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(12) **United States Patent**
Meldolesi et al.

(10) **Patent No.: US 8,776,740 B2**
(45) **Date of Patent: Jul. 15, 2014**

(54) **LOST-MOTION VARIABLE VALVE ACTUATION SYSTEM WITH CAM PHASER**

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(73) Assignee: **Scuderi Group, LLC**, West Springfield, MA (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 80 days.

(21) Appl. No.: **13/359,521**

(22) Filed: **Jan. 27, 2012**

(65) **Prior Publication Data**

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Related U.S. Application Data

(60) Provisional application No. 61/436,735, filed on Jan. 27, 2011.

(51) **Int. Cl.**
F01L 1/34 (2006.01)

(52) **U.S. Cl.**
USPC **123/90.16**

(58) **Field of Classification Search**
USPC 123/90.15–90.17
See application file for complete search history.

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(57) **ABSTRACT**

Devices and related methods are disclosed that generally involve variable actuation of engine valves. In one embodiment, a valve train for a split-cycle internal combustion engine or an air hybrid split-cycle engine is provided that includes a cam phaser, a dwell cam, an adjustable mechanical element for performing a variable valve actuation function, and/or a valve seating control device. The devices and methods disclosed herein also have application in conventional internal combustion engines and can actuate inwardly-opening and/or outwardly-opening valves.

21 Claims, 23 Drawing Sheets

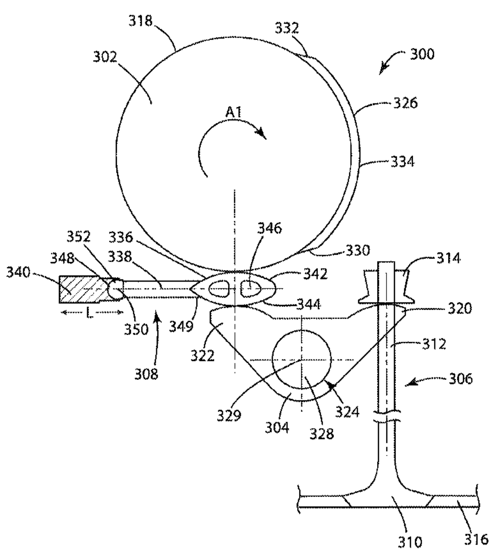

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FIG. 1
Prior Art

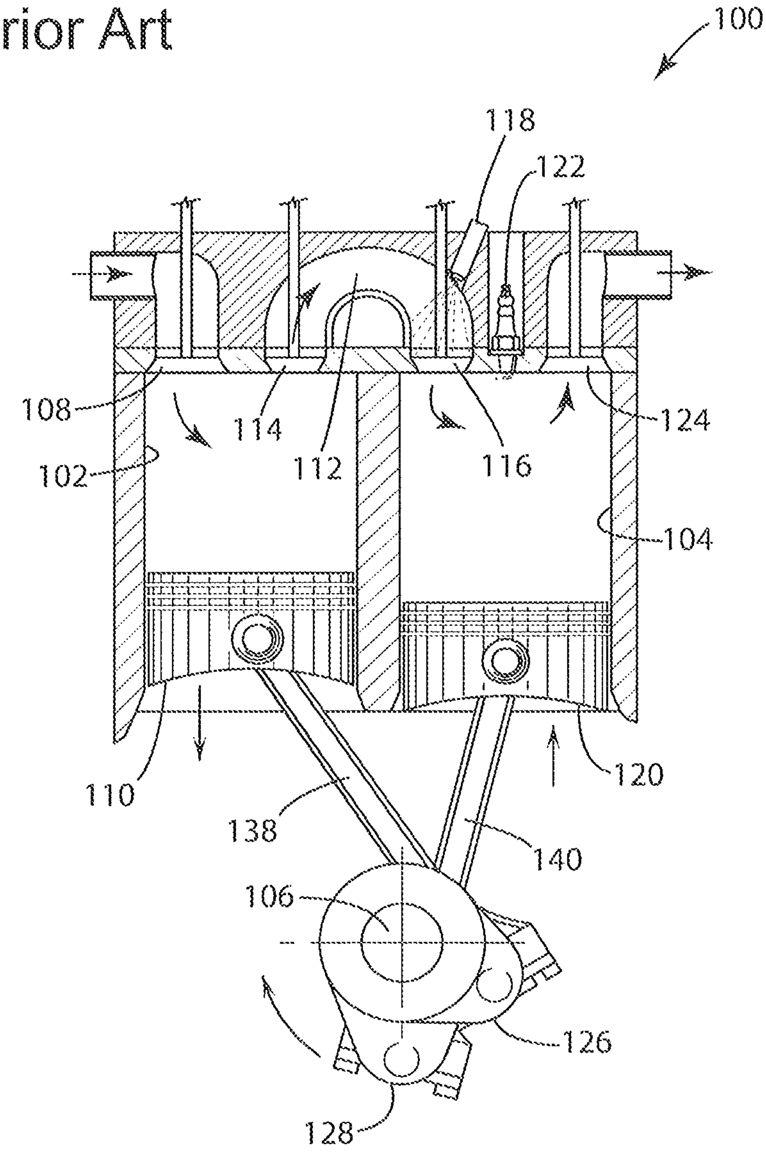

FIG. 2
Prior Art

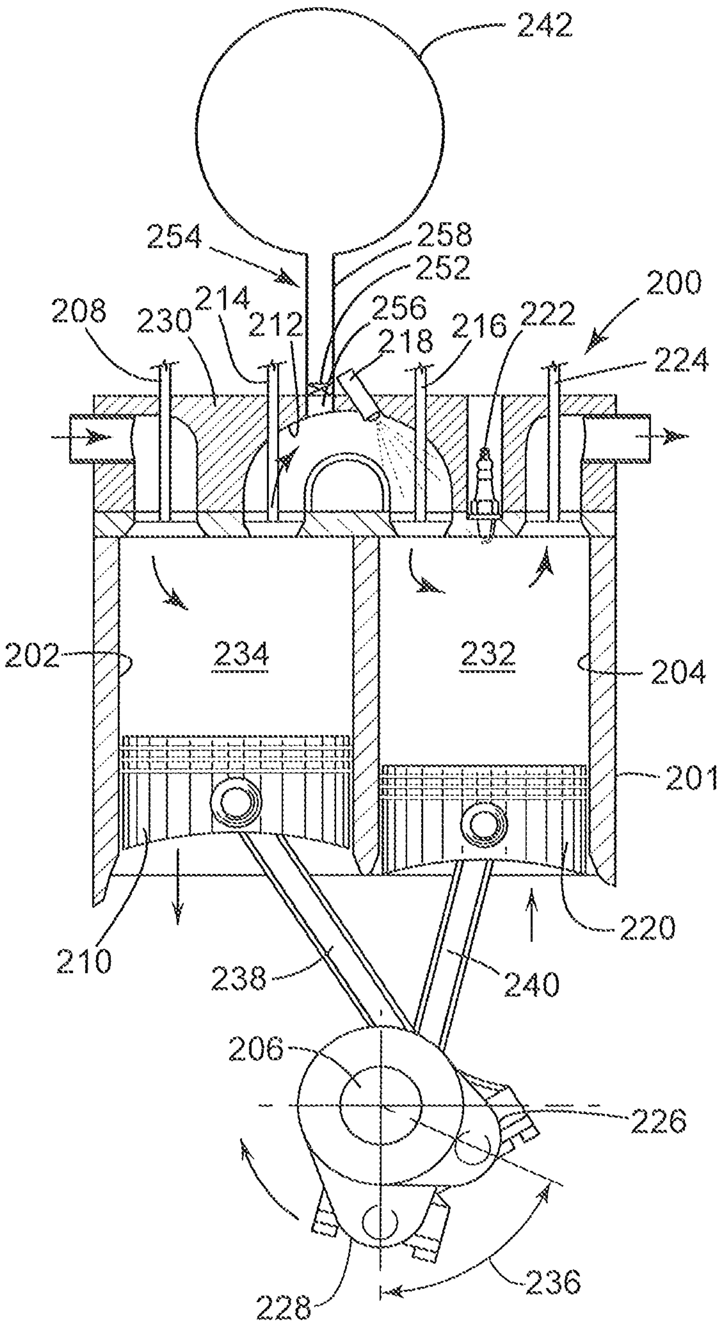

FIG. 3A

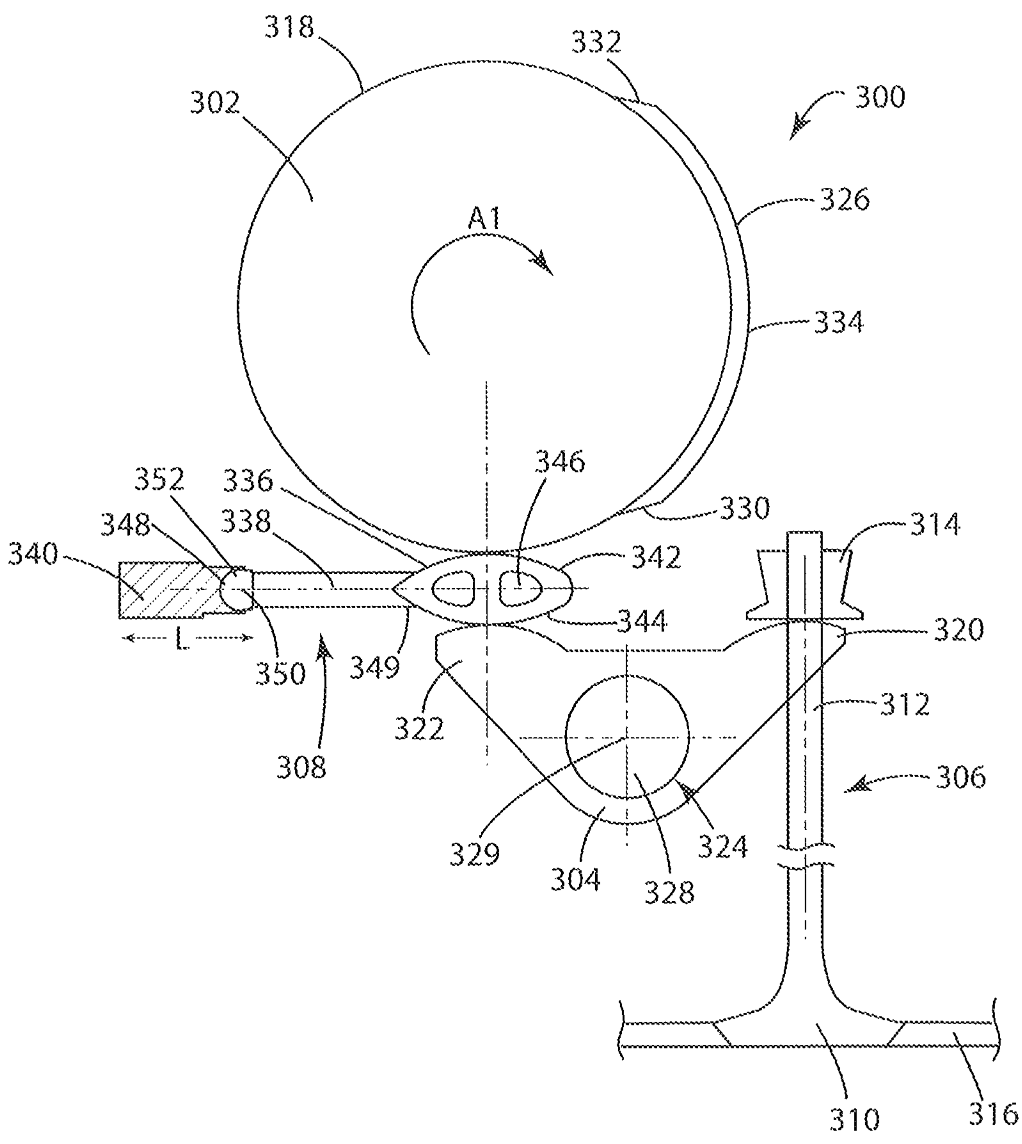

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FIG. 3B

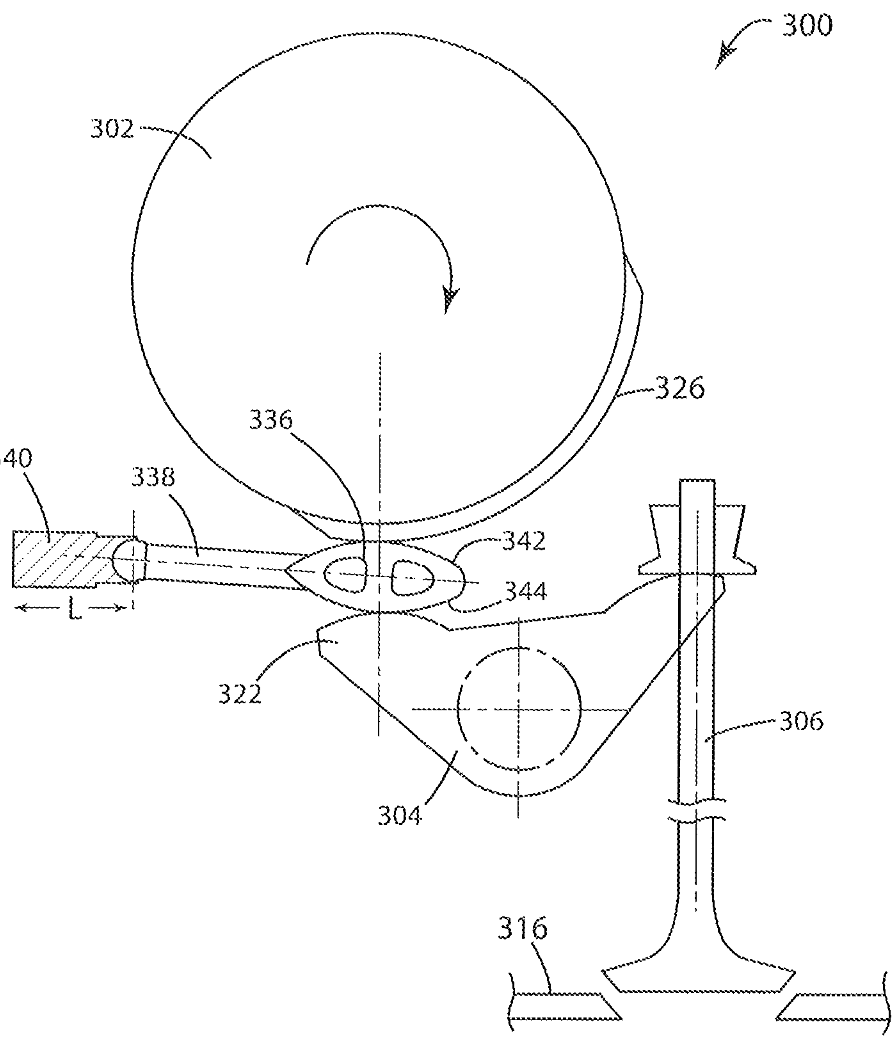

FIG. 3C

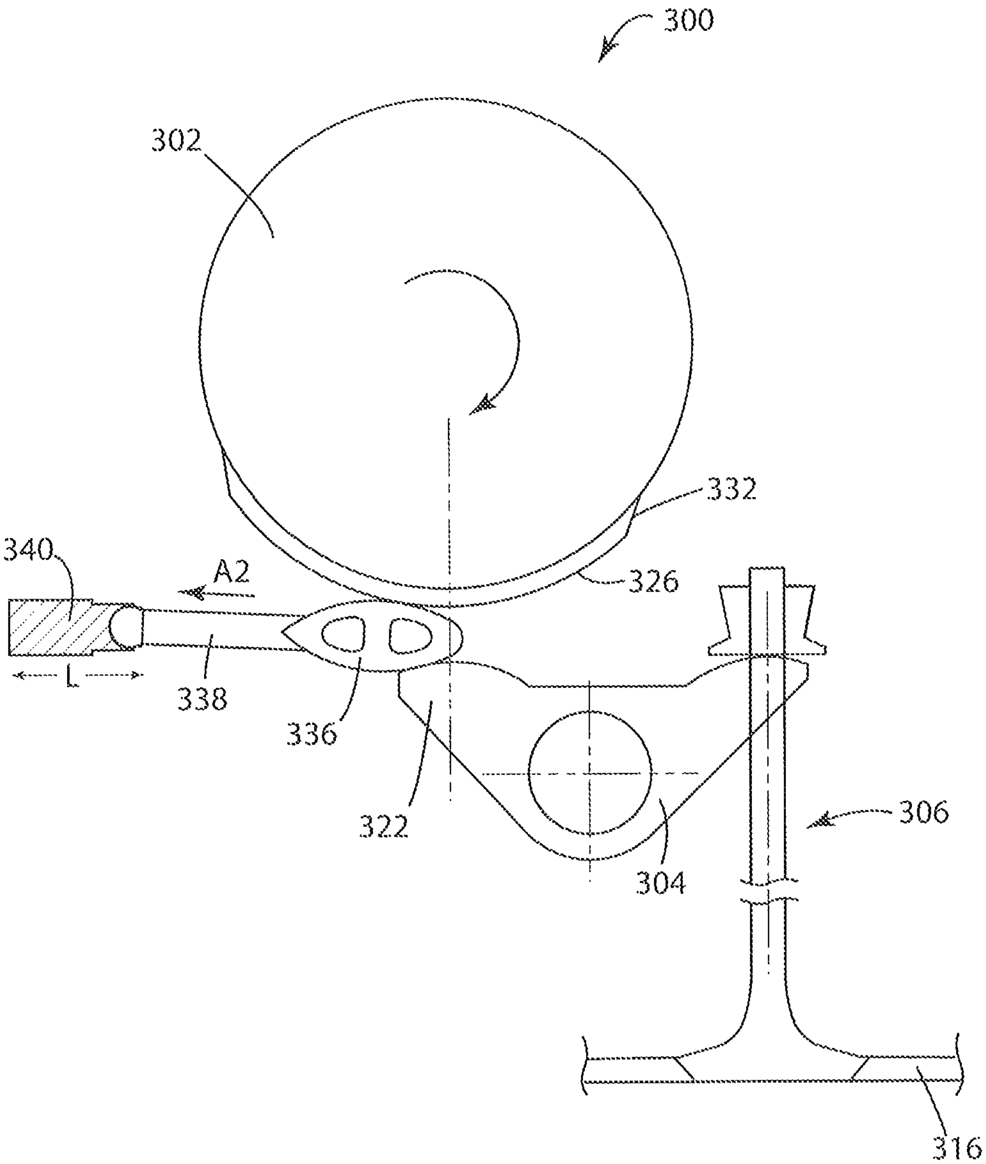

FIG. 4A

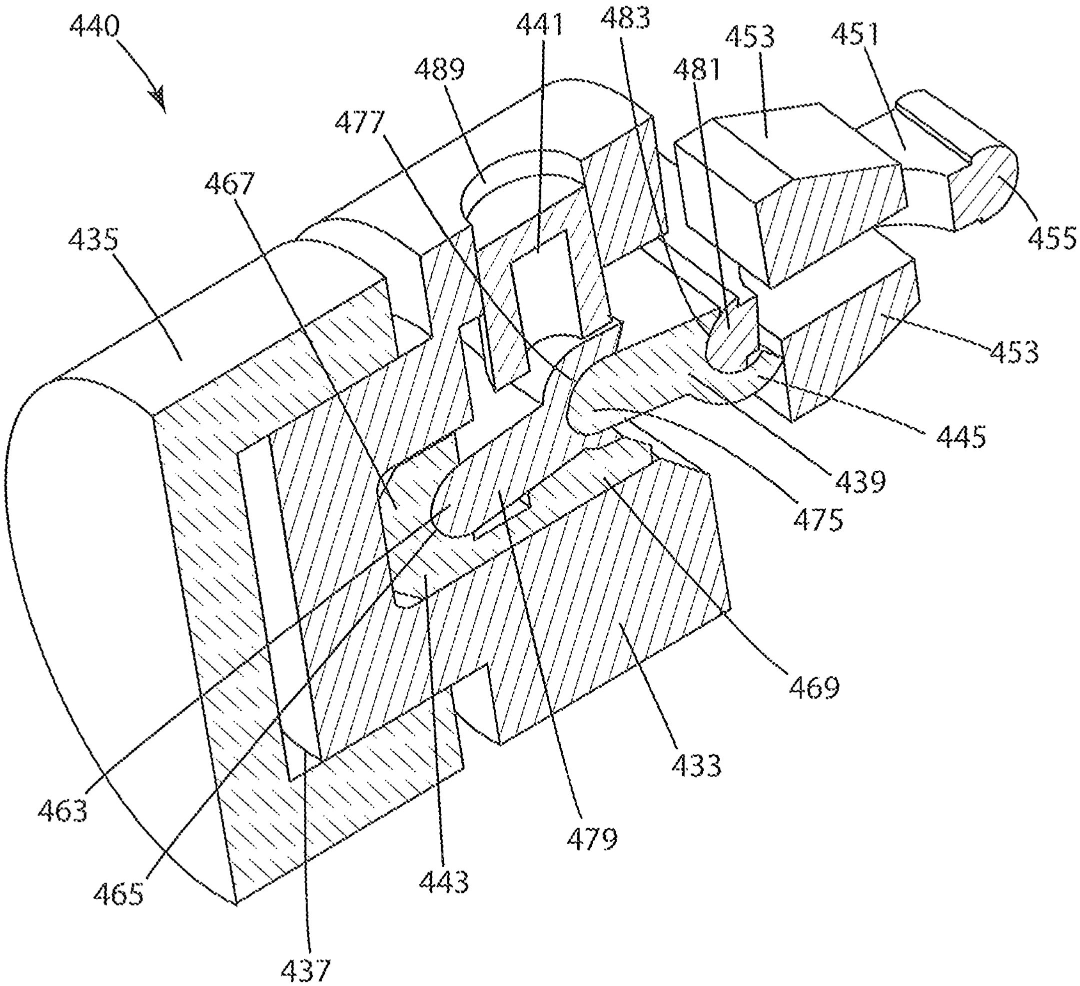

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FIG. 4B

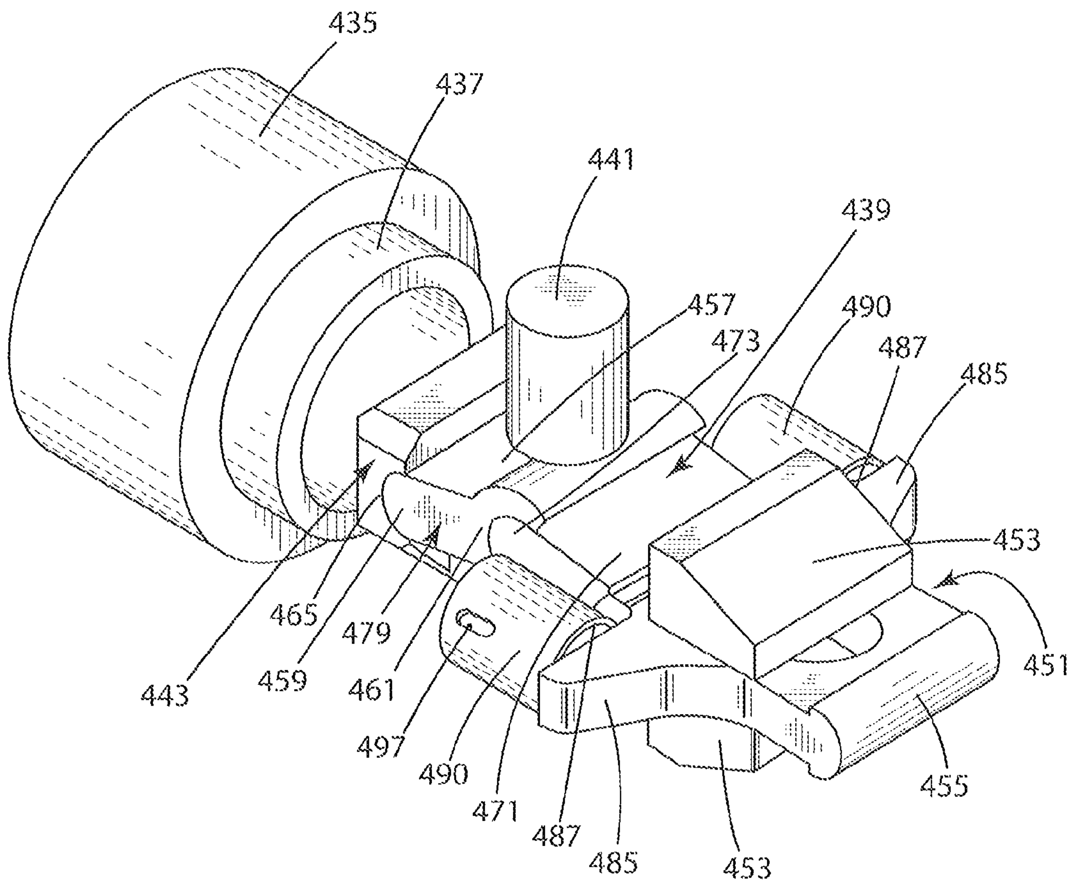
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FIG. 4C

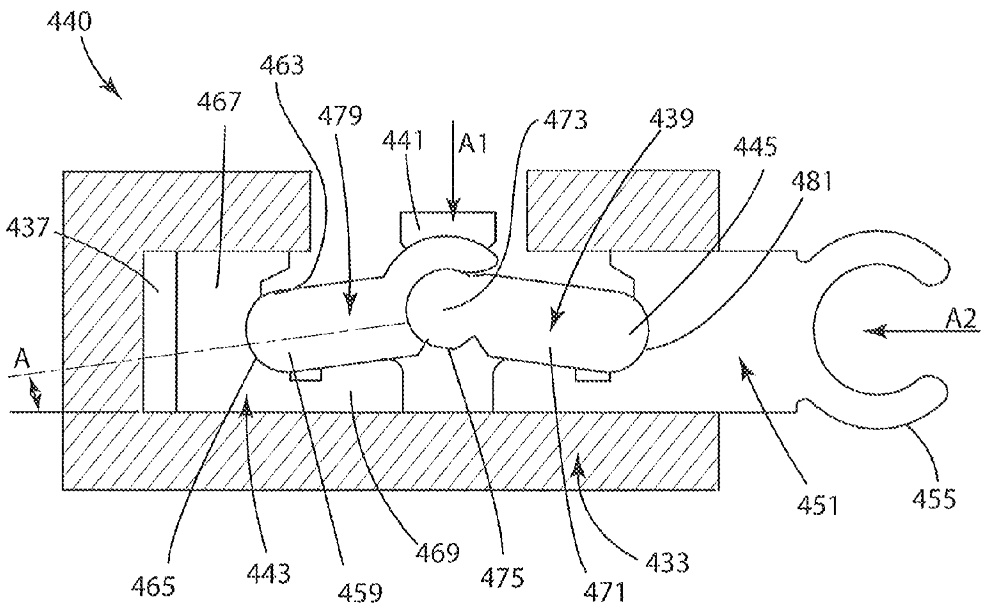
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FIG. 4D

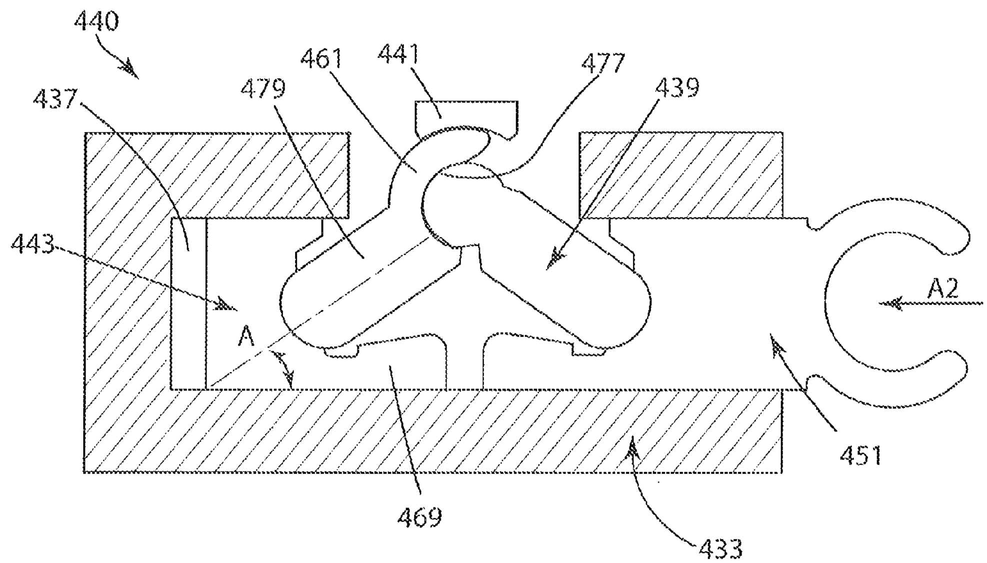
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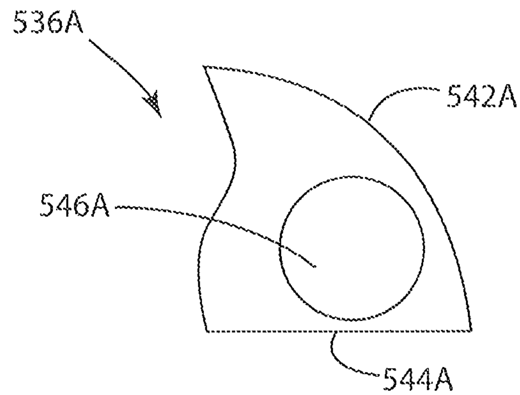
FIG. 5A



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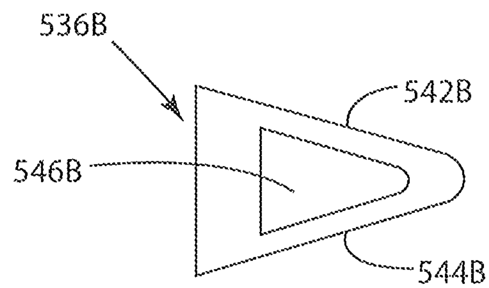


FIG. 5C

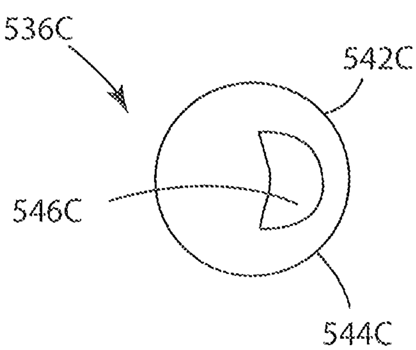


FIG. 6A

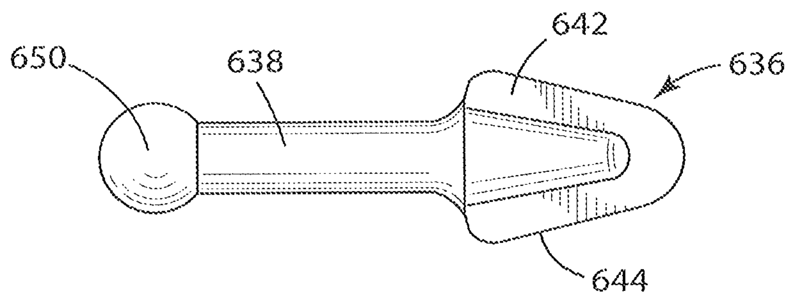


FIG. 6B

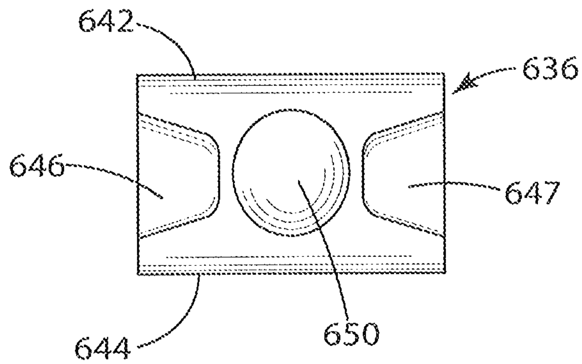


FIG. 6C

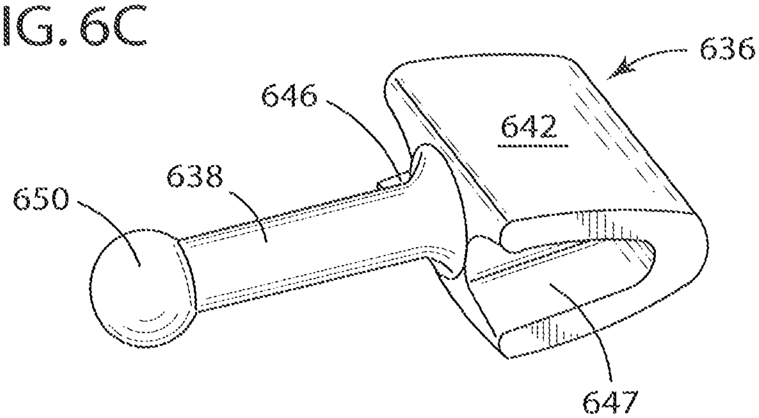


FIG. 7A

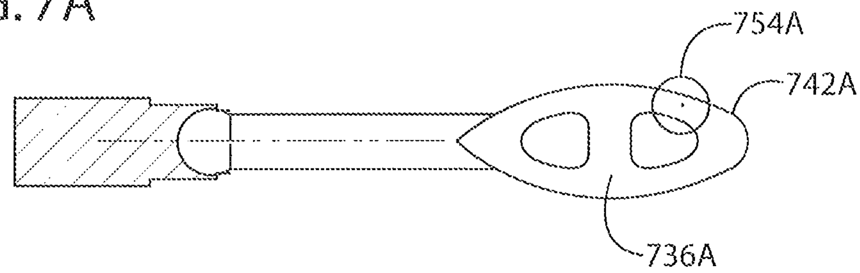


FIG. 7B

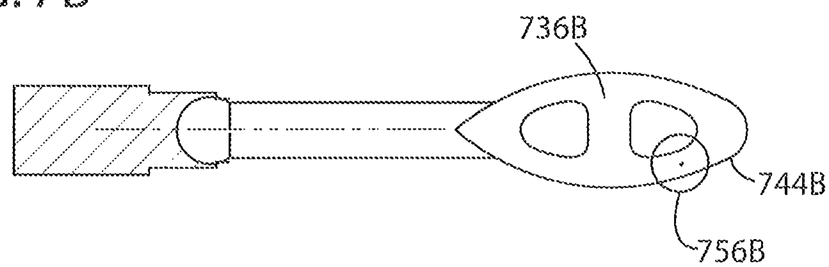
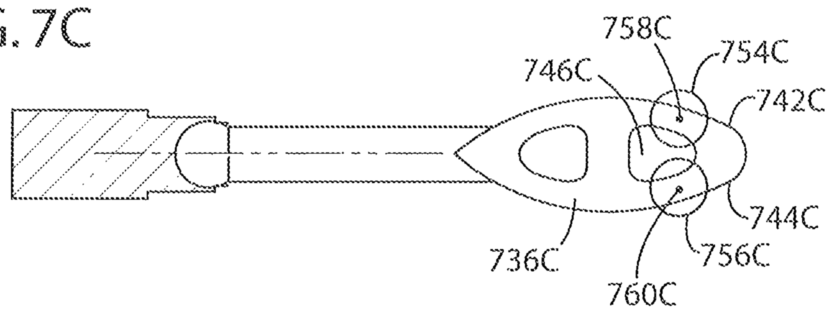


FIG. 7C



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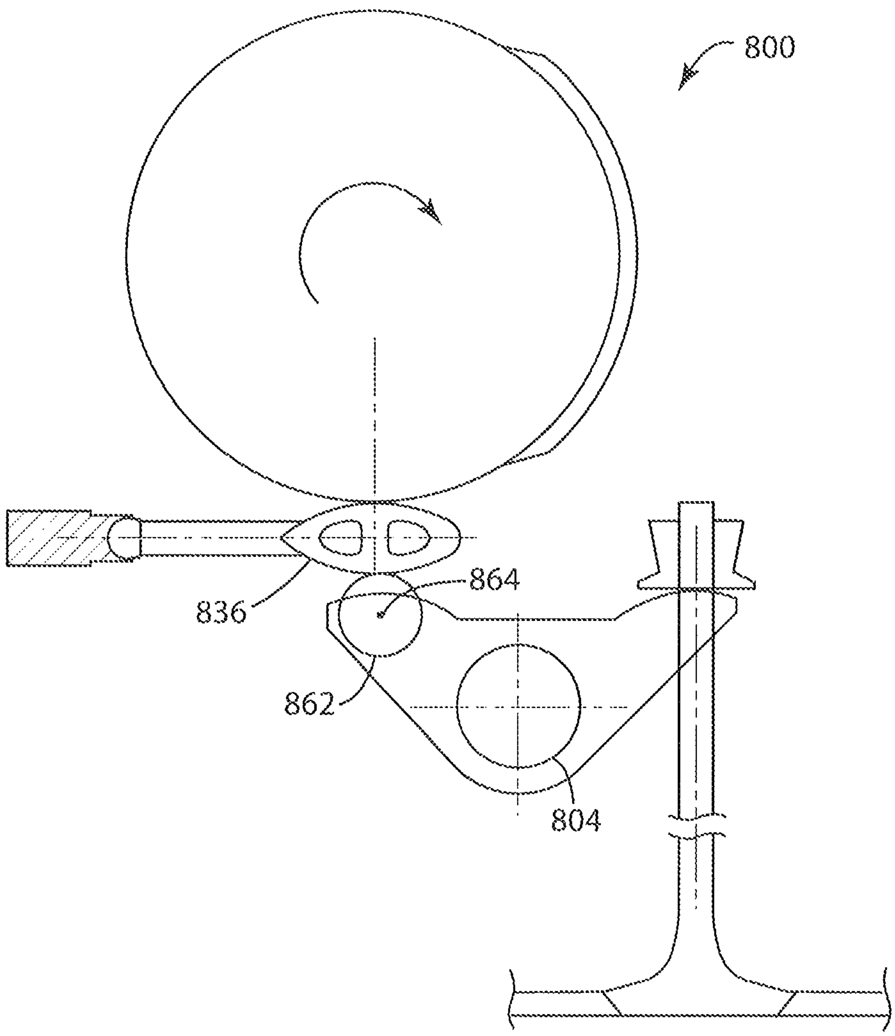

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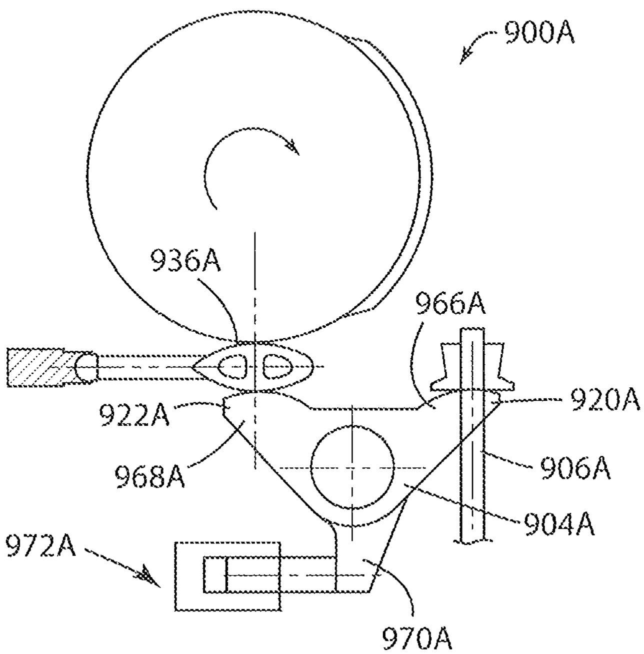
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FIG. 9B

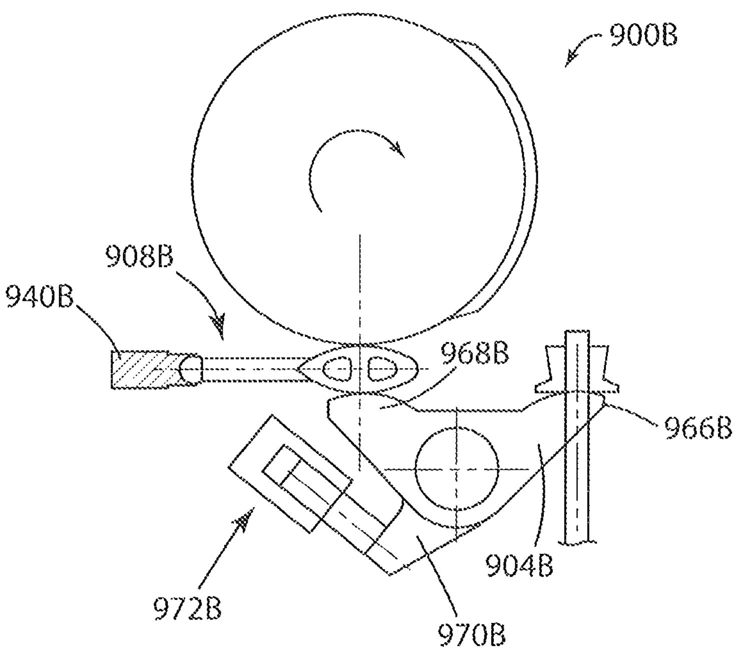
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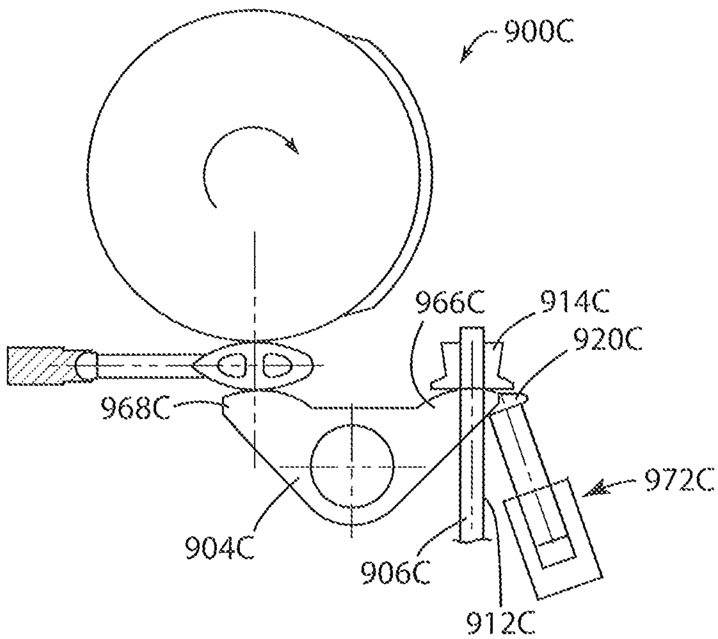
FIG. 9C



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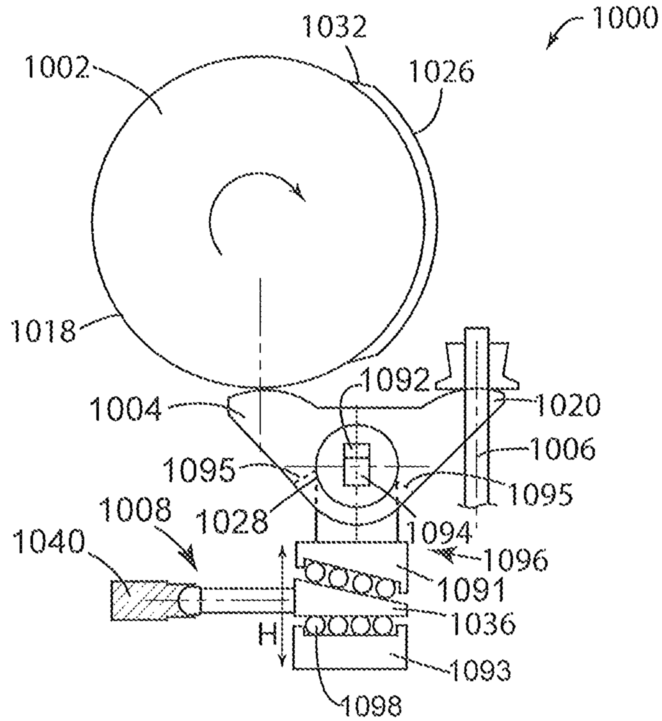
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FIG. 10B

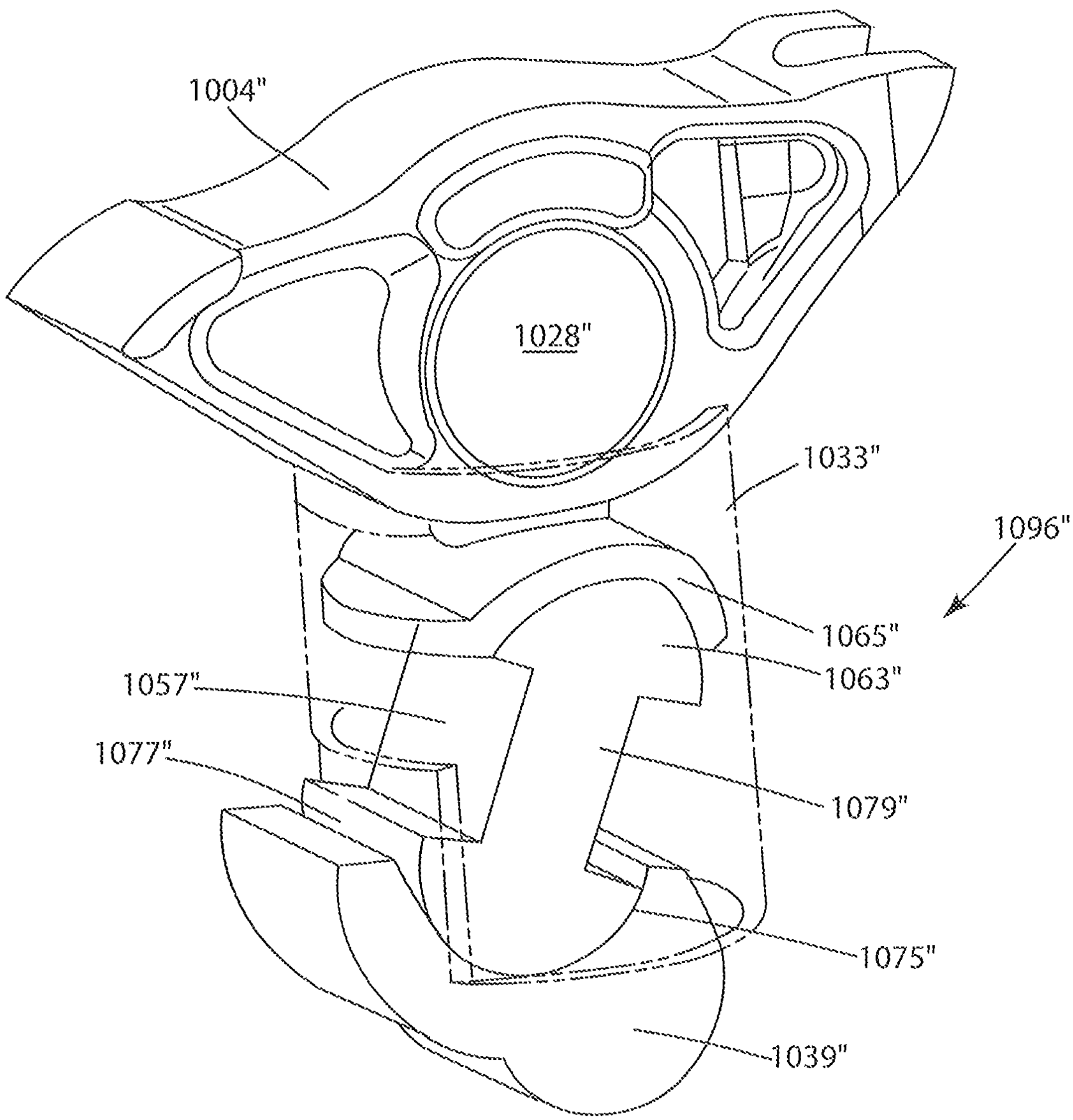
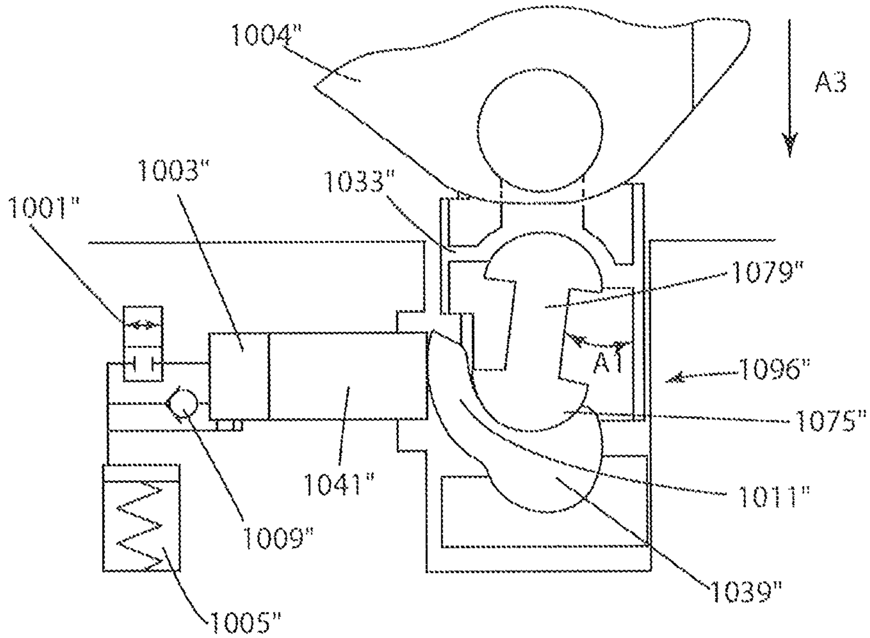


FIG. 10C



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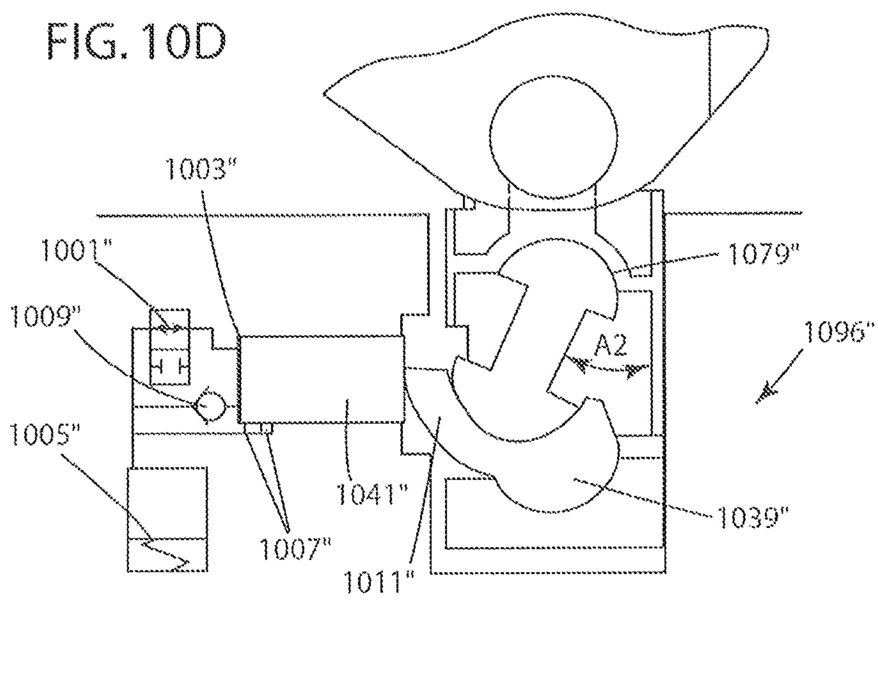


FIG. 10E

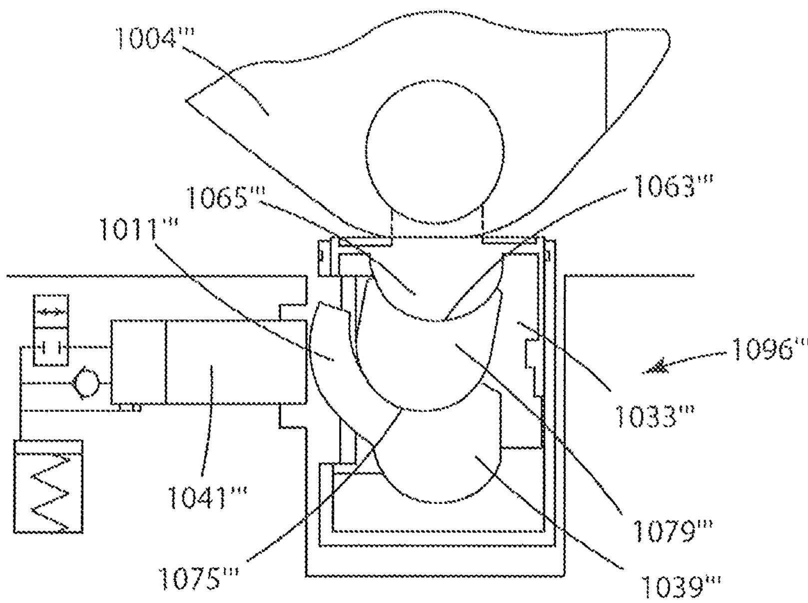


FIG. 10F

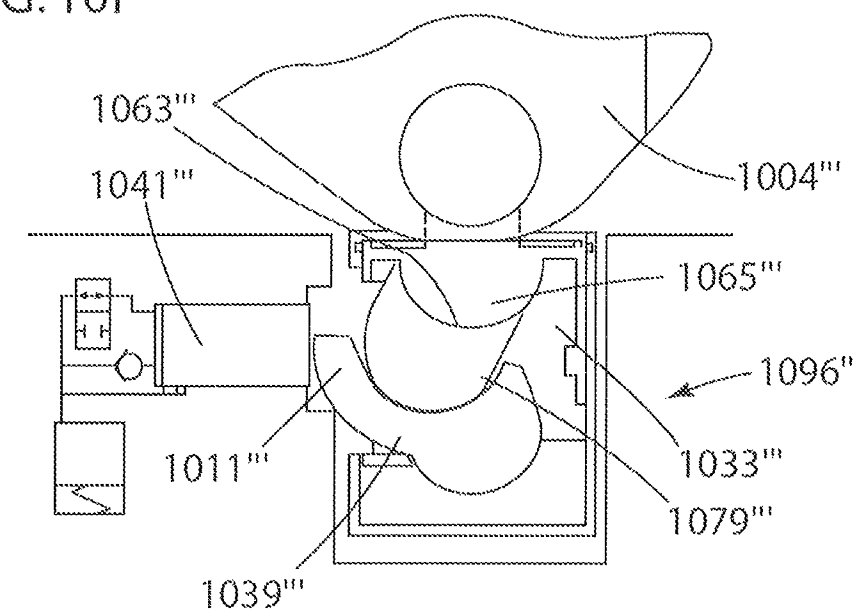


FIG. 11A

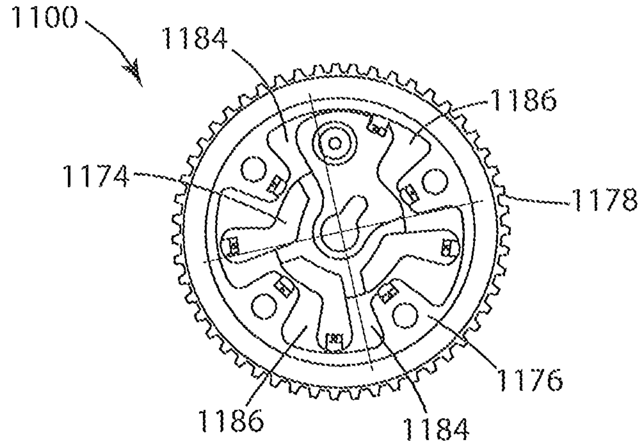
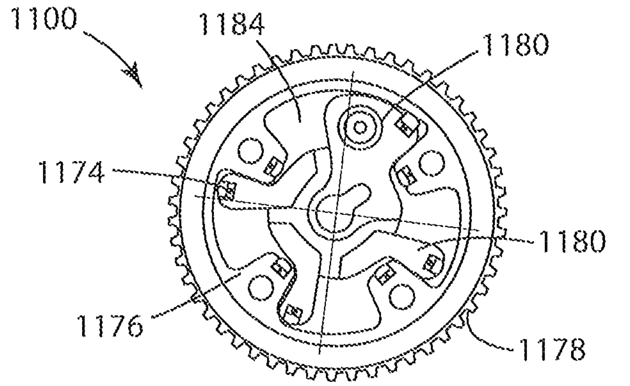


FIG. 11B



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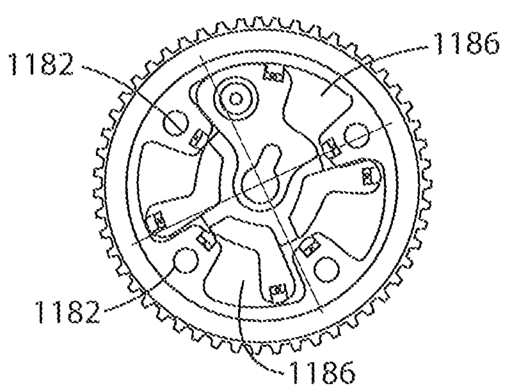
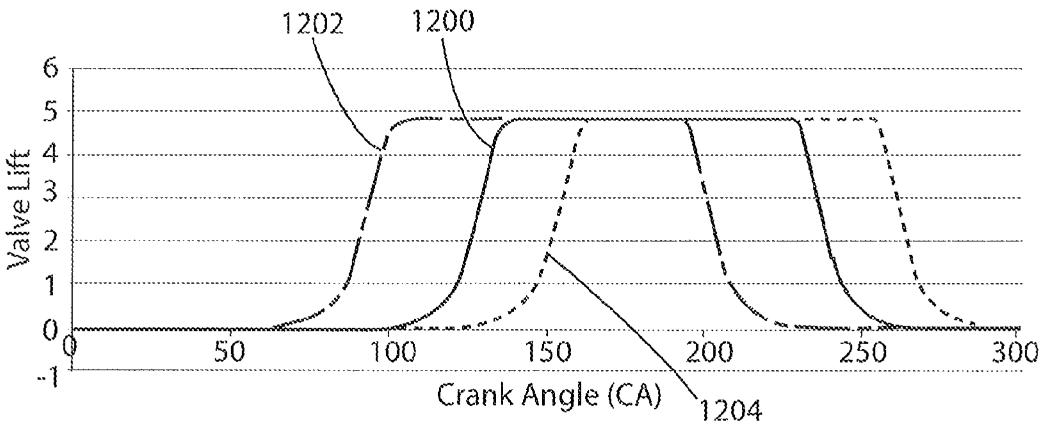


FIG. 12A



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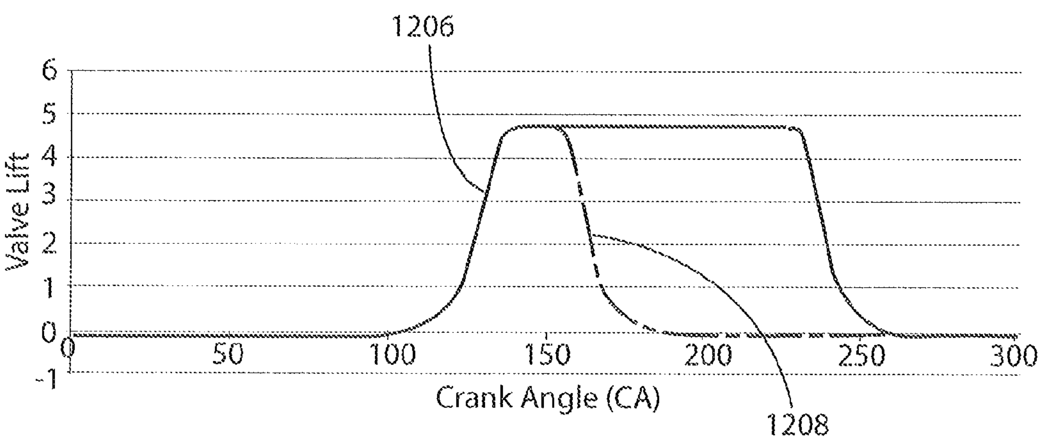

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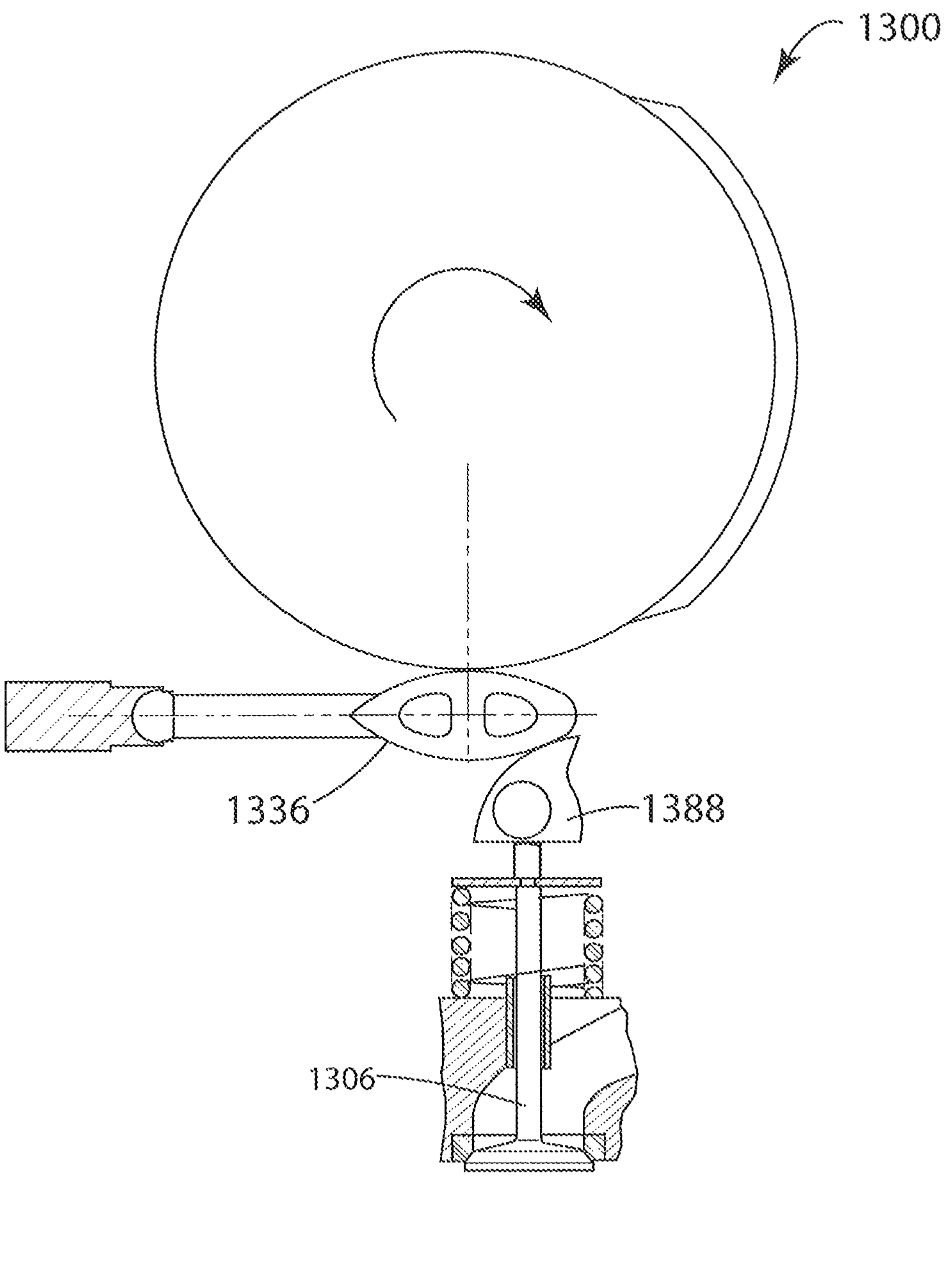

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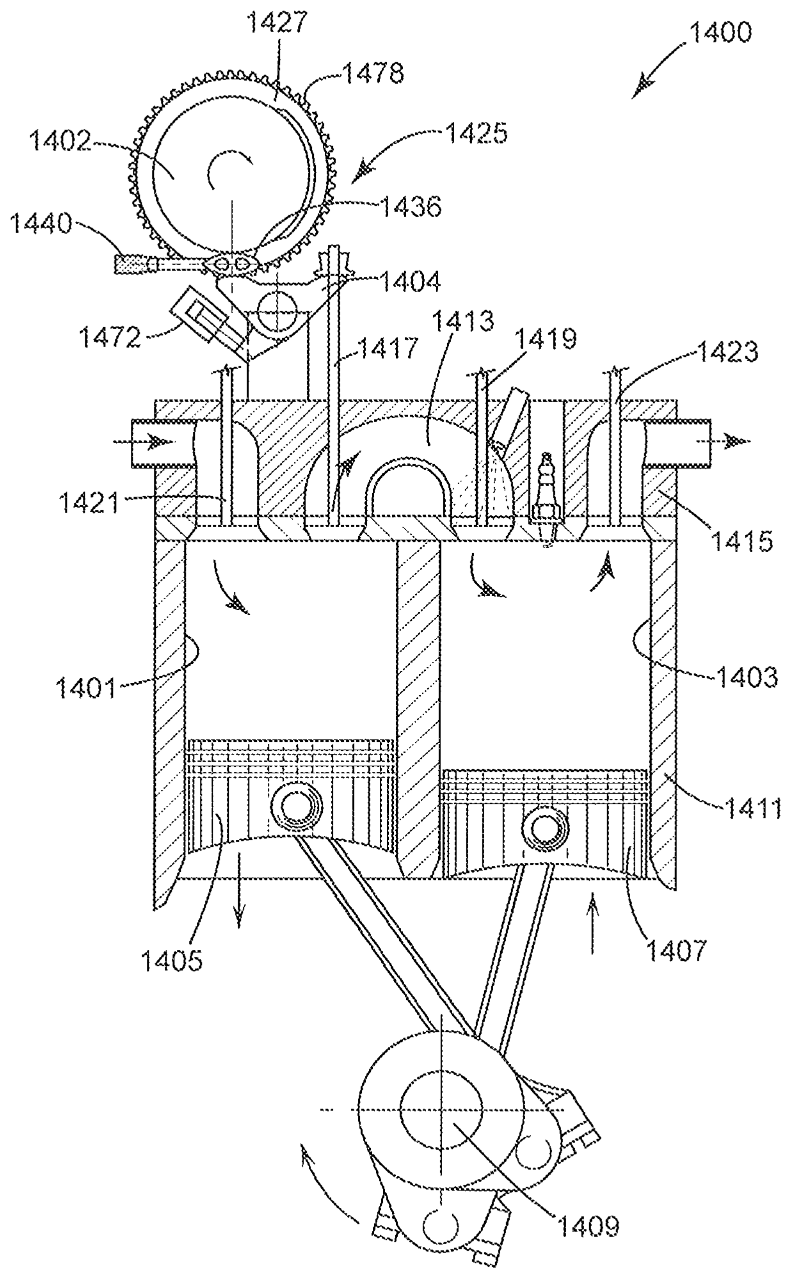
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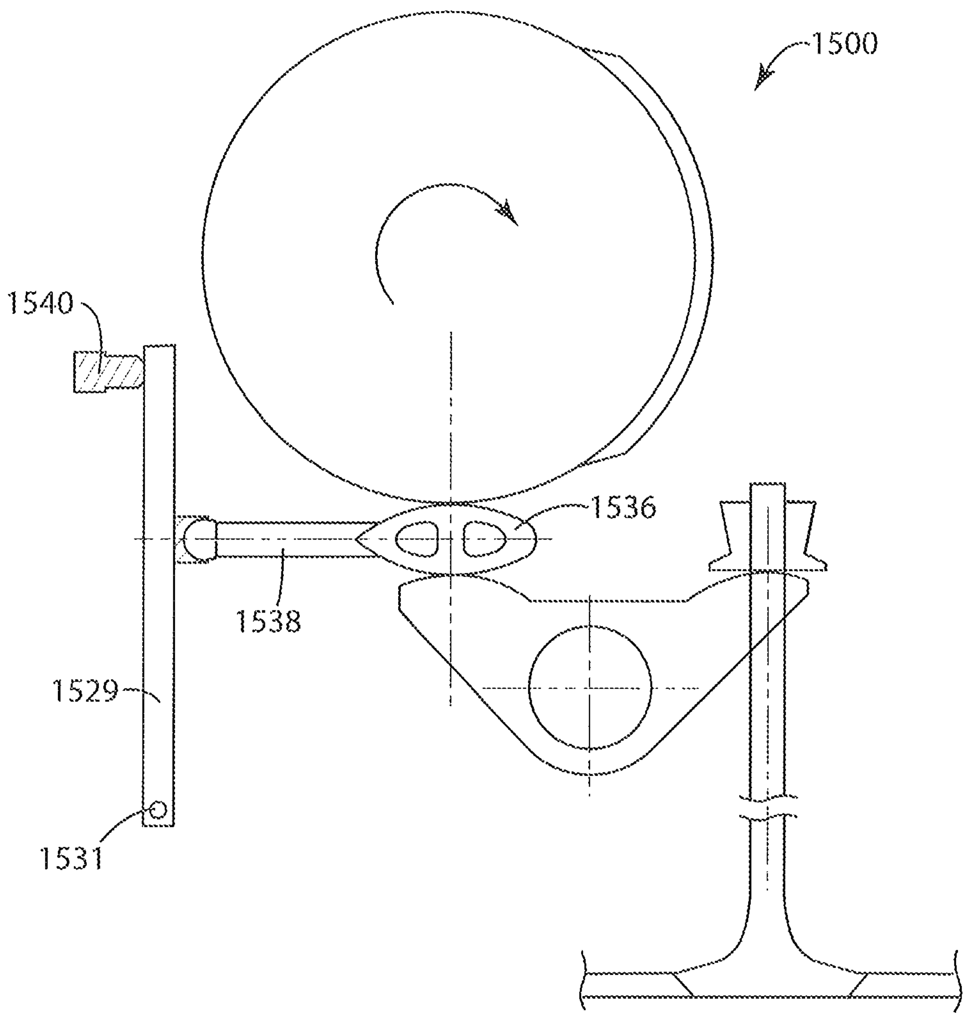
FIG. 15



LOST-MOTION VARIABLE VALVE ACTUATION SYSTEM WITH CAM PHASER

CROSS-REFERENCE TO RELATED APPLICATIONS

This application claims the benefit of priority of U.S. Provisional Patent Application No. 61/436,735, filed on Jan. 27, 2011, the entire contents of which are incorporated herein by reference.

FIELD

The present invention relates to valve actuation systems. More particularly, the invention relates to a split-cycle internal combustion engine having a lost-motion variable valve actuation system.

BACKGROUND

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.

It can be desirable to alter the valve lift, opening rate, opening timing, closing timing, closing rate, and various other valve parameters to achieve optimum engine efficiency for a variety of operating speeds, loads, temperatures, etc. In addition, in an air hybrid engine in which kinetic energy generated from a vehicle's momentum is recycled using air as the medium, certain hybrid operating modes require that one or more of the engine valves stay open longer or shorter than in other operating modes, and longer or shorter than in a non-hybrid, traditional combustion operating mode.

"Lost-motion" systems have been developed to permit a valve to close earlier than what is called for by the cam. Lost-motion systems generally include a lost-motion valve train element that can be selectively actuated to operatively disconnect a cam from a valve during a portion of the cam's rotation. The motion that would have otherwise been imparted to the valve (had the valve not been operatively disconnected) is thus lost.

Existing lost-motion systems, however, suffer from many shortcomings. For example, the moving components of existing systems are either too heavy or lack the requisite stiffness to be used in high speed and high pressure applications.

Accordingly, there is a need for improved methods and devices for varying the opening and closing parameters of an engine valve.

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 approximately 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.

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.

A split-cycle engine generally comprises:

a crankshaft rotatable about a crankshaft axis;

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;

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

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.

FIG. **1** illustrates a prior art split-cycle, non-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. 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.

The four strokes of the Otto cycle are thus "split" over the two cylinders **102**, **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).

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, the 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**.

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 and open ports) 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.

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.

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 top dead center (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.

The XovrE valve **116** is 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.

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.

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** can be 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.

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.

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** can each 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.

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).

Further detail on split-cycle engines can be found in U.S. Pat. No. 6,543,225 entitled Split Four Stroke Cycle Internal Combustion Engine and issued on Apr. 8, 2003; U.S. Pat. No.

6,609,371 entitled Split Four Stroke Engine and issued on Aug. 26, 2003; and U.S. Pat. No. 6,952,923 entitled Split-Cycle Four-Stroke Engine and issued on Oct. 11, 2005, each of which is incorporated by reference herein in its entirety.

FIG. **2** illustrates a prior art air hybrid engine in which a split-cycle engine **200** similar to that shown in FIG. **1** is modified to include an air hybrid system. The split-cycle air hybrid engine **200** 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:

a crankshaft rotatable about a crankshaft axis;

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;

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;

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 including a crossover compression (XovrC) valve and a crossover expansion (XovrE) valve defining a pressure chamber therebetween; and

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.

Like the engine **100** shown in FIG. **1**, the engine **200** includes an engine block **201** having a compression cylinder **202** and an adjacent expansion cylinder **204** extending therethrough. A crankshaft **206** is journaled in the block **201** for rotation about a crankshaft axis. Upper ends of the cylinders **202**, **204** are closed by a cylinder head **230**.

The first and second cylinders **202**, **204** define internal bearing surfaces in which are received for reciprocation a compression piston **210** and a power (or “expansion”) piston **220**, respectively. The cylinder head **230**, the compression piston **210** and the first cylinder **202** define a variable volume compression chamber **234** in the compression cylinder **202**. The cylinder head **230**, the power piston **220** and the second cylinder **204** define a variable volume combustion chamber **232** in the power cylinder **204**.

The crankshaft **206** includes axially displaced and angularly offset first and second crank throws **226**, **228**, having a phase angle **236** therebetween. The first crank throw **226** is pivotally joined by a first connecting rod **238** to the compression piston **210**, and the second crank throw **228** is pivotally joined by a second connecting rod **240** to the power piston **220** to reciprocate the pistons in their respective cylinders 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.

The cylinder head **230** includes any of various passages, ports, and valves suitable for accomplishing the desired purposes of the split-cycle air hybrid engine **200**.

Valves in the cylinder head **230**, which are similar to valves of the engine in FIG. **1**, include four cam actuated poppet valves: an intake valve **208**, an XovrC valve **214**, an XovrE valve **216**, and an exhaust valve **224**. An air reservoir tank valve **252** is also provided. The poppet valves **208**, **214**, **216**, **224** and the air reservoir tank valve **252** can be actuated by camshafts (not shown) having cam lobes for respectively actuating and engaging the valves **208**, **214**, **216**, **224**, **252**.

A spark plug **222** is mounted in the cylinder head with electrodes extending into the combustion chamber **232** for igniting air fuel charges at precise times by an ignition control, not shown. It should be understood that the engine can also be a diesel engine and be operated without a spark plug. Moreover, the engine **200** can be designed to operate on any fuel suitable for reciprocating piston engines in general, such as hydrogen or natural gas.

The split-cycle air hybrid engine **200** also includes an air reservoir (tank) **242**, which is operatively connected to the crossover passage **212** by the air reservoir tank valve **252**. Embodiments with two or more crossover passages **212** may include a tank valve **252** for each crossover passage **212**, which connect to a common air reservoir **242**, or alternatively each crossover passage **212** may operatively connect to separate air reservoirs **242**.

The tank valve **252** is typically disposed in an air tank port **254**, which extends from the crossover passage **212** to the air tank **242**. The air tank port **254** is divided into a first air tank port section **256** and a second air tank port section **258**. The first air tank port section **256** connects the air tank valve **252** to the crossover passage **212**, and the second air tank port section **258** connects the air tank valve **252** to the air tank **242**. The volume of the first air tank port section **256** includes the volume of all additional recesses which connect the tank valve **252** to the crossover passage **212** when the tank valve **252** is closed. Preferably, the volume of the first air tank port section **256** is small relative to the volume of the crossover passage **212** (e.g., less than 25%). More preferably, the first air tank port section **256** is substantially non-existent, that is, the tank valve **252** is most preferably disposed such that it is flush against the outer wall of the crossover passage **212**.

The tank valve **252** may be any suitable valve device or system. For example, the tank valve **252** 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 **252** may comprise a tank valve system with two or more valves actuated with two or more actuation devices.

The air tank **242** is utilized to store energy in the form of compressed air and to later use that compressed air to power the crankshaft **206**. 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 **200** can potentially provide many advantages in fuel efficiency gains and NOx 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.

The engine **200** typically runs in a normal operating mode (engine firing (EF) mode or sometimes called the normal firing (NF) mode) and one or more air hybrid modes. In the EF mode, the engine **200** functions normally as previously described in detail herein (i.e., with respect to FIG. **1**), operating without the use of the air tank **242**. In the EF mode, the air tank valve **252** remains closed to isolate the air tank **242** from the basic split-cycle engine. In the four air hybrid modes, the engine **200** operates with the use of the air tank **242**.

Exemplary Air Hybrid Modes Include:

1) Air Expander (AE) mode, which includes using compressed air energy from the air tank **242** without combustion;

2) Air Compressor (AC) mode, which includes storing compressed air energy into the air tank **242** without combustion;

3) Air Expander and Firing (AEF) mode, which includes using compressed air energy from the air tank **242** with combustion; and

4) Firing and Charging (FC) mode, which includes storing compressed air energy into the air tank **242** with combustion.

Further details on air hybrid engines are disclosed in U.S. Pat. No. 7,353,786 entitled Split-Cycle Air Hybrid Engine and issued on Apr. 8, 2008; U.S. Pat. No. 7,603,970 entitled Split-Cycle Air Hybrid Engine and issued on Oct. 20, 2009; and U.S. Publication No. 2009/0266347 entitled Split-Cycle Air Hybrid Engine and published on Oct. 29, 2009, each of which is incorporated by reference herein in its entirety.

In order to operate the split-cycle engines **100**, **200** described above at a high efficiency, a valve actuation system is required that is capable of (1) opening and closing the crossover valves at an extremely high speed, (2) providing a broad range of crossover valve opening and closing timings, and (3) allowing cycle-to-cycle variation in at least the closing timing. These requirements stem from the unique properties of split-cycle engines and, in particular, split-cycle air hybrid engines.

First, in these split-cycle engines, the dynamic actuation of the crossover valves (i.e. **114**, **116**, **214**, **216**) is very demanding. This is due to the fact that the crossover valves must achieve sufficient lift to fully transfer the fuel-air charge in a very short period of crankshaft rotation (possibly as little as 6 degrees CA) relative to that of a conventional engine, which normally actuates the valves for a period of at least 180 degrees CA. For example, when operating in EF mode, it is desirable to open the XovrE valve, transfer a fluid charge into the expansion cylinder, and close the XovrE valve while the expansion piston is very close to TDC. Thus, the XovrE valve must typically open and close in a window of about 30 degrees CA to about 35 degrees CA. Under full load conditions, this window is even smaller, perhaps as little as about 10 degrees CA to about 20 degrees CA.

Certain air hybrid modes introduce even more stringent requirements. In AEF mode, for example, a volume of compressed air is stored in the air reservoir **242**. Shortly after the expansion piston reaches TDC, the XovrE valve is opened to direct a charge of compressed air (preferably with added fuel) from the reservoir **242** into the combustion chamber where it is then ignited during an expansion stroke. If the engine is operating under only part load and the air reservoir **242** is charged to a high pressure (e.g., above approximately 20 bar), the XovrE valve only needs to be opened for a very short period (e.g., about 6 degrees CA) to transfer the requisite mass of air and fuel into the combustion chamber **232**. In other words, the relatively small mass of air-fuel mixture required for part-load operation will quickly flow into the combustion chamber when the air reservoir **242** is charged to a high pressure and therefore the XovrE valve need only open for a few degrees CA. The crossover valves must therefore be capable of actuation rates that are several times faster than the valves of a conventional engine, which means the valve train associated therewith must be stiff enough and at the same time light enough to achieve such fast actuation rates.

Meanwhile, other operating modes may require that the valves stay open for a relatively long period of time. For example, in AE mode, a volume of compressed air stored in the air reservoir **242** is delivered to the combustion chamber **232** without spark or added fuel, forcing the expansion piston down and providing power to the crankshaft. If, however, the air pressure remaining in the reservoir is low (e.g., less than approximately 15 bar) and there is a high torque requirement (e.g., when a vehicle being powered by the engine is accelerating up a hill), the XovrE valve must remain open much longer to allow a sufficient mass of compressed air into the expansion chamber. In some cases, this can be 100 degrees CA or more. Thus, large variations in closing timing are required, since the XovrE valve might need to close 6 degrees CA after opening in one operating mode while it may need to remain open for 100 degrees CA or more in other operating modes, as presented above.

The engines disclosed herein can also require large variations in the opening timing of the crossover valves **214**, **216**, especially in modes that involve charging the air reservoir (e.g., AC mode and FC mode). In AC mode for instance, the opening timing of the XovrC valve **214** will vary considerably depending on load and the pressure in the air reservoir **242**. If the XovrC valve is opened before the pressure in the compression cylinder is greater than or equal to the pressure in the air reservoir, fluid in the air reservoir will undesirably flow back into the compression cylinder **234**. The energy required to re-compress this backflow reduces the efficiency of the engine. Therefore, the XovrC valve should not be opened until the pressure in the compression cylinder matches or exceeds that of the air reservoir **242**. Thus, a range of approximately 30 to 60 degrees CA of opening timing variability is required for the XovrC valve, depending on the pressure in the air reservoir.

Accordingly, the opening timing, closing timing, and/or various other engine valve parameters must be variable over a wide range of possible values in order to efficiently operate each of the various engine modes.

Moreover, these parameters must be, in some cases, adjustable on a cycle-to-cycle basis. For example, the XovrE valve **216** can be used for load control in operating modes that employ combustion (e.g., EF mode and AEF mode). By closing the XovrE valve at various points along the expansion piston's stroke, the mass of air/fuel supplied to the cylinder can be metered, thereby controlling the engine load. To achieve precise load control in this case, the actuation rate of the XovrE valve must be variable from one cycle to the next.

Existing valve actuation systems are simply incapable of meeting these requirements. They are either too heavy or not stiff enough to be actuated at the required speeds. In addition, they provide only a limited range of opening or closing variability and are not responsive enough for cycle-to-cycle variation.

SUMMARY

Devices and related methods are disclosed that generally involve variable actuation of engine valves. In one embodiment, a valve train for a split-cycle internal combustion engine or an air hybrid split-cycle engine is provided that meets the aforementioned requirements by combining a cam phaser for varying the opening timing of the engine valves, a dwell cam for providing a large maximum possible valve event (e.g., 50-100 degrees CA), and a high speed lost-motion system for varying the closing timing of the engine valves. The devices and methods disclosed herein also have applica-

tion in conventional internal combustion engines and can be adapted to actuate inwardly-opening and/or outwardly-opening valves.

In one aspect of at least one embodiment of the invention, an engine is provided that includes a camshaft having at least one cam formed thereon, the at least one cam being configured to impart motion to at least one engine valve over a maximum valve event measured in degrees crank angle. The engine also includes a cam phaser that selectively adjusts a phase of the at least one cam relative to a crankshaft and a lost-motion system that selectively prevents the at least one cam from imparting motion to the at least one engine valve over the entire maximum valve event.

In another aspect of at least one embodiment of the invention, an adjustable mechanical element is provided that includes a bearing element having opposed convex bearing surfaces, a connecting arm having a proximal end and a distal end, the distal end being fixedly coupled to the bearing element and the proximal end having a cylinder or ball formed thereon, and an adjustable hydraulic tappet having a socket formed in one end thereof for receiving the cylinder or ball of the connecting arm.

In another aspect of at least one embodiment of the invention, a rocker is provided that includes a body portion having an opening formed therein for receiving a rocker shaft. The rocker also includes a first arm extending radially from the body and having a first rocker pad formed thereon for engaging an engine valve and a second arm extending radially from the body and having a second rocker pad formed thereon for engaging a motion element. The rocker further includes a third arm extending radially from the body, the third arm being engaged by a valve seating control device.

In another aspect of at least one embodiment of the invention, a method of varying the opening and closing timing of an engine valve is provided that includes varying the opening timing of the engine valve and varying the closing timing of the engine valve.

In another aspect of at least one embodiment of the invention, a method of actuating an engine valve is provided that includes rotating a cam having an eccentric portion such that the eccentric portion engages a first surface of a bearing element, thereby causing a second surface of the bearing element to engage a rocker coupled to the engine valve, the bearing element being disposed between the cam and the rocker. The method also includes adjusting an opening timing at which the eccentric portion first engages the bearing element by actuating a cam phaser to change a phase of the cam relative to a crankshaft. The method further includes adjusting a closing timing at which the engine valve begins to close such that the engine valve closes earlier than what is called for by the cam by at least partially withdrawing the bearing element from between the cam and the rocker.

In another aspect of at least one embodiment of the invention, an adjustable mechanical element is provided that includes a bell crank having first and second ends, the first end being rotatably mounted about a pivot point. The adjustable mechanical element also includes an adjustable hydraulic tappet configured to selectively apply force to the second end of the bell crank and a connecting arm having a proximal end and a distal end, the distal end being fixedly coupled to a bearing element and the proximal end being pivotally coupled to the bell crank at a location intermediate to the first and second ends.

In another aspect of at least one embodiment of the invention, a rocker assembly is provided that includes a rocker mounted to a rocker pedestal having an adjustable height and a wedge-shaped bearing element slidably disposed between first and second portions of the rocker pedestal. Withdrawing the wedge-shaped bearing element from between the first and second portions is effective to decrease the height of the pedestal.

In another aspect of at least one embodiment of the present invention, a locking knee assembly is provided that includes an outer housing slidably disposed relative to a lash cylinder, a femur having a first end and an opposed second end, the first end being rotatably coupled to an interior of the outer housing, and a shin rotatably coupled to the second end of the femur at a knee joint. The assembly also includes a hydraulic actuation piston configured to selectively apply a force to the knee joint to hold the femur in a fixed angular orientation relative to the shin.

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

BRIEF DESCRIPTION OF THE DRAWINGS

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

FIG. **1** is a schematic cross-sectional view of a prior art split-cycle engine;

FIG. **2** is a schematic cross-sectional view of a prior art air hybrid split-cycle engine;

FIG. **3**A is a schematic view of one embodiment of a valve train according to the present invention in which a valve is closed;

FIG. **3**B is a schematic view of the valve train of FIG. **3**A in which the valve is opened;

FIG. **3**C is a schematic view of the valve train of FIGS. **3**A and **3**B in which the valve is closed earlier than what is called for by a profile of a cam;

FIG. **4**A is a perspective cross-sectional view of one embodiment of a locking knee tappet according to the present invention;

FIG. **4**B is a perspective view of the locking knee tappet of FIG. **4**A with a portion of the outer housing not shown;

FIG. **4**C is a schematic cross-sectional side view of the locking knee tappet of FIGS. **4**A-**4**B in an extended configuration;

FIG. **4**D is a schematic cross-sectional side view of the locking knee tappet of FIGS. **4**A-**4**C in a retracted configuration;

FIG. **5**A is a side view of another embodiment of a bearing element according to the present invention;

FIG. **5**B is a side view of another embodiment of a bearing element according to the present invention;

FIG. **5**C is a side view of another embodiment of a bearing element according to the present invention;

FIG. **6**A is a side view of another embodiment of a bearing element according to the present invention;

FIG. **6**B is an end view of the bearing element of FIG. **6**A;

FIG. **6**C is a perspective view of the bearing element of FIGS. **6**A and **6**B;

FIG. **7**A is a schematic view of one embodiment of an adjustable mechanical element according to the present invention having a roller on a cam engaging surface of a bearing element;

FIG. **7**B is a schematic view of another embodiment of an adjustable mechanical element according to the present invention having a roller on a rocker engaging surface of a bearing element;

FIG. **7**C is a schematic view of another embodiment of an adjustable mechanical element according to the present invention having rollers on both a cam engaging surface and a rocker engaging surface of a bearing element;

FIG. **8** is a schematic view of one embodiment of a valve train according to the present invention having a rocker with a roller mounted thereon;

FIG. **9**A is a schematic view of another embodiment of a valve train according to the present invention having a valve seating control device;

FIG. **9**B is a schematic view of another embodiment of a valve train according to the present invention having a valve seating control device;

FIG. **9**C is a schematic view of another embodiment of a valve train according to the present invention having a valve seating control device;

FIG. **10**A is a schematic view of one embodiment of a valve train according to the present invention having a collapsible rocker;

FIG. **10**B is a perspective view of another embodiment of a valve train according to the present invention having a locking knee collapsible rocker pedestal;

FIG. **10**C is a schematic cross-sectional side view of the locking knee collapsible rocker pedestal of FIG. **10**B in an extended configuration;

FIG. **10**D is a schematic cross-sectional side view of the locking knee collapsible rocker pedestal of FIGS. **10**B-**10**C in a collapsed configuration;

FIG. **10**E is a schematic cross-sectional side view of one embodiment of a low profile locking knee collapsible rocker pedestal according to the present invention in an extended configuration;

FIG. **10**F is a schematic cross-sectional side view of the low profile locking knee collapsible rocker pedestal of FIG. **10**E in a collapsed configuration;

FIG. **11**A is a schematic cross-sectional view of one embodiment of a cam phaser according to the present invention;

FIG. **11**B is a schematic cross-sectional view of the cam phaser of FIG. **11**A in an advanced position;

FIG. **11**C is a schematic cross-sectional view of the cam phaser of FIGS. **11**A and **11**B in a retarded position;

FIG. **12**A is a graph of valve lift as a function of crank angle for a valve actuated by one embodiment of a valve train according to the present invention;

FIG. **12**B is a graph of valve lift as a function of crank angle for a valve actuated by one embodiment of a valve train according to the present invention;

FIG. **13** is a schematic view of one embodiment of a valve train according to the present invention for actuating an inwardly-opening valve;

FIG. **14** is a schematic cross-sectional view of one embodiment of a split-cycle engine according to the present invention; and

FIG. **15** is a schematic view of one embodiment of a valve train according to the present invention that includes a bell crank.

DETAILED DESCRIPTION

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.

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.

As explained above, in order to operate the split-cycle engines disclosed herein at maximum efficiency, and in particular to operate each of the various air hybrid modes contemplated herein, it is desirable to vary the opening timing, opening rate, closing timing, closing rate, lift, and/or various other engine valve parameters.

FIGS. **3**A-**3**C illustrate one exemplary embodiment of a valve train suitable for adjusting the aforementioned valve parameters (i.e., by modifying the valve motion proscribed by a cam profile). The illustrated valve train can be used to actuate any of the valves of the engines **100**, **200**, including without limitation the XovrC and XovrE crossover valves. For purposes herein, a valve train of an internal combustion engine is defined as a system of valve train elements, which are used to control the actuation of the valves. The valve train elements generally comprise a combination of actuating elements and their associated support elements. The actuating elements (e.g., cams, tappets, springs, rocker arms, and the like) are used to directly impart the actuation motion to the valves (i.e., to actuate the valves) of the engine during each valve event. The support elements (e.g., shafts, pedestals or the like) securely mount and guide the actuating elements.

As shown in FIG. **3**A, the valve train **300** generally includes a cam **302**, a rocker **304**, a valve **306**, and an adjustable mechanical element **308**. The valve train **300** can also include one or more associated support elements, which for purposes of brevity are not illustrated.

The valve **306** includes a valve head **310** and a valve stem **312** extending vertically from the valve head **310**. A valve adapter assembly **314** is disposed at the tip of the stem **312** opposite the head **310** and is securely fixed thereto. A valve spring (not shown) holds the valve head **310** securely against a valve seat **316** when the valve **306** is in its closed position. Any of a variety of valve springs can be used for this purpose, including, for example, air or gas springs. In addition, although the illustrated valve **306** is an outwardly-opening poppet valve, any cam actuated valve can be used, including inwardly-opening poppet valves, without departing from the scope of the present invention.

The rocker **304** includes a forked rocker pad **320** at one end, which straddles the valve stem **312** and engages the underside of the valve adapter assembly **314**. Additionally, the rocker **304** includes a solid rocker pad **322** at an opposing end, which slidably contacts the adjustable mechanical element **308**. The rocker **304** also includes a rocker shaft bore **324** extending therethrough. The rocker shaft bore **324** is disposed over a supporting rocker shaft **328** such that the rocker **304** rotates on the rocker shaft **328** about an axis of rotation **329**.

The forked rocker pad **320** of the rocker **304** contacts the valve adapter assembly **314** of the outwardly-opening poppet valve **306** such that a downward direction of the rocker pad **322** caused by the actuation of the cam **302** and adjustable mechanical element **308** translates into an upward movement of the rocker pad **320**, which in turn opens the valve **306**. The geometry of the rocker **304** is selected to achieve a desired ratio of the distance between the forked rocker pad **320** and the axis of the rocker rotation **329** to the distance between the rocker pad **322** and the axis of rocker rotation **329**. In one embodiment, this ratio can be between about 1:1 and about 2:1, and preferably about 1.3:1, about 1.4:1, about 1.5:1, about 1.6:1, or about 1.7:1.

The cam **302** 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. In the illustrated embodiment, the dwell cam **302** rotates clockwise (in the direction of the arrow **A1**). The dwell cam **302** generally includes a base circle portion **318** and an eccentric portion **326**. As the eccentric portion **326** of the cam **302** contacts the adjustable mechanical element **308**, the adjustable mechanical element pivots, which then causes the rocker **304** to rotate about the rocker shaft **328** to lift the valve **306** off of its seat **316**.

The eccentric portion **326** comprises an opening ramp **330**, a closing ramp **332**, and a dwell section **334**. The dwell section **334** can be of various sizes, (i.e., at least 5 degrees CA) and in the illustrated embodiment, is sized to match the longest possible valve event duration (i.e., maximum valve event) needed over a full range of engine operating conditions and/or air hybrid modes. For purposes herein, the dwell section **334** is referred to as being part of the eccentric portion **326** of the cam **302**, even though the dwell section **334** is concentric with the base circle portion **318** of the cam **302** in the illustrated embodiment. The opening ramp **330** of the cam **302** is contoured to a shape that adequately achieves the desired lift of the engine valve **306** at the desired rate. The closing ramp **332** (or "landing" ramp) is shaped to rapidly decelerate the velocity of the valve **306** as it approaches the valve seat **316** and/or provide for refill or resetting of an adjustable hydraulic tappet **340**, as discussed below. Further detail on dwell cams can be found in U.S. application Ser. No. 13/359,525, filed on an even date herewith, entitled "SPLIT-CYCLE AIR HYBRID ENGINE WITH DWELL CAM," which is hereby incorporated by reference in its entirety.

The adjustable mechanical element **308** is used to selectively vary the lift and the opening and closing parameters of the valve **306**. In the embodiment of FIGS. **3A-3C**, the adjustable mechanical element **308** includes a bearing element **336**, a connecting arm **338**, and an adjustable hydraulic tappet **340**.

As shown, the bearing element **336** has a generally elliptical-shaped cross-section defined by opposed first and second bearing surfaces **342**, **344**, each having a generally convex profile. The bearing surfaces **342**, **344** can have any of a variety of cross-sectional shapes, including circular and elliptical. In some embodiments, the bearing surfaces **342**, **344** can be sections of circles having different radii of curvature (e.g., such that the bearing surface **342** has a radius of curvature that is less than a radius of curvature of the bearing surface **344**). The bearing element **336** is selectively positioned between the cam **302** and the rocker **304** such that the first bearing surface **342** slidably engages the cam **302** and the second bearing surface **344** slidably engages the rocker pad **322**. The bearing element **336** has one or more cavities **346** formed therein, for example to reduce the overall mass of the bearing element **336** and thus facilitate faster actuation.

The bearing element **336** is coupled to the adjustable hydraulic tappet **340** via at least one connecting arm **338**. The connecting arm **338** in the illustrated embodiment is a generally cylindrical arm having a proximal end **348** and a distal end **349**. In some embodiments, the connecting arm **338** can have the shape of an I-beam. The distal end **349** of the connecting arm **338** is coupled to the bearing element **336** while the proximal end **348** of the connecting arm **338** is coupled to the tappet **340**.

The connecting arm **338** can be mated to the tappet **340** and to the bearing element **336** in a variety of ways. For example, the connecting arm **338** can be fixedly mated to the tappet **340** and/or the bearing element **336** with, for example, a screw, bolt, snap-fit engagement, etc., can be formed integrally with the tappet **340** and/or the bearing element **336**, or can be pivotally mated to either or both of the tappet **340** and the bearing element **336**. In the illustrated embodiment, the connecting arm **338** is formed integrally with the bearing element **336**. The proximal end **348** of the connecting arm **338** has a generally spherical ball **350** formed thereon. In some embodiments, the spherical ball **350** can be replaced with a cylindrical bearing. The ball **350** is sized and otherwise configured to be received by a corresponding socket **352** formed in a distal end of the tappet **340**, such that the connecting arm **338** is pivotable with respect to the tappet **340**. In other words, the connecting arm **338** is free to rotate about a plurality of rotational axes substantially transverse to a longitudinal axis of the tappet **340**. In embodiments in which a cylindrical bearing is used, rotation of the connecting arm **338** can be limited to rotation about a single axis that is substantially transverse to the longitudinal axis of the tappet **340**. The connecting arm **338** can also be mated to the tappet **340** such that it rotates about a pivot pin, axle, or other coupling. Although the bearing element **336** is formed integrally with the connecting arm **338** in the illustrated embodiment, it can also be pivotally coupled thereto using any of the techniques described above for mating the connecting arm **338** to the tappet **340**.

The tappet **340** is adjustable such that the connecting arm **338** and the bearing element **336** coupled thereto can be selectively advanced towards or retracted from the cam **302** and rocker **304** (i.e., in a lateral direction).

In one embodiment, the tappet **340** is configured to exert both a pulling force and a pushing force on the connecting arm **338** and the bearing element **336**. For example, the tappet **340** can define an internal cavity in which a piston is slidably received. The piston forms a seal with the inner surface of the cavity such that first and second fluid chambers are defined thereby, one on each side of the piston. The piston is operatively coupled to the socket **352** and/or the connecting arm **338** such that linear motion of the piston imparts a corresponding linear motion to the connecting arm. The first and second fluid chambers defined within the tappet **340** are selectively filled with and drained of a hydraulic fluid to move the piston (and thus the bearing element **336**) towards or away from the cam **302** and the rocker **304**.

Alternatively, the tappet **340** can be configured only to exert a pushing force on the bearing element **336**, in which case forces supplied by the cam, the rocker, and/or one or more bias springs are used to force the bearing element **336** into a retracted position. For example, the tappet **340** can include first and second cylindrical telescoping halves defining a fluid chamber between the respective interiors thereof. When the tappet **340** is actuated, fluid can be displaced from the fluid chamber into a hydraulic accumulator allowing the first and second telescoping halves to slide relative to and towards one another, thereby reducing the overall length L of the tappet **340**. In one embodiment, the tappet can be actuated by a solenoid valve and a check valve in communication with the hydraulic circuit including the fluid chamber and accumulator. The solenoid valve can be maintained in a closed position in order to retain hydraulic fluid in the circuit. As long as the solenoid valve remains closed, the length L of the tappet **340** remains substantially constant. When the solenoid valve is opened temporarily, the circuit partially drains, allowing the tappet **340** to partially or fully collapse, thus reducing the length L thereof. When the solenoid valve is again closed, the accumulator selectively refills the tappet **340**, causing it to expand linearly such that the overall length L thereof is increased.

Although the illustrated embodiment includes a hydraulic tappet **340** to advance and/or retract the connecting arm **338** and the bearing element **336**, a variety of other mechanisms can be employed for this purpose without departing from the scope of the present invention. For example, pneumatic, mechanical, electrical, and/or electromagnetic actuators can be used to impart motion to the connecting arm **338** and/or bearing element **336**. As discussed in further detail below, the tappet **340** can be mechanically locked and hydraulically actuated (i.e., a hydraulic system can be used to engage and disengage a mechanical locking device).

In operation, the cam **302** rotates clockwise as a camshaft to which it is mounted is driven by rotation of the engine's crankshaft. As shown in FIG. **3**A, when the base circle portion **318** of the cam **302** engages the bearing element **336**, the rocker **304** remains in a "fully closed" position in which the forked rocker pad **320** does not apply sufficient lifting force to the valve **306** to overcome the bias of the valve spring, and therefore the valve **306** remains closed. In the illustrated embodiment, the thickness of the bearing element **336** and the spacing between the cam **302** and rocker **304** are sized such that even when the thickest portion of the bearing element **336** is positioned between the base circle portion **318** of the cam **302** and the rocker **304**, the valve **306** remains closed.

As shown in FIG. **3**B, the eccentric portion **326** of the cam **302** engages the first bearing surface **342** of the bearing element **336** during a portion of the cam's rotation. The eccentric portion **326** imparts a downward motion to the bearing element **336**, causing the connecting arm **338** to pivot in a clockwise direction about the distal end of the tappet **340**. As the connecting arm **338** pivots, some or all of the downward motion of the bearing element **336** is imparted to the rocker **304**, which engages the second bearing surface **344** of the bearing element **336**. This results in a counterclockwise rotation of the rocker **304**, which in turn is effective to lift the valve **306** off of the seat **316**. Because the bearing surfaces **342**, **344** are curved such that the bearing element **336** has a variable thickness along a length thereof, the degree to which the valve **306** is lifted can be controlled by varying the degree to which the bearing element **336** is inserted between the cam **302** and the rocker **304**. For example, in FIG. **3**B, the bearing element **336** is inserted such that the thickest portion thereof is disposed between the thickest portion of the rocker pad **322** and the cam **302**, thereby imparting maximum lift to the valve **306**. A reduced valve lift is achieved by withdrawing the bearing element **336** slightly in the direction of the tappet **340**, as explained below. Since the solenoid valve (not shown) is closed in FIG. **3**B, the length L of the tappet **340** remains substantially constant and some or all of the motion imparted to the bearing element **336** is transferred to the valve **306**, lifting it off of the seat **316**. In other words, with the tappet **340** maintained at a constant length, the motion of the valve **306** will depend on the shape of the profile of the cam **302**.

As shown in FIG. **3**C, the valve train **300** is capable of closing the valve before the closing ramp **332** of the cam **302** reaches the bearing element **336**, and is capable of reducing the degree to which the valve **306** is opened. For example, the solenoid valve can be actuated to allow a sudden release of hydraulic fluid from the fluid chamber of the tappet **340**. When the fluid is allowed to escape the tappet **340**, a squeezing force acting on the bearing element **336** in the direction of the arrow A**2** is effective to push the bearing element **336** away from the cam **302** and the rocker **304**, compressing or collapsing the tappet **340** and forcing hydraulic fluid through the open solenoid valve. The squeezing force is generated by the combined force of the valve spring biasing the rocker arm **304** in a clockwise direction, coupled with the force of the cam's eccentric portion **326** rotating against the bearing element **336** in a clockwise direction. It will be appreciated that the squeezing force can be only a minor component of the force acting on the bearing element **336**, and that the bearing element **336** can be shaped such that the majority of the force of the cam **302** is applied downwards onto the rocker pad **322** and vice versa. It will also be appreciated that the degree to which the bearing element **336** is forced out from between the cam **302** and the rocker **304**, and thus the degree to which the valve **306** is allowed to close, can be controlled by adjusting the degree to which hydraulic fluid is permitted to escape from the tappet **340**. In other words, if the solenoid valve is opened briefly and then immediately closed, the tappet **340** will only collapse to a degree corresponding to the amount of fluid displaced from the fluid chamber, in which case the valve **306** will only partially close. This can be desirable when it is necessary to adjust the lift height of the valve **306**. Alternatively, the solenoid valve can be left open for a period long enough for the tappet **340** to compress far enough to allow the valve **306** to fully close.

In embodiments in which the tappet **340** is configured to both push and pull the connecting arm **338** and bearing element **336**, the tappet **340** can be controlled to actively pull the bearing element **336** away from the cam **302** and the rocker **304**, instead of relying on the aforementioned squeezing force.

In FIG. **3**C, the bearing element **336** is shown withdrawn far enough from the cam **302** and the rocker **304** such that insufficient motion is imparted from the eccentric portion **336** of the cam **302** to the rocker **304** for the valve **306** to actually be lifted off of the seat **316**, and thus the valve **306** closes or remains closed. The valve train **300** thus provides a lost-motion feature that allows for variable valve actuation (i.e., permits the valve **306** to close at an earlier time than that provided by the profile of the cam **302**). Furthermore, the valve train **300** permits the lift of the valve **306** to be varied, for example by varying the degree to which fluid is drained from the tappet **340** and thus the degree to which the valve is allowed to open or close. The valve train **300** is thus configured to transmit all of the cam motion to the valve **306**, to transmit only a portion of the cam motion to the valve **306**, or to transmit none of the cam motion to the valve **306**.

The adjustable mechanical element **308** can also be configured to take up any lash that may exist in the valve train **300**, for example due to thermal expansion and contraction, component wear, etc. For purposes herein, the terms "valve lash" or "lash" are defined as the total clearance existing within the valve train **300** when the valve **306** is fully seated. The valve lash is equal to the total contribution of all the individual clearances between all individual valve train elements (i.e., actuating elements and support elements) of the valve train **300**. In the valve train **300**, the bearing element **336** is biased towards the cam **302** and the rocker **304** such that any lash that may exist in the valve train **300** is taken up by the gradually increasing thickness of the bearing element **336**. The biasing force can be relatively low, such that once the lash is taken up by the bearing element **336**, the bearing element **336** is not advanced further towards the cam **302** or rocker **304** unless actuated to open the valve **306**. In this manner, the lash is taken up without the valve **306** opening during a period when it should be closed. The biasing force can be supplied in a variety of ways, for example hydraulically, via one or more springs, or via a well-known hydraulic lash adjuster integrally attached to the tappet.

FIGS. **4**A-**4**D illustrate one embodiment of a locking knee tappet **440** that can be used in place of the hydraulic tappet **340** discussed above to selectively advance and/or retract the connecting arm **338** and the bearing element **336**. As shown particularly in FIGS. **4**A-**4**B, the tappet **440** generally includes an outer housing **433** slidably disposed in a lash cylinder **435** such that a variable-volume, oil-filled, lash plenum **437** is defined therebetween. A locking knee joint defined by a femur **479** and a shin **439** is disposed within the outer housing **433**. Flexion of the knee joint is controlled/restricted by a hydraulic actuation piston **441** and a femur support platform **443**. The distal end **445** of the shin **439** is coupled to a foot **451** that is slidably guided along a substantially linear path by opposed guide plates **453**. The distal end **455** of the foot **451** can be coupled to the connecting arm **338** described above or, alternatively, can be coupled directly to the bearing element **336** described above.

The femur **479** has a generally rectangular cross-section and includes a central portion **457**, a proximal end **459**, and a distal end **461**. The proximal end **459** of the femur **479** is radiused so as to form an elongated hemi-cylindrical edge **463**. The hemi-cylindrical edge **463** is received in a corresponding hemi-cylindrical slot **465** formed in an upright portion **467** of the femur support platform **443**, which is in turn fixedly mated to the outer housing **433**. Accordingly, the femur **479** is rotatable relative to the platform **443** and the outer housing **433**. The platform **443** includes a base portion **469** configured to limit the range over which the femur **479** can be rotated. In one embodiment, the base portion **469** is sized and positioned such that an angle A (FIG. **4**C) between the femur **479** and the outer housing **433** cannot be less than approximately 8 degrees. It will be appreciated that the hemi-cylindrical edge **463** of the femur **479** can have a thickness greater than the central portion **457** of the femur **479**, and the sidewalls of the hemi-cylindrical slot **465** can be extended such that the edge **463** is captive within the slot **465**. It will further be appreciated that the term "hemi-cylindrical" as used herein is not limited to shapes having a constant radius or that form exactly half of a cylinder, but rather encompasses a variety of similar shapes. In addition, any of the hemi-cylindrical male-female interfaces described herein can be replaced with a comparable joining mechanism, such as a ball and socket joint, a universal joint, a continuously variable joint, a pin and sleeve joint, etc.

The shin **439** also has a generally rectangular cross-section and includes a central portion **471**, a proximal end **473**, and a distal end **445**. The proximal end **473** of the shin **439** is radiused so as to form an elongated hemi-cylindrical edge **475**. The hemi-cylindrical edge **475** is received in a corresponding hemi-cylindrical slot **477** formed in the distal end **461** of the femur **479** such that the shin **439** is rotatable relative to the femur **479**. The foot **451** likewise has a generally rectangular cross-section and includes a radiused proximal end **481** for mating with a corresponding slot **483** formed in the distal end **445** of the shin **439** such that the foot **451** is rotatable relative thereto. The male/female relationship of the shin **439** and the foot **451** can also be reversed, as shown in FIGS. **4**C and **4**D, as can the male/female relationship of any of the other components of the locking knee tappet **440**. The foot **451** also includes a mating feature (e.g., a male tab as shown in FIGS. **4**A-**4**B or a female receptacle as shown in FIGS. **4**C-**4**D) formed on the distal end **455** thereof for coupling to the connecting arm **338** or bearing element **336** described above. Alternatively, the bearing element **336** can be formed integrally with the foot **451**. As shown in FIG. **4**B, the foot **451** can optionally include opposed lateral ears **485** coupled to respective valve catch pistons **487**, the operation of which is described further below.

The hydraulic actuation piston **441** is reciprocally and sealably disposed within a cylindrical bore **489** formed in the sidewall of the outer housing **433** and is positioned to engage either the femur **479** or the shin **439**, preferably at a "knee" where the femur **479** and the shin **439** are rotatably coupled to one another. A hydraulic control circuit (not shown) is coupled to the hydraulic actuation piston **441** such that hydraulic pressure acting to push the piston **441** towards the "knee" can be selectively applied and relieved.

In operation, referring now to the schematic illustrations of FIGS. **4**C and **4**D, the locking knee tappet **440** has two general configurations. FIG. **4**C illustrates an extended configuration of the tappet **440** in which the femur **479** is rotated against the base portion **469** of the femur support platform **443** such that the femur **479** forms a relatively small angle A (e.g., approximately 8 degrees) with the outer housing **433**. The hydraulic control circuit (not shown) is controlled to maintain the hydraulic pressure that is exerted on the hydraulic actuation piston **441** such that the piston **441** exerts a holding force in the direction of the arrow A**1** against the "knee" (e.g., against the femur **479** as shown). This force prevents the knee from articulating when a lateral force is applied in the direction of the illustrated arrow A**2** (e.g., when an eccentric portion of a cam contacts a bearing element coupled to the end of the foot **451**).

When it is desired to close an engine valve earlier than what is called for by its corresponding cam (e.g., by withdrawing a bearing element from between the cam and a rocker as described above), the locking knee tappet **440** is transitioned to a retracted configuration, as shown in FIG. **4**D. When early valve closing is requested, a solenoid or other control valve in the hydraulic control circuit is opened to allow fluid holding the hydraulic actuation piston **441** in place to flow into an accumulator. Once the control valve is opened, the lateral component of the force exerted on the foot **451** by the cam and the rocker in the direction of the arrow A**2** forces the femur-shin joint to articulate, driving the hydraulic actuation piston **441** upwards and forcing hydraulic fluid through the control valve and into the accumulator. In one embodiment, the femur **479** forms a 35 degree angle A with respect to the outer housing **433** when fully articulated. Once the eccentric portion of the cam has rotated past the rocker, the accumulator refills the fluid chamber above the hydraulic actuation piston **441** and the control valve is closed, again locking the tappet **440** in the extended configuration.

The femur **479** and shin **439** are prevented from being "over-indexed" (e.g., articulating such that the angle A in FIGS. **4**C and **4**D is 0 degrees or a negative angle) by the base portion **469** of the femur support platform **443**, which defines a minimum angle A. In an alternative embodiment, however, the femur support platform **443** can be shaped differently or omitted altogether such that the femur **479** and shin **439** can be over-indexed. For example, the femur **479** and the shin **439** can be positioned such that a central longitudinal axis of the femur **479** is collinear with a central longitudinal axis of the shin **439**. In this embodiment, a second hydraulic actuation piston is provided on the opposite side of the femur **479** from the hydraulic actuation piston **441** to actively buckle the knee when lost-motion is required.

When the hydraulic actuation piston **441** is "unlocked" to allow the femur **479** and the shin **439** to articulate, the rate of articulation (and thus the rate at which the bearing element is withdrawn and the engine valve is closed) is controlled for at least a portion of the valve closing event by the valve catch pistons **487** (FIG. 4B) to which the foot **451** is coupled. The valve catch pistons **487** are disposed in corresponding valve catch cylinders **490** formed in the outer housing **433**. The valve catch cylinders **490** have one or more orifices **497** formed in the sidewalls thereof and are filled with hydraulic fluid. As the valve catch pistons **487** are driven into the valve catch cylinders **490** by retraction of the foot **451**, the hydraulic fluid contained in the cylinders **490** is ejected through the orifices **497**. As the pistons **487** are advanced deeper into the cylinders **490**, the orifices **497** are progressively occluded by the pistons **487**, restricting the rate at which hydraulic fluid can escape from the cylinders **490**. This rate restriction is effective to slow the movement of the pistons **487**, which ultimately reduces the closing velocity of the engine valve. It will thus be appreciated that the orifices **497** are sized, shaped, and positioned such that the speed of the engine valve is greatly reduced as the valve approaches its corresponding valve seat, thereby "catching" the valve and preventing it from crashing against the seat and damaging the engine. While two valve catch cylinder **490** and piston **487** combinations are shown in the illustrated embodiment, any number can be employed without departing from the scope of the present invention, such as zero, one, or more than two.

Throughout the operation of the locking knee tappet **440**, the outer housing **433** remains slidably disposed within the lash cylinder **435**, which is held in a fixed position relative to the valve train in which the tappet **440** is used. Pressurized hydraulic fluid applied to the lash plenum **437** is therefore effective to bias the outer housing **433** (and the bearing element coupled indirectly thereto) towards the cam and the rocker. Accordingly, the tappet **440** provides an automatic valve lash adjustment feature by which lash existing in the valve train is taken up by the graduated thickness of the bearing element **336**.

Additionally, the surface area of the outer housing **433**, which defines the lash plenum **437** and upon which hydraulic pressure is applied to minimize lash, is significantly larger (e.g., 2 to 1, 3 to 1, 4 to 1 or greater) than the surface area of the top of the actuation piston **441**, upon which hydraulic pressure is applied to lock the tappet in its extended configuration. Due to this significantly larger surface area, the stiffness of the lash plenum **437** in the direction of arrow A**2** is much larger than the stiffness of the hydraulic control circuit in the direction of arrow A**1**. Moreover, the stiffness of the lash plenum in the direction of arrow A**2** approaches the stiffness of a purely mechanical linkage.

The locking knee tappet **440** provides numerous advantages. For example, the linkage formed by the femur **479** and the shin **439** provides a variable mechanical advantage at the actuation piston **441** with respect to: 1) the force required to initiate extension of the tappet **440** from the retracted configuration and, 2) the force required to hold the tappet **440** in the extended configuration. When in the extended configuration, the majority of the lateral force applied to the tappet **440** by the valve train (in the direction of the arrow A**2** in FIGS. **4**C and **4**D) is directed longitudinally through the knee joint and into the upright portion **467** of the femur support platform **443**. Thus, only a small portion of the force exerted by the valve train acts in opposition to the holding force (in the direction of arrow A**1**) exerted by the hydraulic actuation piston **441**. In addition, the large mating surface areas defined by the various hemi-cylindrical socket joints (**463**, **465**, **475**, **477**, **481**, **483**) provide a large amount of friction, which resists articulation of the linkage. Thus, the frictional forces assist the hydraulic actuation piston **441** in maintaining the tappet **440** in the extended configuration. Accordingly, the amount of force required to be exerted on the knee by the hydraulic actuation piston **441** to hold the tappet **440** in the extended configuration is relatively small. This permits the use of a small hydraulic actuation piston **441** and a correspondingly small volume of hydraulic fluid, which makes actuation of the hydraulic piston **441** very fast. Since the tappet **440** only needs to be returned to the extended configuration when the bearing element coupled thereto is in contact with the base circle of the cam, relatively little force is required to articulate the knee back to the extended position, and the small hydraulic actuation piston **441** remains adequate.

The locking knee tappet **440** also advantageously provides a hydro-mechanical lost-motion system. Although the tappet **440** is hydraulically actuated (via the hydraulic actuation piston **441** and associated control circuit), the actual locking and longitudinal support of the tappet **440** is provided via a mechanical linkage (the femur **479**, the shin **439**, etc.). The mechanical nature of the locking function, combined with the very stiff lash plenum **437**, provides considerably more stiffness than in a purely hydraulic system. As a result, adequate stiffness can still be achieved with lower mass components, which can be actuated much faster than heavier and bulkier alternatives. The locking knee tappet **440** thus permits fast and consistent actuation of an engine valve.

Although the bearing element **336** of FIGS. **3**A-**3**C is shown as having a generally elliptical-shaped cross-section, the bearing element can have a variety of other cross-sections without departing from the scope of the present invention. FIGS. **5**A-**5**C illustrate various exemplary embodiments of bearing elements.

In FIG. **5**A, the bearing element **536**A has a quarter-circle or "shark-fin" cross section that forms first and second bearing surfaces **542**A, **544**A. The bearing element **536**A also includes a cavity **546**A for reducing a mass thereof.

In FIG. **5**B, the bearing element **536**B has a wedge-shaped cross section that defines first and second bearing surfaces **542**B, **544**B. The bearing element **536**B also includes at least one cavity **546**B.

In FIG. **5**C, the bearing element **536**C has a circular cross section that defines first and second bearing surfaces **542**C, **544**C. The bearing element **536**C also includes at least one cavity **546**C.

FIGS. **6**A-**6**C illustrate another exemplary embodiment of a bearing element **636** formed integrally with a connecting arm **638** for use with the valve trains disclosed herein. As shown, the bearing element **636** has a generally wedge-shaped profile defining first and second bearing surfaces **642**, **644**. First and second cavities **646**, **647** are formed on opposing sides of the bearing element **636** defining a connecting arm base portion therebetween. Although not required, the cavities **646**, **647** can advantageously reduce the mass of the bearing element **636**. The connecting arm **638** is formed integrally with the bearing element **636** and extends from the connecting arm base portion defined by the cavities **646**, **647**. A spherical ball or cylinder **650**, for example for mating with a corresponding socket in an adjustable tappet, is formed on the end of the connecting arm **638**.

It will be appreciated that the size, shape, and materials of the bearing elements disclosed herein can be selected to minimize the overall mass of the bearing element and to maximize its structural rigidity.

Any of the valve train components disclosed herein can also includes various features for reducing friction between the engagement surfaces thereof. For example, the bearing element can have one or more rollers rotatably mounted thereon, as shown for example in FIGS. **7**A-**7**C. In FIG. **7**A, a first, cam-side roller **754**A is provided on the first bearing surface **742**A of the bearing element **736**A for reducing friction between the bearing element **736**A and the cam. As the cam rotates in a clockwise direction against the first bearing surface **742**A, the roller **754**A rotates in a counter-clockwise direction, thus reducing the friction between the bearing element **736**A and the cam. Although only one roller **754**A is provided in the illustrated embodiment, a plurality of rollers can also be provided on the first bearing surface without departing from the scope of the present invention.

In FIG. **7**B, a second, rocker-side roller **756**B is provided on the second bearing surface **744**B for reducing friction between the bearing element **736**B and the rocker. As the rocker opens the valve and rotates counter-clockwise against the second bearing surface **744**B, the roller **756**B rotates in a clockwise direction. Similarly, as the rocker closes the valve and rotates clockwise against the second bearing surface **744**B, the roller **756**B rotates in a counter-clockwise direction. The opposite rotation of the roller **756**B reduces the friction between the bearing element **736**B and the rocker. Although only one roller **756**B is provided in the illustrated embodiment, a plurality of rollers can also be provided on the second bearing surface without departing from the scope of the present invention.

In FIG. **7**C, two rollers **754**C, **756**C are provided, one on each of the bearing surfaces **742**C, **744**C, for reducing friction between the bearing element **736**C and both the cam and the rocker. Operation of the rollers **754**C, **756**C is substantially the same as the operation of the rollers **754**A, **756**B discussed above. In the illustrated embodiment, the rollers **754**C, **756**C are rotatably mounted to the bearing element **736**C via first and second axles **758**C, **760**C. The rollers **754**C, **756**C can be mounted adjacent to a cavity **746**C formed in the bearing element **736**C. This can allow lubricating fluid picked up by the cam and transferred to the cam-side roller **754**C to drip or spray into the cavity **746**C. Once inside the cavity **746**C, the fluid can be picked up by the rocker-side roller **756**C and ultimately supplied to the engagement surfaces of the rocker and the bearing element.

In another embodiment, as shown in FIG. **8**, the valve train **800** can include a rocker **804** which has a roller **862** of its own, rotatably mounted in place of the rocker pad. The roller **862** is rotatably mounted on an axle **864** and is configured to rotate either clockwise or counterclockwise depending on the motion of the rocker **804**. The roller **862** thus reduces friction between the bearing element **836** and the rocker **804**.

Other friction reducing features can also be included in any of the valve trains disclosed herein. For example, friction-reducing coatings can be applied to various surfaces of the components thereof. As another example, one or more fluid jackets can be formed in the bearing element for supplying lubricating fluid to the bearing surfaces thereof through one or more openings formed in the bearing surfaces. It will be appreciated that various combinations of the friction-reducing features can be employed as well.

In cam-driven actuation systems, the closing or landing ramp of the cam generally dictates the speed at which the valve contacts its seat when the valve closes. If the valve is closed earlier than what is called for by the cam, however (i.e., by actuating a lost-motion system before the closing ramp reaches the rocker), the valve can undesirably "free-fall" under the stiffness of the valve spring. This can cause the valve to crash against the valve seat, causing damage to the valve, the seat, and/or other components of the valve train. Accordingly, it can be desirable to control the rate at which the valve closes.

In any of the valve trains **300**, **800** disclosed above, the shape of the bearing element **336**, **836** itself can be used to control the rate at which the valve is allowed to close. In other words, the bearing surfaces of the bearing element can act as a closing ramp, and by controlling the rate at which the bearing element is withdrawn from between the cam and the rocker, the rate at which the valve closes can likewise be controlled.

Alternatively, or in addition, any of the valve trains disclosed herein can include a valve seating control device or "valve catch" to rapidly decelerate the velocity of the valve as it approaches the valve seat when closing. FIGS. **9**A-**9**C illustrate exemplary embodiments of valve trains according to the present invention that include one or more valve seating control devices.

As shown in FIG. **9**A, the valve train **900**A includes a rocker **904**A that generally includes three radial extensions **966**A, **968**A, **970**A. The first extension **966**A includes the forked rocker pad **920**A for engaging the valve **906**A. The second extension **968**A includes the rocker pad **922**A for engaging the bearing element **936**A. The third extension **970**A is coupled to a valve seating control device **972**A.

In the illustrated embodiment, the valve seating control device **972**A is a collapsible hydraulic tappet. The tappet can include a primary orifice that permits fluid to be released relatively freely from a fluid chamber formed within the tappet. The tappet can also include a secondary orifice, smaller than the primary orifice, configured to limit the rate at which fluid is permitted to escape from the fluid chamber. In use, when the valve approaches the valve seat during a closing event, the primary orifice can be closed. Fluid can then only escape from the collapsible tappet through the smaller secondary orifice and thus escapes at a much slower rate. This reduction in the rate at which fluid can escape from the fluid chamber results in a corresponding deceleration of the rocker and thus of the valve.

The primary orifice can be closed in a variety of ways. For example, a high-speed solenoid valve can be actuated to "close" the primary orifice. Alternatively, or in addition, the primary orifice can become occluded as the valve closes by a moving piston or other component of the collapsible tappet. Further details on valve seating control devices can be found in U.S. Patent Publication No. 2010/0180875, entitled "SEATING CONTROL DEVICE FOR A VALVE FOR A SPLIT-CYCLE ENGINE," filed on Jan. 22, 2009, the entire contents of which are hereby incorporated by reference herein.

The various radial extensions of the rocker can be positioned in a variety of angles with respect to each other to attain the clearance and leverage required for successful operation. For example, as shown in FIG. **9**B, a valve train **900**B includes a rocker **904**B in which the third extension **970**B is positioned at a slightly different angle with respect to the first and second extensions **966**B, **968**B and with respect to the rotational axis of the rocker **904**B. This geometry can be used for example when additional clearance is needed for the valve stem or the valve head, or when it is desirable to position the valve seating control device **972**B adjacent to the tappet **940**B of the adjustable mechanical element **908**B (i.e., for easier routing or placement of hydraulic lines and/or controls).

FIG. **9**C illustrates another embodiment of a valve train **900**C in which a valve seating control device **972**C engages a rocker **904**C having only first and second radial extensions **966**C, **968**C. In the illustrated embodiment, the forked rocker pad **920**C has an increased length such that the opposed fork portions thereof extend past the valve stem **912**C and valve adapter assembly **914**C of the valve **906**C when straddling the valve stem. The valve seating control device **972**C engages the underside of the opposed fork portions of the forked rocker pad **920**C where the fork portions extend beyond the valve stem **912**C. This embodiment of the rocker **904**C is desirable when it is necessary to reduce the size and mass of the rocker **904**C.

In the valve train **300** of FIGS. **3**A-**3**C, the lost-motion function is achieved by one or more elements disposed between the cam and the rocker. This need not always be the case, however. For example, lost-motion can also be achieved by one or more elements disposed between first and second portions of an adjustable pedestal on which the rocker is mounted such that the distance between the cam and the pivot point of the rocker can be adjusted. FIG. **10**A illustrates one embodiment of a valve train **1000** having such a configuration. As shown, the valve train **1000** includes a cam **1002**, a rocker **1004**, a valve **1006**, and an adjustable mechanical element **1008** including a bearing element **1036**. The rocker **1004** is mounted on a rocker shaft **1028** having a rectangular aperture **1092** formed therein. The aperture **1092** is sized to slidably receive a rectangular projection **1094** disposed on a rigidly fixed rocker support (not shown). The rectangular projection **1094** has a fixed position relative to the cam **1002** and can thus guide the vertical movement of the rocker **1004** and limit the degree to which the pivot point of the rocker **1004** can be adjusted.

The bearing element **1036** is disposed between opposed halves **1091**, **1093** of the rocker pedestal **1096** that are movable relative to each other such that sliding movement of the bearing element **1036** is effective to adjust a height H of the pedestal **1096**. In the illustrated embodiment, the bearing element **1036** has a wedge-shaped cross-section, although it will be appreciated that a variety of cross-sections can be used without departing from the scope of the present invention. A plurality of roller bearings **1098** can be provided to facilitate sliding movement of the bearing element **1036** relative to the pedestal **1096**. Also, in the illustrated embodiment, the upper half **1091** of the rocker pedestal **1096** extends through a slot **1095** in the rocker **1004** to integrally connect to the rocker shaft **1028**. The slot **1095** is sized to receive the upper half **1091** of the rocker pedestal **1096** and to allow for pivoted movement of the rocker **1004** during a valve event.

In operation, the cam **1002** rotates clockwise as a camshaft to which it is mounted is driven by rotation of the engine's crankshaft. When the base circle portion **1018** of the cam **1002** engages the rocker **1004**, the rocker **1004** remains in a position in which the forked rocker pad **1020** does not apply sufficient lifting force to the valve **1006** to overcome the bias of the valve spring, and therefore the valve **1006** remains closed on its seat.

As the cam **1002** rotates, a dwelled eccentric portion **1026** thereof engages the rocker **1004**. The eccentric portion **1026** imparts a downward force to the rocker **1004**, causing it to rotate counterclockwise and lift the valve **1006** off of its seat until the eccentric portion **1026** rotates past the rocker **1004** or until a lost-motion function is called for.

An adjustable tappet **1040** is used as described above to partially or fully withdraw the bearing element **1036** from the pedestal **1096** when a lost-motion function is called for (i.e., when it is desired to close the valve **1006** before the closing ramp **1032** of the cam **1002** reaches the rocker **1004** or to reduce the degree to which the valve **1006** is opened). As the bearing element **1036** is withdrawn, the downward force applied to the rocker **1004** by the cam **1002** and by the valve spring causes the upper portion **1091** of the pedestal **1096** and the rocker shaft **1028** attached thereto to move away from the cam **1002**. In other words, the pivot point of the rocker **1004** moves downward as the rocker shaft **1028** slides relative to the fixed projection **1094** inserted through the aperture **1092**.

When the bearing element **1036** is withdrawn far enough from the pedestal **1096**, insufficient motion is imparted from the cam **1002** to the rocker **1004** for the valve **1006** to actually be lifted off of its seat, and thus the valve **1006** closes or remains closed. The valve train **1000** thus provides a lost-motion feature that allows for variable valve actuation (i.e., permits the valve **1006** to close at an earlier time than that provided by the profile of the cam **1002**). Furthermore, the valve train **1000** permits the lift of the valve **1006** to be varied, for example by varying the degree to which the bearing element **1036** is withdrawn from or inserted into the pedestal **1096**.

It will be appreciated that the angle of the wedge-shaped bearing element **1036** can be adjusted to alter the magnitude of valve train forces that are exerted on the tappet **1040** and/or the amount of tappet stroke required to accomplish the lost-motion. For example, as the angle of the wedge approaches zero, the axial forces on the tappet **1040** decrease but the amount of stroke required for the tappet **1040** increases. Similarly, as the angle of the wedge approaches 90 degrees, the axial forces on the tappet **1040** increase while the amount of stroke required decreases. Higher axial forces require the use of a larger, sturdier tappet. Longer tappet strokes decrease the reaction time of the system, as it takes longer to actuate the tappet. Also, a shorter stroke reduces the effective mass, which results in a higher actuation speed, while a longer stroke increases the effective mass, which results in a slower actuation speed. The wedge shape of the bearing element **1036** permits these parameters to be optimized such that a reasonably-sized tappet can be used without sacrificing too much in the way of response time. The stroke of the tappet **1040** ranges between a lower value equal to the amount of valve lift to be lost and an upper value equal to about 2-3 times the amount of valve lift to be lost. The angle of the wedge ranges between about 0 degrees and about 25 degrees, and preferably is about 20 degrees. The angle of the wedge can also be adjusted based on the ratio of the rocker being used. It will be appreciated that the locking knee tappet **440** disclosed above can also be employed to advance and retract the bearing element **1036**.

FIGS. **10**B-**10**D illustrate another exemplary mechanism for collapsing the pivot point of a rocker to achieve a lost-motion effect. As shown in FIG. **10**B a locking knee collapsible rocker pedestal **1096"** is provided that includes a rocker shaft support housing **1033"** (shown in phantom) mounted above a knee linkage that includes a femur **1079"** and a shin **1039"**. A rocker **1004"** is rotatably mounted about a rocker shaft **1028"**, which is in turn fixedly mated to the support housing **1033"**. The support housing **1033"** includes a hemi-cylindrical slot **1065"** that receives a corresponding first hemi-cylindrical edge **1063"** of the femur **1079"** such that the femur **1079"** is rotatable relative to the support housing **1033"**. The femur **1079"** also includes a reduced-thickness central portion **1057"** and a second hemi-cylindrical edge **1077"** opposite the first hemi-cylindrical edge **1063"**. The second hemi-cylindrical edge **1077"** is received in a corresponding hemi-cylindrical slot **1075"** formed in the shin **1039"** such that the femur **1079"** and the shin **1039"** are rotatable relative to each other.

Operation of the locking knee collapsible rocker pedestal **1096"** is substantially similar to the operation of the locking knee tappet **440** described above and shown in FIGS. **4**A-**4**E. In particular, as shown in FIG. **10**C, the collapsible rocker pedestal **1096"** has a first extended configuration in which the femur **1079"** is positioned at a first angle **A1** relative to the support housing **1033"** that is relatively small (e.g., about 8 degrees). The first angle A**1** can be controlled by a femur support platform (not shown), as described above. When a lost-motion effect is required, the pivot height of the rocker **1004"** is dropped, thus allowing an engine valve coupled thereto to close earlier than what is called for by the profile of its corresponding cam. This is accomplished by opening a control valve **1001"**, which allows hydraulic fluid to drain from a cylinder **1003"** in which a hydraulic actuation piston **1041"** is reciprocally disposed. When the control valve **1001"** is opened, downward forces (e.g., in the direction of the arrow A**3**) exerted on the rocker **1004"** by the cam and/or the valve spring cause the collapsible rocker pedestal **1096"** to transition to a collapsed configuration, as shown in FIG. **10**D. In this configuration, the "knee" formed at the intersection of the femur **1079"** and the shin **1039"** buckles or articulates, driving the hydraulic actuation piston **1041"** into the cylinder **1003"** and expelling hydraulic fluid out of the cylinder **1003"** and into an accumulator **1005"**. The cylinder **1003"** includes one or more orifices **1007"** that are progressively occluded by the hydraulic actuation piston **1041"** to achieve a valve catch effect as described above. In the collapsed configuration, the femur **1079"** forms a second angle A**2** relative to the housing **1033"** that is greater than the first angle A**1**. In one embodiment, the angle A**2** can be about 23 degrees.

Once the eccentric portion of the cam has rotated past the rocker, the collapsible rocker pedestal **1096"** is transitioned back into the extended configuration. The control valve **1001"** is closed and the accumulator **1005"** forces hydraulic fluid into the cylinder **1003"** through a check valve **1009"** and through the orifices **1007"** as they become no longer occluded by the hydraulic actuation piston **1041"**. As the cylinder **1003"** refills, the hydraulic actuation piston **1041"** forces the femur **1079"** and the shin **1039"** to articulate or "straighten," thereby extending the collapsible rocker pedestal **1096"** and lifting the pivot point of the rocker **1004"**.

FIGS. **10**E-**10**F illustrate an alternative embodiment of a locking knee collapsible rocker **1096'"** having a "low profile" in which the support housing **1033'"** includes a hemi-cylindrical protrusion **1065'"** that is received within a first hemi-cylindrical slot **1063'"** formed in the femur **1079'"**. This embodiment allows for a shorter femur **1079'"** to be used and thus reduces the overall profile of the collapsible rocker pedestal **1096'"**. Operation of this embodiment is substantially similar to the operation of the embodiment shown in FIGS. **10**B-**10**D.

The locking knee collapsible rocker pedestals **1096"**, **1096'"** disclosed herein permit the rocker **1004"**, **1004'"** to be lowered by a distance that is less than the retraction distance of the hydraulic actuation piston **1041"**, **1041'"**. For example, in one embodiment the actuation piston has a maximum stroke of 2.4 mm and permits the rocker to be dropped by 1.5 mm. Because the movement distance of the actuation piston is larger than the change in rocker height, the force required to hold the collapsible rocker pedestal in the extended configuration and the force required to transition it into the extended configuration are reduced. As a result, a smaller hydraulic actuation piston/cylinder combination can be used, which allows the valve train to be stiffer, lighter, and faster. Another advantage to this design is that the leading edge **1011"**, **1011'"** of the hemi-cylindrical slot **1075"**, **1075'"** formed in the shin **1039"**, **1039'"** can be used as a lever to obtain a mechanical advantage. For example, the force required to hold the collapsible rocker pedestal in the extended configuration and the force required to transition it into the extended configuration is reduced by lengthening the leading edge **1011"**, **1011'"** and positioning the actuation piston **1041"**, **1041'"** at or close to the terminal end thereof. Again, the lower force requirement permits the use of a smaller, lighter, and faster hydraulic actuator. It will be appreciated that the hydraulic actuation piston **1041"**, **1041'"** in the embodiments of FIGS. **10**C-**10**E can be replaced with the locking knee tappet **440** disclosed above.

The engines and valve trains disclosed herein can also include other features for varying the valve opening and/or closing parameters. In a traditional engine, one or more camshafts rotate at a fixed rate with respect to the crankshaft (i.e., at a rate of two crankshaft rotations for each rotation of the camshaft). The cams in such engines are fixedly mounted to the camshaft, and thus the position of the cams' eccentric portions relative to the crankshaft and pistons (the cam's "phase") is fixed for any given crank angle. Accordingly, there is no way to vary the crank angle at which the opening ramp of the cam first contacts the rocker (or other intermediate valve train element).

Any of the cam-actuated valve trains disclosed herein can also include one or more cam phasers to overcome this limitation. The one or more cam phasers are controlled to adjust the relative position of the eccentric portion of the cam with respect to the crankshaft and pistons. In other words, the cam phaser is used to selectively adjust a phase of the cam relative to the crankshaft and the top dead center (TDC) positions of the compression **110**, **210** and expansion **120**, **220** pistons.

FIGS. **11**A-**11**C illustrate one exemplary embodiment of a cam phaser **1100** according to the present invention. As shown, the cam phaser **1100** generally includes a rotor **1174** disposed within a housing **1176**. The rotor **1174** is fixedly mated to a camshaft such that the camshaft and rotor **1174** rotate together. A drive sprocket **1178** is formed around or mated to the housing **1176** and is engaged by a timing belt, chain, gear, or the like in order to transfer rotational motion from the crankshaft to the housing **1176** and ultimately to the camshaft.

The rotor **1174** includes a plurality of vanes **1180** that extend radially-outward away from the rotational axis of the rotor **1174**. The housing **1176** includes a corresponding plurality of lobes **1182** that extend radially inward towards the rotational axis of the housing **1176**. When assembled, the rotor vanes **1180** are interspersed with corresponding housing lobes **1182**, thereby defining advance chambers **1184** and retard chambers **1186** on either side of each of the vanes **1180**.

As shown in FIG. **11**B, a substantially incompressible hydraulic fluid can be supplied to the advance chambers **1184** and/or removed from the retard chambers **1186** to rotate the rotor **1174** clockwise relative to the housing **1176**. Since the phase of the rotor **1174** relative to the camshaft is fixed, and since the phase of the housing **1176** relative to the crankshaft is fixed, clockwise rotation of the rotor **1174** relative to the housing **1176** causes the camshaft phase to be advanced relative to the crankshaft and to the TDC position of the expansion piston **120**, **220**.

Similarly, as shown in FIG. **11**C, hydraulic fluid can be supplied to the retard chambers **1186** and/or removed from the advance chambers **1184** to rotate the rotor **1174** counter-clockwise relative to the housing **1176**. This in turn causes the camshaft phase to be retarded relative to the crankshaft.

The cam phaser can also include a hydraulic control circuit including one or more oil control valves for selectively supplying the oil or other hydraulic fluid to the various chambers of the cam phaser. It will be appreciated that the cam phase can be adjusted continuously along a given range by controlling the degree to which fluid is added or removed to the various chambers. The cam phaser can also include one or more locking mechanisms (e.g., locking pins or springs) that can be engaged to prevent relative rotation of the rotor and the housing, thereby locking the phase of the camshaft relative to the crankshaft.

The orientation of the cam phaser can be opposite to that described above, such that the outer housing is fixedly mated to the camshaft and such that the inner rotor is driven by the crankshaft. A variety of other cam phaser technologies known in the art can also be employed without departing from the scope of the present invention.

In one embodiment, a split-cycle engine (such as the engines **100**, **200** discussed above) is provided with a first cam phaser coupled to a first camshaft having a cam disposed thereon for actuating a XovrC valve of the engine. The engine can also include a second cam phaser coupled to a second camshaft having a cam disposed thereon for actuating a XovrE valve of the engine. In this manner, the cam phase for the XovrC valve and XovrE valve can be independently controlled. In one embodiment, the cam phase for the XovrC valve and for the XovrE valve can each be independently advanced or retarded +/−30 degrees CA using the cam phasers. The engine can also include markings on an outer surface of the housing, the camshaft, and/or the crankshaft to facilitate closed-loop feedback control of the cam phasing. For example, one or more sensors can be provided such that a control unit coupled thereto can determine, based on the markings, the actual phase of the camshaft relative to the crankshaft (and pistons) and adjust the phase as desired.

FIGS. **12**A-**12**B illustrate the valve lift profile for an engine valve that is variably actuated using methods and devices disclosed herein.

FIG. **12**A illustrates three exemplary plots of valve lift as a function of position (position being expressed in terms of crank angle CA relative to TDC of a piston, such as an expansion piston) for a valve in one embodiment of an engine according to the present invention. The first plot **1200** illustrates the valve lift when a dwell cam is used and the cam phaser and variable valve actuation system are either not present or are not actuated. As shown, the valve opens with the opening ramp of the eccentric portion of the cam, remains open at a fixed lift throughout the dwell portion of the cam, and then closes with the closing ramp of the cam's eccentric portion.

The second plot **1202** illustrates the valve lift for the same valve when the cam phaser is actuated to advance the cam phase by approximately 30 degrees CA. As shown, the valve opens and closes approximately 30 degrees CA earlier than in the first plot **1200**.

The third plot **1204** illustrates the valve lift for the same valve when the cam phaser is actuated to retard the cam phase by approximately 30 degrees CA. As shown, the valve opens and closes approximately 30 degrees CA later than in the first plot **1200**, and approximately 60 degrees CA later than in the second plot **1202**.

FIG. **12**B illustrates two exemplary plots of valve lift as a function of position (position being expressed in terms of crank angle CA relative to TDC of a piston) for a valve in one embodiment of an engine according to the present invention. The first plot **1206** illustrates the valve lift when a dwell cam is used and the cam phaser and variable valve actuation system are either not present or are not actuated.

The second plot **1208** illustrates the valve lift for the same valve when the variable valve actuation system is actuated to close the valve early. As shown, the valve opens with the opening ramp of the eccentric portion of the cam and remains open at a fixed lift for a fraction of the dwell portion of the cam. When the variable valve actuation system is actuated, in this case at approximately 150 degrees CA, the valve begins to close. This could occur for example when the valve train **300** of FIGS. **3**A-**3**C is used and the bearing element **336** is withdrawn from between the cam **302** and the rocker **304**. As shown, the lift profile of the valve's closing event can be controlled in this situation with a valve seating control device as described herein to substantially mimic the closing ramp of the cam.

FIG. **13** illustrates another embodiment of a valve train according to the present invention for use with inwardly-opening engine valves (i.e., engine valves that open into or towards the cylinder). The valve train **1300** is substantially similar to the valve trains **300**, **800**, **900**A, **900**B, **900**C disclosed herein, and like the valve trains **300**, **800**, **900**A, **900**B, **900**C can include any combination of the features disclosed herein, except that the rocker is omitted from the valve train **1300** such that the bearing element **1336** is in direct contact with the valve **1306** or contacts the valve **1306** via one or more intermediate elements **1388**.

FIG. **14** illustrates one embodiment of a non-hybrid split-cycle engine according to the present invention. As shown, the engine **1400** has a compression cylinder **1401** and an expansion cylinder **1403** in which a compression piston **1405** and an expansion piston **1407** respectively reciprocate. The pistons **1405**, **1407** are coupled to a crankshaft **1409** rotatably journaled into an engine block **1411**. The compression cylinder **1401** and the expansion cylinder **1403** are joined by at least one crossover passage **1413** formed in a cylinder head **1415**. The inlet of the crossover passage **1413** can be selectively opened and closed via an outwardly-opening cam-actuated "XovrC" poppet valve **1417**. The outlet of the crossover passage **1413** can be selectively opened and closed via an outwardly-opening cam-actuated "XovrE" poppet valve **1419**. In addition, inwardly-opening intake and exhaust valves **1421**, **1423** are mounted in the cylinder head **1415**. For clarity of illustration, the valve train or valve trains associated with the intake valve **1421**, the XovrE valve **1419**, and the exhaust valve **1423** are not shown.

The outwardly-opening XovrC valve **1417** can be selectively actuated via a valve train **1425**. It will be appreciated that the outwardly-opening XovrE valve **1419** can also be selectively actuated with the same valve train **1425**. The valve train **1425** includes a dwell cam **1402** mounted on a camshaft coupled to a cam phaser **1427**. A sprocket **1478** of the cam phaser **1427** is driven by a timing belt, chain, and/or gear which is in turn driven by rotation of the crankshaft **1409**. As the cam phaser **1427** is driven, the camshaft and dwell cam **1402** mounted thereon rotate clockwise. When the dwell cam **1402** rotates, it imparts motion to a bearing element **1436** which in turn selectively imparts some or all of the cam's motion to a rocker **1404**, as described above.

The cam phaser **1427** can be selectively actuated to advance or retard the phase of the cam **1402** relative to the crankshaft **1409**. In addition, a hydraulic tappet **1440** coupled to the bearing element **1436** can be selectively actuated to advance the bearing element **1436** towards the cam **1402** and the rocker **1404** or to retract the bearing element **1436** away from the cam **1402** and the rocker **1404**. A valve seating control device **1472** is mounted to the cylinder head **1415** and is configured to selectively "catch" (i.e., decrease the velocity of) the XovrC valve **1417** as it closes.

During operation, the opening timing, opening rate, lift, closing timing, closing rate, and various other valve parameters can be controlled. As the crankshaft **1409** rotates, the cam phaser **1427**, the camshaft (not shown), and the dwell cam **1402** also rotate as a result of their linkage to the crankshaft via a timing belt, chain, gear, or similar mechanism (not shown). As the dwell cam **1402** rotates, an opening ramp portion thereof contacts the bearing element **1436** which in turn contacts the rocker **1404** and imparts motion thereto (e.g., by causing the rocker **1404** to pivot or rotate counter-clockwise. This in turn causes the valve-engaging end of the rocker **1404** to impart motion to the valve **1417** (e.g., by lifting the valve **1417** upwards from its valve seat) and thereby opens the valve **1417**. The timing of this valve opening event can optionally be advanced or retarded by actuating the cam phaser **1427**, as described above.

When a dwell portion of the cam **1402** contacts the bearing element **1436**, the rocker **1404** can be held in a substantially fixed angular orientation which in turn holds the valve **1417** open at a substantially fixed linear distance from the valve seat. If early valve closing control is called for (e.g., by an engine control unit), pressurized hydraulic fluid maintained in the tappet **1440** is rapidly drained, reducing the length of the tappet **1440** and withdrawing the bearing element **1436** from the cam **1402** and the rocker **1404**. As a result, the rocker **1404** pivots or rotates clockwise under the bias of a valve spring (not shown) until the valve **1417** closes against the seat. In this case, the valve seating control device **1472** operates to slow the rotation of the rocker **1404** and thus the velocity of the valve **1417** as the valve **1417** approaches the seat.

Alternatively, when early valve closing control is not called for, the valve **1417** remains open a fixed linear distance until the dwell section of the cam **1402** passes the bearing element **1436** and the bearing element **1436** contacts the closing ramp of the cam **1402**. At that time, the bias of the valve spring forcing the valve **1417** closed causes the rocker **1404** to pivot or rotate clockwise until the valve **1417** is closed. The valve closing velocity can be controlled by the closing ramp profile of the cam and/or by a valve seating control device **1472**, as explained above.

In the illustrated exemplary embodiment, the opening timing of the crossover valves **1417**, **1419** can be independently varied at least plus or minus 30 degrees CA, thus providing at least about a 60 degree CA range over which the opening timing for either valve can be varied. In the same embodiment, the closing timing for either valve can be varied over a range of at least about 100 degrees CA. It will be appreciated that the lost-motion portion of the valve actuation systems disclosed herein permits the closing timing of the valves to be dynamically actuated from one cycle of the engine to the next. In addition, the cam phaser portion of said systems permits the opening timing of the valves to be altered gradually over several cycles of the engine.

The engine **1400** can be readily adapted or modified to include an air hybrid system, as explained above with respect to the engines **100**, **200**. For example, the crossover passage **1413** can be connected to an air reservoir via a tank port having a tank valve disposed therein. It will thus be appreciated that the illustrated valve train can also advantageously provide the dynamic valve actuation characteristics required for efficient operation of an air hybrid engine.

The engine **1400** is configured to operate reliably over a broad range of engine speeds. In certain embodiments, engines according to the present invention can be 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.

FIG. **15** illustrates another embodiment of a valve train according to the present invention. The illustrated valve train **1500** is substantially identical to the valve train **300** described above with respect to FIGS. **3A-3C**, except that a bell crank **1529** is positioned between the tappet **1540** and the connecting arm **1538** to provide a mechanical advantage (i.e., leverage) when adjusting a position of the bearing element **1536**. The bell crank **1529** is rotatably mounted at a pivot point **1531**, which has a fixed position. The tappet **1540** is attached to the bell crank **1529** at an end opposite the pivot point. A ball and socket joint is provided at an intermediate point on the bell crank **1529** to couple the bell crank **1529** to the connecting arm **1538**. Adjusting the position along the bell crank **1529** at which the connecting arm **1538** is attached adjusts the lever ratio of the bell crank **1529**, thereby increasing or decreasing the amount of force required to be supplied by the tappet **1540** and/or the amount of distance that the tappet **1540** must move to effect the desired movement of the bearing element **1536**. This embodiment can be particularly useful in systems where a relatively large amount of force is required to advance and/or withdraw the bearing element **1536**, since it allows a relatively weak tappet **1540** to perform satisfactorily.

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. 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:

1. An engine comprising:

a camshaft having a cam formed thereon;

a rocker coupled to an engine valve; and

a lost-motion system comprising:

a bearing element having a variable thickness between opposed first and second bearing surfaces, the bearing element being positioned between the cam and the rocker such that the first bearing surface slidably engages the cam and the second bearing surface slidably engages the rocker;

an actuator configured to vary the degree to which the bearing element is inserted between the cam and the rocker to vary the degree to which motion of the cam is imparted to the rocker; and

a connecting arm that is rotatably coupled to the actuator and non-rotatably coupled to the bearing element;

wherein the actuator is collapsible along a length thereof to allow the bearing element to be pushed away from the cam and the rocker.

2. The engine of claim **1**, wherein the engine is a split-cycle engine.

3. The engine of claim **2**, further comprising a crossover passage wherein the engine valve is a crossover valve formed in said crossover passage.

4. The engine of claim **1**, wherein the engine is an air hybrid engine.

5. The engine of claim **1**, wherein the engine valve is an outwardly-opening valve.

6. The engine of claim **1**, further comprising a cam phaser that selectively adjusts a phase of the cam relative to a crankshaft to advance or retard an opening timing of the engine valve.

7. The engine of claim **6**, wherein the cam phaser comprises:

a rotor fixedly coupled to the camshaft and having a plurality of vanes extending radially away from a rotational axis of the rotor;

a housing in which the rotor is disposed, the housing including a plurality of lobes extending radially towards a rotational axis of the housing; and

a fluid control valve configured to supply hydraulic fluid to two or more fluid chambers formed between the plurality of vanes and the plurality of lobes, wherein supplying hydraulic fluid to a first chamber of the two or more fluid chambers is effective to rotate the rotor in a first direction relative to the housing and supplying hydraulic fluid to a second chamber of the two or more fluid chambers is effective to rotate the rotor in a second direction opposite to said first direction.

8. The engine of claim **6**, wherein the lost-motion system is operable to close the engine valve earlier than the engine valve would have closed if closed by a profile of the cam.

9. The engine of claim **1**, wherein the bearing element has an elliptical-shaped cross-section.

10. The engine of claim **1**, wherein the first and second bearing surfaces are convex bearing surfaces.

11. The engine of claim **1**, wherein the bearing element has a wedge-shaped cross-section.

12. The engine of claim **1**, wherein the bearing element has a circular cross-section.

13. The engine of claim **1**, wherein the actuator comprises an adjustable hydraulic tappet configured to selectively impart bi-directional linear motion to the bearing element.

14. The engine of claim **1**, wherein at least one of the first and second bearing surfaces includes a roller rotatably mounted to the bearing element.

15. The engine of claim **1**, further comprising a valve seating control device configured to reduce a velocity of the engine valve when the valve approaches a corresponding valve seat after the lost-motion system is actuated.

16. The engine of claim **1**, wherein the cam is a dwell cam.

17. The engine of claim **1**, wherein the actuator comprises a hydraulic tappet configured to selectively allow the bearing element to be pushed away from the cam and the rocker to close the engine valve earlier than the engine valve would have closed if closed by a profile of the cam.

18. The engine of claim **1**, wherein the actuator is collapsible along a length thereof to allow the bearing element to be pushed away from the cam and the rocker by combined forces of the rocker and the cam acting on the bearing element.

19. The engine of claim **1**, wherein the actuator is configured either to hold the bearing element in a position where maximum lift is imparted to the valve or to allow the bearing element to be pushed away from the cam and the rocker until no lift is imparted to the valve.

20. A method of actuating an engine valve, comprising:

rotating a cam having an eccentric portion such that the eccentric portion engages a first bearing surface of a bearing element, thereby causing a second bearing surface of the bearing element that is opposed from the first bearing surface to engage a rocker coupled to the engine valve, the bearing element being disposed between the cam and the rocker and having a variable thickness along a length thereof between the opposed first and second bearing surfaces such that the degree to which motion is imparted to the engine valve can be controlled by an actuator configured to vary the degree to which the bearing element is inserted between the cam and the rocker; and

adjusting a closing timing at which the engine valve begins to close such that the engine valve closes earlier than what is called for by the cam by collapsing the actuator along a length thereof to allow the bearing element to be pushed away from the cam and the rocker, the actuator being rotatably coupled to a connecting arm which is non-rotatably coupled to the bearing element.

21. The method of claim **20**, wherein the engine valve is a component of a split-cycle air hybrid engine.

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