moved to connect port 79 to port 81 which leads to tank 73 via a line 86. In this condition port 80 is blocked against the application of pressure to port 79. This relieves the endwise pressure force on spool 52, permitting spring 85 to urge the spool against the other end stop 54. The pressure in line 84 from the pressure source is held constant by a relief valve 82 which spills excess fluid to tank.

The stroke control mechanism 3 includes a pair of parallel aligned fluid rams or left and right stroking pistons 88 and 89 which engage opposite sides of an extension 90 of the hanger arm 22. Preferably, a pair of compression springs 94, 95 are located between the ends of the piston cylinders 96, 97 and the ends of the pistons 88, 89. These springs 94, 95 are of equal strength and normally hold the arm and thus the attached hanger 17 in a centered or null position.

The pump 1 and servo control system 4 for controlling displacement of the pump are conventional and form no part of the invention of this application. A complete description of both the pump and servo control system may be found in my copending application Ser. No. 858,604, filed Sept. 17, 1969, and assigned to the assignee of this application.

The invention of this application is concerned with a method and apparatus for reducing the total torque required to be applied to the hanger arm 22 by the pistons 88, 89 to effect displacement of the swash plate 15. To that end, the pump 1 includes an inertia piston 100 and a hydraulic network for supplying fluid under pressure to that piston with a force which compensates for and opposes the force applied to the hanger by the inertia of the pistons 16 of the barrel 9.

The pistons 16 act upon the swash plate 15 and attached hanger 17 to apply a torque to that hanger which increases with the angulation of the hanger and with the speed of the barrel 9 of the pump. This torque is referred to as an inertia or acceleration torque because it can be traced to and is attributable to the inertia of the pistons acting upon the swash plate 15. By vector analysis this torque T can be shown to be equal to $\frac{1}{2}R^2\omega^2M \tan a$ where (referring to FIG. 4) R = radius of the bores within which the pistons 16 are reciprocal, a = the hanger angle, M = the piston mass, and ω = the rotational speed of the pump barrel. The inertia piston 100 is so positioned and the pressure applied to it is of such a magnitude as to oppose and preferably exactly balance or cancel out the inertia torque applied to the hanger 17 by the inertia or the acceleration of the pistons acting upon the hanger. The terms "inertia" and "acceleration" are used interchangeably here because the two are inter-related and the one is a function of the other.

In order to apply piston inertia opposing torque to the hanger 17, there is a crank arm 103 secured to and extending from a radial surface 104 of one trunnion 18. A roller bearing 105 is mounted on this crank arm and is engageable with the end surface 106 of the piston 100. This piston 100 is slideable within and extends from one end 108 of a cylinder 107 machined into the case housing 10 of the pump 1. The opposite end 109 of this cylinder is closed by cap 110. Fluid under pressure is supplied to the piston 100 through the cap 110 via a conduit 111 which communicates with the interior chamber 112 of the cylinder 107. Fluid under pressure in the chamber 112 urges the piston 100 toward the roller 105. As may be seen most clearly in FIG. 3, the relative position of the crank arm 103 on the trunnion 18 and of the piston 100 is such that when pressure is supplied to the chamber 112, it urges the piston toward the crank arm so as to force the trunnion into a position in which the hanger is centered or in a zero displacement position. This is the position in which the axis of the swash plate 15 is normal to the axis of the barrel 9 of the pump. As the hanger 17 and the attached trunnion 18 are moved by the pistons 88 and 89 in either direction from this zero displacement position, fluid under pressure in the chamber 112 opposes this movement.

Fluid pressure is supplied to the conduit 111 and consequently to the chamber 112 from a fixed displacement pump 115, the rotor of which is secured to the barrel 9 of the pump 1 by a shaft 116. In practice, this fixed displacement pump 115 is mounted in the port plate 11 and has its pressure or outlet port 118 connected by conduits in port plate 11 to conduit 111. The suction or intake port of this pump 115 is connected by an intake line 117 to tank or a source of fluid.

The pump 115 is a fixed displacement pump and in the preferred embodiment is a conventional gear type pump. It is driven by the shaft 116 and supplies fluid under pressure from its outlet or pressure port 118 through a conduit 119 via a relief valve 120 and a line 125 to the chamber 112. The volume or flow of fluid from this pump 115, since it is a fixed displacement pump, is, of course, proportional to the rotational speed of the barrel 9 to which it is operatively connected.

In order to derive a pressure in the chamber 112 which is proportional to the square of the rotational speed of the barrel 9 and the flow of fluid from the pump 118, the line 119 is connected via a line 121 to the interior 123 of the case or housing 10 of the pump 1. There is a fixed orifice 122 in the line 121 so that the pressure in the line 121, and thus in the line 125, is proportional to the square of the flow in the line 125 since the pressure of a liquid is proportional to the square of the flow of the liquid when the flow is routed through a fixed orifice. The pressure in the line 125 and thus in the chamber 112 is therefore proportional to the square of the flow of the pump 115. Since the flow of the liquid from pump 115 is directly proportional to the rotational speed of the barrel 9, the pressure in the chamber 112 is directly proportional to the square of the rotational speed of the barrel 9 and the pistons 16.

By properly sizing the pump 115 and the orifice 122, it is possible with this construction to exactly balance the torque applied to the hanger 17 by the inertia of the pistons 16 acting upon the hanger. The inertia of the pistons 16 is thereby cancelled or eliminated as a

torque or force upon the hanger which must be overcome by the pistons 88, 89 to effect movement of the hanger. Consequently, a smaller servovalve 4 may be utilized to effect movement of those pistons.

Referring now to FIG. 2, it will be seen that the liquid flow from the auxiliary pump 115 is also utilized as the source of liquid under pressure for the jet pipe 27 of the servovalve. This auxiliary pump thus serves the dual function of supplying fluid to the servovalve 4 and of actuating the inertia piston 100.

Still referring to FIG. 2, It will be seen that in this schematic illustration the inertia piston 100 is illustrated as acting against a cam surface 130. In the preferred embodiment of the invention, the piston 100 acts against a crankarm 103 secured to the trunnion 18 of the pump. It could, though, just as well act against a cam surface secured to the hanger 17.

The primary advantage of an inertia piston to oppose and cancel out torque applied to the hanger of a pump or motor by the inertia or acceleration of the pistons is that it materially reduces the total force or torque required to effectively control hanger displacement. Consequently, smaller valves with less total power output may be utilized to effect that movement and control. The addition of this inertia piston also increases the frequency response of the hanger motors enabling smaller motors with less power output to effect movement of a pump hanger with a particular frequency or speed of response.

While only a single preferred embodiment of the invention has been illustrated and described herein, persons skilled in the art to which this invention pertains will readily appreciate numerous changes and modifications which may be made without departing from the spirit of this invention. Specifically, such persons will appreciate that this invention is applicable to pumps or motors which are manually or mechanically controlled as well as servo controlled pumps. They will also appreciate that any type of servovalve, including jet pipe servovalves, may be utilized to advantage in a pump or motor which incorporates the invention of this application. Additionally, they will appreciate that the valve need not be a servovalve but could as well be a non-feedback type of valve. Therefore, I do not intend to be limited except by the scope of the appended claims.

Having described my invention, I claim:

1. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons,

a jet discharge electrohydraulic valve including a pair of adjacent receptor ports, an armature operated by a torque motor, a jet tube having a nozzle at one end thereof and fixedly mounted at a point separate from said one end, said tube being displaceable by said armature about said point for adjustably dividing a jet stream from said nozzle between said receptor ports, means for supplying pressure fluid to said jet tube,

passageways connecting said receptor ports with the fluid motor means for supplying pressure fluid to said fluid motor means from said receptor ports so as to effect movement of said fluid motor means in response to movement of said jet tube,

feedback means connected to said displacement changing means and to said jet tube for increasingly urging said tube toward a centered position between said receptor ports as said displacement means is moved away from a position in which there is no flow through the device, and the improvement which comprises

means for opposing movement of said displacement changing means away from the position in which there is no flow through said device with a force which increases in direct proportion to the force applied to the displacement changing means by the inertia of the pistons of said device,

said means for opposing movement of said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said pistons upon changes of the rotational speed of said pistons independently of changes of position of said displacement changing means.

2. The variable displacement device of claim 1 in which said means for opposing movement of said displacement changing means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

3. The variable displacement device of claim 2 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

4. The variable displacement device of claim 3 in which said fixed displacement hydraulic pump also supplies fluid under pressure to said jet tube.

5. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly and pivotably movable displacement changing means and including fluid motor means for pivotally moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons,

a jet discharge electrohydraulic valve including a pair of adjacent receptor ports, an armature operated by a torque motor, a jet tube having a nozzle at one end thereof and fixedly mounted at a point separate from said one end, said tube being displaceable by said armature about said point for adjustably dividing a jet stream from said nozzle between said receptor ports,

means for supplying pressure fluid to said jet tube, passageways connecting said receptor ports with the fluid motor means for supplying pressure fluid to

said fluid motor means from said receptor ports so as to effect movement of said fluid motor means in response to movement of said jet tube,

feedback means connected to said displacement changing means and to said jet tube for increasingly urging said tube toward a centered position between said receptor ports as said displacement means is moved away from a position in which there is no flow through the device, and the improvement which comprises

means for opposing movement of said displacement changing means away from the position in which there is no flow through said device with a torque which increases in direct proportion to the torque applied to the displacement changing means by the inertia of the pistons of said device,

said means for opposing movement of said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said pistons upon changes of the rotational speed of said pistons independently of changes of position of said displacement changing means.

6. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and a movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means,

a jet discharge electrohydraulic valve including a pair of adjacent receptor ports, an armature operated by a torque motor, a jet tube having a nozzle at one end thereof and fixedly mounted at a point separate from said one end, said tube being displaceable by said armature about said point for adjustably dividing a jet stream from said nozzle between said receptor ports,

means for supplying pressure fluid to said jet tube, passageways connecting said receptor ports with the fluid motor means for supplying pressure fluid to said fluid motor means from said receptor ports so as to effect movement of said fluid motor means in response to movement of said jet tube,

feedback means connected to said displacement changing means and to said jet tube for increasingly urging said tube toward a centered position between said receptor ports as said displacement means is moved away from a position in which there is no flow through the device, and the improvement which comprises

means for opposing movement of said displacement changing means away from the position in which there is no flow through said device with a force which increases in direct proportion to the movement of said displacement changing means away from the no flow position thereof and in direct proportion to the square of the rotational speed of the barrel.

7. The variable displacement device of claim 6 in which said means for opposing movement of said displacement changing means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

8. The variable displacement device of claim 7 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

9. The variable displacement device of claim 8 in which said fixed displacement hydraulic pump also supplies fluid under pressure to said jet tube.

10. A variable displacement hydraulic energy translating device having reciprocable fluid displacement elements mounted in a rotatable barrel and a movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means, said device being characterized by having a force applied to the displacement changing means by piston inertia which tends to move said displacement changing means away from a position in which there is no flow through the device, which force increases in direct proportion to the square of the rotational speed of the pistons, and the improvement which comprises

means for opposing movement of said displacement changing means away from a position in which there is no flow through the device with a force which increases in direct proportion to the force applied to the displacement changing means by the inertia of the displacement elements of said device, said means for opposing movement of said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said displacement elements upon changes of the rotational speed of said displacement elements independently of changes of position of said displacement changing means.

11. The variable displacement device of claim 10 in which said means for opposing movement of said displacement changing means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said piston and cylinder elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

12. The variable displacement device of claim 11 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

13. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and a movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means, and the improvement which comprises

means for opposing movement of said displacement changing means away from a position in which there is no flow through the device with a force which increases in direct proportion to the movement of said displacement changing means away from the no flow position thereof and in proportion to the square of the rotational speed of said barrel.

14. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly and pivotally movable

displacement changing means and including fluid motor means for pivotally moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons, and the improvement which comprises

means for opposing movement of said displacement changing means with a torque which increases in direct proportion to torque applied to the displacement changing means by the inertia of the pistons of said device,

said means for opposing movement of said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said pistons upon changes of the rotational speed of said pistons independently of changes of position of said displacement changing means.

15. The variable displacement device of claim 14 in which said means for opposing movement of said displacement changing means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

16. The variable displacement device of claim 15 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

17. The variable displacement device of claim 14 in which said torque applying means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

18. The variable displacement device of claim 17 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

19. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons,

a jet discharge electrohydraulic valve including a pair of adjacent receptor ports, an armature operated

by a torque motor, a jet tube having a nozzle at one end thereof and fixedly mounted at a point separate from said one end, said tube being displaceable by said armature about said point for adjustably dividing a jet stream from said nozzle between said receptor ports,

means for supplying pressure fluid to said jet tube, passageways connecting said receptor ports with the fluid motor means for supplying pressure fluid to said fluid motor means from said receptor ports so as to effect movement of said fluid motor means in response to movement of said jet tube,

feedback means connected to said displacement changing means and to said jet tube for increasingly urging said tube toward a centered position between said receptor ports as said displacement means is moved away from a position in which there is no flow through the device, and the improvement which comprises

means for applying a force to said displacement changing means opposed to and in proportion to the force applied to the displacement changing means by the inertia of the pistons of said device, said means for applying a force to said displacement changing means being operable to oppose increased displacement movement of said displacement changing means with a force which remains directly proportional to the square of the rotational speed of said pistons independently of changes of position of said displacement changing means.

20. The variable displacement device of claim 19 in which said force applying means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

21. The variable displacement device of claim 20 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

22. The variable displacement device of claim 21 in which said fixed displacement hydraulic pump also supplies fluid under pressure to said jet tube.

23. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly and pivotably movable displacement changing means and including fluid motor means for pivotally moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons,

a jet discharge electrohydraulic valve including a pair of adjacent receptor ports, an armature operated by a torque motor, a jet tube having a nozzle at one end thereof and fixedly mounted at a point separate from said one end, said tube being displaceable by said armature about said point for adjust-

ably dividing a jet stream from said nozzle between said receptor ports,

means for supplying pressure fluid to said jet tube, passageways connecting said receptor ports with the fluid motor means for supplying pressure fluid to said fluid motor means from said receptor ports so as to effect movement of said fluid motor means in response to movement of said jet tube,

feedback means connected to said displacement changing means and to said jet tube for increasingly urging said tube toward a centered position between said receptor ports as said displacement means is moved away from a position in which there is no flow through the device, and the improvement which comprises

means for applying a torque to said displacement changing means opposed to and in proportion to the torque applied to the displacement changing means by the inertia of the pistons of said device, said means for applying a force to said displacement changing means being operable to oppose increased displacement movement of said displacement changing means with a force which remains directly proportional to the square of the rotational speed of said pistons independently of changes of position of said displacement changing means.

24. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and a movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means,

a jet discharge electrohydraulic valve including a pair of adjacent receptor ports, an armature operated by a torque motor, a jet tube having a nozzle at one end thereof and fixedly mounted at a point separate from said one end, said tube being displaceable by said armature about said point for adjustably dividing a jet stream from said nozzle between said receptor ports,

means for supplying pressure fluid to said jet tube, passageways connecting said receptor ports with the fluid motor means for supplying pressure fluid to said fluid motor means from said receptor ports so as to effect movement of said fluid motor means in response to movement of said jet tube,

feedback means connected to said displacement changing means and to said jet tube for increasingly urging said tube toward a centered position between said receptor ports as said displacement means is moved away from a position in which there is no flow through the device, and the improvement which comprises

means for applying a force to said displacement changing means opposed to and in proportion to the movement of said displacement changing means away from the no flow position thereof and in direct proportion to the square of the rotational speed of the barrel.

25. The variable displacement device of claim 24 in which said force applying means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

26. The variable displacement device of claim 25 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

27. The variable displacement device of claim 26 in which said fixed displacement hydraulic pump also supplies fluid under pressure to said jet tube.

28. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and a movable displacement changing means and including fluid motor means for moving and positioning said displacement changing means, said device being characterized by having a force applied to the displacement changing means by piston inertia which tends to move said displacement changing means away from a position in which there is no flow through the device, which force increases in direct proportion to the square of the rotational speed of the pistons, and the improvement which comprises

means for applying a force to said displacement changing means opposed to and in proportion to the force applied to the displacement changing means by the inertia of the pistons of said device, said means for applying a force to said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said pistons upon changes in the rotational speed of said pistons independently of changes of position of said displacement changing means.

29. The variable displacement device of claim 28 in which said force applying means includes a hydraulic motor having a hydraulic piston element and a hydraulic cylinder element, one of said elements being movable and the other being fixed relative to said displacement changing means, and means for supplying fluid under pressure to said motor at a pressure which increases in direct proportion to the square of the rotational speed of the barrel.

30. The variable displacement device of claim 29 in which said means for supplying fluid to said motor includes a fixed displacement hydraulic pump operatively connected to said barrel of said device so as to supply fluid therefrom at a flow rate directly proportional to the rotational speed of said motor.

31. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly and pivotally movable displacement changing means and including fluid motor means for pivotally moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons, and the improvement which comprises

means for applying a torque to said displacement changing means which opposes and is in proportion to torque applied to the displacement changing means by the inertia of the pistons of said device, said means for applying a force to said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said pistons upon changes in the rotational

speed of said pistons independently of changes of position of said displacement changing means.

32. A variable displacement hydraulic energy translating device having reciprocable pistons mounted in a rotatable barrel and an angularly and pivotally movable displacement changing means and including fluid motor means for pivotally moving and positioning said displacement changing means, said device being characterized by having a torque applied to the displacement changing means by piston inertia which tends to increase angulation of the displacement changing means, which torque increases in direct proportion to the angulation of the displacement changing means and the square of the rotational speed of the pistons, and the improvement which comprises

means for applying a torque to said displacement changing means which opposes and is generally equal to the torque applied to the displacement changing means by the inertia of the pistons of said device,

said means for applying a force to said displacement changing means being operable to maintain said force proportional to the force applied by the inertia of said pistons upon changes in the rotational speed of said pistons independently of changes of position of said displacement changing means.

33. The method of reducing the force required to effect movement of a displacement control element of a variable displacement device of the type which includes reciprocating pistons mounted in a rotatable barrel and a movable displacement changing element for controlling the stroke of the pistons, which device is characterized by having a force applied to the displacement changing element by piston inertia which tends to move said displacement changing element away from a position in which there is no flow through the device, which force increases in direct proportion to the square of the rotational speed of the pistons, which method comprises

the step of applying an opposing force to said displacement changing element opposed to and in direct proportion to the force applied to the displacement changing element by the inertia of the pistons of said device, and

maintaining said opposing force proportional to the inertia of the pistons resulting from changes in rotational speed of said barrel independently of changes of position of said displacement changing element.

34. The method of reducing the torque required to effect pivotal movement of a displacement control element of a variable displacement device of the type which includes reciprocating pistons mounted in a rotatable barrel and an angularly movable displacement changing element for controlling the stroke of the pistons, which device is characterized by having a torque applied to the displacement changing element by piston inertia which tends to increase angulation of the displacement changing element, which torque increases in direct proportion to the angulation of the displacement changing element and the square of the rotational speed of the pistons, which method comprises

the step of applying an opposing torque to said displacement changing element opposed to and in direct proportion to the torque applied to the displacement changing element by the inertia of the pistons of said device, and

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maintaining said opposing torque proportional to the inertia of the pistons resulting from changes in the rotational speed of said barrel independently of changes of position of said displacement changing element.

35. The method of reducing the force required to effect movement of a displacement control element of a variable displacement device of the type which includes reciprocating pistons mounted in a rotatable barrel and a movable displacement changing element for controlling the stroke of the pistons, which device is characterized by having a force applied to the displacement changing element by piston inertia which tends to move said displacement changing element away from a position in which there is no flow through the device, which force increases in direct proportion to the square of the rotational speed of the pistons, which method comprises

the step of applying a force to said displacement changing element opposed to and generally equal to the force applied to the displacement changing element by the inertia of the pistons of said device, and

maintaining said opposing force proportional to the inertia of the pistons resulting from changes in rotational speed of said barrel independently of

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changes of position of said displacement changing element.

36. The method of reducing the torque required to effect pivotal movement of a displacement control element of a variable displacement device of the type which includes reciprocating pistons mounted in a rotatable barrel and an angularly movable displacement changing element for controlling the stroke of the pistons, which device is characterized by having a force applied to the displacement changing element by piston inertia which tends to move said displacement changing element away from a position in which there is no flow through the device, which force increases in direct proportion to the square of the rotational speed of the pistons, which method comprises

the step of applying a torque to said displacement changing element opposed to and generally equal to the torque applied to the displacement changing element by the inertia of the pistons of said device, and

maintaining said opposing torque proportional to the inertia of the pistons resulting from changes in the rotational speed of said barrel independently of changes of position of said displacement changing element.

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