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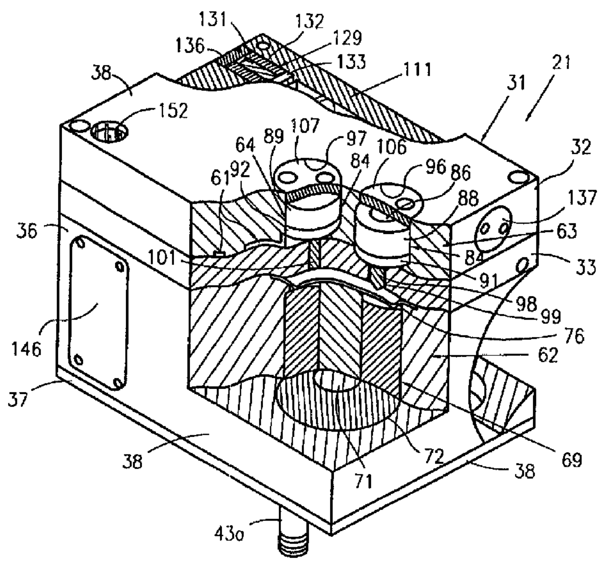
(54) **ACTIONNEUR COMPACT, UNITE DE COMMANDE ET APPAREIL DE POMPAGE AFFERENTS**

(54) **COMPACT ACTUATOR AND CONTROLLER AND PUMPING APPARATUS FOR SAME**



(57) L'invention concerne un actionneur compact (21) destiné à être utilisé avec un fluide, qui comprend un logement (31) présentant une géométrie prédéterminée. Un élément actionneur (41) est monté dans le logement de façon à se déplacer entre la première et la seconde position pour effectuer un certain travail. L'élément actionneur présente une première face (46) et une seconde face (47). Le logement présente un ensemble (61) destiné à fournir un fluide sur la première face de façon à placer l'élément actionneur en première position et sur la seconde face de façon à placer l'élément actionneur en seconde position. L'ensemble comprend

(57) A compact actuator (21) for use with a fluid and including a housing (31) having a predetermined geometry. An actuation member (41) is mounted in the housing for movement between first and second positions for performing work. The actuation member has first and second faces (46, 47). The housing has an assembly (61) for providing fluid to the first face for moving the actuation member to the first position and to the second face for moving the actuation member to the second position. The assembly includes at least one transducer (69, 83 and/or 84) having an active element (71, 86 and/or 87) which is changeable between a first

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au moins un émetteur-récepteur (69, 83 et/ou 84) comportant un élément actif (71, 86 et/ou 87) pouvant passer d'une première forme en l'absence d'un champ électromagnétique à une seconde forme en présence d'un champ électromagnétique. L'invention se rapporte également à une unité de commande (146) et à un appareil de pompage (301) destinés à être utilisés avec l'actionneur.

shape when in the absence of an electromagnetic field and a second shape when in the presence of an electromagnetic field. A controller (146) and pumping apparatus (301) for use with the actuator are provided.

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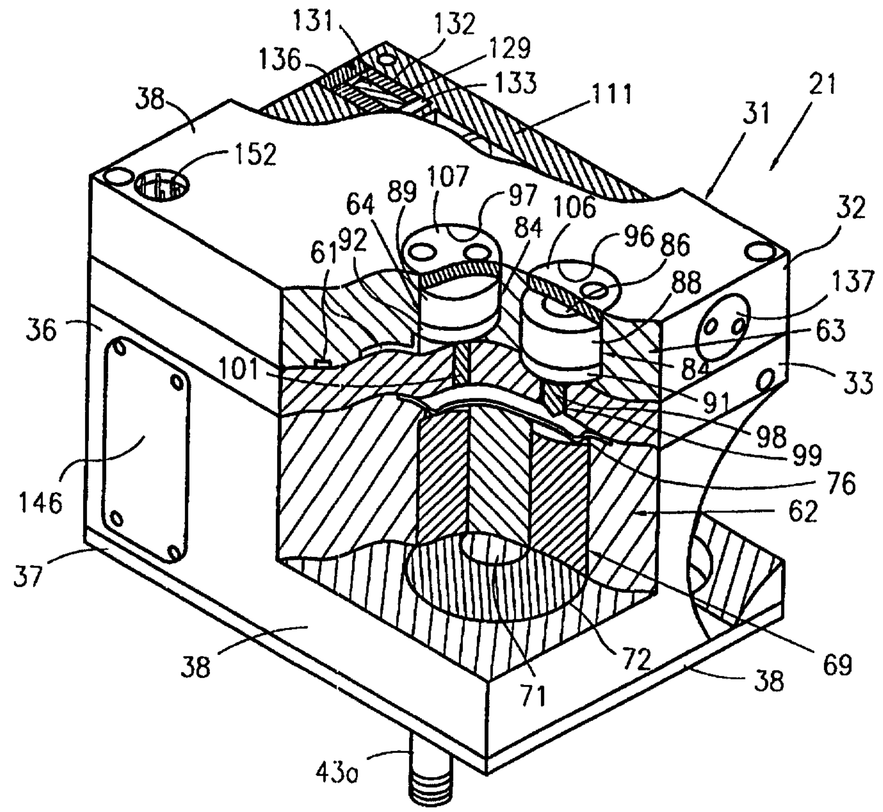
(54) Title: COMPACT ACTUATOR AND CONTROLLER AND PUMPING APPARATUS FOR SAME

(57) Abstract

A compact actuator (21) for use with a fluid and including a housing (31) having a predetermined geometry. An actuation member (41) is mounted in the housing for movement between first and second positions for performing work. The actuation member has first and second faces (46, 47). The housing has an assembly (61) for providing fluid to the first face for moving the actuation member to the first position and to the second face for moving the actuation member to the second position. The assembly includes at least one transducer (69, 83 and/or 84) having an active element (71, 86 and/or 87) which is changeable between a first shape when in the absence of an electromagnetic field and a second shape when in the presence of an electromagnetic field. A controller (146) and pumping apparatus (301) for use with the actuator are provided.



COMPACT ACTUATOR AND CONTROLLER AND PUMPING APPARATUS FOR SAME

This invention pertains generally to fluidic actuators and, more particularly, to compact fluidic actuators.

For many current applications, hydraulic power is still the preferred method of gaining mechanical advantage. On air vehicles, for example, hydraulic power provides the mechanical muscle to operate loads such as primary controls, flaps, slats, landing gear retraction and extension, brakes and steering, thrust reversers and weapons bay doors. In other air vehicles, the power from the engines is used to generate the fluid power. Hydraulic power distribution typically involves routing fluid supply and return lines to connect sources to loads throughout the airframe, often with redundancy requirements to reduce vulnerability. Unfortunately, in many of these applications, this power comes with the penalties of high weight, size, large acoustic signature, cost and maintenance due to the system's distributed nature and redundancy requirements.

Current state-of-the-art solutions that locate the pressure source near the load use an electrical system as a power source. Electrical systems are ubiquitous and flexible and power distribution wiring is relatively small, lightweight and survivable compared with fluid lines. In many of these systems, electrical power is converted to fluid power using a conventional rotating electric motor to drive a pump. These systems, while offering certain advantages, have rotational inertia, bearings to wear out, current inrush, sensitivity to low voltage and weight and bulk. Further, in most cases the unidirectional pump output requires special valving to change output actuation direction.

Other systems utilizing electrical energy as a power source have pumps with magnetostrictive drivers and passive inlet and outlet valves. See, for example, U.S. Patent No. 5,520,522. Unfortunately, these passive mechanical check valves have dynamics that degrade their performance at high frequencies. Such valves also have a single design operating pressure and do not allow the same device to operate at either very low pressure or very high pressure. Nor are these valves shown for use in a fluidic actuator. U.S. Patent No. 5,501,425 separately discloses valves with magnetostrictive drivers, but the emphasis is on fail-safe conditions for two position solenoid valves. The patent does not disclose incorporating such active valves into a pump or a fluidic actuator. As can be seen from the foregoing, there is a need for a new and improved fluidic actuator which overcomes these disadvantages.

In general, it is an object of the present invention to provide a compact actuator which has a self-contained fluid system.

Another object of the invention is to provide an actuator of the above character which is easily scalable for different size applications.

Another object of the invention is to provide an actuator of the above character which provides hydraulic power with little acoustic noise.

Another object of the invention is to provide an actuator of the above character which has relatively few moving parts.

Another object of the invention is to provide an actuator of the above character which uses electricity as a power source.

Another object of the invention is to provide an actuator of the above character which has a relatively small volume of fluid.

Another object of the invention is to provide an actuator of the above character which provides an output of relatively long stroke and high force.

Another object of the invention is to provide an actuator of the above character which utilizes motors having smart material actuation elements made from TERFENOL-D.

Another object of the invention is to provide a controller for operating an actuator of the above character.

Another object of the invention is to provide a pumping apparatus for use with an actuator of the above character which utilizes the momentum of the pump piston to enhance the efficiency of inlet and/or exhaust valves of the pump.

Additional objects and features of the invention will appear from the following description from which the preferred embodiments are set forth in detail in conjunction with the accompanying drawings.

FIG. 1 is a fragmentary cross-sectional view of a portion of a wing assembly utilizing the compact actuator of the present invention.

FIG. 2 is a perspective view of the compact actuator of FIG. 1.

FIG. 3 is a perspective view similar to FIG. 2 of the compact actuator of FIG. 1 which has been cut away at various portions.

FIG. 4 is a cross-sectional view taken along the line 4-4 of FIG. 2 of the compact actuator of FIG. 1.

FIG. 5 is a cross-sectional view taken along the line 5-5 of FIG. 2 of the compact actuator of FIG. 1.

FIG. 6 is a schematic view of the compact actuator of FIG. 1.

FIG. 7 is another schematic view of compact actuator of FIG. 1.

FIG. 8 is a fragmentary view of another embodiment of a normally open flow control valve for use with the compact actuator of FIG. 1.

FIG. 9 is a fragmentary view, similar to FIG. 8, of another embodiment of a normally closed flow control valve for use with the compact actuator of FIG. 1.

FIG. 10 is a schematic view of another embodiment of the compact actuator of the present invention.

FIG. 11 is a flow chart depicting the operation of the compact actuator of FIG. 10.

FIG. 12 is a series of time synchronized graphs showing time histories of sample input currents to the pump and flow control valves of the compact actuator of FIG. 10.

FIG. 13 is a fragmentary view of another embodiment of a pump for use with the compact actuator of FIG. 1.

FIG. 14 is a fragmentary view of a further embodiment of a pump for use with the compact actuator of FIG. 1.

FIG. 15 is a fragmentary view of yet another embodiment of a pump for use with the compact actuator of FIG. 1.

In general, a compact actuator for use with a fluid and including a housing having a predetermined geometry is provided. An actuation member is mounted in the housing for movement between first and second positions for performing work. The actuation member has first and second faces. Means within the housing provides fluid to the first face for moving the actuation member to the first position and to the second face for moving the actuation member to the second position. Said means includes at least one transducer having an active element and means for producing an electromagnetic field which extends through at least a portion of the active element. The active element is changeable between a first shape when in the absence of the electromagnetic field and a second shape when in the presence of the electromagnetic field. A controller and pumping apparatus for use with the actuator are provided.

More particularly, the self-contained actuation apparatus 21 of the present invention can be used with a conventional wing 22 as shown in the fragmentary view of FIG. 1. Wing 22 includes an air foil 23 having a flap 26 at the trailing edge thereof. The segmented flap 26 has a forward portion 26a and an aft portion 26b pivotally coupled to forward portion 26a. A guide pin 27 is connected to the aft portion 26b and rides within a track 28 for moving the aft portion outwardly and downwardly from a first or inline position shown in solid lines in FIG. 1 to a second or deflected position shown in dashed lines in FIG. 1. One or more actuators 21, one of which is shown in FIG. 1, is carried by forward portion 26a for moving aft portion 26b between its inline and deflected positions.

Compact actuation apparatus or solid state smart material actuator 21 has a housing 31 formed with a predetermined geometry. The shape of housing 31 can be tailored to fit in irregularly-shaped envelopes. The housing 31 is made from any suitable material such as hard anodized aluminum or steel and has a weight ranging from 0.1 to 200 pounds and preferably ranging from 20 to 30 pounds and an envelope or displaced volume ranging from 0.5 to 1000 in^3 and preferably ranging from 15 to 25 in^3. The housing 31 shown in FIGS. 2-4 has a weight of less than five pounds. Housing 31 is shaped as a parallelepiped, having a length, width and depth of approximately six, two and two inches, respectively. The segmented housing 31 is formed from a plurality of plate-like members secured together by any suitable fastening means such as bolts and nuts (not shown). These plate members include a top plate 32, a second plate 33, a third plate 36 and a bottom plate 37. The exterior of housing 31 is formed by outer surface 38.

An actuation member in the form of piston or ram 41 is slidably carried by housing 31. Ram 41 is cylindrical in shape and made from any suitable material such as steel or titanium. The ram 41 has an enlarged head 42 formed integral with a rod

43. The rod 43 has a threaded end 43a accessible from the exterior of housing 31 and is shown in FIG. 1 as being connected to aft portion 26b of flap 26. Head 42 has first and second opposite planar faces consisting of upper face or surface 46 and lower face or surface 47. Rod 43 is joined to head 42 at lower surface 47. Third plate 36 of housing 31 is provided with a cylindrical bore or chamber 56 for slidably receiving ram head 42. As such, the cylindrical internal surface of housing 31 forming chamber 56 is diametrically sized slightly larger than head 42. An access bore 57 extends through bottom plate 37 and communicates with chamber 56. Rod 43 slidably extends through access bore 57. Conventional fluid sealing means (not shown) is provided between head 42 and the inner cylindrical surface of third plate 36 for inhibiting the flow of fluid between surfaces 46 and 47 of ram 41. Head 42 thus divides ram chamber 56 into first and second portions 56a and 56b. Similarly, conventional sealing means (not shown) is provided between the outer surface of rod 43 and the inner cylindrical surface forming bore 57 in bottom plate 37. As shown most clearly in FIG. 5, ram 41 is movable between a first or extended position, shown in solid lines in FIG. 5, and a second or retracted position, shown in dashed lines in FIG. 5.

Self-contained fluid means or system 61 is provided within housing 31 and coupled to ram 41 for causing movement of the ram between its first and second positions. Fluid or hydraulic system 61 is shown schematically in FIGS. 6 and 7 and includes at least one transducer for providing liquid to upper surface 46 of ram head 42 for moving ram 41 to the first position and, alternatively, for providing liquid to lower surface 47 for moving ram 41 to the second position. More specifically, hydraulic system 61 includes a positive displacement pump 62, a first control valve 63 at the inlet of the pump 62 and a second control valve 64 at the outlet of the pump 62. Each of these three components has a transducer therein. For example, Pump 62, as shown most clearly in FIGS. 3 and 4 and schematically in FIG. 7, has a transducer or motor 69 which includes a cylindrical actuation element or piston 71. A suitable liquid for use in actuator 21 shown in FIGS. 2-5 is a fire-resistant phosphate ester hydraulic fluid or a petroleum based hydraulic fluid and the actuator 21 operates with a volume of such liquid ranging from 0.05 to 1 liter and preferably ranging from 0.20 to 0.25 liter. Liquid metals can also be used for the fluid or liquid of actuator 21.

Piston or rod 71 is made from a suitable active or smart material which changes shape when energized by being placed in an electromagnetic field. Such materials can include electrostrictive materials, piezoelectric materials and magnetostrictive materials. A preferred electrostrictive material is lead magnesium niobate and its variants and a preferred piezoelectric material is lead zirconate titanate and its variants. Magnetostrictive materials change shape in response to an applied magnetic field. Specifically, rod 71 is changeable between a first or shortened shape when in the absence of a magnetic field and a second or elongated shape when in the presence of a magnetic field. A giant magnetostrictive material is preferred because such a material can tolerate high mechanical stress so as to have a relatively high energy density. High energy density enables more mechanical power output from a given electrical power input and volume of smart material which thus reduces the size and weight of actuator 21. Preferred giant magnetostrictive materials are rare earth materials, rare earth-transition metal materials and compositions having rare earth materials, transition metals and other elements.

Preferred rare earth materials for operating temperatures ranging from 0° to 200° K are rare earth binary alloys such as Tb_xDy_{1-x}, where x ranges from 0 to 1. Other rare earth elements can be added or substituted for either terbium or dysprosium in this base alloy. For example, holmium, erbium or gadolinium can be used in place of either terbium or dysprosium. Other preferred rare earth materials for operating temperatures ranging from 0° to 200° K are body centered cubic intermetallic compounds such as $(Tb_xDy_{1-x})(Zn_yCd_{1-y})$, where x ranges from 0 to 1, y ranges from 0 to 1 and $x + y = 1$. Other rare earth elements, such as holmium, erbium or gadolinium, can be added or substituted for either terbium or dysprosium in these body centered cubic intermetallic compounds.

Preferred rare earth-transition metal materials are rare earth-iron materials such as TERFENOL based alloys. These alloys are suited for operating temperatures ranging from 0° to 700° K. One of these alloys is $TbFe_2$. Particularly preferred rare earth-iron materials for operating in the 0° to 700° K temperature range are disclosed in U.S. Patent Nos. 4,308,474; 4,609,402; 4,770,704; 4,849,034 or 4,818,304, incorporated herein by this reference, and include the material known as TERFENOL-D sold by ETREMA

Products, Inc. of Ames, Iowa. TERFENOL-D is a metal alloy formed from the elements terbium, dysprosium and iron and has the formula of $Tb_xDy_{1-x}Fe_{2-w}$, where x ranges from 0 to 1 and w ranges from 0 to 1. A preferred formula for TERFENOL-D is $Tb_xDy_{1-x}Fe_{1.90-1.95}$, where x ranges from 0.25 to 1.0. A particularly preferred formula for the TERFENOL-D material of rod 71 is $Tb_{0.3}Dy_{0.7}Fe_{1.92}$. Other rare earth materials, such as cerium, praseodymium, neodymium, holmium, erbium or gadolinium, can be added or substituted for terbium or dysprosium for property enhancement purposes. For example, a giant magnetostrictive material having the rare earth materials R^1_{x1}, R^2_{x2}, R^3_{x3} ... R^n_{xn} can be provided where R^1, R^2, R^3 ... R^n constitute rare earth materials and $x1 + x2 + x3 \ldots + xn = 1$. Other transition metals, such as manganese, cobalt or nickel, can be added or substituted for iron as disclosed in U.S. Patent No. 5,110,376, incorporated herein by this reference. Elements which are not transition metals, such as aluminum, can also be added or substituted for iron. For example, a giant magnetostrictive material having the elements T^1_{y1}, T^2_{y2}, T^3_{y3} ... T^n_{yn} can be provided where T^1, T^2, T^3 ...T^n constitute transition metals or elements such as aluminum and $y1 + y2 + y3 \ldots + yn = 2-w$, and w ranges from 0 to 1. Alternatively, an intermetallic compound can be provided having combinations or variations of TERFENOL-D, such as $(Tb_{x1},Dy_{x2},R^3_{x3},R^4_{x4} \ldots R^n_{xn})(Fe_{y1},T^2_{y2},T^3_{y3} \ldots T^n_{yn})_{2-w}$ where $x1 + x2 + x3 \ldots + xn = 1$, $y1 + y2 + y3 \ldots + yn = 2-w$, and w ranges from 0 to 1.

Giant magnetostrictive materials which contract and thus exhibit negative magnetostriction when placed in a magnetic field can be used for the material of rod 71 and be within the scope of the present invention. These negative magnetostrictive materials have formulations similar to the giant magnetostrictive materials described above except that they include the rare earth element samarium. Preferred negative magnetostrictive materials for operating temperatures ranging from 0° to 700° K are SAMFENOL based alloys such as $SmFe_2$. A particularly preferred SAMFENOL based alloy is SAMFENOL-D, which is also disclosed in U.S. Patent Nos. 4,308,474; 4,609,402; 4,770,704; 4,849,034 or 4,818,304 and has the formula $Sm_xDy_{1-x}Fe_{2-w}$, where x ranges from 0 to 1 and w ranges from 0 to 1. Other rare earth materials, such as cerium, praseodymium, neodymium, holmium, erbium or gadolinium, can be added or substituted for samarium or dysprosium in the same manner as discussed above with respect to TERFENOL based alloys. In addition, other transition metals, such as manganese, cobalt or nickel, and elements which are not transition metals, such as aluminum, can be added or substituted for iron in the same manner as also discussed above.

Means which includes coil 72 is provided in motor 69 for producing an electromagnetic field which extends through at least a portion of drive member or rod 71. Coil 72 is made from any suitable material such as copper or aluminum and is concentrically disposed about rod 71 for producing a magnetic field having a flux which extends through the rod 71. Coil 72 is mounted within third plate 36, as shown in FIG. 4. Rod 71 has an end 71a which extends from one end of coil 72 to abut the center of a diaphragm 76 made from any suitable material such as stainless steel. The other end of rod 71 is secured to the inside of coil 72. Diaphragm 76 extends across an opening in the form of cylindrical bore 77 provided in third plate 36 and is generally cup-like in shape. In this regard, diaphragm 76 extends upwardly from coil 72 at its outer periphery to sealably secure to the third plate 36. A plate member or disk 81 made from any suitable material such as a noncorrosive stainless steel alloy overlies diaphragm 76 and is sealably secured at its outer periphery to the diaphragm 76. In this manner, disk 81 and diaphragm 76 form a pumping chamber 82 therebetween.

Inlet and outlet check valves 63 and 64 respectively control the flow of liquid into and out of pumping chamber 82 in a single direction and are included within the means of actuator 21 for selectively directing pressurized liquid between pump 62 and ram 41. As shown in FIGS. 3 and 4 and schematically in FIG. 7, valves 63 and 64 have respective motors 83 and 84 which include respective actuation elements in the form of rods or pistons 86 and 87 made from any suitable material such as the materials discussed above with respect to rod 71. Preferably, pistons or drive members 86 and 87 are each made from a magnetostrictive material and more preferably from a giant magnetostrictive material such as TERFENOL-D. Coils 88 and 89, each substantially similar to coil 72, are included in motors 83 and 84 and are concentrically mounted about respective pistons 86 and 87. Coils 88 and 89 serve to energize pistons 86 and 87 by producing a magnetic field which extends through the pistons 86 and 87. First and second amplifiers 91 and 92 are included within valves 63 and 64 for amplifying the stroke of respective pistons 86 and 87 of motors 83 and 84. Stroke amplifiers 91 and 92, not cross sectioned in the drawings, can each be of any conventional type such as a hydraulic stroke amplifier and preferably a Bernoulli-type stroke amplifier. Amplifiers 91 and 92 have respective output rods 93 and 94 slidably carried therein and movable longitudinally as a function of the longitudinal movement of respective pistons 86 and 87. Inlet and outlet valves 63 and 64 and amplifiers 91 and 92 are disposed in first and second spaced-apart bores 96 and 97 extending downwardly into top plate 32 of housing 31.

Valves 63 and 64 each utilize a poppet valve configuration. Specifically, a cylindrical member or valve rod 98 is secured to the end of amplifier rod 93. The lower portion of valve rod 98 is formed as a valve head 99. A cylindrical member or valve rod 101 is similarly secured to the end of amplifier rod 94 of outlet valve 64. Valve rod 101 has a lower portion formed as a valve head 102. Rods 98 and 101 are each made from any suitable material such as a noncorrosive stainless steel alloy. Valve heads or plugs 99 and 102 seat respectively in spaced-apart openings or orifices 103 and 104 provided in disk 81. The orifices 103 and 104 communicate with pumping chamber 82. Caps 106 and 107 are provided for retaining inlet and outlet control valves 63 and 64 within top plate 32.

The elastic modulus of the smart material and the bulk modulus of the fluid within system 61 determine the stiffness matching geometry of pumping chamber 82 and the smart material driver or piston 71. The maximum fluid power point (pressure and flow) is optimized for a given electrical input, size, weight, thermal environment and/or startup transient for actuator 21. Plugs 99 and 102 and orifices 103 and 104 are shaped and contoured to optimize fluid flow characterization and thus maximize the performance of actuator 21. In this regard, the maximized fluid power is matched with ram 41 so that the minimum and maximum physical geometries of plugs 99 and 102 and orifices 103 and 104, as well as the minimum and maximum physical geometries of pumping chamber 82 and piston 71, can be determined. Additional design considerations are the output pressure and flow and frequency response over the desired temperature range.

Hydraulic system 61 further includes a spool valve or servo valve 111 which serves as means for directing the flow of liquid between inlet and outlet control valves 63 and 64 and the ram 41. Servo valve 111 is shown in FIG. 5 and schematically in FIG. 7. An inlet line 112 connects inlet control valve 63 to the servo valve 111 and an outlet line 113 connects outlet control valve 64 to the servo valve 11. First and second lines 116 and 117 connect the servo valve 111 to respective first and second portions 56a and 56b of ram chamber 56. Servo valve 111 has a central chamber 118 provided in top plate 32 which communicates with each of lines 112 and 113 and lines 116 and 117. Chamber 118 is generally cylindrical in shape. An elongate member or rod 121 made from any suitable materials such as steel or aluminum extends through chamber 118. Rod 121 has first and second spaced-apart spools 122 and 123 mounted thereon. Rod 121 is slidably disposed on first and second bearing assemblies 126 and 127 carried by top plate 32 at opposite ends of internal chamber 118. Rod 121 is longitudinally movable within chamber 118 from a first position shown in solid lines in FIG. 5 and in dashed lines in FIG. 7, in which liquid is directed from line 113 to first line 116, and a second position shown in solid lines in FIG. 7, in which liquid is directed from line 113 to second line 117. When rod 121 is in its first position, liquid returning from ram chamber second portion 56b by means of second line 117 is directed to line 112. When rod 121 is in its second position, liquid returning from ram chamber first portion 56a by means of first line 116 is directed to the line 112. For simplicity, not all of the hydraulic lines of fluid system 61 are shown in FIGS. 2-5. The hydraulic lines are milled into housing 31.

Servo valve 111 includes means in the form of a transducer or motor 129 for driving rod 121 between its first and second longitudinal positions within chamber 118. Motor 129 includes a cylindrical rod-like element in the form of actuation element 131 made from any suitable material such as the materials discussed above with respect to rod 71. Preferably, actuation element 131 is made from a magnetostrictive material and more preferably from a giant magnetostrictive material such as TERFENOL-D. A coil 132 substantially similar to coil 72 is concentrically carried or disposed about actuation element 131. An amplifier 133 is included within servo valve 111 for amplifying the stroke of actuation element 131 of motor 129. Stroke amplifier 133, not cross sectioned in the drawings, is of a conventional type and substantially similar to stroke amplifiers 91 and 92. The amplifier has an output rod 134 slidably carried therein and movable longitudinally as a function of the longitudinal movement of actuation element 131. The output rod 134 has an end secured to one end of rod 121. Caps 136 and 137 are provided for respectively retaining rod 121 and motor 129 within top plate 32.

Fluid system 61 in compact actuator 21 further includes a conventional high pressure accumulator 138 and a conventional low pressure reservoir 139, as shown schematically in FIGS. 6 and 7. Both high pressure accumulator 138 and low pressure reservoir 139 are similar in configuration and have a temperature compensating variable volume chamber 141 which is separated from a fluid storage chamber 142 by a flexible membrane 143. Chamber 141 is filled with pressurized gas for accommodating pressure changes due to such factors as piston motion or thermal changes. Low pressure liquid in fluid system 61 is routed to low pressure reservoir 139 where it is ready for recycling through the fluid system. The elements of each of high pressure accumulator 138 and low pressure reservoir 139 are carried inside third plate 36 of housing 31, as shown in FIG. 5 with respect to low pressure reservoir 139.

Electrical means in the form of control electronics or controller 146 is electrically coupled to coils 72, 88-89 and 132. The multiple-input, multiple-output controller is built from conventional off-the-shelf parts. As shown most clearly in FIGS. 6 and 7, electrical lead 147 couples controller 146 to pump motor 69 and leads 148 and 149 respectively couple the controller to inlet and outlet control valve motors 83 and 84. The controller 146 is electrically coupled to motor 129 of servo valve 111 by means of lead 151. Controller 146 is provided with a power and signal connector 152 accessible at the exterior of housing 31 for permitting an external power supply 153 to be electrically coupled to controller 146 and for permitting commanded signals to be provided to the controller and other electrical communications therewith. The controller 146 contains power conditioning electronics for directing electrical energy to the coils and thus causing each of the coils to generate an electromagnetic and specifically a magnetic field around the coil. The magnetic fields actuate and thus control the movement of the respective actuation elements.

Sensing or measuring means is carried by housing 31 for determining the external axial force being exerted on threaded end 43a of ram 41 and the position of ram 41 within chamber 56. In this regard, as shown in FIG. 6, a first pressure transducer or sensor 161 communicates with first line 116 and a second pressure transducer or sensor 162 communicates with second line 117 for determining the liquid pressure in such lines and thus the pressure in first and second chamber portions 56a and 56b on each side of ram head 42. The differential pressure across ram head 42 can thus be determined by controller 146. Conventional pressure sensors 161 and 162 can be of the type manufactured by Kulite Tungsten Corporation of East Rutherford, New Jersey. A position sensor in the form of a linear variable differential transformer or stroke transducer 163 is carried by housing 31 and coupled to ram 41 for determining the position of ram 41 and its threaded end 43a relative to housing 31. Conventional stroke transducer 163 can be of the type manufactured by Trans-Tek, Inc. of Ellington, Connecticut. Ram 41 includes a force sensor in the form of a load cell or force transducer 164 disposed in the load path between head 42 and rod 43 for determining the force upon ram 41. Conventional force transducer 164 can be of the type manufactured by Sensotec, Inc. of Columbus, Ohio. Transducers 161-163 and load cell 164 are electrically coupled to controller 146 by means of electrical leads 166. Pressure sensing or measuring means in the form of conventional pressure sensor 167 is carried by housing 31 and communicates with fluid storage chamber 142 of high pressure accumulator 138. Pressure sensor 167 is electrically connected to controller 146 by electrical lead 168 for permitting the pressure within accumulator 138 to be monitored by the controller 146.

In operation and use, compact hydraulic actuator 21 can be used for moving or pivoting aft portion 26b of flap 26 relative to forward portion 26a of the flap and thus change the camber of wing 22. For initiating movement of ram 41 and thus flap aft portion 26b, the desired command signal is sent to controller 146 which in turn supplies electrical energy to pump 62, inlet and outlet control valve 63, 64 and servo valve 111 at the appropriate durations and levels to cause these components to provide liquid to either upper surface 46 or lower surface 47 of ram head 42 and thus move ram 41 within chamber 56.

Controller 146 of actuator 21 can be electrically coupled by means of connector 152 to any suitable computer such as a personal computer (not shown). Actuator 21 can be electronically configured by the personal computer to function in a prescribed manner for the given application. In this regard, the length of movement or displacement of ram 41, the speed or velocity at which the ram 41 is moved and the force that is available at threaded end 43a of ram 41 can be set. Other parameters of actuator 21 which can be so set include: the characterization of the ram 41 in relation to the input control signal to actuator 21, the response time or acceleration of ram 41, the damping ratio of ram 41, the accuracy of the desired position of ram 41, and the type of control strategy (for example, phase-locked-loop or frequency-locked-loop) the actuator 21 will execute and the functional parameters of the control strategy. Additional actuator parameters which can be set include the communications parameters (for example, peer-to-peer, supervisory, report by exception, networked or stand alone), the alarm and shutdown parameters such as operating temperature limits on the internal diagnostics of actuator 21 and the database parameters, alarm and event logs and user security passwords of the actuator 21.

Actuator 21 can be programmed with a variety of failure modes for output ram 41 in the event of loss of electrical power to the actuator. In a "lock-in-last position" failure mode, ram 41 is locked in the position along its stroke length that it was in at the time of loss of electrical power or control signal to actuator 21. This is accomplished by closing both flow control valves 63 and 64 to prevent flow from occurring between the ram 41 and the pump 62. Ram 41 is allowed to move freely to any position along its stroke length upon loss of electrical power or control signal to actuator 21 in a "move-to position" failure mode. This is accomplished by opening both of valves 63 and 64 to enable flow to occur between the ram 41 and pump 62. If a "fail-retracted" failure mode is selected, ram 41 is forced to a zero (minimum) displacement position along its stroke length by opening both of valves 63 and 64 to permit flow between the ram 41 and pump 62 and by providing a mechanical spring force (not shown) to push the ram to its fully retracted position. In a "fail-extended" failure mode, ram 41 is forced to its full (maximum) displacement position along its stroke length. This is accomplished by opening both of valves 63 and 64 and by providing a mechanical spring force (not shown) to push the ram to its fully extended position.

The operation of pump 62 and valves 63, 64 and 111 is substantially similar in that these components are each activated by an electrical signal from controller 146. Controller 146 operates at the fixed frequency of a standard power supply, for example 400 Hz for an aircraft. Pump 62 and valves 63, 64 and 111 are each driven by an actuation member made from TERFENOL-D. These magnetostrictive actuation elements are changeable between a first condition in the absence of a magnetic field and a second elongated or extended condition in the presence of a magnetic field for performing work in said components. The amount of elongation of an actuation element is proportional to the amplitude of the electrical current signal to actuation coil 72, 88-89 or 132 and the resulting strength of the magnetic field generated by the coil. The frequency of elongation corresponds to the frequency of the electrical signal to the coil. For example, the energizing of coil 72 in pump 62 causes rod 71 to elongate and rod end 71a to move diaphragm 76 toward disk 81 and thus pressurize the liquid within pumping chamber 82. The cessation of electrical energy to coil 72 causes rod 71 to contract and thus return diaphragm 76 to its home position shown in FIG. 4. Likewise, normally open inlet and outlet valve pistons 86 and 87 extend when respective coils 88 and 89 are energized by energy supplied to them by controller 146. Pistons 86 and 87 cause respective valve heads 99 and 102 to close and seat within respective valve seat orifices 103 and 104 as shown in FIG. 4. Upon the cessation of electrical energy to coil 88 or 89, the corresponding valve head 99 or 102 unseats from the respective orifice 103 or 104 to cause the control valve to open.

The drive current waveform provided to pump 62 and valves 63 and 64 is controlled to provide the desired pressure in hydraulic accumulator 138 and thus fluid system 61. The pump waveform is obtained by modulating the desired direct current amplitude signal with a fixed frequency sinusoid, producing a variable amplitude, fixed frequency waveform. Increased performance in the terms of pumping efficiency is obtained by driving pump 62 with an electrical signal having a frequency equal to the mechanical resonant frequency of the pump 62. The resonant input signal to pump 62 amplifies the stroke of the pump to provide relatively large stroke and pumping capacity. Thus, pump 62 provides relatively more stroke for less energy than conventional pumps.

The mechanical resonant frequency of pump 62 is determined by the size and weight of the magnetostrictive piston 71, the amplitude of the drive signal to piston 71, the prestressing and other mechanical loading of the piston 71. For a fixed pump design, the optimal resonant frequency is a function of both the power level and the pump load characteristics, which can change during operation. In the simplest implementation, the drive frequency is fixed, thus requiring a tradeoff to determine the single frequency that provides the best efficiency given the operation power levels and pump load characteristics. Once the drive frequency is chosen and fixed, the output of pump 62 is controlled by the amplitude of the drive current to piston 71.

Pump 62 appears to the drive electronics or circuit as an inductor and a resistor in series. Hence, the drive electronics within controller 146 must supply both real and reactive power to pump motor 69. The efficiency of the drive electronics can be increased if designed to advantageously use the inductive features of the magnetrostrictive motor 69. In general, the drive electronics consist of a sinusoidal voltage source in series with a capacitor. The value of the capacitor is chosen to provide a resonant circuit with the inductance of the magnetostrictive motor 69 when operated at the mechanical resonant frequency of the pump 62. That is, the capacitor is chosen to produce in conjunction with the inductance of the magnetostrictive motor 69 an electronic resonance at the desired drive frequency. In this case, the voltage source need only provide real power, and not reactive power, to pump 62 so as to reduce the cost and increase the efficiency of the drive electronics.

Alternate implementations of the drive electronics can use switched mode electronics. In these cases, the drive electronics can use the inductive load of pump motor 69 for power smoothing. Increased performance and decreased cost can be obtained by designing these switched-mode drive electronics for the specific load characteristics of pump motor 69 and the chosen drive frequency.

Pump 62 can operate at one frequency independent of the fluid demands within actuator 21. As can be appreciated, for a fixed surface area of piston head 42, the flow through pump 62 is determined by the frequency and stroke length of the piston head. When the frequency of the input signal to pump motor 69 is fixed, the flow through the pump is thus determined by the stroke length of the piston head 42. The stroke of pump 62 is a function of the magnetic field strength applied by drive coil 72, which is determined by the current in the coil 72. This current is a function of the voltage across the drive coil 72 since the impedance of drive coil 72 is generally constant at a fixed frequency. In this manner, the power to pump 62 can be switched on and off at the fixed frequency between zero volts and the control voltage to produce the reciprocating stroke of the pump. As can be seen, the foregoing simplifies power control to ON/OFF and flow control to voltage control.

Inlet and outlet control valves 63 and 64 actively control the flow of liquid from pump 62 to accumulator 138 and from the reservoir 139 to the pump 62. Valves 63 and 64 are driven by a simple on/off current waveform which has a frequency substantially equal to the frequency of the electrical signal driving pump 62, but at a phase angle shifted from the frequency supplied to the pump 62. In this manner, pump 62 and flow control valves 63 and 64 are coordinated so as to work in concert for creating the desired hydraulic pressure within hydraulic system 61. More specifically, when flow control valves 63 and 64 are closed and the elongation of the smart material of pump piston 71 begins, the stroke of the piston 71 is reduced by its compliance as compression stress increases and the resulting pump stroke is matched by hydraulic fluid compression. Once the pressurization portion of the total pump stroke is complete, outlet flow control valve 64 is directed by controller 146 to open and the remaining stroke length is used to pump out the pressurized liquid. At the end of this pump stroke, valve 64 is directed to close before piston 71 begins to retract. Since the entire pumping chamber 82 cannot be evacuated at the end of the output stroke, the amount of remaining liquid within chamber 82 expands once piston 71 retracts. After chamber 82 is depressurized, inlet flow control valve 63 opens to admit a new liquid charge. The amplitude of the electrical signal to coils 88 and 89 can be equal to or different than the amplitude of the actuation signal to pump coil 72.

The waveform and timing of the electrical signals driving flow control valves 63 and 64 enables the valves to open and close at the precise times corresponding to the optimized pressure and flow needed to meet the output requirements of compact actuator 21. The ability to precisely control the opening and closure of valves 63 and 64 also permits actuator 21 to be operated at the resonant frequency of pump 62. In addition, controller 146 can synchronously open and close the flow control valves 63 and 64 with the motion of piston 71 so that the valves open and close with near zero pressure differential across them. Active control valves 63 and 64 thus contribute to maximizing the efficiency of pump 62 and actuator 21.

The waveform shaping of active flow control valves 63 and 64 permits the valves to open fast and close softly so as to reduce valve seat wear. This is particularly important at higher frequencies. The TERFENOL-D composition of valve pistons 86 and 87 permits relatively large displacements of the pistons over a wide frequency range. Valves 63 and 64 open and close at precise times corresponding to peak pump pressures, independent of the system pressure. This precise electrical control of each of the valves 63 and 64 enables suppression of pump ripple, present in most hydraulic systems having passive valves therein, thus removing the need for large ripple suppression equipment and integration with load characteristics to suppress water hammer.

With each cycle of pump 62, a small volume of liquid is pumped through outlet control valve 64 to servo valve 111. The servo valve 111 alternatively directs the pressurized liquid from outlet control valve 64 to the desired surface 46 or 47 of ram 41 so as to cause the ram to move in the desired direction relative to housing 31 and/or to change the relative pressure across head 42 and thus the external force being applied by ram 41. Controller 146 regulates servo valve 111 by means of motor 129. Unlike the oscillating or pulsed signal provided by the controller 146 to pump 62 and flow control valves 63 and 64, a constant current is provided by controller 146 to motor 129 so as to cause actuation element 131 to elongate and move rod 121 to the position shown in solid lines in FIG. 7. In the absence of such signal, rod 121 is in the position shown in FIG. 5. As can be appreciated, ram 41 is moved to its extended position shown in FIG. 5 when pressurized liquid is directed by servo valve 111 to first line 116. Ram 41 is retracted to its position shown in FIG. 7 when pressurized liquid is directed by servo valve 111 to second line 117. Ram 41 can be stopped at any time between its extreme positions within chamber 56 by stopping the supply of electrical energy to pump 62 and closing one or both of flow control valves 63 and 64.

Differential pressure or displacement signals from transducers 161-163 provide an inner feedback control loop which allows for a wide range of control schemes required by the multiple applications of compact hydraulic actuator 21. Stroke transducer 163 provides a feedback signal to controller 146 indicating the position of ram 41 within chamber 56. Pressure transducers 161 and 162 provide a feedback signal to the controller 146 indicating the external force being applied on the ram 41. These feedback signals are incorporated into the control algorithm of the controller to modify, if necessary, the signals provided by the controller to pump 62, flow control valves 63 and 64 and servo valve 111.

The compact hydraulic actuator 21 of the present invention is a modular, self-contained linear motion device capable of producing dynamic output strokes similar to those of conventional hydraulic actuators yet at higher frequency and at significantly reduced weight, complexity and volume. All of the components of unitized actuator 21 are located inside hermetically sealed housing 31 so as to be protected from moisture and contamination. The integrated package of actuator 21 ensures dynamic stability of the components thereof and eliminates the need for multiple attachment points. In contrast to conventional hydraulic approaches, actuator 21 utilizes a high authority smart material such as TERFENOL-D. The smart material motors for pump 62 and valves 63, 64 and 111 have no moving parts such as linkages, cams and the like. As such, actuator 21 offers the advantages of reduced cost, longer life, improved maintainability and increased reliability. The compactness and integration of actuator 21 facilitates its removal and replacement from wing 22 thus reducing installation and maintenance costs.

Actuator 21 utilizes actuation members made from a magnetostrictive material such as TERFENOL-D in pump 62, flow control valves 63 and 64 and servo valve 111. The actuator 21 converts the mechanical output of these smart material actuation members, that is, high force, short stroke and high frequency, into a more suitable output, that is, high force, long stroke and low frequency, by locally generating and using hydraulic power. In this regard, actuator 21 can produce any desirable output force and stroke. The TERFENOL-D actuation elements of these components have a response time ranging from ten microseconds to ten seconds, preferably ranging from one to twenty milliseconds, and can operate over a very broad frequency range (DC to ultrasonic) without material or performance degradation. Pump 62 can thus operate at a frequency up to 10 kHz.

Pump 62 and flow control valves 63 and 64 can be configured by means of controller 146 to provide pressure and flow on demand. Unlike conventional hydraulic systems, actuator 21 is not constantly absorbing power to keep the system under pressure in order to react in time to a sudden demand. In position and hold applications, there is also no need for constant power absorption. As a result, actuator 21 can remain off for long periods of time before being quickly actuated.

Pump 62 is relatively simple in comparison to conventional rotating pumps having a multitude of moving parts. Pump 62 consists only of a diaphragm 76 driven directly by motor 69. This direct drive of the diaphragm eliminates seal friction present in conventional pump designs. Pump 62 in concert with flow control valve 63 and 64 is capable of generating pressures in excess of 2,000 psi and, under certain configurations, in excess of 4,000 psi. More specifically, pump 62 produces a pressure within fluid system 61 ranging from 1 to 10,000 psi and preferably ranging from 2500 to 3500 psi and a flow rate ranging from 0 to 100 in^3/minute and preferably ranging from 10 to 20 in^3/minute. The power requirements of actuator 21 range from zero to 5000 watts and preferably from 100 to 200 watts. Preliminary calculations show that before flow losses and without storage, an actuator 21 measuring 12"x9"x3" and weighing under 20 pounds can generate 1 GPM at 5000 psi pressure differential from 5 kVA rms of 400 Hz ac line power. Size can be further reduced by increasing the output pressure differential to around 8000 to 10,000 psi and again by increasing operating frequency. The operating power level can be reduced by providing enough internal pressurized fluid storage to cover a duty cycle.

The current from controller 146 actuating coil 132 in servo valve 111 results in effective positioning of spools 122 and 123 by the magnetostriction of actuation element 131. This direct positioning of spools 122 and 123 by a drive current allows for significantly higher bandwidths in servo valve 111 than can be achieved with conventional solenoid-based servo valves.

Although the compact fluidic actuator of the present invention has been described as being an hydraulic actuator, it should be appreciated that the actuator can utilize pneumatics instead of hydraulics and be within the scope of the present invention. In addition, a compact actuator can be provided having one active flow control valve of the type disclosed above and one passive flow control valve. Alternatively, two passive flow control valves can be used in place of active valves 63 and 64. A compact actuator can also be provided having a pump and/or a servo valve which does not include an active actuation element. Pump 62 can further include a stroke amplifier similar to amplifiers 91, 92 and 133 for amplifying the movement of piston 71. It should be further appreciated that pump 62 and flow control valves 63 and 64, each utilizing magnetostrictive or other smart material actuation elements, can be produced as separate stand alone devices.

Actuator 21 is easily scalable for use in other applications. In addition, actuator 21 is easily reshaped and reconfigured to suit the particular application. Further aeronautics applications include helicopter rotor blade noise and vibration control and steering and brake actuation systems. Other exemplary applications include process control applications that are remote operating, such as well head control, gauge stations and pipeline valves that only have electric power available for operation. These control valves can be remotely activated by means of an electric signal with the self-contained actuation apparatus herein, which provides as advantages improved control and reduced size. The actuators herein are useful for well applications such as submersible pumps for oil and water and downhole drill bit operation and seismic sources. They are also useful for micropositioning applications such as precision drug delivery and ink control on computer printers where accurate metering and delivery is required. Automotive applications include hydraulic clutches, hydraulic clamps, hydraulic or pneumatic brakes, power steering pumps, fuel injection, brake, suspension and steering systems. Since the actuators of the present invention have little vibration and thus a low acoustic signature, they are well suited for use in submarines for replacing conventional hydraulic systems. Applications further include any other industrial application in which a distributed control algorithm in lieu of a centralized hydraulic source and master cylinder architecture is utilized.

It should be appreciated that the flow control valves of actuator 21 can have other configurations or embodiments. For example, in FIG. 8 there is shown a fragmentary view of actuator 21 having therein a normally open flow control valve 171 substantially similar to inlet control valve 63 and outlet control valve 64 described above. Like reference numerals have been used in FIG. 8 to describe like components between valve 171 and valves 63 and 64. As shown in FIG. 8, flow control valve 171 is disposed inside housing 31 and has a motor 172 substantially identical to motors 83 and 84. Motor 172 has an electrical connector 173 for electrically coupling the motor to controller 146. An amplifier 176 is included within valve 171 for amplifying the stroke of the smart material actuation element or piston of motor 172. Stroke amplifier 176, not cross sectioned in the drawings, is of a conventional type and substantially similar to stroke amplifiers 91, 92 and 133. The amplifier has an output rod 177 slidably carried therein and movable longitudinally as a function of the longitudinal movement of the piston within motor 172.

Flow control valve 171 has a poppet valve 178 which includes an elongate cylindrical valve stem 181. The stem 181 extends through a plug 182 disposed within a bore 183 provided in actuator housing 31. Fluid system 61 of actuator 21 has a first or inlet passageway portion 186 and a second or outlet passageway portion 187 which communicate with each other by means of poppet valve 178. Valve stem 181 extends across outlet passageway portion 187. A conventional seal 191 made from any suitable material such as a compliant polymer material is carried by the plug 182. Seal 191 is concentrically mounted about valve stem 181 for inhibiting the flow of fluid through plug 182. A valve plug 192 is mounted to the end of the valve stem 181. Valve stem and plug 181 and 192 are each made from any suitable noncorrosive material such as a stainless steel alloy or a polymer material. Plug 192 is sized and shaped to sealably engage a valve seat 193 secured within a bore 196 extending between passageway portions 186 and 187. Valve seat 193 has an orifice 197 which permits the liquid within fluid system 61 to flow from inlet passageway portion 186 to outlet passageway portion 187.

In operation and use, normally open flow control valve 171 permits the liquid within fluid system 61 to flow through orifice 197 in the absence of power to motor 172. Valve stem and plug 181 and 192 are movable from their first or open positions, shown in FIG. 8, to second or closed positions (not shown) upon actuation of motor 172. In the closed positions, valve plug 192 engages valve seat 193 to close off orifice 197 and thus preclude fluid from flowing between passageway portions 186 and 187. Stroke amplifier 176 provides the necessary longitudinal movement of valve stem 181, without the need for increasing the size of the piston in motor 172, for operation of poppet valve 178. Stroke amplifier 176 may be particularly preferred in a two-stage valve.

In FIG. 9, a normally closed flow control valve 204 is shown. Valve 204 is substantially similar to valve 171 and thus like reference numerals have been used in FIG. 9 to identify like components in valves 204 and 171. Flow control valve 204 has a poppet valve 206 which includes a valve seat 207 made from the same material as valve seat 193 and disposed within bore 196. Poppet valve 206 further includes an elongate valve stem 208 substantially similar to valve stem 181. Valve seat and stem 207 and 208 are made from the same materials as valve seat and stem 193 and 181. Valve seat 207 is provided with an orifice 209 sized and shaped to permit valve stem 208 to extend therethrough. A valve plug 212, made from the same material as valve plug 192, is secured to the end of valve stem 208 and is disposed within outlet passageway portion 187. A further bore 213 is provided in housing 31 and is longitudinally aligned with bores 196 and 183 to permit access to the poppet valve 206 and plug 182. A cap 214 is sealably disposed and secured within bore 213.

In operation and use, flow control valve 204 is in its normally closed position shown in FIG. 9 when motor 172 is not being energized. As such, the liquid of fluid system 61 is precluded from traveling through passageway portions 186 and 187 of the valve 204. Upon actuation of motor 172, valve plug 212 disengages from seat 207 and moves into inlet passageway portion 186 to permit liquid to flow through orifice 209 provided in the valve seat 207.

In a further embodiment of the integrated electric actuator of the present invention, a schematic of a compact actuator 221 is shown in FIG. 10 which is substantially identical to compact actuator 21. Like reference numerals have been used in FIG. 10 to describe the like components of actuators 221 and 21. Pump 62 of compact actuator 221 is bi-directional in comparison to uni-directional pump 62 of compact actuator 21. Flow control valves 63 and 64 of actuator 221 can alternatively serve as inlet valve or outlet valves to pump 62. First flow control valve 63 is fluidly connected to first line 116 and second flow control valve 64 is fluidly connected to second line 117. Controller 146 is programmed to cause the bi-directional flow control valves 63 and 64 to alternatively move the liquid within fluid system 61 in a first direction through a pump 62, in which the liquid flows out of second flow control valve 64 to second portion 56b of ram chamber 56, or in a second direction opposite of the first direction, in which the liquid flows out of first flow control valve 63 to first portion 56a of the ram chamber.

The bi-directionality of pump 62 eliminates the need for servo valve 111 in compact actuator 221. In place of high pressure accumulator 138 or a low pressure reservoir 139, actuator 221 has a single hydraulic tank 222 coupled to first and second lines 116 and 117 by means of respective lines 226 and 227. Tank 222 is vented, however a low-pressure gas or spring accumulator could also be used. Conventional poppet valves 228 and 229 regulate the flow of liquid between lines 112 and 113 and the hydraulic tank 222 and allow the tank 222 to provide liquid, as needed, to the low pressure side of fluid system 61 and isolate the tank from the high pressure side of the system 61.

Controller 146 synchronizes the drive currents to pump 62 and first and second flow control valves 63 and 64 to achieve pumping in either direction within fluid system 61. FIG. 12 shows time histories for the synchronized input signals to pump 62 and valves 63 and 64 for pumping liquid out second valve 64. A sinusoidal current is shown driving pump 62, although other periodic wave forms can be used in the alternative. The output rate of pump 62 is preferably controlled by modulating the amplitude of the input wave form. Alternatively, frequency control of the input wave form to pump 62 can also be used for determining the output rate of the pump. The time history for the position of pump piston 71 is similar to the input wave form for the pump, varying slightly due to the non-linear nature of the magnetostrictive material of piston 71 and the drive load exerted on the piston 71.

The second and third time histories shown in FIG. 12 are for the first and second check valves 63 and 64. As seen, the input signals for each of the valves 63 and 64 is similar to a square wave, except that the current ramps upwardly to and downwardly from the maximum current. First check valve 63 is acting as the input check valve and opens during the downward stroke of pump 62 so as to allow liquid to be drawn into pumping chamber 82. The first check valve 63 closes just as pump piston 71 reaches its minimum position, in which pumping chamber 82 has maximum volume. Second check valve 64 is acting as the output check valve in the synchronization shown in FIG. 12, and opens during the upward, pressurizing motion of pump piston 71. Second check valve 64 closes as the pump stroke reaches its maximum value, in which pumping chamber 82 has minimum volume.

To reverse the direction of liquid through pump 62, the synchronization of the input current to first and second check valves 63 and 64 is reversed so that second check valve 64 becomes the inlet check valve and first check valve 63 becomes the output check valve.

Controller 146 directs ram 41 to move to an extended position by causing pump 62 to move the liquid of fluid system 61 through first flow control valve 63 and line 116. The liquid enters first chamber portion 56a and engages upper surface 46 of ram head 42 to thus cause threaded end 43 of the ram 41 to extend outwardly from housing 31. When it is desired to retract rod 43 relative to housing 31, controller 146 resynchronizes the signals to check valve motors 83 and 84. Liquid within fluid system 61 now flows from second flow control valve 64 into second line 117 and then second chamber portion 56b to increase the hydraulic pressure upon lower surface 47. The closed looped fluid system 61 is simultaneously moving liquid from first chamber portion 56a so as to reduce the hydraulic pressure on upper surface 46 of ram head 42. In this manner, the redistribution of pressure across head 42 causes rod 43 to retract relative to housing 31.

The schematic of FIG. 11 includes a flow chart depicting the operation of controller 146. The inputs to controller 146 include the commanded position signal 231 for ram 41, the measured position for the ram 41 obtained by position sensor 163 and the measured differential pressure across ram 41 obtained by first and second pressure sensors 161 and 162. Controller 146 compares the commanded position signal 231 to the measured position signal in position comparison step 236 to form a position error signal 237 which is the difference between the commanded and the measured position of the ram 41. The position error signal 237 is used as an input to a position controller 238 which includes a control filter or algorithm that implements a differential equation in calculation step to determine the commanded force signal 241 required to drive ram 41 to the commanded position. The algorithm utilized in calculation step within position controller 238 accounts for the stability requirements of the system in which compact actuator 221 is utilized, uncertainties and variations in the actuator 221, desired closed-loop accuracy and dynamic range and any noise within sensors 161-163.

The commanded force signal 241 is passed through a pressure estimator 242, which in a pressure estimation step derives a commanded pressure signal 243. The pressure estimator is, in general, a dynamic signal processing element which, in its simplest form, divides the commanded force signal 241 by the hydraulic actuator area to determine the commanded pressure signal 243.

In pressure comparison step 246, a pressure difference error signal 247 is determined from a comparison of the commanded pressure signal 243 to the differential pressure on upper and lower surfaces 46 and 47 of ram head 42 as measured by first and second pressure sensors 161 and 162. The measured pressures from sensors 161 and 162 are differenced to provide the differential pressure acting on piston head 42. The pressure difference error signal 247 is used as the input to pressure controller and pump and check valve drive electronics 248. The pressure difference error signal 247 has both sign and magnitude so as to indicate to pressure controller or filter 248 both the amount of desired pressure that pump 62 must produce and which output of the pump should be at higher pressure.

The signal flow portions of controller 146 are implemented by any suitable means such as one or more microcontrollers or digital signal processing hardware/software. The drive currents are produced by any suitable means such as power semiconductor technologies.

The bi-directional operation of pump 62 permits compact actuator 221 to be relatively smaller, simpler and less expensive than compact actuator 21. The direction of liquid through pump 62 is changed by merely changing the phasing of the electrical signals to flow control valves 63 and 64. Pump 62 requires no operating point adjustment.

The inclusion of first and second pressure sensors 161 and 162 provides a closed loop pressure control system in compact actuator 221 which allows more accurate control of the power delivered to the hydraulic load. This closed loop pressure control system compensates for the influence of operation conditions such as temperature, wear and hydraulic load variations by delivering controlled pressure to the hydraulic load in the presence of these variations. This system allows controller 146 to deliver power only when required to maintain desired pressure levels, thus increasing the efficiency of actuator 221.

The closed loop or pressure control system described above can be used to provide force control of the output at ram 41 of the hydraulic system of compact actuator 221. The force, and even torque, output of the hydraulic system can be regulated by controlling by means of controller 146, as described above, the input pressure to ram 41 from the feedback signals of sensors 161-162.

The inclusion of position sensor 163 with the closed loop pressure control system described above provides a closed loop motion control system in compact actuator 221. In such a system, the inner pressure or force loop derived from pressure sensors 161-162 and lines 166, shown in FIG. 11, is controlled by an outer position loop derived from position sensor 163 and the related line 166. The use of these two loops to provide position control allows the outer position loop to be designed without detailed knowledge of the generally nonlinear dynamics of compact actuator 221. The high bandwidth of the inner force loop shields the outer position loop from these undesirable dynamics. Actuator 221 so configured as a motion controller can be specialized to the case where feed-forward information about the desired motion profile is available.

It should be appreciated that an integrated electric actuator of the present invention can be provided having only closed loop pressure control, that is no position sensor 163 for providing motion control, or alternatively open loop control, that is no sensors 161-163 for providing pressure and motion control. In an open loop configuration, the simplified controller 146 has as input electrical power and a desired pump output flow rate. Other implementations of the integrated electric actuator herein can be provided having a controller which provides a voltage modulated control signal to pump 62 and first and second flow control valves 63 and 64, instead of a current modulated control signal as described above.

It should be appreciated that the pump and valve apparatus of compact actuators 21 and 221 can have other configurations or embodiments. In FIG. 13, for example, a pump and valve apparatus 301 having inlet and outlet passive, spring-loaded check valves is provided in a housing 302 substantially similar to housing 31. A portion of actuator housing 302 is shown in FIG. 13. An elongate cylindrical cavity or pumping chamber 303 extending along a longitudinal axis 304 is provided in housing 302. Pumping chamber 303 is formed from an inner cylindrical surface 306 and a generally planar end surface 307 extending perpendicularly to cylindrical surface 306. A cylindrical piston member or piston 311 is disposed in pumping chamber 303. Housing 302 and piston 311 are each made from any suitable material such as stainless steel. Piston 311 has an outer cylindrical surface 312 and a planar end surface 313 extending at a right angle to outer cylindrical 312. The diameter of the outer cylindrical surface 312 is slightly smaller than the diameter of inner cylindrical surface 306. Piston 311 is slidable along longitudinal axis 304 between a first position in which end surface 313 is in close proximity to end surface 307, shown in solid lines in FIG. 13, and a second position in which the insert 313 is spaced farther away from end surface 307, shown in phantom lines in FIG. 13.

Pump and valve apparatus 301 includes means in the form of a transducer or motor 316 for driving piston 311 between its first and second longitudinal positions within pumping chamber 303. Motor 316 has a cylindrical rod-like element in the form of actuation element 317 made from any suitable active or smart material which changes shape when energized by being placed in an electromagnetic field. Preferred materials for actuation element 317 are the magnetostrictive materials discussed above with respect to rod 71 and more preferably a giant magnetostrictive material such as TERFENOL-D. A drive coil 318 substantially similar to coil 72 is concentrically carried or disposed about actuation element 317 and serves as means for creating an electromagnetic field through at least a portion of actuation element 317. More specifically, drive coil 318 creates an electromagnetic field through the entire actuation element 317. The actuation element 317 elongates when in the presence of such a magnetic field. An amplifier 321 is included within motor 316 for amplifying the stroke of actuation element 317. Stroke amplifier 321, not cross sectioned in FIG. 13, is of a conventional type and is substantially similar to stroke amplifiers 91 and 92. The amplifier 321 has an output rod 322 slidably carried therein and movable longitudinally as a function of the longitudinal movement of actuation element 317. The output rod 322 has an end secured to piston 311 by any suitable means such as a weld (not shown). Means for prestressing and/or magnetically biasing the magnetostrictive material of actuation element 317 can be included within motor 316 and be within the scope of the present invention.

A first or inlet poppet valve 331 is provided in piston 311 for controlling the flow of liquid from inlet line 112 into pumping chamber 303. In this regard, a cylindrical valve chamber 332 extends through end surface 313 along longitudinal axis 304 and terminates at a valve seat 333 formed in the piston 311. A fluid passageway 334 extends longitudinally through valve seat 333 and then transversely through piston 311 and outer cylindrical surface 312 thereof. Inlet line 312 extends through inner cylindrical surface 306 to communicate with fluid passageway 333. In this regard, a channel 336 extends longitudinally from fluid passageway 333 along outer cylindrical surface 312 toward piston end surface 313 for permitting communication between the fluid passageway 333 and inlet line 112 throughout the slidable movement of piston 311 in pumping chamber 303.

A valve head in the form of ball 337 is disposed in valve chamber 332 for engaging valve seat 333 and inhibiting the flow of liquid from fluid passageway 334 into valve chamber 332. Ball 337 is movable along longitudinal axis 304 within valve chamber 332 between a first or closed position in engagement with valves 333, shown in solid lines in FIG. 13, and a second or open position spaced longitudinally from valve seat 333, shown in phantom lines in FIG. 13. Inlet valve 331 has means which includes helical spring 341 for urging ball 337 to its closed position within the valve seat 333. Spring 341 is longitudinally disposed within valve chamber 332 and has one end engaging ball 337 and a second opposite end engaging a tubular retainer 342 press fit or otherwise suitably secured within chamber 332 flush with piston end surface 313. The tubular retainer 342 is provided with a bore 343 extending longitudinally therethrough for permitting liquid to pass from valve chamber 332 through end surface 313 into pumping chamber 303. Ball 337, spring 341 and retainer 342 are each made from any suitable material such as stainless steel.

Means is carried by housing 302 and piston 311 for precluding rotation of the piston 311 about longitudinal axis 304 within pumping chamber 303. Said means includes a longitudinally extending spline 346 extending radially inwardly from a cylindrical surface 306 into a longitudinal groove 347 provided in outer cylindrical surface 312 of piston 311. The interengagement of spline

346 and groove 347 serves to retain channel 336 and fluid passageway 334 in communication with inlet line 112 throughout the longitudinal movement of piston 311 within pumping chamber 303.

Annular sealing means in the form of first and second longitudinally spaced-apart O-rings 351 and 352 are carried by piston 311 for precluding liquid passing from inlet line 112 into fluid passageway 334 from flowing around the ends of piston 311. O-rings 351 and 352 are disposed within respective first and second annular grooves 353 and 354 extending circumferentially around outer cylindrical surface 312 of piston 311.

A second or exhaust valve 361 is provided within housing 302 for permitting liquid to flow out of pumping chamber 303 into outlet line 313. Exhaust or outlet poppet valve 361 is substantially similar to inlet valve 331 and includes a longitudinally-extending valve chamber 362 formed with a valve seat 363 at one end thereof. A fluid passageway 364 extends longitudinally through valve seat 363 and end surface 307 for permitting communication between valve chamber 362 and pumping chamber 303. A ball 366 is disposed in valve chamber 362 for longitudinal movement therein between a closed position in engagement with valve seat 363 and an open position spaced apart from the valve seat 363. A helical spring 367 serves to retain ball 366 within valve chamber 362 and urge the ball against valve seat 363. Helical spring 367 is retained in place by a tubular retainer 368 secured within valve chamber 362 and provided with a bore extending longitudinally therethrough for permitting fluid communication between the valve chamber 362 and outlet valve 113. Ball 366, spring 367 and retainer 368 are substantially similar to ball 337, spring 341 and retainer 342 described above.

In operation and use, inlet valve 331 controls the flow of liquid from inlet line 112 into pumping chamber 303 and exhaust valve 361 controls the flow of the pressurized fluid from the pumping chamber into outlet line 113. Ball 337 unseats from valve seat 333 when the pressure in inlet line 112 exceeds the pressure in pumping chamber 303. Similarly, ball 366 of exhaust valve 361 unseats from valve seat 363 when the pressure within pumping chamber 303 exceeds the pressure in outlet line 113.

The incorporation of inlet valve 331 within piston 311 increases the efficiency of pump and valve apparatus 301. Specifically, the longitudinal movement of ball 337 in valve chamber 332 is aligned and in the same direction with the longitudinal movement of piston 311 in pumping chamber 303 so that the longitudinal motion of piston assists in unseating and seating ball 337 with respect to valve seat 333. For example, when piston 311 is changing its direction from its downward stroke to its upward stroke in FIG. 13, the downward momentum of ball 337 and the liquid within both valve chamber 332 and fluid passageway 334 assists in unseating the ball 337 from valve seat 333. The stiffness of spring 341 can thus be increased without increasing the differential fluid pressure required to open inlet valve 331. In addition, the time required to open valve 331 is decreased. As such, the motion of piston 311 enhances the operation of valve 331 and increases the allowable operating frequency of pump and valve apparatus 301. The efficiency of pump and valve apparatus 301 is particularly enhanced when operated at its resonant frequency. Since the operating frequency of pump and valve apparatus 301 correlates directly with the flow rate, increased flow rates are provided.

Although motor 316 is shown with a stroke amplifier 321, it should be appreciated that a motor having an actuation element 317 and drive coil 318 can be provided without a stroke amplifier and be within the scope of the present invention. Furthermore, it should be appreciated that other types of motors or means for moving piston longitudinally within pumping chamber 303 can be utilized.

In an alternate embodiment of pump and valve apparatus 301, an annular groove can be provided around piston 311 in place of channel 336 to ensure continued fluid communication between fluid passageway 333 and inlet line 112 throughout the stroke of the piston. The annular groove would be disposed between O-rings 351 and 352 and would have a length and depth similar to the length and depth of channel 336. Such an annular groove would eliminate the need for spline 346 and groove 347.

Another embodiment of the pump and valve apparatus of the present invention is shown in FIG. 14. Pump and valve apparatus 381 therein is substantially similar to pump and valve apparatus 301 and like reference numerals have been used to describe like components of apparatus 301 and 381. Inlet valve 331 is longitudinally centered on a longitudinal axis 382 spaced-apart from and aligned parallel to central longitudinal axis 304.

Second or exhaust valve 386 is incorporated in piston 311 and extends along another longitudinal axis 387 spaced-apart and parallel to both axes 382 and 304. Exhaust valve includes a cylindrical valve chamber 388 which opens onto end surface 313 of piston 311. An annular shoulder 391 forms the terminus of valve chamber 388 within piston 311. A fluid passageway 392 has a longitudinal portion which opens into valve chamber 388 at shoulder 391 and a transverse portion which extends through outer cylindrical surface 312 of piston 311. Outlet line 113 opens into inner cylindrical surface 306 of pumping chamber 303 to communicate with fluid passageway 392. A channel 393, substantially similar to channel 336, extends longitudinally from fluid passageway 392 toward piston end surface 313 for permitting communication between the fluid passageway 392 and the outlet line 113 throughout the longitudinal movement of piston 311 within pumping chamber 303. A tubular member or retainer 396 is press fit or otherwise suitably secured within valve chamber 388 adjacent piston end surface 313. A bore 397 extends through tubular member 396 to a valve seat 398 formed by the inner end of tubular member 396. Ball 406 and helical spring 407 are disposed in valve chamber 388. Ball 406 is movable between a first position in engagement with valve seat 398 and a second position spaced apart from the valve seat. Helical spring 407 abuts shoulder 391 at one end and engages ball 406 at the other end for urging the ball against valve seat 398. Ball 406, spring 407 and retainer 396 are substantially similar to ball 337, spring 341 and retainer 342 described above. Third O-ring 411 disposed within third groove 412 extending circumferentially around the piston 311 serves as annular seal means between channel 393 and piston end surface 313. Thus, liquid traveling between fluid passageway 392 and outlet line 113 is bound between second and third O-rings 352 and 411.

The operation and use of pump and valve apparatus 381, and specifically inlet valve 331 thereof, is substantially similar to the operation of pump and valve apparatus 301. Incorporation of exhaust valve 386 into piston 311 further enhances the operating efficiency of pump and valve apparatus 381. Exhaust valve 386 operates in substantially the same manner as inlet valve 331. In this regard, when piston 311 changes directions between its upward stroke and its downward stroke in FIG. 14, the change in momentum of ball 406 and the liquid in valve chamber 388 and bore 397 provide an additional force for unseating ball 406 from valve seat 398.

Active valves can be substituted for the passive poppet valves of pump and valve apparatus 301 and 381. Pump and valve apparatus 421 illustrated in FIG. 15 is similar in certain respects to pump and valve apparatus 301 and 381 and like reference numerals have been used to describe like components of apparatus 301, 361 and 421. Pump and valve apparatus 421, shown somewhat schematically in FIG. 15, is formed from a pump housing 422 having a removable end cap 423. Pump housing 422 and end cap 423 are each made from any suitable material such as stainless steel and are part of housing 302 described above. Pump housing 422 is provided with a cylindrical pumping chamber 426 extending along a central longitudinal axis 427. Chamber 426 is formed in part by an inner cylindrical surface 428. End cap 423 has a cylindrical portion 431 formed with an outer cylindrical surface 432 and a planar end surface 433 extending at a right angle to cylindrical surface 432. Cylindrical portion 431 is disposed in housing 422 and secured to the pump housing by any suitable means so that end surface 433 forms one end of pumping chamber 426. Annular seal means in the form of an O-ring 436 disposed within an annular groove 437 provided in outer cylindrical surface 432 provides a fluid-tight seal between the outer and inner cylindrical surfaces 432 and 428.

A cylindrical piston 441 substantially similar to piston 311 described above and having an outer cylindrical surface 442 and a planar end surface 432 extending perpendicularly to outer surface 442 is disposed within pumping chamber 426. Piston 441 is longitudinally movable by motor 316 from a first position where end surface 443 thereof is in close proximity to end surface 433 and a second position in which piston end surface 443 is moved away from end surface 433. First and second O-rings 451 and 452 disposed within respective first and second annular grooves 453 and 454 provided in outer cylindrical surface 442 provide a fluid-tight seal between piston 441 and inner cylindrical surface 428 of pump housing 422.

A first or inlet valve 456 is provided in cylindrical portion 431 for controlling the flow of liquid from inlet line 112 into pumping chamber 426. Inlet valve 456 is aligned along longitudinal axis 427 and includes a longitudinally-extending valve chamber 458 extending through end surface 433 to a valve seat 459 formed within cylindrical portion 431. A first fluid passageway 461 is provided in valve seat 459 and cylindrical portion 431 for communicating with inlet line 112. A second fluid passageway 462 extends between the inner cylindrical surface of valve chamber 458 and outer cylindrical surface 432 of cylindrical portion 431 where it opens into a channel 463 provided in surface 432. Channel 463 extends longitudinally through end surface 433.

A valve head 466 made from steel or any other suitable material is disposed within valve chamber 458 for longitudinal movement and cooperative engagement with valve seat 459. Means for moving valve head 466 between opened and closed positions relative to valve seat 459 includes a motor 467 substantially similar to motor 83 and having an active drive element or piston 471 energized by a coil 472 concentrically carried thereabout. Piston 471 can be made from any of the materials discussed above with respect to piston 71, and is preferably made from a giant magnetostrictive material such as TERFENOL-D. Motor 467 is diametrically sized smaller than valve chamber 458 so that liquid passing through valve seat 459 travels between motor 467 and the inner surface of valve chamber 458 to second fluid passageway 462. Motor 467 is mounted on a cylindrical plug 473 made from stainless steel or any other suitable material. Plug 473 is secured within the end of valve chamber 458 flush with end surface 433 by any suitable means. A stroke amplifier and means for prestressing and/or magnetically biasing the magnetostrictive material of piston 471 can be included within motor 467 and be within the scope of the present invention.

A second or exhaust valve 481 is longitudinally disposed within piston 441. Exhaust valve 481 has a valve chamber 482 which extends longitudinally through piston end surface 443. Valve chamber 482 is longitudinally aligned with valve chamber 458. A tubular member or retainer 483 made from stainless steel or any other suitable material is secured within valve chamber 482 flush with end surface 443. The inner end of tubular retainer 483 is formed as a valve seat 486. A central bore 487 extends longitudinally through valve seat 486 and tubular retainer 483. A valve head 491 made from steel or any other suitable material is provided for sealable engagement with valve seat 486.

Means for moving valve head 491 from a first or closed position in engagement with valve seat 486 and a second or open position spaced away from valve seat 486 includes a motor 492 substantially identical to motor 467. Motor 492 has a cylindrical piston 493 and a coil 494 concentrically disposed about piston 493. Piston 493 is in its de-energized state and valve head 491 in an open position in FIG. 15. Motor 492 is mounted to the end surface of valve chamber 482 formed by piston 471. Cylindrical motor 492 has an external diameter less than the internal diameter of valve chamber 482 so as to permit liquid passing through valve seat 486 to flow through valve chamber 482 between motor 492 and the inner cylindrical surface of piston 471 to a fluid passageway 496 extending radially between valve chamber 482 and outer cylindrical surface 442. A longitudinally-extending channel 497 provided in outer cylindrical surface 442 between first and second O-rings 451 and 452 opens into fluid passageway 496. Outlet line 113 extends through inner cylindrical surface 428 for communication with fluid passageway 496 or channel 497 throughout the longitudinal movement of piston 441 for the egress of liquid from pumping chamber 426.

Means (not shown) similar to spline 346 and groove 347 are carried by pump housing 422 and piston 441 for precluding rotational movement of piston 441 about central longitudinal axis 427 for thus ensuring the registration of fluid passageway 496 and channel 497 with outlet line 113.

Pump and valve apparatus 421 operates in substantially the same manner as described above with respect to pump and valve apparatus 301 and 381. When energized by controller 146, motors 467 and 492 serve to close respective inlet and exhaust valves 456 at appropriate times and for appropriate durations for the operation of pump and valve apparatus 421. The operating efficiency of exhaust valve 481 is enhanced by its disposition within piston 441. As discussed above, the change in momentum of valve head 491 and the liquid within bore 487 and valve chamber 482 provide an additional force for opening valve head 491 when piston 441 changes direction from moving upwardly in FIG. 15 to moving downwardly.

Although pump and valve apparatus 421 is shown as having only exhaust valve 481 disposed within piston 441, it should be appreciated that one or both of inlet and exhaust valves 456 and 481 can be provided in piston 441 and be within the scope of the present invention.

The integrated electric actuator herein can have means for prestressing and/or magnetically biasing the magnetostrictive drive element, as described in copending U.S. patent application Serial No. 08/855,228 filed May 13, 1997 (File No. A-64718) incorporated herein by this reference, and be within the scope of the present invention. In addition, and as also described in U.S. patent application Serial No. 08/855,228 filed May 13, 1997 (File No. A-64718), ac magnetic flux return means can be provided in the actuator of the present invention.

From the foregoing, it can be seen that a new and improved fluidic actuator has been provided. The fluidic or integrated electric actuator is relatively compact in size and is easily scalable for different applications. The actuator provides hydraulic power with little acoustic noise. It has relatively few moving parts and uses electricity as a power source. A self-contained fluid system having a relatively small volume of fluid is utilized in the actuator. An output of relatively long stroke and high force is provided by the actuator. The actuator utilizes motors having smart material actuation elements made from TERFENOL-D. A method for operating the actuator is provided. The actuator can have a pumping apparatus which utilizes the momentum of the pump piston to enhance the efficiency of inlet and/or exhaust valves of the pump.

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What is claimed is:

1. A compact actuator for use with a fluid comprising a housing having a predetermined geometry, an actuation member mounted in the housing for movement between first and second positions for performing work, the actuation member having first and second surfaces, means within the housing for providing fluid to the first surface for moving the actuation member to the first position and to the second surface for moving the actuation member to the second position, said means including at least one transducer having an active element of a material which changes shape when in the presence of an electromagnetic field and means for producing an electromagnetic field which extends through at least a portion of the active element, the active element changeable from a first shape when in the absence of the electromagnetic field to a second shape when in the presence of the electromagnetic field.

2. The actuator of Claim 1 wherein the active element is made from an electrostrictive material and wherein the means for producing an electromagnetic field consists of means for producing an electric field.

3. The actuator of Claim 1 wherein the active element is made from a piezoelectric material and wherein the means for producing an electromagnetic field consists of means for producing an electric field.

4. The actuator of Claim 1 wherein the active element is made from a magnetostrictive material and wherein the means for producing an electromagnetic field consists of means for producing a magnetic field.

5. A compact actuator for use with a fluid comprising a housing having a predetermined geometry, an actuation member mounted in the housing for movement between first and second positions for performing work, the actuation member having first and second surfaces, means within the housing for providing fluid to the first surface for moving the actuation member to the first position and to the second surface for moving the actuation member to the second position, said means including at least one transducer having a magnetostrictive element of a material which changes shape when in the presence of a magnetic field and means for producing a magnetic field which extends through at least a portion of the magnetostrictive element, the magnetostrictive element changeable from a first shape when in the absence of the magnetic field to a second shape when in the presence of the magnetic field.

6. The actuator of Claim 5 wherein the magnetostrictive element is made from a rare earth-iron magnetostrictive material.

7. The actuator of Claim 6 wherein the magnetostrictive element is made from TERFENOL-D.

8. The actuator of Claim 5 wherein the magnetostrictive element is in the form of a rod and wherein the means for producing a magnetic field consists of a coil concentrically carried about the rod.

9. The actuator of Claim 5 wherein the at least one transducer includes a pump.

10. The actuator of Claim 9 wherein the at least one transducer further includes a control valve having such magnetostrictive element for directing fluid relative to the pump.

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11. The actuator of Claim 10 wherein such magnetostrictive element changes the valve between opened and closed positions.

12. The actuator of Claim 11 wherein the valve is normally open.

13. The actuator of Claim 11 wherein the valve is normally closed.

14. The actuator of Claim 11 wherein the valve includes means for amplifying the movement created by such magnetostrictive element changing from a first shape to a second shape.

15. The actuator of Claim 5 wherein the at least one transducer includes a pump and first and second control valves for directing fluid relative to the pump, the pump and the first and second control valves each having a magnetostrictive element changeable from a first shape in the absence of the magnetic field to a second shape in the presence of the magnetic field.

16. A self contained actuation apparatus comprising a housing having an exterior, an actuation member mounted in the housing and having a portion thereof accessible from the exterior, the actuation member having first and second surfaces and being movable between first and second positions, self contained fluid means within the housing for providing fluid to the first surface for moving the actuation member to the first position and to the second surface for moving the actuation member to the second position and electrical means carried within the housing and coupled to the fluid means for controlling the movement of the actuation member.

17. The apparatus of Claim 16 wherein the fluid means includes a pump.

18. The apparatus of Claim 17 wherein the pump includes a drive member made from a magnetostrictive material and a coil concentrically mounted about the drive member for causing the drive member to change from a first shape when in the absence of a magnetic field and a second shape when in the presence of the magnetic field.

19. An apparatus as in Claim 17 wherein the fluid means includes first and second valves for controlling the flow of fluid into and out of the pump, each of the valves having a drive member made from a magnetostrictive material and a coil concentrically mounted about the drive member for opening and closing the valve.

20. An apparatus as in Claim 19 wherein the electrical means includes a controller electrically coupled to the first and second valves for causing the valves to alternatively direct fluid through the pump in first and second opposite directions, the first valve directing the fluid to the first surface of the actuation member for causing movement of the actuation member in a first direction and the second valve directing fluid to the second surface of the actuation member for causing movement of the actuation member in a second direction.

21. A compact actuator comprising a housing having a predetermined geometry, an actuation member mounted in the housing for movement between first and second positions for performing work, the actuation member having first and second surfaces, a self contained fluid system carried within the housing having a pump for providing a pressurized fluid, the pump including a drive member made of a magnetostrictive material and a coil concentrically mounted about the drive member, a controller mounted within the housing for directing energy to the coil so as to cause the drive member to change shape for pressurizing the fluid,

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means coupled between the pump and the actuation member for selectively directing the pressurized fluid to the first surface for moving the actuation member to the first position and to the second surface for moving the actuation member to the second position.

22. An actuator as in Claim 21 wherein the means coupled between the pump and the actuation member includes first and second valves for controlling the flow of fluid into and out of the pump, each of the valves having a drive member of a magnetostrictive material and a coil concentrically mounted about the drive member, the controller electrically coupled to the coils of the valves for selectively causing the drive members of the valves to elongate for closing or opening the valves.

23. An actuator as in Claim 22 wherein the means coupled between the pump and the actuation member includes a spool valve, the controller being electrically coupled to the spool valve.

24. An actuator as in Claim 23 wherein the controller causes the first and second valves to direct the fluid through the pump in a single direction and the spool valve to alternatively direct the fluid to the first and second surfaces of the actuation member.

25. An actuator as in Claim 22 wherein the controller causes the first and second valves to alternatively direct the fluid through the pump in first and second opposite directions, the first valve directing the fluid to the first surface of the actuation member and the second valve directing the fluid to the second surface of the actuation member.

26. A controller for operating a compact actuator having an actuation member movable from a first position to a second position for performing work and a self-contained fluid system for providing a pressurized fluid, the fluid system including a pump and first and second valves for directing fluid through the pump in first and second directions to respectively move the actuation member to the first and second positions, comprising electrical means for providing a first set of synchronized electrical signals to the pump and to the first and second valves to direct the fluid through the pump in the second direction and thus move the actuation member to the second position and for providing a second set of synchronized electrical signals to the pump and to the first and second valves to direct the fluid through the pump in the first direction and thus move the actuation member to the first position.

27. A controller as in Claim 26 wherein the pump and the first and second valves each have a drive member made from a magnetostrictive material and a coil concentrically mounted about the drive member.

28. A method for operating a compact actuator having an actuation member movable from a first position to a second position for performing work and a self-contained fluid system for providing a pressurized fluid, the fluid system including a pump and first and second valves for directing fluid through the pump in first and second directions to respectively move the actuation member to the first and second positions, comprising the steps of providing a set of synchronized electrical signals to the pump and the first and second valves to direct the fluid through the pump in the second direction and thus move the actuation member to the second position.

29. A method as in Claim 28 further comprising the step of providing another set of synchronized electrical signals to the pump and the first and second valves to direct the fluid through the pump in the first direction.

30. A method as in Claim 28 wherein the fluid exerts a pressure on the actuation member, further comprising the step of providing a commanded pressure electrical signal indicating the commanded pressure to be exerted on the actuation member,

measuring the actual pressure on the actuation member, providing a measured pressure electrical signal corresponding to the actual pressure on the actuation member, comparing the commanded pressure electrical signal to the measured pressure electrical signal and providing another set of synchronized electrical signals to the pump and the first and second valves for causing the commanded pressure to be exerted on the actuation member.

31. A method as in Claim 30 wherein the actuation member has an actual position further comprising the step of providing a commanded position electrical signal indicating the commanded position to which the actuation member should be moved, measuring the actual position of the actuation member, providing a measured position electrical signal corresponding to the actual position of the actuation member, comparing the commanded position electrical signal to the measured position electrical signal to provide the commanded pressure electrical signal.

32. A method as in Claim 28 wherein the pump has a reciprocating piston, further comprising the step of reciprocating the piston at the resonant frequency of the pump.

33. A method as in Claim 32 wherein the piston is made from a magnetostrictive material.

34. A pumping apparatus for use with a fluid comprising a housing provided with a chamber, a piston slidably disposed in the chamber for movement along an axis between first and second spaced-apart positions, the piston having a valve in communication with the chamber for the intake or exhaust of fluid in the chamber, the valve including a valve seat and a valve head in the piston, the valve head in the piston being movable in the piston along the axis between a first position in engagement with the valve seat for inhibiting the flow of fluid through the valve seat and a second position spaced-apart from the valve seat for permitting fluid to flow through the valve seat and means carried by the piston for urging the valve head to one of its first and second positions.

35. A pumping apparatus as in Claim 34 wherein the urging means includes means for urging the valve head to its first position.

36. A pumping apparatus as in Claim 34 wherein the urging means is a spring.

37. A pumping apparatus as in Claim 34 wherein the urging means is a motor having an active element and means for producing an electromagnetic field which extends through at least a portion of the active element.

38. A pumping apparatus as in Claim 37 wherein the active element is made from a magnetostrictive material and wherein the means for producing an electromagnetic field includes a coil concentrically disposed about the active element.

39. A pumping apparatus as in Claim 34 wherein the valve is for the intake of fluid in the chamber, the piston having an additional valve in communication with the chamber for the exhaust of fluid in the chamber, the additional valve including a valve seat and a valve head movable along the axis between a first position in engagement with the valve seat and a second position spaced-apart from the valve seat of the additional valve.

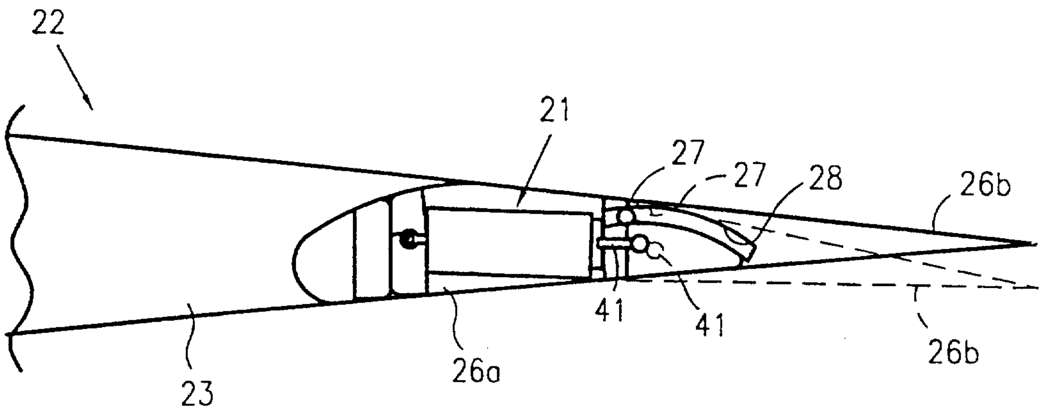
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FIG. 1

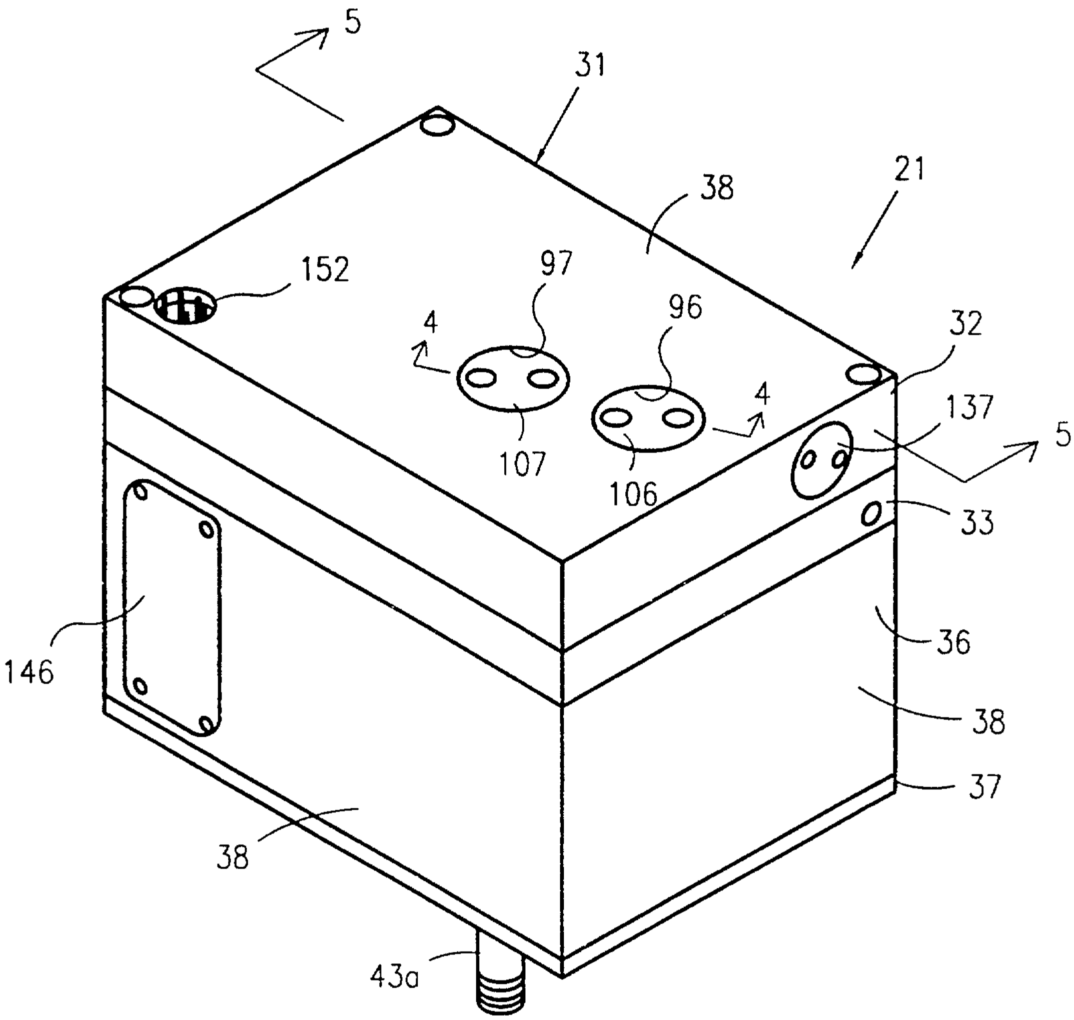
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FIG. 2

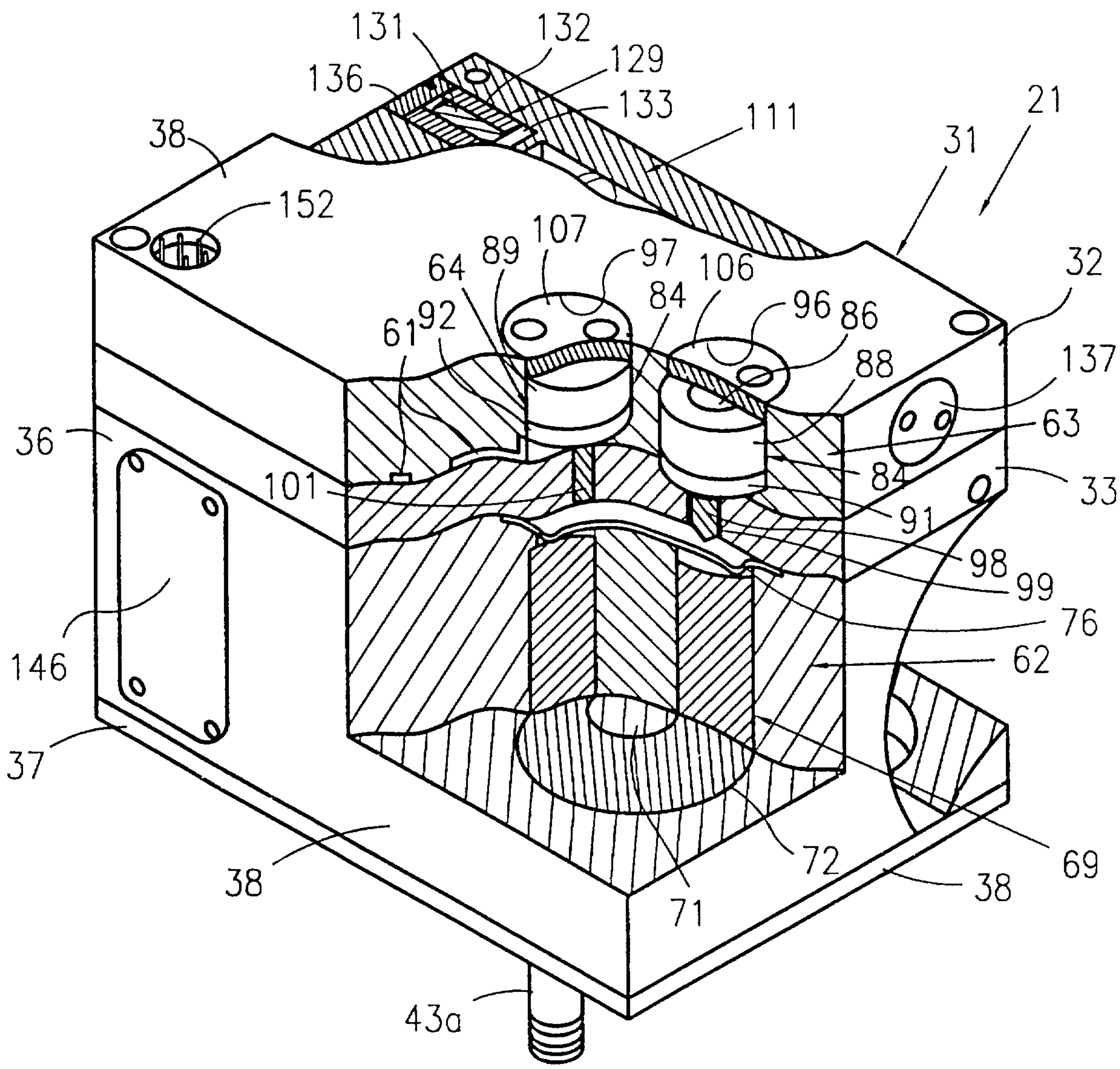

FIG. 3

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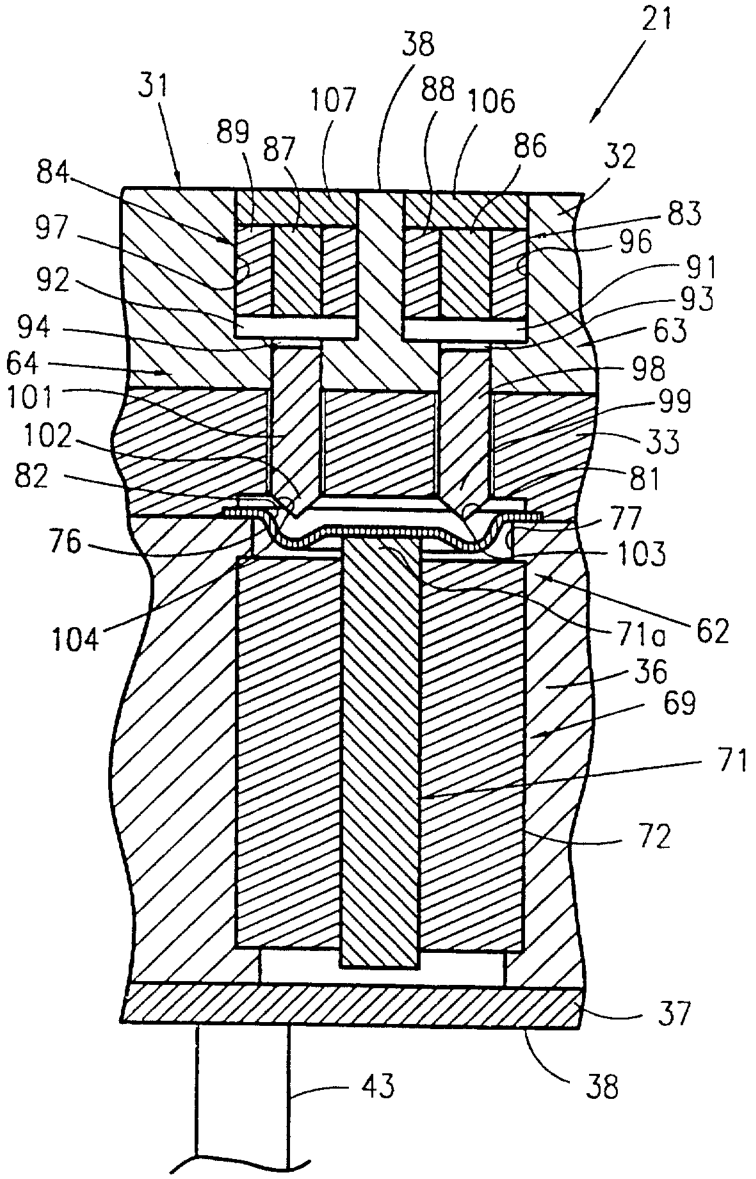
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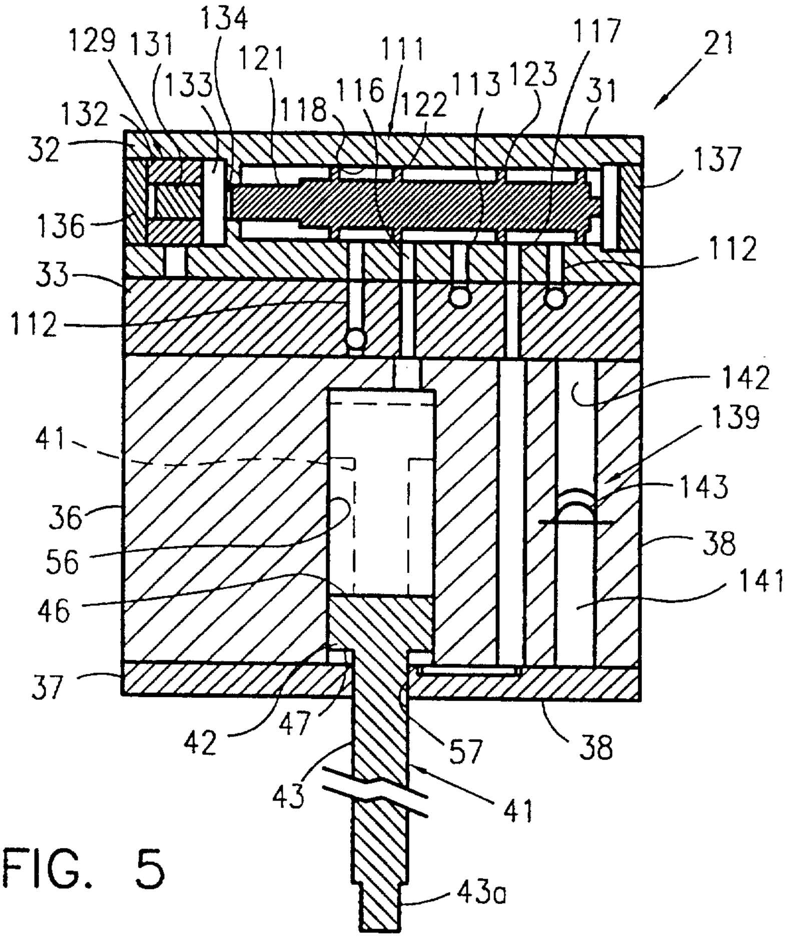
FIG. 4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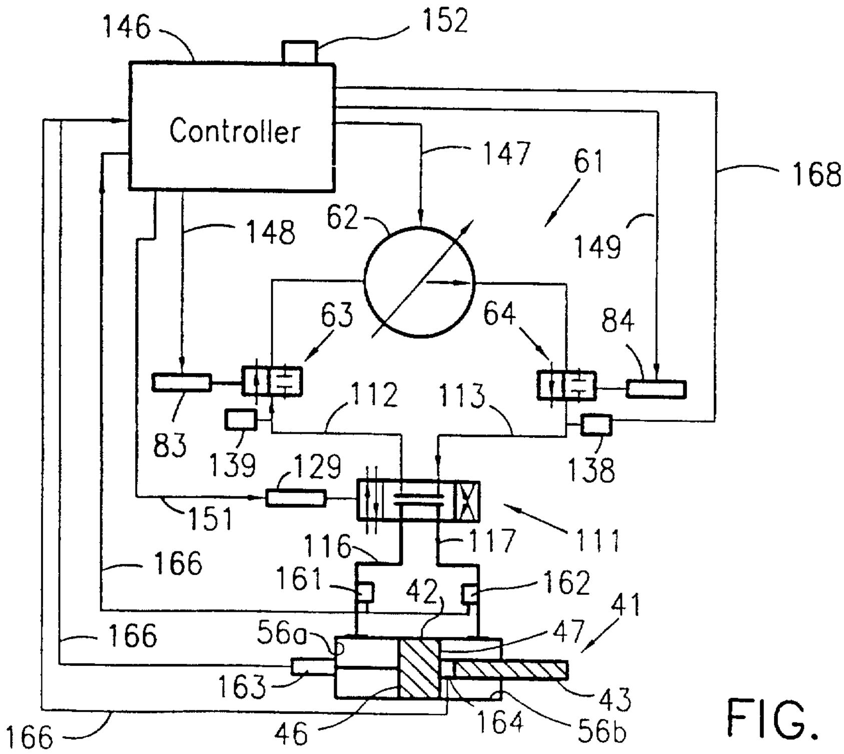
43a

FIG. 5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Controller
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166
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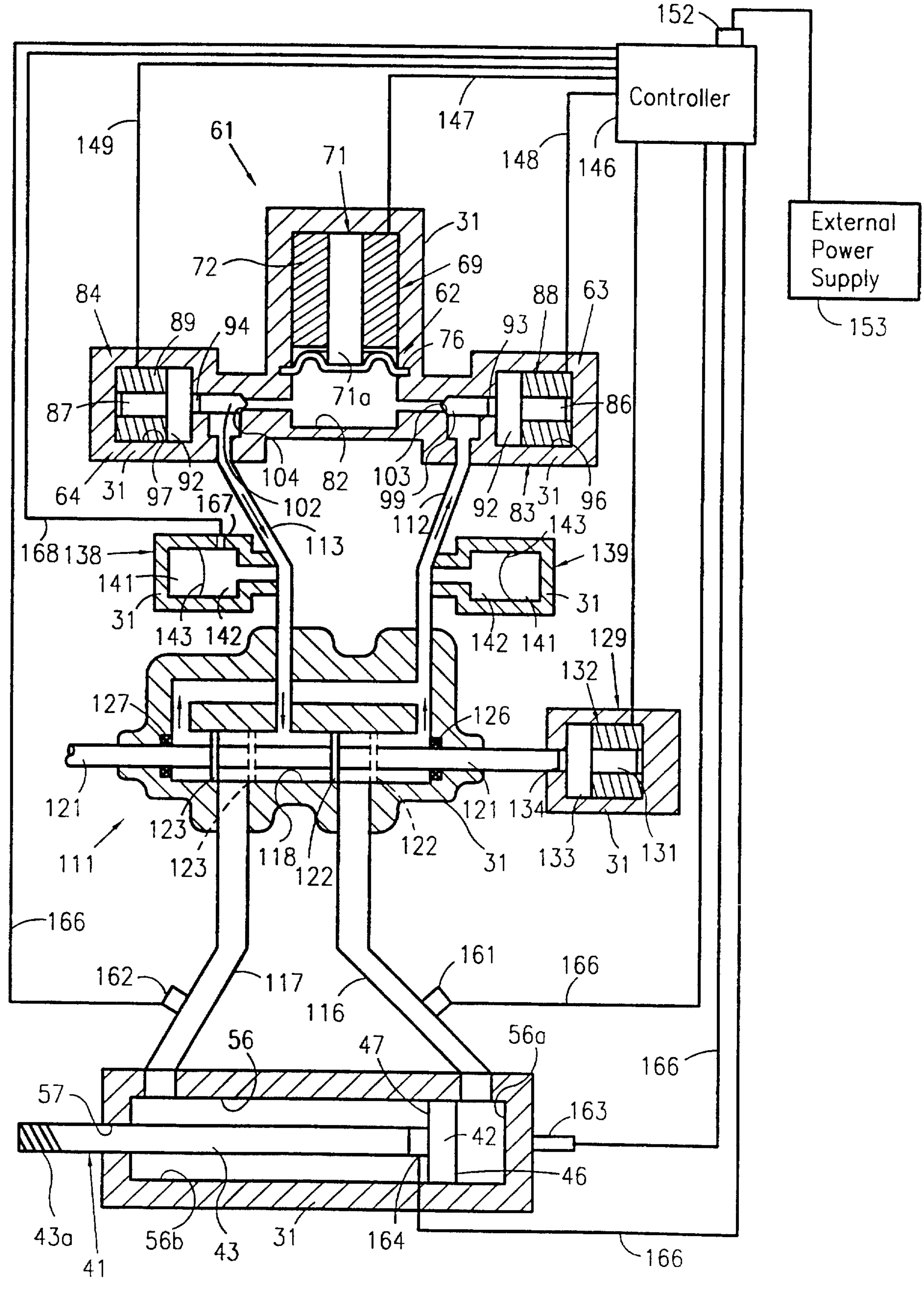
43
166
46
164
56b

FIG. 6


Controller
External Power Supply
152
153
146
147
148
149
61
71
72
31
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76
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71a
87
86
64
97
92
104
82
103
99
102
83
96
167
113
112
168
138
141
143
142
139
127
126
129
132
121
123
118
122
111
134
133
131
166
162
161
117
116
56
47
56a
57
42
163
46
43a
41
56b
43
164

FIG. 7

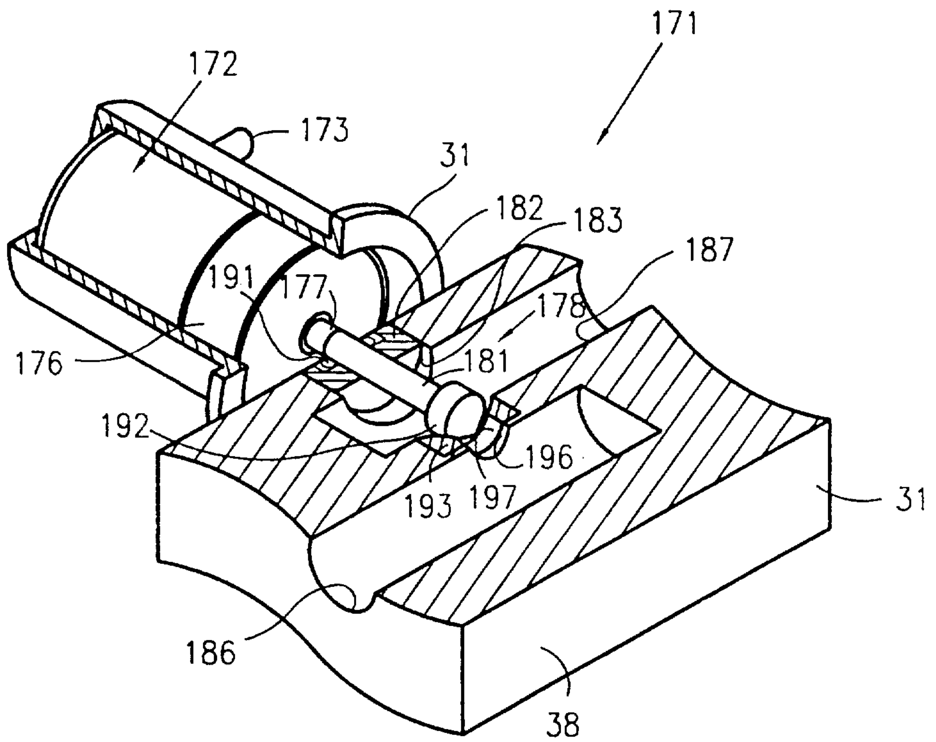

171
172
173
31
182
183
187
191
177
178
176
181
192
196
193
197
31
186
38

FIG. 8

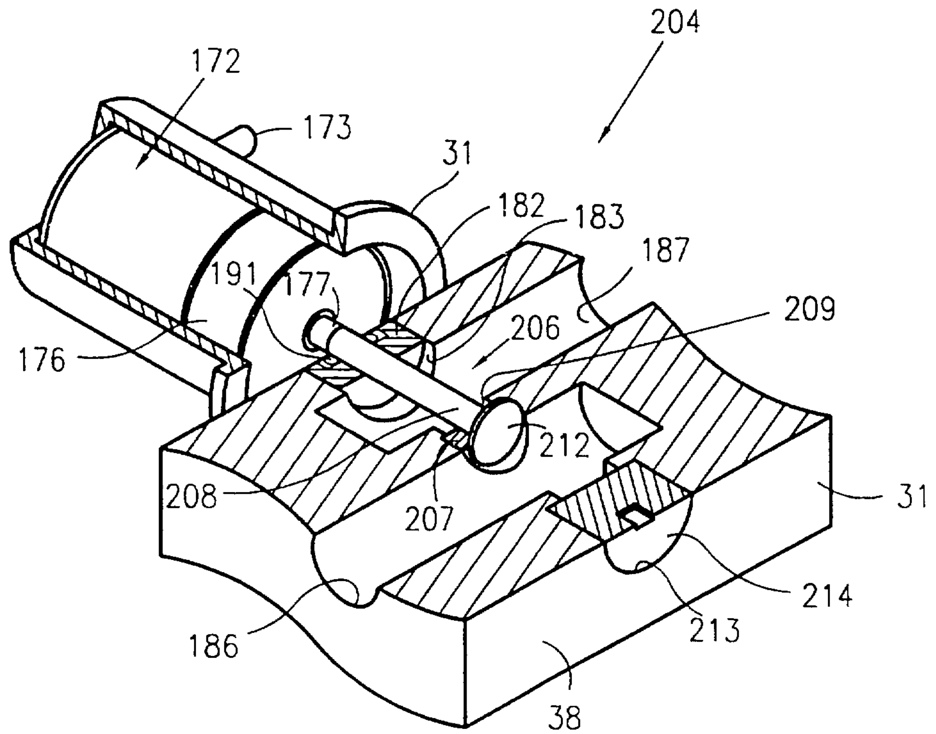

204
172
173
31
182
183
187
191
177
206
209
176
212
208
207
31
214
186
213
38

FIG. 9

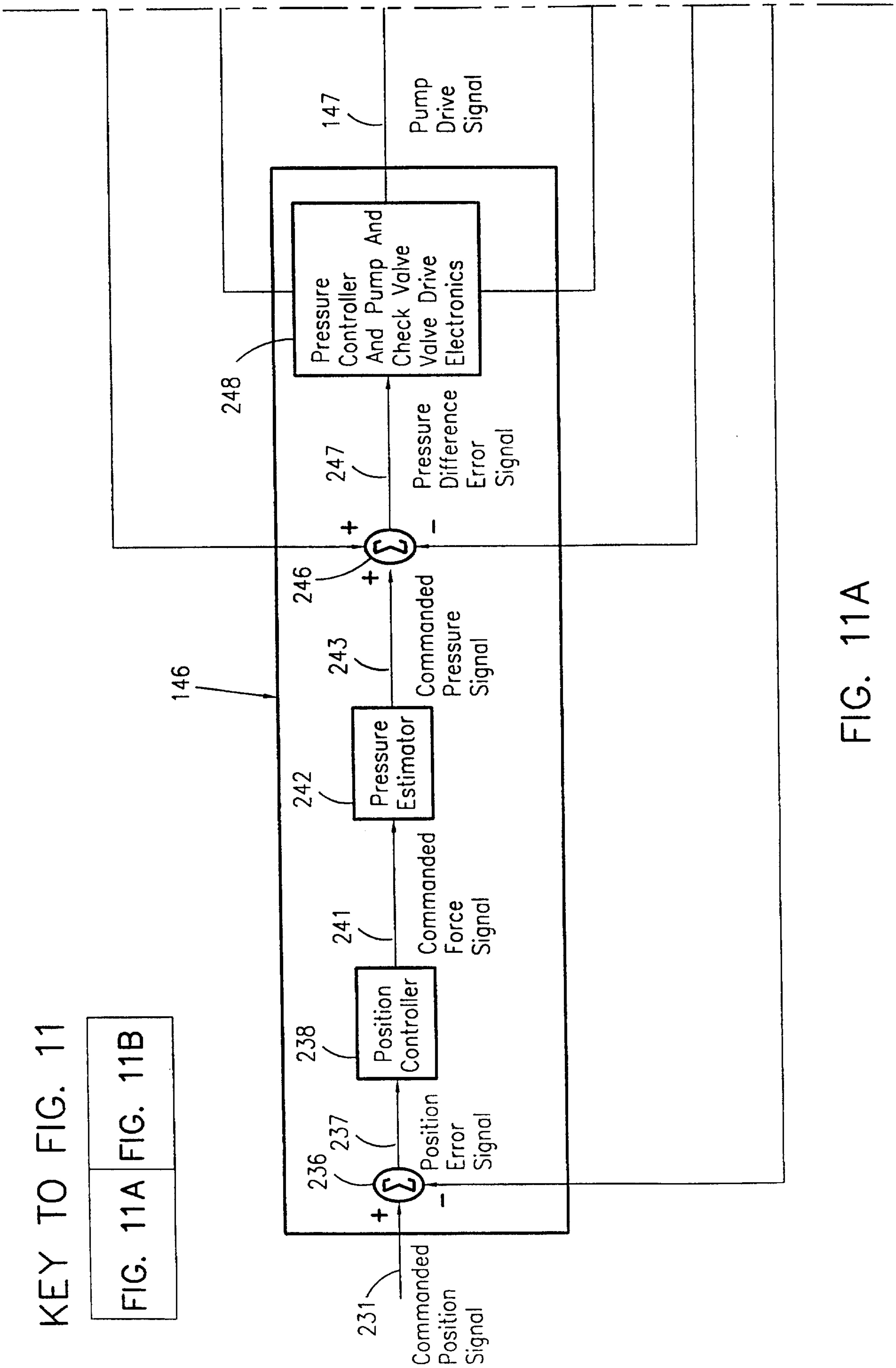

Pump Drive Signal
147
Pressure Controller And Pump And Check Valve Valve Drive Electronics
248
247
Pressure Difference Error Signal
246
243
Commanded Pressure Signal
146
Pressure Estimator
242
241
Commanded Force Signal
Position Controller
238
236
237
Position Error Signal
231
Commanded Position Signal
KEY TO FIG. 11
FIG. 11A
FIG. 11B

FIG. 11A

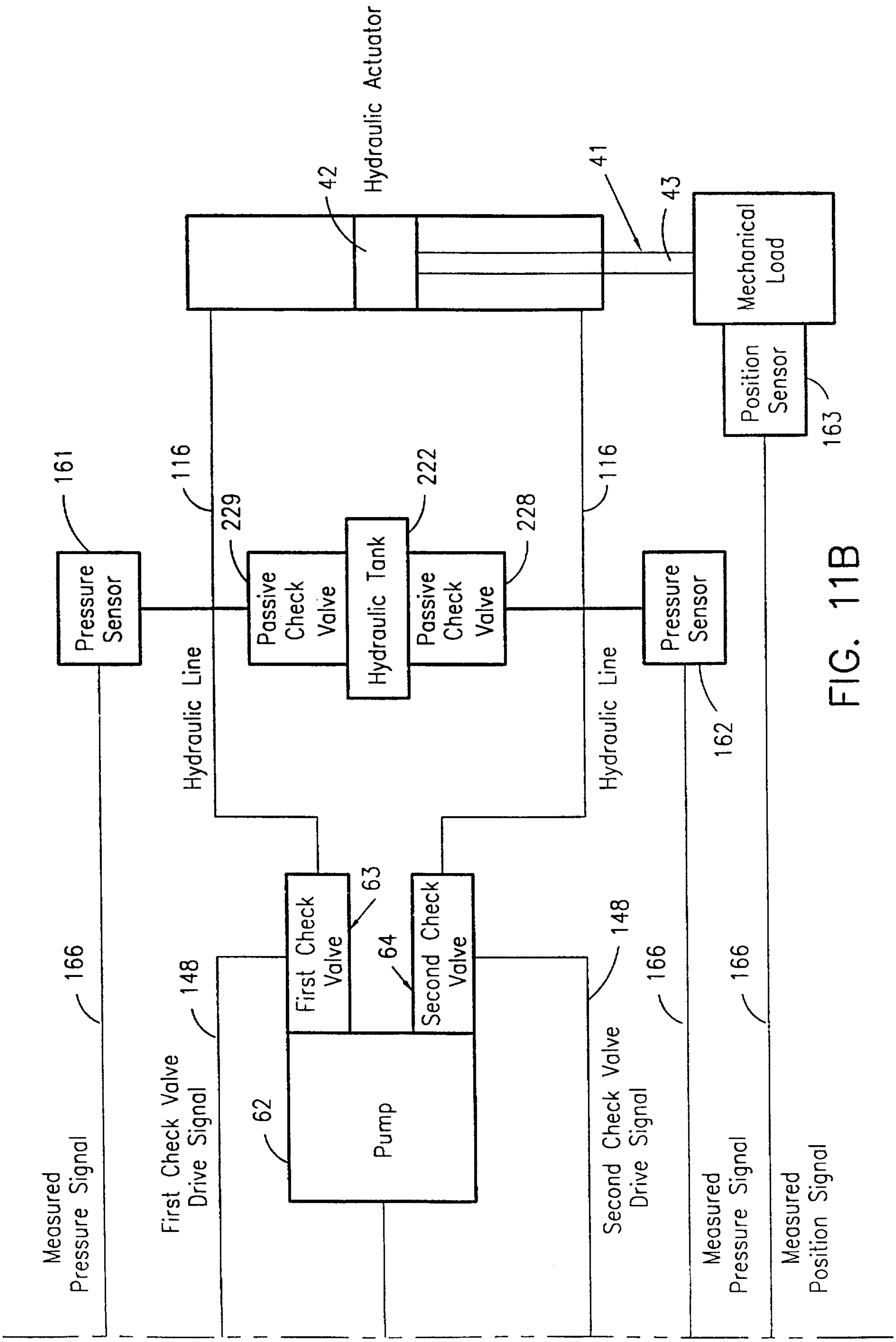

Hydraulic Actuator
42
41
43
Mechanical Load
Position Sensor
163
161
Pressure Sensor
116
229
Passive Check Valve
222
Hydraulic Tank
228
Passive Check Valve
116
Pressure Sensor
162
Hydraulic Line
Hydraulic Line
First Check Valve
63
64
Second Check Valve
148
148
Pump
62
166
166
166
Measured Pressure Signal
First Check Valve Drive Signal
Second Check Valve Drive Signal
Measured Pressure Signal
Measured Position Signal

FIG. 11B

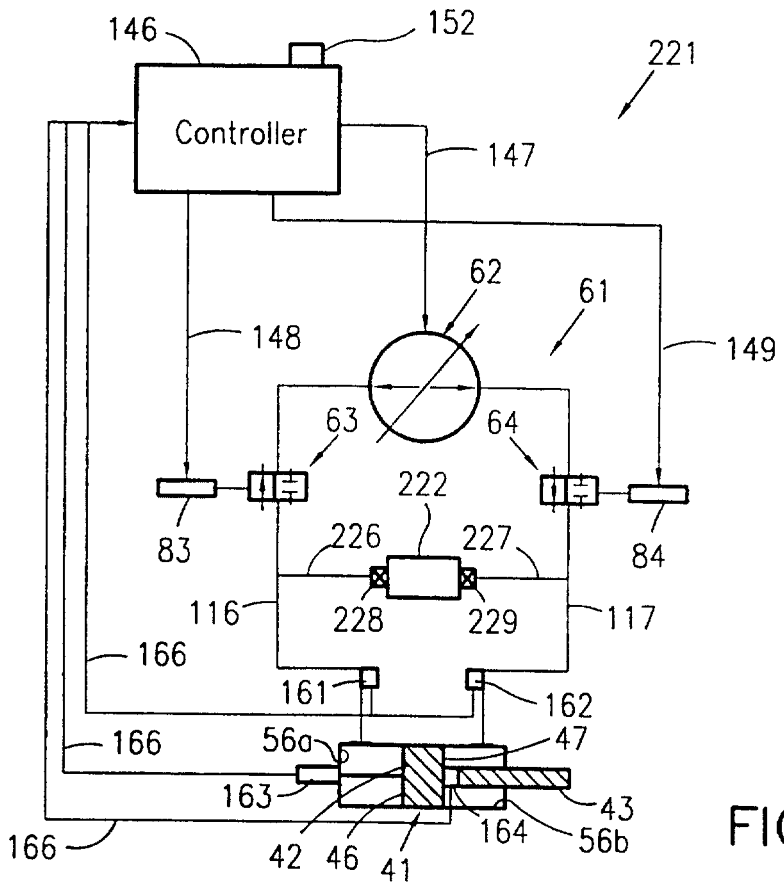

146
152
221
Controller
147
62
61
148
149
63
64
222
83
84
226
227
116
228
229
117
166
161
162
166
56a
47
163
43
166
42
46
41
164
56b

FIG. 10

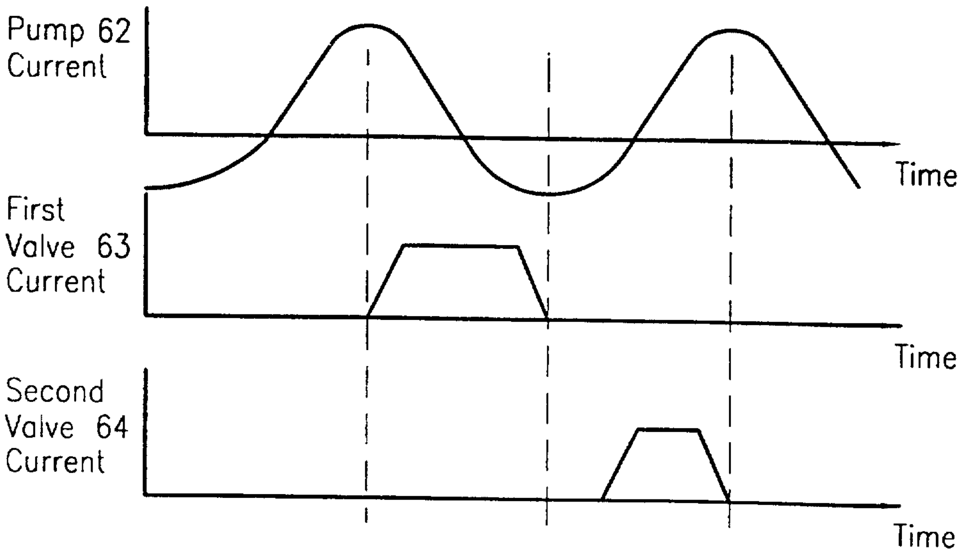

Pump 62
Current
Time
First
Valve 63
Current
Time
Second
Valve 64
Current
Time

FIG. 12

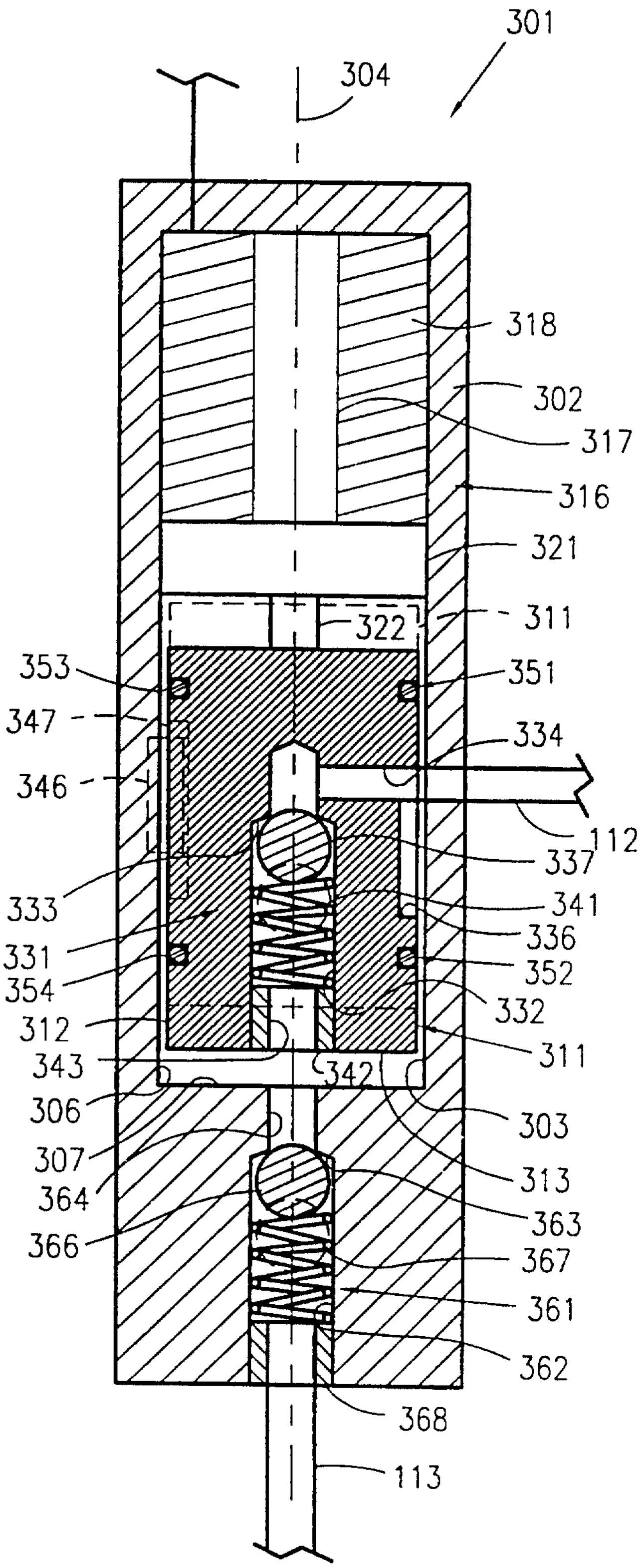

301
304
318
302
317
316
321
311
322
353
351
347
334
346
112
337
333
341
336
331
352
354
332
312
311
343
342
306
303
307
313
364
363
366
367
361
362
368
113

FIG. 13

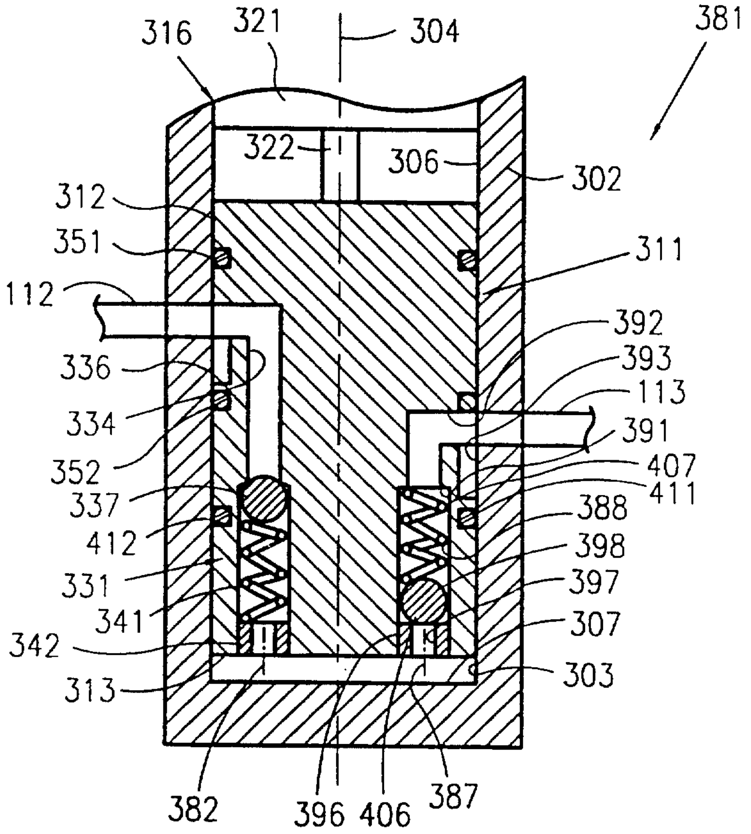

316
321
304
381
322
306
302
312
351
311
112
392
393
336
113
334
391
352
407
337
411
412
388
398
331
397
341
307
342
303
313
382
396
406
387

FIG. 14

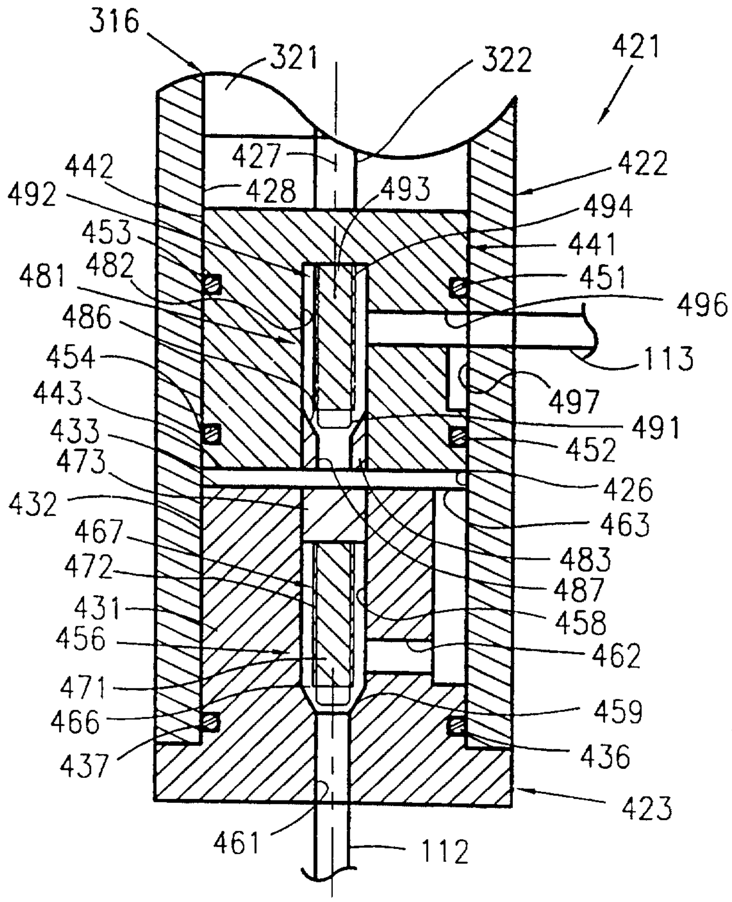
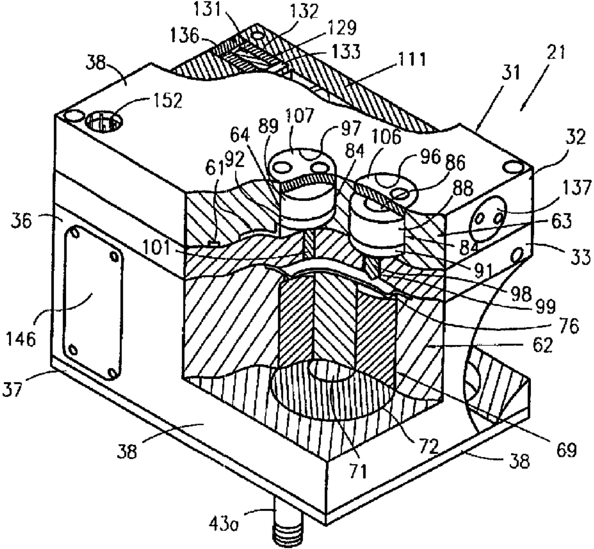

316
321
322
421
427
422
442
428
492
493
494
453
441
481
482
451
496
486
113
454
497
443
491
433
452
473
426
432
463
467
483
472
487
431
458
456
462
471
459
466
436
437
423
461
112

FIG. 15


131
132
136
129
133
38
111
21
31
152
107
64
89
97
106
32
84
92
96
86
61
88
137
36
63
101
84
33
91
98
99
76
146
62
37
38
72
69
71
38
43o