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(54) HYDRAULICALLY ACTUATED, ELECTRICALLY CONTROLLED LINEAR MOTOR

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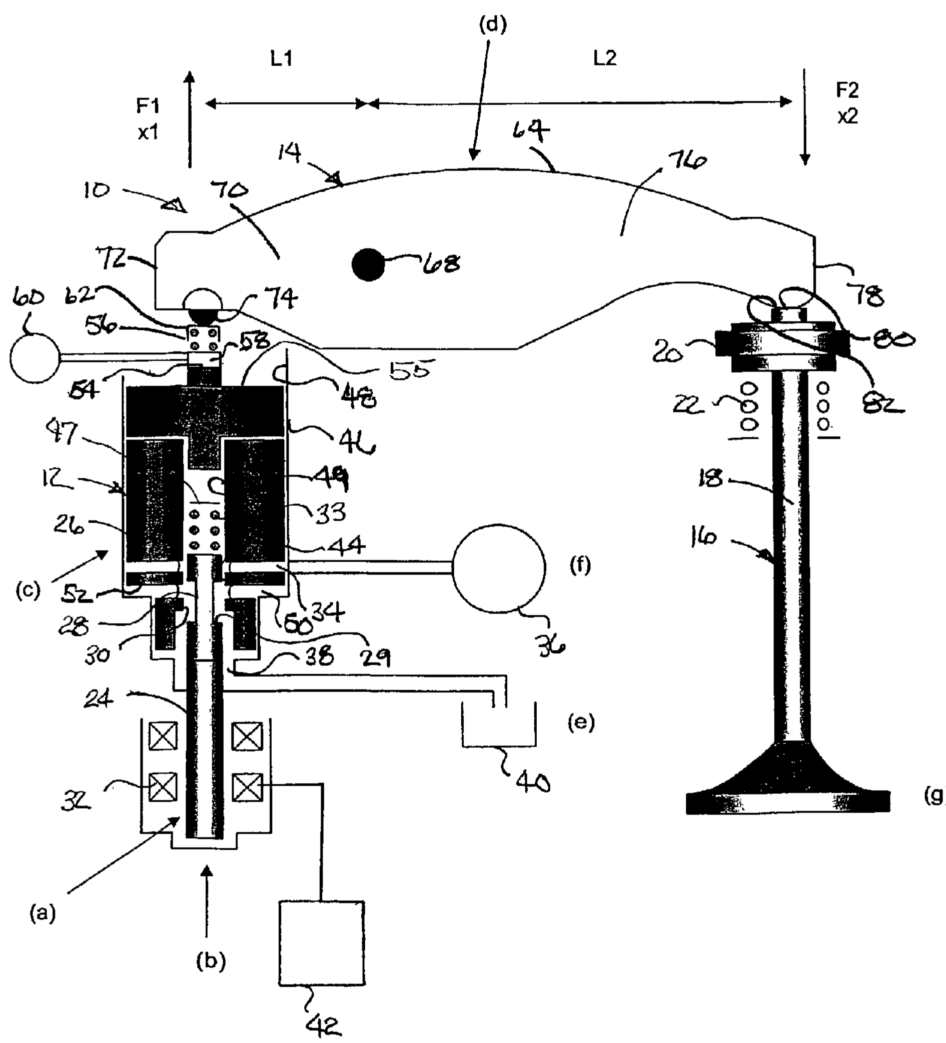
(57) **ABSTRACT**

A hydraulically actuated, electrically controlled linear motor includes a trigger that is electrically controllable. A hydraulic cartridge has a needle and a piston, the needle being operably coupled to the trigger and the piston being operably fluidly in communication with a source of high pressure actuation fluid, translation of the needle effected by the trigger acting to selectively port high pressure actuation fluid to the piston and to vent actuation fluid from the piston. A method of linear actuation is also included.



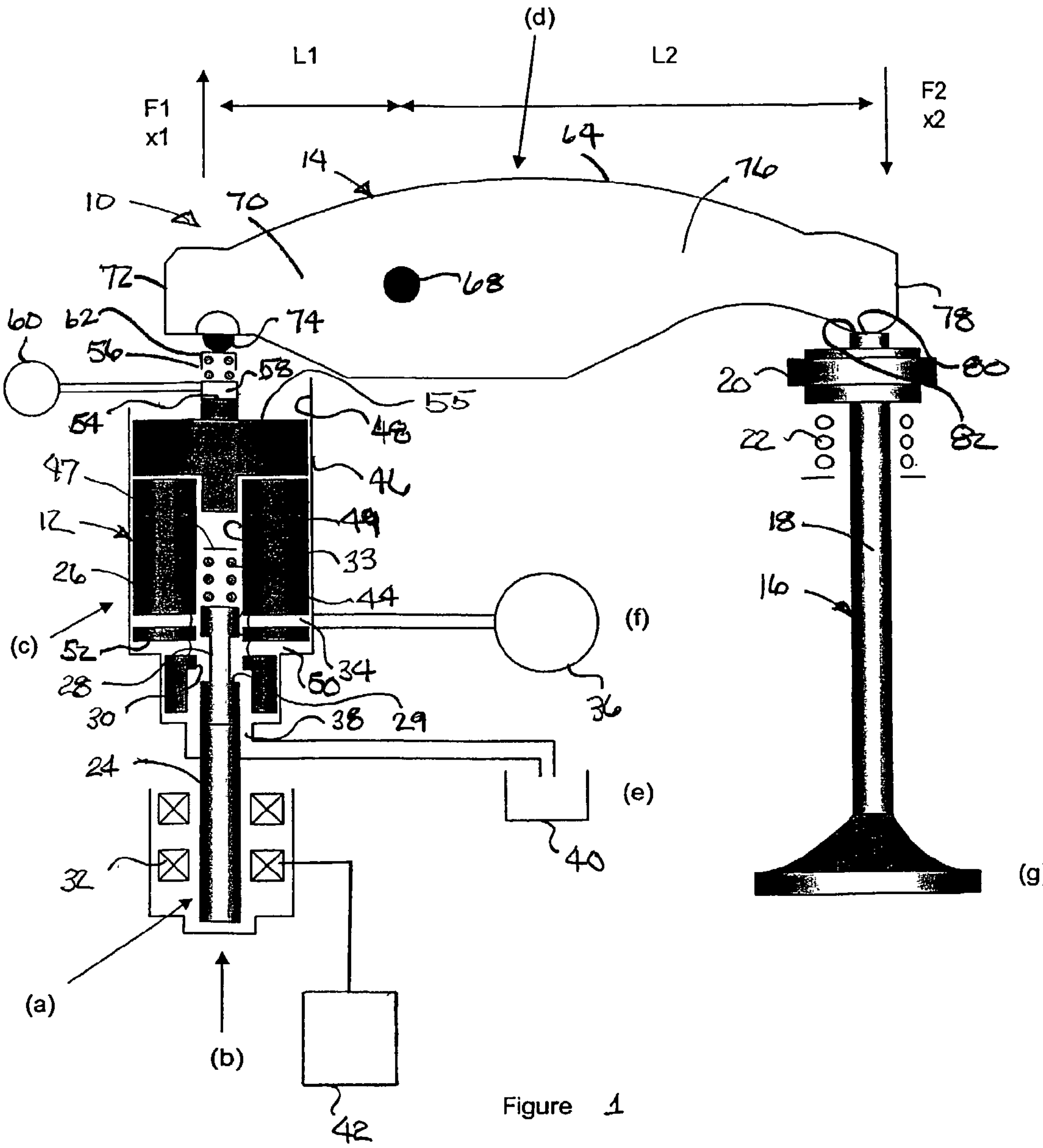
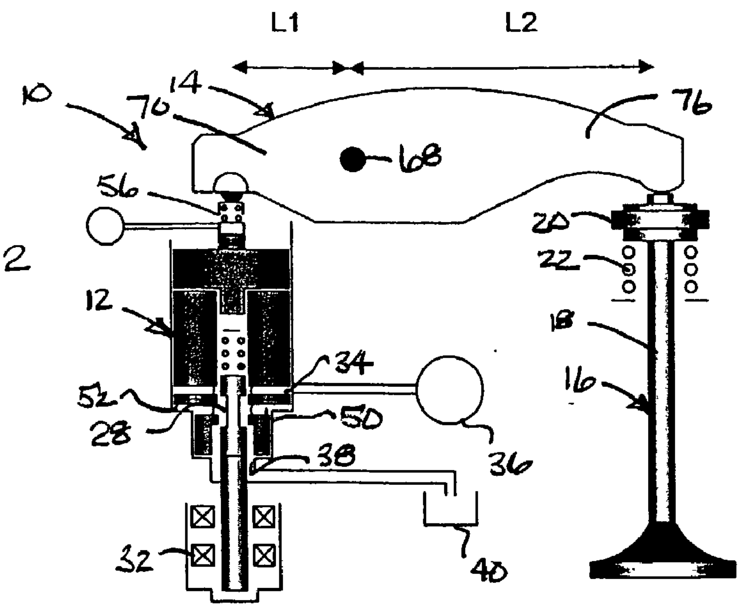

(d)
L1
L2
F1
x1
F2
x2
64
14
76
10
70
72
68
78
60
62
74
56
58
80
20
54
55
48
82
22
47
46
12
49
18
26
33
16
44
(f)
(c)
52
34
50
28
30
38
29
36
24
(e)
40
32
(g)
(a)
(b)
42

Figure 1

FIG. 2



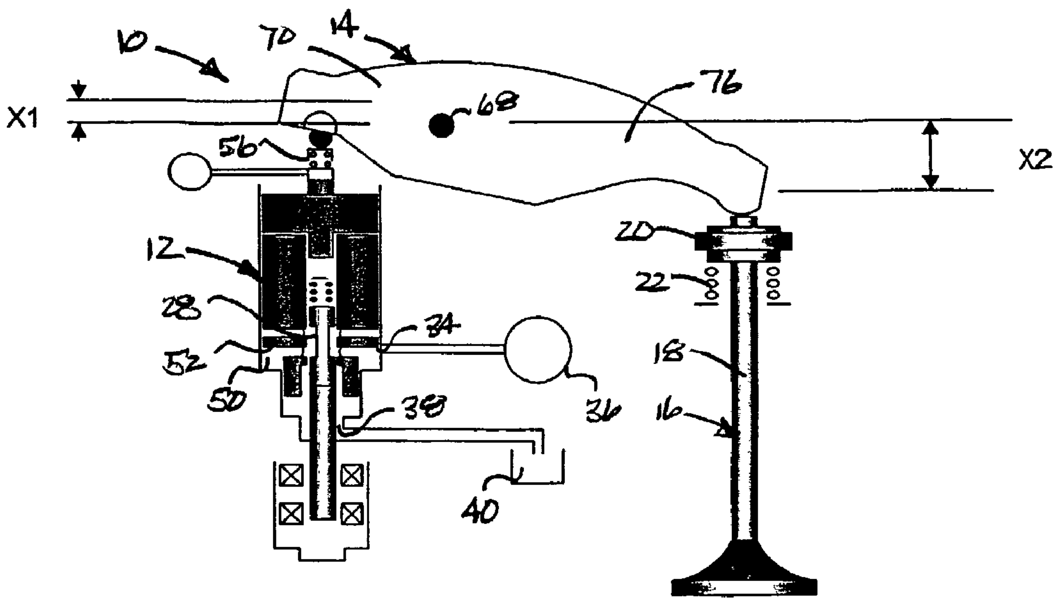


Figure 3.

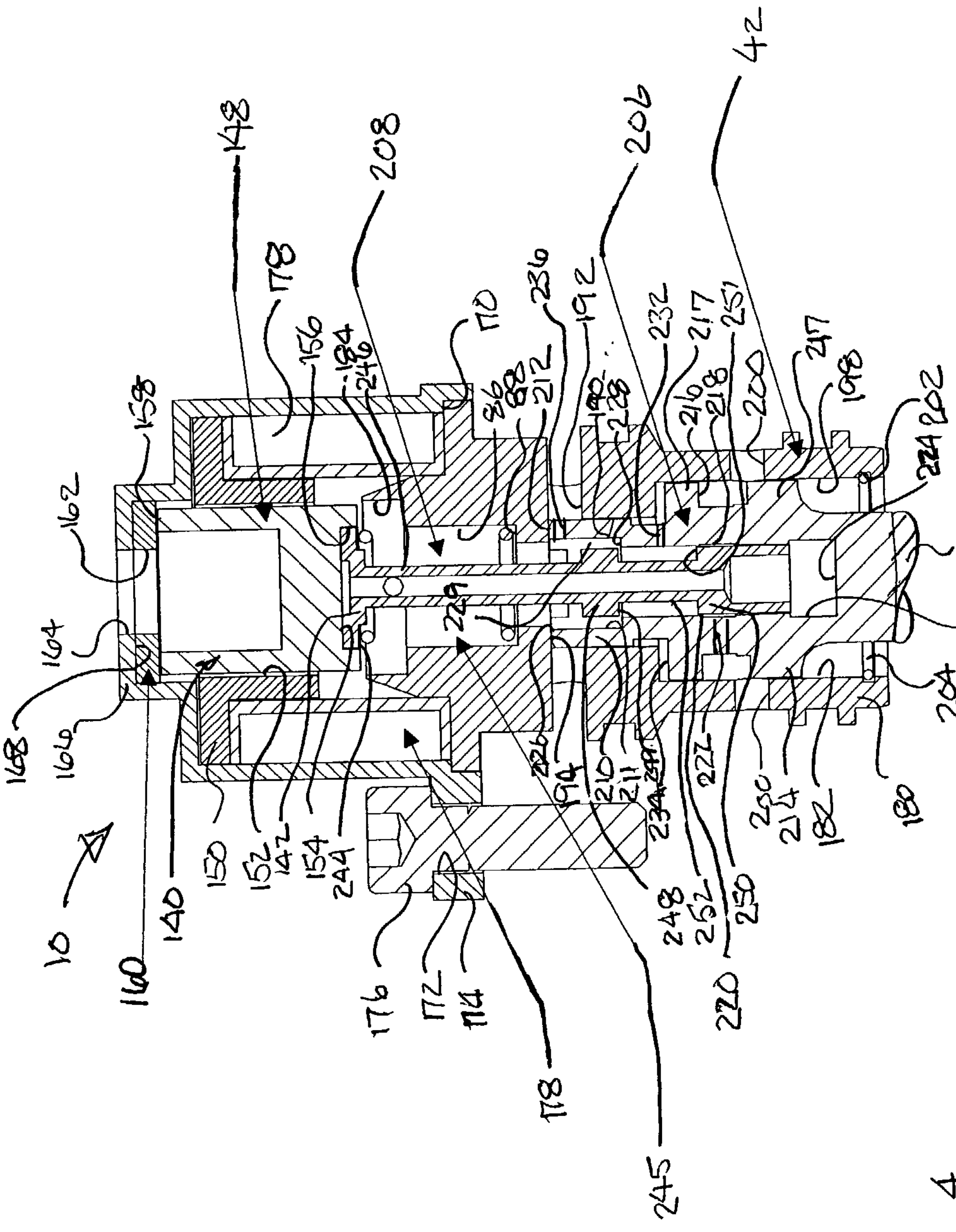


Figure 4

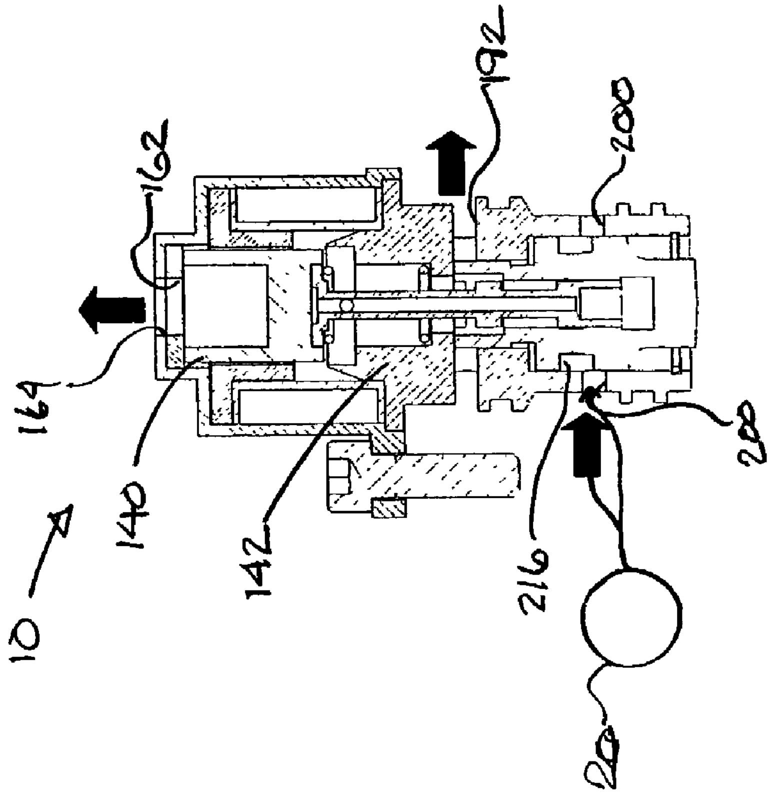


Figure 10

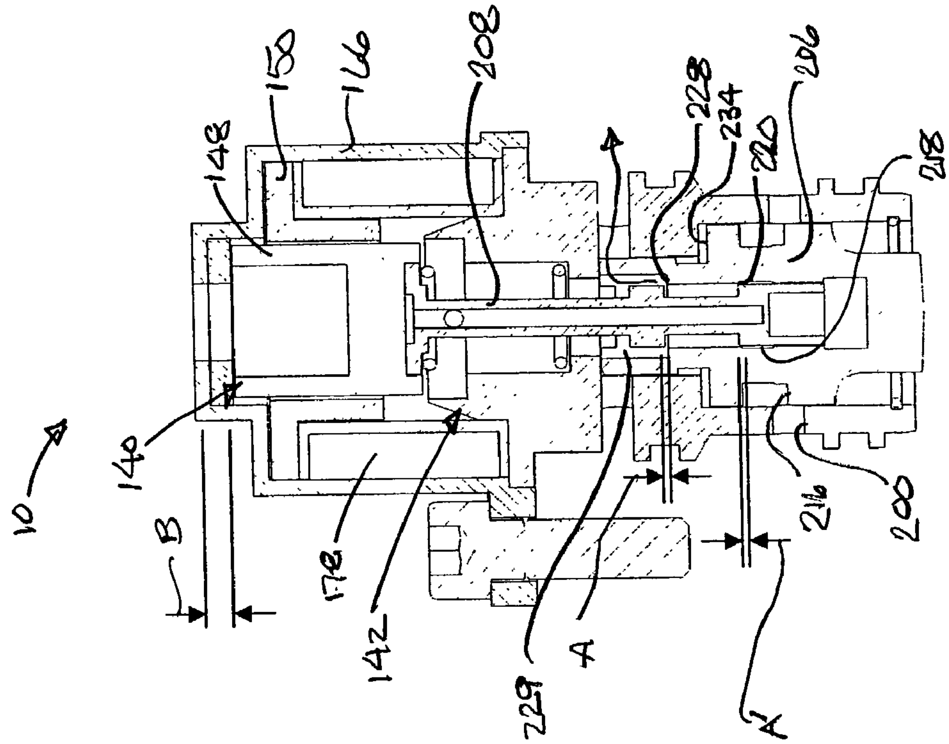
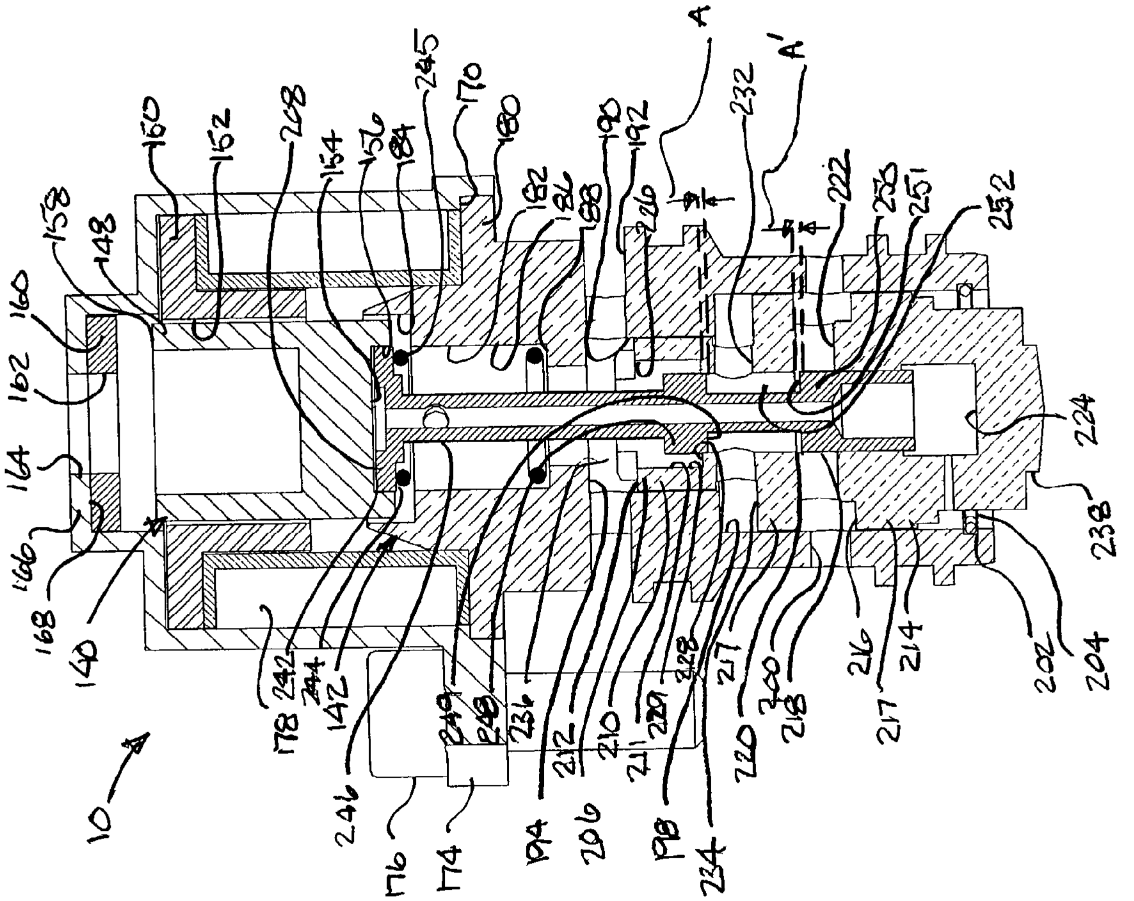

10
140
148
158
166
208
228
234
220
206
218
200
216
A
A'
B
172
142
229

Figure 6

Figure 7



HYDRAULICALLY ACTUATED, ELECTRICALLY CONTROLLED LINEAR MOTOR

RELATED APPLICATIONS

[0001] This application is a continuation-in-part of U.S. patent application Ser. No. 10/044,867, filed Jan. 10, 2002, which is a continuation-in-part of U.S. patent application Ser. No. 09/457,908, filed Dec. 8, 1999, now U.S. Pat. No. 6,338,320, which is a continuation-in-part of U. S. patent application Ser. No. 09/152,497, now U.S. Pat. No. 6,044, 815, all incorporated herein by reference.

TECHNICAL FIELD

[0002] The present application is related to motors. More particularly, the present application relates to a linear motor adaptable for linear actuation of diverse devices.

BACKGROUND OF THE INVENTION

[0003] There is a need for linear motors in many industries. This need is felt in the automotive industry in particular. There is increasing sophistication in microprocessor control of major automotive components in order to enhance vehicle control and safety. Many of these components have been previously controlled by cables and direct hydraulic actuation. With the shift to microprocessor control, there is a growing need for linear motors, particularly, hydraulically-actuated, electrically-controlled linear motors.

[0004] For example, it is usual to provide each vehicle wheel with a brake having a disk or drum shaped friction surface as well as a friction element and an actuator. The actuator presses the friction element against the friction surface when the brake is operated. In a motor vehicle, the service and parking brake system generally employ a common brake. Commonly, the parking brake system is operated manually, by hand or foot, and maintained in the operating condition by mechanical means, typically a cable or direct hydraulic control. There is a need in the industry for the more sophisticated control afforded by a linear motor.

[0005] Additionally, the service brake system may have brake slip control and automatic brake management for active brake operations such as traction control and driving dynamics control. Such traction control systems are commonly known as ABS, antilock brake system. Driving dynamics control devices have been marketed under a number of acronyms. There is a need in the industry for more sophisticated actuation of such brakes, such as could be afforded by a linear motor.

[0006] In another application, small internal combustion engines are an attractive choice for portable electric generators. These generators are commonly used to provide electric power in places without access to the national electric grid, and are particularly popular for use on construction sites, and recreational vehicles in remote areas, and during power outages.

[0007] One problem with the use of internal combustion engines in portable generators is that many electrical appliances require alternating current at almost exactly 60 hertz. Operating the IC engine to maintain 60 hertz requires that the speed of the IC engine be very accurately controlled. There is a need in the industry for sophisticated governance of such IC engines, and a linear motor could prove effective in providing such governance.

[0008] In another application, in compression ignition engines it is desirable to have both a positive power mode of operation and a braking mode of power operation. A highly effective way of operating an engine in braking mode is to cut off the fuel supply to the engine and then to open the exhaust valves in the engine near top dead center of the compression strokes of the engine cylinders. This causes air that the engine has compressed in its cylinders to escape to the exhaust system of the engine before the engine can recover the work of compressing the air during the subsequent "power" stroke of the engine piston. There is a need in the industry for a means of opening the exhaust valves to effect the compression release engine braking. The linear motor could be effective in opening the valves as described, either directly or through a rocker arm device.

[0009] In a further application, there has been a continual problem of how to successfully bring two rotatable gears selectively into and out of gear meshing engagement, such that both gears when engaged will then rotate together to do some form of useful work. One environment in which this problem exists is in the area of power take off devices (PTO). PTO's have found commercial uses in a wide variety of vehicular areas such as where the PTO is attached to the transmission of the vehicle for translating the power of the engine into the rotation of a shaft for doing useful work. Such work may be for example operating a hydraulic pump, compacting a garbage compactor, operating a separate attachable implement such as a snow blower, operating a separate non-attachable implement such as a grain auger, and the like. There is a need in the industry for a linear motor effective in engaging and disengaging the gears to effect the rotation of the PTO. A linear motor could be effective in making the gear engagement and disengagement as noted above.

SUMMARY OF THE INVENTION

[0010] The linear motor of the present invention substantially meets the aforementioned actuation needs set forth above. The mechanical components needed to effect the actuation by the linear motor are relatively simple, thereby minimizing the additional components required. The linear motor of the present invention is designed to provide for uniform actuation over a wide range of hydraulic fluid temperatures.

[0011] The foregoing advantages of the present invention are effected by the use of fine needle control. The fine needle control provides for modulation of the device being actuated as needed and provides for aggressive or less aggressive actuation as needed. Further, the device being actuated very closely follows the input of the linear motor. Therefore, the linear motor of the present invention does not require the added complexity requiring a sensor to measure the position of the device being the actuated for feedback control. The accurate control of the device being actuated is attained by accurate control of the needle at the end of needle stroke.

[0012] The present invention is a hydraulically actuated, electrically controlled, linear motor and includes a trigger that is electrically controlled. A hydraulic cartridge has a needle and a piston, the needle being operably coupled to the trigger and the piston being operably fluidly in communication with a source of high pressure actuation fluid, translation of the needle effected by the trigger acting to selectively port high pressure actuation fluid to the piston and to vent actuation fluid from the piston. The present invention is further a method of linear actuation.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013] **FIG. 1** is a sectional side perspective view of a further embodiment of a linear motor of the present invention and a device to be actuated (an engine valve);

[0014] **FIG. 2** is a sectional side perspective view of the embodiment of the linear motor of **FIG. 1** and engine valve, the engine valve being in the closed disposition;

[0015] **FIG. 3** is a sectional side perspective view of the embodiment of the linear motor of **FIG. 1** and engine valve, the engine valve being in the open disposition;

[0016] **FIG. 4** is a sectional view of a further embodiment of the linear motor of the present invention;

[0017] **FIG. 5** is a sectional view of the embodiment of **FIG. 4** depicting actuating fluid flow;

[0018] **FIG. 6** is a sectional view of the embodiment of **FIG. 4** depicting overlap and opening dimensions; and

[0019] **FIG. 7** is an enlarged sectional view of the embodiment of **FIG. 4**.

DETAILED DESCRIPTION OF THE DRAWINGS

[0020] The use of the linear motor in the application noted below is but one of a myriad of uses of the linear motor of the present invention. With this point in mind, the present invention is detailed below.

[0021] A preferred embodiment of the linear motor **10** is presented in FIGS. **1-3**. In this embodiment of the linear motor **10**, the linear motor is employed to act on a rocker arm **14** to effect the opening and closing of an engine valve **16**. Engine valve **16** has a valve stem **18**, a keeper **20** and a return spring **22**. The return spring **22** typically biases the engine valve **16** in the closed disposition and opposes the action of the linear motor **10**. Valve **16** has a valve longitudinal axis **23**.

[0022] The rocker arm **14** has an elongate arm member **64**. The arm member **64** is pivotable about a pivot point **68**. A first arm portion **70** extends leftward in the depiction of **FIG. 3** from the pivot point **68** to the proximal end **72** of the arm member **64**. A ball bearing **74** is disposed proximate the proximal end **72** for coupling to the drive piston **26**. The first arm portion **70** has a length L1, defined between the pivot point **68** and the point of contact of the ball bearing **74** with the piston **26**.

[0023] A second arm portion **76** of the arm member **64** extends rightward from the pivot point **68** to the distal end **78** of the arm member **64**. A bearing surface **80** is presented proximate the distal end **78**. The bearing surface **80** bears upon the stem upper margin **82** of the engine valve **16**. The second arm portion **76** has a length, L2, defined between the pivot point **68** and the point of contact of the bearing surface **80** with the stem upper margin **82**. L1 is preferably less than L2 to provide stroke amplification as discussed in more detail below.

[0024] The linear motor **10** has two major subcomponents: the actuator valve **24** and drive piston **26**. The actuator valve **24** is an elongate piston comprising a spool valve **28** at a first end and including a spool groove **29**. The actuator valve **24** is translatably disposed in part in spool bore **30**. A solenoid **32** is disposed proximate a second end of the actuator valve **24**. A return spring **33** bears on the end of the actuator valve **24** (top margin) that is disposed proximate the spool groove **29**.

[0025] An inlet port **34** is fluidly coupled to the source of high pressure actuating fluid **36**. This source may be a high pressure rail **36**. The rail **36** may convey any suitable high pressure fluid. Preferably, the fluid in the rail **36** is engine oil at approximately 450-3,500 psi. The rail **36** provides for hydraulic actuation of the linear motor **10**. The inlet port **34** is defined in the drive piston **26** and is in fluid communication with the spool bore **30** via the bore **49**, described in greater detail below. A vent port **38** is also in fluid communication with the spool bore **30** and is further fluidly coupled to a ambient reservoir **40**. The ambient reservoir **40** may be at substantially ambient pressure of 0 to 100 psi.

[0026] A controller **42** is in communication with the solenoid **32** for energizing and deenergizing the solenoid **32**. The controller **42** effects electric control of the linear motor **10**. The controller **42** may be a microprocessor that is in communication with a sensor or sensors for sensing parameters or other inputs, some of which may be manually inputted, necessary to exert control over the particular device to which linear actuation (in this case, valve **16**) is to be imparted by the linear motor **10**. For example, in an ABS application, the sensors might sense a certain difference in rotational speed of two or more vehicle wheels and the controller **42** would then command the linear motor coupled to at least on wheel brake to linearly actuate the brake with a single stroke or a modulated stroke to effect a single brake actuation or a pulsed brake actuation, as desired.

[0027] The second subcomponent of the actuator **12** is the drive piston **26**. The drive piston **26** includes a cylinder housing **46** having a cylinder **48** defined therein. The drive piston **26** is translatably disposed in the cylinder **48**. In the depiction of FIGS. **3-5**, the drive piston **26** is depicted as having two components, a cylinder capped by a cap. It should be noted that the drive piston **26** could as well be a single unitary component.

[0028] An axial central actuator piston bore **49** is defined in the drive piston **26**. The actuator piston bore **49** is an extension of the spool bore **30**. The actuator valve **24** projects into the actuator piston bore **49**, the actuator piston bore **49** accommodating, in cooperation with the spool bore **30**, the translation of the actuator valve **24**. Additionally, the return spring **33** is housed within the actuator piston bore **49**, bearing on the top margin of the actuator valve **24** and a spring stop **47** and being compressed therebetween.

[0029] A fluid chamber **50** is defined beneath the lower margin of the drive piston **26**. The fluid chamber **50** is selectively in communication with the inlet port **34** as a function of the disposition of the spool groove **29** relative to both the fluid chamber **50** and the inlet port **34**. The fluid chamber **50** is a variable volume chamber defined in part by the actuating surface **52**, the actuating surface **52** defining the lower margin of the drive piston **26**.

[0030] A bearing surface **54** is presented proximate the upper margin of the drive piston **26**. The bearing surface could be an elongate rod operably coupled to the device to which linear motion is to be applied by the linear motor.

[0031] The actuator valve **24** is electromagnetically actuated by a solenoid **32**. The actuator valve **24** is constrained to move linearly between a lower and an upper limit. Motion of the actuator valve **24** relative to the hydraulically actuated drive piston **26** sequentially opens and closes orifices (the groove **29** of spool **28**) that control hydraulic fluid in fluid chamber **50** acting on the actuating surface **52** of the drive piston **26**. In the closed disposition, depicted in **FIG. 2**, the vent port **38** and L.P. reservoir **40** are in fluid communication with fluid chamber **50**. In the open disposition, depicted in **FIG. 3**, the inlet port **34** is in fluid communication with the fluid chamber **50** by means of the spool groove **29**.

[0032] The function of this system is described below. The actuator valve **24** is actuated from rest (see **FIGS. 1 and 2**) to the open valve disposition (see **FIG. 3**) by the controller **42** applying voltage to the solenoid **32**. The actuator valve **24** then moves upward against its return spring **33** due to the magnetic force generated at the solenoid **32** responsive to the input signal from the controller **42**. Displacement of the actuator valve **24** relative to the drive piston **26** sequentially closes the vent **38** connected to tank **40** and opens the inlet port **34** that allows high-pressure fluid to flow from the rail **36** through the spool groove **29** into the actuating chamber **50**. The resulting hydraulic force acting on the actuating surface **52** displaces the drive piston **26** upward against the rocker arm **14**. It should be noted that actuator valve **24** moves upward in concert with the drive piston **26** after moving independently the small distance necessary to effect porting of the high pressure actuating fluid to the actuating chamber **50**.

[0033] Motion of the drive piston **26** displaces the rocker arm **14** about its pivot point **68**. The linear motion of the drive piston **26** is amplified and transmitted to the poppet valve **16** according to:

$$x_{poppetvalve} = x_{drivepiston}\left(\frac{L_2}{L_1}\right),$$

[0034] where L2>L1. See **FIGS. 2 and 3**.

[0035] At the appropriate time, dictated by engine performance and emissions constraints, the poppet valve **16** is returned to its seat as follows. Return motion of the actuator valve **24** to its initial position is initiated by the controller **42** removing the applied solenoid **32** voltage. The return spring **33** overcomes any residual magnetic force and returns the actuator valve **24** to its seat, as depicted in **FIG. 2**. The initial motion of the actuator valve **24** relative to the drive piston **26** sequentially closes the inlet port **34** connected to the high-pressure rail **36** and opens the vent orifice **38** connected to the tank **40**. Hydraulic pressure is thus removed from the drive piston **26**, which is then forced to return to its seat by the return spring **22** connected to the poppet valve **16**. A second return spring acting only on the drive piston could also be used where the device being actuated by the linear motor **10** does not have its own closing or return bias. Such a return spring could be disposed to act on the upper margin **55** of the drive piston **26**. It should be noted that actuator valve **24** moves downward in concert with the drive piston **26** after moving independently the small distance necessary to effect venting of the high pressure actuating fluid from the actuating chamber **50**. Further, a second solenoid could be utilized to oppose the solenoid **32** by generating an opposing linear motion of the drive piston **26** when actuated. Such second solenoid would be disposed external to the drive piston **26** in a manner similar to the disposition of the solenoid **32** relative to the actuator valve **24** and exert an electromagnetic force thereon when activated.

[0036] A constraint on the hydraulic surfaces of the second drive piston **26** is as follows:

$$F_{hydraulic} = F_{returnspring}\left(\frac{L_2}{L_1}\right)$$

[0037] In other words, the hydraulic force supplied to the drive piston **26** must overcome the "amplified" return spring force exerted by the return spring **22**. This force requirement may be accommodated by a larger area actuation surface **52** on the drive piston **26**, or by supplying higher pressure actuating fluid from the rail **36**.

[0038] A further preferred embodiment of the linear motor **10** is presented in FIGS. **4-77**. The first component of the linear motor **10** is the solenoid **140**. The solenoid **140** includes an armature **148** that is translatably disposed within an armature guide **150**. The armature guide **150** provides a cylindrical surface that is coincident with at least a portion of the outside margin of the armature **148** and constrains the armature **148** during translation thereof. Accordingly, the cylindrical surface of the armature guide **150** comprises a guide bore **152**.

[0039] A needle bearing surface **154** comprises a portion of the bottom margin of the armature **148**. The needle bearing surface **154** resides within a recess **156** defined in the bottom surface margin of the armature **148**.

[0040] The upper margin **158** of the armature **148** comprises an armature stop. The upper margin **158** is stopped by a shim **160** when the armature **148** is in the retracted disposition. The shim **160** has a selected depth dimension B, as will be described in greater detail below with reference to **FIG. 6**.

[0041] A hydraulic vent **162** is defined in the shim **160**. The hydraulic vent **162** is preferably in registry with a hydraulic vent **164** defined in the cover **166**. The underside margin **168** of the top of the cover **166** constrains the shim **160** between the cover **166** and the upper margin **158** of the armature **148**.

[0042] The cover **166** further has retaining groove **170** that is formed proximate the lower margin of the cover **166**. The retaining groove **170** bears on a peripheral margin of the cartridge **142**. A bore **172** is defined in a flange **174** that projects to the side of the cover **166**. A bolt **176** may be passed through the bore **172** and threaded into a threaded bore defined in the rail **20**. By such means, the cover **1166** secures both the solenoid **140** and the cartridge **142** in the integral receiver **130** defined in the rail **120** and holds the shim **160** in place.

[0043] The cover **166** additionally provides a retaining element for the coil **178** that is associated with the armature

148. The coil 178 is generally cylindrical in shape and resides outward of the armature 148. The armature guide 150 is preferably disposed between the coil 178 and the armature 148. Suitable electrical leads (not shown) couple the coil 178 to an external controller (not shown) for providing actuation commands to the linear motor 10.

[0044] The second component of the linear motor 10 is the hydraulic cartridge 142. The hydraulic cartridge 142 includes a cartridge body 180. An actuator bore 182 is centrally defined within the cartridge body 180. The actuator bore 82 extends all the way through the cartridge body 180 and has a number of varying diameters. Commencing at the top margin of the cartridge body 180, the first such diameter defines an armature receiver 184. When the armature 48 is in the actuated, extended disposition, the armature 148 translates downward from a retracted disposition to a extended disposition and is encompassed in part by the armature receiver 184.

[0045] The portion of the actuator bore 182 immediately below the armature receiver 184 has a lesser diameter than the armature receiver 184 and comprises a spring cage 186. A step formed at the bottom margin of the spring cage 186 comprises a spring seat 188.

[0046] A portion of the actuator bore 182 that comprises a piston neck receiver 190 is beneath the spring cage 186 and has a diameter that is greater than the step forming the spring seat 188. A hydraulic vent 192 extends radially outward from the piston neck receiver 190 and fluidly connects the piston neck receiver 190 to ambient conditions exterior to the hydraulic cartridge 142 (see in particular FIG. 5). A piston stop 194 is formed at the upper margin of the piston neck receiver 190. Preferably, the piston stop 194 is formed by the step that also forms the spring seat 188, the stop 194 being the lower margin of the step and the seat 188 being the upper margin of the step.

[0047] The greatest diameter of the actuator bore 182 comprises the lowermost portion of the actuator bore 182. This portion defines a power section receiver 198. A hydraulic inlet 200 extends through the cartridge body 180 and fluidly couples the power section receiver 198 to actuating fluid under pressure provided by the rail 20. See in particular FIG. 5. A retainer groove 202 is defined in the power section receiver 198 proximate the lower margin of the cartridge body 180. A retainer 204, preferably a snap ring, may be disposed in the retainer groove 202.

[0048] A piston 206 and a needle (sometimes referred to as a spool) 108 reside within the actuator bore 182 and are retained in place by the retainer 204.

[0049] The piston 206 is preferably a unitary device having a generally cylindrical outside margin of varying diameters. The piston 206 has, in descending order as depicted, a neck 210, a power section 214, and an actuator rod 238. A blind needle bore 211 is centrally defined within the piston 206 and extends downward to approximately the juncture of the power section 214 and the actuator rod 238. The needle bore 211, being blind, is closed at the bottom 224. The needle bore 211 is open at a top opening 226 formed by an upper margin 212.

[0050] The neck 210 of the piston 206 preferably translatably resides within the piston neck receiver 190 of the cartridge body 180. When the piston 206 is in the retracted, venting disposition, the upper margin 212 of the neck 210 is stopped by the piston stop 194 defined in the actuator bore 182 of the cartridge body 180.

[0051] The power section 214 of the piston 206 has an annular groove 216 defined between a pair of spaced apart lands 217. The annular groove 216 is preferably in registry with the hydraulic inlet 200 and, accordingly, is in fluid communication with actuating fluid provided by the rail 20. An annulus 218 is defined in the needle bore 211 substantially in registry with the annular groove 216. A fluid passageway 222 fluidly connects the annular groove 216 to the annulus 218. As noted in more detail below, the upper margin 220 of the annulus 218 becomes a critical interface in the operation of the piston 206 and needle 208.

[0052] A second critical interface of the piston 206 and needle 208 is the sealing shoulder 228. Sealing shoulder 228 is formed by a step increase in the diameter of the needle bore 211. The increase in diameter of the needle bore 211 forms an annulus 229 that is at least in part in registry with the hydraulic vent 192 defined in the cartridge body 180.

[0053] A high pressure fluid passageway 232 extends between the needle bore 211 and a piston head 234. A vent passageway 236 extends between the annulus 229 formed by the increased diameter of the needle bore 211 upward up the step 228 and the vent 192.

[0054] The actuator rod 238 depends from the power section 214. The distal end of the actuator rod 138 bears on the device to which linear motion is to be imparted.

[0055] The needle (spool) 208 is operably coupled to the armature 148 of the solenoid 140 and is translatably disposed within the needle bore 211 of the piston 206. The needle 208 has a head 242 that bears on the needle bearing surface 154 defined within the recess 156 of the armature 148. The underside margin of the head 242 comprises a spring seat 244. A coil return spring 245 is captured between the spring seat 244 and the spring seat 188 formed at the bottom of the spring cage 186 defined in the cartridge body 180. The return spring 245 is generally always in a state of compression and exerts an upward bias on the needle 208.

[0056] The return spring 245 is disposed concentric with a shank 246 that depends from the head 242. The shank 246 has a spool groove 252 defined between an upper land 248 and a lower land 250. As described in greater detail below, the positional interaction between the lower margin 249 of the upper land 248 and the sealing shoulder 228 and upper margin 251 of the lower land 250 with the upper margin 220 of the annulus 218 is critical to the operation of the piston 206 and needle 208.

[0057] When the trigger (solenoid 140) actuates the spool (needle 208) in the linear motor 10, the device to be actuated is also shifted linearly. The only delay in actuation of the device to be actuated between the time of actuation of the solenoid 140 and movement of the device to be actuated is the amount time the it takes the needle 208 to translate through the dimension A, A' noted in FIG. 6.

[0058] Prior to actuation of the solenoid 140, the piston 206 and needle 208 are in their retracted dispositions as indicated in FIG. 6. The incoming high-pressure actuation fluid is sealed off where the overlap is indicated by the dimension A'. The spool 252 is opened as indicated by the dimension A and actuating fluid is free to flow from the spool **252** upward through the vent passageway **236** and out the hydraulic vent **192**. See also **FIG. 5**.

[**0059**] Upon actuation of the solenoid **140**, the armature **148** is drawn downward by the magnetic force generated in the coil **178**, overcoming the bias of the return spring **245**. The armature **148** carries with it the needle **208**. The needle **208** translates downward relative to the piston **206**. Such motion closes the opening indicated by dimension A, thereby sealing off the venting of actuating fluid. Simultaneously, the overlap indicated by dimension A' is eliminated, thereby opening the spool **252** to the annulus **218** and causing the flow of high pressure actuating fluid into the spool **252**, as depicted in **FIG. 7**. The high pressure actuating fluid flows radially outward through high-pressure fluid passageway **232** to bear downward on the piston head **234**.

[**0060**] The force generated by the high pressure actuating fluid acting on the piston head **234** drives the piston **206** downward (in conjunction with the continued downward travel of the spool **208**). This downward translation of the piston **206** exerts a downward pressure and linear translational motion on the device to be actuated.

[**0061**] Retraction or closing of the device to be actuated occurs when the actuation command to the solenoid **140** is withdrawn. The magnetic field collapses and the return spring **245** shifts the needle **208** upward relative to the piston **208** to the retracted venting disposition depicted in **FIG. 4**. The accumulator **268** of rail **20** is sealed off from the hydraulic cartridge **142** by the overlap A' of **FIG. 4**. Hydraulic fluid in the hydraulic cartridge **142** escapes through opening A and out hydraulic vent **192**, as depicted in **FIG. 5**.

[**0062**] It will be obvious to those skilled in the art that other embodiments in addition to the ones described herein are indicated to be within the scope and breadth of the present application. Accordingly, the applicant intends to be limited only by the claims appended hereto.

1. A hydraulically-assisted linear motor for imparting linear actuation to a device, comprising:

an actuator piston being operably coupled to the device; and

a translatable needle valve being in fluid communication with a source of hydraulic fluid under pressure and with the actuator piston and further being operably coupled to a needle positioning mechanism, the needle valve being translatable at a desired rate and effecting the metering of hydraulic fluid under pressure to and from the actuator piston in response to translational control inputs from the needle positioning mechanism, the metered hydraulic fluid at least partially effecting translational motion of the actuator piston to follow the translation of the needle valve to effect a desired profile of linear translational motion of the device.

2. The hydraulically-assisted linear motor of claim 1 wherein the needle valve has a generally elongate cylindrical shape and has a first end defining a first end groove and a second end opposed thereto, the second end being operably coupled to the needle positioning mechanism.

3. The hydraulically-assisted linear motor of claim 1 wherein the needle positioning mechanism is selected from mechanisms consisting of a solenoid and a stepper motor.

4. The hydraulically-assisted linear motor of claim 1 wherein the actuator piston has a generally cylindrical shape and has a first end operably coupled to the device and a second end opposed thereto, an axial bore being defined in the actuator piston extending from the second end at least a portion of a longitudinal dimension of the actuator piston.

5. The hydraulically-assisted linear motor of claim 1 further including an actuator housing, the actuator housing having an axial housing cylinder bore defined therein, the actuator piston having a piston head, the piston head being translatably disposed in the housing cylinder bore.

6. The hydraulically-assisted linear motor of claim 5 wherein the actuator housing is fluidly coupled to a source of relatively high pressure hydraulic fluid.

7. The hydraulically-assisted linear motor of claim 5, wherein the needle valve has a generally elongate cylindrical shape and has a first end defining a first end groove and a second end opposed thereto, the second end being operably coupled to the needle positioning mechanism, the needle valve being translatably disposed in an axial bore defined in the actuator piston.

8. The hydraulically-assisted linear motor of claim 7 wherein the needle valve first end groove acts to meter hydraulic fluid to and from the actuator piston head responsive to translation of the needle valve relative to the actuator piston.

9. A method of linear actuation of a device, comprising:

translating an actuator valve responsive to control inputs by a pilot valve positioning system;

metering hydraulic fluid under pressure to and from a servo piston by means of translation of the pilot valve relative to the servo piston;

operably coupling the servo piston to the device; and

translating the servo piston and the device by means of a force exerted on the servo piston by the hydraulic fluid under pressure, the hydraulic fluid under pressure causing the servo piston to closely follow the translation of the pilot valve to effect a desired profile of linear translational motion of the device.

10. The method of claim 9 including biasing the servo piston by a return spring, the force exerted on the servo piston by the hydraulic fluid under pressure acting in opposition to the bias exerted by the return spring.

11. The method of claim 9, including actuating the actuator valve by at least one solenoid.

12. The method of claim 9, including actuating the actuator valve by a first solenoid and an opposed spring.

13. The method of claim 9, including shifting a spool valve to selectively port actuating fluid to the servo piston and to selectively vent actuating fluid from the servo piston.

14. A hydraulically actuated, electrically controlled linear motor, comprising:

a trigger being electrically controllable;

a hydraulic cartridge having a needle and a piston, the needle being operably coupled to the trigger and the piston being operably fluidly in communication with a source of high pressure actuation fluid, translation of the needle effected by the trigger acting to selectively port high pressure actuation fluid to the piston and to vent actuation fluid from the piston for effecting linear translation of the piston.

15. The linear motor of claim 14, the needle having a spool, the spool selectively being in fluid communication with the source of high pressure actuation fluid and being vented by translatory motion of the needle relative to the piston.

16. The linear motor of claim 14 being an open loop system.

17. A linear motor for actuating a device, comprising:

an electrohydraulic actuator having a piston, the piston being operably couplable to the device and being translatable responsive to an actuating fluid bearing on a piston surface, the piston surface being in fluid communication with an actuator valve, the actuator valve being in selective fluid communication with a source of actuating fluid under pressure, the actuator valve being selectively shiftable relative to the piston to selectively port and vent actuating fluid to and from the piston surface to effect at least in part linear translatory motion of the piston and the device as desired.

18. The linear motor of claim 17, a return spring being operably coupled to the actuator valve and exerting a bias on the actuator valve.

19. The linear motor of claim 17, the actuator valve being actuated by at least one solenoid.

20. The linear motor of claim 19, the actuator valve being actuated by a first solenoid and a second opposed solenoid.

21. The linear motor of claim 20, the actuator valve being actuated by a first solenoid and by a bias exerted by an opposed spring.

22. The linear motor of claim 17, the source of actuating fluid under pressure being a high pressure rail.

23. The linear motor of claim 17, the actuating fluid being engine lubricating oil.

24. The linear motor of claim 17, the actuator valve being a spool valve translatably disposed at least in part in a bore defined in the piston.

25. The linear motor of claim 17, the actuator valve being in selective fluid communication with a reservoir at substantially ambient pressure.

26. A linear motor for actuating a device, comprising:

a hydraulically actuated, electrically controlled servomechanism having an actuator valve and a drive piston, motion of the actuator valve relative to the drive piston acting to open and close certain orifices for controlling fluid acting on the drive piston, the drive piston being operably coupled to the device for imparting linear motion in at least one direction to the device.

27. The linear motor of claim 26, a return spring being operably coupled to the actuator valve and exerting a bias on the actuator valve.

28. The linear motor of claim 26, the actuator valve being actuated by at least one solenoid.

29. The linear motor of claim 28, the actuator valve being actuated by a first solenoid and an opposed spring.

30. The linear motor of claim 26, the actuator valve including a spool valve, the spool valve being shiftable to selectively port actuating fluid to the drive piston and to selectively vent actuating fluid from the drive piston.

31. The linear motor of claim 26, the source of actuating fluid under pressure being a high pressure rail.

32. The linear motor of claim 26, the actuator valve being a spool valve translatably disposed at least in part in a bore defined in the piston.

33. The linear motor of claim 26, the actuator valve being translatable through a known stroke, a first portion of the stroke being relative to the drive piston and a second portion of the stroke being in concert with the translation of the drive piston.

34. The linear motor of claim 33, the first portion of the stroke of the actuator valve selectively effecting porting actuating fluid to the power piston and venting actuating fluid from the power piston.

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