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(12) **United States Patent**
Cotton

(10) **Patent No.: US 8,776,756 B2**
(45) **Date of Patent: Jul. 15, 2014**

(54) **SLIDING VALVE ASPIRATION**

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **13/863,710**

(22) Filed: **Apr. 16, 2013**

(65) **Prior Publication Data**

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Related U.S. Application Data

(60) Continuation-in-part of application No. 13/443,077, filed on Apr. 10, 2012, now Pat. No. 8,459,227, which is a division of application No. 12/387,184, filed on Apr. 29, 2009, now Pat. No. 8,210,147.

(60) Provisional application No. 61/135,267, filed on Jul. 18, 2008.

(51) **Int. Cl.**
F01L 5/00 (2006.01)

(52) **U.S. Cl.**
CPC ***F01L 5/00*** (2013.01)
USPC .... **123/188.5**; 123/190.8; 123/81 C

(58) **Field of Classification Search**
CPC F01L 5/00; F01L 5/04; F01L 7/00; F01L 7/02; F01L 3/08
USPC .... 123/188.4, 188.5, 188.12, 190.8, 123/190.12, 81 R, 81 B, 81 C, 81 D, 84
See application file for complete search history.

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(Continued)

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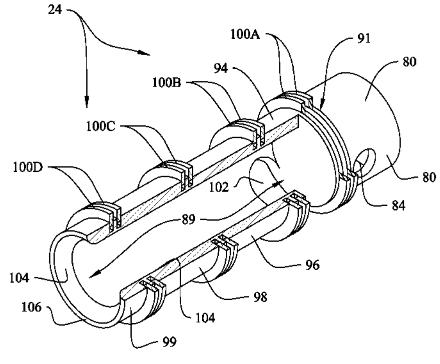
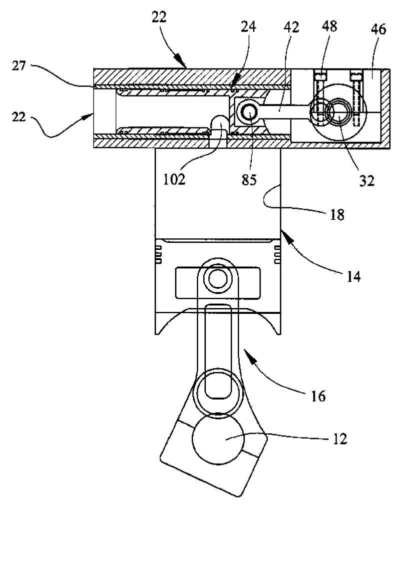
(57) **ABSTRACT**

Multi-section sleeve valves for internal combustion engines for improved aspiration. An open connecting rod section is separated from an internal, tubular passageway by a closed wall. A port section proximate the wall defines valve ports. A power stroke midsection borders the port section. An oiling section borders the midsection, and an open section adjacent the oiling section is in fluid flow communication with the tubular passageway. The lower-diameter midsection forms a relief annulus between the valve and the tunnel or sleeve in which the valve is disposed. Fluid flow occurs through the valve interior and through ports dynamically positioned above the compression cylinder, proximate aligned sleeve and head ports. Sleeve ports are separated by bridges that maintain valve rings in compression during reciprocation to prevent damage. High pressure gas is confined between axially spaced apart, stepped sealing rings that prevent gases from flowing axially about the valve exterior.

8 Claims, 21 Drawing Sheets





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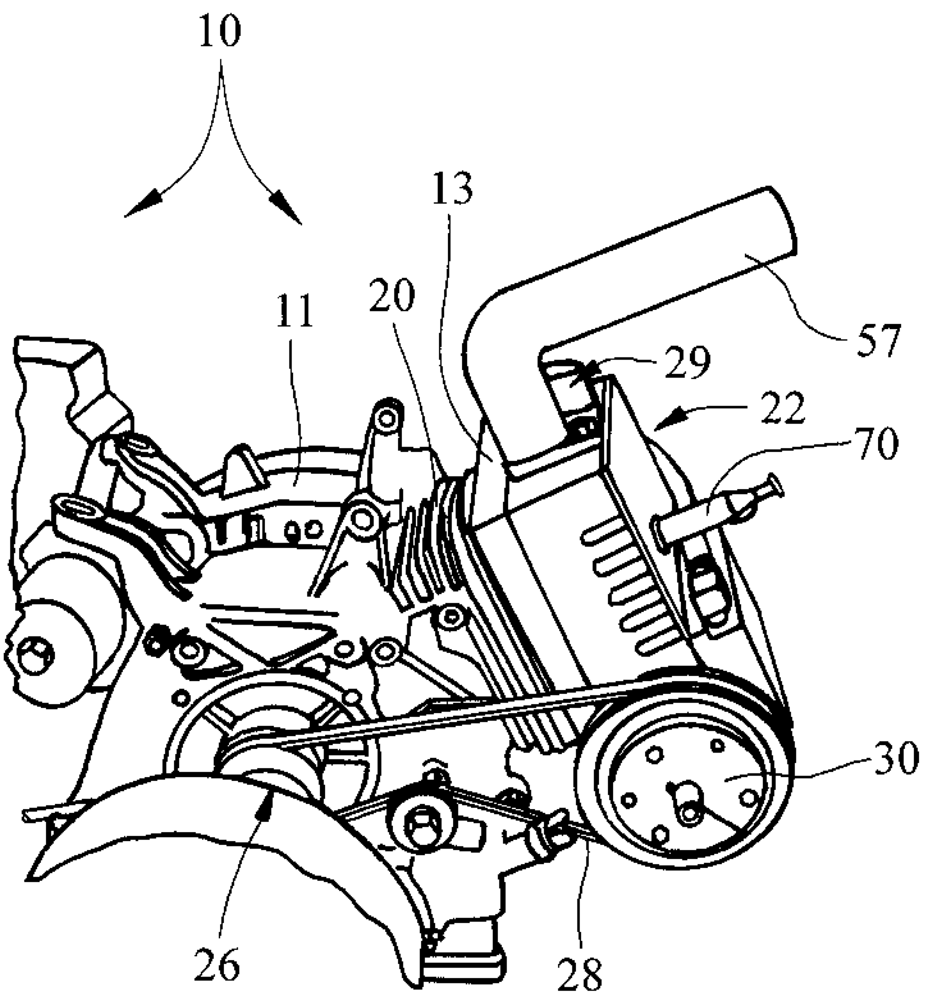


Fig. 1

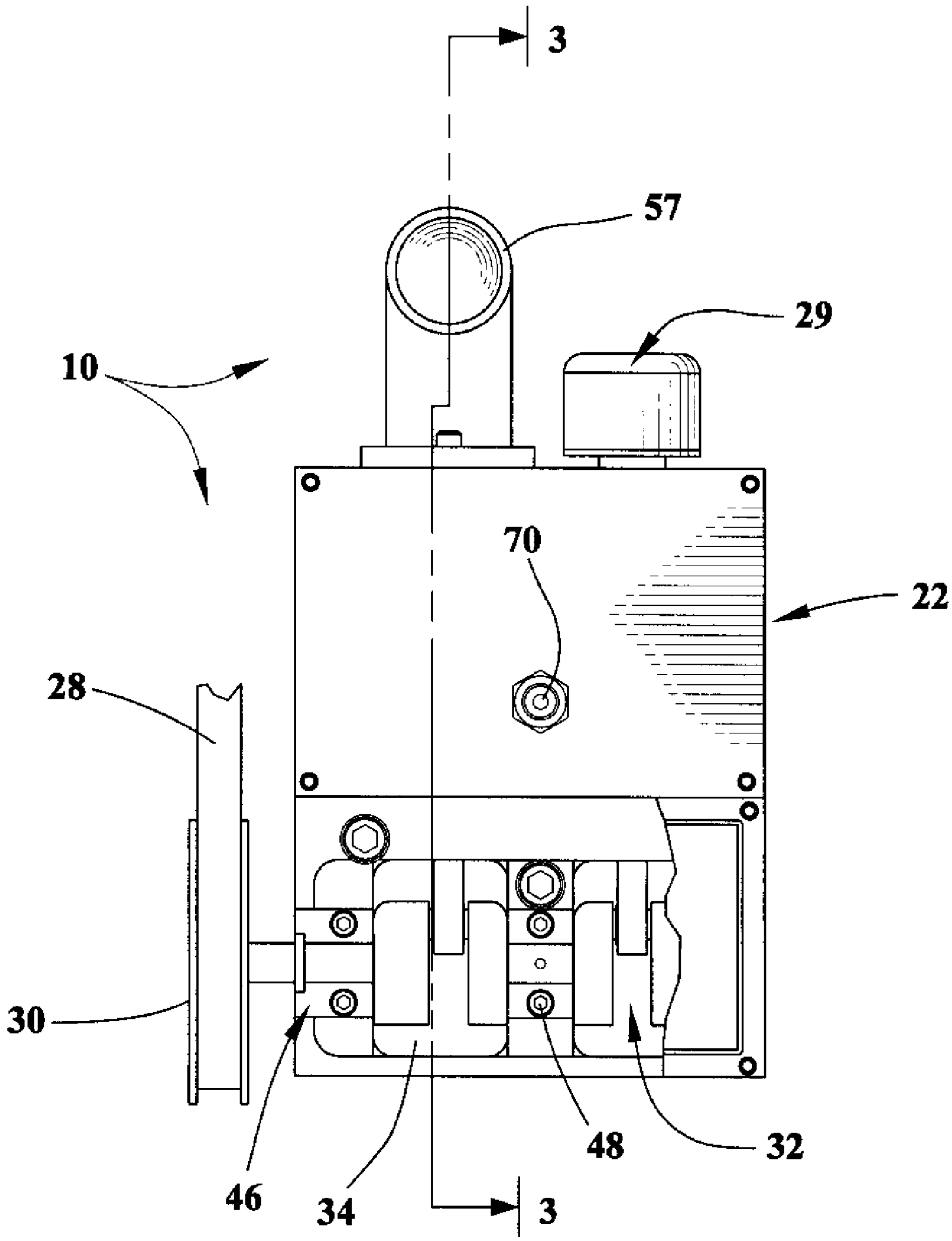


Fig. 2

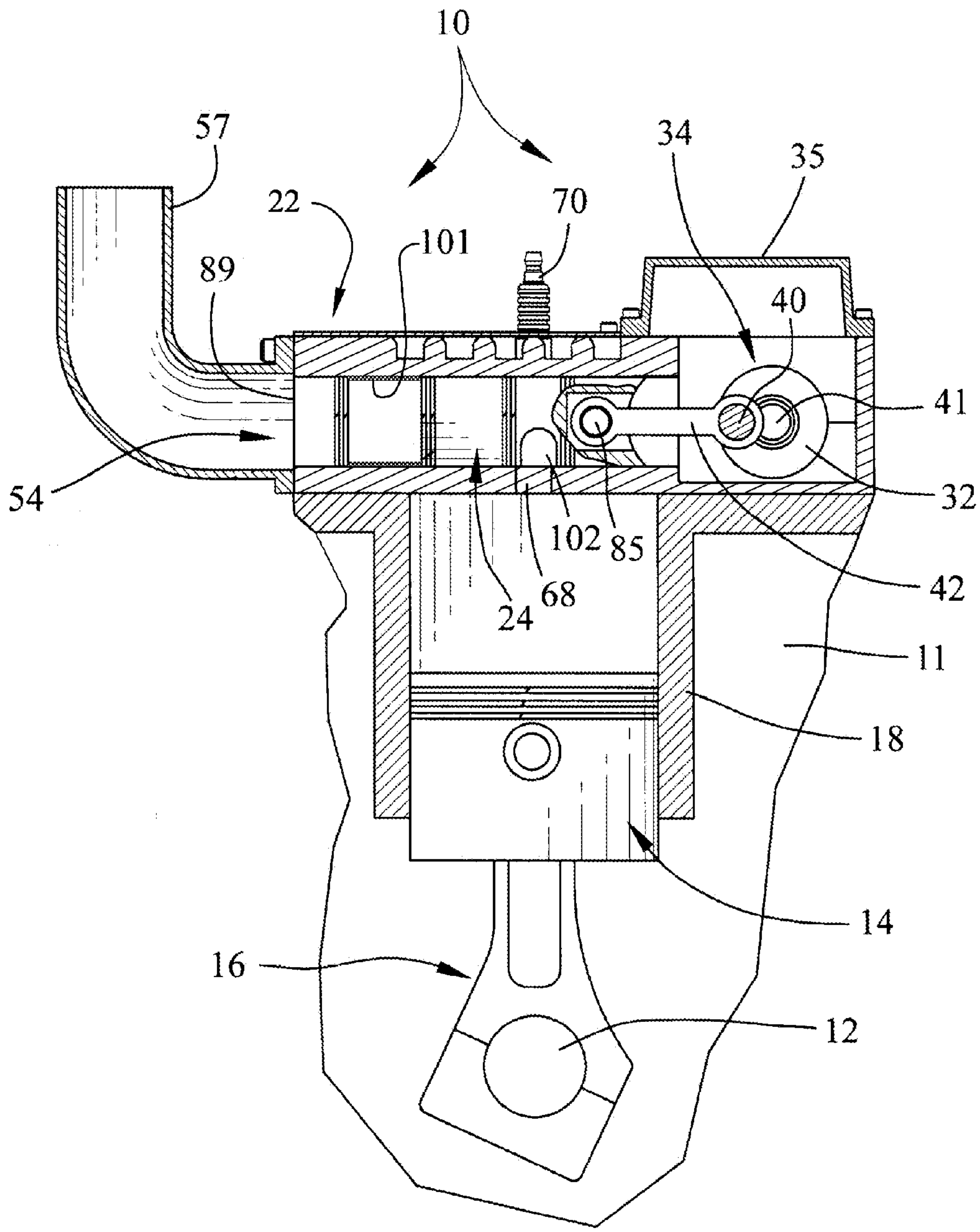


Fig. 3

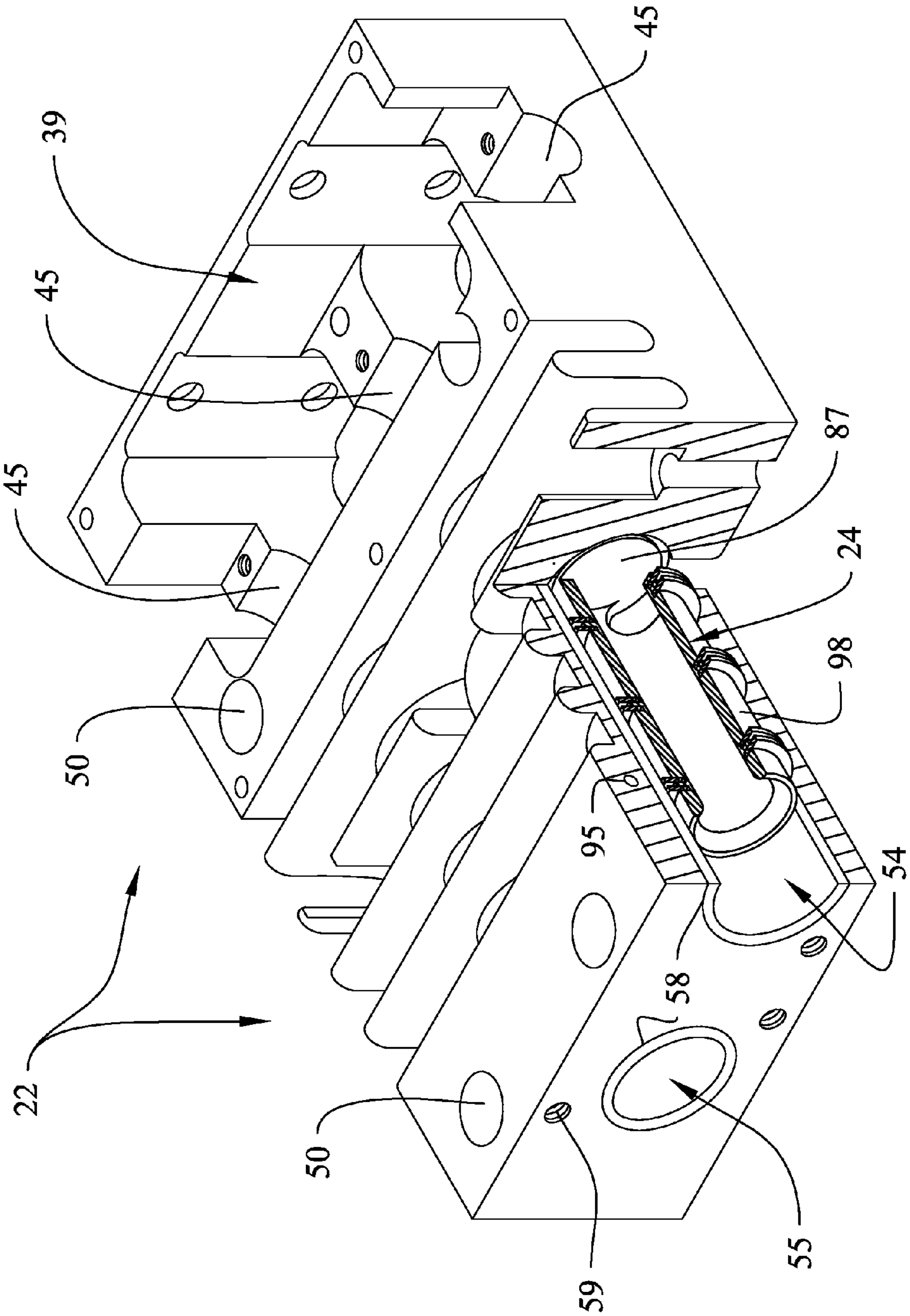


Fig. 4

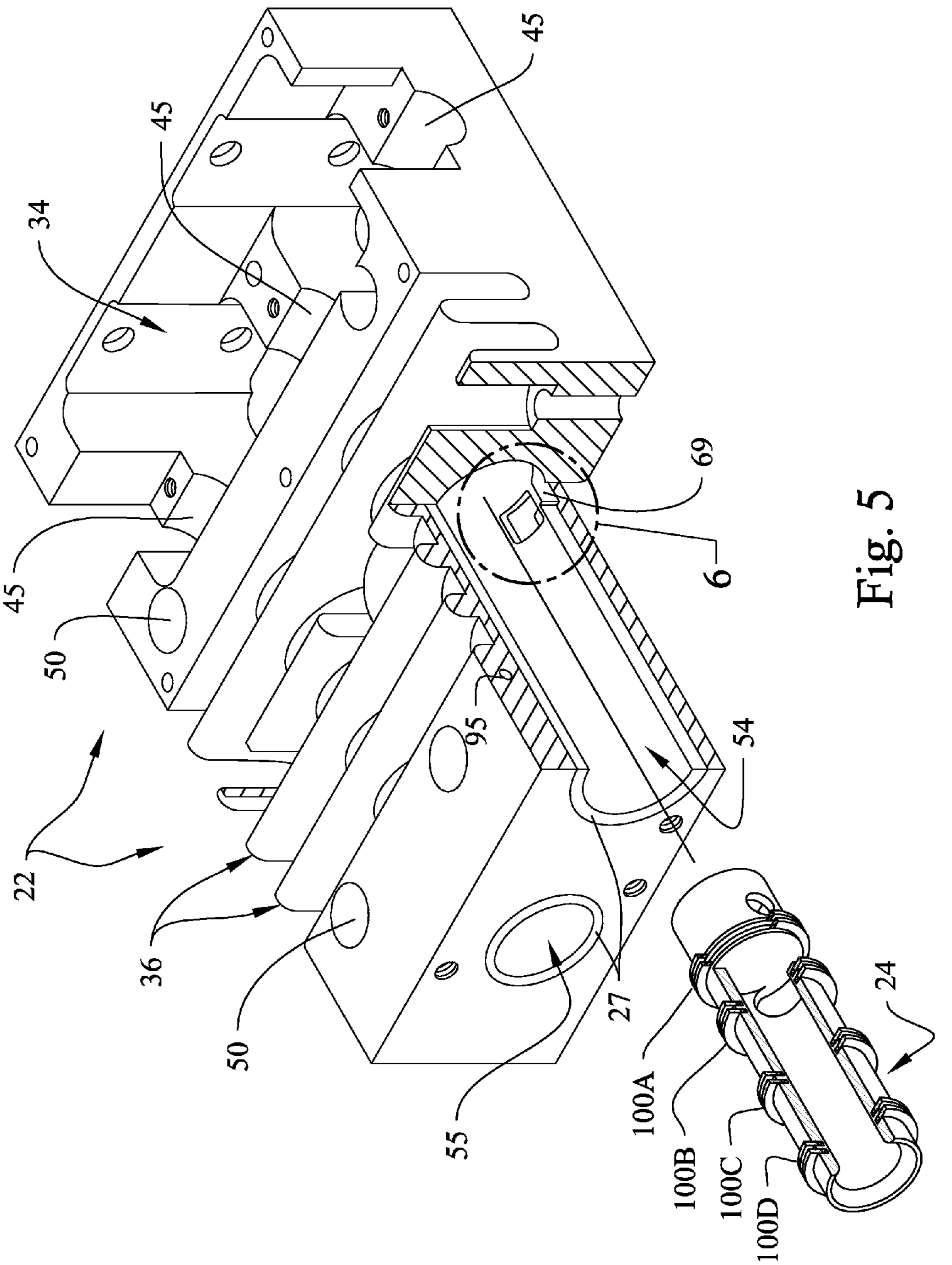


Fig. 5

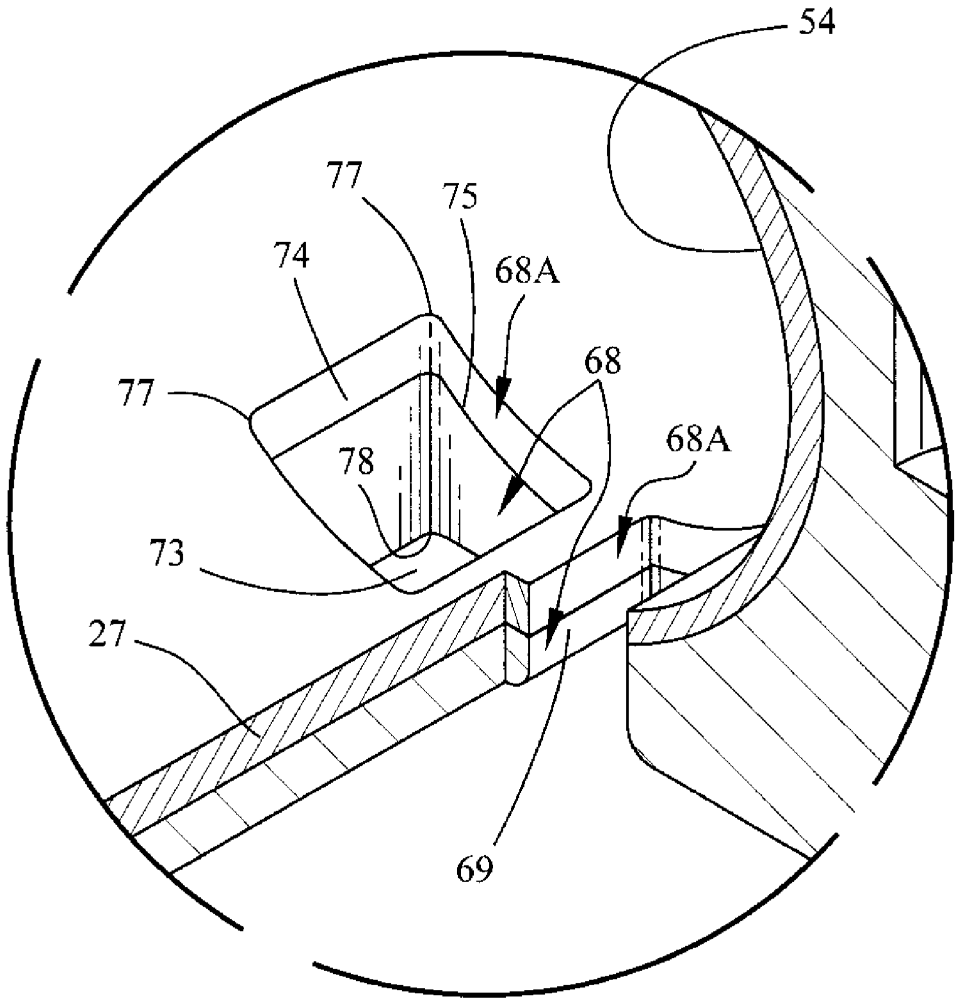


Fig. 6

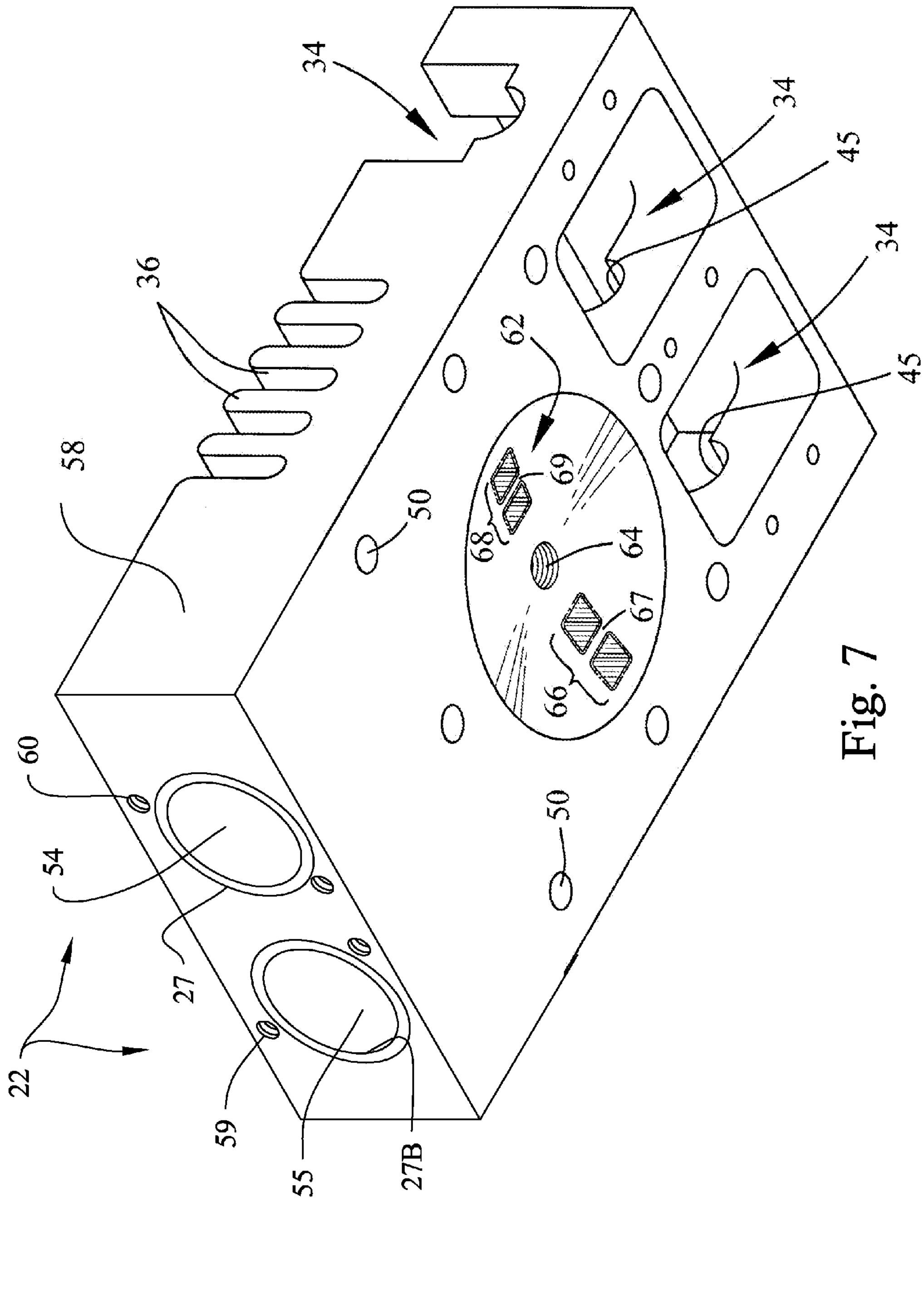


Fig. 7

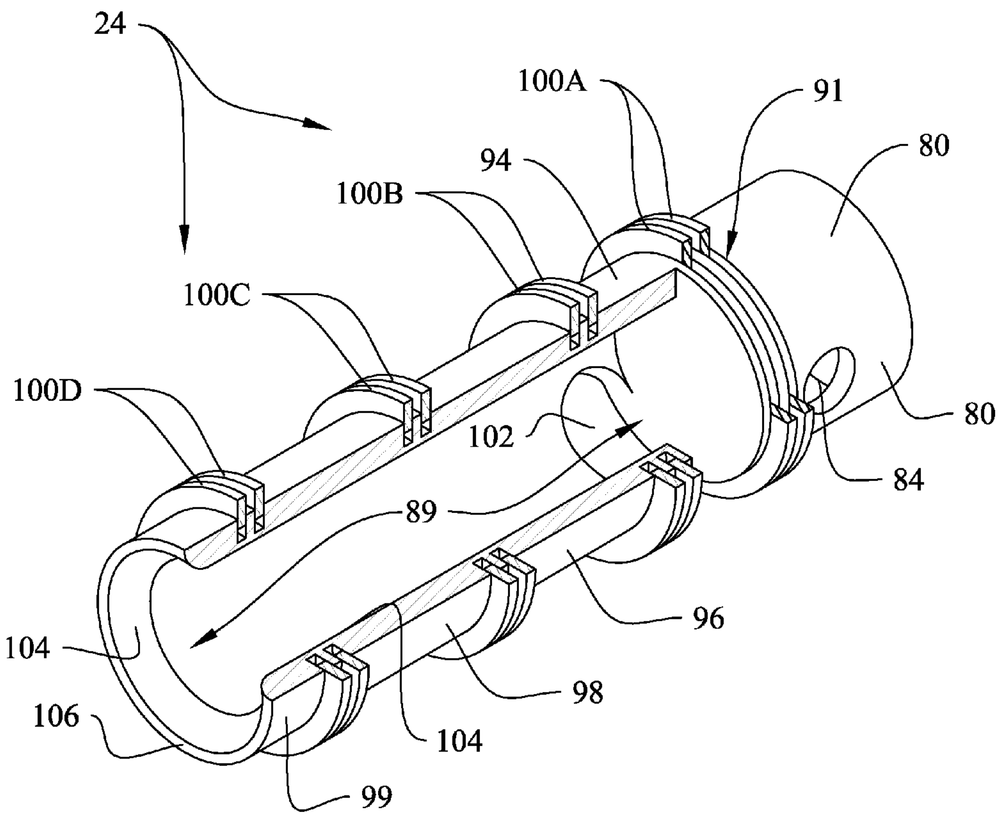


Fig. 8

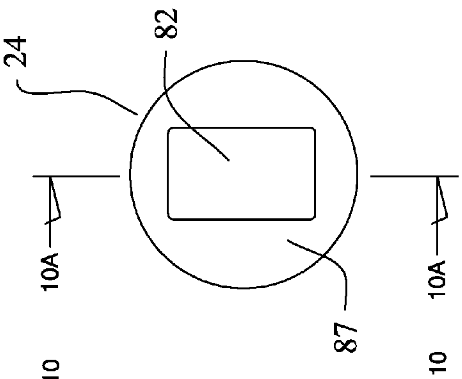


Fig. 10

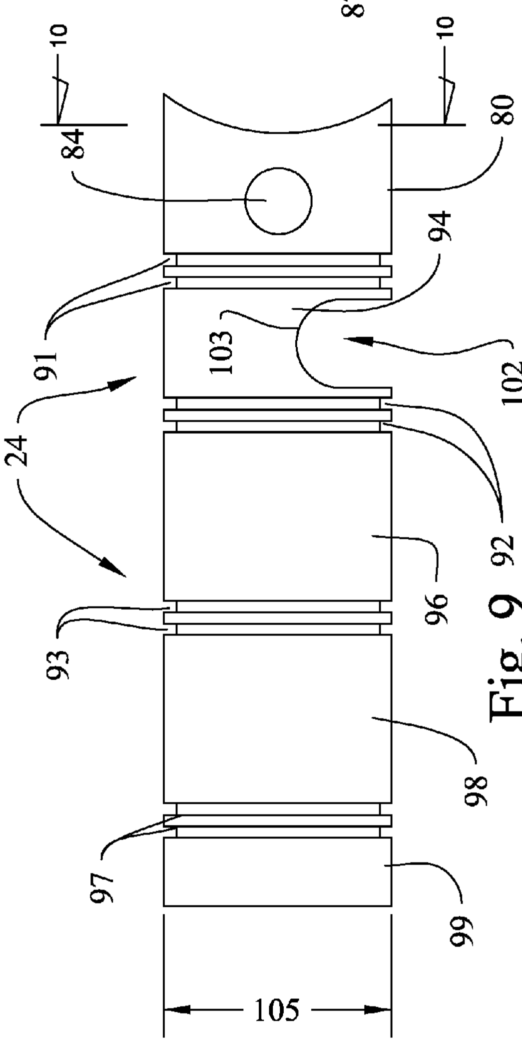


Fig. 9

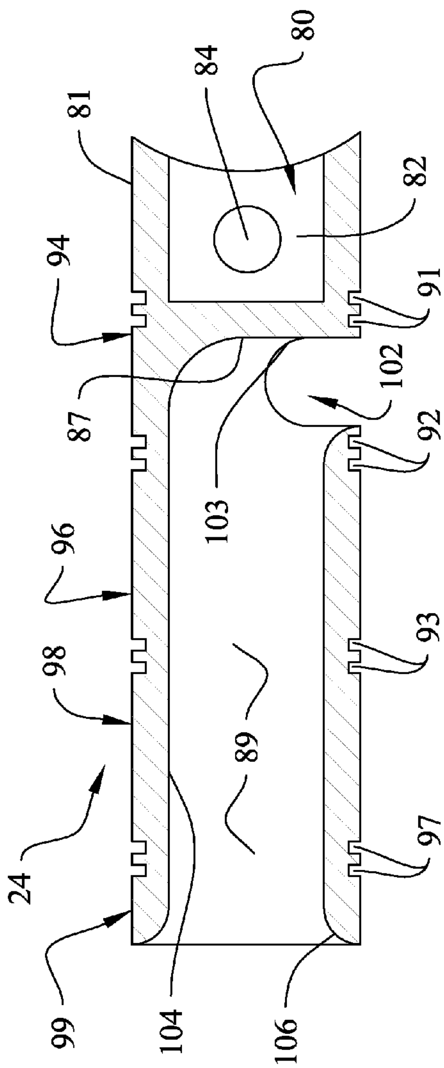


Fig. 10A

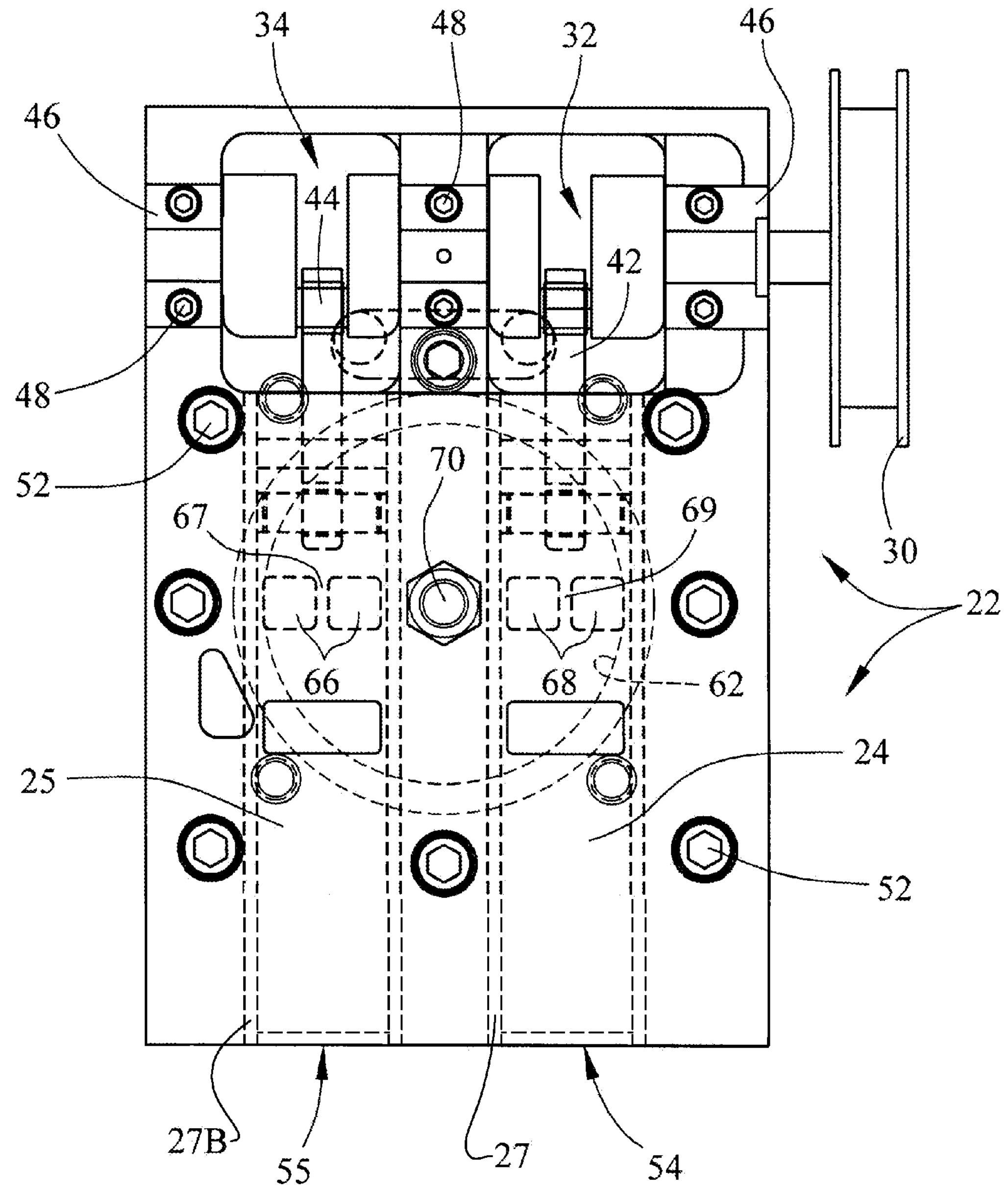


Fig. 11

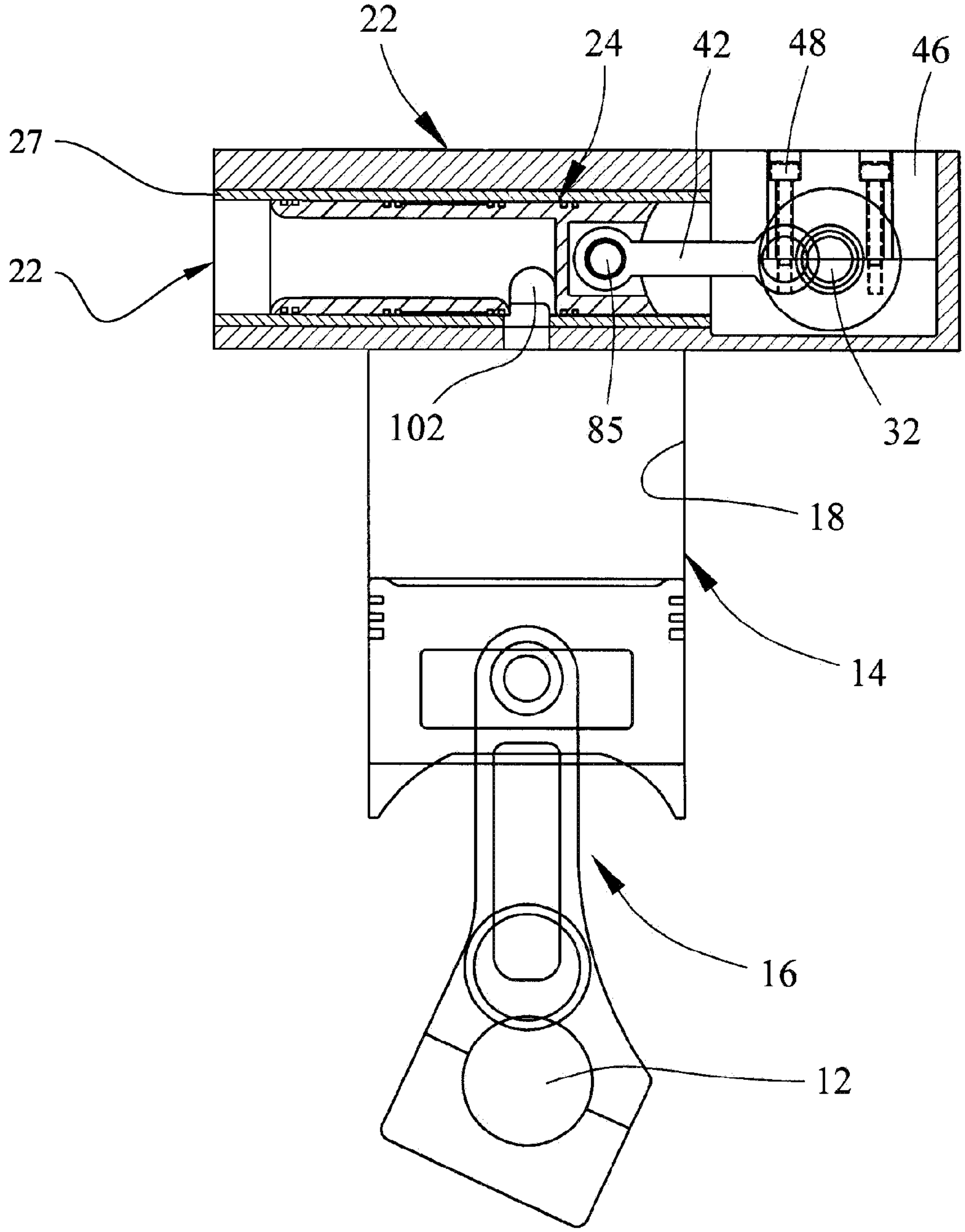
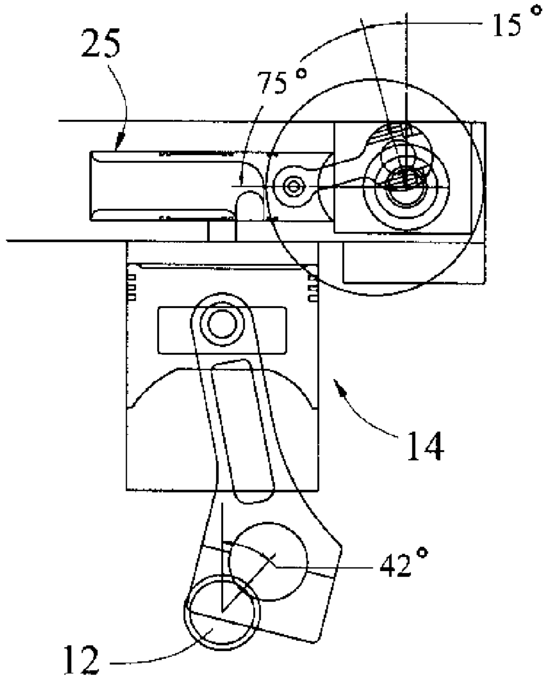
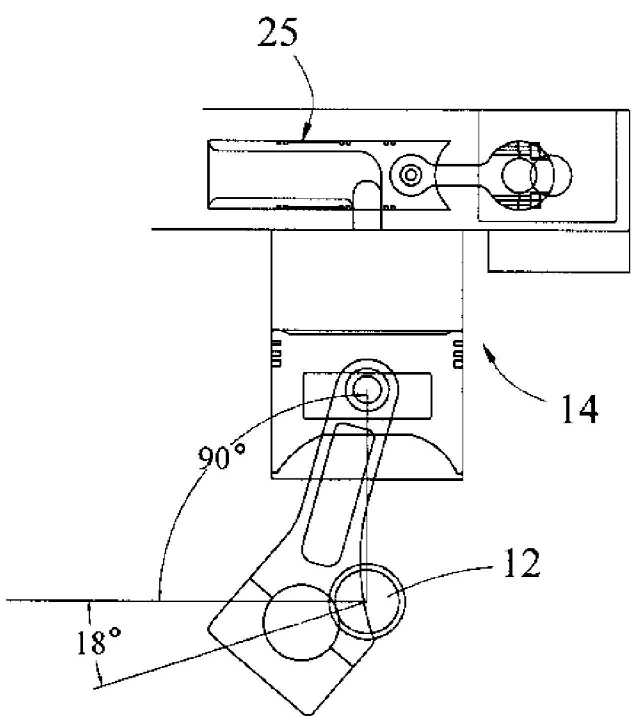


Fig. 12



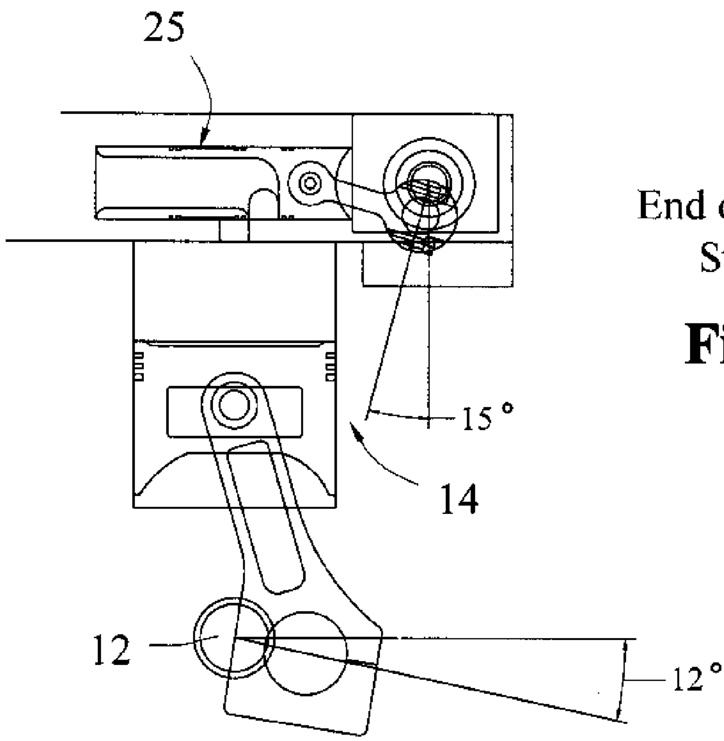
Beginning of
Intake Stroke

Fig. 13



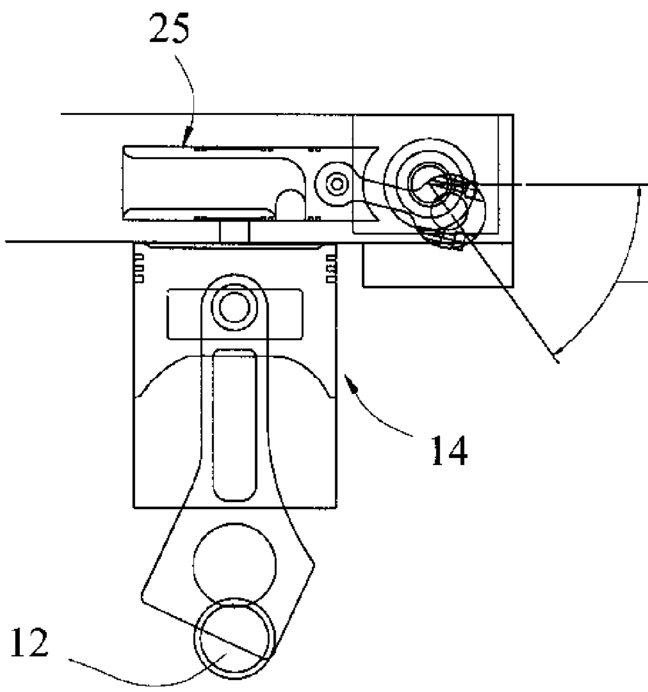
Intake Valve
Open at 108°

Fig. 14



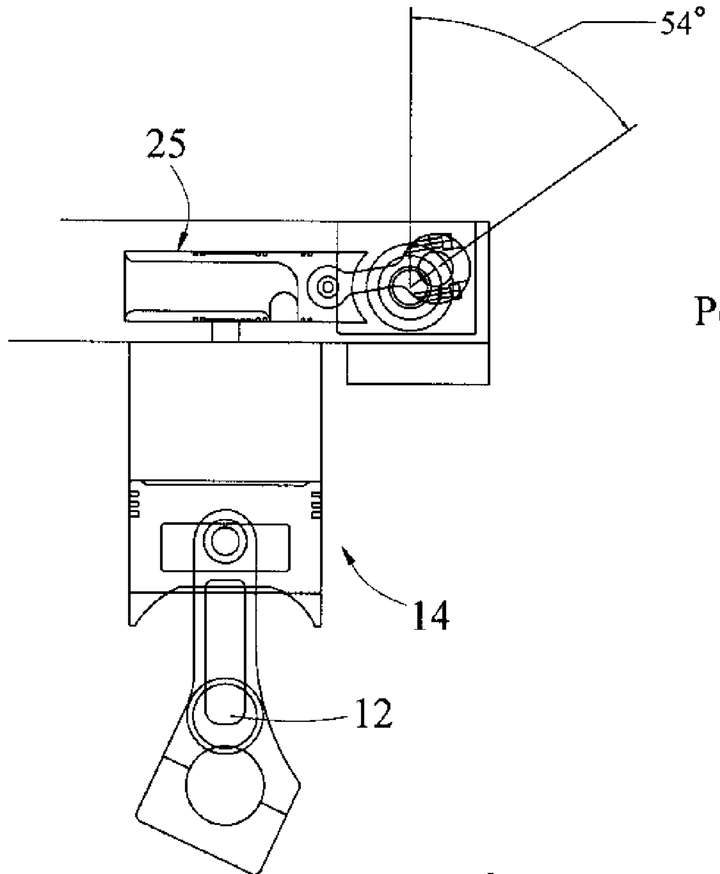
End of Intake Stroke

Fig. 15



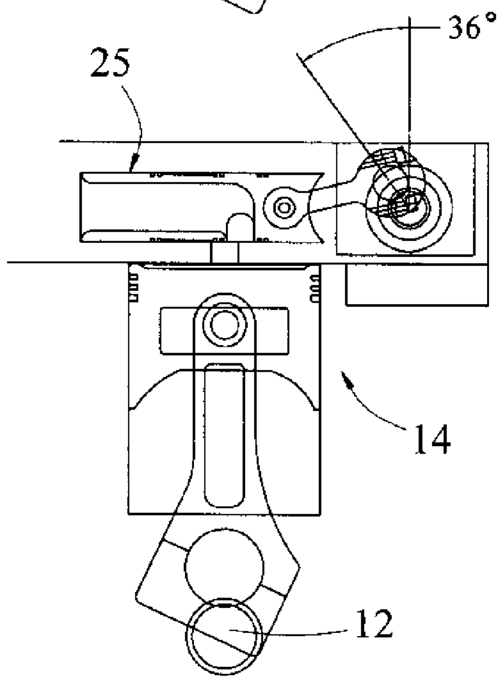
Beginning of Power Stroke

Fig. 16



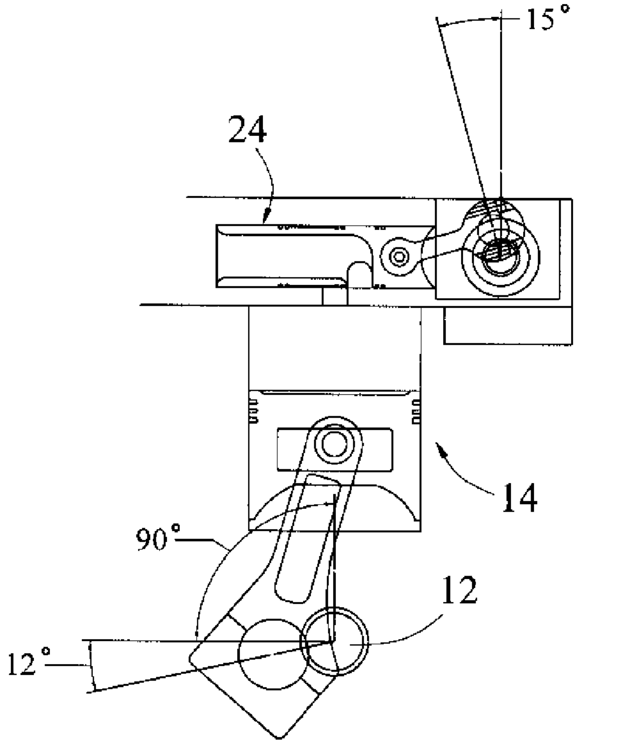
Bottom of
Power Stroke

Fig. 17



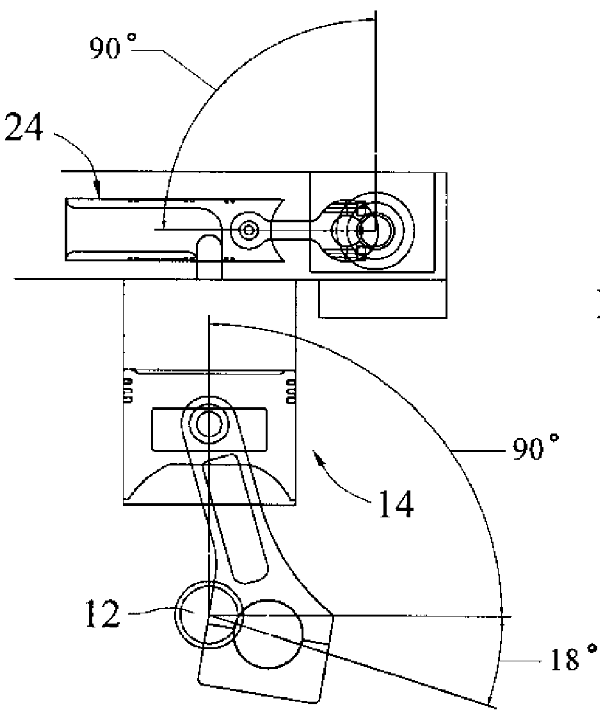
End of Exhaust
Stroke

Fig. 18



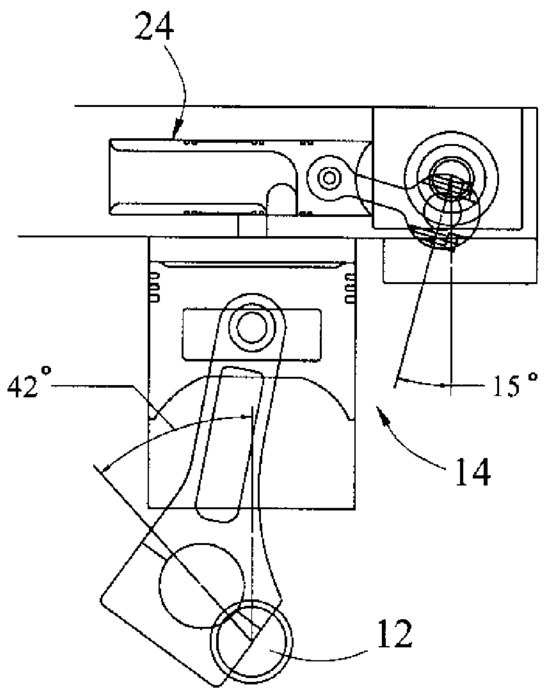
Exhaust Valve at
Start of Exhaust Stroke

Fig. 19



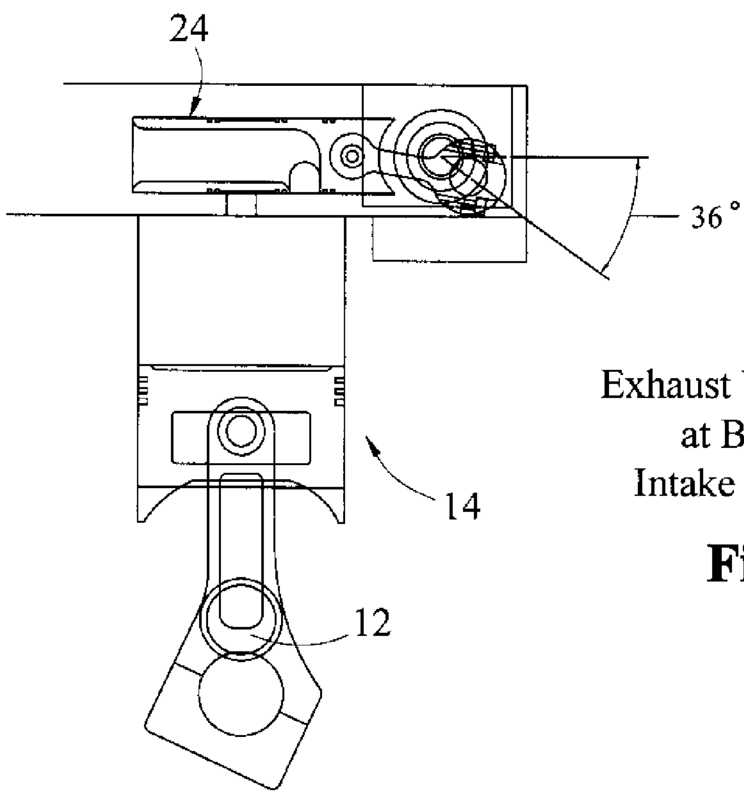
Exhaust Valve
Fully Open 251°

Fig. 20



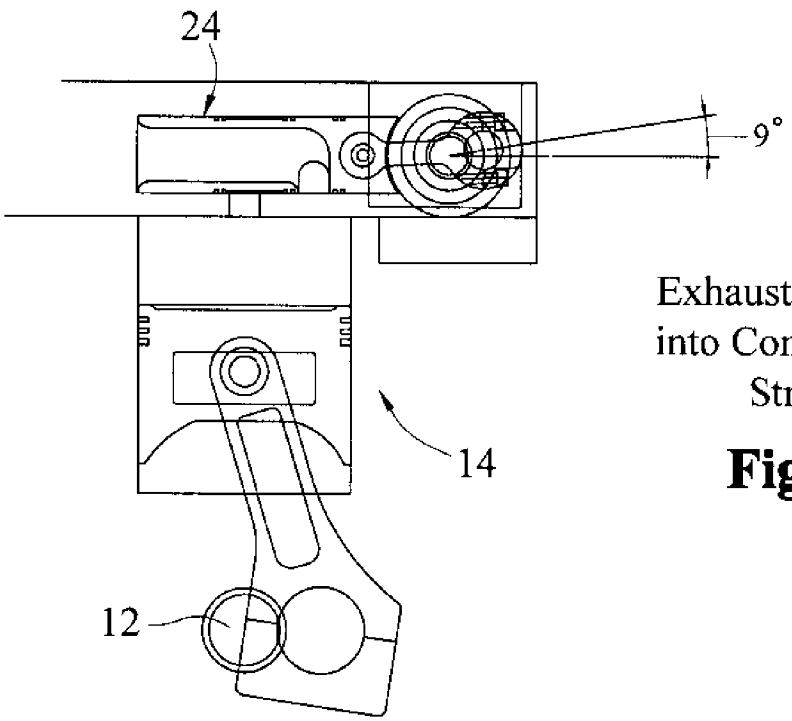
Exhaust Valve Beginning
Closure at Beginning of
Intake Stroke 222°

Fig. 21



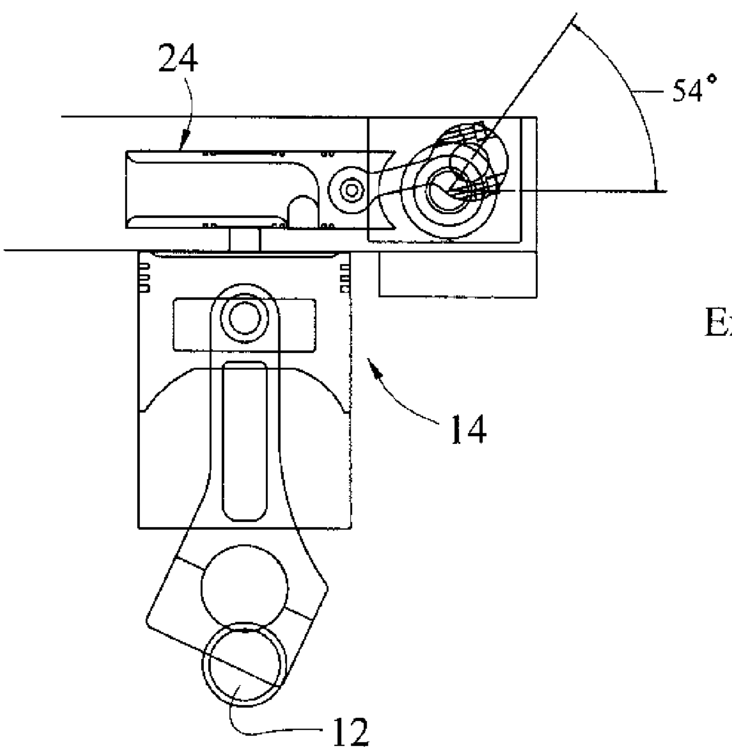
Exhaust Valve Closed
at Bottom of
Intake Stroke 180°

Fig. 22



Exhaust Valve 90° into Compression Stroke

Fig. 23



Exhaust Valve at 0° TDC

Fig. 24

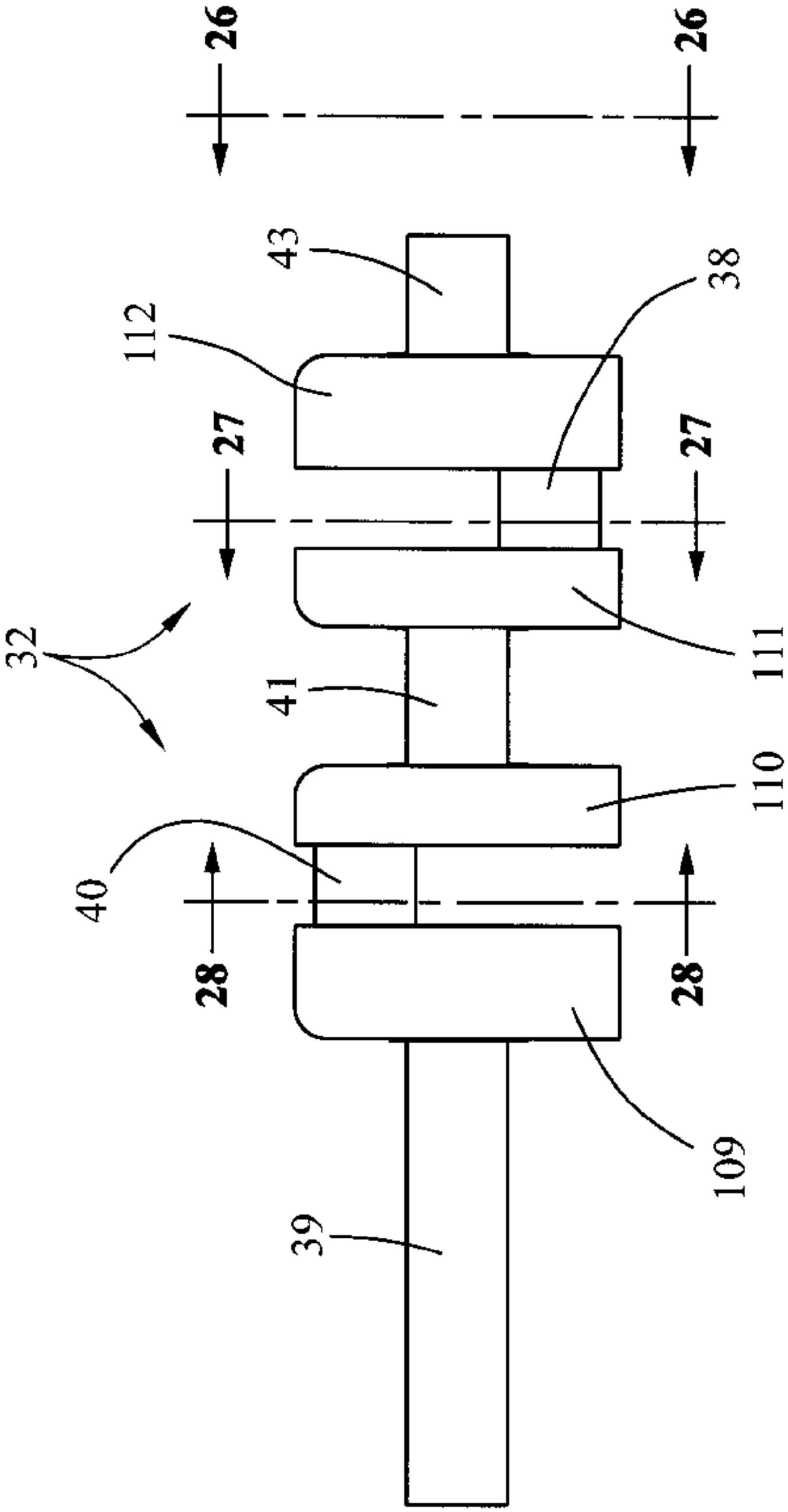


Fig. 25

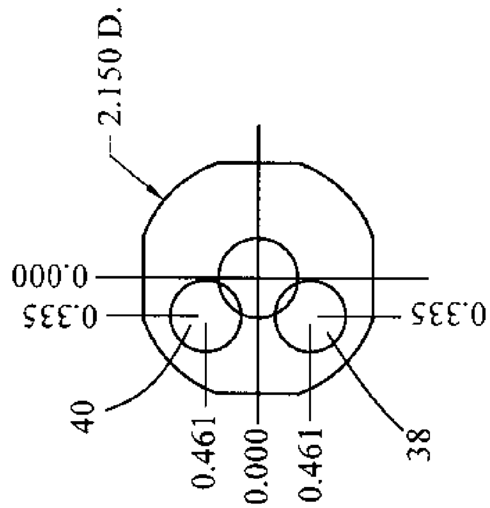


Fig. 28

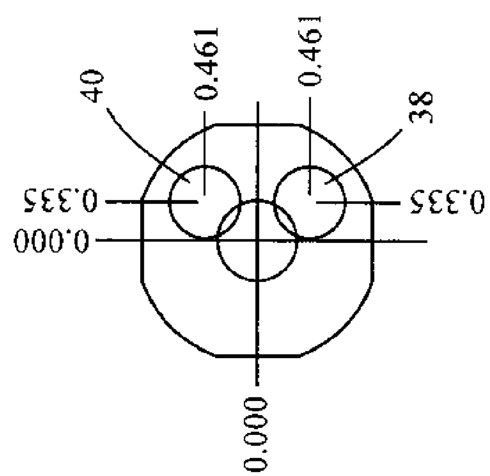


Fig. 27

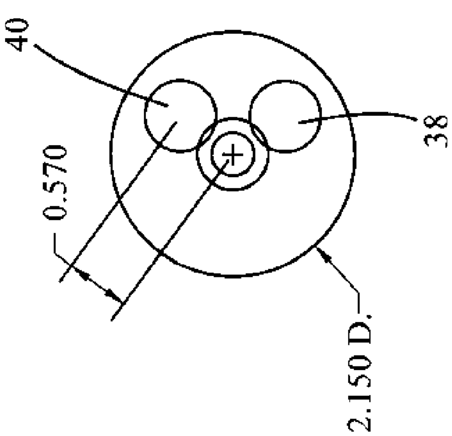
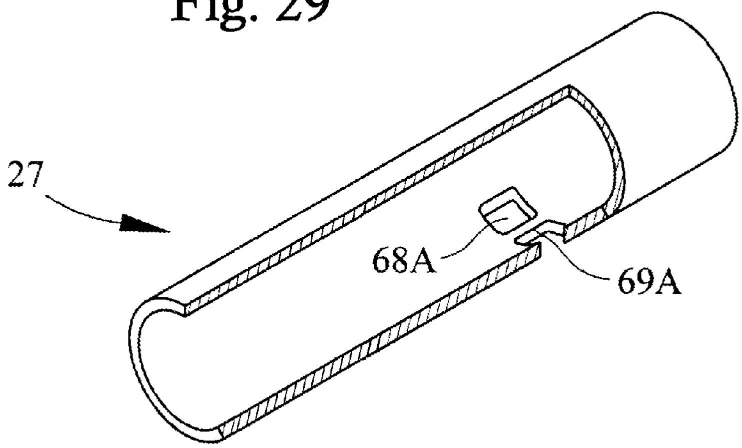


Fig. 26

Fig. 29



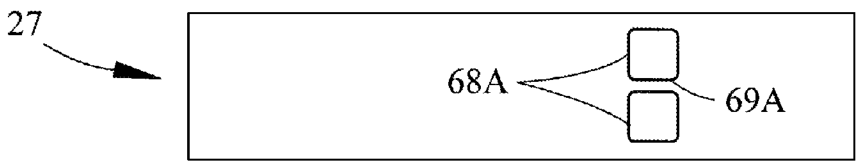


Fig. 30

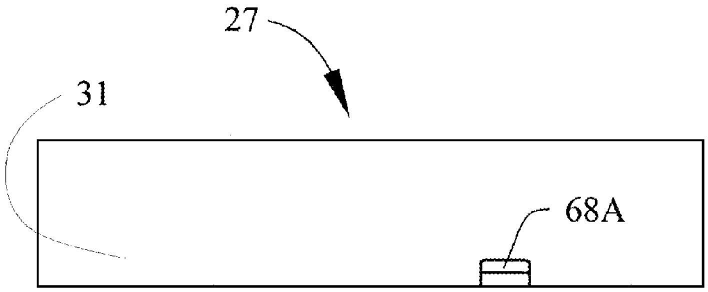


Fig. 31

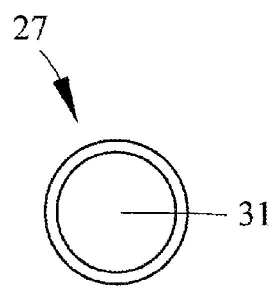


Fig. 32

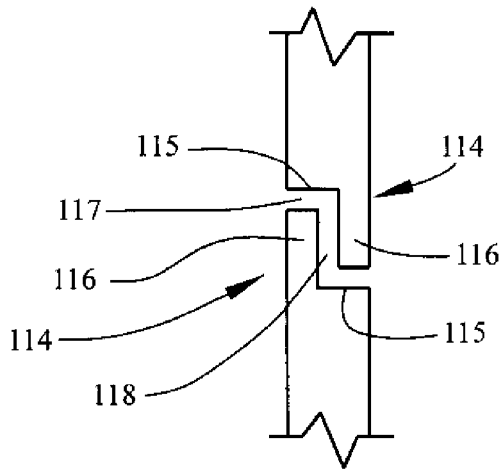


Fig. 35

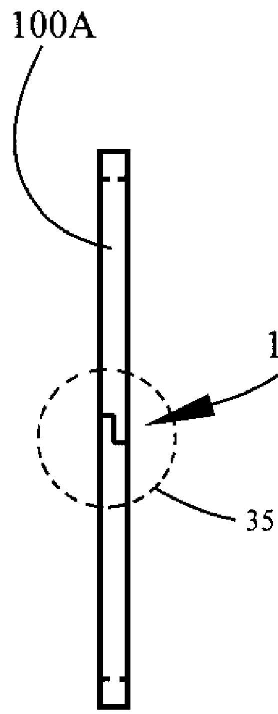


Fig. 33

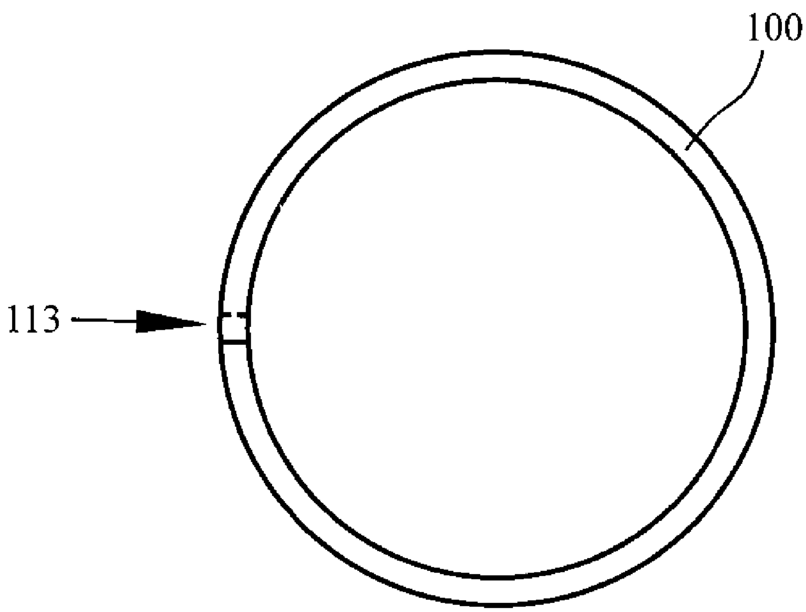


Fig. 34

SLIDING VALVE ASPIRATION

CROSS REFERENCE TO RELATED APPLICATIONS

This utility patent application is a Continuation in Part of Ser. No. 13/443,077, Filed Apr. 10, 2012, entitled "Sliding Valve Aspiration," by inventor Gary W. Cotton, which was a divisional application based upon prior U.S. utility patent application Ser. No. 12/387,184, filed Apr. 29, 2009, Entitled "Sliding Valve Aspiration System," by inventor Gary W. Cotton, now U.S. Pat. No. 8,210,147 issued Jul. 3, 2012, which was based upon a prior U.S. Provisional application entitled "Sliding Valve Aspiration Engine," Ser. No. 61/135,267, filed Jul. 18, 2008, by inventor Gary W. Cotton.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates generally to sleeve valve systems for aspirating internal combustion engines, and to internal combustion engines with tubular sliding valves for enhanced aspiration. More particularly, the present invention relates to reciprocating sleeve valve systems for engines equipped therewith of the general type classified in United States Patent Class 123, Subclasses 84, 188.4, and 188.5.

2. Description of the Related Art

A variety of aspiration schemes are recognized in the internal combustion motor arts. In a typical four-cycle firing sequence, gases are first inputted and then withdrawn from the combustion chamber of each cylinder interior during reciprocating piston movements caused by the crankshaft. Gas pathways must be opened and closed during a typical cycle. During the intake stroke, for example, an air/fuel mixture is suctioned through an open intake passageway into the combustion chamber as the piston is drawn downwardly within the cylinder. The intake passageway is typically opened and closed by some form of reciprocating valve mechanism that is ultimately driven by mechanical interconnection to the crankshaft. The combustion chamber must be sealed during the following compression and power strokes, and the valve mechanisms must be closed to block the ports. During the following exhaust stroke, exhaust ports must be opened to discharge spent gases from the combustion chamber.

Spring-biased poppet valves are the most common form of internal combustion engine valve. Typically, poppet valves associated with the intake and exhaust passageways are seated within the cylinder head above the combustion chamber proximate the cylinder and piston. Typical reciprocating poppet valves are spring biased, assuming a normally closed position when not deflected. In a typical arrangement, the bias spring coaxially surrounds the valve stem to maintain the integral valve within the matingly-configured valve seat. Poppet valves are typically opened by mechanical deflection from valve train apparatus driven by camshafts. Typical overhead-valve motor designs include rocker arms comprising reciprocating levers driven by push rods in contact with camshaft lobes. When the camshaft lobe deflects a pushrod to raise one end of the rocker arm, the opposite arm end pivots downwardly and opens the valve. When the camshaft rotates further, the rocker arm relaxes and spring pressure closes the valve. With overhead-cam designs camshafts are disposed over the valves above the head, and valve deflection is accomplished without push rods or rocker arms. Overhead camshafts push directly on the valve stem through cam followers or tappets. Some V-configured engines use twin overhead camshafts, one for each head. Some enhanced DOHC designs use two camshafts in each head, one for the intake valves and one for the exhaust valves. The camshafts are driven by the crankshaft through gears, chains, or belts.

Despite the overwhelming commercial success of poppet-valve designs, there are numerous deficiencies and disadvantages associated with poppet valves. Although poppet valve designs provide manufacturing advantages and cost savings, substantial spring pressure must be repeatedly overcome to properly open the valves. Spring pressure results in considerable drag and friction which increases fuel consumption and limits engine RPM. Poppet valve heads are left within the fluid flow passageway, despite camshaft deflection, and the resulting obstruction in the gas flow pathway promotes inefficiency. For example, back pressure is increased by the valve mass obstructing fluid flow, which contributes to turbulence. Poppet valves are exposed to high combustion chamber temperatures, particularly during the exhaust stroke, that can promote deformation and wear. Thermal expansion of exhaust valves, for example, can interfere with proper valve seating and subsequent sealing, which can decrease combustion performance.

Many of these disadvantages are amplified in high-horsepower or "high R.P.M." applications. Valve deflection in high power applications is often extreme, increasing the amplitude of valve deflection or travel. Damaging valve-to-piston contact can result. As a means of attenuating the latter factor, some pistons are designed with valve clearance regions, but these piston surface irregularities can deleteriously affect the combustion charge and fluid flow through the combustion chamber. Another problem is that the applied drive forces experienced by the valves are asymmetric. The extreme forcing pressure applied by the camshaft to open the valves, for example, is not as uniform as the spring closing pressure. Disharmony between the opening and closing forces contributes to valve lash and concomitant timing problems that interfere with power generation and limit engine R.P.M. Of course, in high power systems involving four or more valves per cylinder, the problems and disadvantages with poppet valve engines are increased proportionally.

So-called "rotary valves" have been proposed for replacing reciprocal poppet valves. Typical rotary valve designs include an elongated tube or cylinder machined with a plurality of gas flow passageways that admit or pass gases. The rotary valves are not reciprocated; they are rotated about their axis to expose passages defined in them in directions normal to their longitudinal axis. Rotary valves must be timed properly to dynamically align their internal passageways with the fluid flow paths of the engine during operation. When rotated to a closing position, the rotary valve passageways are radially displaced, obstructing the normal flow pathways and sealing the engine for firing or compression strokes.

One advantage espoused by rotary valve proponents is the relative simplicity of the design. Further, rotary valves do not penetrate or extend into the cylinder, avoiding potential mechanical contact with the piston, and minimizing fluid flow obstructions. However, the biggest problem with rotary valves relates to ineffective sealing. Although much activity and research has been directed to rotary valve sealing designs, commercially feasible systems have not been perfected. Rotary systems provide inefficient cylinder sealing, lessening firing efficiency, and reducing compression pressure because of leakage. Further, rapid wear of such systems increases the aforementioned problems.

Sliding valves of many configurations are also known in the art. Typical slide valves may be hollow and tubular, or planar, or cylindrical. They are reciprocated within a tubular valve seat region proximate the combustion chamber to alternately open and then close the intake and exhaust passageways. Like rotary valves, sliding valve designs have hitherto been difficult to seal effectively, with predictable negative results.

U.S. Pat. No. 2,080,126 issued May 11, 1937 to Gibson shows a sliding valve arrangement involving a tubular valve driven by a secondary crankshaft. Its reciprocating axis is parallel to the axis of piston deflection. Ports arranged at the side of the piston are alternately opened and closed by piston movements, and gases are conducted through and around portions of the piston exterior.

A similar arrangement is seen in U.S. Pat. No. 1,995,307 issued Mar. 26, 1935, and U.S. Pat. No. 2,201,292, issued May 21, 1940, both to Hickey. The latter patents show designs that aspirate a single working cylinder with a pair of tubular, reciprocating valves that are mounted on either side of the piston and driven by secondary crankshafts. The aspirating valves are forcibly reciprocated between port blocking and port aligning positions. The valves are aligned at an angle slightly off of parallel with the axis of the cylinder.

Other examples of engines with tubular, reciprocating slide valves that move in a direction generally parallel with the drive piston axis are provided by U.S. Pat. Nos. 1,069,794; 1,142,949; 1,777,792; 1,794,256; 1,855,634; 1,856,348; 1,890,976; 1,905,140; 1,942,648; 2,160,000; and 2,164,522 that are largely cumulative.

Hickey U.S. Pat. No. 2,302,442 issued Nov. 17, 1942 shows a tubular, reciprocating sliding valve disposed atop a piston head. The valve slides in an axis generally perpendicular to the axis of the lower drive piston.

U.S. Pat. No. 5,694,890 issued to Yazdi on Dec. 9, 1997 and entitled "Internal Combustion Engine With Sliding Valves" discloses an internal combustion engine aspirated by slidable valves. Tapered, horizontally disposed valve seats are defined near inlet and exhaust ports at the top of the combustion chambers. The slidable valves are tapered to conform to the valve seats. Valve movement is caused by a crankshaft driving a rocker arm that is oriented substantially orthogonal to the rod, whereby crankshaft rotation is translated into horizontal, sliding movements of the planar valves, which reciprocate in a direction normal or transverse to the axis of the piston.

U.S. Pat. No. 7,263,963 issued to Price on Sep. 4, 2007 and entitled "Valve Apparatus For An Internal Combustion Engine" discloses a cylinder head with a cam-driven valve slidably disposed within a valve pocket. The valve, which is displaceable along its longitudinal axis has a tapered portion defining multiple fluid flow passageways. The valve is displaced by cam rotation between a configurations passing gases through the passageways and a configuration wherein the valve flow passageways are closed.

BRIEF SUMMARY OF THE INVENTION

This invention provides an improved sliding valve system for aspirating internal combustion engines, and engines equipped therewith. The system employs tubular, reciprocating sliding valves disposed within sleeves defined within the head secured above the motor's reciprocating pistons. The valves are driven by an independent crankshaft that is exteriorly driven through a pulley.

The sliding valves are positioned within suitable exhaust and intake tunnels in the head. Preferably, sleeves are concentrically disposed around the valves and concentrically fitted within the tunnels. Fluid flow through the valves results through ports defined in the body of the tubular slide valves that are aligned with similar ports in their sleeve, that are in turn aligned with ports dynamically positioned above the compression or combustion region of the cylinder located below the head. Gas pressure develops shearing forces on valve sides. Gases are routed through the tubular interior of the sliding intake valve or valves during intake strokes, and exhaust gases are likewise forced out of the combustion cylinder through the interior of the exhaust valve or valves during exhaust strokes. Pressured gases traveling longitudinally through the valve interior passageways are inputted or outputted through lateral valve ports in fluid flow communication with the internal valve passageways. High pressure gas is confined between axially spaced apart sealing rings that prevent gases from flowing axially about the valve exterior.

All intake and exhaust gas flow is thus confined within the tubular interior of the sliding valves. As a result, gas pressure does not develop a substantial resistive force upon leading surfaces of the valve in a direction coincident with the direction of valve travel. Instead gas pressure that might otherwise resist valve travel, and add to friction, is applied as a shear force, and pressure is evenly distributed in the relief annulus. Gas flow is distributed through the valve interior rather than around it, and friction is substantially reduced.

Importantly, the port sizes are maximized for efficient breathing. However, in the past, large sliding valve ports have contributed to inefficiency, reduced sealing, and premature valve failure. In the present design, the slide-valve sleeves are provided with a unique connecting bridge that traverses the port area, aligned with the direction of sliding valve travel. When the valves slidably reciprocate through this region, their sealing rings are supported tangentially by the bridges, to maintain ring integrity. Importantly, the present design includes an oiling section on the sliding valves with an additional sealing ring.

Thus a basic object of my invention is to provide a highly efficient, sliding valve aspiration system for internal combustion engines, particularly four-cycle designs.

A related object is to provide an improved sliding valve that is ideally employed with four cycle, internal combustion engine.

A related object is to improve combustion efficiency within an internal combustion engine. It is a feature of our invention that its advantageous overhead valve geometry and the reduction of valve-train parts needed for the invention increase overall efficiency.

Another important object is to preserve the sealing integrity of sliding valves. One important feature of the invention in this regard is that the head ports are provided with bridges that support the valve sealing rings during motion. Another important feature is the addition of a fourth sealing ring proximate a separate oiling section.

Another basic object is to provide a valve system for internal combustion engines that provides an enhanced power stroke. In other words, it is a feature of this invention that a higher proportion of the total 720 degrees of crankshaft rotation during typical four cycle operation occurs during the power stroke.

Another important object is to provide a sliding valve system of the character described that does not affect combustion chamber volume during operation. Important features of my invention are the fact that chamber expansion during valve displacement is avoided, and that the porting path does not consume the operational compression volume.

A related object is to provide a valve system of the character described wherein the valve structure does not enter the combustion chambers.

Another object is to provide a valve deflection system that applies force symmetrically, to minimize valve lash and allow higher engine speeds.

Yet another basic object is to minimize friction. It is a feature of my invention that spring-biased poppet valves and the typical frictional cam shafts and associate linkages such as rocker arms used to reciprocate poppet valves are avoided.

A still further object is to provide a valve system of the character described that is driven externally by a belt, so that efficiency is increased and complexity is reduced.

Another important object is to avoid so-called "split-lift" applications used in the prior art for aspirating motors.

These and other objects and advantages of the present invention, along with features of novelty appurtenant thereto, will appear or become apparent in the course of the following descriptive sections.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

In the following drawings, which form a part of the specification and which are to be construed in conjunction therewith, and in which like reference numerals have been employed throughout wherever possible to indicate like parts in the various views:

FIG. **1** is a fragmentary, isometric view of a one-cylinder, internal combustion engine constructed in accordance with the best mode of the invention known at this time;

FIG. **2** is an enlarged, fragmentary, plan view of the engine taken generally from a position to the right of FIG. **1**, with portions thereof broken away or shown in section for clarity;

FIG. **3** is an enlarged, fragmentary sectional view taken generally along line **3-3** of FIG. **2**;

FIG. **4** is an enlarged, fragmentary, isometric view of the preferred cylinder head assembly, with portions thereof broken away or shown in section for clarity or omitted for brevity;

FIG. **5** is an enlarged, partially exploded fragmentary isometric view of the cylinder head assembly of FIG. **4**, with my new sliding valve removed from its sleeve, and with portions thereof broken away or shown in section for clarity;

FIG. **6** is an enlarged, fragmentary isometric view taken generally from circled region "6" in FIG. **5**;

FIG. **7** is an enlarged bottom isometric view of the preferred cylinder head;

FIG. **8** is an enlarged isometric view of my new valve, with portions thereof broken away or shown in section for clarity;

FIG. **9** is a side elevational view of my new sliding valve;

FIG. **10** is an end elevational view of the valve of FIG. **9**, looking generally in the direction of arrows **10-10**;

FIG. **10**A is a longitudinal sectional view of a preferred sliding valve, derived generally in the direction of arrows **10**A-**10**A in FIG. **10**;

FIG. **11** is an enlarged top plan view of the preferred cylinder head, with phantom lines illustrating various internal parts, and with portions broken away or shown in section for clarity;

FIG. **12** is an enlarged, fragmentary diagrammatic view showing the basic arrangement of the engine power cylinder, the head, the overhead exhaust valve, and the exhaust valve sleeve;

FIGS. **13-15** are diagrammatic views of progressive intake sliding valve movements during the intake stroke as the power crankshaft rotates;

FIG. **16** is a diagrammatic view showing the intake valve position when the spark plug fires at the beginning of the power stroke;

FIG. **17** is a diagrammatic view showing the intake valve position at the bottom of the power stroke;

FIG. **18** is a diagrammatic view showing the intake valve position at the end of the exhaust stroke;

FIG. **19** is a diagrammatic view showing the exhaust valve position at the start of the exhaust stroke;

FIG. **20** is a diagrammatic view showing the fully open exhaust valve position at 251 degrees of engine crankshaft angle;

FIG. **21** is a diagrammatic view showing the closing exhaust valve at the beginning of the intake stroke at 222 degrees of crankshaft angle;

FIG. **22** is a diagrammatic view showing the fully closed exhaust valve at the bottom of the intake stroke at 180 degrees of crankshaft angle;

FIG. **23** is a diagrammatic view showing the closed exhaust valve 90 degrees into the compression stroke;

FIG. **24** is a diagrammatic view showing the closed exhaust valve at zero degrees TDC;

FIG. **25** is a longitudinal diagrammatic view of the preferred secondary crankshaft that operates the intake and exhaust valves and moves them between positions illustrated in FIGS. **13-24**;

FIGS. **26-28** are sectional views taken respectively along lines **26-26**, **27-27**, and **28-28** of FIG. **25**;

FIG. **29** is an isometric view of a preferred valve sleeve, with portions broken away for clarity;

FIG. **30** is a bottom plan view of the sleeve of FIG. **29**;

FIG. **31** is a side elevational view of the sleeve of FIG. **29**;

FIG. **32** is an end elevational view of the sleeve of FIG. **29**;

FIG. **33** is an enlarged, side elevational view of a preferred sealing ring used with the sliding valves;

FIG. **34** is an enlarged, plan view of a preferred sealing ring used with the sliding valves; and,

FIG. **35** is an enlarged, fragmentary plan view of circled region **35** in FIG. **33**.

DETAILED DESCRIPTION OF THE INVENTION

For purposes of providing an enabling disclosure, prior U.S. patent application Ser. No. 13/443,077, Filed Apr. 10, 2012, entitled Sliding Valve Aspiration, by inventor Gary W. Cotton, and U.S. Pat. No. 8,210,147 issued Jul. 3, 2012, Entitled "Sliding Valve Aspiration System," by inventor Gary W. Cotton, are hereby incorporated by reference as if fully set forth herein.

With initial reference directed to FIGS. **1-5** of the appended drawings, a basic single-cylinder, four-cycle internal combustion engine equipped with the aspiration system constructed in accordance with the best mode of the invention has been generally designated by the reference numeral **10**. It should be understood that the aspiration system as herein described is suitable for use with engines equipped with multiple cylinders, arrayed in the popular V-configuration or other configurations. The engine **10** has a rigid block **11** housing a primary crankshaft **12** (FIG. **3**) of conventional construction that drives a reciprocating power piston **14** (FIG. **3**) with a conventional connecting rod **16**. The basic engine illustrated comprises a Honda thirteen-horsepower motor, which is modified as hereinafter described. The engine configuration as illustrated can be varied considerably according to recognized standards known to those with skill in the art.

The standard combustion power piston **14** reciprocates within a cylinder **18** (FIG. **3**) that is externally air-cooled with multiple, external heat dissipation fins **20** (FIG. **1**) proximate the engine deck **13** (FIG. **1**). The basic construction of piston **14** and its accessories is substantially conventional and is not critical to practice of the invention. The instant sliding valve system is disposed within a head, generally indicated by the reference numeral **22** (i.e., FIGS. **3-5**, **7**, **11**), that mounts conventionally above the engine deck **13** above the conventional piston **14** and cylinder **18** described previously. Piston **14** moves it upwardly and downwardly in a direction substantially perpendicular to head **11**. For purposes of this invention, the term "head" shall generally designate that region of an internal combustion engine enclosing the combustion chambers, above the pistons. Such a head may be a conventional, separate part bolted atop the engine, or in some cases the "head" may be integral with the engine block in a single casting that is thereafter appropriately machined.

With additional reference directed primarily now to FIGS. **4-11**, head **22** houses a pair of tubular, slide valves **24**, **25** (FIGS. **4**, **5** **8-11**) that aspirate the cylinder **18**. In the best mode known at this time, the tubular exhaust valve **24** and the tubular intake valve **25** are made from titanium. While those skilled in the art will recognize that several alloys of titanium and/or titanium steel are available, my experiments have yet to reveal the ideal composition of these critical valves. Ordinary steel compositions however, result in heat damage and premature wear and failure. Furthermore, as illustrated in FIG. **5**, for example, the sliding valves **24**, **25** are mounted in appropriately ported sleeves **27** that fit into the cylinder head **22** and line up and register with the appropriate ports in the head. While sleeveless sliding valve designs are functional, sleeves are much preferred. It is also preferred that the sleeves be coated by treating them with Nickel-boron.

A drive pulley **26** (FIG. **1**) driven by conventional internal crankshaft **12** (FIG. **3**) is connected via drive belt **28** to a valve pulley **30** that drives the slide valve crankshaft **32** housed within head **22**. Crankshaft **32**, best seen in FIG. **11**, is mounted perpendicularly relative to sliding valves **24**, **25**. It extends across and through compartmentalized crankshaft mounting region **34** (FIG. **5**) across the top (i.e., as viewed in FIGS. **4**, **5**) of the head **22**. Region **34** (FIG. **3**) contains liquid oil for lubricating the crankshaft and the slide valves to be described and it is normally covered by shroud **35**. The crankshaft exhaust journal **38** and the crankshaft intake valve journal **40** (i.e., FIG. **25**) of crankshaft **32** support connecting rods **42**, **44** that respectively operate exhaust slide valve **24**, and intake slide valve **25**. Aligned and integral crankshaft portions **39**, **41**, **43** (i.e., FIG. **25**) are rotatably constrained within conventional saddles **45** within mounting region **34** (i.e. FIG. **4**, **5**) and mounted with conventional bearing assemblies **46** (FIG. **2**) as known in the art. In the best mode it is proposed that the counterweight sections **109**, **110**, **111**, and **112** of the crankshaft (FIG. **25**) be drilled appropriately for crankshaft balancing. Preferably the rotating and reciprocating aspiration slide valve assembly may thus be "balanced" and "tuned" for optimal aspiration performance.

The crankshaft bearing assemblies **46** are bolted within crankshaft region **34** to mount the slide valve crankshaft **32** over the saddles **45** are secured with a plurality of bolts **48**. As best seen in FIGS. **4**, **5** and **7**, head **22** includes a plurality of spaced apart mounting orifices **50** through which head bolts **52** (FIG. **11**) extend when mounting the head **22** to the deck **13**.

The intake sliding valve **25** (i.e., FIG. **11**) is slidably received within a sleeve **27**B disposed within head tunnel **55** (FIGS. **4**, **11**), that is spaced apart from and parallel with exhaust tunnel **54** and sleeve **27**. Tunnels **54** and **55** are oriented generally perpendicularly to the stroke of the power piston **14**. Exhaust sliding valve **24** slidably reciprocates within sleeve **27** concentrically disposed within tunnel **54**. Sleeves **27**, **27**B (FIGS. **5**, **29-32**) require ports aligned with head ports and valve described hereinafter, as appreciated by those skilled in the art. An air-fuel mixture is drawn into intake valve tunnel **55** from a conventional carburetor **29** (FIG. **2**) mounted with screws received within orifices **59** (FIG. **4**). Alternatively the invention may be used with fuel injection systems.

As best viewed in FIGS. **29-32**, each sleeve **27** is elongated and tubular. Each has a pair of spaced apart open ends **31** defining opposite ends of an elongated cylindrical passageway in which the sliding valves **24** and/or **25** are inserted. A pair of ports **68**A are separated by a bridge **69**A (FIG. **29**) that maintains pressure on the sliding valve rings during operation. While both sleeves are identical in dimensions and geometry, the exhaust sleeve should be of a more expensive heat resistant alloy. It is preferred that the exhaust sleeve be made of Steelite or Nickalloy heat resistant titanium steel alloy.

This invention requires maximal air flow quickly. In other words, it is preferred that the carburetor **29** have a relatively large throat with a relatively short venturi. In the model depicted in the drawings, which has been thoroughly tested, a Honda 350 cc. "dirt bike" motorcycle carburetor is preferred.

Exhaust valve **24** is slidably constrained within its sleeve **27** in tubular tunnel **54** (FIGS. **5**, **7**, **11**). The exhaust header **57** (FIG. **1**) is preferably screw-mounted upon the head's end surface **58** (FIGS. **4**, **7**) with suitable screws that penetrate orifices **60** (FIG. **7**). Head cooling is encouraged by fin areas **36** (FIGS. **5**, **7**).

As best seen in FIG. **7**, the circular combustion chamber **62** includes a central, threaded spark plug passageway **64** that is spaced between intake ports, collectively numbered **66**, and exhaust ports, collectively numbered **68** (FIG. **7**). A conventional spark plug **70** (i.e., FIGS. **1**, **11**) is threadably mated to passageway **64**, with its electrodes positioned and centered within combustion chamber **62**.

As seen in FIGS. **29-30**, for example, adjacent sleeve ports **68**A are separated from one another by a central bridge **69**A. Similarly intake ports **66** in the head (FIG. **7**) built into the combustion chamber may be separated with a bridge **67** that is integral with the head **22**. Similarly, a rigid, centered bridge **69** in the head separates the twin exhaust ports **68** (FIGS. **6**, **7**). These ports in the head must align with the valve sleeve ports **68**A seen in FIGS. **29-32**.

As best seen in FIG. **6**, each head exhaust port **68** aligns with sleeve port **68**A. The composite ports have smooth, downwardly inclined sidewalls **74**, **75** that are polished for maximal fluid flow. These walls communicate with a lower orifice **73** in the head that opens to the combustion chamber **62**. The intake ports **66** (i.e., FIG. **7**) are similarly configured. Importantly, it is desired that corner ridges of the structure be radiused for maximum fluid flow, as illustrated by gently radiused corner regions

Importantly, rigid, transverse bridges **69**A are integrally formed in the sleeve port regions and bisect these regions into twin, side by side orifices **68**A (FIG. **29**). The head is similarly ported. In FIG. **7**, for example, there are two pairs of ports **66** and **68** respectively separated by bridges **67**, **69**. Sleeve **69**A bears against critical sealing rings associated with the sliding valves **24** and **25**, as discussed below. By pressuring the sealing rings during valve travel, deformation of the critical sealing rings in the region of the various exhaust ports **68** and intake ports **66** is prevented. As sealing of the tubular slide valves **24**, **25** is critical to the invention, bridges **67** and **69** are vital to the best mode of the invention.

With joint reference directed now primarily to FIGS. **8-12** and **10**A, valves **24** and **25** are structurally virtually identical, so only exhaust valve **24** will be detailed. However, the exhaust valve **24** runs at higher temperatures, and thus requires more heat resistance, so it is preferably fabricated from a premium grade of titanium alloy steel.

With emphasis directed to FIGS. **8**-**10** and **10**A, each slide valve **24**, **25** is elongated, substantially tubular, and multi-sectioned. An open connecting rod section **80** (i.e., FIG. **10**A) enables mechanical connection to the connecting rod **42** (FIG. **12**). The end of rod **42** extends into the interior **82** of section **80** and is journalled by wrist pin **85** (FIG. **3**) and is conventionally secured between wrist pin orifices **84** (FIGS. **9**, **10**A). Importantly, valve section **80** ends in a closed interior wall **87** that separates interior region **82** and the connecting rod structure from the rest of the tubular interior passageway **89** (FIG. **10**A) of the valve **24**. The open end of the interior passageway **89** within each valve directly communicates through tubular tunnels **54**, or **55** (FIG. **4**) for aspiration fluid flow. The exterior surface **81** of valve rod section **80** (FIGS. **9**, **10**A) is preferably cross hatched by machining to promote oil flow and distribution.

In the best mode each valve has four pairs of external ring grooves to seat suitable sealing rings. For example, a pair of concentric and parallel ring grooves **91** separate valve rod section **80** from the adjacent port section **94** (FIGS. **9**, **10**A). Ring grooves **92** separate port section **94** from the adjacent "power stroke" midsection **96**. Similarly, ring grooves **93** separate midsection **96** from adjacent oiling section **98**. Finally, a fourth set of ring grooves **97** separates oiling section **98** from terminal open section **99**. Establishment of the separate oiling section **98** aids in lubrication and sealing, and cooling effects.

A comparison of FIGS. **8** and **9** reveals that ring groove pairs **91**, **92**, **93**, and **97** seat pairs of spaced apart, concentric sealing rings **100**A, **100**B, **100**C, and **100**D respectively, that are externally, coaxially, mounted about each valve exterior. Valve rod sections **80** and oiling section **98** are in fluid flow communication with head region **34** that contains lubricating oil. Thus rings **100**A and **100**D are oil rings. Port **95** (FIG. **4**) delivers a mist of oil to the preferably reduced diameter valve oiling section **98**. It will be recognized by those skilled in the art that when the valves **24** or **25** are fitted within their sleeves **27**, (i.e., FIG. **5**) the rings **100**A, **100**B, **100**C or **100**D will substantially. flushly seat within the respective ring grooves **91**, **92**, **93** and/or **97** (i.e., FIG. **9**) and the exterior of the rings will be flush with the cylindrical outside body of the valves **24**, **25**, slidably touching the interior surfaces of the captivating sleeves **27**.

Each sealing ring **100**A, **100**B, **100**C, and **100**D is preferably made of heat treated and heat resistant nickel alloy steel. As best seen in FIGS. **33**-**35**, the compressively touching ends of the rings are stepped in the best mode to form an overlapped intersection **113** that forms an improved pressure seal. Preferably, each end of a given ring is configured in the overlapping or stepped configuration of FIG. **35**, where abutting ring ends comprise a notched region **115** and a bordering, elongated tabbed region **116**. The tabbed regions **116** are variably spaced apart from notched regions **115**, with end gaps **117** therebetween. The parallel, spaced apart ring end gaps **117** allow for thermal expansion and contraction of the rings during operation. However, a sealing gap **118**, which is perpendicular to gaps **117**, is defined between mutually aligned and abutting tabbed regions **116**. Gap **118** is much smaller than indicated, and provides a seal, as end regions **116** abut in operation, and seal the gaps for compression. At the same time gaps **117** allow for normal thermal expansion and contraction.

Importantly, the valve port section **94** (FIGS. **8**, **9**) includes an enlarged, arcuate cutout **102** functioning as an aspiration port (i.e., either exhaust or intake). Port **102** radially extends about approximately 30-40 percent of the radial periphery of the valve. A gently radiused arch **103** above port **102** (FIGS. **8**, **10**A) leads to the smoothly configured, generally cylindrical passageway **89** that leads to the exterior of the valve. Passageway **89** (FIG. **10**A) comprises tubular interior passageway walls **104**, terminating in gently radiused, flared lips **106** (FIG. **10**A) at the valve end that maximize fluid flow. Aspiration occurs when valve ports **102** are aligned with sleeve ports **68**A (FIG. **32**) which are in turn aligned with head port pairs **66** or **68** (FIG. **7**), in response to timed, reciprocal movements caused by the valve crankshaft **32** previously described. Thus when port **102** (FIGS. **3**, **9**) of the exhaust valve **24** overlies sleeve ports **68**A (FIG. **32**) and head ports **68** (FIG. **7**), hot exhaust gases may be vented away from the combustion chamber **62** and lower cylinder **18** in response to upward movement of the power piston **14** towards top-dead-center. At this time exhaust gases are vented to the left (as viewed in FIG. **9**) through port **102**, along the valve interior passageway **89** (FIG. **8**) and through head tunnel **54** (FIG. **7**) and out header **57** (FIGS. **1**, **3**). Similarly, during the intake stroke, air and raw fuel is drawn through carburetor **29** into the head **22** through tunnel **55** (FIG. **7**), and into the passageway **89** in the intake valve **25**, through its port **102** and into the cylinder combustion region through head ports **66** (FIG. **7**) and aligned sleeve ports **68**A.

Importantly, as slide valves **24**, **25** reciprocate, their multiple sealing rings **100** are prevented from deformation while traversing sleeve ports **68**A by the bridges **69**A (i.e., FIG. **32**).

Referencing FIG. **9**, the arrow **105** indicates the outside diameter of the majority of the length of valve **24**. Sections **80**, **94**, **96** and **99** are all of this diameter. Valve oiling section **98** however, has a slightly reduced diameter in the best mode. Thus a cylindrical or annular region **101** (FIG. **3**) is defined radially around the external periphery of valve oiling section **98**.

Operation:

In FIG. **13** intake valve **25** has started to open at the beginning of the intake stroke. In FIG. **14** the intake valve **25** is now open at approximately 108 degrees BTDC.

FIG. **15** shows the intake valve **25** closing at the end of the intake stroke. Full closure of valve **25** is indicated in FIG. **16** at the beginning of the power stroke.

FIG. **17** shows the bottom of the power stroke, with the intake valve **25** fully closed. In FIG. **18** at the end of the exhaust stroke the intake valve **25** is seen starting to open.

The exhaust valve **24** is seen in FIG. **19** at the start of the exhaust stroke. In FIG. **19**, the plug and cylinder have fired, and at 108 degrees ATDC the exhaust valve **24** starts to open. In FIG. **20** the exhaust valve **24** is completely open, with 251 degrees crankshaft angle.

At the beginning of the intake stroke in FIG. **21** the exhaust valve **24** begins to close, at approximately 222 degrees. The bottom of the intake stroke is seen in FIG. **22**, at which time the exhaust valve **24** is fully "closed," and the power stroke midsection **96** is positioned over the exhaust ports **68**.

In FIG. **23** the exhaust valve **24** is completely open, 90 degrees into the compression stroke. In the positions of FIG. **24** the plug fires, and the exhaust valve **24** is completely closed at zero degrees TDC.

In FIGS. **25**-**28** the configuration and position of the crankshaft **32** is illustrated. The exhaust valve journal **40** and the intake journal **38** are seen in critical rotational positions.

Example:

Dyno Test Chart-December 2008		
	FACTORY ENGINE	G1 ENGINE
LOW LOAD		
Load %	33%	33%
RPM	2900	2900
Run Time	1:30 minutes	1:30 minutes
lb-ft Torque	7.5	7.5
Brake Horsepower	4.1	4.1
Fuel Usage - Milliliters	12.07	10.86
Nitrogen Oxide—NOX	10.97	10.97
Carbon Monoxide—CO	0.95	1.07
Hydrocarbons—HC	21.9	2.39
Carbon Dioxide—CO2	2.1	2
Oxygen—O2	1.41	1.43
G1 FUEL USAGE RESULTS PER UNIT OF BRAKE HORSEPOWER		
Low Load Fuel Usage: 10% less than Factory Engine (12.07 − 10.86 = 1.21/12.07)		
HIGH LOAD		
Load %	80%	80%
RPM	3550	3550
Run Time	1:30 minutes	1:30 minutes
lb-ft Torque	10	14
Brake Horsepower	6.7	9.4
Fuel Usage - Milliliters	13.19	8.65
Nitrogen Oxide—NOX	5.97	8.65
Carbon Monoxide—CO	0.58	0.44
Hydrocarbons—HC	11.04	1.07
Carbon Dioxide—CO2	1.29	0.8
Oxygen—O2	1.34	0.67
G1 FUEL USAGE RESULTS PER UNIT OF BRAKE HORSEPOWER		
High Load Fuel Usage: 34.4% less than Factory Engine 13.19 − 8.65 = 4.54/13.19)		
G1 HIGH LOAD EMISSION RESULTS PER UNIT OF BRAKE HORSEPOWER		
NOX: 23.4% less than Factory Engine	HC: 90.3% less than Factory Engine	
CO: 24.1% less than Factory Engine	CO2: 37.9% less than Factory Engine	

Two GX 390 Honda 13 hp engines were used for testing and comparisons (i.e., a "stock" engine versus one modified in accordance with the instant invention). Both engine specifications were as follows:

Four stroke valve single cylinder
3.5×2.5 bore & stroke
4.412 rod length
Forced air cooling systems
Gravity feed fuel systems
87 octane gasoline
23.7 cu/in displacement
Transistorized magnet ignition systems

The muffler was removed on both engines to confine exhaust emissions for analysis purposes. The engine with the stock head is named the "Factory" engine on the above chart. The engine with our proprietary head is named the "G1" on the above chart.

All tests were conducted on the same day in a controlled and isolated environment. Fuel and emission measurements were made using the following equipment:

Land & Sea Water Brake Dyno, the Dyno-Max 2000 Model
Dyno-Max 2000 Data Analysis Software and Multimedia PC Demonstration, 9.38 SPI Version
UEI AGA 5000 Emissions Analyzer
ASTME rated ⅜ inch Bellwether 100 cc Tube

The primary objective of house testing was to determine the fuel usage of the modified engine. We kept run time, load and rpm constant. To compare and measure the efficiency, input was divided by output. In our particular case, fuel usage was our input variable and our output variable was the pound-foot of torque produced. Fuel usage and all emissions results of both engines were calculated based on a unit of brake horsepower (torque×rpm/5252).

The low load fuel usage per unit of brake horsepower for the G1 engine was 10% less than the Factory engine. The high load fuel usage per unit of brake horsepower for the G1 engine above. It was determined that fuel consumption of the modified engine G1 was 34.4% less than the Factory engine. The high load emissions per unit of brake horsepower for the G1 engine resulted in 23.4% less nitrogen oxide (NOX), 24.1% less carbon monoxide (CO), 90.3% less hydrocarbons (HC) and 37.9% less carbon dioxide (CO2) compared to the Factory engine.

From the foregoing, it will be seen that this invention is one well adapted to obtain all the ends and objects herein set forth, together with other advantages which are inherent to the structure.

It will be understood that certain features and subcombinations are of utility and may be employed without reference to other features and subcombinations.

As many possible embodiments may be made of the invention without departing from the scope thereof, it is to be understood that all matter herein set forth or shown in the accompanying drawings is to be interpreted as illustrative and not in a limiting sense.

What is claimed is:

1. A slide valve for aspirating internal combustion engines, the slide valve comprising:

a tubular body adapted to be slidably disposed within a tubular tunnel or sleeve, said body comprising at least one aspiration port and an elongated, internal tubular passageway in fluid flow communication with said aspiration port for intaking or exhausting gases;

an open connecting rod section with an interior enabling mechanical connection to a rod for reciprocating the slide valve;

a closed interior wall that separates the connecting rod section from the internal tubular passageway;

a port section proximate said closed wall in which said at least one aspiration port is defined, wherein an arcuate cutout defined in said port section functions as said aspiration port, the cutout contacting said closed wall and the cutout comprising at least one radiused arch;

a tubular power stroke midsection adjacent the port section;

a tubular oiling section adjacent the power stroke midsection;

a terminal open section adjacent said oiling section that is in fluid flow communication with said tubular passageway;

said elongated, internal tubular passageway extending coaxially longitudinally between said closed wall and a spaced-apart, open valve end, the open valve end comprising radiused lips;

at least one concentric ring groove externally separating the valve rod section from the port section;

at least one concentric ring groove externally separating the valve port section from the adjacent valve power stroke midsection;

at least one concentric ring groove externally separating the valve midsection from the adjacent oiling section;

at least one concentric ring groove externally separating the valve oiling section from the valve open section; and,

at least one sealing ring seated in said ring grooves.

2. The valve as defined in claim **1** wherein said connecting rod section interior is of a size substantially less than the diameter of said tubular passageway.

3. The valve as defined in claim **1** wherein each arcuate cutout radially extends between 30-40 percent around the radial periphery of the valve.

4. The valve as defined in claim **3** wherein the sealing rings are stepped for enhanced compression and comprise:

abutting ring ends with a notched region and a bordering tabbed region;

the tabbed regions variably spaced apart from said notched regions;

end gaps between the notched and tabbed regions compensating for thermal expansion and contraction; and,

wherein tabbed regions of abutting ring ends abut one another and laterally seal the ring ends.

5. A slide valve for aspirating internal combustion engines, the slide valve comprising:

a tubular body adapted to be slidably disposed within a tubular tunnel or sleeve, said body comprising at least one aspiration port and an elongated, internal tubular passageway in fluid flow communication with said aspiration port for intaking or exhausting gases;

an open connecting rod section with an interior enabling mechanical connection to a rod for reciprocating the slide valve;

a closed interior wall that separates the connecting rod section from the internal tubular passageway;

a port section proximate said closed wall in which said at least one aspiration port is defined, wherein an arcuate cutout defined in said port section functions as said aspiration port, the cutout contacting said closed wall and the cutout comprising at least one radiused arch;

a power stroke midsection adjacent the port section;

an oiling section adjacent the midsection;

an open section adjacent said oiling section that is in fluid flow communication with said tubular passageway;

said elongated, internal tubular passageway extending coaxially longitudinally between said closed wall and a spaced-apart open valve end at said open section, the open **31** valve end comprising radiused lips;

the oiling section having a diameter reduced from that of the diameters of the port 2 section or open section to distribute oil about the circumference of the valve;

at least one concentric ring groove externally separating the valve rod section from the port section;

at least one concentric ring groove externally separating the valve port section from the adjacent valve power stroke midsection;

at least one concentric ring groove externally separating the valve midsection from the valve oiling section;

at least one concentric ring groove externally separating the valve oiling section from the valve open section; and,

at least one sealing ring seated in each of said ring grooves.

6. The valve as defined in claim **5** wherein said connecting rod section interior is of a size substantially less than the diameter of said tubular passageway.

7. The valve as defined in claim **6** wherein each arcuate cutout radially extends between 30-40 percent around the radial periphery of the valve.

8. The valve as defined in claim **7** wherein the sealing rings are stepped for enhanced compression and comprise:

abutting ring ends with a notched region and a bordering tabbed region;

the tabbed regions variably spaced apart from said notched regions;

end gaps between the notched and tabbed regions compensating for thermal expansion and contraction; and,

wherein tabbed regions of abutting ring ends abut one another and laterally seal the ring ends.

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