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Lou

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(54) **VARIABLE VALVE ACTUATOR WITH A PNEUMATIC BOOSTER**

(75) Inventor: **Zheng Lou**, Plymouth, MI (US)

(73) Assignee: **LGD Technology, LLC**, Plymouth, MI (US)

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F01L 1/34 (2006.01)

(52) **U.S. Cl.** **123/90.16**; 123/90.15; 123/90.12

(58) **Field of Classification Search** 123/90.11, 123/90.12, 90.16, 90.15; 137/906; 251/25; 91/356, 392, 508
See application file for complete search history.

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Schuderi Group, LLC Plaintiff v. *LGD Technology, LLC and Zheng* (*David*) *Lou, Ph.D., Defendents,* Answer to Complaint, Affirmative Defenses, Counterclaims and Jury Demand.

Schuderi Group, LLC Plaintiff v. *LGD Technology, LLC and Zheng*(*David*) *Lou, Ph.D., Defendents,* Answer to Defendent's Counterclaim and Affirmative Defenses.

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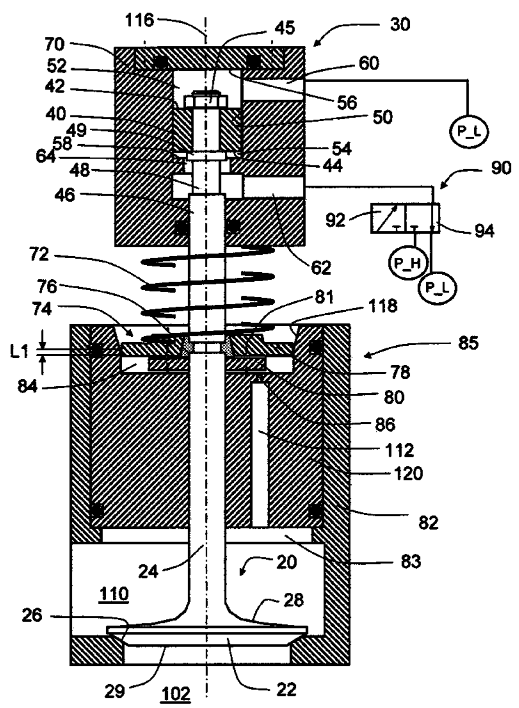
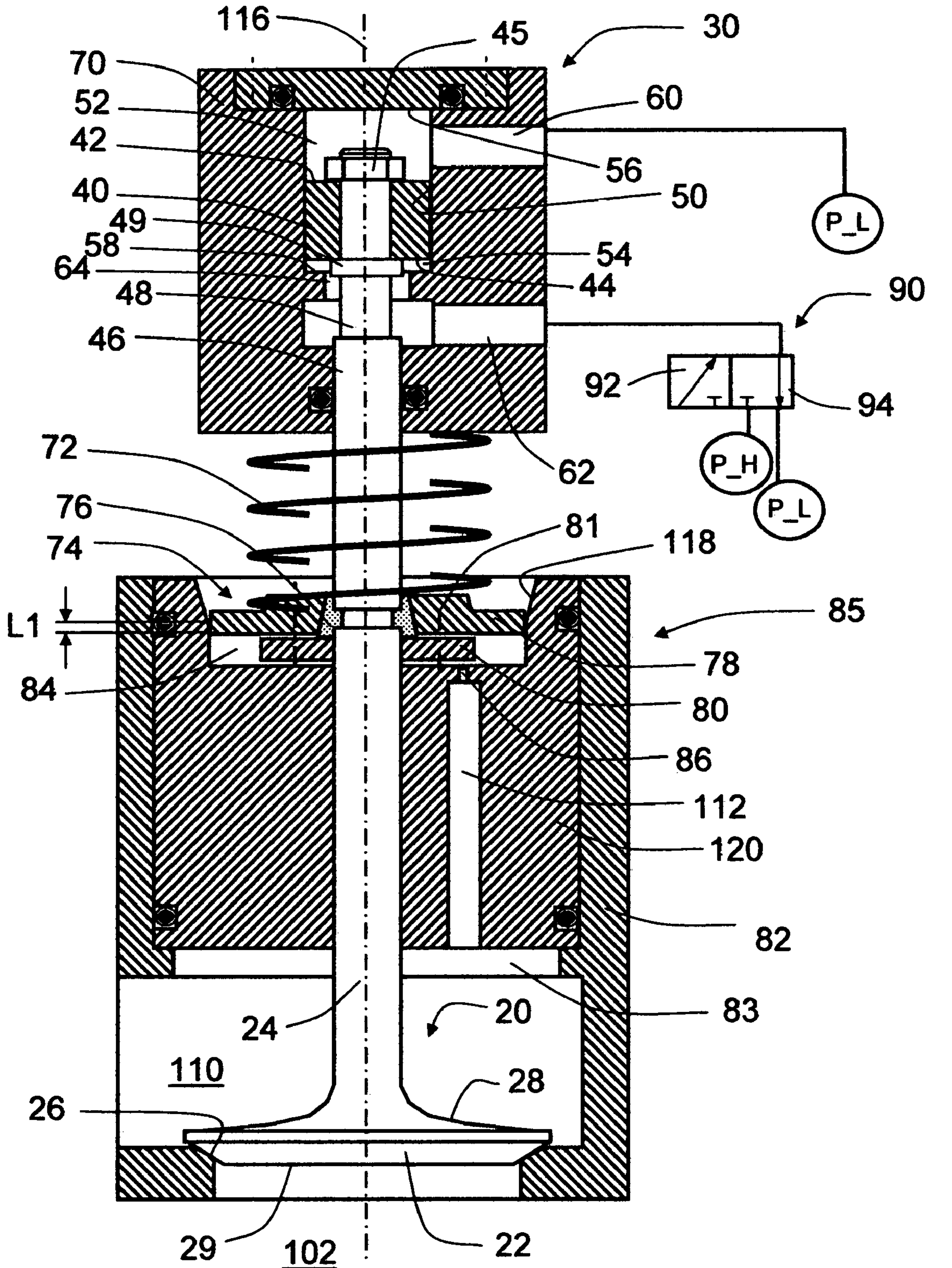
Primary Examiner—Zelalem Eshete

(57) **ABSTRACT**

Actuators, and corresponding methods and systems for controlling such actuators, provide independent valve control with a large initial or opening force. In an exemplary embodiment, an actuator includes a driver further including a housing defining a longitudinal axis and first and second directions, an actuation mechanism capable of generating actuation force at least in the first direction, and a rod with one end operably connected with at least one part of the actuation mechanism and with the other end available for an operable connection with a load such as an engine valve; at least one return spring operably connected with the rod through a spring retainer assembly and biasing the rod in the second direction; and a pneumatic booster further including a pneumatic cylinder, a pneumatic piston operably connected with the rod through the spring retainer assembly and biasing the rod in the first direction, a charge mechanism providing a controlled fluid communication between the pneumatic cylinder and a high-pressure gas source, and a bleed mechanism providing a controlled fluid communication between the pneumatic cylinder to a low-pressure gas sink.

15 Claims, 5 Drawing Sheets





116
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FIGURE 1

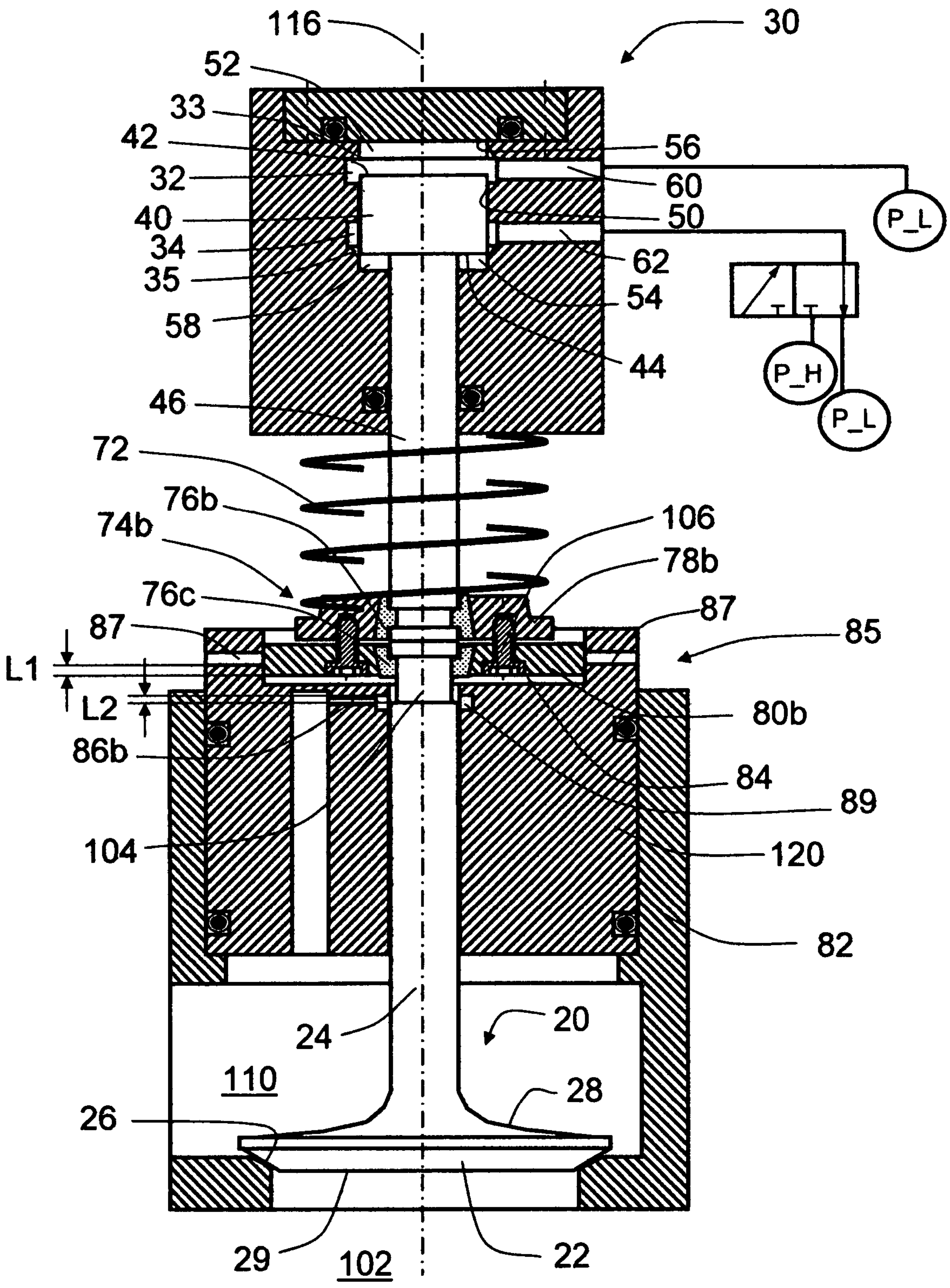
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FIGURE 2

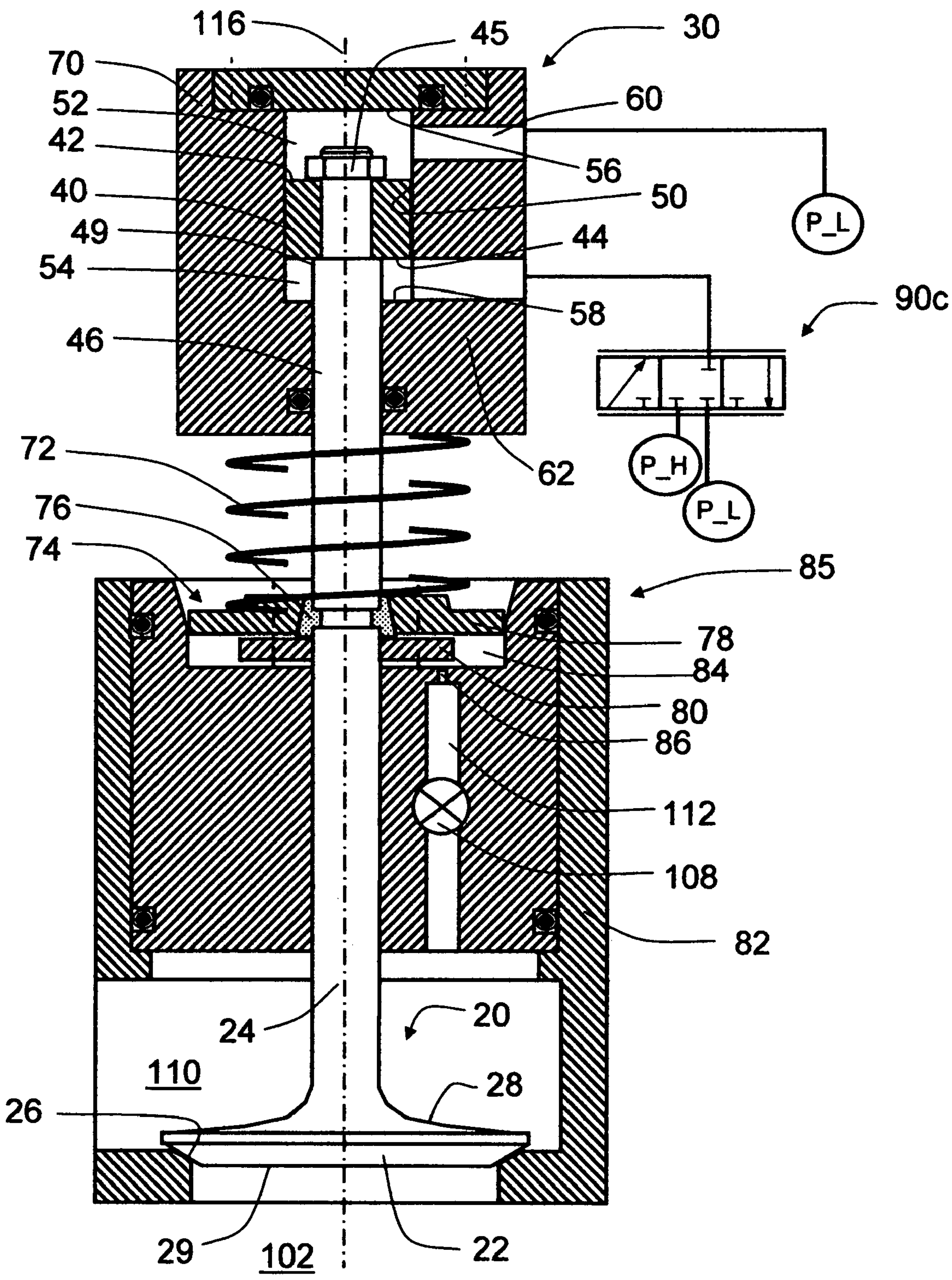
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FIGURE 3

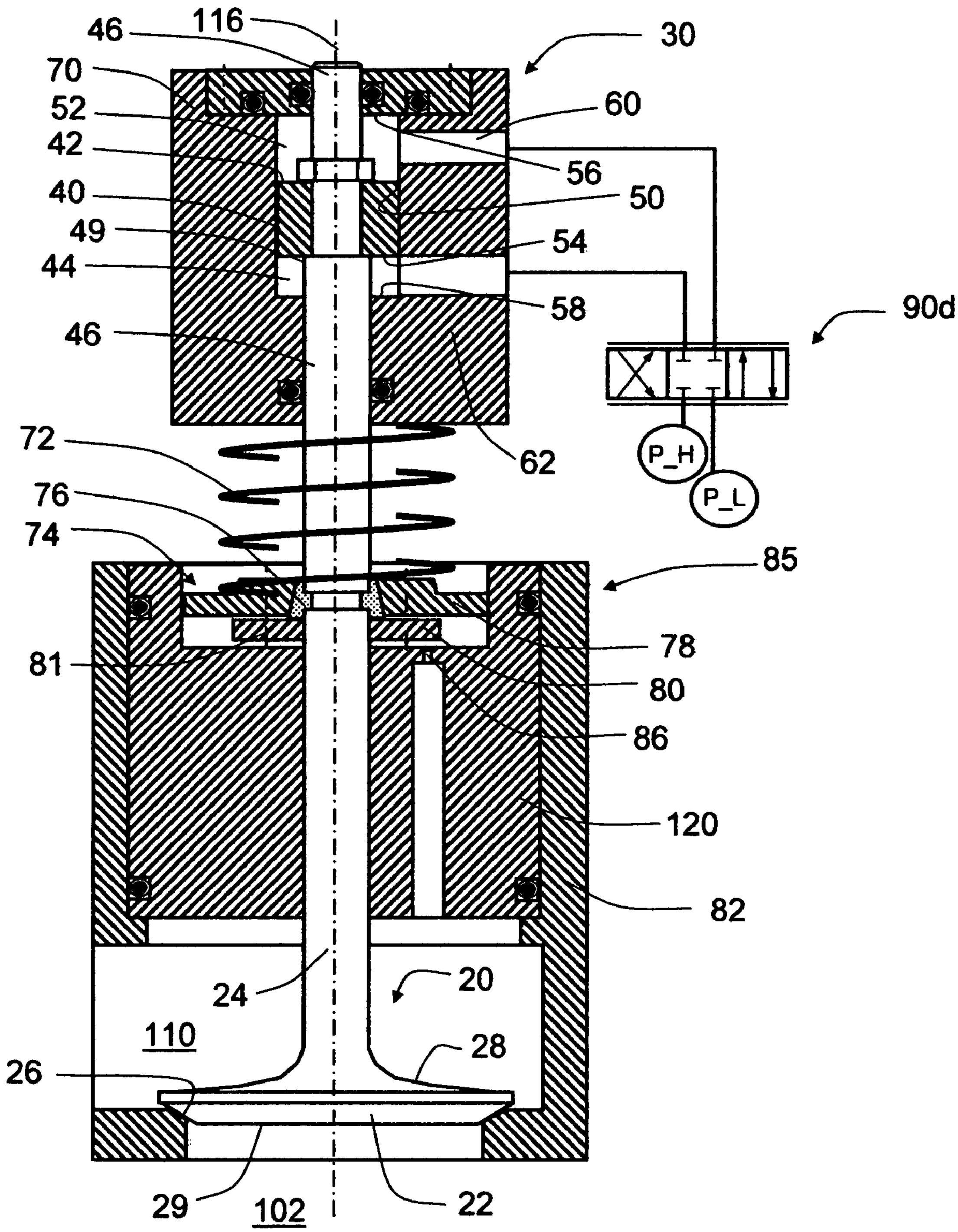
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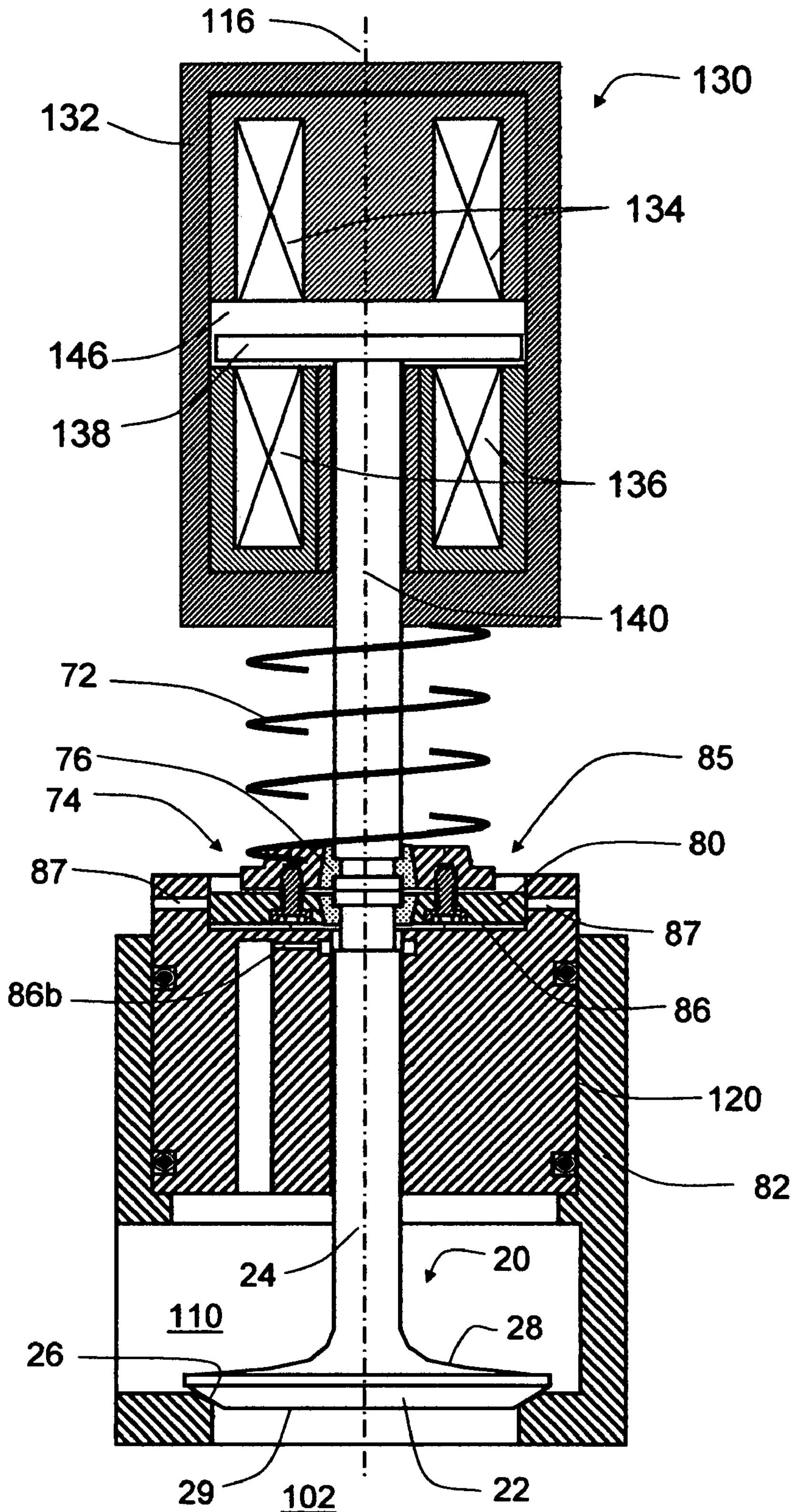
FIGURE 4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102
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FIGURE 5

VARIABLE VALVE ACTUATOR WITH A PNEUMATIC BOOSTER

FIELD OF THE INVENTION

This invention relates generally to actuators and corresponding methods and systems for controlling such actuators, and in particular, to actuators offering efficient, fast, flexible control with large opening forces.

BACKGROUND OF THE INVENTION

A split four-stroke cycle internal combustion engine is described in U.S. Pat. No. 6,543,225. It includes at least one power piston and a corresponding first or power cylinder, and at least one compression piston and a corresponding second or compression cylinder. The power piston reciprocates through a power stroke and an exhaust stroke of a four-stroke cycle, while the compression piston reciprocates through an intake stroke and a compression stroke. A pressure chamber or cross-over passage interconnects the compression and power cylinders, with an inlet check valve providing substantially one-way gas flow from the compression cylinder to the cross-over passage, and an outlet or cross-over valve providing gas flow communication between the cross-over passage and the power cylinder. The engine further includes an intake and an exhaust valve on the compression and power cylinders, respectively. The split-cycle engine according to the referenced patent and other related developments potentially offers many advantages in fuel efficiency, especially when integrated with an additional air storage tank interconnected with the cross-over passage, which makes it possible to operate the engine as an air hybrid engine. Relative to an electrical hybrid engine, an air hybrid engine can potentially offer as much, if not more, fuel economy benefits at much lower manufacturing and waste disposal costs.

To achieve the potential benefits, the air or air-fuel mixture in the cross-over passage has to be maintained at a predetermined firing condition pressure, e.g. approximately 270 psi or 18.6 bar gage-pressure, for the entire four stroke cycle. The pressure may go much higher to achieve better combustion efficiency. Also, the opening window of the cross-over valve has to be extremely narrow, especially at medium and high engine speeds. The cross-over valve opens when the power piston is at or near the top dead center (TDC) and closes shortly after that. The total opening window in a split cycle engine may be as short as one to two milliseconds, compared with a minimum period of six to eight milliseconds in a conventional engine. To seal against a persistently high pressure in the cross-over passage, a practical cross-over valve is most likely a poppet or disk valve with an outward (i.e. away from the power cylinder, instead of into it) opening motion. When closed, the valve disk or head is pressured against the valve seat under the cross-over passage pressure. To open the valve, an actuator has to provide an extremely large initial opening force to overcome the pressure force on the head as well as the inertia. The pressure force drops dramatically once the cross-over valve is open because of a substantial pressure-equalization between the cross-over passage and the power cylinder. Once the combustion is initiated, the valve should be closed as soon as desired to prevent the spread of the combustion into the cross-over passage, which also entails, during a certain period of combustion, a need to keep the valve seated against a power cylinder pressure that is higher than the cross-over passage pressure. In addition, the cross-over valve needs to be deactivated when the power stroke is not active in certain phases of the air hybrid operation. Like conventional engine valves, the seating velocity of the cross-over valve has to be kept under a certain limit to reduce noise and maintain adequate durability.

In summary, a cross-over valve actuator has to offer a large initial opening force, a substantial seating force, a reasonably low seating velocity, a high actuation speed, and timing flexibility while consuming minimum energy by itself. Most, if not all, conventional engine valve actuation systems are not able to meet these demands.

SUMMARY OF THE INVENTION

Briefly stated, in one aspect of the invention, one preferred embodiment of an actuator includes a driver further including a housing defining a longitudinal axis and first and second directions, an actuation mechanism capable of generating actuation force at least in the first direction, and a rod with one end operably connected with at least one part of the actuation mechanism and with the other end available for an operable connection with a load such as an engine valve; at least one return spring operably connected with the rod through a spring retainer assembly and biasing the rod in the second direction; and a pneumatic booster further including a pneumatic cylinder, a pneumatic piston operably connected with the rod through the spring retainer assembly and biasing the rod in the first direction, a charge mechanism providing a controlled fluid communication between the pneumatic cylinder and a high-pressure gas source, and a bleed mechanism providing a controlled fluid communication between the pneumatic cylinder and a low-pressure gas sink.

In operation, the actuator holds the load to a second-direction end position with the force from the at-least-one return spring biasing in the second direction and overcoming the sum of the rest of the forces including those from the pneumatic booster and the load, without generating the actuation force in the first direction from the actuation mechanism, and with the pneumatic booster being charged through the charge mechanism to yield a substantial force in the first direction to oppose a substantial load force in the second direction.

The actuator initiates the travel of the load in the first direction by generating the actuation force in the first direction from the actuation mechanism, with the combination of the actuation force and the force from the pneumatic booster being able to overcome the sum of the rest of the forces including those from the at-least-one return spring and the load and accelerate the load in the first direction.

The actuator keeps the travel in the first direction with the actuation force in the first direction until reaching the target stroke, and keeping the actuation force in the first direction if the load needs to be held at the target stroke. The actuator initiates the return travel of the load in the second direction at least by turning off the actuation force in the first direction so that the load is accelerated in the second direction at least by the return spring.

The actuator bleeds off excess air in the booster cylinder through the bleed mechanism during at least part of the time period described in the above paragraph to reduce the force from the pneumatic booster, which is otherwise too excessively resistant to the return travel of the load. It completes the return travel with a decreasing force from the return spring and an increasing force from the pneumatic booster, which help slow down the load.

In another embodiment, the driver is a fluid driver, the actuation mechanism comprising an actuation piston, an actuation cylinder, first and second fluid spaces in fluid communication with first and second ports, respectively; and the rod being a piston rod operably connected with the actuation piston and the load.

In another embodiment, the driver is an electromagnetic driver; the actuation mechanism comprising an armature disposed in an armature chamber, and at least a first electromagnet on the first direction side of the armature chamber, whereby being able to pull the armature in the first direction when energized; and the rod being an armature rod operably connected with the armature and the load.

In another embodiment, the charge mechanism includes a charge orifice, whereby substantially restricting the charge flow rate. It may also includes a control mechanism that substantially closes off charge flow at least when the bleed mechanism is actively bleeding off excess air.

The present invention provides significant advantages over the prevailing fluid actuators and their control, especially those needed for the cross-over passage engine valve that needs a large initial opening force, a substantial seating force, a reasonably low seating velocity, a high actuation speed, and timing flexibility while consuming minimum energy by itself. The pneumatic booster is able to provide that large initial force, without adding too much construction complexity or demanding too much energy consumption or stretching the capacity and functional limits of the fluid or electromagnetic actuators, by tapping directly into the cross-over passage or the air storage tank. With the charge mechanism, the boost force can be directly adjusted to the varying operating pressure in the cross-over passage, without sophisticated active control. With the bleed mechanism, the engine valve return force can be greatly reduced by making the boost force to be substantially lower during the return stroke.

With the pneumatic booster, the driver, be it a fluid or electromagnetic one, is able to concentrate on more or less conventional valve actuation, without the design, function and cost burden associated with the large initial opening force, which conventionally entails large flow rate and package size for fluid drivers and high, if not impossible, magnetic force and electrical power for electromagnetic drivers.

The present invention, together with further objects and advantages, will be best understood by reference to the following detailed description taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic illustration of one preferred embodiment of the engine valve actuator, which is at a closed state;

FIG. **2** is a schematic illustration of another preferred embodiment, which includes design variations in the fluid driver, the spring retainer assembly, and the pneumatic booster;

FIG. **3** is a schematic illustration of another preferred embodiment, which includes a 3-way proportional valve and a charge valve;

FIG. **4** is a schematic illustration of another preferred embodiment, which includes a 4-way proportional valve, a fluid driver with a double-ended piston rod, and a pneumatic booster without a bleed mechanism; and

FIG. **5** is a schematic illustration of another preferred embodiment, which includes an electromagnetic driver.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to FIG. **1**, a preferred embodiment of the invention provides an actuator including a fluid driver **30**, an actuation 3-way valve **90**, an return spring **72**, and a pneumatic booster **85**. The load or control target of the actuator is an engine valve **20**.

The actuation 3-way valve **90** supplies the fluid driver **30** through a second port **62** of the fluid driver **30**. The 3-way valve **90** has two of its three ways connected with a low-pressure P_L fluid line and a high-pressure P_H fluid line, and the third way connected with the second port **62**. A first port **60** of the fluid driver **30** is in fluid communication directly with the low-pressure P_L fluid line.

The actuation 3-way valve **90** is switched either to a left position **92** or a right position **94**. At the left and right positions **92** and **94**, the second port **62** is in fluid communication with the P_H and P_L lines, respectively.

The pressure P_H can be either constant or continuously variable. When variable, it is to accommodate variability in system friction, engine valve opening, air pressure, the engine valve seating velocity requirement, etc., and/or to save operating energy when possible. The pressure P_L can be simply the fluid tank pressure, the atmosphere pressure, or a fluid system backup pressure. The fluid system backup pressure can be simply supported or controlled, for example, by a spring-loaded check valve, with or without an accumulator. The P_L value is preferred to be as low as possible to increase the system efficiency, and yet high enough to help prevent fluid cavitation. When necessary, The P_L can be more tightly controlled as well. When necessary and/or allowed, the two P_L lines connected with the two ports **60** and **62** may maintain two pressure values. For example, the first port **60** may be simply used to dump some leakage flow to the fluid tank (not shown in FIG. **1**). In this case, much of the first fluid space may be simply filled with air, instead of the working fluid (assuming the working fluid is not air).

The engine valve **20** includes an engine valve head **22** and an engine valve stem **24**. The engine-valve head **22** includes a first surface **28** and a second surface **29**, which in the case of a split-cycle engine, are exposed to a cross-over passage **110** and the engine cylinder **102**, respectively. The engine valve **20** is operably connected with the fluid driver **30** along a longitudinal axis **116** through the engine valve stem **24**, which is slideably disposed in an engine valve guide **120**. For ease of description, the assembly and the longitudinal axis **116** have first and second directions, which are the same as the top and bottom directions, respectively, in FIG. **1**. The engine valve guide **120** as illustrated in FIG. **1** does not look like a traditional engine valve guide, which normally is a sleeve with a much limited wall thickness. The guide **120** is designed to be situated in the cylinder head **82**, over a valve assembly opening **83**, which is large enough to slide through the engine valve head **22** during assembly. This is just one of many potential assembly options. This does not exclude the possibility of adding a traditional-looking sleeve inside the guide **120**. The guide **120** may contain necessary engine coolant and lubricant passages (not shown in FIG. **1**).

When the engine valve **20** is fully closed, the engine valve head **22** is in contact with an engine valve seat **26**, sealing off the fluid communication between the cross-over passage **110** and the engine cylinder **102**.

The fluid driver **30** comprises an actuator housing **70**, an actuation piston **40**, and an actuation cylinder **50**. The actuation piston **40** is slideably disposed in the actuation cylinder **50**. The actuation piston **40** is fixed on to a piston rod **46** between a fastening element **45** and a shoulder **49**. The actuation piston **40** includes a first surface **42** and a second surface **44**, and longitudinally divides the actuation cylinder **50** into a first fluid space **52** (between an actuation-cylinder first end **56** and an actuation-piston first surface **42**) and a second fluid space **54** (between the actuation-piston second surface **44** and the actuation-cylinder second end **58**). The radial clearances around the actuation piston **40** and the piston rod **46** are substantially tight, provide substantial fluid seal, and yet offer tolerable resistance to relative motions.

The second fluid space **54** is in fluid communication with the second port **62** through a second flow passage **64** around a neck feature **48** on the piston rod. The second flow passage **64** becomes substantially more restrictive when the actuation piston **40** is close to the actuation-cylinder second end **58**, with the shoulder **49** longitudinally approaching and/or overlapping the second flow passage **64**. If a second flow mechanism is defined to include the second flow passage **64**, the neck **48**, and the shoulder **49**, then the second flow mechanism provides substantially open fluid communication between the second; fluid space and the second port. It provides a snubbing function when the actuation piston **40** is close to the actuation-cylinder second end **58**. When desired, the second flow mechanism may also include a one-way or check valve (not shown in FIG. **1**), providing a parallel, substantially-open fluid communication from the second port **62** to the second fluid space **54**.

The first fluid space **52** is in fluid communication with the first port **60** without much flow restriction.

The piston rod **46** is operably connected with the engine valve stem **24**, and in this embodiment (as illustrated in FIG. **1**) the rod **46** and stem **24** are structurally the same part, which is not the only design option.

A spring retainer assembly **74** is designed to help hold the return spring **72** and transfer its force on to the engine valve stem **24**. The return spring **72** as illustrated in FIG. **1** is a single mechanical compression spring. This does not exclude other design options, such as a pair of compression springs in parallel. The spring **72** may also be in the form of the Belleville type or pneumatic nature.

The spring retainer assembly **74** includes a first and second spring retainers **78** and **80** and a set of valve keepers **76**. The first spring retainer **78** also functions or doubles as a pneumatic piston, which is slideably disposed inside a pneumatic cylinder **84**, a cavity at the top of the engine valve guide **120**, to form the pneumatic booster **85**. The side, sliding walls of the first spring retainer **78** and the pneumatic cylinder **84** maintain an air-tight seal and yet reasonable level of friction with necessary lubrication and sealing mechanism (details not in FIG. **1**). The return spring **72** and the pneumatic booster **85** apply forces to the first retainer **78**, and thus the engine valve stem **24**, in the second and first directions, respectively. The spring retainer assembly **74** is thus designed to sustain forces in both directions. The force from the return spring **72** is applied to the first spring retainer **78**, and is transferred, through the valve keepers **76**, to the engine valve stem **24**. The pneumatic force from the pneumatic cylinder **84** is primarily applied to the first spring retainer **78**, and is transferred to the valve stem **24** through spring-retainer fastening means **81** (details of which are not illustrated in FIG. **1**), the second spring retainer **80**, and the valve keepers **76**.

The pneumatic cylinder **84** is charged or supplied with the pressurized gas or air from the cross-over passage **110**, a high-pressure gas source, through a charge mechanism including a charge passage **112** and a charge orifice **86**. The charge orifice **86** is designed to be more restrictive than the charge passage **112**. The passage **112** and orifice **86** may be combined into a single restrictive long orifice (not shown in FIG. **1**). The separate construction or existence of the charge orifice **86** may ease the manufacturing process. The pneumatic cylinder **84** is also intentionally designed to have an expansion **118** in its top portion so that a substantially air-tight seal between the first retainer **78** and the pneumatic cylinder **84** is kept only when the engine valve **20** is seated and within a predefined distance L**1** of the engine valve travel in the first direction, beyond which there is a substantial clearance or bleed passage between the pneumatic cylinder **84** and the first retainer **78**, and the pneumatic cylinder **84** is in substantial fluid communication with the atmosphere or a lower-pressure gas sink and yet is in a restrictive fluid communication with the cross-over passage **110**.

The actuation cylinder **50** offers substantial room longitudinally such that the actuation piston **40** does not touch the first and second ends **56** and **58** of the cylinder **50** when the load or engine valve **20** is at its first-direction and second-direction end positions, respectively. When the engine valve **20** is seated or at its second-direction end position as shown in FIG. **1**, there is still a distance between the actuation-piston second surface **44** and the actuation-cylinder second end **58** to accommodate the engine valve lash adjustment. When the engine valve **20** is fully open or at its first-direction end position, there is enough force from the return spring **72** and/or enough longitudinal space in the cylinder **50** to prevent a direct contact between the actuation-piston first surface **42** and the actuation-cylinder first end **56**.

Alternatively, one may design for the engine valve opening travel to be limited or defined by the physical contact between the actuation-piston first surface **42** and the actuation cylinder first end **56**, or between their equivalent surfaces, with necessary snubbing or control measures, like those shown later in FIGS. **2** and **5**.

The engine valve head **22** is generally exposed to the pressure of the cross-over passage **110** on the first surface **28** and the pressure of the engine cylinder **102** on the second surface **29**.

The cross-section area of the first spring retainer or the pneumatic: piston **78** is to be substantially equal to that of the engine-valve head so that the pneumatic pressure force on the pneumatic piston **78** substantially cancels the pressure force on the engine-valve first surface **28** when the pressure in the pneumatic cylinder **84** is substantially equal to that in the cross-over passage, due to fluid communication through the charge orifice **86**. Alternatively, the cross-section area of the pneumatic piston **78** is to be appreciably, but not necessarily substantially, different from, either larger or smaller than, that of the engine valve head **22**. A larger pneumatic piston cross-section area, for example, offers an extra engine valve opening force so that a relatively more compact fluid driver **30** is sufficient.

The system also experiences various friction forces, steady-state flow forces, transient flow forces, and other inertia forces. Steady-state flow forces are caused by the hydrostatic pressure redistribution due to flow-induced velocity variation, i.e. the Bernoulli effect. Transient flow forces are fluid inertial forces. Other inertial forces result from the acceleration of objects, excluding fluid here, with inertia, and they are substantial in an engine valve assembly because of the large magnitude of the acceleration or the fast timing.

Power-Off State

At power-off state, all fluid supply sources P_H and P_L are at low or zero gage pressure. The total fluid force on the actuation piston **40** is substantially equal to zero. The engine valve can be seated or closed by the return spring **72** alone. The seating is even more secure if the pneumatic piston **78** has a smaller diameter than the engine valve head **22**, and the cross-over passage **110** is still sufficiently pressurized, especially for an air-hybrid application with an air storage tank.

At the power-off, the default position of the actuation 3-way valve **90** is preferably, but not necessarily, to be in its right position **94** as shown in FIG. **1** so that the second fluid space **54** is in fluid communication with the low pressure P_L fluid line and is surely at a low or zero gage pressure if a secure engine valve seating is important or critical. Immediately after engine off, the high pressure P_H fluid line may be still pressurized. At the engine start, the engine valve **20** can be kept at the closed position without actively switching the valve **90**.

Start-Up

To start-up the system from the power-off state, all fluid supply sources are pressurized, and the actuation 3-way valve **90** is secured, either by default or active control, at its right position **94** as shown in FIG. **1**. The engine valve **20** is secured, at least by the return spring **72**, at the closed or seated position as shown in FIG. **1**.

Valve Opening and Closing

To open the engine valve **20**, the actuation 3-way valve **90** is switched to its left position **92**. The second fluid space **54** is open to the high pressure P_H supply through the second flow mechanism, while the first fluid space **52** remains to be exposed to the low pressure P_L supply. The resulting differential pressure force on the actuation piston **40** is in the first direction (or upward in FIG. **1**) to overcome primarily the spring force, driving open the engine valve **20**. At the same time, the downward differential air pressure force on the engine valve **20** is substantially balanced by the upward differential air pressure force on the pneumatic piston **78**, considering that the pneumatic cylinder **84** is under the same pressure as the cross-over passage **110**. In a split-cycle engine, the dominant force on an engine valve is the air pressure force from the cross-over passage **110**. The incorporation of the pneumatic piston **78** helps balance out and counter this large force, which otherwise demands an extremely large and energy-intensive actuator.

As soon as the engine valve **20** cracks open, the engine cylinder **102** is filled rapidly, and its pressure reaches the cross-over passage pressure within a short period of time, well before the engine valve **20** passes the middle point of the opening stroke, resulting in a rapid disappearance of the differential pressure on the engine valve surfaces **28** and **29**. During the same short period of time, the pressure in the pneumatic cylinder **84** and the differential pressure on the pneumatic piston **78** drop rapidly as well because of its limited, predefined initial volume, its rapid volume expansion associated with the engine valve movement, a limited amount of air inflow through the charge orifice **86**, and the bleeding off of the air as the pneumatic piston **78** moves up a predefined distance L**1**, as shown in FIG. **1**, to the expanded top portion **118** of the pneumatic cylinder **84**.

For the rest of the opening stroke or beyond the distance L**1**, the air pressure forces on the pneumatic piston **78** and the engine valve **20** are minimum, and the actuation piston **40** continues to drive the engine valve **20** in the first direction (or upward in FIG. **1**) against an increasing spring force from the return spring **72** until the engine valve reaches its full open position, when the spring force and the fluid differential force across the actuation piston **40** are balanced, which is expected to be dynamic with certain overshoot and damped oscillation, considering the spring-mass nature of the construction. There are however measures, as shown in other preferred embodiments (FIGS. **2** and **4**), to have a more definitive lift or full open position.

The engine valve **20** remains open as long as the actuation 3-way valve **90** remains at its left position **92**. During this period, the pneumatic cylinder **84** keeps receiving a small stream of air flow from the charge orifice **86** and keeps bleeding the air out through the substantial gap between the pneumatic piston **78** and its top, expanded cylinder wall **118**. This energy loss will continue until the pneumatic piston **78** is back at the lower portion of the pneumatic cylinder **84**. However, the energy loss is minimized by the restrictive nature of the charge orifice **86** and the limited engine valve opening period relative to the entire thermal cycle.

To start closing the engine valve, the actuation 3-way valve **90** is switched to its right position **94**, and the second fluid space **54** is open back to the low pressure P_L fluid supply, resulting in a substantially zero pressure differential across the actuation piston **40**. The return spring **72** is able to drive the engine valve **20** downward. When the pneumatic piston **78** passes the expanded part **118** of the pneumatic cylinder **84**, a substantially air-tight seal is established again between the pneumatic piston **78** and the wall of the pneumatic cylinder **84**, and the pressure in the pneumatic cylinder starts building up primarily because of a shrinking cylinder volume as the engine valve **20** and thus the pneumatic piston **78** move downward. The pressure build-up is also assisted by the flow from the charge orifice **86**. The pneumatic cylinder **84** functions like a pneumatic spring, slowing down the advancement of the engine valve **20** and eventually helping achieve a soft-seating when the engine valve **20** reaches the engine-valve seat **26**.

Around the engine valve seating or landing and shortly after that, the pressure in the engine cylinder momentarily exceeds the cross-over passage pressure because of the effect of the combustion, resulting in a transient differential pressure force in the first direction or upward. The preload of the return spring **72** should be designed to be able to hold the engine valve **20** in seated position against this transient upward differential force on the engine valve and also against the pressure force from the pneumatic cylinder **84**. The pneumatic cylinder pressure, at this moment, is however not equal to the full cross-over pressure. It is purposely so by earlier bleeding off through the expanded portion **118** of the pneumatic cylinder **84** and the restrictive nature of the charge orifice **86**.

Thereafter, the engine cylinder pressure drops below the cross-over passage pressure as the volume expands further. The pneumatic cylinder pressure rises up further through the restricted flow from the charge orifice **86** during the rest of the engine thermal cycle, which is slow but sure enough to be ready for the next engine valve opening event.

FIG. **2** depicts an alternative embodiment of the invention that features some variations in the design of the fluid driver **30**. The first flow mechanism, which is the means of the fluid communication between the first port **60** and the first fluid space **52**, includes a first undercut **32** and at least one first snubbing groove **33**. When the actuation-piston first surface **42** passes the first undercut **32** longitudinally in the first direction during an opening stroke, the working fluid is substantially trapped in the first fluid space **52**, with only a limited outlet through the at-least-one first snubbing groove **33**, resulting in a snubbing action to help slow down the travel speed and reduce potential oscillation. When so desired, the actuation-cylinder first end can be longitudinally arranged to provide a solid stop to the actuation-piston first surface **42**, thus a well defined engine valve lift. If so desired, a check valve (not shown in FIG. **2**) can be arranged to allow one-way flow from the first port **60** into the end of the first fluid space **52** during the starting phase of the engine valve closing stroke to avoid cavitation.

Similarly, the second flow mechanism, which is the means of fluid communication between the second port **62** and the second fluid space **58**, includes a second undercut **34** and at least one second snubbing groove **35**. When the actuation-piston second surface **44** passes the second undercut **34** longitudinally in the second direction during an closing stroke, the working fluid is substantially trapped in the second fluid space **58**, with only a limited outlet through the at-least-one second snubbing groove **35**, resulting in a snubbing action to help slow down the travel speed and achieve soft-seating for the engine valve **20**. It is desired to leave a predefined longitudinal distance between the actuation-cylinder second end and the actuation-piston second surface **44** to ensure a solid contact and tight seal between the engine valve head **22** and the valve seat **26** when the engine valve **20** is seated, which has to be accommodated at all engine operating conditions and throughout the engine's service life. When necessary, additional engine valve lash adjustment device (not shown in FIG. **2**) is to be integrated in this and other embodiments.

The embodiment in FIG. **2** further features variations in the design of the spring retainer assembly **74**. The second spring retainer **80**, instead of the first spring retainer **78**, functions or doubles as the pneumatic piston **80**. It also includes two sets of valve keepers **76*b*** and **76*c***. This embodiment allows the engine valve stem **24** and the piston rod **46** to be physically two separate pieces, united operably by the spring retainer assembly **74** with necessary fastening means **106** or the equivalent.

This embodiment also shows variations in the charging and bleeding mechanisms for the pneumatic booster **85**. It adopts at least one bleed hole **87** as the bleed passage, instead of an expanded wall **118** in FIG. **1**, for the pneumatic cylinder **84** to discharge its extra gas when the pneumatic piston **80*b*** travels up a predefined distance L**1** as shown in FIG. **2**. The bleed holes **87** may be fitted with porous materials or filters (not shown) to reduce noise associated the bleeding process. To save the effort and cost of drilling or casting the bleed holes **87**, one may also simply design the engine valve guide **120**, and thus the pneumatic cylinder **84**, up to that height, causing the pneumatic piston **80*b*** to be disengaged from the pneumatic cylinder **84** once it travels up to that point, resulting in a wide open bleeding process.

One can also use some predefined variation (not shown in FIG. **2**) in the radial clearance between the pneumatic piston **80*b*** and the pneumatic cylinder **84**. Adopting an opposite approach, some diaphragm (not shown in FIG. **2**) may be used to completely seal off leakage through the radial clearance, totally depending on the at-least-one bleed hole **87** or its equivalent for the control of the air or gas mass discharge. Also, when desired, one may use a control valve (not shown in FIG. **2**) to control its on/off state.

The charge orifice **86*b*** in FIG. **2** is regulated by a control mechanism including an orifice gate **89** and a stem undercut **104**, which are not open to each other when the engine valve **20** travels up a predefined distance L**2** (as shown in FIG. **2**). The distance L**2** is preferably to be equal or shorter than the distance L**1** so that the flow through the charge orifice **86*b*** and thus the charging process are substantially blocked when the discharging process, through the bleed hole **87** or its equivalent, is active. This variation in the charge mechanism will help reduce unnecessary, however small, energy loss.

Refer now to FIG. **3**, which is a drawing of yet another alternative embodiment of the invention. In this fluid driver **30**, a proportional or servo 3-way valve **90*c*** is used to control the fluid supply to the second fluid space **54**. The engine valve or actuator position signal can be collected via a position sensor (not shown in FIG. **3**). The feedback control will help achieve more precise control over the engine valve lift and seating velocity. The proportional or servo valve **90*c*** itself can be actuated directly via various means (not shown in FIG. **3**), including solenoids or other electromagnetic means, electro-hydraulic pilot valves, and piezoelectric actuators.

This embodiment further features a charge valve **108**, as a control mechanism, along the charge passage **112** to help achieve better control over the charging process for the pneumatic cylinder **84**. The charge valve **108** has at least one of two major functions: (1) to open the charge passage **112**, allowing the pneumatic cylinder **84** to be charged, before the engine valve opening stroke, and close the charge passage **112** especially if the restrictive charge orifice **86** is not used, eliminating or reducing leak flow when the pneumatic cylinder **84** is being bled; (2) to completely close off the charge passage **112** when the engine or that particular engine cylinder is power-off, as in an air hybrid vehicle, minimizing leakage and preserving the pressurized air in the cross-over passage and/or the air storage tank. For the first function, one charge valve **108** is needed for each power cylinder of the split four-stroke cycle engine because each power cylinder has its unique timing. If only the second function is needed, one may optionally use only one charge valve **108** for an entire engine, with the valve **108** controlling a common charge passage (not shown in FIG. **3**) that eventually branches into tributary charge passages (not shown in FIG. **3**) for individual power cylinders (not shown in FIG. **3**). Further for the first function, the charge valve **108** may be optionally a proportional valve, instead of an on/off valve. By being a proportional valve, the charge valve **108** is able to actively control, for example, the air pressure in the pneumatic cylinder **84** for various functional, durability and NVH needs.

At this and other figures, the charge passage **112** is connected to the cross-over passage **110**. Optionally, it can be connected to the air storage tank (in the case of an air hybrid vehicle) or a separate reservoir (not shown in the figures). The separate reservoir may have its own pressure, which may be regulated to help achieve optimum charging process for the pneumatic cylinder **84**.

Refer now to FIG. **4**, which is a drawing of yet another alternative embodiment of the invention. In this case, a proportional or servo 4-way valve **90*d*** is used to control the fluid supply both to the first and second fluid spaces **52** and **54**. This embodiment is able to provide actively-controlled actuation forces both in the first and second directions. Optionally, the piston rod **46** extends longitudinally through the first fluid space **52**, becoming a double-ended piston rod. To have a biased or asymmetric differential fluid force, the two ends of the piston rod may possess two different diameters, with the side with a smaller rod diameter having a larger effective fluid pressure surface area.

Still another variation or option is its lack of a bleed mechanism. The actuation force in the second direction will easily help overcome the high air pressure force from the pneumatic booster **85** during the engine valve closing. The elimination of the bleed mechanism will help simplify the construction of the pneumatic booster **85**. Without a bleed mechanism or substantial leakage, the charge mechanism, including the charge orifice **86**, is still needed to compensate for potential minor leakages, and adjust the pressure and air mass level in the pneumatic booster **85** to accommodate the pressure level variation in the cross-over passage or air storage tank. The actuator needs a lower boost force, for example, when the cross-over passage pressure is lower. In this sense, the charge mechanism also has a balance function, which is even true for the pneumatic boosters with a bleed mechanism.

Depending on the application, the rest of the embodiment in FIG. **4** may still be integrated with one of the bleed mechanisms featured in earlier embodiments (illustrated in FIGS. **1-3**) if a lower air pressure force is ideal for the engine valve seating process.

Refer now to FIG. **5**, which is a drawing of yet another alternative embodiment of the invention. In this embodiment, an electromagnetic driver **130** replaces the fluid drivers **30** in FIGS. **1-4**. The electromagnetic driver **130** includes a housing **132**, within which from the top to the bottom are a first electromagnet **134**, an armature chamber **146**, and a second electromagnet **136**. The first and second electromagnets **134** and **136** further include their electrical windings and lamination stacks, details of which are not shown in FIG. **5**. An armature **138** is disposed inside the armature chamber **146** and between the first and second electromagnets **134** and **136** and is rigidly connected to an armature rod **140**. The armature rod **140** is slideably disposed through the second electromagnet **136** and the housing **132**, and is operably connected with the engine valve stem **24**.

When powered, the first and second electromagnets **134** and **136** attract the armature **138** in the first (top) and second (bottom) directions, respectively. The first electromagnet **134** is able to catch the armature **138** and keep the engine valve **20** open at the full lift. To crack open the engine valve **20** when the air pressure forces on the engine valve **20** and the pneumatic piston **80** are substantially balanced, the first electromagnet **134** only needs to overcome the preload from the return spring **72**, which is achievable despite the highly nonlinear nature of the electromagnetic force because the overall lift for the cross-over engine valve and thus the air gap between the armature **138** and the electromagnet **134** are small. This can be further assisted, if necessary, by designing the pneumatic piston **80** to be appreciably larger than the engine valve head **22** and thus introducing a differential air pressure force in the first direction.

To close the engine valve **20** from the full open position, the first electromagnet **134** is de-energized, and the engine valve **20** is pushed down by the returning force of the return spring **72**, with the pulling assistance, if necessary, from an energized second electromagnet **136**. During the later phase of the closing, the pneumatic cylinder **86** is pressurized by volumetric contraction and optional charging action through the charge orifice **86***b*, and it helps slow down the engine valve **20** to achieve soft-seating. A further retarding action can be achieved by re-energizing the first electromagnet **134** in a controlled way, resulting in a desired pulling force in the first direction depending on the operational needs or feedback signal.

The pulling force in the second direction from the second electromagnet **136** may also assist the return spring **72**, if a low spring preload is desired otherwise, in keeping the engine valve **20** seated during at least part of combustion, when the pressure in the power cylinder **102** appreciably exceeds that in the cross-over passage **110**.

If the pneumatic booster **85** includes a bleed mechanism like the bleed holes **87** in FIG. **5**, the second electromagnet **136** is an optional component, which can be eliminated if the return spring **72** and other related components are sufficient for various functions.

The second electromagnet **136** is indispensable, however, if one is to adopt a pneumatic booster design without, as shown in FIG. **4**, a bleed mechanism. In this case, the second electromagnet **136** needs to generate an actuation force in the second direction to help overcome a high air pressure force from the pneumatic booster during the engine valve closing, when there is no high differential air pressure force on the engine valve to balance the force from the pneumatic booster.

In FIGS. **1-5**, various embodiments of the pneumatic booster **85** are specially developed to overcome the initial pressure force on the engine-valve first surface **28** to crack open the engine valve. Yet, through its bleed mechanism, the pneumatic booster **85** is able to scale down its pressure force for the valve closing when the differential pressure force across the engine valve head is substantially smaller. With this pneumatic booster **85**, the fluid drivers **30** in FIGS. **1-4** and the electromagnetic driver **130** in FIG. **5** are able to handle the less forceful part of the engine valve opening and closing. The effective integration of the various embodiments of the pneumatic booster **85** is not limited to those fluid and electromagnetic drivers **30** and **130** discussed above. In fact, any driver with sufficient force and control for the engine valve acceleration, deceleration, and seating control will do, with the large initial opening force being taken care of by the pneumatic booster **85**.

In all the above descriptions, each of the switch and/or control valves may be either a single-stage type or a multiple-stage type. Each valve can be either a linear type (such as a spool valve) or a rotary type. Each valve can be driven or piloted by an electric, electromagnetic, mechanic, piezoelectric, or fluid means.

In some illustrations and descriptions, the fluid medium may be assumed or implied to be in hydraulic or in liquid form. In most cases, the same concepts can be applied, with proper scaling, to pneumatic boosters and systems. As such, the term "fluid" as used herein is meant to include both liquids and gases. Also, in many illustrations and descriptions so far, the application of the invention is defaulted to be in split four-stroke cycle internal combustion engine valve control, and it is not limited so. The invention can be applied to other situations where a fast and/or high-initial-force control of the motion is needed.

Although the present invention has been described with reference to the preferred embodiments, those skilled in the art will recognize that changes may be made in form and detail without departing from the spirit and scope of the invention. As such, it is intended that the foregoing detailed description be regarded as illustrative rather than limiting and that it is the appended claims, including all equivalents thereof, which are intended to define the scope of this invention.

I claim:

1. An actuator, comprising

a fluid driver including a housing defining a longitudinal axis and first and second directions, an actuation mechanism comprising an actuation cylinder and an actuation piston slideably disposed in the actuation cylinder, with the actuation mechanism generating actuation force at least in the first direction;

at least one return spring operably connected with the actuation mechanism and biasing the actuation mechanism in the second direction; and

a pneumatic booster operably connected with the actuation mechanism and biasing the actuation mechanism in the first direction, whereby giving the driver a boost force in the first direction; and wherein:

the pneumatic booster further including a pneumatic cylinder; a pneumatic piston, slideably disposed in the pneumatic cylinder for at least part of its travel range; and a charge mechanism, whereby charging the pneumatic cylinder and balancing its pressure with pressurized air from a high-pressure gas source during at least part of an actuation cycle of the actuator;

the charge mechanism including a control mechanism, whereby substantially closing off the charge flow during a substantial portion of a period when the pneumatic cylinder is not at its minimum volume;

the actuation mechanism being operably connected with a load of the actuator through a stem;

the control mechanism of the charge mechanism including an orifice gate and an undercut on the stem; and

the charge flow being open and closed when the orifice gate and the undercut longitudinally overlap and underlap respectively.

2. An actuator, comprising

a fluid driver including a housing defining a longitudinal axis and first and second directions, an actuation mechanism comprising an actuation cylinder and an actuation piston slideably disposed in the actuation cylinder, with the actuation mechanism generating actuation force at least in the first direction;

at least one return spring operably connected with the actuation mechanism and biasing the actuation mechanism in the second direction; and

a pneumatic booster operably connected with the actuation mechanism and biasing the actuation mechanism in the first direction, whereby giving the driver a boost force in the first direction; and wherein:

the pneumatic booster further including a pneumatic cylinder; a pneumatic piston, slideably disposed in the pneumatic cylinder for at least part of its travel range; and a charge mechanism, whereby charging the pneumatic cylinder and balancing its pressure with pressurized air from a high-pressure gas source during at least part of an actuation cycle of the actuator;

the pneumatic booster further including a bleed mechanism, whereby bleeding off excess air from the pneumatic cylinder to a low-pressure gas sink during at least part of an actuation cycle of the actuator; and

the bleed mechanism being at least one bleed passage, which becomes substantially exposed to the pressurized air in the pneumatic cylinder when the pneumatic piston travels beyond a predefined distance from an initial position, whereby bleeding off excess air.

3. An actuator, comprising

a fluid driver including a housing defining a longitudinal axis and first and second directions, an actuation mechanism comprising an actuation cylinder and an actuation piston slideably disposed in the actuation cylinder, with the actuation mechanism generating actuation force at least in the first direction;

at least one return spring operably connected with the actuation mechanism and biasing the actuation mechanism in the second direction; and

a pneumatic booster operably connected with the actuation mechanism and biasing the actuation mechanism in the first direction, whereby giving the driver a boost force in the first direction; and wherein:

the actuation piston dividing the inner volume of the actuation cylinder into first and second fluid spaces and being operably connected with a piston rod; and

the actuation mechanism further comprising first and second ports in fluid communication with the first and second fluid spaces, respectively.

4. The actuator of claim **3**, wherein

the second port being alternately supplied with high-pressure and low-pressure fluid lines, through an actuation 3-way valve.

5. The actuator of claim **3**, wherein

the second port being supplied through a proportional 3-way valve.

6. The actuator of claim **3**, wherein

the first and second ports being supplied through a 4-way valve.

7. A method of controlling an actuator comprising:

(a) providing an actuator including the following components:

a driver further including

a housing defining a longitudinal axis and first and second directions,

an actuation mechanism capable of generating actuation force at least in the first direction, and

a rod with one end operably connected with at least one part of the actuation mechanism and with the other end available for an operable connection with a load of the actuator;

at least one return spring operably connected with the rod and biasing the rod in the second direction; and

a pneumatic booster further including

a pneumatic cylinder,

a pneumatic piston operably connected with the rod and biasing the rod in the first direction, and

a charge mechanism, whereby providing a controlled fluid communication between the pneumatic cylinder and a high-pressure gas source;

(b) holding the load of the actuator to a second-direction end position

with the force from the at-least-one return spring biasing in the second direction and overcoming the sum of the rest of the forces including those from the pneumatic booster and the load,

without generating the actuation force in the first direction from the actuation mechanism, and

with the pneumatic booster being charged through the charge mechanism to yield a substantial force in the first direction to oppose a substantial load force in the second direction;

(c) initiating the travel of the load of the actuator in the first direction by generating the actuation force in the first direction from the actuation mechanism, with the combination of the actuation force and the force from the pneumatic booster being able to overcome the sum of the rest of the forces including those from the at-least-one return spring and the load and accelerate the load in the first direction;

(d) continuing the travel in the first direction with the actuation force in the first direction at least until reaching the target stroke;

(e) initiating the return travel of the load of the actuator in the second direction at least by turning off the actuation force in the first direction so that the load is accelerated in the second direction at least by the return spring;

(f) completing the return travel with a decreasing force from the return spring and an increasing force from the pneumatic booster, whereby slowing down the load.

8. The method of claim **7**, wherein:

the pneumatic booster further including a bleed mechanism, whereby providing a controlled fluid communication between the pneumatic cylinder and a low-pressure gas sink.

9. The method of claim **8**, further comprising

bleeding off excess air in the booster cylinder through the bleed mechanism, during at least part of the time period after the initial travel in the first direction and before near the close of the travel in the second direction, to reduce the force from the pneumatic booster.

10. The method of claim **7**, wherein:

the charge mechanism including a charge orifice, whereby substantially restricting the charge flow rate.

11. The method of claim **7**, wherein:

the charge mechanism including a control mechanism, whereby substantially closing off the charge flow during a substantial portion of a period when the pneumatic cylinder is not at its minimum volume.

12. The method of claim **8**, wherein:

the bleed mechanism being at least one passage, which becomes exposed to the pressurized air in the pneumatic cylinder when the pneumatic piston travels beyond a predefined distance from an initial position, whereby bleeding off excess air.

13. The method of claim **7**, wherein

the driver being a fluid driver;

the actuation mechanism comprising an actuation cylinder, an actuation piston slideably disposed in the actuation cylinder and dividing the inner volume of the actuation cylinder into first and second fluid spaces, and first and second ports in fluid communication with the first and second fluid spaces, respectively; and

the rod being a piston rod operably connected with the actuation piston.

14. The method of claim **7**, wherein

the driver being an electromagnetic driver;

the actuation mechanism comprising an armature chamber, an armature disposed in the armature chamber, and at least a first electromagnet on the first direction side of the armature chamber, whereby being able to pull the armature in the first direction when energized; and

the rod being an armature rod operably connected with the armature.

15. The method of claim **14**, further including a second electromagnet on the second direction side of the armature chamber, whereby being able to pull the armature in the second direction when energized.

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