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**Fitzgerald**

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(54) **ELECTRO-HYDRAULIC VALVE ACTUATOR WITH INTEGRAL ELECTRIC MOTOR DRIVEN ROTARY CONTROL VALVE**


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(65) **Prior Publication Data**

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**Related U.S. Application Data**

(60) Provisional application No. 60/788,783, filed on Apr. 3, 2006.

(51) **Int. Cl.**
***F01L 9/02*** (2006.01)

(52) **U.S. Cl.** ............ **123/90.12**; 123/90.11; 137/625.21; 251/129.11

(58) **Field of Classification Search** .............. 123/90.12, 123/90.13, 90.11; 137/625, 625.15, 625.65, 137/625.21; 251/65, 129.01, 129.11
See application file for complete search history.


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*Primary Examiner*—Ching Chang

(57) **ABSTRACT**


An improved electro-hydraulic intake and exhaust valve actuator for a "camless" internal combustion reciprocating engine. The present invention integrates an electric motor driven "plug type" rotary control valve and a single acting hydraulic cylinder in one housing for the actuation of an engine valve. The geometry of the hydraulic ports in the rotary control valve may be tailored for desired valve actuation profiles. The electronic control of the rate of rotation and angular position of the rotary control valve are used to infinitely vary the engine valve operating parameters. Thus, engine valve timing, speed, cycle duration and lift may be varied. A rotary control valve permits high speed operation and accommodates a broad range of valve sizes. Availability of a wide range of commercial open frame brushless electric motors and dedicated integrated circuit controllers contribute to the cost effectiveness of the present design.


**4 Claims, 10 Drawing Sheets**

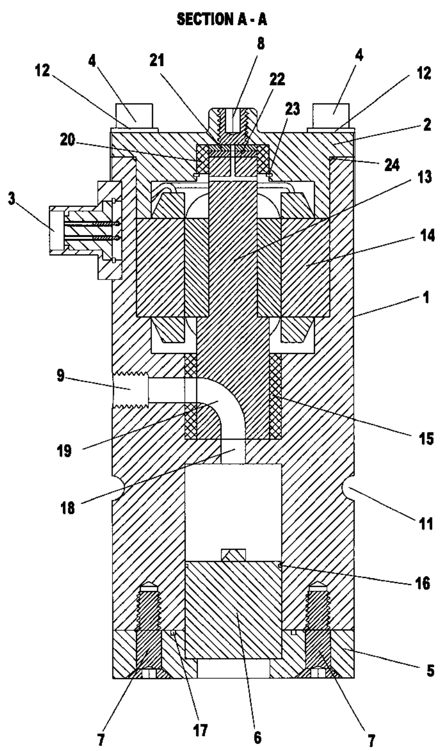


FIG. 1

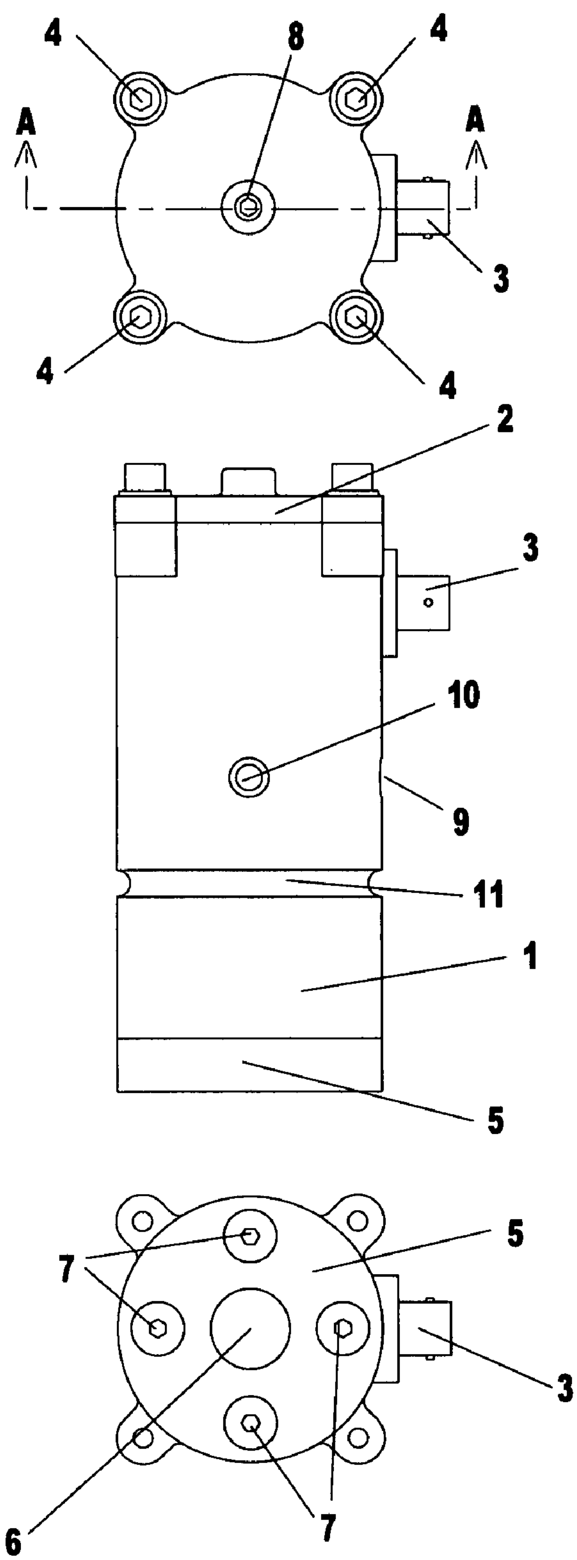


FIG. 2

SECTION A - A

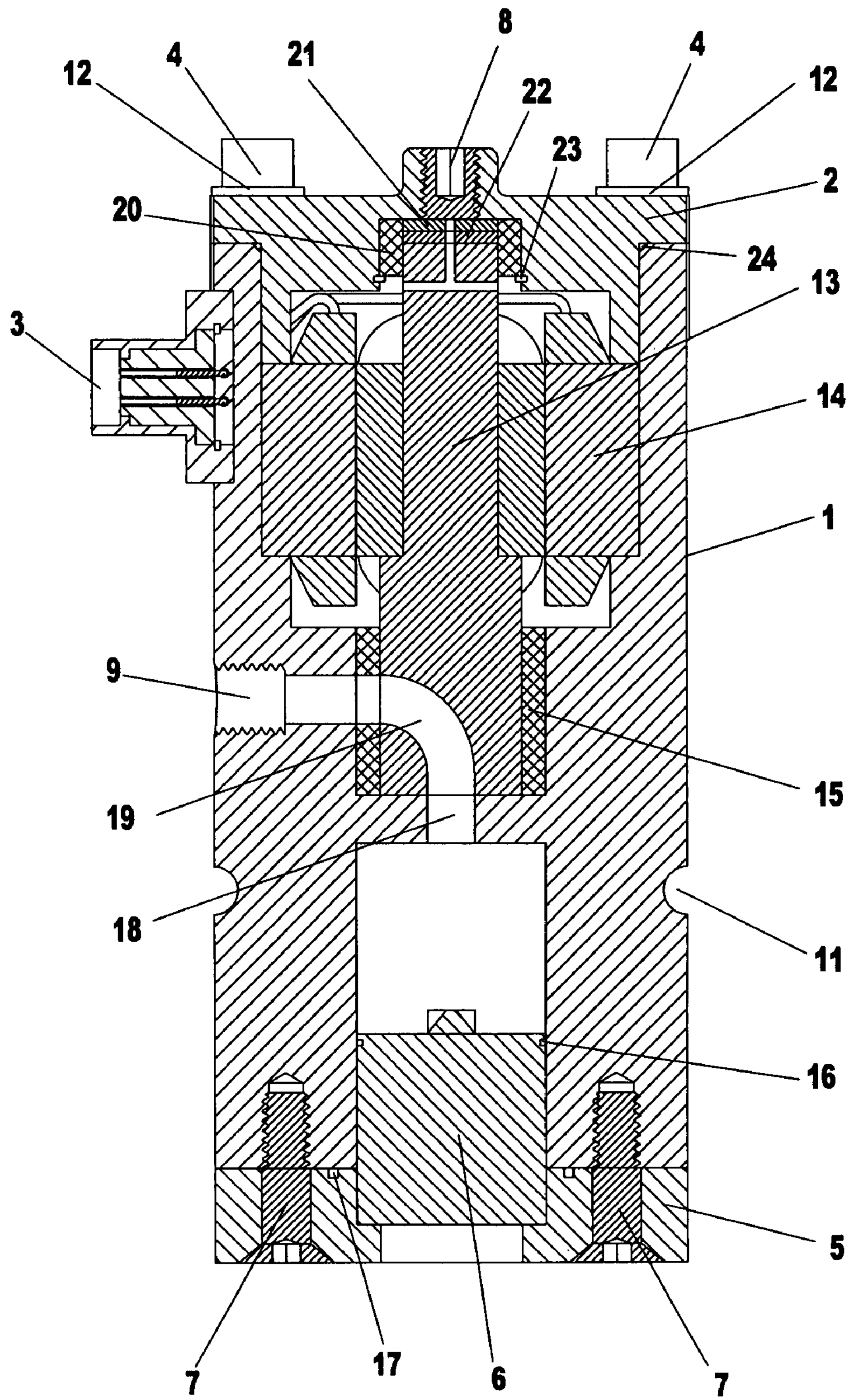


FIG. 3

SECTION A - A

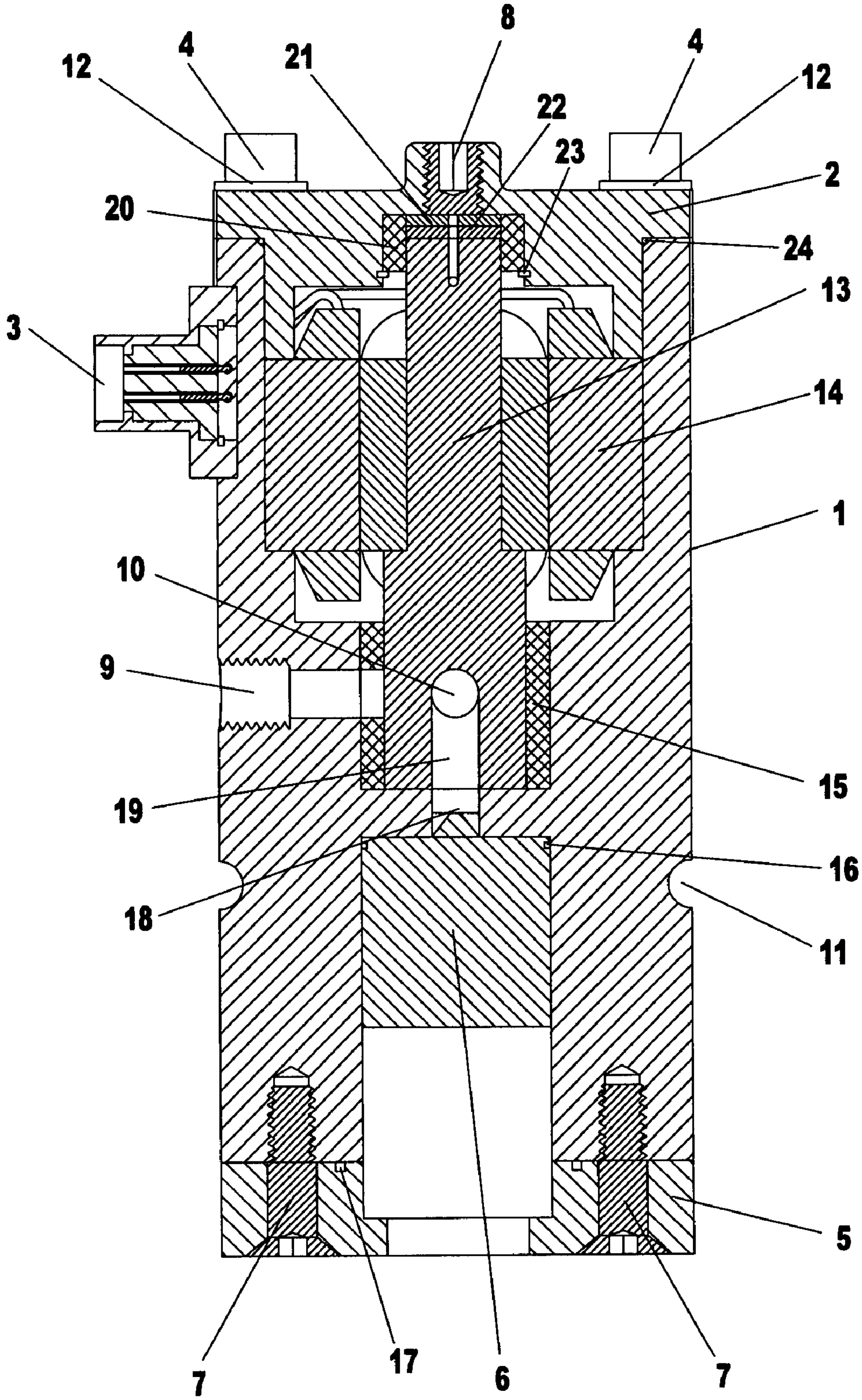
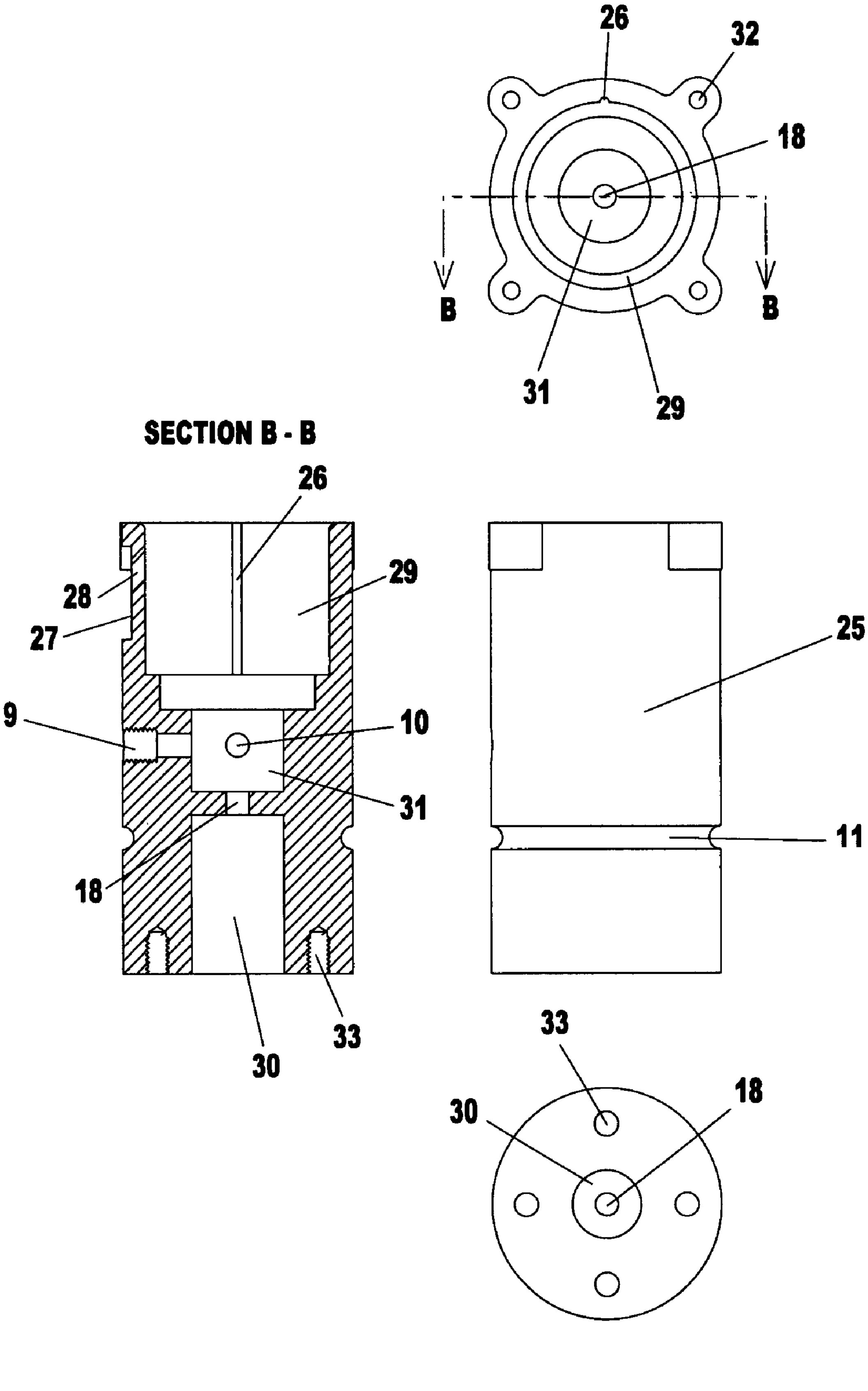


FIG. 4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18
B
B
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SECTION B - B
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FIG. 5

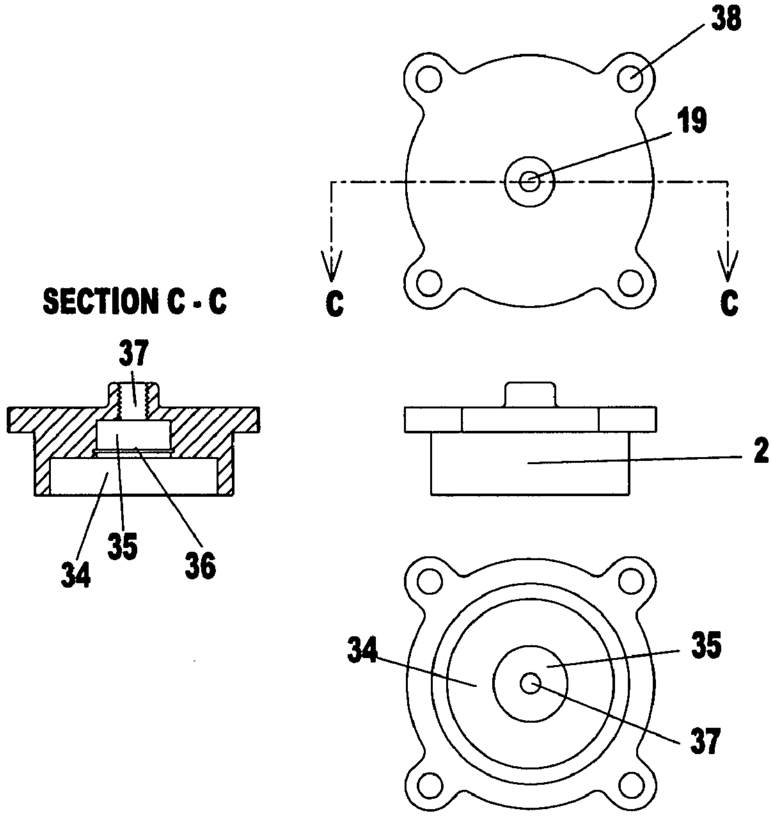



FIG. 6

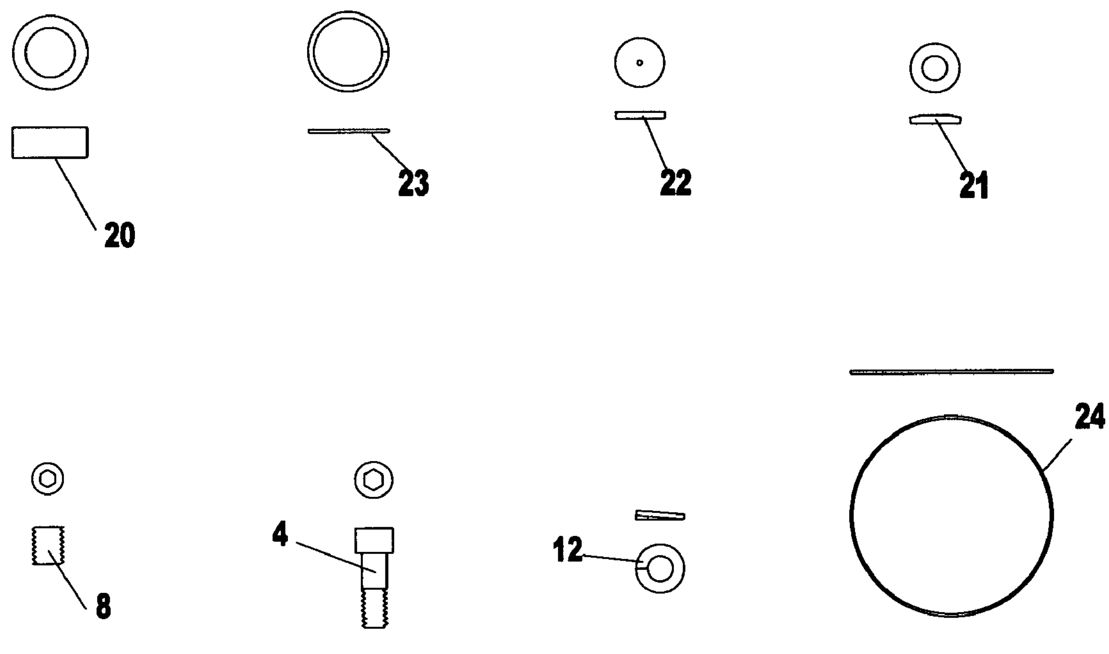


FIG. 7

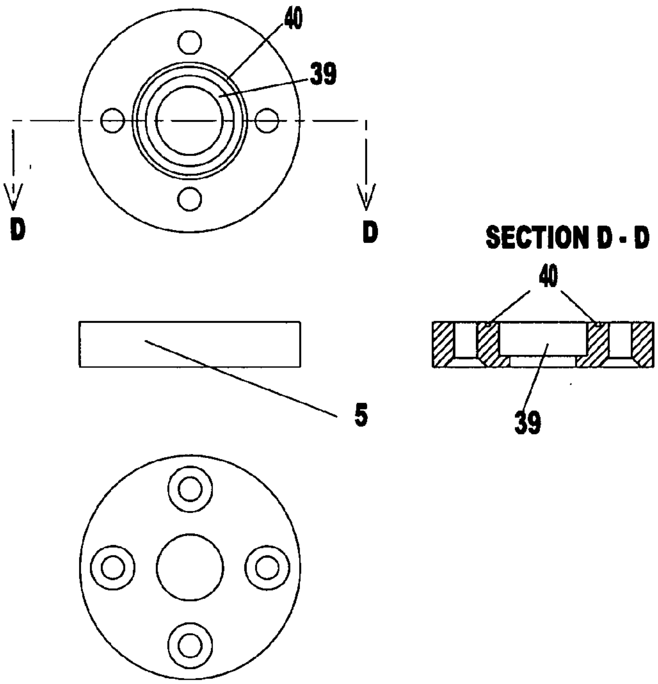



FIG. 10

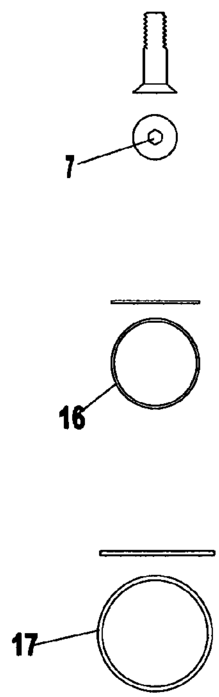



FIG. 9

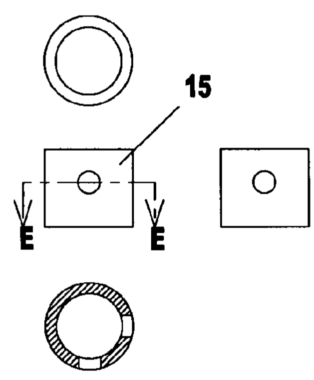



FIG. 8

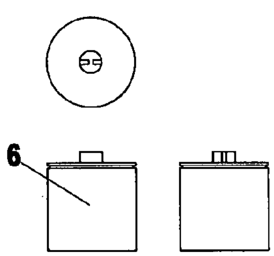


FIG. 11

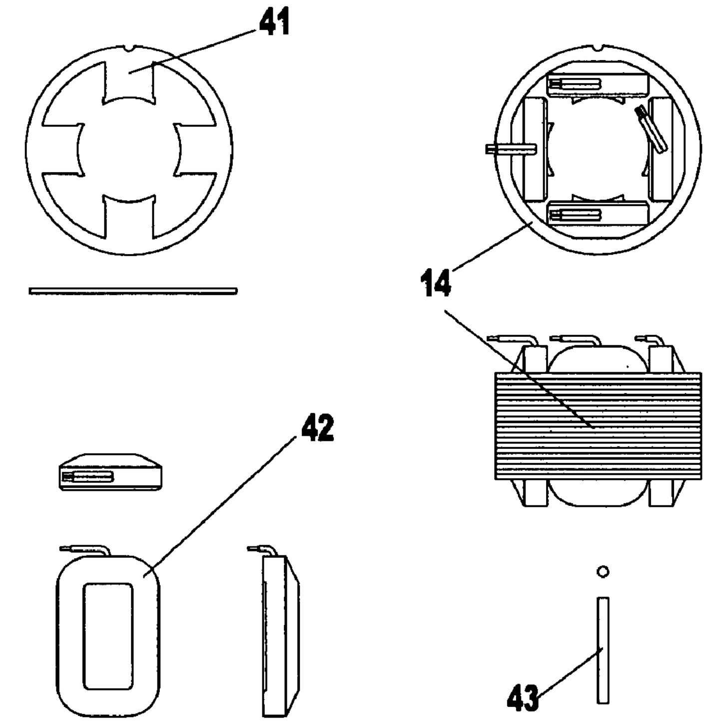



FIG. 12

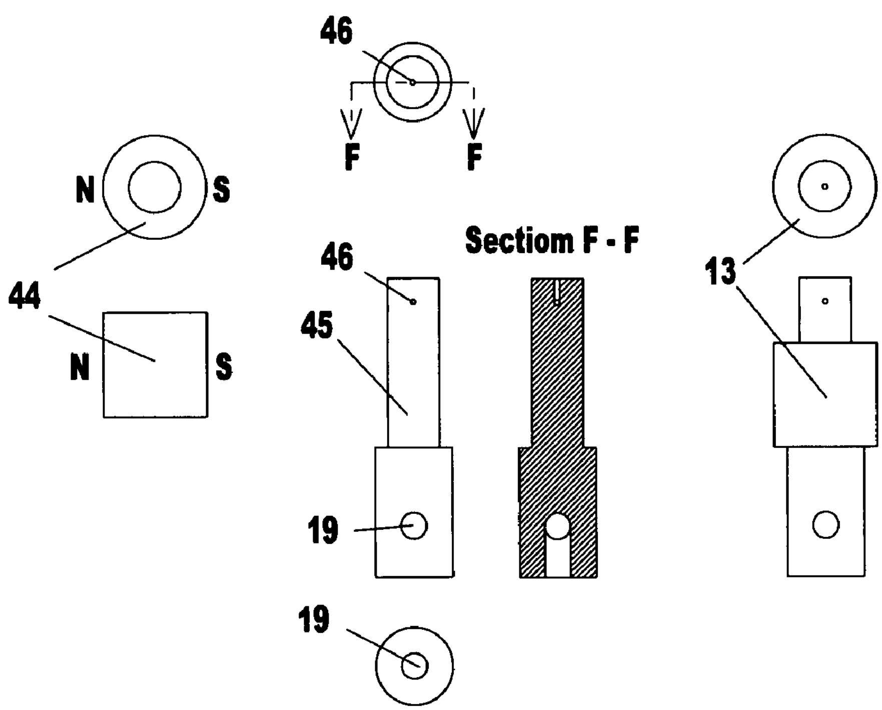


FIG. 13

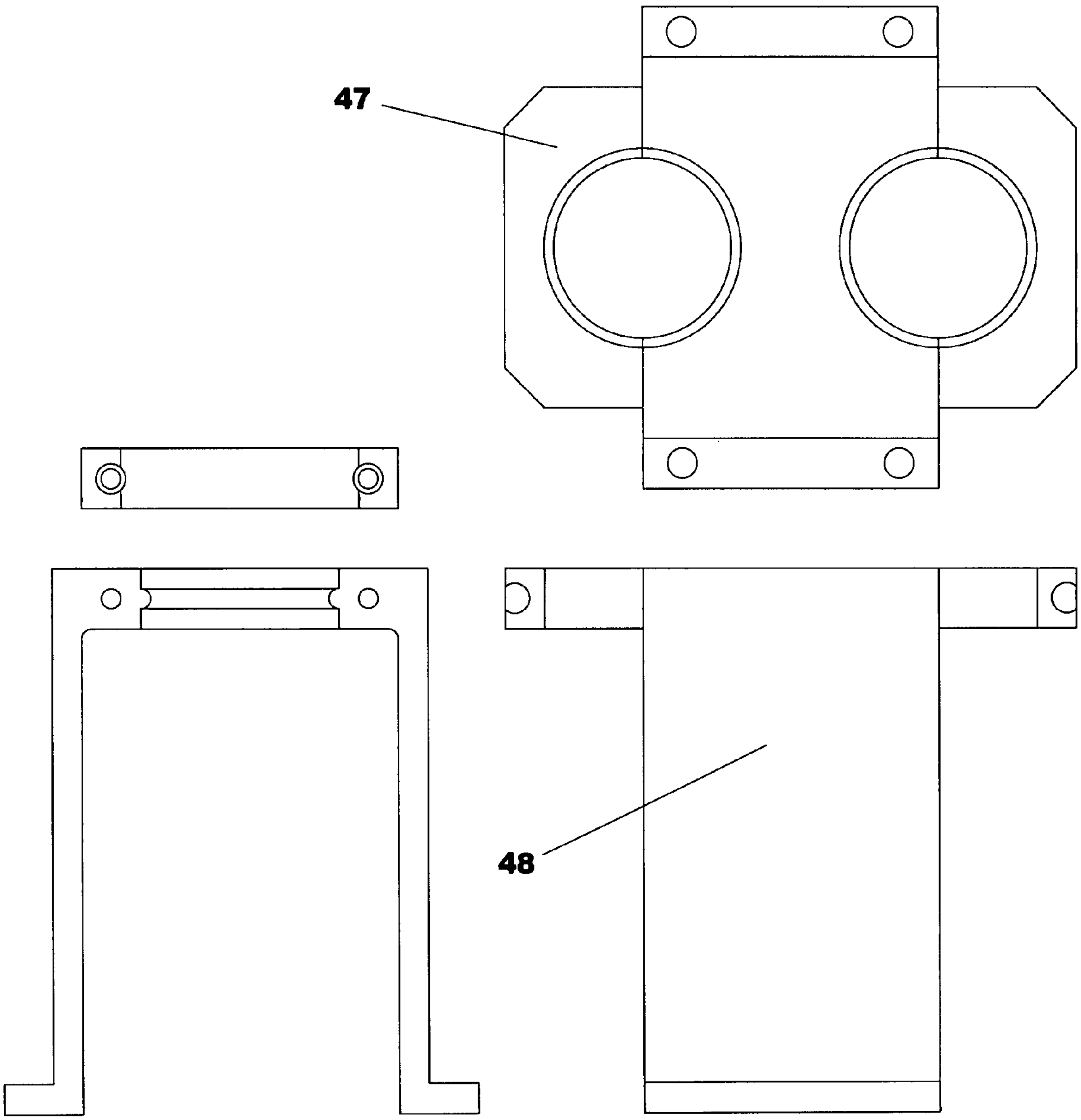


FIG. 14

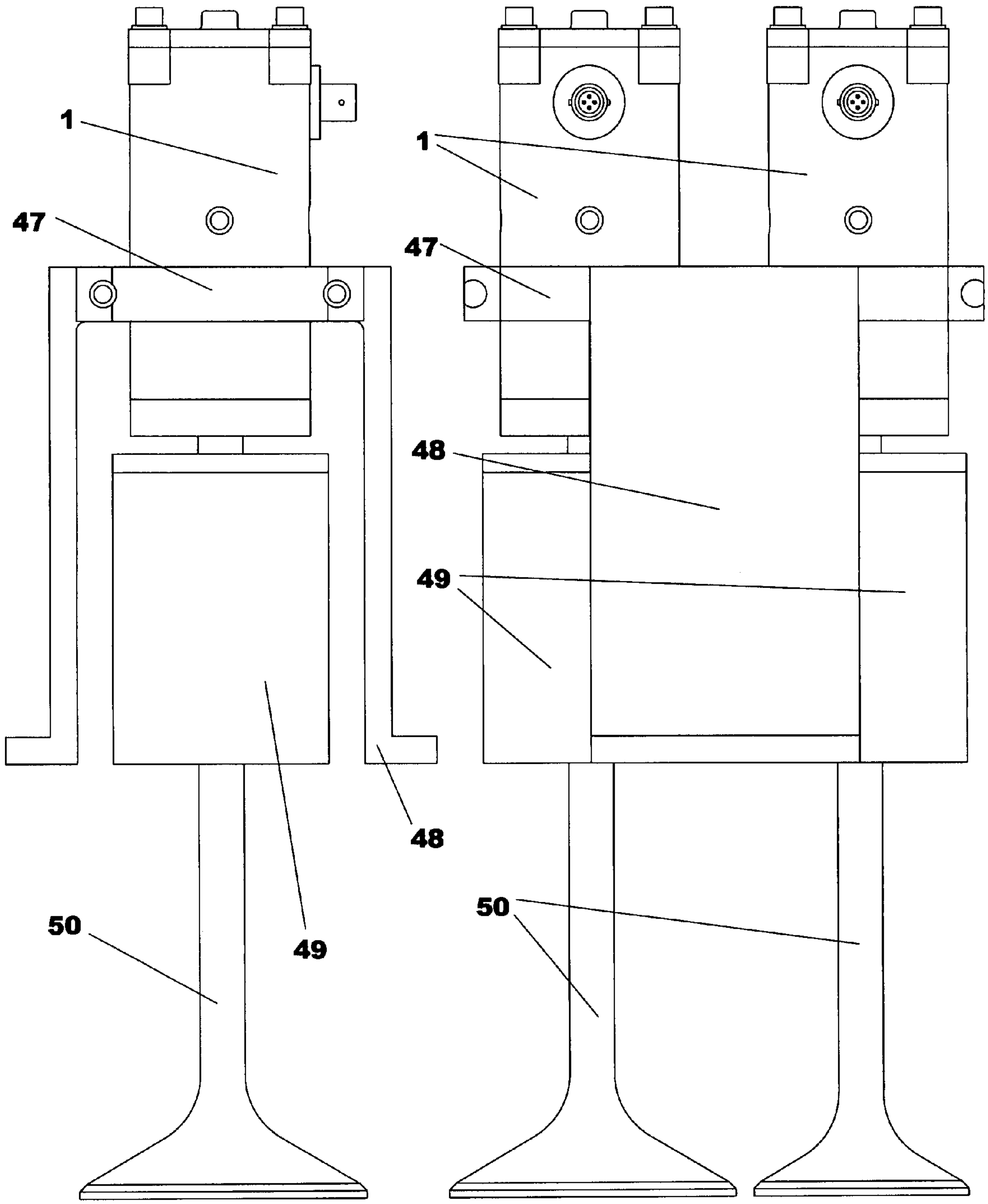
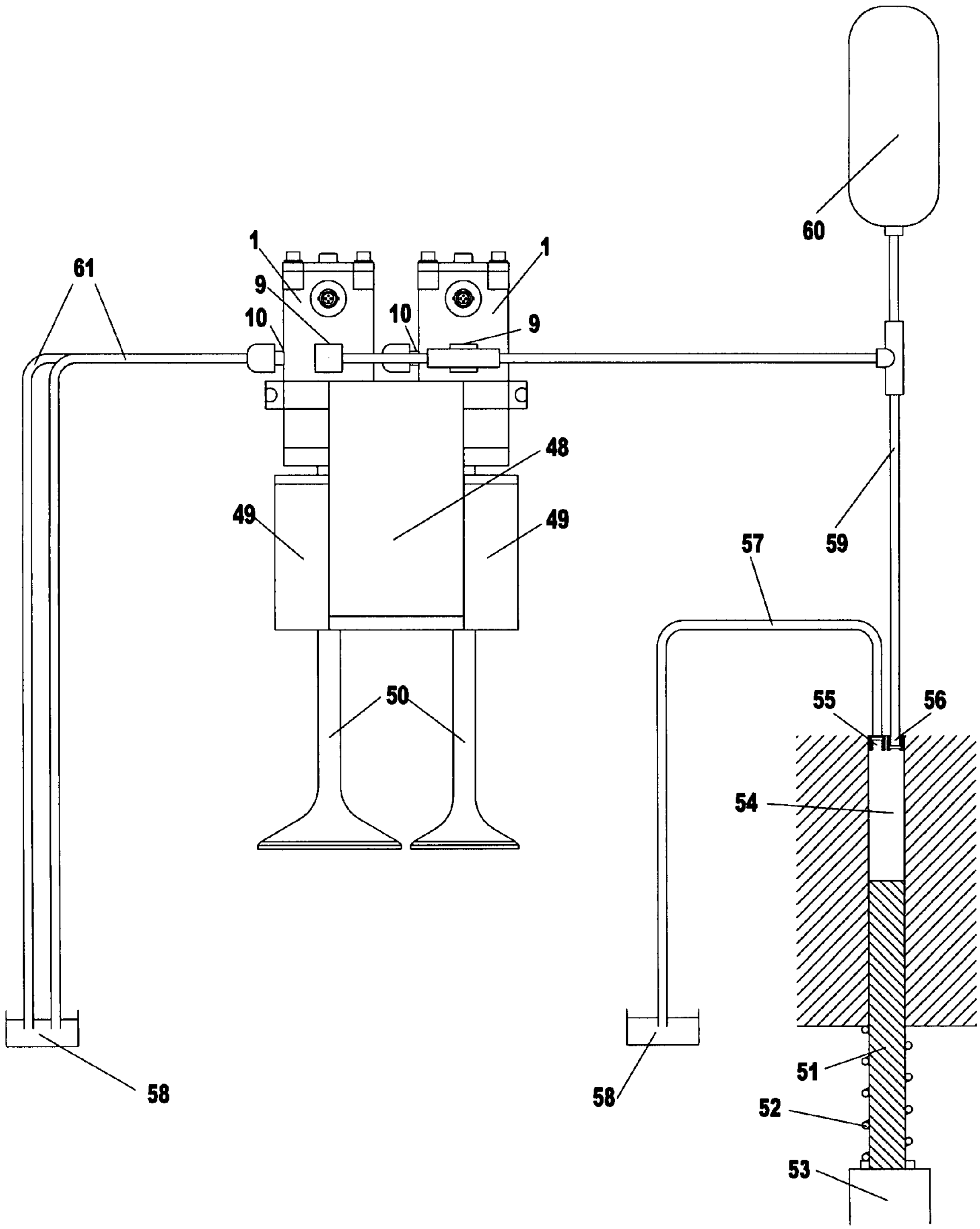


FIG. 15



# ELECTRO-HYDRAULIC VALVE ACTUATOR WITH INTEGRAL ELECTRIC MOTOR DRIVEN ROTARY CONTROL VALVE

## CROSS-REFERENCE TO RELATED APPLICATIONS

The benefit of Provisional Application 60/788,783, filed Apr. 3, 2006 by the same named inventor, entitled "Electro-hydraulic valve actuator with integral electric motor driven rotary control valve" and of substantially the same subject matter, is hereby requested.

## FEDERALLY SPONSORED RESEARCH OR DEVELOPMENT

Not Applicable

## COMPACT DISC APPENDIX

Not Applicable

## BACKGROUND OF THE INVENTION

Internal combustion reciprocating engine (ICRE) design has been in transformation for some time due to the demands for increased engine efficiency and lower emissions. Non-conventional fuel blends, and ultimately alternative fuels, are anticipated to come into increasing use. In response, engine designers have been re-examining engine attributes, including the actuation of the gas exchange valve (GEV), i.e. the intake and exhaust valve. In its present forms the ubiquitous poppet valve, with cam shaft actuation and coiled metal spring valve closure, are generally seen as inadequate for future engine requirements. Over the last several years there has been considerable effort expended on valve actuation (VA) as well as variable valve actuation (VVA) and a great number of patents have been issued in this area. Of these, the electro-hydraulic valve actuator (EHVA) is the focus the present invention. This class includes both the basic function of valve actuation (valve opening and valve closure) and variable valve actuation (varied valve timing, open/close duration and amount of valve lift).

Notable among the EHVA designs are the valve actuators disclosed by Sturman or its assignees—see: U.S. Pat. Nos. 7,025,326, 6,557,506, 6,360,728, 6,308,690, 6,148,778, 5,829,396, 5,713,316, 5,640,987, and 5,638,781. The foregoing patents are based primarily on the original Sturman design of a latching solenoid, disclosed in U.S. Pat. Nos. 3,743,898 and 3,683,239 (first applied to Diesel fuel injectors). This latching solenoid device is employed in the Sturman EHVA to move a linear hydraulic spool valve, which then provides hydraulic pressure and flow to an actuating hydraulic cylinder. In this design, as disclosed in U.S. Pat. No. 5,638,781, the valve operation is either open or closed. Quoting from its abstract: "—Energizing one solenoid moves the spool and valve into an open position. The valve spool is maintained in the open position by the residual magnetism of the valve housing and spool even when power is no longer provided to the solenoid. Energizing the other solenoid moves the spool and valve to a closed position. The solenoids are digitally latched by short pulses provide by a microcontroller. The valve is therefore opened by providing a digital pulse of a short duration to one of the solenoids and closed by a digital pulse that is provided to the other solenoid.—". That is, the valve is either fully open or fully closed. Sturman discloses, in U.S. Pat. No. 5,638,781, an EHVA with integrated double acting hydraulic cylinder (which eliminates the need for a GEV return spring) and digital solenoid spool valve. To add an additional degree of valve control, Sturman further discloses, in U.S. Pat. No. 7,025,326, a design and method which adds a proportional hydraulic control valve function, with the objective of reducing the power consumption of the valve actuation system. However, this addition has a higher degree of complexity and an associated cost increase compared to the "digital" version. Sturman valve actuators have demonstrated satisfactory on-engine performance and the introduction of a Sturman EHVA into a production truck engine is imminent. Nonetheless, the latching solenoid principle appears to be limited to relatively modest sized EHVA—due the required properties of the magnetic circuit.

Schechter discloses in U.S. Pat. No. 5,456,222 (assigned to Ford Motor company) a reversing electric motor with a threaded shaft coupled to a threaded hydraulic valve spool—to convert the motor rotary motion to linear motion for the reciprocation of the spool valve. The hydraulic spool valve produces reversible hydraulic fluid flow to an integral double acting actuating cylinder (no valve spring) for a GEV. The requirement for reversing the motor is a disadvantage as it degrades valve response compared to a motor with continuous rotation.

Eaton discloses in U.S. Pat. No. 5,682,846 an EHVA with solenoid spool valve and an integral double acting hydraulic cylinder actuator with dual pistons of two different diameters, providing greater actuation force onto the GEV—than similar prior devices.

Buehrle discloses in U.S. Pat. No. 6,024,060 a unique rotationally oscillating electric motor directly driving a hydraulic control valve supplying hydraulic fluid to a separate single acting hydraulic cylinder actuating the GEV.

Cummins discloses in U.S. Pat. No. 6,067,946 a device utilizing one or more hydraulic pressure sources applied through solenoid valves to a separate single acting hydraulic cylinder actuator for a GEV with varying return spring configurations.

Each of these inventors devices, Sturman, Schechter (Ford), Eaton, Cummins, and Buehrle, have limitations such as speed, operating range, capacity, cost, power consumption, etc.—which other designers are endeavoring to overcome. For example, see "Development of a Piezoelectric Controlled Hydraulic Actuator for a Camless Engine" Thesis of J. S. Brader, University of South Carolina, 2001—that demonstrated a successful proof of concept piezoelectric stack, hydraulic spool valve and actuator device. Also see: "Dynamic simulation of an electro-hydraulic open center gas-exchange valve actuator system for camless internal combustion engines." Thesis, J. M. Donaldson, P. E., Milwaukee School of Engineering, 2003—in which modeling of an open-center hydraulic series valve system demonstrated the feasibility of the concept.

The present invention is an electro-hydraulic valve actuator (EHVA) intended to provide a more optimal balance of the wide range of design aspects required of EHVA, including: capacity, speed, lift, profile, cost, etc.—thereby satisfying the requirements of a broader range of ICRE and providing an improvement over the existing EHVA art. It utilizes a rotary "plug" valve which has the potential for very high speed, (>10,000 rpm or 20,000 rpm engine speed) thus allowing the present invention to meet the speed requirement of any known ICRE. As a single acting actuator, the present invention's speed is however, ultimately limited by the valve spring. The present invention is scalable over the entire range of ICRE sizes from micro engines to the largest Diesel contemplated. In addition, the present invention may be implemented with a varying range of components to meet cost objectives—for example a switched reluctance motor versus a permanent magnet motor. The recent commercial availability of a wide range of brushless electric motors and dedicated integrated driver circuits has made the present invention viable. Nonetheless, it is unlikely there will be just one solution to improved ICRE valve actuation as the range of engine requirements is highly diverse.

## SUMMARY OF THE INVENTION

The general objective is to provide variable valve actuation for the gas exchange valves of a "camless" ICRE. Electro-hydraulic valve actuators have shown to be able to provide far greater actuating force than competing valve actuation technology. Given the trends in ICRE operation, the GEV is expected to operate with greater pressures and at faster rates than in previous engines—which requires higher actuating force—thus the selection of an EHVA for the basis of the present invention. Furthermore, economics favor the use of a single acting hydraulic cylinder type actuator, as fewer actuator components are required versus a double acting cylinder. (Although valve springs are needed with a single acting cylinder they are a mature and cost effective component.) Integrating the actuating cylinder with the control valve has also shown to be cost effective and provides for the most compact geometry. Both features have been adopted for the present invention.

The present invention is an EHVA with a rotary valve and integral single acting linear hydraulic cylinder. Hydraulic pressure and flow to the hydraulic cylinder is controlled by an electric motor driver rotary "plug" valve (which may be incorporated into the motor shaft). The rotary "plug" valve is ported in such a manner that, to open the GEV, the high pressure hydraulic fluid—from an external pump—is directed from the EHVA inlet port to the hydraulic cylinder causing it to move linearly, which compresses the valve spring and opens the GEV. As the rotary valve is turned further, by the electric motor, the inlet port and valve port are no longer aligned and the pressure is retained in the hydraulic cylinder—thereby holding the GEV open. Additional rotation of the rotary valve aligns its port with the EHVA outlet port and pressure is relieved from the hydraulic cylinder and the valve spring forces the hydraulic cylinder piston to return to the original position closing the GEV and discharging the hydraulic fluid in the cylinder to the external pump return. The cycle repeats as long as the rotary valve is turned by the electric motor. The EHVA motor speed and angular position are controlled in such a manner as to match the ICRE speed and attain the desired valve timing, duration and lift. The present design is an improvement on existing designs in that it is scalable over a wide size range and capable of actuating the GEV at speeds greater than existing devices and is producible at a competitive cost.

## BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** shows the electro-hydraulic valve actuator assembly **1** in front, top, bottom, and right side views. It consists of valve top cap **2**, valve body **25**, electrical connector **3**, valve bottom cap **5**, and piston **6**. Hydraulic fluid intake port **9** and hydraulic fluid outlet port **10** provide the means of supplying hydraulic pressure and flow into and out of, respectively, the electro-hydraulic valve actuator assembly **1**.

FIG. **2** is cross section A-A of FIG. **1** showing stator assembly **14** and rotor assembly **13**—which is rotated by the revolving magnetic field generated by stator assembly **14**. Rotor assembly **13** is shown oriented in the position open to the hydraulic fluid inlet port **9** providing hydraulic pressure and flow first through the hydraulic passage-rotary valve **19** then through the hydraulic passage-internal **18** to piston **6**. Piston **6** is forced by hydraulic pressure and flow to move away from the hydraulic passage **18** and toward valve bottom cap **5**—thus providing a linear actuating force along the axis of travel. Concurrently, the rotor assembly **13** blocks hydraulic pressure and flow to the hydraulic fluid outlet port **10**.

FIG. **3** is cross section A-A of FIG. **1** with rotor assembly **13** shown rotated toward the hydraulic fluid outlet port, **10**,—which relieves hydraulic pressure on the piston, **6**, allowing it to move away from the valve bottom cap, **5**, and return back toward the hydraulic passage—internal, **18**. The return travel force is provided by the external GEV spring (not shown) in contact with the external face of the piston **6**. Concurrently, rotor assembly **13** blocks hydraulic pressure and flow from the hydraulic fluid inlet port **19**.

FIG. **4** shows valve body **25** which is the main housing of the electro-hydraulic valve actuator assembly **1**.

FIG. **5** shows valve top cap **2** in top, side, bottom and cross section views.

FIG. **6** illustrates hardware items associated with the valve top cap **2**: bushing—motor shaft **20**, retaining ring **23**, thrust washer **22**, Belleville washer **21**, plug **8**, socket head cap screw **4**, lock washer **12** and seal ring **24**.

FIG. **7** shows front, top, bottom and cross section views of valve bottom cap **5**. Piston bore **39** and seal ring groove **40** are illustrated in the top and cross section views.

FIG. **8** shows three views of the piston **6** illustrating a seal ring groove and integral snubber.

FIG. **9** shows front, top, bottom and cross section views of the bushing—rotary valve, **15** illustrating the position of hydraulic fluid passages.

FIG. **10** shows hardware associated with valve bottom cap **5** and piston **6**: flat head cap screw **7**, piston seal **16**, and bottom cap seal ring **17**.

FIG. **11** shows front and top views of stator **14**. It can be seen that stator **14** consists of magnetic metal laminations **41** and insulated wire coils **42**.

FIG. **12** shows a front and top view of rotor assembly **13**. Motor shaft and valve rotor **45** is shown in front, top, bottom and cross section views. Permanent magnet **44** is shown in front and top views.

FIG. **13** shows mounting bracket clamp **47** and mounting bracket **48**.

FIG. **14** Mounting bracket **48** and mounting bracket clamp **47** are shown in a front and side view with two of the electro-hydraulic valve actuator assembly **1** installed. Also shown are two of valve spring-coiled **49** and valve-poppet **50** in the relative position of a typical engine cylinder installation.

FIG. **15** portrays a typical hydraulic fluid pressure and return circuit for a single cylinder of an internal combustion reciprocating engine with the electro-hydraulic valve actuator assembly **1** providing the opening and closing of the intake and exhaust valves **50**. Hydraulic pump piston **51** provides hydraulic power to a pair of electro-hydraulic valve actuator assemblies **1**.

## DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIG. **1**, the electro-hydraulic valve actuator assembly **1** is shown in front, top, bottom and right side views.

It consists of four major external components: valve body **25**, valve top cap **2**, valve bottom cap **5** and piston **6**. Valve top cap **2** primarily provides the means of locating and securing internal components and sealing the valve body **25**. Valve top cap **2** is fastened to valve body **25** by four (4) socket head cap screws **4** which are secured by four (4) lock washers **12** and is sealed with vent plug **8**. (Alternatively, vent plug **8** may be removed and a "case drain" line connected.) Detail of valve top cap **2** may be seen in FIG. **5** which shows front, top, bottom views and cross section C-C. Note cross section C-C showing stator bore **34**, bushing bore **35**, retaining ring slot **36** and threaded hole **37**. Also note fastener holes (4) **38** in the top view. See FIG. **6** for miscellaneous hardware items associated with valve top cap **2**.

FIG. **4** details valve body **25**, which is the main housing of the electro-hydraulic valve actuator assembly **1**. Anti-rotation pin slot **26** is machined axially into the wall of motor stator bore **29**. Electrical connector seat **27** provides the locating and seating surface on valve body **25** for electrical connector **3**. Electrical connector **3** is the means of providing electrical power and control signals into valve body **25**. The electrical connector **3** is required to seal against the internal hydraulic pressure of the valve as well as sufficiently isolate conductors such that electrical conduction through hydraulic fluid does not occur. Commercial hermetic connectors are available for this purpose, with varying methods of attachment to valve body **25**. Wiring passage **28** provides the route through valve body **25** by which electrical connector **3** wiring is routed to stator assembly **14**. Piston bore **30** serves to hold piston **6** and also contains the hydraulic fluid during the GEV open period. Rotary valve bushing bore **31** serves to hold bushing—rotary valve **15**. The threaded bolt holes—top cap fasteners **32** are for threading in top cap fasteners—socket head cap screws **4**. The threaded holes—bottom cap fasteners **33** are for threading in bottom head fastener—flat head cap screw **7**. FIG. **7** shows front, top, bottom views and cross section D-D of valve bottom cap **5**. Piston bore **39** and seal ring groove **40** are illustrated in the top view and cross section D-D. Valve bottom cap **5** is fastened to valve body **25** by four flat head cap screws **7**. Piston **6** provides the linear reciprocating motion by which the electro-hydraulic valve actuator assembly **1** opens and closes valve-poppet **50** for intake and exhaust of the cylinder gasses (shown in FIG. **14**). Hydraulic fluid intake port **9** and hydraulic fluid exhaust port **10** provide the means of supplying hydraulic pressure and flow into and out of electro-hydraulic valve actuator assembly **1**. Locating groove **11** provides the means of locating electro-hydraulic valve actuator assembly **1** in respect to the valve-poppet **50**.

One of ordinary skill in the art will recognize that electro-hydraulic valve actuator assembly **1** can be constructed in a variety of ways and the foregoing is intended only to serve as an example of many satisfactory means of constructing the present invention. For instance, valve top cap **2**, valve bottom cap **5** and valve body **25** could be welded instead of bolted together, and a bolted flange could replace locating groove **11**.

FIG. **2** shows cross section A-A of FIG. **1**, electro-hydraulic valve actuator assembly **1** illustrating stator assembly **14** and rotor assembly **13** which is rotated by the revolving magnetic field generated by stator assembly **14**—which, as illustrated, is functioning as a two phase synchronous electric motor. Rotor assembly **13** is shown oriented in the open position to hydraulic fluid inlet port **9** providing hydraulic pressure and flow first through hydraulic passage-rotary valve **19** then through hydraulic passage **18** to piston **6**. Piston **6** is forced by hydraulic pressure and flow to move along, piston bore **30** (see FIG. **4**) away from hydraulic passage **18** and toward valve bottom cap **5**—thus providing a linear actuating force along the axis of travel to an external member (valve cap etc.) in contact with piston **6** external face.

Electrical connector **3** is connected to an external control and power source (not shown) and is internally electrically wired to stator assembly **14**. Note: Commercial integrated circuits are available for the purpose of providing control and power to stator assembly **14**. FIG. **11** shows front and top views of stator **14**. It can be seen that the stator consists of a stack of magnetic metal laminations **41** and insulated wire coil **42**. These are of typical construction to that used in existing small electric servo motors.

FIG. **12** shows a front and top view of rotor assembly **13**, then front, top, bottom and cross section F-F of motor shaft and integral rotary valve **45**, as well as front and top views of permanent magnet **44**. Permanent magnet **44** is made from high strength permanent magnet material, preferably with a high temperature rating. Such materials would be commonly known to one of ordinary skill in the art and the choice from available materials is a trade-off between cost and performance for the particular engine requirements. Vent hole **46** is shown in motor shaft and integral rotary valve **45** the purpose of which is to facilitate purging of entrapped air on the initial filling of the hydraulic fluid. It can be seen that rotor assembly **13** is an assembly of permanent magnet **44** and motor shaft and integral rotary valve **45**. The fit and assembly of these items is typical of that used in permanent magnet servo motors. Such information would be known to one of ordinary skill in the art and is also available from a variety of texts on motor design. Note the magnetic field orientation of permanent magnet **44**. Cross section F-F of motor shaft and integral rotary valve **45** shows hydraulic passage **19**. This is illustrated with a round hole as the hydraulic passage. However, this need not be the case and other passage cross sectional geometries may be used to alter the hydraulic fluid flow rate (thus providing different actuator movement profiles)—in conjunction with bushing-rotary valve **15**. The hydraulic fluid flow rate alters the actuated GEV rate of travel and/or opening and closing profile. Thus, the control over the rate of rotation and angular position, along with the port geometry, can be used to infinitely vary the valve operating parameters. These parameters are a function of the desired operating characteristics of the specific engine application. Rotor assembly **13** is located and supported at the lower end by bearing bushing-rotary valve **15** (see FIG. **9**). The finish and dimensional tolerances of bearing bushing-rotary valve **15** would be those typically found on hydraulic spool valves. Such information would be known to one of ordinary skill in the art and is available from a variety of texts on the subject of hydraulic valve design and in particular on hydraulic servo valve design.

Referring to FIG. **8**, piston **6** has incorporated into the internal (upper) face a boss with a pair of radial slots, the function of which is to act as a hydraulic snubber as the piston **6** reaches the end of the return stroke and the valve **50** seats. This snubbing action provides a so called "soft landing" for the valve **50** as it seats. A person of ordinary skill in the art would recognize that there are a variety of ways to accomplish this snubbing action, in either direction of travel of the piston **6**. Referring to FIG. **10**, piston seal **16** provides dynamic sealing of piston **6** and bottom cap seal ring **17** provides static sealing of hydraulic pressure to valve bottom cap **5**. Piston seal **16** would typically be a conventional hydraulic cylinder lip seal. Sealing ring **17** would typically be an "O" ring. The fit and finish requirements of piston **6** and piston bore **30** are typical of hydraulic pistons and cylinders, which is available from a variety of texts on hydraulic cylinder design and would be known by a person of ordinary skill in the art.

Referring to FIG. **6**, bearing bushing-motor shaft **20** is retained in valve top cap **2** by retaining ring **23**. Belleville washer **21** and thrust washer **22** provide axial thrust on rotor assembly **13**. The purpose of this thrust is to hold the rotor assembly **13**, in a manner to minimize the clearance between the end of and rotor assembly **13** and actuator body **25**—so that leakage of hydraulic fluid from hydraulic passage-rotary valve **19** to the hydraulic fluid outlet port **10** is minimized. Sealing ring-top cap **24** provides sealing of hydraulic pressure for valve top cap **2**.

FIG. **3** is cross section A-A of FIG. **1**. It can be seen that rotor assembly **13** is rotated by a revolving magnetic field generated by two phase electrical power created by stator assembly **14**—which is electrically wired through electrical connector **3** to an appropriate external electronic control module (of which a number of commercially available devices are suitable). The stator assembly **14** and rotor assembly **13** preferably operate as a two phase servo motor with infinitely variable control over the angular position and rotational speed. A person of ordinary skill in the art would recognize that, alternatively, the motor could also function in the so called "stepper or indexing mode" of rotation. Also, a person of ordinary skill in the art would recognize that, alternatively, a three phase (or more) motor and power source could be utilized in place of the basic two phase motor illustrated. It is appropriate to note that commercial open frame motors are widely available and are quite suitable for the purpose intended herein. Furthermore, alternate motor types, such as the switched reluctance motor, may be utilized.

Referring to FIG. **3**, it may be seen that rotor assembly **13** is shown rotated toward hydraulic fluid outlet port **10** which relieves hydraulic pressure on piston **6** de-actuating valve—poppet **50** allowing it to close under pressure from valve spring-coiled **49**. Concurrently, rotor assembly **13** also blocks hydraulic pressure and flow from the hydraulic fluid inlet port **9**. With valve-poppet **50** (intake or exhaust valve) held in the closed position by valve-poppet **50**, the rotation of motor shaft and valve rotor **45** continues at a rate as determined by the external electronic control (not shown) until the hydraulic passage **19** again aligns with hydraulic fluid inlet port **9** and the valve-poppet **50** again opens. Hydraulic fluid inlet port **9** and hydraulic fluid outlet port **10** are shown located ninety degrees apart in valve body **25**, thus the speed of rotation of rotor assembly **13** is one half that of the engine speed (similar to a conventional camshaft arrangement). Alternative angular location of the inlet port **9** and outlet port **10** is possible but the 90 degree orientation is preferred as it allows for slower valve rotation (one half engine speed).

FIG. **13** shows mounting bracket clamp **47** and mounting bracket **48** which are suitable for mounting two of the electro-hydraulic valve actuator assembly **1**. This provides for valve actuation of a single cylinder of an internal combustion engine. One of ordinary skill in the art would recognize that a wide range of suitable mounting brackets can be developed for a variety of on-engine conditions and that the one shown herein serves only as an example.

In FIG. **14** valve spring-coiled **49** and valve-poppet **50** are shown in a front and side view with mounting bracket clamp **47**, mounting bracket **48** and two electro-hydraulic valve actuator assemblies **1** which illustrate a typical installation for a single cylinder. Note: The cylinder head to which mounting bracket **48** would be fastened and on which valve-poppet **50** would be located has been omitted for clarity.

Referring to FIG. **15**, hydraulic pump piston **51** provides hydraulic power for two electro-hydraulic valve actuator assemblies **1** for actuating valve-poppet **50** for a single cylinder of an engine. Thus, as illustrated, a hydraulic piston pump is required for each cylinder in an engine. The pump cylinder **54** houses the piston **51** and pump inlet valve **55** and pump outlet valve **56**. The piston **51** is driven by cam **53** during the output stroke of piston **51**. Spring **52** drives the piston **51** during the hydraulic fluid intake stroke of the pump. During the intake stroke, hydraulic fluid is drawn from the hydraulic reservoir **58** through suction line **57** and intake valve **55** by the piston **51**. The cams **53**, driving all the pump pistons for each cylinder may be on a common shaft driven by a takeoff from the engine shaft or from a separate drive—such as by an electric motor synchronized to the engine speed and piston position. Alternatively the pump pistons **51** may be directly driven by the reciprocating motion of the engine pistons. One skilled in the art would recognize that hydraulic pressure and flow could also be provided by a variety of hydraulic pumps driven in a number of different ways. In the method as shown, hydraulic fluid under pressure is driven out of the cylinder **54** and through outlet valve **56** and into the high pressure lines **59**. The timing of the illustrated pump operation is such that the valve actuators are closed during the discharge of hydraulic fluid from the pump. Thus, hydraulic fluid under pressure flows into accumulator **60**, where it remains under pressure until it is required to open an engine intake or exhaust valve. When required by an electro-hydraulic valve actuator assembly **1**, the hydraulic fluid flows out of the accumulator **60** through high pressure lines **59**, then through hydraulic fluid inlet port **9**. The high pressure hydraulic fluid drives the actuating piston **6**, forcing valve spring **49** to compress and the valve-poppet **50** to open. When the electro-hydraulic valve actuator assembly **1**, moves to the close position (by the rotor assembly **13**, turning such that hydraulic passage-rotary valve **19**, aligns with outlet port **10**), the hydraulic fluid in the valve discharges through the outlet port **10**, where it then flows through the return lines **61** to the hydraulic fluid reservoir **58**.

One of ordinary skill in the art would recognize that the invention herein disclosed can be implemented over a wide range of size and capacity to suite the requirements of a wide range of engine types and size. Further, one of ordinary skill in the art would readily recognize that suitable material and components must be selected for the specific on-engine operating conditions, with particular attention to the temperature and chemical environmental properties. Additionally, one of ordinary skill in the art would foresee that piston **6** could be arranged other than co-axially with rotor assembly **13**, as shown herein, and that a wide variety of configurations is possible. One skilled in the art would also recognize that multiple electro-hydraulic valve actuator assemblies **1** could be installed in one housing for a single engine cylinder. Also, one of ordinary skill in the art would readily recognize that alternate types of valve springs, such as pneumatic or magnetic springs, could be employed and in addition, valve springs of varying types could be made integral within electro-hydraulic valve actuator assembly **1**.

What is claimed is:

**1**. An actuator for operation of an internal combustion reciprocating engine gas exchange poppet valve, comprising:

- a linear acting hydraulic cylinder arranged to actuate said gas exchange poppet valve;
- a rotary valve for directing hydraulic fluid pressure and flow into and out of said linear acting hydraulic cylinder;
- ports and means of passage in said rotary valve for conveying said hydraulic fluid flow;
- an electric motor for driving said rotary control valve;
- a housing for containing said linear acting hydraulic cylinder, said rotary valve and said electric motor;

ports and means of passage in said housing for conveying hydraulic pressure and flow in and out of said actuator and to and from said rotary valve;

an electrical connector in said housing for conducting electrical power and electronic signals in an out of said housing; and,

a means of mounting said housing adapted to affect the actuation of said internal combustion reciprocating engine gas exchange valve.



**2**. The actuator of claim **1** wherein said linear acting hydraulic cylinder, said rotary valve and said electric motor are arranged in line within said housing.

**3**. The actuator of claim **2** wherein the hydraulic cylinder is of a single acting configuration.

**4**. The actuator of claim **1** wherein the hydraulic cylinder is of a single acting configuration.

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