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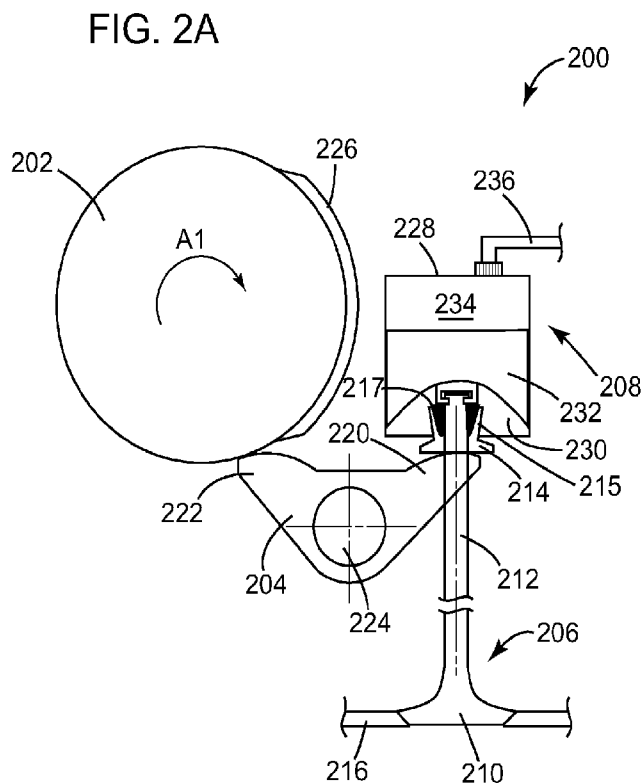
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(54) Title: VARIABLE FORCE VALVE SPRING



(57) Abstract: Devices and related methods are disclosed that generally involve variable force valve springs for controlling the motion of an engine valve. The force exerted by the valve spring can be adjusted by altering the pressure at which a fluid is supplied to a fluid chamber thereof, by altering the volume of the fluid chamber, and/or by changing the aggregate surface area over which fluid pressure is coupled to the engine valve. Associated fluid control systems are also disclosed herein, as are various methods for adjusting the force of a valve spring based on a variety of engine parameters, such as engine speed, engine load, and/or a combination thereof.



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VARIABLE FORCE VALVE SPRING

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application claims the benefit of priority of U.S. Provisional Patent Application Number 61/436,739, filed on January 27, 2011, the entire contents of which are incorporated herein by reference.

FIELD

[0002] The present invention relates to methods and devices for actuating an engine valve. More particularly, the invention relates to variable force valve springs and methods related thereto.

BACKGROUND

[0003] Internal combustion engines generally include one or more valves for controlling the flow of air and fuel through the engine. These valves are usually actuated by a mechanical cam. For example, a rotating shaft having a teardrop-shaped cam lobe can be configured to impart motion to the valve, either directly or via one or more intermediate elements. As the shaft rotates, the eccentric portion of the cam lobe imparts a linear motion to the valve over a range of the shaft's rotation.

[0004] It is often necessary to bias the valve towards a particular position (i.e., opened or closed). In conventional internal combustion engines, the valves are generally inwardly-opening (i.e., opening into the combustion chamber and toward the piston) and are biased to a closed position by one or more mechanical valve springs. One problem with this system is that there is no way to alter or control the amount of force required to compress the valve spring. The mechanical springs are generally chosen such that they have sufficient fitted force and are stiff enough to handle the most demanding of the conditions the engine will experience over its operating range, which may only be encountered in very rare situations. In other words, the valve spring is far stronger than what is required for the majority of the engine's operating range. In conventional internal combustion engines, a stronger valve spring produces higher rocker

shaft and cam shaft bearing friction, contributing to the parasitic losses of the engine and reducing overall engine efficiency.

[0005] Traditional mechanical valve springs are also poorly suited for “lost-motion” valve train systems. A lost-motion system is generally a system in which a lost-motion valve train element is selectively actuated to operatively disconnect the valve from a cam during a portion of the cam’s rotation. In a valve train having no lost-motion system, some of the energy exerted to compress the valve spring is returned to the cam and ultimately to the rotation of the engine when the spring expands against the closing ramp of the cam. In lost-motion systems, however, the valve can be closed earlier than what is called for by the cam by operatively disconnecting the valve from the cam. In this case, the energy stored in the compressed valve spring of an open valve is not returned to the cam, and is instead applied only to the valve as it closes. Since none of the energy is returned to the cam, the energy spent to compress the valve spring in the first place is essentially wasted.

[0006] Accordingly, there is a need for improved valve springs and related methods.

[0007] For purposes of clarity, the term “conventional engine” as used in the present application refers to an internal combustion engine wherein all four strokes of the well-known Otto cycle (the intake, compression, expansion and exhaust strokes) are contained in each piston/cylinder combination of the engine. Each stroke requires one half revolution of the crankshaft (180 degrees crank angle (“CA”)), and two full revolutions of the crankshaft (720 degrees CA) are required to complete the entire Otto cycle in each cylinder of a conventional engine.

[0008] Also, for purposes of clarity, the following definition is offered for the term “split-cycle engine” as may be applied to engines disclosed in the prior art and as referred to in the present application.

[0009] A split-cycle engine generally comprises:

[0010] a crankshaft rotatable about a crankshaft axis;

[0011] a compression piston slidably received within a compression cylinder and operatively connected to the crankshaft such that the compression piston reciprocates through an intake stroke and a compression stroke during a single rotation of the crankshaft;

[0012] an expansion (power) piston slidably received within an expansion cylinder and operatively connected to the crankshaft such that the expansion piston reciprocates through an expansion stroke and an exhaust stroke during a single rotation of the crankshaft; and

[0013] a crossover passage interconnecting the compression and expansion cylinders, the crossover passage including at least a crossover expansion (XovrE) valve disposed therein, but more preferably including a crossover compression (XovrC) valve and a crossover expansion (XovrE) valve defining a pressure chamber therebetween.

[0014] A split-cycle air hybrid engine combines a split-cycle engine with an air reservoir and various controls. This combination enables the engine to store energy in the form of compressed air in the air reservoir. The compressed air in the air reservoir is later used in the expansion cylinder to power the crankshaft. In general, a split-cycle air hybrid engine as referred to herein comprises:

[0015] a crankshaft rotatable about a crankshaft axis;

[0016] a compression piston slidably received within a compression cylinder and operatively connected to the crankshaft such that the compression piston reciprocates through an intake stroke and a compression stroke during a single rotation of the crankshaft;

[0017] an expansion (power) piston slidably received within an expansion cylinder and operatively connected to the crankshaft such that the expansion piston reciprocates through an expansion stroke and an exhaust stroke during a single rotation of the crankshaft;

[0018] a crossover passage (port) interconnecting the compression and expansion cylinders, the crossover passage including at least a crossover expansion (XovrE) valve disposed therein, but more preferably a crossover compression (XovrC) valve and a crossover expansion (XovrE) valve defining a pressure chamber therebetween; and

[0019] an air reservoir operatively connected to the crossover passage and selectively operable to store compressed air from the compression cylinder and to deliver compressed air to the expansion cylinder.

[0020] FIG. 1 illustrates one exemplary embodiment of a prior art split-cycle air hybrid engine. The split-cycle engine 100 replaces two adjacent cylinders of a conventional engine with a combination of one compression cylinder 102 and one expansion cylinder 104. The compression cylinder 102 and the expansion cylinder 104 are formed in an engine block in which a crankshaft 106 is rotatably mounted. Upper ends of the cylinders 102, 104 are closed by a cylinder head 130. The crankshaft 106 includes axially displaced and angularly offset first and second crank throws 126, 128, having a phase angle therebetween. The first crank throw 126 is pivotally joined by a first connecting rod 138 to a compression piston 110 and the second crank throw 128 is pivotally joined by a second connecting rod 140 to an expansion piston 120 to reciprocate the pistons 110, 120 in their respective cylinders 102, 104 in a timed relation determined by the angular offset of the crank throws and the geometric relationships of the cylinders, crank, and pistons. Alternative mechanisms for relating the motion and timing of the pistons can be utilized if desired. The rotational direction of the crankshaft and the relative motions of the pistons near their bottom dead center (BDC) positions are indicated by the arrows associated in the drawings with their corresponding components.

[0021] The four strokes of the Otto cycle are thus “split” over the two cylinders 102 and 104 such that the compression cylinder 102 contains the intake and compression strokes and the expansion cylinder 104 contains the expansion and exhaust strokes. The Otto cycle is therefore completed in these two cylinders 102, 104 once per crankshaft 106 revolution (360 degrees CA).

[0022] During the intake stroke, intake air is drawn into the compression cylinder 102 through an inwardly-opening (opening inward into the cylinder and toward the piston) poppet intake valve 108. During the compression stroke, a compression piston 110 pressurizes the air charge and drives the air charge through a crossover passage 112, which acts as the intake passage for the expansion cylinder 104. The engine 100 can have one or more crossover passages 112.

[0023] The volumetric (or geometric) compression ratio of the compression cylinder 102 of the split-cycle engine 100 (and for split-cycle engines in general) is herein referred to as the

“compression ratio” of the split-cycle engine. The volumetric (or geometric) compression ratio of the expansion cylinder 104 of the engine 100 (and for split-cycle engines in general) is herein referred to as the “expansion ratio” of the split-cycle engine. The volumetric compression ratio of a cylinder is well known in the art as the ratio of the enclosed (or trapped) volume in the cylinder (including all recesses) when a piston reciprocating therein is at its bottom dead center (BDC) position to the enclosed volume (i.e., clearance volume) in the cylinder when said piston is at its top dead center (TDC) position. Specifically for split-cycle engines as defined herein, the compression ratio of a compression cylinder is determined when the XovrC valve is closed. Also specifically for split-cycle engines as defined herein, the expansion ratio of an expansion cylinder is determined when the XovrE valve is closed.

[0024] Due to very high volumetric compression ratios (e.g., 20 to 1, 30 to 1, 40 to 1, or greater) within the compression cylinder 102, an outwardly-opening (opening outwardly away from the cylinder and piston) poppet crossover compression (XovrC) valve 114 at the crossover passage inlet is used to control flow from the compression cylinder 102 into the crossover passage 112. Due to very high volumetric compression ratios (e.g., 20 to 1, 30 to 1, 40 to 1, or greater) within the expansion cylinder 104, an outwardly-opening poppet crossover expansion (XovrE) valve 116 at the outlet of the crossover passage 112 controls flow from the crossover passage 112 into the expansion cylinder 104. The actuation rates and phasing of the XovrC and XovrE valves 114, 116 are timed to maintain pressure in the crossover passage 112 at a high minimum pressure (typically 20 bar or higher at full load) during all four strokes of the Otto cycle.

[0025] At least one fuel injector 118 injects fuel into the pressurized air at the exit end of the crossover passage 112 in coordination with the XovrE valve 116 opening. Alternatively, or in addition, fuel can be injected directly into the expansion cylinder 104. The fuel-air charge fully enters the expansion cylinder 104 shortly after the expansion piston 120 reaches its top dead center (“TDC”) position. As the piston 120 begins its descent from its TDC position, and while the XovrE valve 116 is still open, one or more spark plugs 122 are fired to initiate combustion (typically between 10 to 20 degrees CA after TDC of the expansion piston 120). Combustion can be initiated while the expansion piston is between 1 and 30 degrees CA past its TDC position. More preferably, combustion can be initiated while the expansion piston is between 5 and 25 degrees CA past its TDC position. Most preferably, combustion can be initiated while

the expansion piston is between 10 and 20 degrees CA past its TDC position. Additionally, combustion can be initiated through other ignition devices and/or methods, such as with glow plugs, microwave ignition devices, or through compression ignition methods.

[0026] The XovrE valve 116 is then closed before the resulting combustion event enters the crossover passage 112. The combustion event drives the expansion piston 120 downward in a power stroke. Exhaust gases are pumped out of the expansion cylinder 104 through an inwardly-opening poppet exhaust valve 124 during the exhaust stroke.

[0027] With the split-cycle engine concept, the geometric engine parameters (i.e., bore, stroke, connecting rod length, compression ratio, etc.) of the compression and expansion cylinders are generally independent from one another. For example, the crank throws 126, 128 for the compression cylinder 102 and expansion cylinder 104, respectively, have different radii and are phased apart from one another with TDC of the expansion piston 120 occurring prior to TDC of the compression piston 110. This independence enables the split-cycle engine to potentially achieve higher efficiency levels and greater torques than typical four-stroke engines.

[0028] The geometric independence of engine parameters in the split-cycle engine 100 is also one of the main reasons why pressure can be maintained in the crossover passage 112 as discussed earlier. Specifically, the expansion piston 120 reaches its top dead center position prior to the compression piston 110 reaching its top dead center position by a discrete phase angle (typically between 10 and 30 crank angle degrees). This phase angle, together with proper timing of the XovrC valve 114 and the XovrE valve 116, enables the split-cycle engine 100 to maintain pressure in the crossover passage 112 at a high minimum pressure (typically 20 bar absolute or higher during full load operation) during all four strokes of its pressure/volume cycle. That is, the split-cycle engine 100 is operable to time the XovrC valve 114 and the XovrE valve 116 such that the XovrC and XovrE valves 114, 116 are both open for a substantial period of time (or period of crankshaft rotation) during which the expansion piston 120 descends from its TDC position towards its BDC position and the compression piston 110 simultaneously ascends from its BDC position towards its TDC position. During the period of time (or crankshaft rotation) that the crossover valves 114, 116 are both open, a substantially equal mass of gas is transferred (1) from the compression cylinder 102 into the crossover passage 112 and (2) from

the crossover passage 112 to the expansion cylinder 104. Accordingly, during this period, the pressure in the crossover passage is prevented from dropping below a predetermined minimum pressure (typically 20, 30, or 40 bar absolute during full load operation). Moreover, during a substantial portion of the intake and exhaust strokes (typically 90% of the entire intake and exhaust strokes or greater), the XovrC valve 114 and XovrE valve 116 are both closed to maintain the mass of trapped gas in the crossover passage 112 at a substantially constant level. As a result, the pressure in the crossover passage 112 is maintained at a predetermined minimum pressure during all four strokes of the engine's pressure/volume cycle.

[0029] For purposes herein, the method of opening the XovrC 114 and XovrE 116 valves while the expansion piston 120 is descending from TDC and the compression piston 110 is ascending toward TDC in order to simultaneously transfer a substantially equal mass of gas into and out of the crossover passage 112 is referred to herein as the "push-pull" method of gas transfer. It is the push-pull method that enables the pressure in the crossover passage 112 of the engine 100 to be maintained at typically 20 bar or higher during all four strokes of the engine's cycle when the engine is operating at full load.

[0030] The crossover valves 114, 116 are actuated by a valve train that includes one or more cams (not shown). In general, a cam-driven mechanism includes a camshaft mechanically linked to the crankshaft. One or more cams are mounted to the camshaft, each having a contoured surface that controls the valve lift profile of the valve event (i.e., the event that occurs during a valve actuation). The XovrC valve 114 and the XovrE valve 116 each can have its own respective cam and/or its own respective camshaft. As the XovrC and XovrE cams rotate, eccentric portions thereof impart motion to a rocker arm, which in turn imparts motion to the valve, thereby lifting (opening) the valve off of its valve seat. As the cam continues to rotate, the eccentric portion passes the rocker arm and the valve is allowed to close.

[0031] For purposes herein, a valve event (or valve opening event) is defined as the valve lift from its initial opening off of its valve seat to its closing back onto its valve seat versus rotation of the crankshaft during which the valve lift occurs. Also, for purposes herein, the valve event rate (i.e., the valve actuation rate) is the duration in time required for the valve event to occur within a given engine cycle. It is important to note that a valve event is generally only a fraction

of the total duration of an engine operating cycle (e.g., 720 degrees CA for a conventional engine cycle and 360 degrees CA for a split-cycle engine).

[0032] The split-cycle air hybrid engine 100 also includes an air reservoir (tank) 142, which is operatively connected to the crossover passage 112 by an air reservoir tank valve 152.

Embodiments with two or more crossover passages 112 may include a tank valve 152 for each crossover passage 112, which connect to a common air reservoir 142, or alternatively each crossover passage 112 may operatively connect to separate air reservoirs 142.

[0033] The tank valve 152 is typically disposed in an air tank port 154, which extends from the crossover passage 112 to the air tank 142. The air tank port 154 is divided into a first air tank port section 156 and a second air tank port section 158. The first air tank port section 156 connects the air tank valve 152 to the crossover passage 112, and the second air tank port section 158 connects the air tank valve 152 to the air tank 142. The volume of the first air tank port section 156 includes the volume of all additional recesses which connect the tank valve 152 to the crossover passage 112 when the tank valve 152 is closed. Preferably, the volume of the first air tank port section 156 is small relative to the second air tank port section 158. More preferably, the first air tank port section 156 is substantially non-existent, that is, the tank valve 152 is most preferably disposed such that it is flush against the outer wall of the crossover passage 112.

[0034] The tank valve 152 may be any suitable valve device or system. For example, the tank valve 152 may be a pressure-activated check valve, or an active valve which is activated by various valve actuation devices (e.g., pneumatic, hydraulic, cam, electric, or the like). Additionally, the tank valve 152 may comprise a tank valve system with two or more valves actuated with two or more actuation devices.

[0035] The air tank 142 is utilized to store energy in the form of compressed air and to later use that compressed air to power the crankshaft 106. This mechanical means for storing potential energy provides numerous potential advantages over the current state of the art. For instance, the split-cycle air hybrid engine 100 can potentially provide many advantages in fuel efficiency gains and NO_x emissions reduction at relatively low manufacturing and waste disposal costs in relation to other technologies on the market, such as diesel engines and electric-hybrid systems.

[0036] The engine 100 typically runs in a normal operating or firing (NF) mode (also commonly called the engine firing (EF) mode) and one or more of four basic air hybrid modes. In the EF mode, the engine 100 functions normally as previously described in detail herein, operating without the use of the air tank 142. In the EF mode, the air tank valve 152 remains closed to isolate the air tank 142 from the basic split-cycle engine. In the four air hybrid modes, the engine 100 operates with the use of the air tank 142.

[0037] The four basic air hybrid modes include:

[0038] 1) Air Expander (AE) mode, which includes using compressed air energy from the air tank 142 without combustion;

[0039] 2) Air Compressor (AC) mode, which includes storing compressed air energy into the air tank 142 without combustion;

[0040] 3) Air Expander and Firing (AEF) mode, which includes using compressed air energy from the air tank 142 with combustion; and

[0041] 4) Firing and Charging (FC) mode, which includes storing compressed air energy into the air tank 142 with combustion.

[0042] Further details on split-cycle engines can be found in U.S. Patent No. 6,543,225 entitled Split Four Stroke Cycle Internal Combustion Engine and issued on April 8, 2003; and U.S. Patent No. 6,952,923 entitled Split-Cycle Four-Stroke Engine and issued on October 11, 2005, each of which is incorporated by reference herein in its entirety.

[0043] Further details on air hybrid engines are disclosed in U.S. Patent No. 7,353,786 entitled Split-Cycle Air Hybrid Engine and issued on April 8, 2008; U.S. Patent Application No. 61/365,343 entitled Split-Cycle Air Hybrid Engine and filed on July 18, 2010; and U.S. Patent Application No. 61/313,831 entitled Split-Cycle Air Hybrid Engine and filed on March 15, 2010, each of which is incorporated by reference herein in its entirety.

SUMMARY

[0044] Devices and related methods are disclosed that generally involve variable force valve springs for controlling the motion of an engine valve. The force exerted by the valve spring can be adjusted by altering the pressure at which a fluid is supplied to a fluid chamber thereof, by altering the volume of the fluid chamber, and/or by changing the aggregate surface area over which fluid pressure is coupled to the engine valve. Associated fluid control systems are also disclosed herein, as are various methods for adjusting the force of a valve spring based on a variety of engine parameters, such as engine speed, engine load, and/or a combination thereof.

[0045] In one aspect of at least one embodiment of the invention, an engine is provided that includes a pneumatic valve spring coupled to an engine valve and a control module configured to adjust a spring force of the pneumatic valve spring based on at least one of engine speed and engine load.

[0046] In another aspect of at least one embodiment of the invention, a valve spring system is provided that includes a vessel having a piston reciprocally disposed therein, an engine valve coupled to the piston, a first source of pressurized air having a first pressure, and a second source of pressurized air having a second pressure that is greater than the first pressure. The system also includes a control valve configured to adjust a force exerted on the piston by controlling a first fluid communication between the vessel and the first source and a second fluid communication between the vessel and the second source.

[0047] In another aspect of at least one embodiment of the invention, a valve spring system is provided that includes a cap defining a first bore therein and a piston coupled to an engine valve, the piston being slidably disposed within the first bore such that the piston and the cap define a first fluid chamber. The system also includes an outer housing defining a second bore therein, the cap being slidably disposed within the second bore such that the cap and the outer housing define a second fluid chamber, and a fluid port configured to selectively release pressurized fluid from the second fluid chamber, thereby allowing the cap to slide relative to the outer housing such that a volume of the first fluid chamber is increased.

[0048] In another aspect of at least one embodiment of the invention, a valve spring system is provided that includes a first housing defining a first bore therein in which a first piston is reciprocally disposed, the first piston being coupled to a valve stem of an engine valve. The

system also includes a second housing defining a second bore therein in which a second piston is reciprocally disposed, the second piston being coupled to an extension stem. The system also includes a third housing defining a hydraulic plenum, the hydraulic plenum being in fluid communication with a proximal end of the valve stem, a distal end of the extension stem, and a control valve. The plenum is in fluid communication with an accumulator when the control valve is opened such that the valve stem is movable independently from the extension stem, and the plenum is sealed when the control valve is closed such that movement of the valve stem requires movement of the extension stem.

[0049] In another aspect of at least one embodiment of the invention, a method of actuating a valve of an engine is provided that includes increasing a spring force of a valve spring associated with the valve based on an increase in an engine parameter, and decreasing the spring force of the valve spring based on a decrease in the engine parameter.

[0050] The present invention further provides devices, systems, and methods as claimed.

BRIEF DESCRIPTION OF THE DRAWINGS

[0051] The invention will be more fully understood from the following detailed description taken in conjunction with the accompanying drawings, in which:

[0052] FIG. 1 is a schematic cross-sectional view of a prior art split-cycle air hybrid engine;

[0053] FIG. 2A is a schematic view of one embodiment of a valve train according to the present invention in which a valve is closed;

[0054] FIG. 2B is a schematic view of the valve train of FIG. 2A in which the valve is opened;

[0055] FIG. 3A is a schematic view of one embodiment of a fluid control system according to the present invention, shown with a spool valve in a first position;

[0056] FIG. 3B is a schematic view of the fluid control system of FIG. 3A, shown with a spool valve in a second position;

[0057] FIG. 3C is a schematic view of the fluid control system of FIGS. 3A and 3B, shown with a spool valve in a third position;

[0058] FIG. 4 is a schematic view of another embodiment of a fluid control system according to the present invention;

[0059] FIG. 5A is a schematic view of one embodiment of a valve train according to the present invention in which an inwardly-opening valve is closed;

[0060] FIG. 5B is a schematic view of the valve train of FIG. 5A in which the inwardly-opening valve is opened;

[0061] FIG. 6A is a schematic view of another embodiment of a valve spring according to the present invention shown in a reduced filling volume configuration;

[0062] FIG. 6B is a schematic view of the valve spring of FIG. 6A shown in an increased filling volume configuration;

[0063] FIG. 7 is a schematic view of another embodiment of a valve spring according to the present invention;

[0064] FIG. 8A is a plot of required spring pressure as a function of engine speed for one embodiment of an engine according to the present invention;

[0065] FIG. 8B is a plot of required spring pressure normalized to required idle spring pressure as a function of engine speed for one embodiment of an engine according to the present invention;

[0066] FIG. 9A is a series of plots of spring force as a function of engine speed for various embodiments of the present invention; and

[0067] FIG. 9B is a series of plots of spring force as a function of engine load for various embodiments of the present invention.

DETAILED DESCRIPTION

[0068] Certain exemplary embodiments will now be described to provide an overall understanding of the principles of the structure, function, manufacture, and use of the devices and methods disclosed herein. One or more examples of these embodiments are illustrated in the accompanying drawings. Those skilled in the art will understand that the devices and methods specifically described herein and illustrated in the accompanying drawings are non-limiting exemplary embodiments and that the scope of the present invention is defined solely by the claims. The features illustrated or described in connection with one exemplary embodiment may be combined with the features of other embodiments. Such modifications and variations are intended to be included within the scope of the present invention.

[0069] Although certain methods and devices are disclosed herein in the context of a split-cycle engine and/or an air hybrid engine, a person having ordinary skill in the art will appreciate that the methods and devices disclosed herein can be used in any of a variety of contexts, including, without limitation, non-hybrid engines, two-stroke and four-stroke engines, conventional engines, diesel engines, etc.

[0070] In order to operate an engine at maximum efficiency, it is desirable to vary the force applied by the valve spring or springs acting upon the various engine valves. For example, efficiency gains can be realized by varying the spring force based on engine speed. In split-cycle engines (such as the engine 100 detailed above), the dynamic actuation of the crossover valves (i.e., 114, 116) is very demanding. This is because the crossover valves must achieve sufficient lift to fully transfer the fuel-air charge in a very short period of crankshaft rotation (typically in a range of about 30 to 45 degrees CA) relative to that of a conventional engine, which normally actuates the valves for a period of approximately 180 degrees CA. As a result, the crossover valves are required to actuate about four to six times faster than the valves of a conventional engine. Thus, the valve train and valve springs associated with these valves must be capable of relatively fast actuation rates. As engine speed increases, the speed of the valves likewise increases, generating higher valve acceleration and higher valve inertia. As a result, the force of the valve spring required to overcome the valve inertia increases with engine speed. Efficiency can therefore be improved by using a low spring force when the engine is operating at a low speed, and using a high spring force when the engine is operating at a high speed. Changes in

engine speed are relatively slow, and therefore the valve spring force can be adjusted gradually to compensate for changes in engine speed.

[0071] Efficiency gains can also be realized by adjusting valve spring force based on engine load. In the split-cycle engine 100 detailed above, the crossover valves 114, 116 are outwardly-opening and therefore the valve springs associated therewith must exert enough force to hold the crossover valves closed during at least the peak pressure portion of the compression and expansion strokes of the engine. Since the peak combustion pressure in the expansion cylinder increases with engine load, the force requirement of the XovrE valve spring likewise increases. Efficiency can therefore be improved by using a low spring force when the engine is operating under a relatively low load, and using a high spring force when the engine is operating at a relatively high load. Changes in engine load can occur almost instantaneously, for example when the throttle is suddenly opened to a full throttle position or suddenly closed. Accordingly, the valve spring force must likewise be adjustable almost instantaneously in order to successfully vary spring force based on load. A variety of adjustable valve springs is disclosed herein, including some that are gradually adjustable, some that are substantially instantaneously adjustable, and some that are both gradually and substantially instantaneously adjustable.

[0072] FIGS. 2A-2B illustrate one exemplary embodiment of a valve train 200 that includes a variable force valve spring 208. Although the operation of the valve train 200 is described with respect to an outwardly-opening poppet valve, it will be appreciated that it can be readily adapted for use with almost any engine valve, including inwardly-opening poppet valves.

[0073] As shown in FIG. 2A, the valve train 200 generally includes a cam 202, a rocker 204, a valve 206, and a variable force valve spring 208. The valve train 200 also includes one or more associated support elements which, for purposes of brevity, are not illustrated.

[0074] The valve 206 comprises a valve head 210 and a valve stem 212 extending vertically from the valve head 210. A valve adapter assembly 214 is disposed at the tip of the stem 212 opposite the head 210 and is securely fixed thereto. The valve adapter assembly 214 includes an adapter portion 215 and a collet 217 disposed therein.

[0075] The rocker 204 comprises a forked rocker pad 220 at one end, which straddles the valve stem 212 and engages the underside of the valve adapter assembly 214. Additionally, the rocker 204 includes a solid rocker pad 222 at an opposing end, which slidably contacts the cam 202. The rocker 204 pivots about a rocker shaft 224 extending therethrough. The forked rocker pad 220 of the rocker 204 contacts the valve adapter assembly 214 of the outwardly-opening poppet valve 206 such that a downward direction of the rocker pad 222 caused by the actuation of the cam 202 translates into an upward movement of the rocker pad 220, which in turn opens the valve 206.

[0076] The cam 202 is a “dwell cam,” which as used herein is a cam that includes a dwell section (i.e., a section of the eccentric portion of the cam having a constant radius) of at least 5 degrees CA. For purposes herein, the dwell section is referred to as being part of the eccentric portion of the cam, even though the dwell section is concentric with the base circle portion of the cam in the illustrated embodiment. In the illustrated embodiment, the dwell cam 202 rotates clockwise (in the direction of the arrow A1). As an eccentric portion 226 of the cam 202 contacts the rocker 204, the rocker 204 rotates about the rocker shaft 224 to lift the valve 206 off of its seat 216.

[0077] As will be appreciated by those having ordinary skill in the art, the valve train 200 can optionally include a lost-motion system for selectively operatively disconnecting and connecting the rocker from the cam, the rocker from the valve, etc. For example, the rocker 204 can be mounted on a collapsible hydraulic tappet that can be selectively drained of and filled with hydraulic liquid to dynamically change the pivot height of the rocker 204. Other examples of lost-motion systems are described at length in U.S. Application Serial No. 13/359,521, entitled “LOST-MOTION VARIABLE VALVE ACTUATION SYSTEM WITH CAM PHASER” filed on an even date herewith, which is hereby incorporated by reference in its entirety.

[0078] The valve spring 208 biases the valve 206 towards a closed position and holds the valve head 210 securely against the valve seat 216 when the valve 206 is fully closed. The valve spring 208 also provides sufficient closing force to collapse a lost-motion valve train element when valve closing control is requested. As shown, the valve spring 208 includes an outer housing 228 that defines a cylindrical vessel 230 therein. A piston 232 coupled to the engine

valve 206 is reciprocally positioned within the vessel 230. The piston 232 is directly coupled to the engine valve 206 (i.e., to the valve stem 212) or is coupled thereto via one or more intermediate elements, including for example the valve adapter assembly 214. In one embodiment, the piston has a diameter of about 13 mm. The piston 232 includes one or more sealing features formed on or mounted to a sidewall thereof to facilitate a fluid-tight sealing engagement between the piston 232 and the housing 228. It will be appreciated that the term “fluid” as used herein includes compressible gasses, such as air, nitrogen, and the like. In this manner, the piston 232 and housing 228 together define a variable-volume fluid chamber 234. The housing 228 includes a fluid port 236 for supplying or removing fluid to or from the chamber 234. In FIG. 2A, the valve 206 is shown in a closed position. When the valve is so-positioned, the piston 232 is substantially at the bottom of its stroke within the vessel 230 and the chamber 234 has a maximum volume.

[0079] As shown in FIG. 2B, when the cam 202 rotates clockwise, as a camshaft to which it is mounted is driven by rotation of the engine's crankshaft, the eccentric portion 226 of the cam 202 imparts a downward motion to the rocker 204. This results in a counter-clockwise rotation of the rocker 204, which in turn is effective to lift the valve 206 off of the seat 216, thereby opening the valve. As the valve 206 is lifted, the piston 232 coupled thereto slides upwards within the vessel 230 of the valve spring 208, reducing the volume of the chamber 234. Any fluid disposed in the chamber 234 is either compressed or forced out of the chamber 234 through the fluid port 236. When the valve 206 is fully opened as shown in FIG. 2B, the piston 232 is substantially at the top of its stroke within the vessel 230 and the chamber 234 has a minimum volume.

[0080] FIGS. 3A-3B illustrate one embodiment of a fluid control assembly 300 for controlling a spring force of the valve spring 208. The fluid control assembly generally includes a control valve 338, a high pressure source 340, and a pressure regulator 344 that supplies a low pressure source 345. In the illustrated embodiment, the control valve 338 is in the form of a high-speed, solenoid-actuated spool valve, though a variety of valves known in the art can also be employed without departing from the scope of the present invention. The high pressure source 340 is coupled via a first fluid line 341 to a first input 346 of the control valve 338. The high pressure source 340 is supplied by an air compressor operatively coupled to the engine in which the valve

spring 208 is installed. The high pressure source 340 can also be supplied from the crossover passage or the air hybrid reservoir of the engine in which the valve spring 208 is installed. U.S. Patent Publication No. 2010/0282225 entitled "AIR SUPPLY FOR COMPONENTS OF A SPLIT-CYCLE ENGINE," filed on May 7, 2010, provides further details on generating a high pressure source from the crossover passage of a split-cycle engine, and is incorporated herein by reference in its entirety. In some embodiments, the high pressure source has a pressure of at least about 20 bar, at least about 30 bar, and/or at least about 40 bar.

[0081] The high pressure source 340 is also fed through the pressure regulator 344, where it is converted into a comparatively low pressure source 345 (e.g., a second fluid line, a low pressure reservoir, or the like) and coupled to a second input 348 of the control valve 338. The regulator 344 can supply the low pressure source 345 at a variety of pressures, and can optionally be adjustable, electronically or otherwise. In some embodiments, the low pressure source 345 can have a pressure of less than about 1 bar, less than about 5 bar, and/or less than about 20 bar. For example, the low pressure source can have a pressure between about 1 bar and about 20 bar, between about 1 bar and about 10 bar, between about 5 bar and about 10 bar, etc.

[0082] The control valve 338 also includes an output 352 coupled to the fluid port 236 of the valve spring 208 shown in FIGS. 2A-2B.

[0083] In use, an electric current is selectively applied to a solenoid disposed within a solenoid body 366 of the control valve 338 to impart a linear pushing or linear pulling motion to a spool 356. The electric current is selectively supplied from or at the request of a control module (i.e., an engine control computer). The control module or engine control computer includes a microprocessor coupled to a memory and executes one or more programs for controlling various aspects of an engine's operation, including, without limitation, engine valve timing, valve spring rate, air hybrid operating mode, etc. The spool 356 is sized to reciprocate within the body 358 of the control valve 338 and to form a sealing engagement with an interior surface thereof. The spool 356 includes first and second end portions 360, 362 that form a seal with the surrounding valve body 358. The first and second end portions 360, 362 are joined by a connecting element 364 which has a smaller diameter than the first and second end portions 360, 362 such that fluid can flow between the connecting element 364 and the interior surface of the control valve body

358. The connecting element 364 can also include one or more apertures to facilitate fluid flow therethrough.

[0084] In FIG. 3A, the spool 356 is illustrated in a first position in which the first input 346 of the control valve 338 is in fluid communication with the output 352. Accordingly, the high pressure source 340 is in fluid communication with the fluid port 236 of the valve spring 208. Since the second input 348 is blocked by the spool 356, the fluid chamber 234 of the valve spring 208 is supplied only by the high pressure source 340. In this position of the control valve 338, the valve spring 208 can be said to have a “high” spring force, as the valve spring 208 will bias the engine valve 206 towards a closed position under the relatively high pressure of the high pressure source 340. In this position, more force and more energy is required to open the engine valve 206 and to hold the engine valve 206 open. In addition, this position causes the valve 206 to close faster.

[0085] Referring now to FIG. 3B, the solenoid can be actuated to push the spool 356 downwards away from the solenoid body 366. As the spool 356 is advanced, the first and second inputs 346, 348 are blocked by the spool 356, thereby isolating the output 352 and sealing the fluid chamber 234.

[0086] As shown in FIG. 3C, the spool 356 can be advanced further downwards away from the solenoid body 366. In this position, the second input 348 of the valve 338 is placed in fluid communication with the output 352 while the first input 346 is blocked by the spool 356. In this position, the fluid chamber 234 of the valve spring 208 is charged with low pressure fluid from the pressure regulator 344. In this position of the control valve 338, the valve spring 208 can be said to have a “low” spring force, as the valve spring 208 will bias the engine valve 206 towards a closed position under the relatively low pressure of the regulated pressure source 345. In this position, considerably less force and less energy is required to open the engine valve 206 and to hold the engine valve 206 open than would be required with the control valve 338 in the position shown in FIG. 3A. In addition, this position causes the valve 206 to close more slowly than when the control valve 338 is in the position shown in FIG. 3A.

[0087] In use, the control valve 338 is selectively actuated (e.g., under the control of an engine computer or control module) to vary the spring force of the valve spring 208. For example, the

valve spring 208 can be set to a “high” spring force by simply holding the control valve 338 in the position shown in FIG. 3A. Similarly, the valve spring 208 can be set to a “low” spring force by holding the control valve 338 in the position shown in FIG. 3C. Alternatively, the spring force of the valve spring 208 can be adjusted to any of a variety of levels between the “high” and “low” settings by manipulating the control valve 338. For example, the pressure in the fluid chamber 234 can be maintained at a particular level by positioning the control valve 338 as shown in FIG. 3B to effectively isolate and seal off the fluid chamber 234. To raise the pressure in the fluid chamber 234 slightly from the current level, the control valve 338 can be moved briefly to the position shown in FIG. 3A. Once the desired pressure is reached, which in some embodiments can be indicated by a pressure sensor configured to measure the pressure within the fluid chamber 234, the control valve can be moved back to the position shown in FIG. 3B to maintain the valve spring 208 at the new pressure level. Likewise, to lower the pressure in the fluid chamber 234 slightly from the current level, the control valve 338 can be moved briefly to the position shown in FIG. 3C. Once the desired pressure is reached (e.g., as indicated by an output of a pressure sensor), the control valve 338 can be moved back to the position shown in FIG. 3B to maintain the valve spring 208 at the new pressure level. Thus, by momentarily moving the control valve 338 up or down from the position shown in FIG. 3B, slight increases or decreases in valve spring pressure can be effected. These changes can be dictated by the engine control computer, based at least in part on a feedback loop provided by a pressure sensor configured to measure the pressure within the fluid chamber 234.

[0088] FIG. 4 illustrates another embodiment of a fluid control assembly 400 for controlling the spring force of the valve spring 208. The fluid control assembly 400 generally includes a high pressure source 440, a continuously-adjustable pressure regulator 444, and a release valve 468. The pressure regulator 444 is configured to convert the high pressure source 440 to a pressure-regulated output source 445 having any of a variety of pressures in a prescribed range. The regulator 444 can be configured to generate an output source of fluid at a pressure of about 1 bar to about 100 bar, about 1 bar to about 85 bar, about 1 bar to about 20 bar, and/or about 10 bar to about 20 bar. The regulator 444 can be electronically adjustable, and/or can be hydraulically, magnetically, mechanically, and/or manually adjustable. It will be appreciated that the regulator 444 permits the pressure supplied to the fluid chamber 234 of the valve spring 208 to be varied across a wide range of pressures. This advantageously permits dynamic control of the spring

force of the valve spring 208. The release valve 468 can be selectively actuated to release pressurized fluid stored in the fluid chamber 234. For example, the release valve 468 can be electronically actuated and can be configured to open briefly when instructed by an engine control computer or control module to release the pressurized charge from the fluid chamber 234. In one embodiment, the release valve 468 is an integral component of the regulator 444 and is configured to release stored pressure automatically when the regulator 444 is actuated to reduce the output pressure.

[0089] In use, the regulator 444 is selectively adjusted (e.g., under the control of an engine computer or control module) to vary the spring force of the valve spring 208.

[0090] FIGS. 5A-5B illustrate another embodiment of a valve spring according to the present invention for use with inwardly-opening engine valves (i.e., engine valves that open into the cylinder and towards the piston). As shown, the valve spring 508 biases the valve 506 towards a closed position and holds the valve head 510 securely against a valve seat 516 when the valve 506 is fully closed. The valve spring 508 also provides sufficient closing force to collapse a lost-motion valve train element when valve closing control is requested. The valve spring 508 includes an outer housing 528 that defines a cylindrical vessel 530 therein. A piston 532 coupled to the engine valve 506 is positioned within the vessel 530 such that it reciprocates therein. The piston 532 can be directly coupled to the engine valve 506 (i.e., to the valve stem 512) or can be coupled thereto via one or more intermediate elements. The piston 532 includes one or more sealing features formed on or mounted to a sidewall thereof to facilitate a fluid-tight sealing engagement between the piston 532 and the housing 528. In this manner, the piston 532 and housing 528 together define a variable-volume fluid chamber 534. The housing 528 includes a fluid port 536 for supplying or removing fluid to or from the chamber 534. The piston 532 also includes an engagement portion 533 extending therefrom for engaging the cam 502 or an intermediate valve train element. In FIG. 5A, the valve 506 is shown in a closed position. When the valve is so-positioned, the piston 532 is substantially at the top of its stroke within the vessel 530 and the chamber 534 has a maximum volume.

[0091] As shown in FIG. 5B, when the cam 502 rotates clockwise as a camshaft to which it is mounted is driven by rotation of the engine's crankshaft, an eccentric portion 526 of the cam 502

imparts a downward motion to the piston 532, which in turn is effective to lift the valve 506 off of the seat 516. As the valve 506 is lifted, the piston 532 slides downwards within the vessel 530 of the valve spring 508, reducing the volume of the chamber 534. Any fluid disposed in the chamber 534 is either compressed or forced out of the chamber 534 through the fluid port 536. When the valve 506 is opened as shown in FIG. 5B, the piston 532 is substantially at the bottom of its stroke within the vessel 530 and the chamber 534 has a minimum volume.

[0092] It will be appreciated that any of the fluid control systems described herein (e.g., the fluid control systems 300, 400) can be coupled to the fluid port 536 as described above to dynamically control the spring force of the valve spring 508 and thereby improve the efficiency of an engine in which it is installed. For example, the fluid control systems 300, 400 can be used to selectively vary the pressure of a fluid charge in the chamber 534, thereby varying the force of the valve spring 508.

[0093] FIGS. 6A-6B illustrate another embodiment of a valve spring 608 according to the present invention in which the filling volume of the valve spring's fluid chamber can be altered to change the peak spring force of the valve spring at peak valve lift, even though the filling pressure (pressure to which the valve spring's fluid chamber is filled when the valve is in its closed position on its valve seat) is held substantially constant. The valve spring 608 biases an engine valve 606 towards a closed position and holds the valve 606 securely against a valve seat 616 when the valve 606 is fully closed. The valve spring 608 also provides sufficient closing force to collapse a lost-motion valve train element when valve closing control is requested. As shown, the valve spring 608 includes a sliding cap 628 that defines a cylindrical vessel 630 therein. A piston 632 coupled to the engine valve 606 is reciprocally positioned within the vessel 630. In addition, the sliding cap 628 is reciprocally disposed within a cylindrical vessel 631 formed in an outer housing 629. In this manner, the piston 632 and sliding cap 628 together define a first fluid chamber 634. An aperture 636 is provided in the sliding cap 628 for supplying or removing fluid to or from the first fluid chamber 634. In some embodiments, the fluid supplied to or removed from the first fluid chamber 634 is a substantially compressible fluid (e.g., air or nitrogen). The sliding cap 628 and the outer housing 629 together define a second fluid chamber 635. The outer housing 629 includes a fluid port 637 for supplying or removing fluid to or from the second fluid chamber 635. The fluid supplied to or removed from

the second fluid chamber 635 can be a substantially compressible fluid (e.g., air or nitrogen), a substantially incompressible fluid (e.g., oil), or some combination thereof.

[0094] In use, the first fluid chamber 634 remains coupled to a pressurized fluid supply via the aperture 636. In one embodiment, the filling pressure (e.g., the pressure in the first fluid chamber 634 when the valve 606 is closed long enough to equalize the supply pressure and the pressure in the first fluid chamber 634) is about 20 bar.

[0095] In FIG. 6A, the valve spring 608 is shown in a “high” force configuration in which the amount of peak pressure, and therefore peak spring force exerted by the valve spring 608 when the valve 606 is in its peak lift position (i.e., peak valve lift) to close the engine valve 606, is greatest. In this configuration, the second fluid chamber 635 is pressurized such that the force pushing the sliding cap 628 down with respect to the valve 606 (e.g., towards the valve seat 616) exceeds the force pushing the sliding cap 628 up with respect to the valve 606 (e.g., away from the valve seat 616). As a result, the sliding cap 628 is moved to or held at the bottom of its stroke within the outer housing 629 against a mechanical stop (not shown). It will be appreciated that the pressure in the second fluid chamber 635 need not necessarily be greater than the pressure in the first fluid chamber 634 for this to occur, as the surface area of the sliding cap 628 that is exposed to the second fluid chamber 635 exceeds the surface area of the sliding cap 628 that is exposed to the first fluid chamber 634. Thus, if both the first and second fluid chambers 634, 635 are pressurized to the same pressure (e.g., 20 bar), the sliding cap 628 will still move to, or be held at, the bottom of its stroke within the outer housing 629. In this configuration, the first fluid chamber 634 has a minimum filling volume (e.g., the volume of the first fluid chamber 634 when the valve 606 is fully closed).

[0096] When the valve 606 is opened, the piston 632 is forced upwards relative to the sliding cap 628, causing the pressure in the first fluid chamber 634 to increase. As a result, fluid in the first fluid chamber 634 flows back through the aperture 636 and into the fluid supply. The aperture 636 is sized to choke the flow of fluid back into the supply such that enough fluid is maintained in the first fluid chamber 634 to maintain the requisite closing pressure. In one embodiment, the peak pressure in the first fluid chamber 634 when the valve 606 is opened is 45 bar and the supply pressure is 20 bar.

[0097] When the valve 606 is no longer held open by the valve train, the pressure in the first fluid chamber 634 forces the piston 632 downwards relative to the sliding cap 628, thereby closing the valve 606. As the piston 632 slides downwards and the volume of the first fluid chamber 634 increases, the first fluid chamber 634 is refilled through the aperture 636 until the pressure therein matches the supply pressure. In other words, the aperture 636 is sized such that the chamber pressure and the supply pressure equalize during the period of cam rotation during which the valve 606 is closed.

[0098] When desired, the peak spring force exerted by the valve spring 608 at peak valve lift can be decreased by transitioning the valve spring 608 to the configuration shown in FIG. 6B. In FIG. 6B, the valve spring 608 is shown in a “low” force configuration in which the amount of peak pressure, and therefore force exerted by the valve spring 608 at peak valve lift, to close the engine valve 606, is at a minimum. In this configuration, the pressure in the second fluid chamber 635 is released, allowing the sliding cap 628 to slide upwards relative to the valve 606 (e.g., away from the valve seat 616) to the top of its stroke within the outer housing 629 against a mechanical stop (not shown). The pressure in the second fluid chamber 635 can be released in a variety of ways, for example by switching one or more control valves such that the fluid port 637 is disconnected from the pressurized supply and instead connected to a lower pressure source or vented to ambient. When the pressure in the second fluid chamber 635 is released, the sliding cap 628 is urged upwards by the pressure in the first fluid chamber 634 and/or by an optional mechanical bias spring (not shown). With the sliding cap 628 positioned at the top of its stroke within the outer housing 629, the first fluid chamber 634 has a maximum filling volume and the force exerted by the valve spring 608 at peak valve lift is at a minimum. When it is desired to again increase the force exerted by the valve spring 608 at peak valve lift, the second fluid chamber 635 is re-pressurized, returning the valve spring 608 to the configuration shown in FIG. 6A.

[0099] Since the lift of the engine valve 606 remains the same, increasing the filling volume of the first fluid chamber 634 is effective to decrease the ratio by which the fluid disposed therein must be compressed when the valve is opened. In other words, even if the supply pressure to the chamber 634 is kept constant, the spring force at peak valve lift is decreased as the filling volume is increased. Thus, the force exerted by the valve spring 608 at peak valve lift is a function of the

filling volume of the first fluid chamber 634, and decreasing the filling volume (as shown in FIG. 6A) increases the force exerted by the valve spring 608 at peak valve lift, while increasing the filling volume (as shown in FIG. 6B) decreases the force exerted by the valve spring 608 at peak valve lift.

[00100] It will be appreciated that, in the embodiment of FIGS. 6A-6B, pressure can be supplied to or removed from the second fluid chamber 635 substantially instantaneously (e.g., within one engine cycle). Accordingly, the spring force can be altered substantially instantaneously, which as noted above is desirable in situations where the spring force must be altered to reflect the engine load.

[00101] It can also be desirable, however, to adjust the spring force gradually such that it is proportional to parameters that also change gradually, such as engine speed. (As noted above, engine speed generally changes gradually due to the inertia of the flywheel and/or vehicle in which the engine is installed). Thus, in the embodiment of FIGS. 6A-6B, the spring pressure can also be adjusted gradually by adjusting the supply pressure supplied to the first and second fluid chambers 634, 635 (e.g., using the fluid control devices 300, 400 discussed above).

[00102] FIG. 7 illustrates another embodiment of a valve spring 708 according to the present invention in which the spring force can be adjusted by increasing or decreasing the aggregate surface area over which fluid pressure is coupled to the engine valve 706. The valve spring 708 biases the valve 706 towards a closed position and holds the valve head 710 securely against a valve seat 716 when the valve 706 is fully closed. The valve spring 708 also provides sufficient closing force to collapse a lost-motion valve train element when valve closing control is requested. As shown, the valve spring 708 includes first and second outer housings 728A, 728B that define respective cylindrical vessels 730A, 730B therein. First and second pistons 732A, 732B are reciprocally positioned within the vessels 730A, 730B, respectively, thereby defining first and second fluid chambers 734A, 734B, respectively. The first and second fluid chambers 734A, 734B are coupled to a pressurized supply via first and second apertures 736A, 736B. The first piston 732A is directly coupled to the valve stem 712 of the engine valve 706 by a collet 717. The second piston 732B is coupled to an extension stem 719.

[00103] The proximal end 721 of the valve stem 712 and the distal end 723 of the extension stem 719 are slidably received in a third housing 728C that defines a hydraulic plenum 725. A stop 701 is provided in the third housing 728C to limit the travel of the extension stem 719 in the valve closing direction. It will be appreciated that the stop 701 can also be positioned elsewhere in the system, such as beneath the second piston 732B. The plenum 725 is coupled via a control valve 727 to a hydraulic accumulator 715, which is coupled via a check valve 703 to a low pressure fluid supply 705 (e.g., the engine oil supply at a pressure of 2-4 bar). The control valve 727 is actuated under the direction of an engine control computer using any of a variety of actuation techniques known in the art, including hydraulic, electrical, pneumatic, mechanical, and/or magnetic actuation. The check valve 703 allows one-way flow of fluid from the fluid supply 705 to the accumulator 715, for example to provide make-up for piston leakage.

[00104] In use, the spring force of the valve spring 708 can be varied by opening and closing the control valve 727. When a reduced spring force is required, the control valve 727 is opened, placing the accumulator 715 in fluid communication with the plenum 725. When the engine valve 706 is opened, the valve stem 712 slides further into the plenum 725, forcing hydraulic fluid through the control valve 727 and into the accumulator 715. Since the force required to fill the accumulator 715 is less than the force required to move the second piston 732B relative to the second housing 728B, the extension stem 719 does not move and instead remains at the bottom of its stroke within the plenum 725. Accordingly, only the pressure in the first fluid chamber 734A and the relatively small pressure supplied by the accumulator 715 act on the engine valve 706. When the engine valve 706 is subsequently closed and the valve stem 712 is partially withdrawn from the plenum 725, the extension stem 719 rests against the stop 701 and the accumulator 715 forces hydraulic fluid back through the open control valve 727 to refill the plenum 725.

[00105] When an increased spring force is required, the control valve 727 is closed, locking a volume of hydraulic fluid in the plenum 725. When the engine valve 706 is opened and the valve stem 712 is advanced into the plenum 725, the relatively incompressible hydraulic fluid can no longer escape through the control valve 727 into the accumulator 715, and instead the extension stem 719 is forced upwards. As a result, the pressure in both the first and second fluid chambers 734A, 734B acts on the valve 706, effectively increasing the spring force.

[00106] Since the spring force is a function of the aggregate surface area over which fluid pressure is coupled to the valve 706, the degree to which the spring force is adjusted can be controlled in a variety of ways. For example, the size of the second piston 732B and/or the second fluid chamber 734B can be varied to adjust the degree to which the spring force changes when the second piston 732B is engaged. Alternatively, or in addition, the plenum 725 can be coupled to a plurality of extension stems, each having its own respective control valve and associated piston/fluid chamber combination. In such embodiments, the number of control valves that are opened or closed can be controlled to provide more granularity in spring force adjustment. For example, when only a small increase in spring force is required, a first control valve can be closed to couple the engine valve to a single extension stem. If more spring force is required, two or more control valves can be closed to couple the engine valve to multiple extension stems.

[00107] FIGS. 8A and 8B illustrate valve spring pressure requirements as a function of engine speed for an exemplary split-cycle engine. As shown in FIG. 8A, the pressure in the valve spring required to overcome the momentum of the valve need only be about 1-2 bar at the 660 rpm idle speed of the engine. As engine speed increases, however, the valve momentum increases and the valve spring pressure required to overcome that momentum also increases. For example, at 4000 rpm, 20 bar of valve spring pressure is required to maintain the desired valve opening and closing characteristics. FIG. 8B shows the spring pressure requirement normalized to the idle spring pressure requirement. In other words, the spring pressure requirement at 2000 rpm is 10 times the spring pressure requirement at idle, the spring pressure requirement at 3000 rpm is 20 times the spring pressure requirement at idle, etc.

[00108] The methods and devices disclosed herein can be used to adjust the spring force of a valve spring based on a variety of parameters, including engine speed, engine load, throttle position, engine temperature, ambient temperature, ambient pressure, intake air pressure, intake air temperature, exhaust temperature, engine age, and the like. The spring force can also be adjusted based on a manual setting by a user (e.g., a control switch on a vehicle's dashboard that permits a user to select an "economy mode" or a "sport mode" of engine operation). FIGS. 9A-9B illustrate various exemplary embodiments of methods for varying the spring force of a valve spring based on such parameters.

[00109] FIG. 9A illustrates spring force as a function of engine speed when various valve spring systems disclosed herein are used to adjust the spring force based on the speed of the engine. The first plot 900 illustrates the spring force when the valve spring 208 of FIGS. 2A-2B is used with the fluid control system 300 of FIGS. 3A-3C in a first mode of operation. As shown, the control valve 338 remains in the position shown in FIG. 3C such that the fluid chamber 234 of the valve spring 208 is in fluid communication with the regulated low pressure source 345 until a threshold engine speed is reached (about 3000 rpm in the illustrated embodiment). When the threshold is reached, the control valve 338 is actuated to switch from the low pressure source 345 to the high pressure source 340 to increase the spring force of the valve spring 208. In one embodiment, a crank position sensor is employed to detect the speed of the engine. The crank position sensor is coupled to an engine control computer or control module, which in turn selectively actuates the control valve 338 based on the input engine speed. It will be appreciated that any engine speed between the engine's idle speed and the engine's redline speed can be used for the threshold speed. For example, the threshold can be about 1000 rpm, about 3000 rpm, and/or about 5000 rpm. It will also be appreciated that the fluid control system 300 can be used to set the spring forces used when engine speed is below or above the threshold to any of a variety of levels intermediate to the high pressure source 340 and the low pressure source 345, as described above.

[00110] The first plot 900 also typifies the spring force when the valve spring 508 of FIGS. 5A-5B, the valve spring 608 of FIGS. 6A-6B, or the valve spring 708 of FIG. 7 is used. For example, when the predetermined speed threshold is reached, the fluid port 637 of the valve spring 608 of FIGS. 6A-6B can be coupled to a pressurized source. This allows pressure to build in the second fluid chamber 635, thereby reducing the volume of the first fluid chamber 634 and generating a sudden increase in spring force of the valve spring 608. Similarly, the normally-open control valve 727 of the valve spring 708 of FIG. 7 can be closed when the speed threshold is reached, effectively coupling the valve stem 712 to the extension stem 719 and rapidly increasing the spring force of the valve spring 708.

[00111] The second plot 902 illustrates the spring force when the valve spring 208 of FIGS. 2A-2B is used with the fluid control system 400 of FIG. 4. As shown, the regulator 444 remains set to a minimum output pressure 445 when the engine is idling and/or operating below a certain

threshold speed (about 1200 rpm in the illustrated embodiment). As the engine speed increases beyond that threshold, the regulator 444 is adjusted to increase the output pressure 445 supplied to the fluid chamber 234 of the valve spring 208, thereby increasing the spring force of the valve spring. The output pressure 445 is increased to a degree commensurate with the present engine speed until a second threshold is reached (about 4800 rpm in the illustrated embodiment), at which point the regulator 444 is set to the maximum output pressure. When the engine speed decreases, the release valve 468 is selectively actuated to release pressure from the fluid charge in the chamber 234, while the regulator 444 is dialed down to a lower output pressure. As described above, a crank position sensor can be employed to detect the speed of the engine. The crank position sensor is coupled to an engine control computer or control module, which in turn selectively actuates the release valve 468 and/or the regulator 444 based on the input engine speed.

[00112] The second plot 902 is also illustrative of the spring force when the valve spring 208 of FIGS. 2A-2B is used with the fluid control system 300 of FIGS. 3A-3C in a second mode of operation, or when the valve spring 508 of FIGS. 5A-5B, the valve spring 608 of FIGS. 6A-6B, or the valve spring 708 of FIG. 7 is used. For example, the control valve 338 of the fluid control system 300 of FIGS. 3A-3C can be manipulated to gradually increase the pressure in the fluid chamber 234 when a speed threshold is reached. By way of further example, when a speed threshold is reached, the supply pressure to the first and second fluid chambers 634, 635 of the valve spring 608 of FIGS. 6A-6B can be gradually increased to increase the spring force. Similarly, the supply pressure to the first and second fluid chambers 734A, 734B of the valve spring 708 of FIG. 7 can be gradually increased to increase the spring force when a speed threshold is reached.

[00113] It will thus be appreciated that according to the methods of FIG. 9A, the same valve spring can have a relatively low spring force when the engine is idling or operating at low speed, yet also have a relatively high spring force when the engine is operating at high speeds. It is thus possible to save energy that would otherwise be wasted compressing a stiff valve spring when the engine is operating at a low speed, without compromising the valve train's ability to perform at higher engine speeds. The illustrated thresholds and ramp rate are merely exemplary, and any

of a wide range of values can be selected for these parameters without departing from the scope of the present invention.

[00114] As shown in FIG. 9B, the principles discussed above for varying spring force with respect to engine speed can be applied to vary spring force based on engine load. For example, the plot 904 illustrates spring force as a function of engine load when the valve spring 208 of FIGS. 2A-2B is used with the fluid control system 300 of FIGS. 3A-3C in a first mode of operation. As shown, the control valve 338 remains in the position shown in FIG. 3C such that the fluid chamber 234 of the valve spring 208 is in fluid communication with the regulated low pressure source 345 until a threshold engine load is reached (about 45% of the maximum engine load in the illustrated embodiment). When the threshold is reached, the control valve 338 is actuated to switch from the low pressure source 345 to the high pressure source 340 to increase the spring force of the valve spring 208. In one embodiment, the engine load is estimated based on readings from one or more sensors, which can include an intake air flow sensor, a throttle position sensor, an engine speed sensor, and/or a vehicle speed sensor. Any or all of these sensors can be coupled to an engine control computer or control module, which in turn selectively actuates the control valve 338 based on the input sensor readings. It will be appreciated that any engine load or range of engine loads in the range of 0% to 100% can be used for the threshold engine load. The engine load threshold can also be specified relative to the engine load at idle. For example, the engine load threshold can be about twice the idle load, about three times the idle load, about four times the idle load, etc. It will also be appreciated that the fluid control system 300 can be used to set the spring forces used when engine load is below or above the threshold to any of a variety of levels intermediate to the high pressure source 340 and the low pressure source 345, as described above.

[00115] The first plot 904 also typifies the spring force when the valve spring 508 of FIGS. 5A-5B, the valve spring 608 of FIGS. 6A-6B, or the valve spring 708 of FIG. 7 is used. For example, when the predetermined load threshold is reached, the fluid port 637 of the valve spring 608 of FIGS. 6A-6B can be coupled to a pressurized source. This allows pressure to build in the second fluid chamber 635, thereby reducing the volume of the first fluid chamber 634 and generating a sudden increase in spring force of the valve spring 608. Similarly, the normally-open control valve 727 of the valve spring 708 of FIG. 7 can be closed when a load threshold is

reached, effectively coupling the valve stem 712 to the extension stem 719 and rapidly increasing the spring force of the valve spring 708.

[00116] The second plot 906 illustrates the spring force when the valve spring 208 of FIGS. 2A-2B is used with the fluid control system 400 of FIG. 4. As shown, the regulator 444 remains set to a minimum output pressure when the engine is idling and/or the engine load is below a first threshold (about 20% load in the illustrated embodiment). As the engine load increases beyond that threshold, the regulator 444 is adjusted to increase the output pressure supplied to the fluid chamber 234 of the valve spring 208, thereby increasing the spring force of the valve spring. The output pressure is increased to a degree commensurate with the present engine load until a second threshold is reached (about 80% engine load in the illustrated embodiment), at which point the regulator 444 is set to the maximum output pressure. When the engine load decreases, the release valve 468 is selectively actuated to release pressure from the fluid charge in the chamber 234, while the regulator 444 is dialed down to a lower output pressure. As described above, the engine load can be estimated based on readings from one or more sensors, which can include an intake air flow sensor, a throttle position sensor, an engine speed sensor, and/or a vehicle speed sensor. Any or all of these sensors can be coupled to an engine control computer or control module, which in turn selectively actuates the release valve 468 and/or the regulator 444 based on the input sensor readings.

[00117] The second plot 906 is also illustrative of the spring force when the valve spring 208 of FIGS. 2A-2B is used with the fluid control system 300 of FIGS. 3A-3C in a second mode of operation, or when the valve spring 508 of FIGS. 5A-5B, the valve spring 608 of FIGS. 6A-6B, or the valve spring 708 of FIG. 7 is used. For example, the control valve 338 of the fluid control system 300 of FIGS. 3A-3C can be manipulated to gradually increase the pressure in the fluid chamber 234 when a load threshold is reached. By way of further example when a load threshold is reached, the supply pressure to the first and second fluid chambers 634, 635 of the valve spring 608 of FIGS. 6A-6B can be gradually increased to increase the spring force. Similarly, the supply pressure to the first and second fluid chambers 734A, 734B of the valve spring 708 of FIG. 7 can be gradually increased to increase the spring force when a load threshold is reached.

[00118] It will thus be appreciated that according to the methods of FIG. 9B, the same valve spring can have a relatively low spring force when the engine is idling or operating under a low load, yet also have a relatively high spring force when the engine is operating under a high load. It is thus possible to save energy that would otherwise be wasted compressing a stiff valve spring when the engine is operating under a low load, without compromising the valve train's ability to perform under higher engine loads. The illustrated thresholds and ramp rate are merely exemplary, and any of a wide range of values can be selected for these parameters without departing from the scope of the present invention.

[00119] The engines disclosed herein can be configured to operate reliably over a broad range of engine speeds. In certain embodiments, engines according to the present invention are capable of operating at a speed of at least about 4000 rpm, and preferably at least about 5000 rpm, and more preferably at least about 7000 rpm.

[00120] Although the invention has been described by reference to specific embodiments, it should be understood that numerous changes may be made within the spirit and scope of the inventive concepts described. For example, one or both of the crossover valves can be actuated by a cam having no dwell section or can be actuated using a cam-less system. Also, one or both of the crossover valves can be inwardly-opening. There can also be more than two crossover valves, and more than one crossover passage. The intake and exhaust valves, and any other valve in the engine for that matter, can also include a valve spring as described herein with respect to the crossover valves. The engines disclosed herein are not limited to having only two cylinders. Accordingly, it is intended that the invention not be limited to the described embodiments, but that it have the full scope defined by the language of the following claims.

What is claimed is:

CLAIMS:

1. An engine, comprising:
a pneumatic valve spring coupled to an engine valve; and
a control module configured to adjust a spring force of the pneumatic valve spring based on at least one of engine speed and engine load.
2. The engine of claim 1, wherein the control module is configured to increase the spring force when the engine speed exceeds a first speed threshold.
3. The engine of claim 1, wherein the control module is configured to increase the spring force when the engine load exceeds a first load threshold.
4. The engine of claim 1, wherein the control module is configured to decrease the spring force when the engine speed is below a first speed threshold.
5. The engine of claim 1, wherein the control module is configured to decrease the spring force when the engine load is below a first load threshold.
6. The engine of claim 2, wherein the first speed threshold is 3000rpm.
7. The engine of claim 2, wherein the first speed threshold is four times higher than an idle speed of the engine.
8. The engine of claim 3, wherein the first load threshold is 50 percent of a maximum load of the engine.
9. The engine of claim 3, wherein the first load threshold is three times higher than an idle load of the engine.
10. A valve spring system, comprising:

a vessel having a piston reciprocally disposed therein;
an engine valve coupled to the piston;
a first source of pressurized air having a first pressure;
a second source of pressurized air having a second pressure that is greater than the first pressure; and

a control valve configured to adjust a force exerted on the piston by controlling a first fluid communication between the vessel and the first source, and a second fluid communication between the vessel and the second source.

11. The valve spring system of claim 10, wherein the engine valve is actuated by a lost motion system.

12. The valve spring system of claim 10, wherein the engine valve is outwardly-opening.

13. The valve spring system of claim 10, wherein the engine valve is a crossover valve in a split-cycle engine.

14. The valve spring system of claim 13, wherein the second source is supplied from a crossover passage of the split-cycle engine.

15. The valve spring system of claim 13, wherein a first force exerted on the engine valve by a crossover charge in the crossover passage and a second force exerted on the engine valve by the piston together exceed a combustion force exerted on the engine valve.

16. The valve spring system of claim 10, wherein the first pressure is between about 1 bar and about 10 bar.

17. The valve spring system of claim 10, wherein the second pressure is between about 20 bar and about 85 bar.

18. The valve spring system of claim 10, further comprising a control module configured to actuate the control valve.
19. An engine comprising the valve spring system of claim 10.
20. A valve spring system, comprising:
 - a cap defining a first bore therein;
 - a piston coupled to an engine valve, the piston being slidably disposed within the first bore such that the piston and the cap define a first fluid chamber;
 - an outer housing defining a second bore therein, the cap being slidably disposed within the second bore such that the cap and the outer housing define a second fluid chamber; and
 - a fluid port configured to selectively release pressurized fluid from the second fluid chamber, thereby allowing the cap to slide relative to the outer housing such that a volume of the first fluid chamber is increased.
21. The valve spring system of claim 20, wherein the cap includes at least one aperture in fluid communication with the first fluid chamber.
22. The valve spring system of claim 21, wherein the aperture remains in fluid communication with a pressurized fluid supply line throughout a full engine cycle.
23. The valve spring system of claim 21, wherein fluid disposed in the first fluid chamber is forced out of the first fluid chamber through the aperture when the engine valve is open.
24. The valve spring system of claim 21, wherein the first fluid chamber is filled with fluid through the aperture when the engine valve is closed.
25. The valve spring system of claim 20, wherein the pressure in the first fluid chamber is substantially equal to the pressure in the second fluid chamber when the engine valve is closed for at least 100 degrees crank angle.

26. A method of varying a spring force exerted on an engine valve by the valve spring system of claim 20, comprising:

coupling the fluid port to a vent to release pressure from the second fluid chamber and increase the spring force exerted on the engine valve; and

coupling the fluid port to a pressurized fluid supply to increase pressure in the second fluid chamber and decrease the spring force exerted on the engine valve.

27. The method of claim 26, wherein the spring force is increased when a load of the engine exceeds a first predetermined threshold and is decreased when the load of the engine is below a second predetermined threshold.

28. The method of claim 26, wherein the spring force is increased when a speed of the engine exceeds a first predetermined threshold and is decreased when the speed of the engine is below a second predetermined threshold.

29. A valve spring system, comprising:

a first housing defining a first bore therein in which a first piston is reciprocally disposed, the first piston being coupled to a valve stem of an engine valve;

a second housing defining a second bore therein in which a second piston is reciprocally disposed, the second piston being coupled to an extension stem;

a third housing defining a hydraulic plenum, the hydraulic plenum being in fluid communication with a proximal end of the valve stem, a distal end of the extension stem, and a control valve;

wherein the plenum is in fluid communication with an accumulator when the control valve is opened such that the valve stem is movable independently from the extension stem; and

wherein the plenum is sealed when the control valve is closed such that movement of the valve stem requires movement of the extension stem.

30. The valve spring system of claim 29, wherein the first housing, the second housing, and the third housing are formed integrally.

31. The valve spring system of claim 29, wherein a first fluid chamber defined by the first housing and the first piston is supplied with pressurized fluid through a first aperture formed in the first housing and wherein a second fluid chamber defined by the second housing and the second piston is supplied with pressurized fluid through a second aperture formed in the second housing.

32. The valve spring system of claim 31, wherein fluid disposed in the first fluid chamber is forced out of the first fluid chamber through the first aperture when the engine valve is open.

33. The valve spring system of claim 31, wherein fluid disposed in the second fluid chamber is forced out of the second fluid chamber through the second aperture when the engine valve is open and the control valve is closed.

34. A method of varying a spring force exerted on an engine valve by the valve spring system of claim 29, comprising:

closing the control valve to increase the aggregate surface area over which fluid pressure is coupled to the engine valve and thereby increase the spring force; and

opening the control valve to decrease the aggregate surface area over which fluid pressure is coupled to the engine valve and thereby decrease the spring force.

35. The method of claim 34, wherein the spring force is increased when a load of the engine exceeds a first predetermined threshold and is decreased when the load of the engine is below a second predetermined threshold.

36. The method of claim 34, wherein the spring force is increased when a speed of the engine exceeds a first predetermined threshold and is decreased when the speed of the engine is below a second predetermined threshold.

37. A method of actuating a valve of an engine, comprising:

increasing a spring force of a valve spring associated with the valve based on an increase in an engine parameter; and

decreasing the spring force of the valve spring based on a decrease in the engine parameter.

38. The method of claim 37, wherein the engine parameter is selected from the group consisting of engine speed, engine load, engine temperature, throttle position, and engine age.

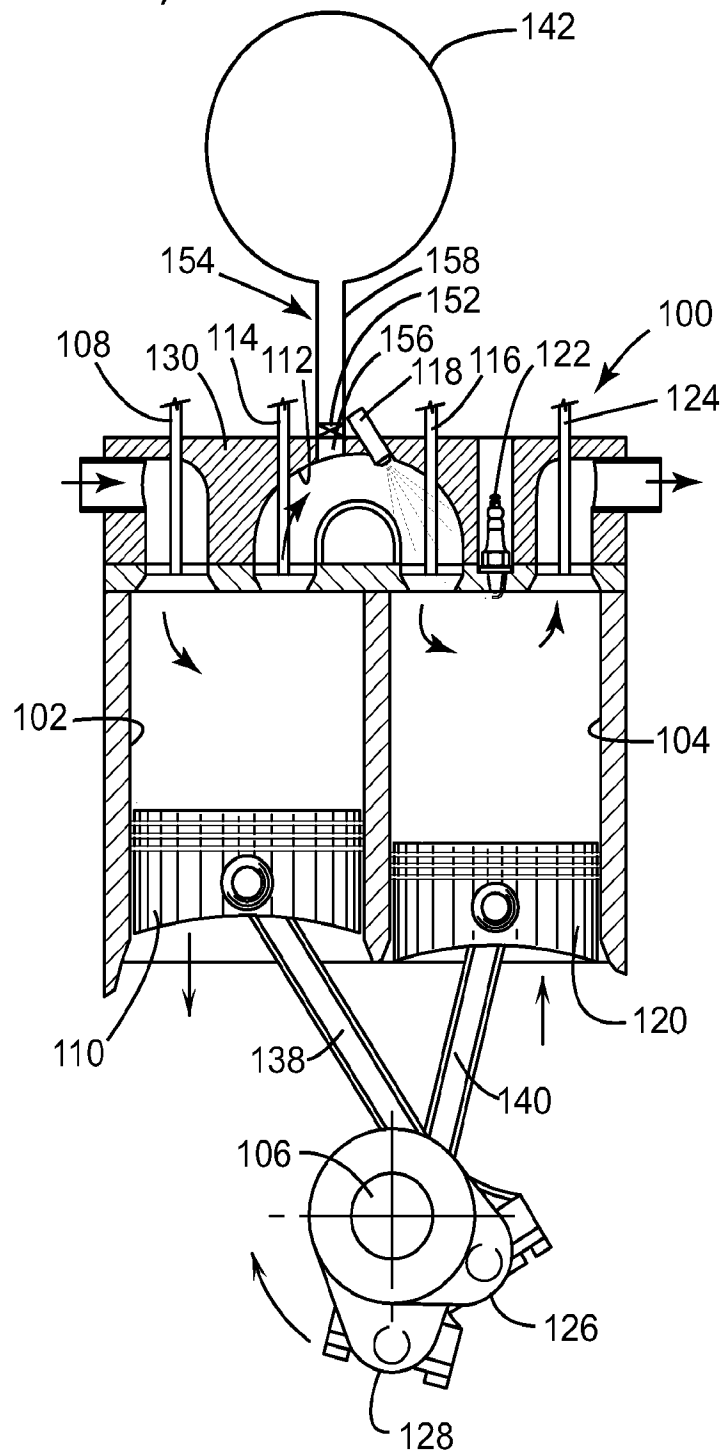
39. The method of claim 38, wherein the spring force is increased by increasing a pressure supplied to a fluid chamber of the valve spring.

40. The method of claim 38, wherein the spring force is increased by decreasing a volume of a fluid chamber of the valve spring.

41. The method of claim 38, wherein the spring force is increased by increasing an aggregate surface area over which fluid pressure is coupled to the engine valve.

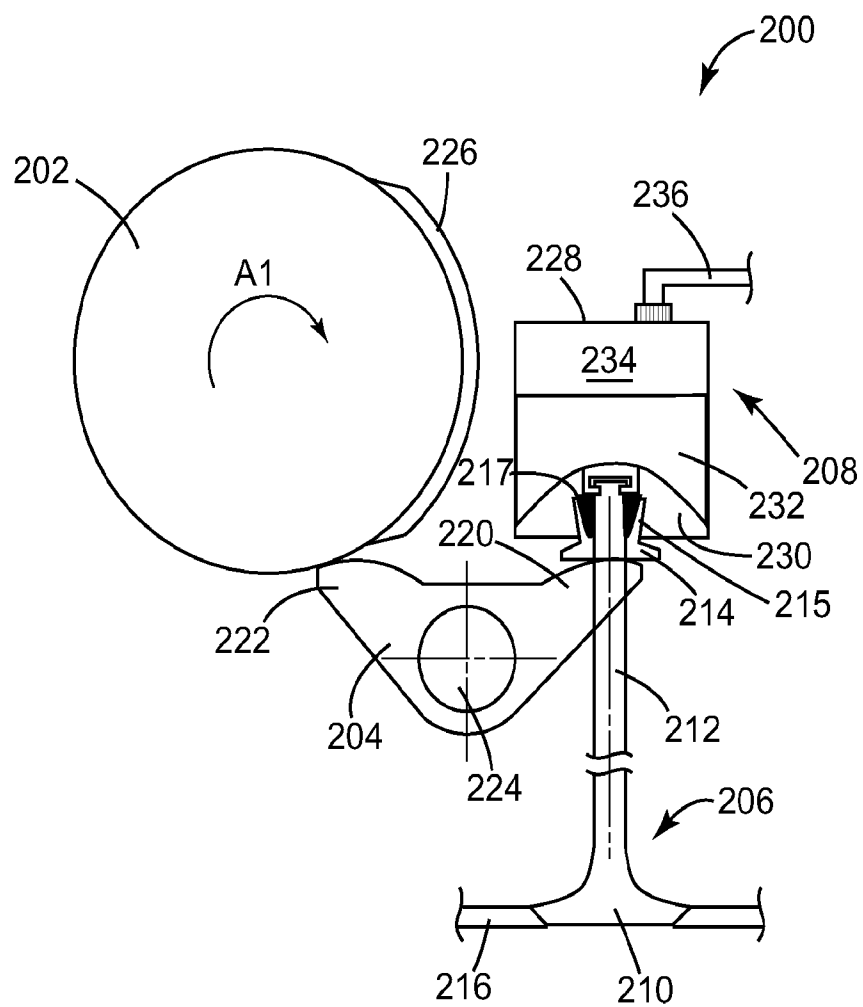
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FIG. 1
(Prior Art)



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FIG. 2A



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FIG. 2B

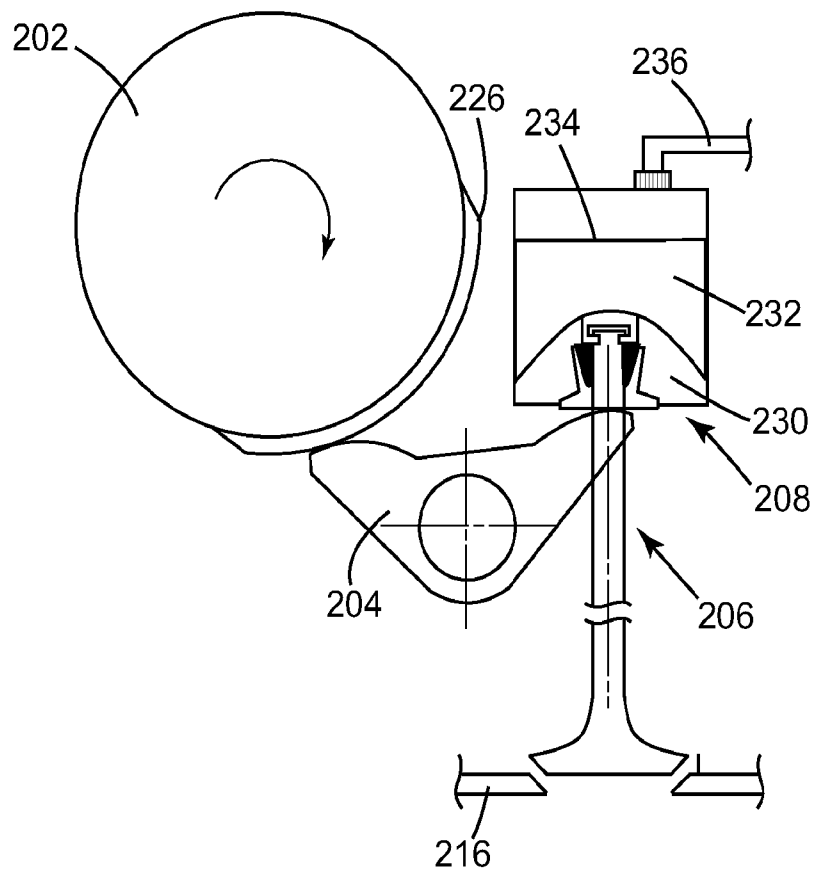


FIG. 3A

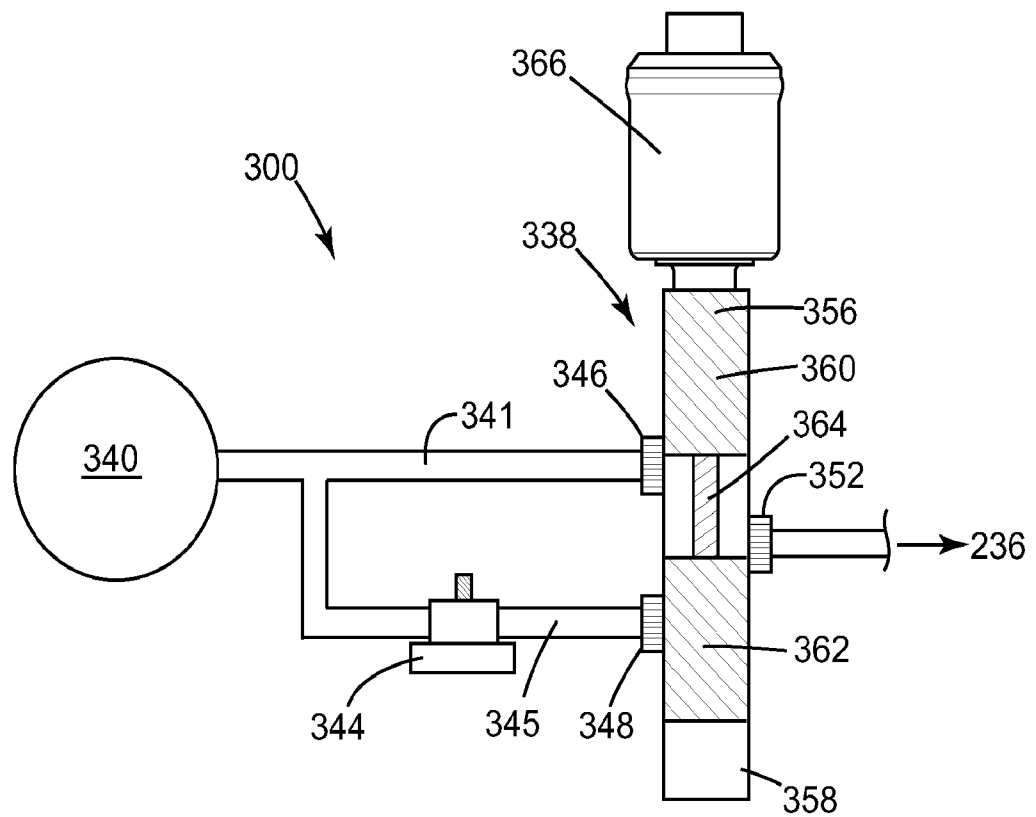


FIG. 3B

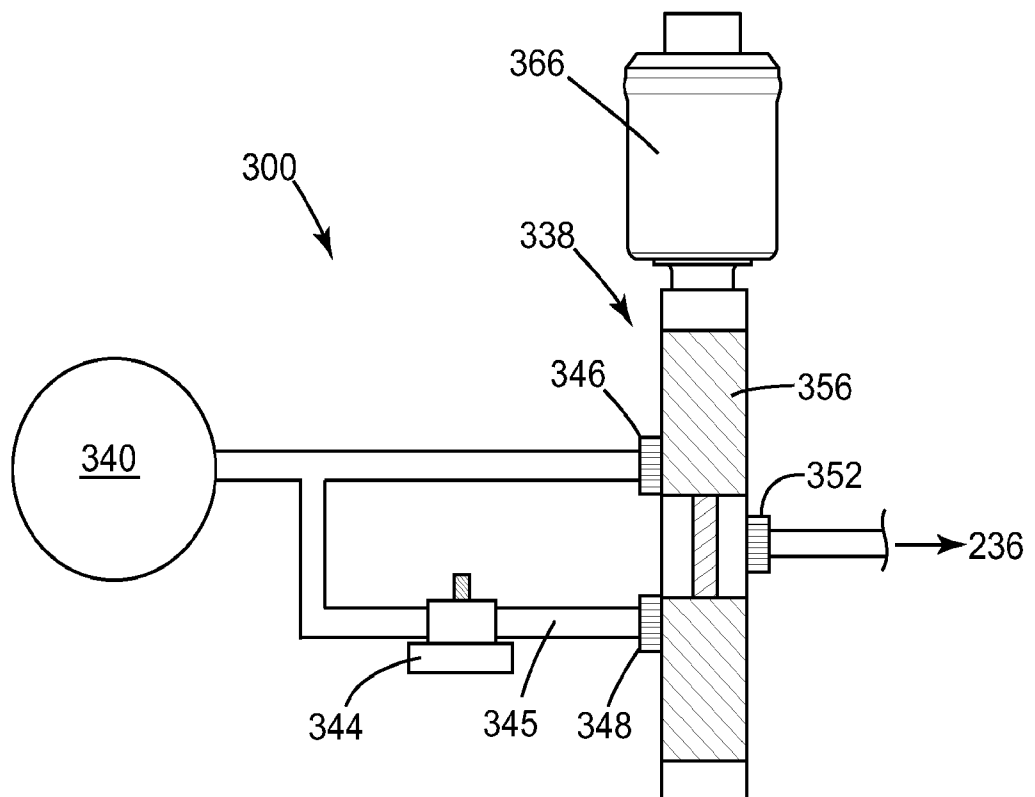
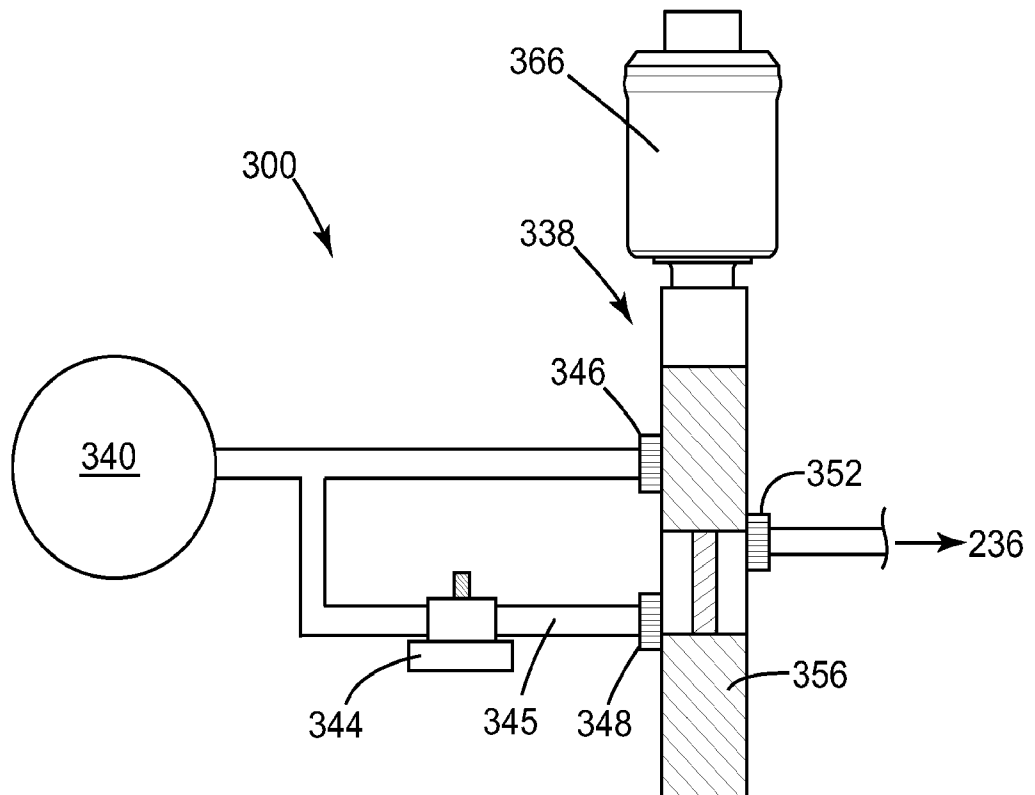


FIG. 3C



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FIG. 4

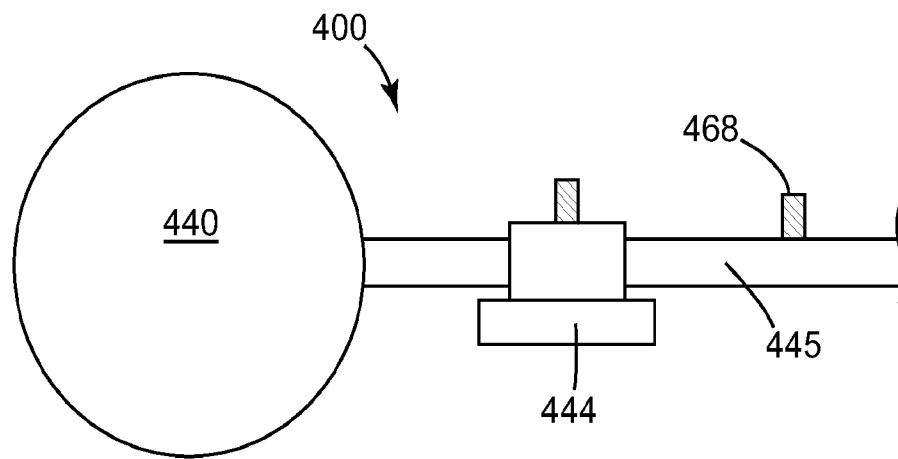
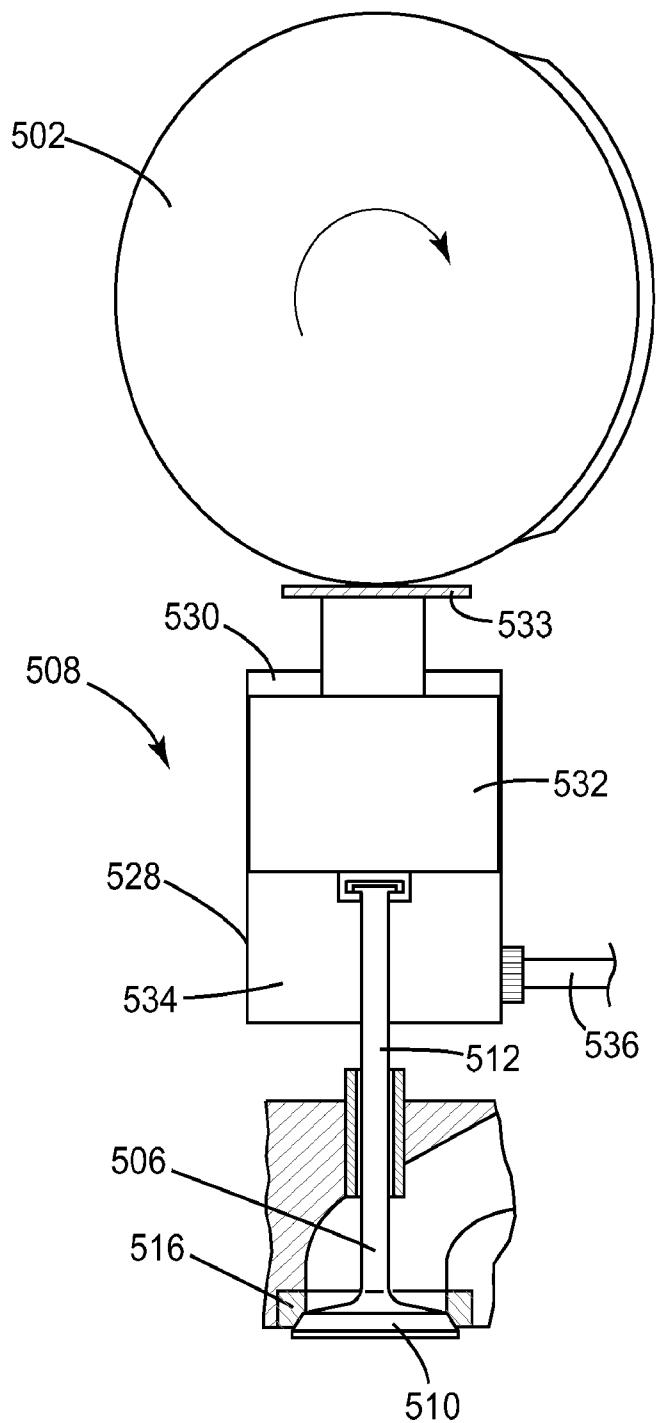


FIG. 5A



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FIG. 5B

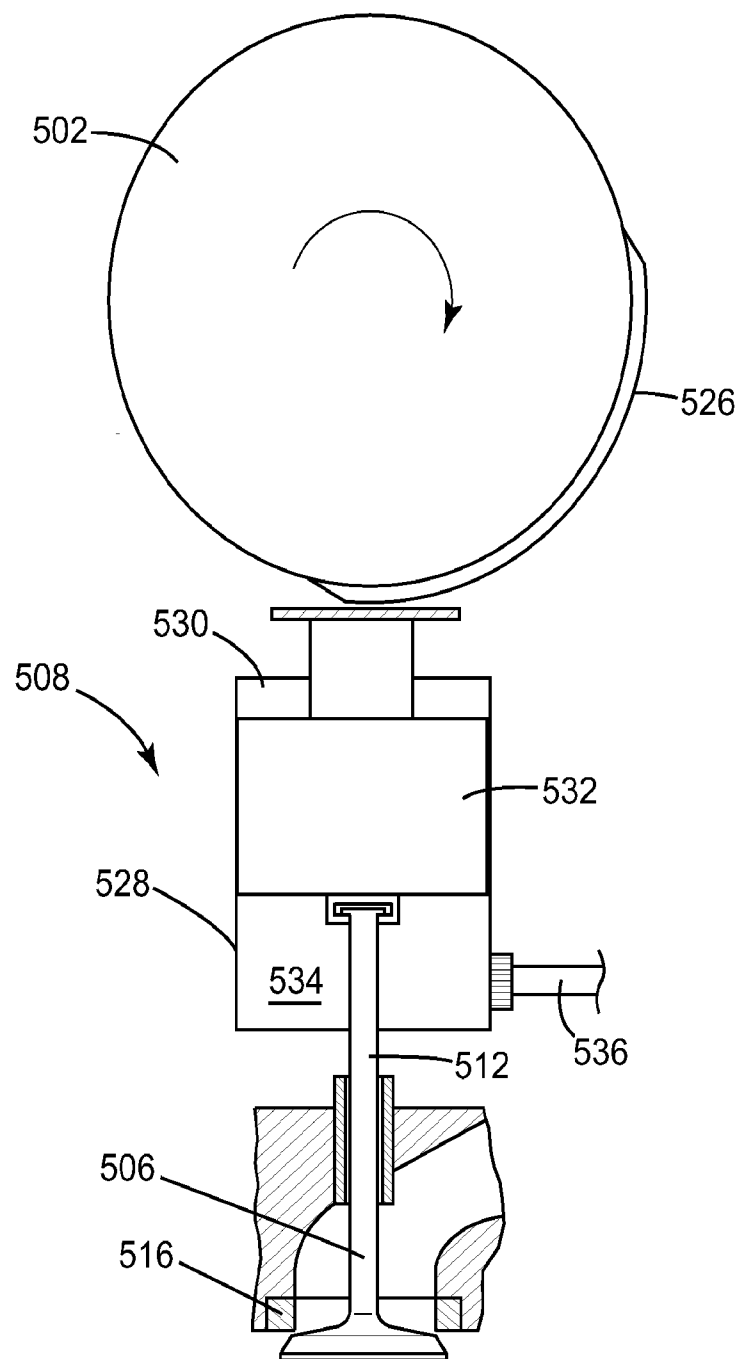


FIG. 6A

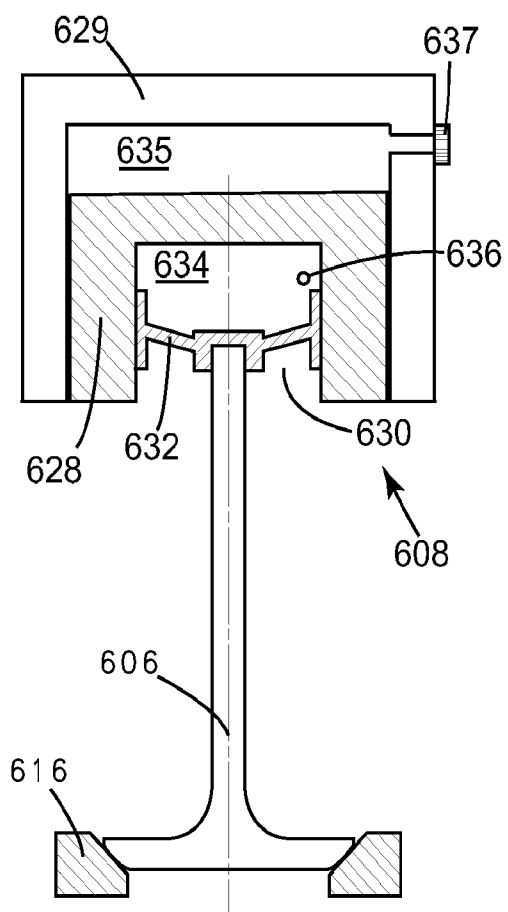
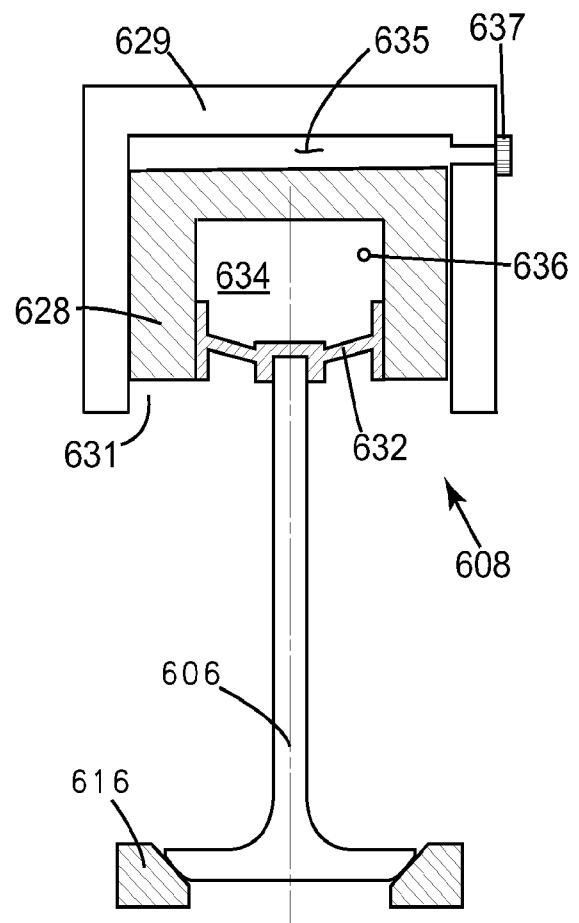
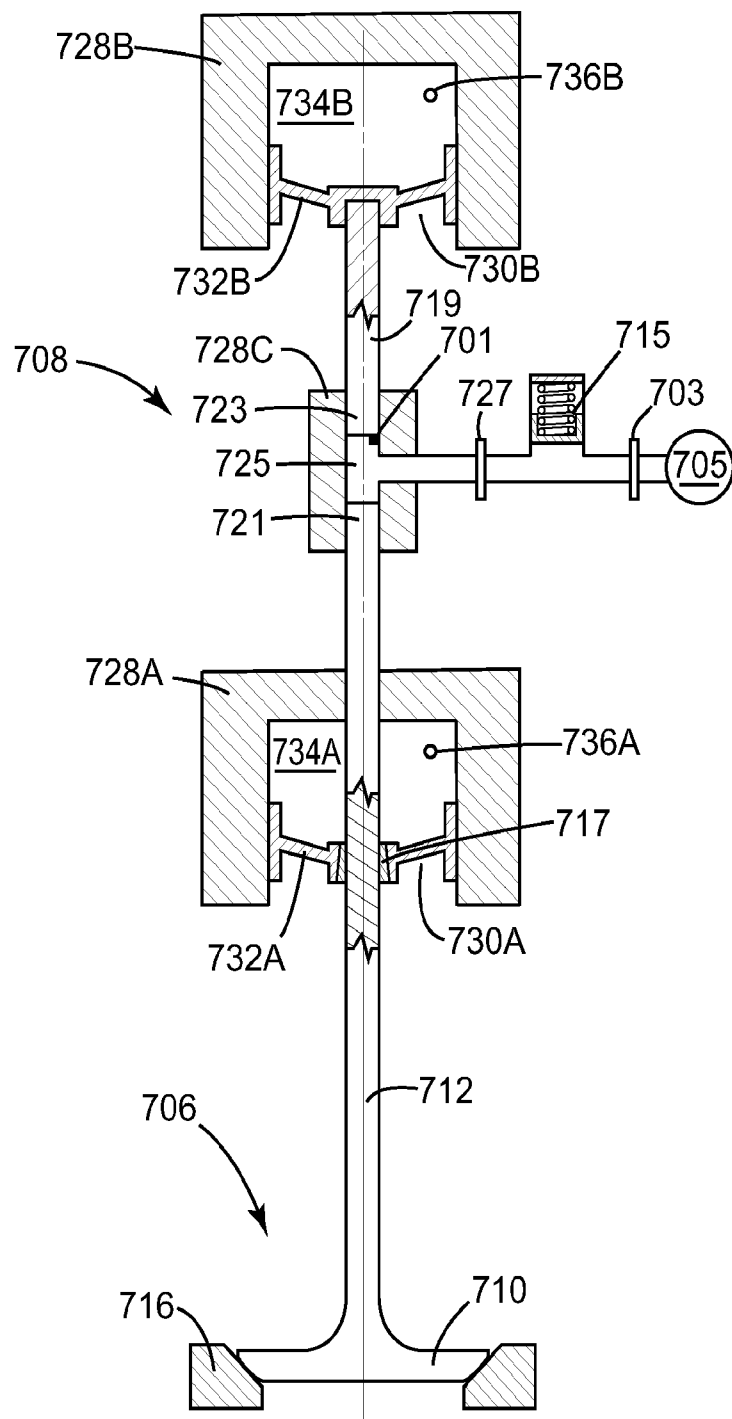


FIG. 6B



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FIG. 7



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FIG. 8A

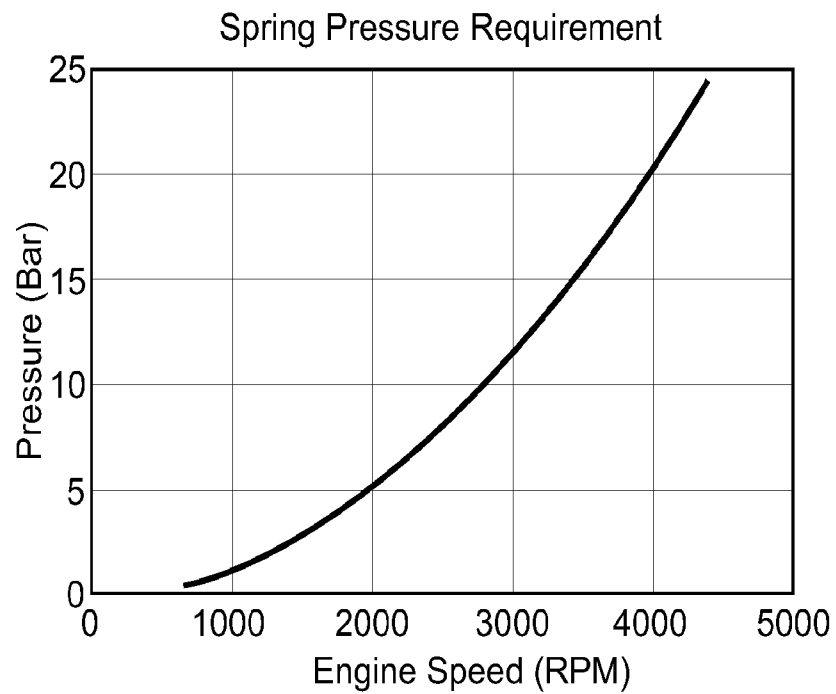


FIG.8B

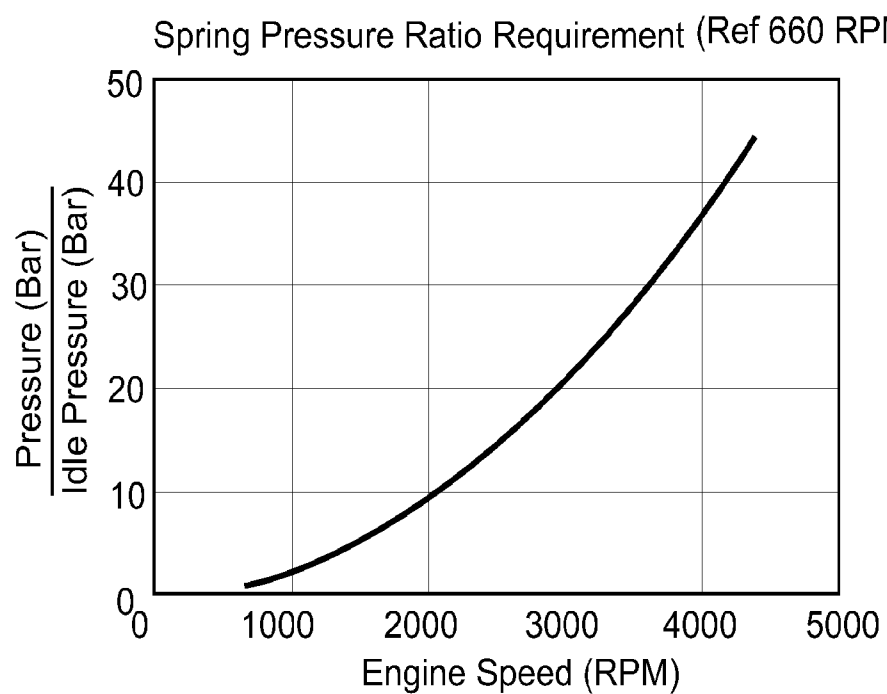


FIG. 9A

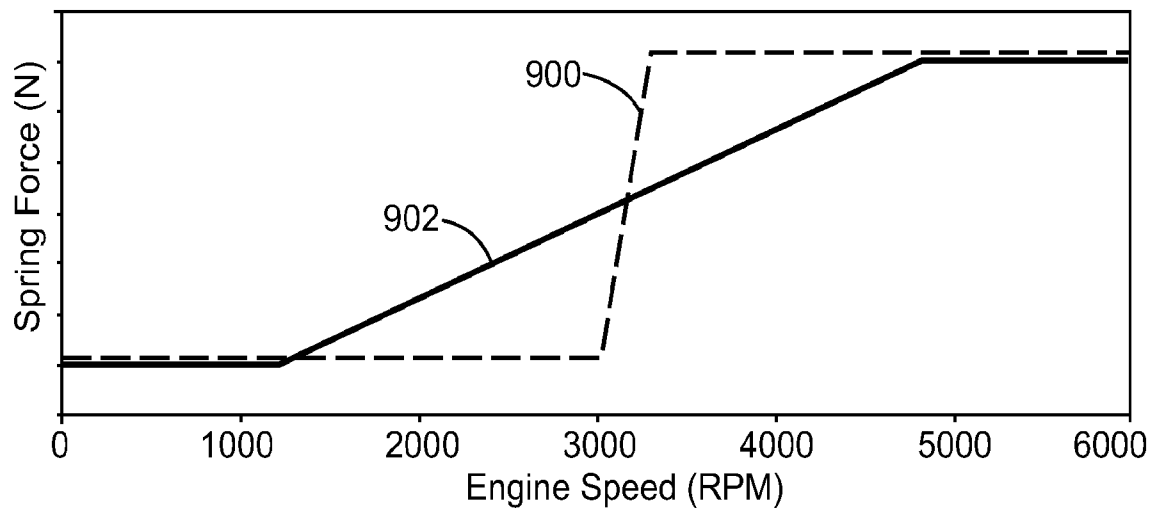
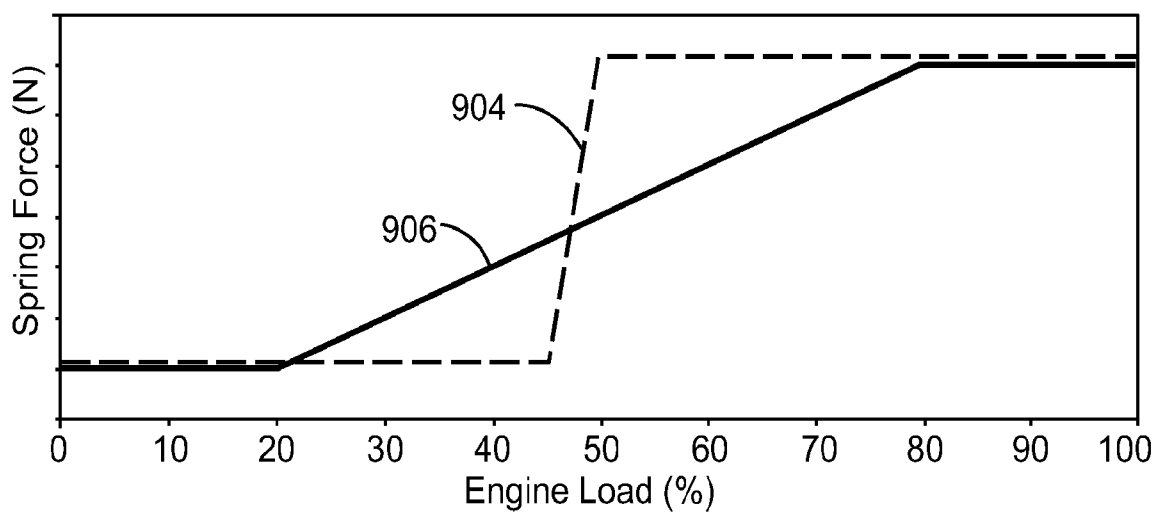


FIG. 9B



INTERNATIONAL SEARCH REPORT

International application No.

PCT/US 12/22834

A. CLASSIFICATION OF SUBJECT MATTER

IPC(8) - F01L 3/10

USPC - 123/90.65

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

USPC:123/90.65

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

USPC:123/90.1, 90.65-90.67; F01L 3/10

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

PubWEST:PGPB,USPT,EPAB,JPAB,DWPI,TDBD; Google Scholar

Search Terms: valve, spring, piston, cap, cover, housing, chassis, enclosure, fluid, pressure, plenum, air, engine, load, speed, threshold

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5,213,072 A (Dohring) 25 May 1993 (25.05.1993) entire document; especially fig 3, col 5, ln 10-18.	1-9, 37-41
X ----- Y	US 5,664,531 A (Kim) 09 September 1997 (09.09.1997) entire document; especially fig 2A&B; col 3, ln 5-46.	20-28 ----- 10-19
X	US 6,886,511 B1 (Tong et al.) 03 May 2005 (03.05.2005) entire document; especially fig 1; col 2, ln 45-60; col 3, ln 20-44.	29-36
Y	US 7,077,083 B2 (Dingle et al.) 18 July 2006 (18.07.2006) entire document; especially fig 1, 2A&B; col 6, ln 25-39; col 5, ln 53.	10-19
Y	US 2008/0105225 A1 (Scuderi et al.) 08 May 2008 (08.05.2008) entire document; especially fig 2; para [0060].	13-15
A	US 5,445,119 A (Regueiro) 29 August 1995 (29.08.1995) entire document.	1-41
A	US 2006/0011154 A1 (Scuderi et al.) 19 January 2006 (19.01.2006) entire document.	1-41



Further documents are listed in the continuation of Box C.



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"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search

03 May 2012 (03.05.2012)

Date of mailing of the international search report

11 MAY 2012

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