

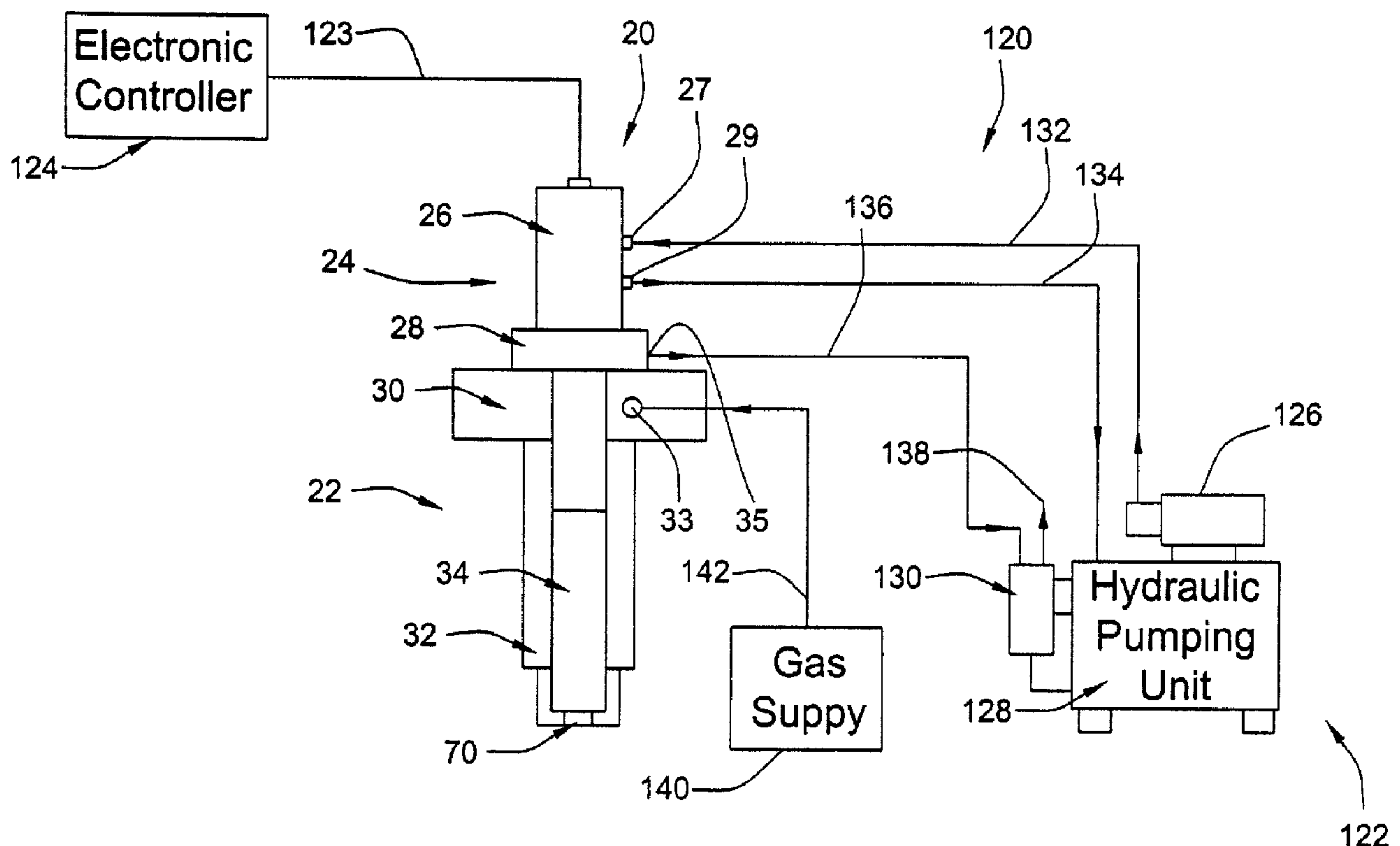
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(54) Titre : INJECTION DE COMBUSTIBLE GAZEUX HAUTE-PRESSION
(54) Title: HIGH-PRESSURE GASEOUS FUEL INJECTION



A high-pressure fuel injection system for injecting high-pressure gaseous fuel from a gaseous fuel supply into cylinders of an engine. The injection system comprises an oil reservoir of hydraulic oil, a hydraulic oil supply of hydraulic oil at a pressure higher than the oil reservoir, and a gas/oil separator located in a safe location remote from the cylinders of the engine. The gas/oil

(57) Abrégé(suite)/Abstract(continued):

separator is adapted to separate any gaseous fuel from the hydraulic oil, and return the separated oil to the oil reservoir. A plurality of high-pressure, fuel-injector valve assemblies inject fuel into the cylinders of the engine; each fuel-injector valve assembly including an electrohydraulic valve and a fuel injector. The electrohydraulic valve is connected to the high-pressure hydraulic oil supply, the oil reservoir and an electronic controller providing electrical control signals, and the electrohydraulic valve creates successive hydraulic signals in response to electrical control signals. The fuel injector includes a fuel valve operated by a piston, and an outlet port in communication with the gaseous fuel supply. The piston reciprocates the fuel valve in response to hydraulic signals to open and close the outlet port, and thereby successively injects gaseous fuel into the cylinder. Each fuel-injector valve assembly further includes means for collecting leakage of oil and leakage of gaseous fuel, and communicates any such leaked oil and leaked gas to the gas/oil separator.

ABSTRACT

A high-pressure fuel injection system for injecting high-pressure gaseous fuel from a gaseous fuel supply into cylinders of an engine. The injection system comprises an oil reservoir of hydraulic oil, a hydraulic oil supply of hydraulic oil at a pressure higher than the oil reservoir, and a gas/oil separator located in a safe location remote from the cylinders of the engine. The gas/oil separator is adapted to separate any gaseous fuel from the hydraulic oil, and return the separated oil to the oil reservoir. A plurality of high-pressure, fuel-injector valve assemblies inject fuel into the cylinders of the engine; each fuel-injector valve assembly including an electrohydraulic valve and a fuel injector. The electrohydraulic valve is connected to the high-pressure hydraulic oil supply, the oil reservoir and an electronic controller providing electrical control signals, and the electrohydraulic valve creates successive hydraulic signals in response to electrical control signals. The fuel injector includes a fuel valve operated by a piston, and an outlet port in communication with the gaseous fuel supply. The piston reciprocates the fuel valve in response to hydraulic signals to open and close the outlet port, and thereby successively injects gaseous fuel into the cylinder. Each fuel-injector valve assembly further includes means for collecting leakage of oil and leakage of gaseous fuel, and communicates any such leaked oil and leaked gas to the gas/oil separator.

HIGH-PRESSURE GASEOUS FUEL INJECTION

Field Of The Invention

5 The present invention generally relates to fuel injectors, and more particularly high-pressure gaseous fuel injection systems for internal combustion engines. This application is divided from and comprises the disclosure of Canadian Patent Application 2,330,226, filed January 04, 2001. Consequently the expression "the invention", and the like, as used herein is not restricted to subject
10 matter specifically claimed in this divided application, but may encompass additional subject matter described but not claimed.

Background Of The Invention

15 The natural gas transmission industry and chemical process industries use a large number of large-bore, 2-stroke and 4-stroke natural gas engines for compressing natural gas. For example, industries use these engines for such purposes as maintaining pressure in the extensive network of natural gas pipelines that supply residential housing and commercial businesses. The network of natural
20 gas pipelines typically operate at high pressures in the neighborhood of between 500 psig and 1000 psig.

 These large-bore, natural gas engines may be powered by a small portion of the natural gas passing through the pipelines. However, before being injected into the engine, the pressure of the gas is significantly and substantially reduced.
25 Gaseous fuel is typically injected into these cylinders at low pressures (for example, 15 psig to 60 psig by mechanically actuated fuel injectors, such as that disclosed in Fisher, United States Patent 4,365,756). The problem with low pressure injection is that the fuel pressure provides little kinetic energy with which to induce cylinder charge mixing. There is ample evidence that the fuel and air in
30 these large bore engines are not well mixed and as such exhibit poor combustion

stability, high misfire rates and significant cycle-to-cycle variations in peak pressure. As a result, these engines are not efficient and also are environmentally detrimental, contributing to approximately 10% of the total NO_x production in the United States from stationary combustion sources according to estimates.

5 The concept of using high pressure fuel delivery to enhance fuel mixing in these engines has been proposed as a means to improve efficiency and environmental emissions from these engines. However, retrofitting existing engines provides a significant hurdle because these engines are manufactured by different companies and also vary in size. Moreover, injecting fuel at high
10 pressure as opposed to low pressure requires the fuel injectors to operate under extremely high operating pressures which in turn greatly increases stresses and powering requirements for opening and closing the valves. A key requirement for any proposed high-pressure fuel injector is reliability. These large-bore, natural gas engines typically run continuously over long time periods, meaning that any
15 suitable fuel injector must be capable of reliably enduring very long operating cycles of the engine. It is desirable for example, that the fuel injectors reliably operate over several hundred million continuous cycles of the engine (about one to two years before replacement). As such, a valve must achieve reliability over this long time period or operating interval. Fuel injectors of the prior art such as that
20 disclosed in Fisher, U.S. Patent 4,365,756 are not capable of reliably sealing and accurately controlling the injection of gas at high pressure. Only recently have economic and environmental pressures on the gas industry resulted in justification for advances in fuel injection technology. For at least the foregoing reasons, commercial large bore 2-stroke and 4-stroke natural gas engines continue to be
25 fueled at low pressure by conventional low pressure fuel injectors.

Summary Of The Invention

It is a general aim of the present invention to provide commercially reliable and practical fuel injection systems for injecting high-pressure gaseous fuel (e.g. around 300-700 psi or more) into combustion engines.

5 In one aspect it provides a fuel injector that can withstand the forces of high-pressure gaseous fuel and has a long service operation but does not leak either gaseous fuel or hydraulic fluid to the external environment.

It also can provide a fuel injector that is universal in that the fuel injector assembly can be easily adapted without any or any substantial redesign to fit and
10 operate as desired on the various types and sizes of combustion engines in the industry.

Another aspect provides a fuel injection system with increased operating life, whereby gas leakage, eventually expected from o-rings and sliding gas seals, is captured and safely and properly disposed of, on an ongoing basis, not requiring
15 engine shut-down to replace the injector valve.

In accordance with this aspect of the invention a fuel injection system for injecting gaseous fuel into the cylinders of an engine comprises an oil reservoir of hydraulic oil, and a supply of hydraulic oil at a higher pressure than the oil reservoir. A gas/oil separator is adapted to separate gaseous fuel from the
20 hydraulic oil, and return the separated oil to the oil reservoir. A plurality of fuel-injector valve assemblies each include an electrohydraulic valve and a fuel injector; the electrohydraulic valve actuating the fuel-injector valve with hydraulic signals derived from the hydraulic oil supply to inject gaseous fuel from the gaseous fuel supply into at least one of the cylinders of the engine. The fuel-
25 injector valve assemblies have at least one collection chamber adapted to collect hydraulic oil and leaked gaseous fuel; the at least one collection chamber returning the hydraulic oil and leaked gaseous fuel to the gas/oil separator.

As mentioned, according to another aspect the invention is directed towards a novel fuel injector cartridge, a novel fuel injector that includes a tubular
30 cartridge housing and the fuel injector cartridge inserted therein, and an

electrohydraulic fuel injector assembly incorporating the fuel injector and an electrohydraulic valve, all for facilitating the injection of high-pressure gaseous fuel. The fuel injector is operated by hydraulic oil to successively open and close to inject gas into the cylinders of the engine in sequence with the engine cycles.

5 The invention includes at least one oil collection chamber for collecting leakage of hydraulic oil and/or any gas for safe removal. According to a preferred embodiment, two interconnected chambers are provided. One chamber collects oil leakage past the piston and gas leakage past the gas valve. The second chamber collects any other leakage between the electrohydraulic valve and the fuel injector.

10 Gas seal leakage is not anticipated for new fuel injectors and their cartridges, but it will be appreciated that over the course of several hundred million operating cycles, wear of gas seals can occur, and particularly, the gas valve seal engaging the moving valve, and as such, gas leakage can occur. The collected oil and/or gas is directed through a separate outlet port away from the entire assembly to a

15 remote gas and oil separator to remove gas from the oil and recycle the oil. A small clearance may be provided between the piston and the piston chamber or bore in which it reciprocates to regulate and limit the amount of leakage and thereby provide a controlled amount of leakage.

 Several advantages may result from the availability of leaked oil in the fuel

20 injector assembly. One advantage is that the leaked oil can be used to lubricate the contact surfaces between an upper guide and the valve. This allows for a precision hardened steel upper guide without wear concerns that provides accurate guiding of the gas valve, prolonging the life of the gas valve seat and sliding dynamic gas valve seal. Another advantage is that the collection chambers and associated

25 passages provide a low pressure buffer between the high-pressure gaseous fuel and high pressure hydraulic operating fluid, as well as between these high pressure locations inside the fuel injector and the external atmosphere. In a preferred embodiment, two collection chambers are provided, one between the

 electrohydraulic valve and the fuel injector cartridge, and one between the valve

30 and piston inside of the cartridge.

The novel fuel injector cartridge of the present invention may be replaced periodically, typically more frequently than other components of the system. The structure of the preferred embodiment provides a cost effective means for periodically replacing the fuel injector cartridge.

5 Other features and advantages of the invention will become more apparent from the following detailed description when taken in conjunction with the accompanying drawings, in which:

Brief Description Of The Drawings

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FIG. 1 is a cross-sectional view of a high-pressure fuel injector assembly, illustrated in a closed position.

FIG. 2 is a cross-sectional view of a high-pressure fuel injector assembly similar to that in FIG. 1, but in an open position.

15 FIG. 3 is an enlarged cross sectional view of a portion of the high-pressure fuel injector illustrated in FIG. 1.

FIG. 4 is an enlarged cross sectional view of another portion of the high-pressure fuel injector illustrated in FIG. 1.

20 FIG. 5 is an enlarged cross-sectional view of the high-pressure fuel injector assembly of FIG. 1, but taken about a different plane to indicate the provision of the gas/oil outlet.

FIG. 6 is a schematic illustration of the high-pressure fuel injector assembly of FIG. 1, in an engine system environment.

25 FIG. 7 is a perspective and partly schematic illustration of multiple high-pressure fuel injector assemblies in an engine system environment and mounted to an engine.

FIG. 8 is a perspective view of the high-pressure fuel injector assembly of FIG. 1.

FIG. 9 is a perspective view of the high-pressure fuel injector assembly of FIG. 1, but with a different embodiment of the fuel injector housing.

While the invention will be described in connection with certain preferred embodiments, there is no intent to limit it to those embodiments. On the contrary, the intent is to cover all alternatives, modifications and equivalents as included within the spirit and scope of the invention as defined by the appended claims.

5

Detailed Description

Referring to the cross section of FIG. 1, the present invention is embodied in a high-pressure fuel injector assembly 20. The high-pressure fuel injector
10 assembly 20 generally comprises a fuel injector 22 and an electrohydraulic valve assembly 24. In general, the electrohydraulic valve assembly 24 hydraulically operates the fuel injector 22 to successively inject gaseous fuel such as natural gas into the cylinders of an engine 121. A partly schematic illustration of an engine 121 with multiple fuel injector assemblies 20 is illustrated in FIG. 7. The disclosed
15 fuel injector assembly 20 provides a commercially reliable and practical fuel injector for injecting high-pressure gaseous fuel (eg. around 300-700 psig, but also including greater and lesser pressure) into combustion engines, thereby improving the efficiency of the engine and reducing the environmental emissions therefrom. Detail below will first be given to the structure and function of the high-pressure
20 fuel injector assembly 20 as shown in FIG. 1 and then to an exemplary engine operating environment for the assembly 20.

Although electrohydraulic valves are not believed to have been previously applied to the present art, it should be noted that electrohydraulic valves and associated mounting assemblies are generally known in other related fields of art.
25 As such, for purposes of the present invention, the electrohydraulic valve assembly 24 is intended to have a broad meaning and may include an electrohydraulic valve 26, and a mounting block 28 for mounting the electrohydraulic valve 26 to the fuel injector 22. A mounting flange 30 secures the entire assembly 20 to the engine 121 via conventional fasteners or bolts as shown
30 in FIG. 7. In the preferred embodiment, the electrohydraulic valve 26 includes an

electrical driver 23 such as an on/off solenoid and a three-way control valve 25.

The three way control valve 25 has a high pressure inlet 27 connected to a pressurized hydraulic supply of oil or other suitable hydraulic fluid and a low pressure outlet 29 connected to a lower pressure sump of oil. In response to

5 external electrical pulses or signals from the electronic engine control, the electrical driver 23 switches the control valve 25 between two positions to successively connect an output 31 alternatively to the high pressure inlet 27 and the low pressure outlet 29. This provides successive hydraulic signals and also alternates the direction of hydraulic flow between the electrohydraulic valve 26
10 and the fuel injector 22. As illustrated in FIG. 5, the mounting flange 30 may provide an external gas inlet port 33 for connecting a fuel supply to the inlet of the cartridge and an external gas/oil outlet port 35 for connection to a gas oil separator, the function of which will be described in further detail below.

Many aspects of the present invention are directed toward the fuel injector
15 22 which is operated by any suitable form of hydraulic signals of a hydraulic type fluid such as oil. Hydraulic actuation provides sufficient force to actuate the valve 46 despite large opposing forces due to the high gaseous fuel pressures, friction, and mechanical spring forces in the cartridge 34. The fuel injector 22 generally includes an outer tubular cartridge housing 32 and an fuel injector cartridge 34
20 mounted therein. In the preferred embodiment, the cartridge housing 32 includes a hollow and cylindrical body tube 36, a nose piece 38, and a mounting flange 30, all brazed together, and a nozzle 40 press fit into the nose piece 38. Although the nozzle 40 could be integrally provided by the nose piece 38, providing a separate nozzle 40 allows for easy design modifications of the nozzle which can be suited
25 to different sizes or types of engines. The nozzle 40 regulates and optimizes dispersion and mixing of the gaseous fuel in the cylinders of the engine and as such improves environmental emissions and efficiency of the engine. In the preferred embodiment, one end of the cartridge housing 32 is closed by the electrohydraulic valve assembly 24 and the other end of the outer housing 32 is
30 closed by the combination of the nose piece 38 and the end portion of the fuel

injector cartridge 34. The cartridge housing 32 contains a gas passageway 41 for communicating gaseous fuel from the gas inlet port 33 to the nozzle 40. In the preferred embodiment, the gas passageway 41 is a large annular chamber between the housing 32 and the cartridge 34. The volume of this gas passageway 41 is
5 maximized to provide a large local reservoir. This large gas reservoir serves to maintain desirably high gas injection pressure throughout the injection event.

The fuel injector cartridge 34 generally comprises a generally cylindrical cartridge body 42 that houses a cylindrical piston 44 and an elongate valve 46. In the preferred embodiment, the cartridge body 42 is generally of two piece
10 construction, including a lower valve body 48 and an upper actuator body 50 screwed together via interfitting threads or otherwise secured together. The combination of the stationary components, eg. the cartridge body 42 and the outer cartridge housing 32, provide a stationary support housing that provides the gas passageway 41 into the engine cylinder and supports the moving components such
15 as the piston 44 and valve 46. Although it will be appreciated that in alternative embodiments the support housing may be provided by fewer or more components. The actuator body 50 defines a cylindrical bore or control chamber 52 in which the piston 44 is slidably mounted for linear reciprocating movement. The control chamber 52 is connected by a drilled passage 54, connector tube 56, and orifice
20 plug 71 to the output 31 of the electrohydraulic valve 26 for receiving hydraulic operating signals. The end of the drilled passage 54 provides a hydraulic input for receipt of hydraulic signals.

The valve body 48 generally includes a steel body sleeve 58 and upper and lower spaced apart cylindrical collar guides 60, 62. In the preferred embodiment,
25 the upper guide 60 is a solid machined steel member while the lower guide 62 is a self lubricating, high temperature, carbon/graphite bushing formed from a commercially available material. The lower guide 62 is press fit into the sleeve 58. One potential problem with use of carbon/graphite material is fragility and the susceptibility to chipping at the edges. As such, a steel washer or other bushing
30 retainer 63 is seated in a recess in the sleeve 58 below the graphite bushing. The

bushing retainer 63 prevents graphite or carbon chips from dropping down and potentially lodging between the valve 46 and valve seat 68. The valve 46 is slidably mounted through axially aligned bores in the guides 60, 62 for linear reciprocating movement between open and closed positions. The guides 60, 62 thus support and guide the linear reciprocating movement of the valve 46. As illustrated, the end portion of the valve body 48 closes one end of the cartridge housing 32. The sleeve 58 defines a frusto-conical valve seat 68 surrounding an outlet orifice 70 that provides for discharge of gaseous fuel into the cylinders of the engine. To ensure correct alignment of the valve seat 68 and the bore of the lower guide 62, the inner diameter of the conical seat 68 and the inner diameter of the bore in the guide 62 are simultaneously or sequentially precision ground, thereby assuring accurate alignment. This provides precise alignment of the valve with its seat, resulting in long seat life, low gas leakage, and therefore more precise and accurate control over fuel injection. The lower end portion of the valve body 48 also includes cross holes 72 formed in the sleeve 58 and below the lower guide 62 to extend the gas passageway 41 to the outlet orifice 70.

A helical compression spring 66 is mounted in a spring chamber 64 between the sleeve 58 and the valve 46 (surrounding the valve 46). The spring 66 biases the valve 46 to a closed position as shown in FIG. 1 in which an enlarged frusto-conical closure member 74 on the valve 46 is seated against the valve seat 68 along a circular contact. Preferably, the respective slope or angles of the mating conical surfaces between the seat 68 and the closure member 74 are offset slightly by a degree or more to ensure tight circular contact which prevents leakage of gaseous fuel into the cylinders of the engine. As shown in FIG. 3, the spring 66 engages a disc shaped spring retainer 76 which is secured to the valve 46 by keepers 78. The spring provides a large force sufficient to prevent the high pressures of the gaseous fuel from causing fuel leakage into the engine's cylinder while the valve is closed. Although the spring 66 could be eliminated if a 4-way actuating valve was provided in the electrohydraulic valve in which the piston would be configured to be hydraulically actuated both ways to both open and

closed positions by high pressure hydraulic signals, the spring 66 performs the necessary function of a fail-safe, in that the spring 66 mechanically maintains the valve 46 in the closed position in the event of failure of the electrohydraulic valve or the hydraulic pressure supply.

5 In the preferred embodiment, the valve 46 is a separate member from the piston 44, but another embodiment of the present invention may integrally provide the two or otherwise connect the two together. These and other possibilities are intended to be covered by all of the claims appended hereto. The piston 44 includes a reduced diameter nose 80 which contacts the top surface of the valve.

10 Surrounding the nose 80 is a seating surface 82 which is adapted to engage the top surface of the upper valve guide 60 acting as a mechanical stop to control the stroke or maximum distance of linear movement of the valve 46, and thereby the fuel injection rate.

 To open the valve 46, the piston 44 is actuated in response to high pressure

15 hydraulic signals or pulses from the electrohydraulic valve 26. High pressure hydraulic signals result by a connection between the output 31 and the high pressure inlet 27. High pressure hydraulic signals received in the control chamber 52 overpower the force of the spring and linearly actuate the valve 46 to an open position as illustrated in FIG. 2 in which the closure member 74 is lifted off of the

20 valve seat 68 to allow passage of gas through the outlet orifice 70 and into the corresponding cylinder of the engine.

 As or after the valve 46 opens, the electrohydraulic valve 26 ends the high pressure hydraulic signal and switches the connection to the output 31 by connecting the output to the lower pressure outlet 29. The spring 66 automatically

25 returns the valve 46 to the closed position, causing hydraulic oil in the control chamber to flow to the lower pressure outlet 29.

 The preferred embodiment also includes an orifice plug 71 located in the input passage regulating flow between the electrohydraulic valve 26 and the control chamber 52. It is an advantage that the orifice plug 71 is more restrictive

30 one way and less restrictive the other way, such that the valve 46 moves more

quickly from the closed position to the open position than the movement from the open position to the closed position. Because the orifice plug 71 is less restrictive in the direction associated with valve opening, reduced fluid pressure is required to achieve acceptable valve opening velocity. Reduced fluid pressure has the advantage of lower hydraulic power consumption, reduced fluid heating, and less hydraulic system stress. Reduced closing velocity reduces the impact and resulting wear between the valve seat 68 and the closure member each time the valve 46 closes. To accomplish this flow regulation, each side of the orifice plug 71 has a different discharge coefficient. In particular, the plug 71 includes a restriction orifice 73 and a conical or otherwise chamfered surface 75 on one side of the restriction orifice 73 and a substantially flat surface 77 on the other side of the restriction orifice 73. The restriction orifice 73 determines the maximum speed of actuation by limiting hydraulic flow. The chamfered surface 75 directs the pressure of the hydraulic signals like a nozzle and increase the amount of flow through the orifice 73. The substantially flat surface 77 does not direct the flow into the orifice 73 and acts as a barrier thereby reducing the amount of flow through the orifice 73. As a result, the valve 64 moves more quickly towards the open position and more slowly towards the closed position. The force of the spring 66 is also selected to control the return rate.

In accordance with an aspect of the present invention relating to practicality and reliability of the fuel injector 22 and the entire assembly 20, a small controlled amount of hydraulic oil leakage is allowed past the piston 44 for collection in a collection chamber 83 between the actuator body 50 and the valve body 48. In the preferred embodiment, the collection chamber 83 is provided by recesses in the actuator body 50 and mounting block 28. The piston 44 and its mating bore in actuator body 50 are made with hardened, wear resistant surfaces. When lubricated by hydraulic oil, these sliding surfaces exhibit long cyclic life with negligible wear. Conventional sliding seals are commonly known to not provide the required cyclic life and are therefore considered to be not satisfactory for sealing between the piston 44 and its bore. It is therefore an advantage to avoid

using sliding seals, and to simply incorporate the lubricated, hardened, steel surfaces. The lubricating oil leakage passing the piston 44 is limited by the small annular clearance between the piston 44 and its mating bore. There are several other advantages of this leakage. One significant advantage it that the leaked
5 hydraulic oil lubricates the sliding movement between metal to metal contact surfaces between the inner bore of the upper guide 60 and the valve 46. This increases wear resistance and significantly prolongs the life of the components in the cartridge 34. Another advantage is that the oil lubricates and prolongs the life of a gas seal 84 between the upper guide 60 and the valve 46. The leaked oil
10 collected in the collection chamber 83 is directed via an outlet in the form of an axial outlet passage 86 in the cartridge body 42 that is connected to the gas/oil outlet port 35 for removal to an external location where gas and oil separation can occur. It should be noted that the leakage is controlled to be a very small flow rate.

The O-ring gasket 85 prevents gas leakage from the gas passageway 41 to
15 the collection chamber 83. The gas seal 84 prevents gaseous fuel leakage between the valve 46 and the upper guide 60. The gas seal 84 is located at the far lower end of the upper guide 60 such that oil lubricates all or substantially all of the contacting surfaces between the upper guide 60 and the valve 46. When initially installed, the gas seal 84 and the O-ring gasket 85 provide zero leakage of gas
20 from the gas passageway 41 (including spring chamber 64) to the collection chamber 83. However, it will be appreciated that over the lifetime of operation (eg. during several hundred million operating cycles) wear can occur, which in turn, may and often causes slow gaseous leakage past the gas seal 84. Indeed, the intense gas pressure exerted by the fuel (eg. typically around 300-700 psig) greatly
25 increases the likelihood of such leakage occurring. The provision of the collection chamber 83 provides a fail safe, tolerates such leakage and vastly extends the operating life for the fuel injector cartridge 34, because small gas leakage is carried away to an acceptable disposal means. If it were not for this gas leakage disposal means, the engine would have to be stopped, and the leaking cartridge
30 replaced, at the first sign of gas leakage past the seals.

A second collection chamber 87 is also provided at the other axial end of the passage 86, generally between the actuator body 50 and the mounting block 28 of the electrohydraulic valve assembly 24. A number of O-ring gaskets 88-91 are provided in this general vicinity and serve to prevent leakage. Two connector tube
5 O-rings 88, 89 between the connector tube 56 and the actuator body 50 and the mounting block 28 of the electrohydraulic valve 26 prevent leakage of oil into the collection chamber 87. However, the continuous and cyclic pulses of hydraulic oil through the connector tube 56 presents a possibility of oil leakage after a long time period. As such, small amounts of oil leakage can be allowed or is tolerated as it is
10 collected in the second collection chamber 87. An O-ring 90 is also provided between the mounting flange 30 and the actuator body 50 to prevent leakage of high-pressure gaseous fuel from the gas passageway 41. However, a small amount of gas leakage is also tolerated at this location, in which gas would be collected in the second collection chamber 87 for removal. The outer O-ring 91 prevents
15 leakage of oil and gas to the external environment. It will be appreciated that the oil and any combined oil/gas in the second collection chamber 87 is at relatively low pressure, much lower pressure than either the high-pressure gaseous fuel supply or the hydraulic oil at the high pressure inlet 27. As a result, little pressure and thus minimal forces are exerted on this gasket 91 thereby providing a highly
20 reliable seal at this location and avoiding leakage to the external environment.

From the foregoing, it will be appreciated by those skilled in the art that the first and second collection chambers 83, 87 each provide a fail-safe for oil leakage or gas leakage at several locations and two separate means for tolerating leakage of oil and gas at least one location in the fuel injector assembly and for removal of
25 any leakage of hydraulic fluid and gas from the fuel injector assembly.

In accordance with another aspect of the present invention relating to universality of the valve cartridge 22, a shim 92 is used to control the maximum stroke or distance of reciprocating movement of the valve 46. As shown in FIGS. 1 and 4, the upper guide 60 is compressed axially between the actuator body 50
30 and a shoulder/recess 93 formed in the valve body sleeve 58. A stop plate 100 and

shim 92 are positioned axially between the upper guide 60 and the shoulder/recess to control the amount that the upper guide 60 protrudes from the valve body 48 and the resulting overall axial length of the valve body 48. Excess threads in the two bodies 48, 50 are provided to accommodate variations in their engagements
5 due to different sizes of shims. A thicker shim 92 will increase the protrusion of the upper guide 60 relative to the upper face of the valve 46. As a result, the distance between the outer face of the piston 44 and the face of the upper guide will be reduced, causing the stroke of the valve to be reduced. This in turn results in less gaseous fuel being injected into the cylinders of the engine during each
10 cycle. A thinner shim 92 will increase the allowable stroke of the valve 46, resulting in more gaseous fuel being injected into the cylinders of the engine during each cycle. The selection of the shim 92 thickness allows the valve cartridge 22 to be easily adjusted for larger and smaller types of engines which have different fueling requirements. Thus the preferred embodiment provides a
15 valve cartridge and fuel injector that are universal for a variety of different engines. Using the shim 92, conventional opening distances for the valve closure member 74 of the preferred embodiment can be conveniently adjusted over the range of opening distance range desired for these types of engines. This is an important advantage when considering that the fuel injector 22 is used to retrofit
20 existing engines which exist in a wide variety of models and sizes. Also as shown, the shim 92 and a stop plate 100 axially retain the gas seal 84.

In accordance with another aspect of the present invention relating to cooling and reliability, openings in the form of cross-holes 94 are drilled into the valve body sleeve 58 at radially spaced intervals. The cross-holes 94 allow the
25 cool gaseous fuel entering the gas inlet 33 and flowing through the gas passageway 41 to cool the exposed surface of the valve 46 inside the spring chamber 64. During operation, the nozzle 40 and closure member 74 are exposed to extreme temperatures inside the cylinder of the engine, eg. up to about 2000 degrees Fahrenheit. In contrast, the conventional material of gas seal and other
30 conventional material gaskets and spring materials start to thermally deteriorate at

around 300-400 degrees Fahrenheit. By cooling the exposed surface of the valve 46, life of the gaskets/seals and spring and therefore life of the cartridge 34 is prolonged. During operation, the pressure in the gas passageway 41 rises and falls as the valve 46 opens and closes. This in turn causes relatively cool gas to pulsate
5 into and out of the spring chamber vastly enhancing the cooling effect achieved. These cross-holes 94 direct this gas flow towards the valve and spring and improve the life span and reliability of the cartridge 34 by removing heat that would otherwise travel up the valve and spring, undesirably raising the operating temperature of the spring and seals.

10 Still another function of the cross-holes 94 is to provide a means of restraining the gas valve body 48 while tightening/loosening the threaded joint joining the actuator body 50 and the gas valve body 48. This is accomplished by engaging pins in the cross-holes 94 using a holding fixture designed for that purpose.

15 Another novel feature of the preferred embodiment is the provision of a metal O-ring 95 for sealing the contacting surfaces between the cartridge housing 32 and the cartridge 34. The metal O-ring provides a highly reliable seal in a location proximate the engine cylinders where the temperatures are extreme. It will be appreciated that current materials for other more conventional types of
20 gaskets would likely fail from thermal damage in this type of environment. To maintain the metal O-ring 95 in sealing relationship, a large axial force, eg. of about 10,000 pounds, is applied by a spring in the form of two Belleville load or spring washers 96 supported by the body of the electrohydraulic valve 26 and engaging the other axial end of the cartridge 34. Specifically, the load washers 96
25 engage a load pad 97 situated in the second collection chamber 87 and seated in a formed recess in the actuator body 50. Shim 98 is interposed between the load pad 97 and the recess in the actuator body 50. It should be noted that the thickness of the shim 98 is selected to maintain the desired force on the metal O-ring 95. In particular, recalling that the thickness of the shim 92 is variable depending upon
30 the fueling rate requirements of the intended engine, the thickness of the second

shim 98 depends upon the thickness of the first shim 92. The thicker the first shim 92, the thinner the second shim 98 is to thereby maintain the same force on the metal O-ring 97. The thickness of second shim 98 is also adjusted to achieve deflection of the load washers as required to generate the desired metal O-ring clamping force, compensating for the effects of manufacturing tolerances in the parts. A threaded hole 99 is also drilled 180 degrees apart from the axial passage 54 to facilitate insertion of screw which can then be used to lift the cartridge 34 out of the housing 32. Together, threaded hole 99 and passage 86, diametrically opposite each other in the face of actuator body 50, conveniently accommodate a common spanner wrench adapter to facilitate tightening/loosening the threaded joint connecting the actuator body 50 and the valve body 48.

Because of the number of moving components and seals, the fuel injector cartridge 34 is intended to have a lifespan of about one to two years. As such, the cartridge 34 is easily replaced by removing the electrohydraulic valve assembly 24, and the various parts between the electrohydraulic valve and the cartridge and pulling the cartridge 34 from the cartridge housing 32. The electrohydraulic valve 26, cartridge housing 32 and interposed parts can be reused with a new replacement fuel injector cartridge 34 and new metal O-ring 95.

FIGS. 6 and 7 illustrate a high-pressure fuel injection system 120 incorporating the high-pressure fuel injector assembly 20. The primary advantage of this type of system is that the fuel injector 22 injects fuel at high pressures greatly increasing air and fuel mixing in the cylinders and thereby resulting in fewer harmful environmental emissions and increasing engine efficiency. FIG. 6 illustrates the system 120 in schematic form with a single fuel injector valve assembly 20 while FIG. 7 illustrates the system 120 on an engine 121 with multiple valve assemblies 20, one for each cylinder of the engine 121. The system 120 includes a hydraulic pumping unit 122 for supplying high pressure hydraulic oil to the electrohydraulic valve 26 and an electronic controller 124 for driving the electrical driver 23 via electrical signals on electrical line 123. The hydraulic pumping unit 122 in this case is located remote from the engine cylinders and may

be electrically or pneumatically powered. The preferred embodiment illustrated in FIG. 7 is an engine driven pump 126, a low pressure sump or reservoir 128, and a gas/oil separator 130. The pump 126 is adapted to pump hydraulic oil from the reservoir 128 to the high pressure inlet 27 of the electrohydraulic valve 26 via a high pressure hydraulic oil supply line 132. The pressure in this line 132 may be in the rough neighborhood of around 800 psig. A low pressure hydraulic return line 134 connects the low pressure outlet 29 with the reservoir 128. This pressure in this line 134 may be in the rough neighborhood of about 45 psig. A gas/oil return line 136 connects the gas/oil outlet port 35 to the gas/oil separator 130. The gas/oil separator 130 allows any combined gas and oil to sit for a sufficient time at which the gas separates and is exhausted via a gas vent 138 to a non-explosive location. A gaseous fuel supply 140 of a combustible gas is connected to the gas inlet 33 by a gas line 142 that may have a pressure in the neighborhood of between about 300-700 psig, or other suitable lower or higher pressure. Other associated equipment includes a hydraulic oil filter 144 for keeping the hydraulic oil clean and a gas leakage indicator 146 for sensing excessive gas leakage which could indicate hazardous conditions.

While this invention has been described with an emphasis upon preferred embodiments, it will be obvious to those of ordinary skill in the art that variations of the preferred embodiments may be used and that it is intended that the invention may be practiced otherwise than as specifically described herein. Accordingly, this invention includes all modifications encompassed within the spirit and the scope of the invention as defined by the following claims.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

1. A fuel injection system for injecting gaseous fuel from a gaseous fuel supply into cylinders of an engine, the injection system comprising:
 - an oil reservoir of hydraulic oil;
 - a hydraulic oil supply of hydraulic oil at a pressure higher than that in the oil reservoir;
 - a gas/oil separator adapted to separate gaseous fuel from the hydraulic oil, and to return the separated oil to the oil reservoir;
 - a plurality of fuel-injector valve assemblies for injecting gaseous fuel into the cylinders of the engine, each fuel-injector valve assembly including an electrohydraulic valve and a fuel injector, the electrohydraulic valve actuating the fuel injector with hydraulic signals derived from the hydraulic oil supply to inject gaseous fuel from the gaseous fuel supply into at least one of the cylinders of the engine, and the fuel-injector valve assemblies having at least one collection means adapted to collect hydraulic oil and leaked gaseous fuel, the at least one collection means returning said hydraulic oil and leaked gaseous fuel to the gas/oil separator.
2. A fuel injection system as defined in claim 1, wherein the electrohydraulic valve successively creates hydraulic signals in response to electrical control signals, and wherein the fuel injector includes a fuel valve operated by a piston and an outlet port in communication with the gaseous fuel supply, the piston reciprocating the fuel valve in response to hydraulic signals to open and to close the outlet port and thereby successively inject gaseous fuel into the cylinder.
3. A high-pressure fuel injection system for injecting high-pressure gaseous fuel from a gaseous fuel supply into cylinders of an engine, the injection system comprising:
 - an oil reservoir of hydraulic oil;

a hydraulic oil supply of hydraulic oil at a pressure higher than that in the oil reservoir;

a gas/oil separator located in a safe location remote from the cylinders of the engine, the gas/oil separator adapted to separate any gaseous fuel from the hydraulic oil and return the separated oil to the oil reservoir;

an electronic controller providing electrical control signals;

a plurality of high-pressure fuel-injector valve assemblies for injecting fuel into the cylinders of the engine, each fuel-injector valve assembly including an electrohydraulic valve and a fuel injector, the electrohydraulic valve being connected to the electronic controller, to the high-pressure hydraulic oil supply and to the oil reservoir, the electrohydraulic valve successively creating hydraulic signals in response to the electrical control signals, the fuel injector including a fuel valve operated by a piston and an outlet port in communication with the gaseous fuel supply, the piston reciprocating the fuel valve in response to hydraulic signals to open and to close the outlet port and thereby successively inject gaseous fuel into the cylinders, each fuel-injector valve assembly further including means for collecting leaked oil and leaked gaseous fuel and communicating any such leaked oil and gaseous fuel to the gas/oil separator.

4. A fuel injection system as defined in claim 2 or 3, wherein each fuel-injector valve assembly comprises a dynamic gas seal engaging the fuel valve for preventing gaseous fuel from mixing with the leaked oil.

5. A fuel injection system as defined in claim 2, 3 or 4, wherein the electrohydraulic valve is a three-way valve, and the fuel injector further comprises a spring biasing the fuel valve to a closed position, whereby hydraulic oil flows into the fuel injector to cause the valve to open and the spring returns the fuel valve to the closed position causing the hydraulic oil to vent from the fuel injector.

6. A fuel injection system as defined in any one of claims 2 to 5, wherein gaseous fuel is injected through the outlet port and into the cylinder of the engine at a pressure of at least about 300 psig.
7. A fuel injection system as defined in any one of claims 2 to 6, wherein a controlled leakage of hydraulic oil occurs from the hydraulic signals past the piston to provide continuous lubrication, and the collection means collects the controlled leakage of hydraulic oil.
8. A fuel injection system as defined in claim 7, wherein a controlled leakage of hydraulic oil lubricates a portion of the fuel valve.
9. A fuel injection system as defined in any one of claims 2 to 8, wherein the collection means includes a collection chamber between the piston and the fuel valve.
10. A fuel injection system as defined in claim 9, further comprising seal means for preventing gaseous fuel from entering the collection chamber.
11. A fuel injection system as defined in any one of claims 1 to 8, wherein the collection means includes a collection chamber between the electrohydraulic valve and the fuel injector.
12. A fuel injection system as defined in claim 11, further comprising seal means for preventing gaseous fuel from entering the collection chamber.
13. A fuel injection system as defined in claim 12, further comprising second seal means for preventing oil from entering the collection chamber.

14. A fuel injection system as defined in any one of claims 1 to 8, wherein the collection means is a collection chamber contained in the fuel injector substantially fluidically-isolated from said gaseous fuel supply.
15. A fuel injection system as defined in claim 14, wherein a lubricating portion of the hydraulic oil used by the electrohydraulic valve lubricates moving parts of the fuel injector, the lubricating portion being collected in the collection chamber, and further comprising at least one seal in the fuel injector arranged to prevent gaseous fuel from leaking into the collection chamber.
16. A fuel injection system as defined in claim 14 or 15, further comprising a gas vent extending from the gas/oil separator and venting gaseous fuel separated from hydraulic oil to a safe, non-explosive location remote from the engine.
17. A fuel injection system as defined in claim 16, further comprising a gas-leakage indicator fluidically-connected to the gas/oil separator and indicating whether an excessive amount of gaseous fuel is being vented.

FIG. 2

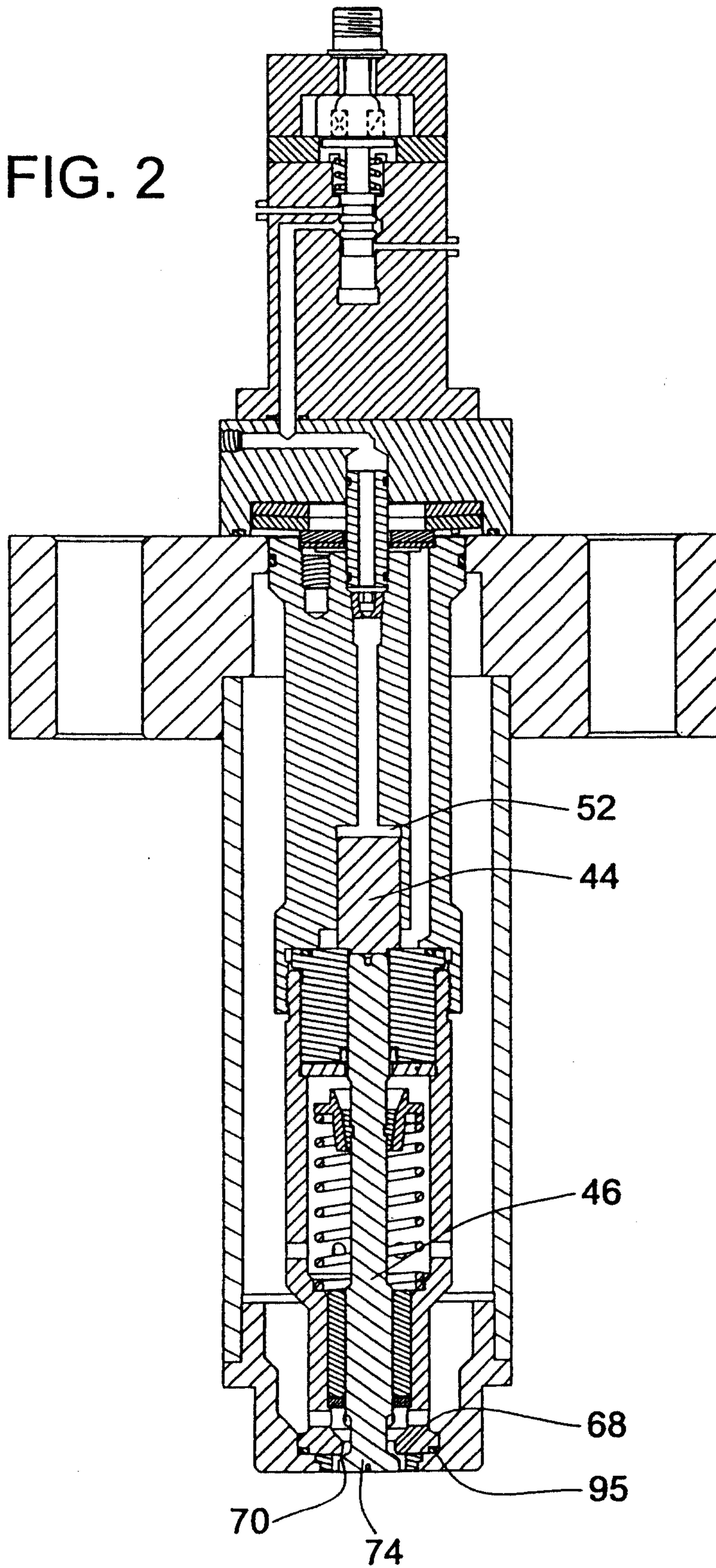


FIG. 3

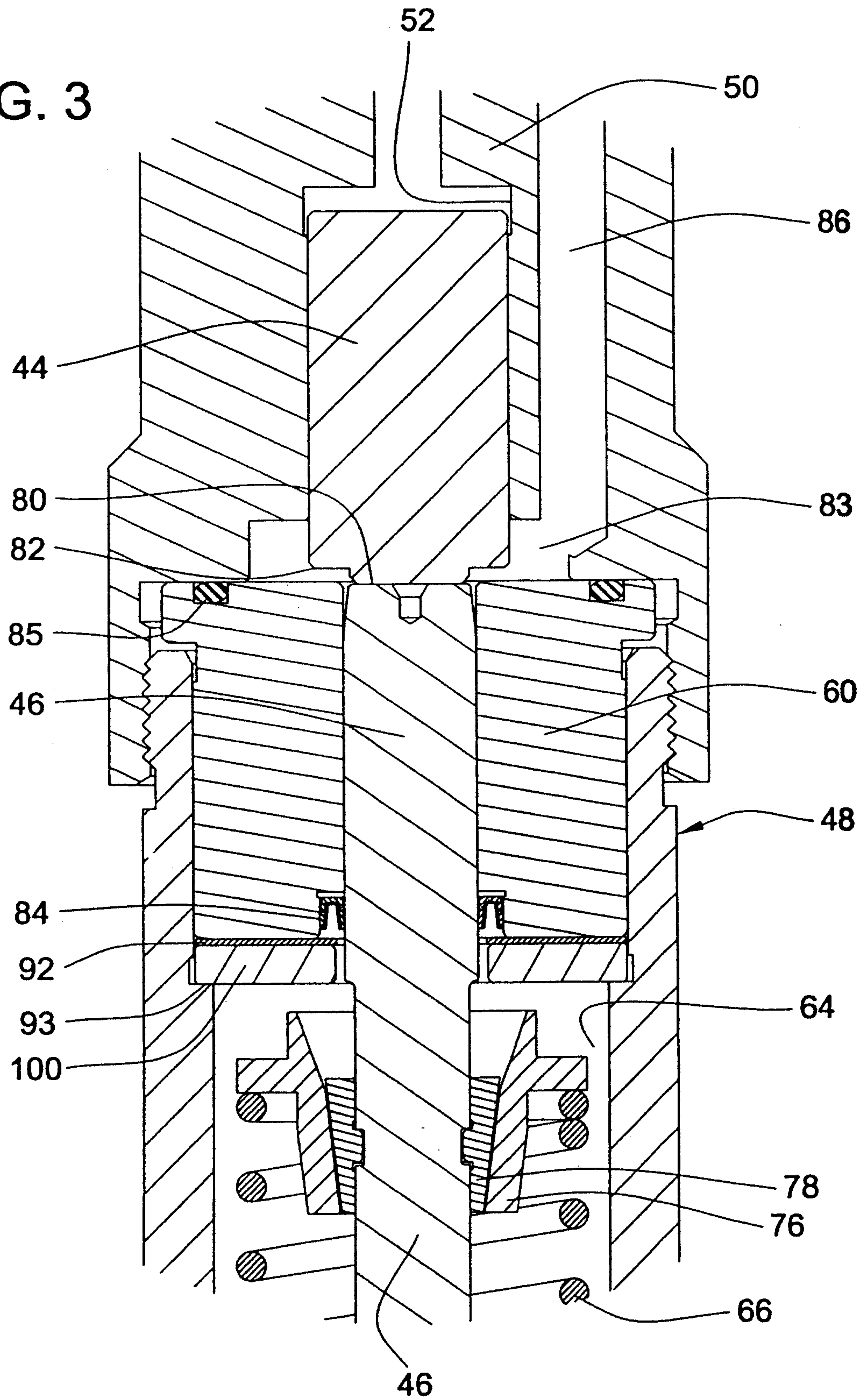


FIG. 4

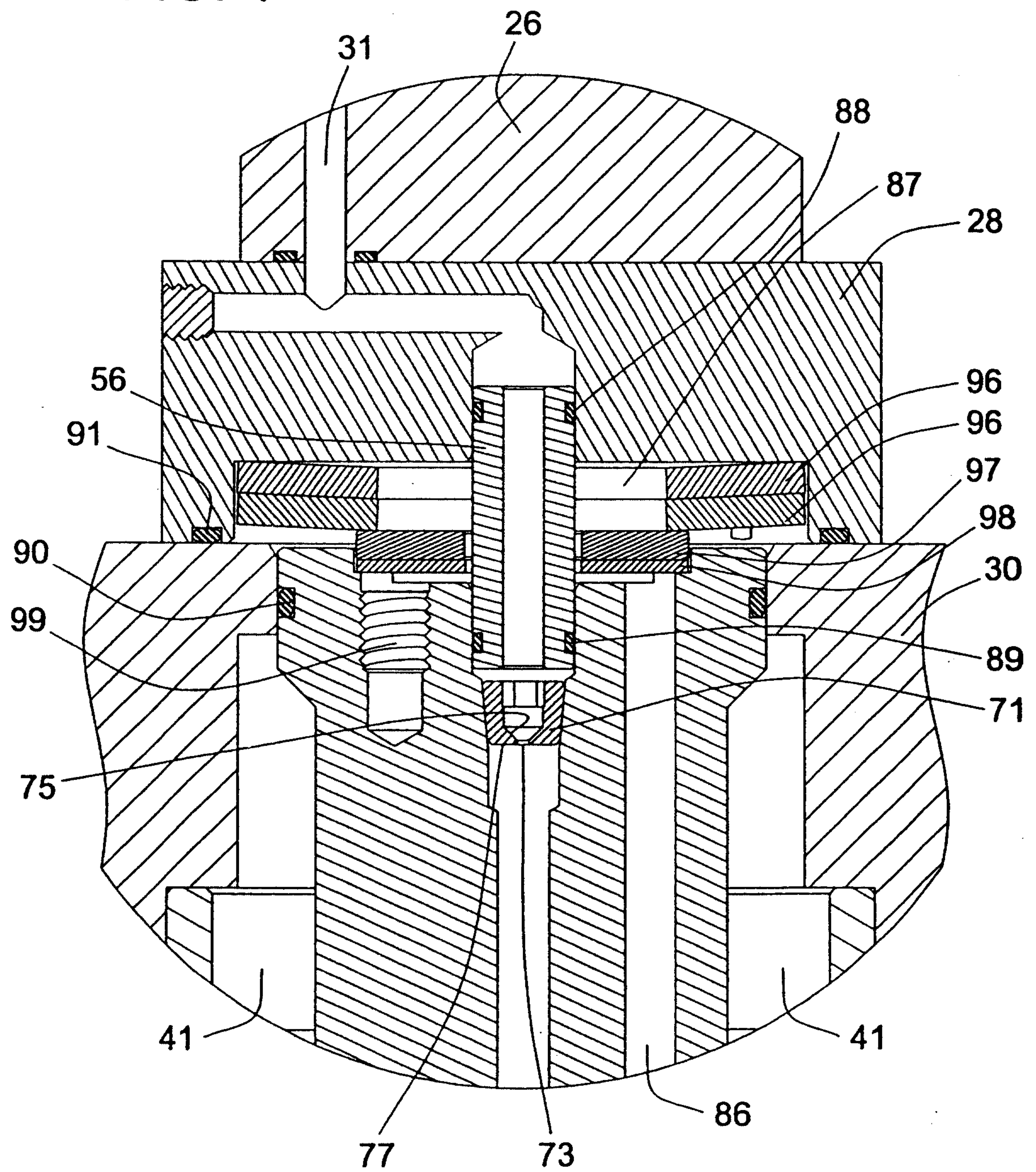
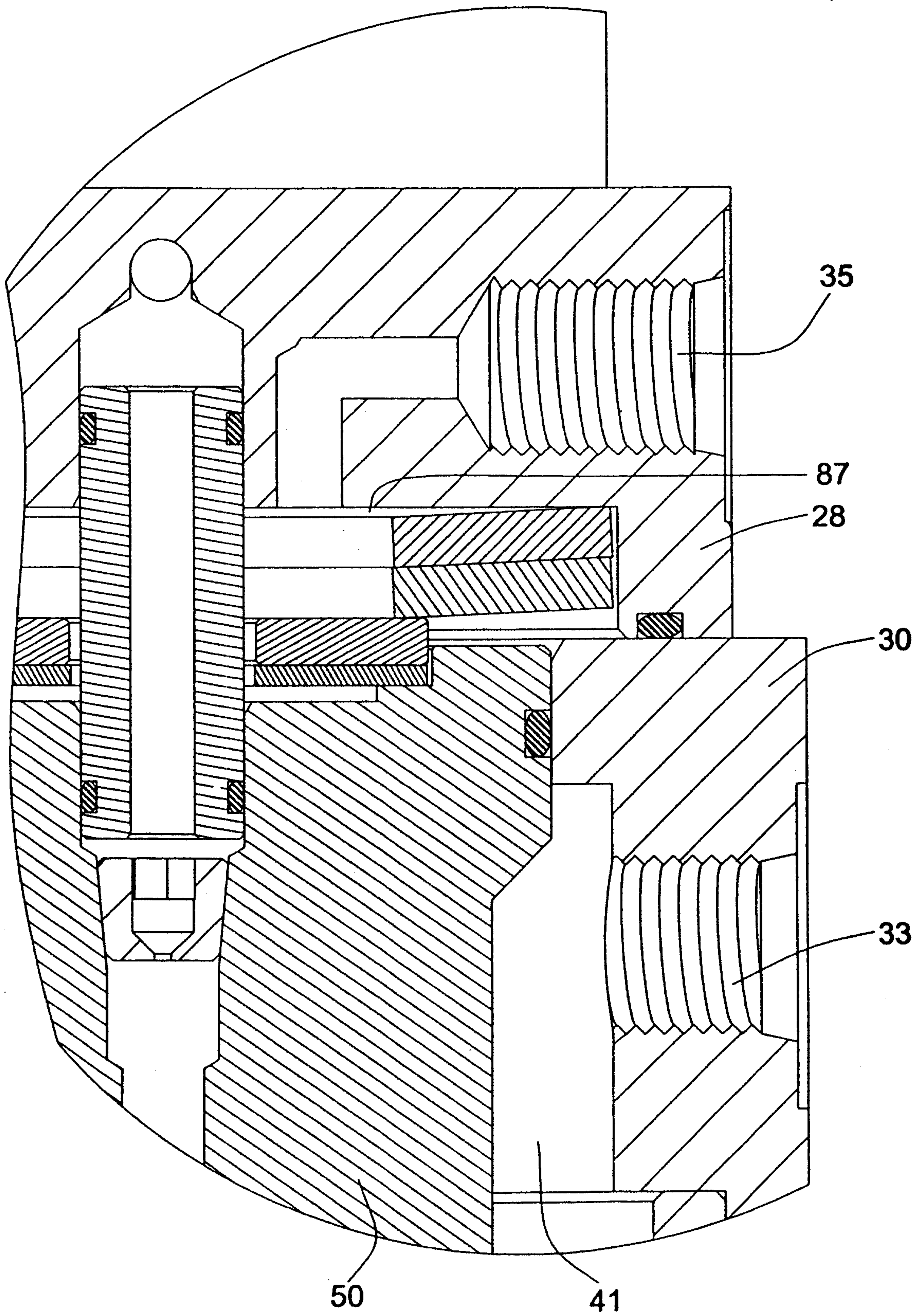


FIG. 5



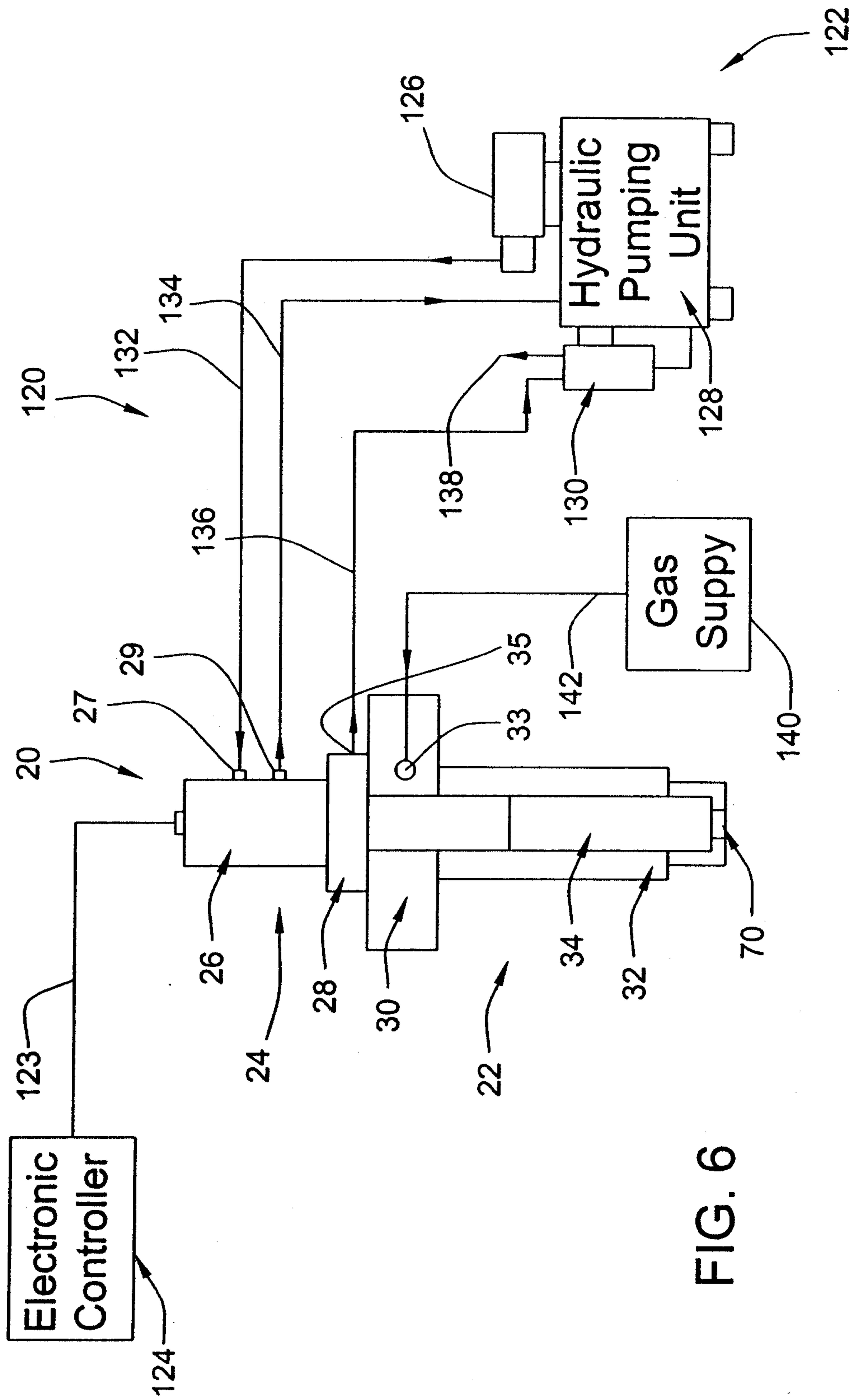


FIG. 6

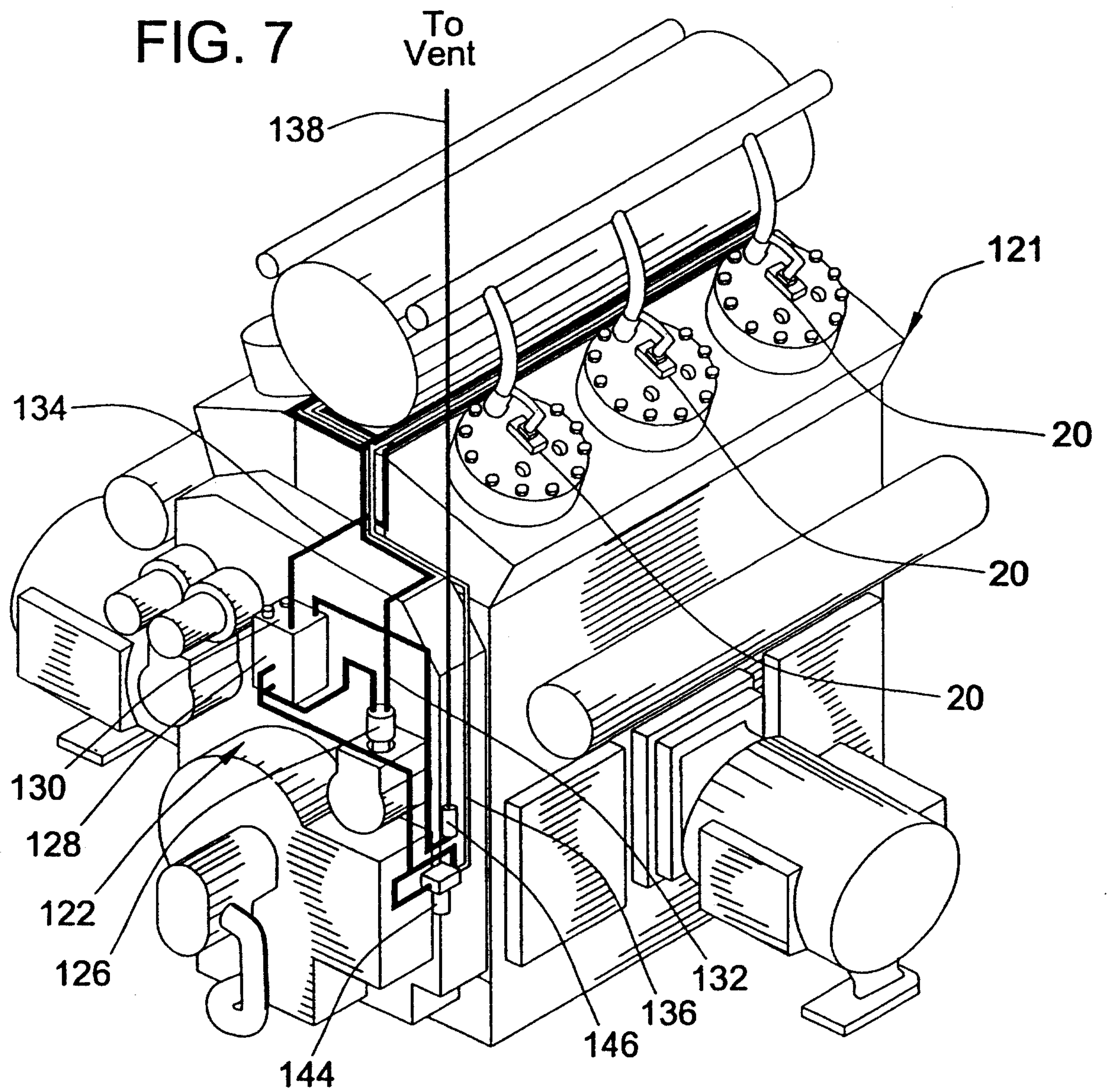


FIG. 8

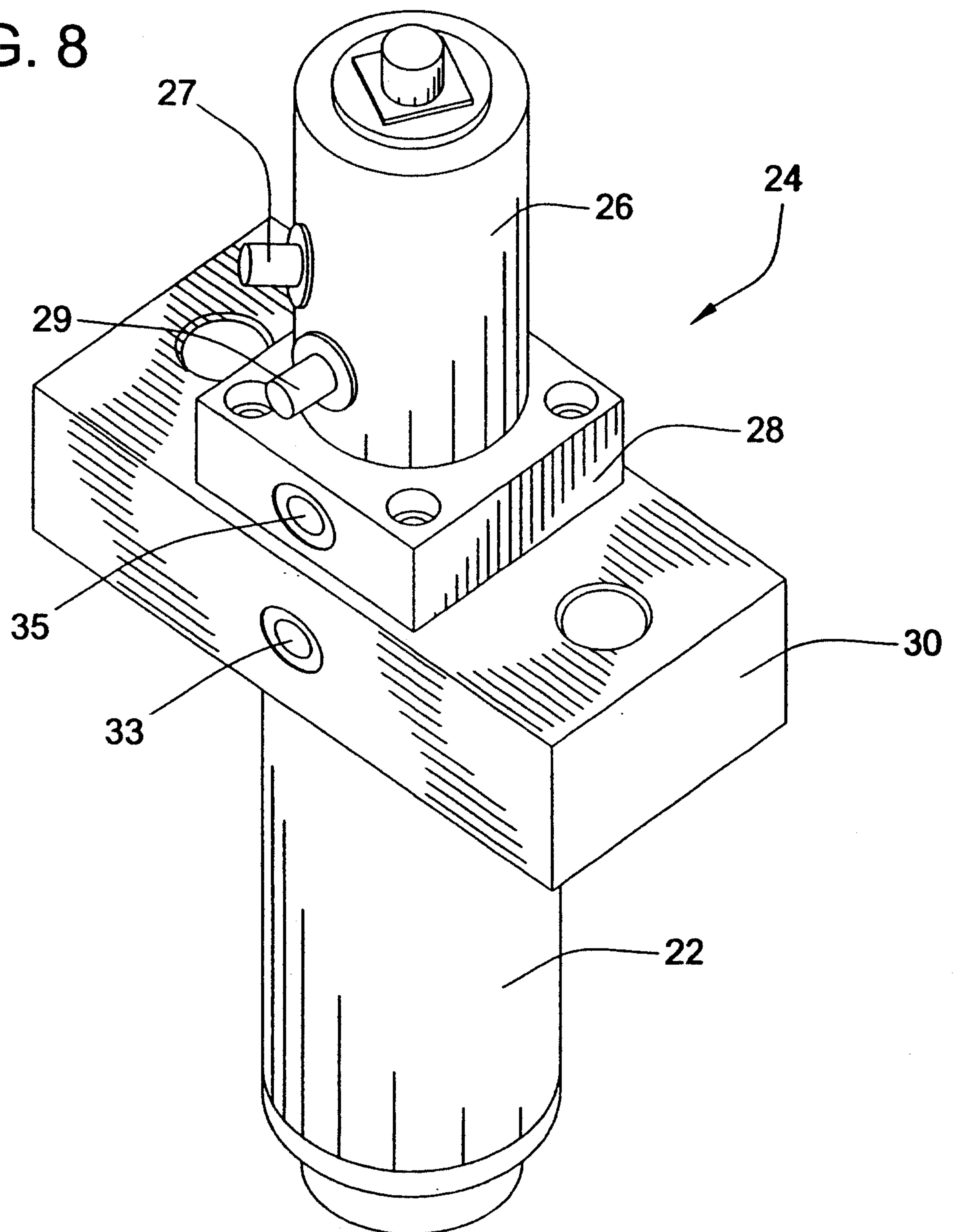


FIG. 9

