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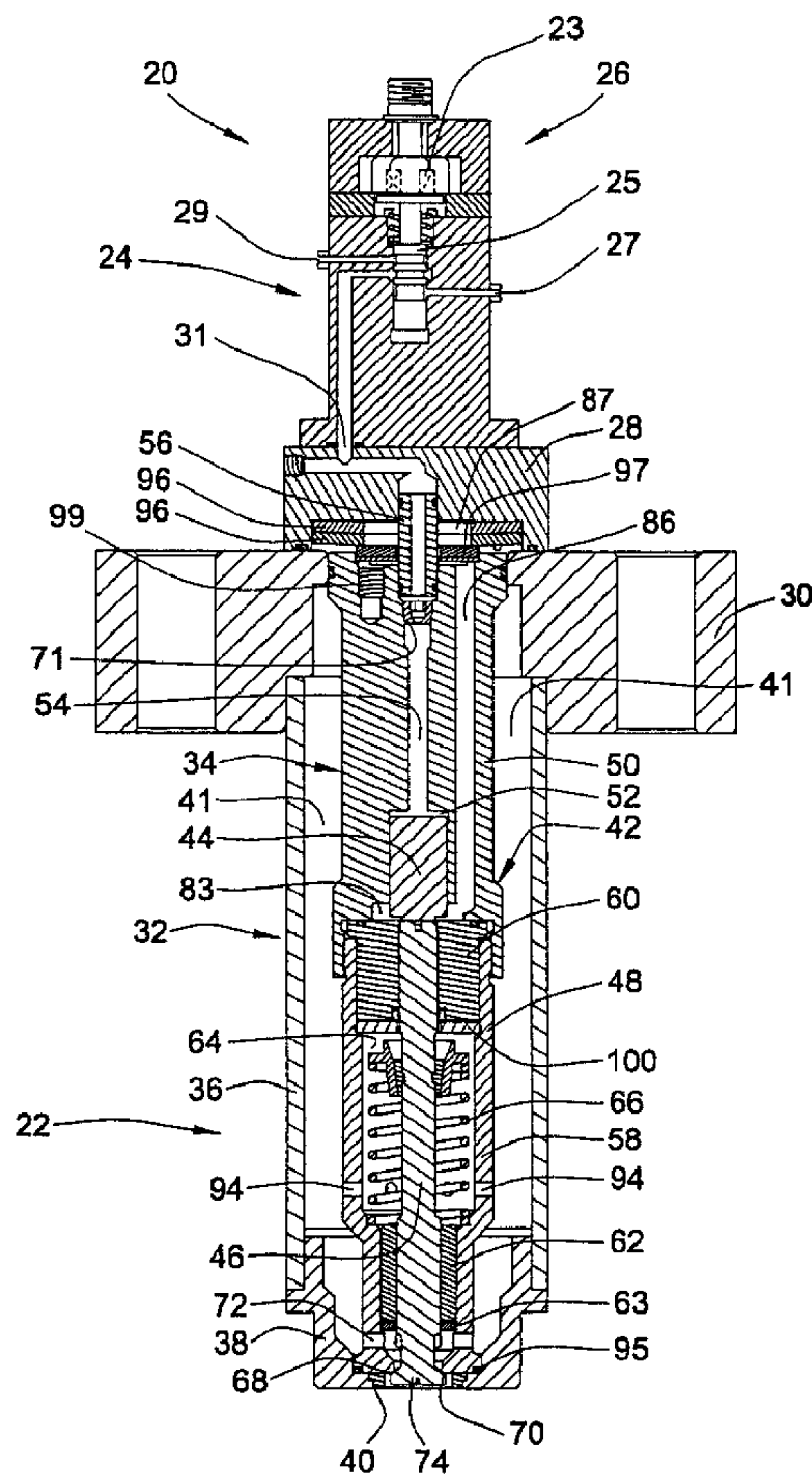
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(54) Title: GASEOUS FUEL INJECTOR HAVING HIGH HEAT TOLERANCE



(57) Abrégé/Abstract:

A high pressure gaseous fuel injector for injecting natural gas or other gaseous fuels at high pressures (eg. 300 to 700 psig more or less) into combustion engines for improved efficiency, better performance and reduced environmental emissions. The fuel



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

injector is powered by hydraulic signals from an electrohydraulic valve. The fuel injector includes an outer cartridge housing and a universal valve cartridge mounted therein. The cartridge comprises an activator body and a valve body secured together, and the valve body has a gas valve slidable therein. The stroke of the valve is adjustable by controlling the size of shims in the valve body assembly, thereby providing a universal valve cartridge that can be easily adapted to differing fueling requirements for models and sizes of engines. The valve body includes a spring chamber between upper and lower valve guides. The valve body also includes cross-holes to allow cool gaseous fuel to successively pulsate into and out of the spring chamber and thereby cool the exposed portion of the valve between guides, preventing heating of the dynamic gas seal located in the upper guide. An unsealed small clearance between the actuating piston and its bore in the actuator body allows controlled leakage of oil. The leakage of oil lubricates metal to metal contact between the upper guide and the valve and lubricates the dynamic gas seal. Gaseous fuel leakage into the leaked oil is tolerated and any combined oil/gaseous fuel is removed to an external location for separation. Spring washers are used to urge the valve assembly insert against a metal O-ring supported by the outer cartridge body.

ABSTRACT OF THE DISCLOSURE

A high pressure gaseous fuel injector for injecting natural gas or other gaseous fuels at high pressures (eg. 300 to 700 psig more or less) into combustion engines for improved efficiency, better performance and reduced environmental emissions. The fuel injector is powered by hydraulic signals from an electrohydraulic valve. The fuel injector includes an outer cartridge housing and a universal valve cartridge mounted therein. The cartridge comprises an activator body and a valve body secured together, and the valve body has a gas valve slidable therein. The stroke of the valve is adjustable by controlling the size of shims in the valve body assembly, thereby providing a universal valve cartridge that can be easily adapted to differing fueling requirements for models and sizes of engines. The valve body includes a spring chamber between upper and lower valve guides. The valve body also includes cross-holes to allow cool gaseous fuel to successively pulsate into and out of the spring chamber and thereby cool the exposed portion of the valve between guides, preventing heating of the dynamic gas seal located in the upper guide. An unsealed small clearance between the actuating piston and its bore in the actuator body allows controlled leakage of oil. The leakage of oil lubricates metal to metal contact between the upper guide and the valve and lubricates the dynamic gas seal. Gaseous fuel leakage into the leaked oil is tolerated and any combined oil/gaseous fuel is removed to an external location for separation. Spring washers are used to urge the valve assembly insert against a metal O-ring supported by the outer cartridge body.

GASEOUS FUEL INJECTOR HAVING HIGH HEAT TOLERANCE**Field Of The Invention**

5 The present invention generally relates to fuel injectors, and more particularly high pressure gaseous fuel injectors for internal combustion engines.

Background Of The Invention

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 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
15 pipelines that supply residential housing and commercial businesses. The network of natural 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
20 being injected into the engine, the pressure of the gas is significantly and substantially reduced. 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, U.S. Patent No. 4,365,756). The problem with low pressure injection is that the fuel pressure provides
25 little kinetic energy with which to induce cylinder charge mixing. There is ample evidence that the fuel and air in 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.

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 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 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 disclosed in Fisher, U.S. Patent No. 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 fueled at low pressure by conventional low pressure fuel injectors.

Summary Of The Invention

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

5 It is an object of the present invention according to one aspect to provide 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.

10 It is another object of the present invention according to another aspect to 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 operate as desired on the various types and sizes of combustion engines in industry.

15 It is a another object of the present invention according to another aspect to provide a highly reliable fuel injector, and specifically one that is not susceptible to thermal damage from the engine.

It is another object to provide a fuel injector 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 engine shut-down to replace the injector valve.

20 In accordance with these and other objectives, the present invention provides a fuel injector cartridge and a fuel injector incorporating the same, that uses the relatively cool gaseous fuel passing through the valve to directly cool the exposed surface of the valve and therefore limit the amount of heat transferred from the engine cylinder to the gas seals. The fuel injector
25 cartridge includes a valve body having an outer sleeve, and upper and lower guide collars mounted in the sleeve. The stem of an elongate valve extends up into and through the guide collars for radial retention. The valve is slidable through the guide collars for linear reciprocating movement between open and closed positions. The guide collars are separated by a cooling chamber

(which in the preferred embodiment doubles as a spring chamber) in which a portion of the valve stem is exposed. The sleeve has at least one cooling port (that takes the preferred form of a plurality of cross-holes) that allows the cool gaseous fuel to pulsate into and out of the cooling chamber correspondingly as the valve opens and closes. During opening of the valve, gas pressure drops in the gas passageway, resulting in a suction effect sucking the now heated gas (by virtue of direct contact with the valve, spring and guides) out of the cooling chamber. During closing of the valve, the pressure increases in the gas passageway forcing new more cool gaseous fuel into the cooling chamber.

10 It is an aspect of the present invention that the lower collar guide is a self lubricated high temperature graphite/carbon bushing. A bushing retainer is provided below the bushing to prevent any chips which may form from dropping into the outlet port.

15 It is another aspect of the present invention that a metal O-ring is used at the between the bottom of the cartridge and the cartridge housing to provide a seal axially between the cartridge housing and the cartridge body. The metal O-ring can withstand the high temperatures nearest to the cylinders of the engine and provide a highly reliably seal at the same time. Means in the preferred form of load washers engage the other axial end of the valve housing to axially compress the metal O-ring. A force in the rough neighborhood of about 10,000 is necessary to maintain a seal for high pressure fuel injection over about 300 psi.

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Brief Description Of The Drawings

Figure 1 is a cross-sectional view of a high pressure fuel injector assembly in accordance with a preferred embodiment of the present invention, illustrated in a closed position.

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

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

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

Figure 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.

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

Figure 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.

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

Figure 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.

Detailed Description Of The Preferred Embodiments

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 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. 5. The disclosed 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 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. 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 cartridge 22. A mounting flange 30 secures the entire assembly 20 to the engine 121 via conventional fasteners or bolts as shown 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 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 28 and the fuel injector cartridge 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 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 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 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 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 chamber (gas passageway 41) is 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 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 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 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, 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 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
5 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
10 the valve 46 in the closed position in the event of failure of the
electrohydraulic valve or the hydraulic pressure supply.

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
15 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. 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
20 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 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
25 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 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 returns the valve 46 to the closed position, causing hydraulic oil in the control chamber to flow to the lower pressure outlet 29.

5 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 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
10 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.
15 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
20 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
25 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 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 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 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 from the gas passageway 41 (including spring chamber 64) to the collection chamber 83. However, it will be appreciated that over the
5 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 increases the likelihood of such leakage occurring. The provision of the collection chamber 83 provides a fail
10 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 replaced, at the first sign of gas leakage past the seals.

15 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 O-rings 88, 89 between the connector tube 56
20 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 collected in the
25 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 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 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 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 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 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 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 valve
5 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
10 retrofit 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
15 allow the 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
20 gas seal and other 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.
25 This in turn causes relatively cool gas to pulsate 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.

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.

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 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 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 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,
5 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.
10 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-
15 ring 95.

FIGS. 6 and 7 illustrate one such 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
20 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
25 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
5 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
10 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
15 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
20 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 present invention in which an exclusive property or privilege is claimed are defined as follows:

1. A fuel injector for injecting gaseous fuel into an engine, comprising:
 - a tubular cartridge housing;
 - a valve body in the cartridge housing, including a sleeve, and upper and lower guide collars in the sleeve, the upper and lower guide collars being spaced apart and separated by a cooling chamber;
 - a gas passageway generally between the cartridge housing and the valve body for communicating gaseous fuel to the engine through an outlet port;
 - an elongate valve retained by the guide collars, having an exposed surface in the cooling chamber, the valve being slidable between open and closed positions to open and close the outlet port, respectively;
 - at least one cooling port defined in the sleeve connecting the gas passageway to the cooling chamber, the at least one cooling port located between the upper and lower guide collars communicating with the cooling chamber adapted to communicate gaseous fuel into and out of the cooling chamber, located for cooling the exposed surface of the valve; and
 - a gas seal carried by the valve body engaging the elongate valve, the cooling chamber located between the outlet port and the gas seal such that heat traveling through the elongate valve from the outlet port is cooled by the cooling chamber prior to reaching the gas seal.
2. The fuel injector of claim 1, further comprising a spring in the cooling chamber biasing the valve to the closed position.
3. The fuel injector of claim 2, further comprising an actuator body secured to the valve body, and a piston, the actuator body defining a control chamber containing the piston, the piston being separate from the valve and contacting an end of the valve, the piston being operated by a hydraulic working fluid to actuate the valve to the open position against the action of the spring.

4. The fuel injector of claim 1, 2 or 3, further comprising:
a metal O-ring axially between the valve body and the cartridge housing; and
means for urging the valve body axially against the tubular cartridge housing to compress the metal O-ring therebetween and provide a seal between the tubular cartridge housing and the valve body.
5. The fuel injector of any one of claims 1 to 4, wherein the at least one cooling port comprises a plurality of cross-holes radially about the sleeve.
6. The fuel injector of any one of claims 1 to 5, wherein the valve includes a closure member adapted to open and close the outlet port, the closure member being exposed to a cylinder of the engine and a thermal environment associated therewith wherein the gas seal has a thermal limit below the thermal environment of the closure member, gaseous fuel adapted to pulsate into and out of the cooling chamber and provide sufficient cooling of the exposed surface of the valve to prevent thermal deterioration of the gas seal.
7. The fuel injector of any one of claims 1 to 6, wherein the lower guide collar is a self lubricated high temperature, carbon bushing, and further comprising a bushing retainer between the carbon bushing and the outlet port preventing chips from the bushing from reaching the outlet port.
8. The fuel injector of any one of claims 1 to 7, wherein the sleeve defines the outlet port at its axial end and a conical valve seat surrounding the outlet port, the sleeve further comprising cross-holes between the lower guide collar and the outlet port communicating the gas passageway to the outlet port.
9. A fuel injector cartridge for insertion into a tubular cartridge housing for injecting gaseous fuel into an engine, a gas passageway being provided generally between the cartridge housing and the cartridge for communicating gaseous fuel to the engine through an outlet port, the cartridge comprising:

a valve body including a sleeve, and upper and lower guide collars in the sleeve, the upper and lower guide collars being spaced apart and separated by a cooling chamber;

an elongate valve retained by the guide collars having an exposed surface in the cooling chamber, the valve being linearly movable between open and closed positions to open and close the outlet port;

at least one cooling port defined in the sleeve connecting the gas passageway to the cooling chamber, the at least one cooling port located between the upper and lower guide collars communicating with the cooling chamber adapted to communicate gaseous fuel into and out of the cooling chamber for cooling the exposed surface of the valve; and

a gas seal carried by the valve body engaging the elongate valve, the cooling chamber located between the outlet port and the gas seal such that heat traveling through the elongate valve from the outlet port is cooled by the cooling chamber prior to reaching the gas seal.

10. The fuel injector cartridge of claim 9, further comprising a spring in the cooling chamber biasing the valve to the closed position.

11. The fuel injector cartridge of claim 10, further comprising an actuator body secured to the valve body and a piston, the actuator body defining a control containing the piston, the piston being separate from the valve and contacting the end of the valve, the piston being operated by a hydraulic working fluid to actuate the valve to the open position against the action of the spring.

12. The fuel injector cartridge of claim 9, 10 or 11, further comprising:

a metal O-ring axially between the tubular cartridge body and the valve housing; and
means for urging the valve housing axially against the tubular cartridge body to compress the metal O-ring therebetween and provide a seal between the tubular cartridge body and the valve housing.

13. The fuel injector cartridge of any one of claims 9 to 12, wherein the at least one cooling port comprises a plurality of cross-holes radially about the sleeve.

14. The fuel injector cartridge of any one of claims 9 to 13, wherein the valve includes a closure member adapted to open and close the outlet port, the closure member being exposed to a cylinder of the engine and a thermal environment associated therewith wherein the gas seal has a thermal limit below the thermal environment of the closure member, gaseous fuel adapted to pulsate into and out of the cooling chamber in correspondence with the opening and closing of the valve and provide sufficient cooling of the exposed surface of the valve to prevent thermal deterioration of the gas seal.

15. The fuel injector cartridge of any one of claims 9 to 14, wherein the lower guide collar is a self lubricated, high temperature, carbon bushing, and further comprising a bushing retainer between the carbon bushing and the outlet port preventing chips from the bushing from reaching the outlet port.

16. The fuel injector cartridge of any one of claims 9 to 15, wherein the sleeve defines the outlet port at its axial end and a conical valve seat surrounding the outlet port, the sleeve further comprising cross-holes between the lower guide bushing and the outlet port communicating the gas passageway to the outlet port.

17. A fuel injector for injecting gaseous fuel into an engine, comprising:

- a tubular cartridge housing;

- a valve body in the cartridge housing, including a sleeve, and upper and lower guide collars in the sleeve, the upper and lower guide collars being spaced apart, a cooling chamber in the valve body adjacent the lower guide collar;

- a gas passageway generally between the cartridge housing and the valve body for communicating gaseous fuel to the engine through an outlet port;

- an elongate valve retained by the guide collars, having an exposed surface in the cooling chamber, the valve being slidable between open and closed positions to open and close the outlet port, respectively;

at least one cooling port defined in the sleeve adapted to communicate gaseous fuel into the cooling chamber for cooling the exposed surface of the valve;

a spring in the cooling chamber biasing the valve to the closed position; and

an actuator body secured to the valve body, and a piston, the actuator body defining a control chamber containing the piston, the piston being separate from the valve and contacting an end of the valve, the piston being operated by a hydraulic working fluid to actuate the valve to the open position against the action of the spring.

18. A fuel injector cartridge for insertion into a tubular cartridge housing for injecting gaseous fuel into an engine, a gas passageway being provided generally between the cartridge housing and the cartridge for communicating gaseous fuel to the engine through an outlet port, the cartridge comprising:

a valve body including a sleeve, and upper and lower guide collars in the sleeve, the upper and lower guide collars being spaced apart, a cooling chamber in the valve body adjacent the lower guide collar;

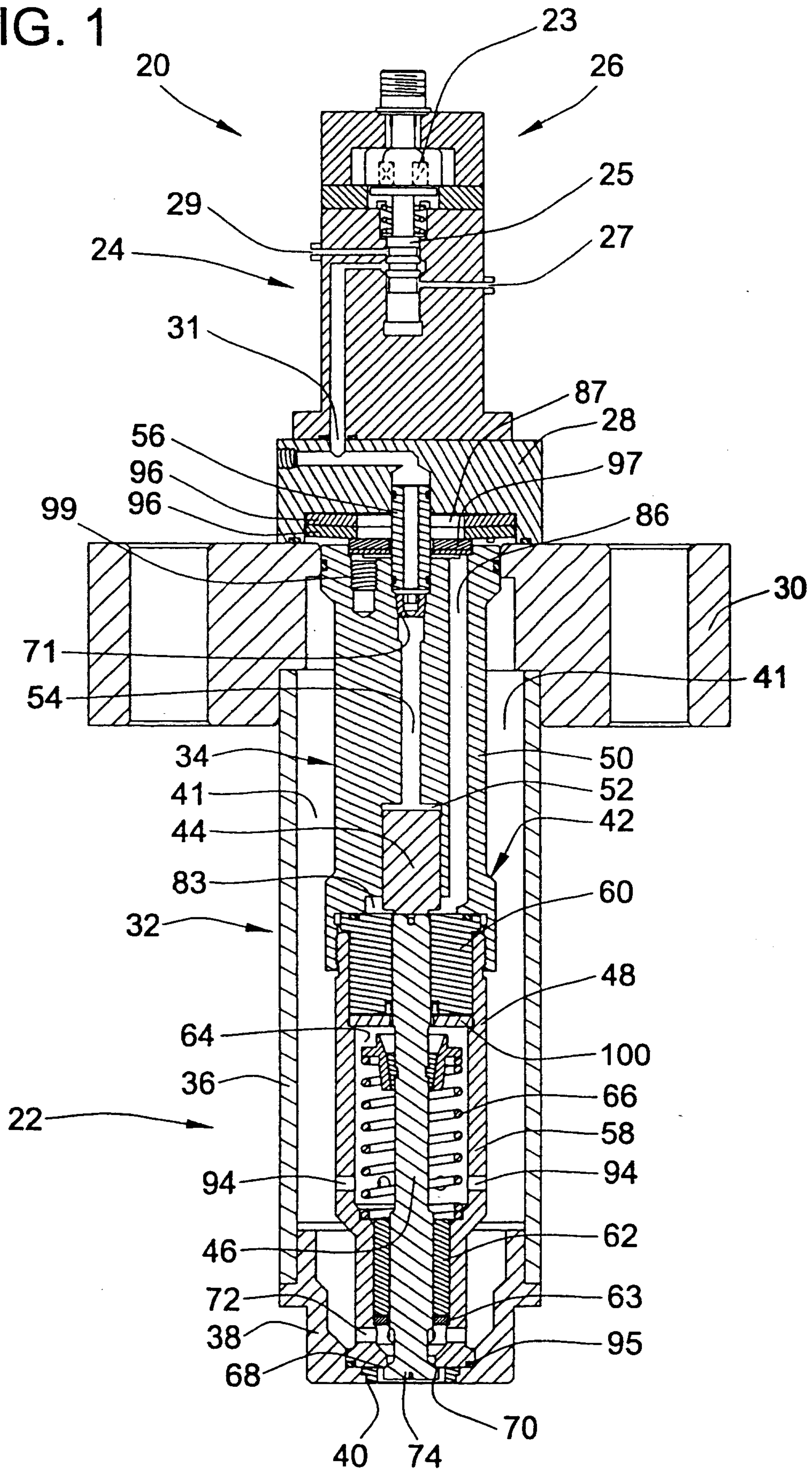
an elongate valve retained by the guide collars having an exposed surface in the cooling chamber, the valve being linearly movable between open and closed positions to open and close the outlet port;

at least one cooling port defined in the sleeve adapted to communicate gaseous fuel into the cooling chamber for cooling the exposed surface of the valve;

a spring in the cooling chamber biasing the valve to the closed position; and

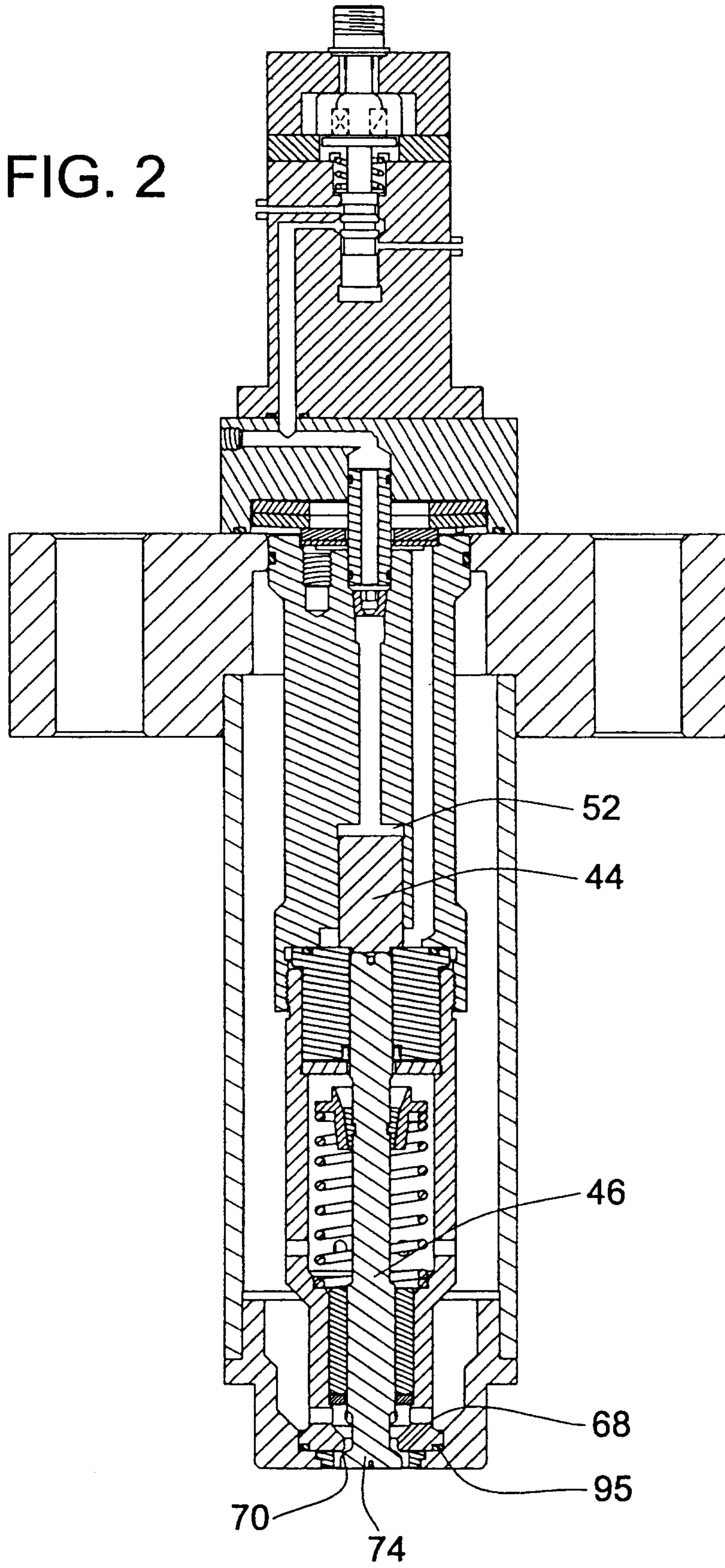
an actuator body secured to the valve body and a piston, the actuator body defining a control containing the piston, the piston being separate from the valve and contacting the end of the valve, the piston being operated by a hydraulic working fluid to actuate the valve to the open position against the action of the spring.

FIG. 1



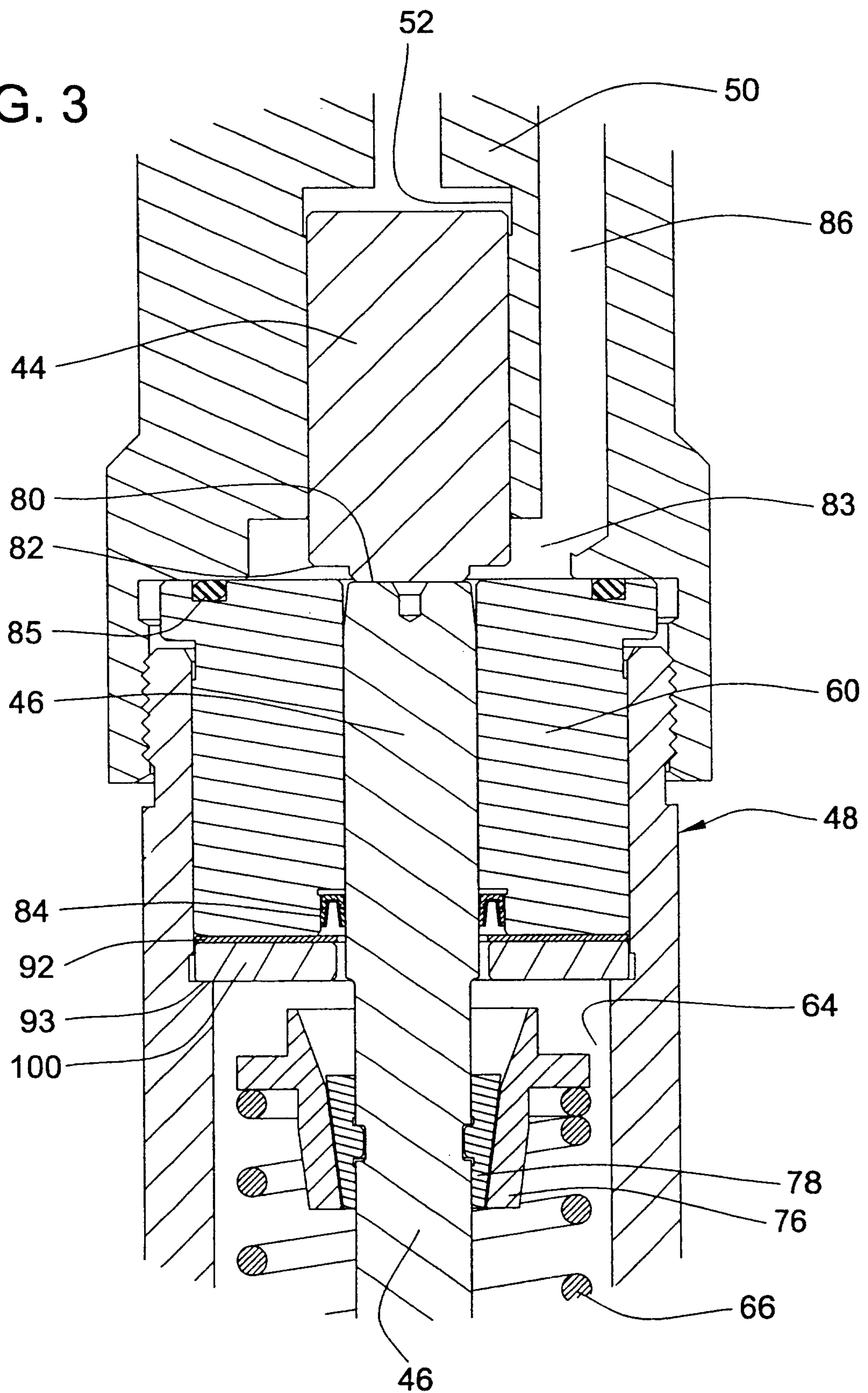
Marks & Clerk

FIG. 2



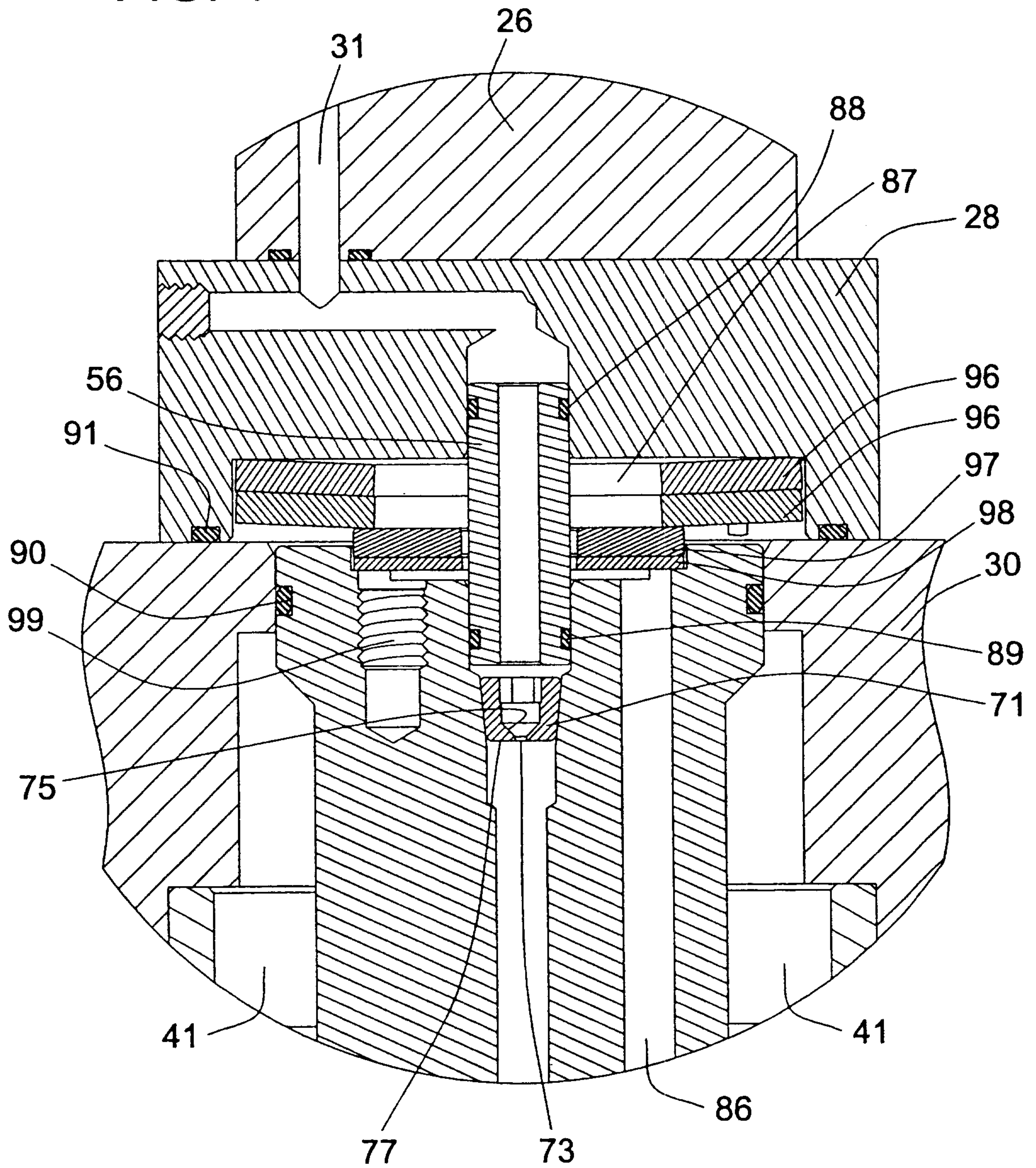
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FIG. 3



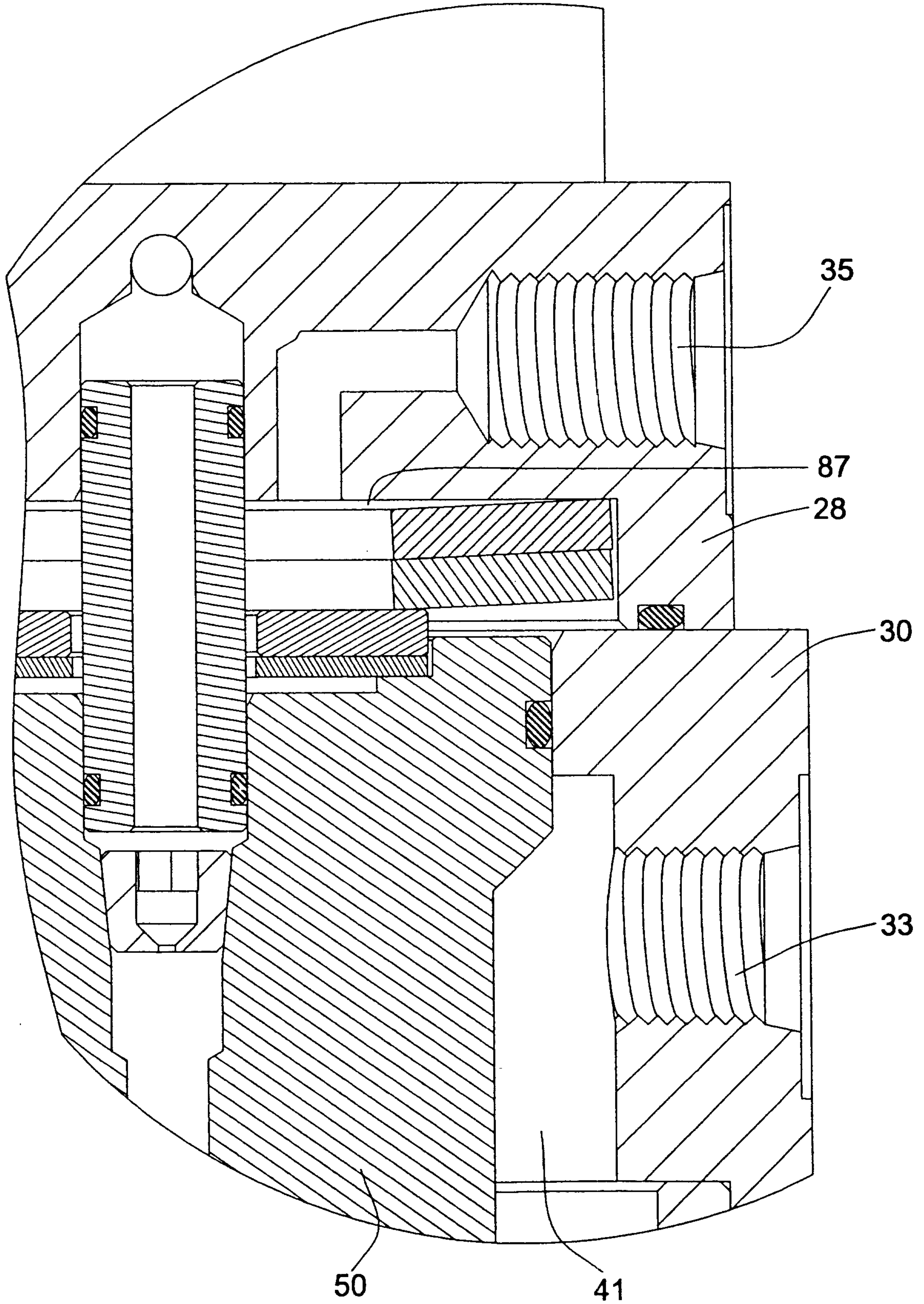
Wicks & Clark

FIG. 4

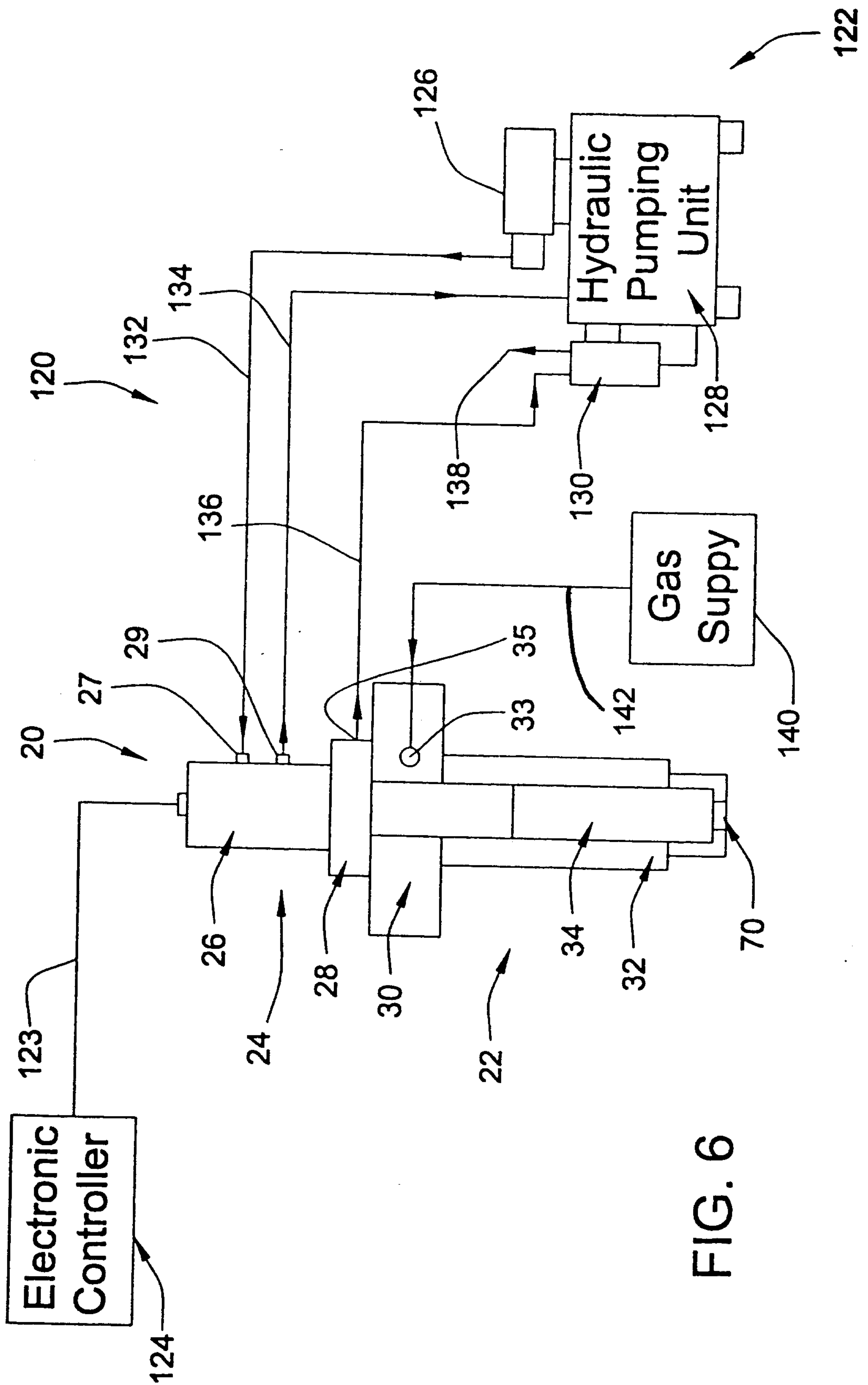


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FIG. 5

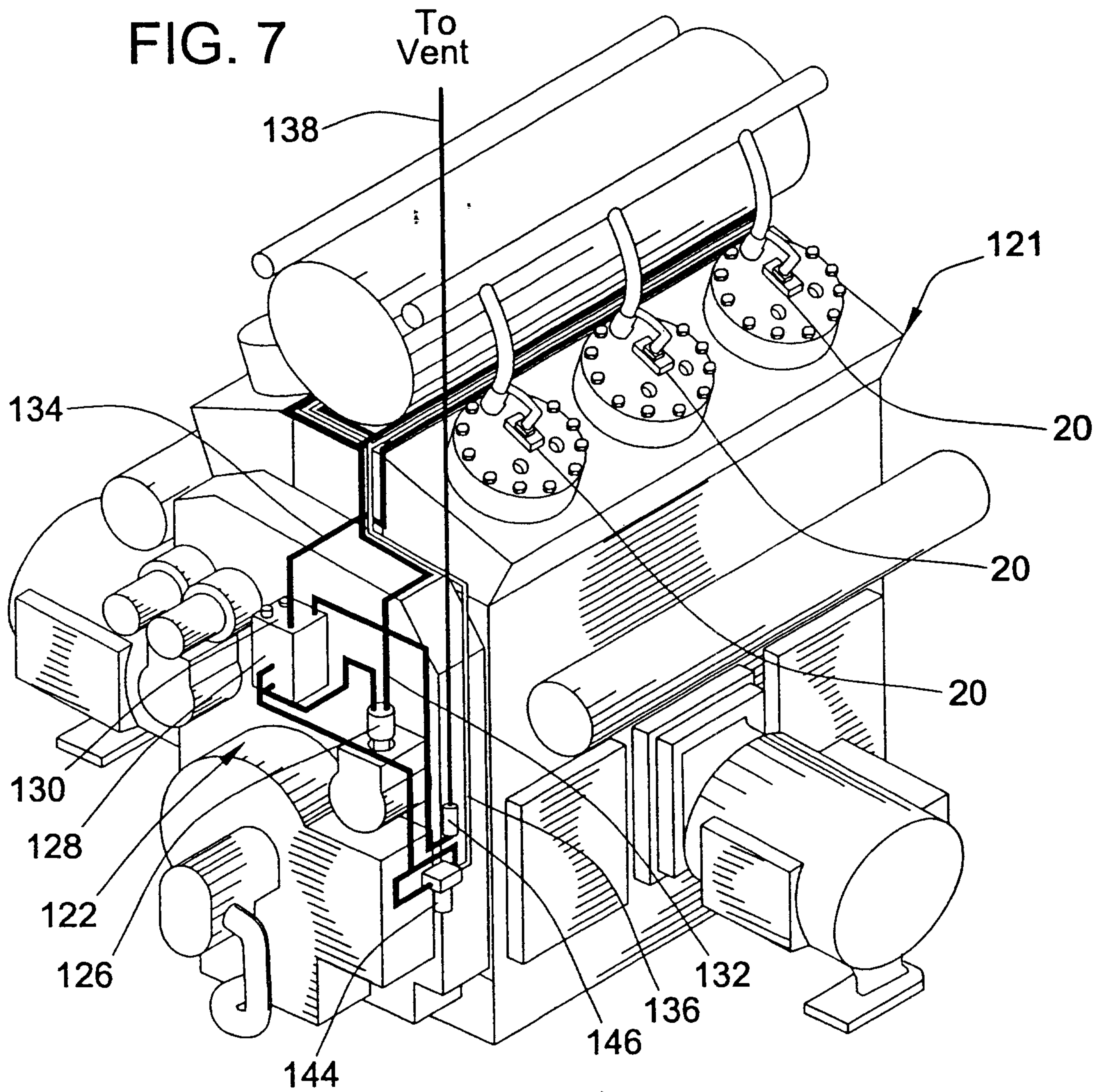


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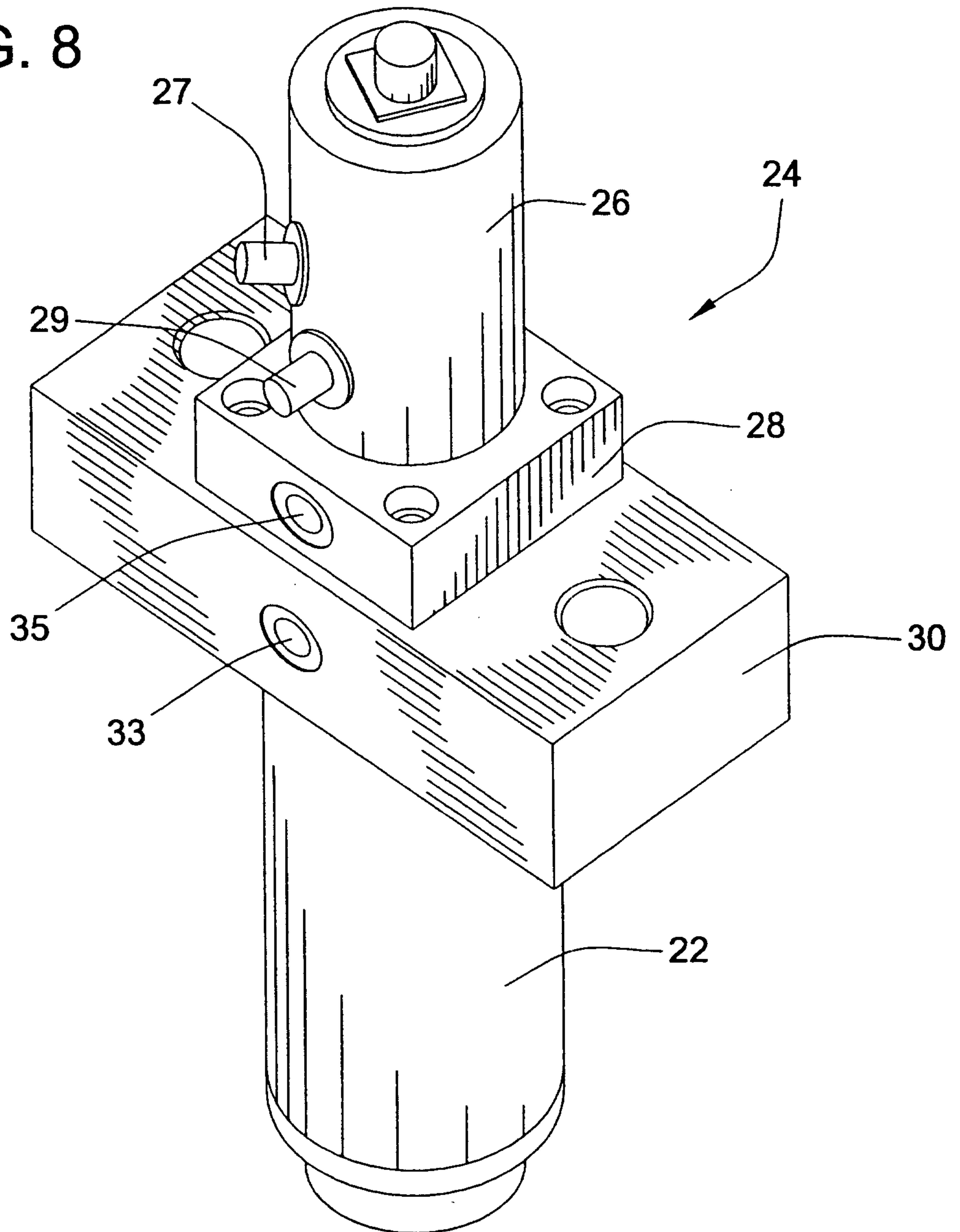
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FIG. 7



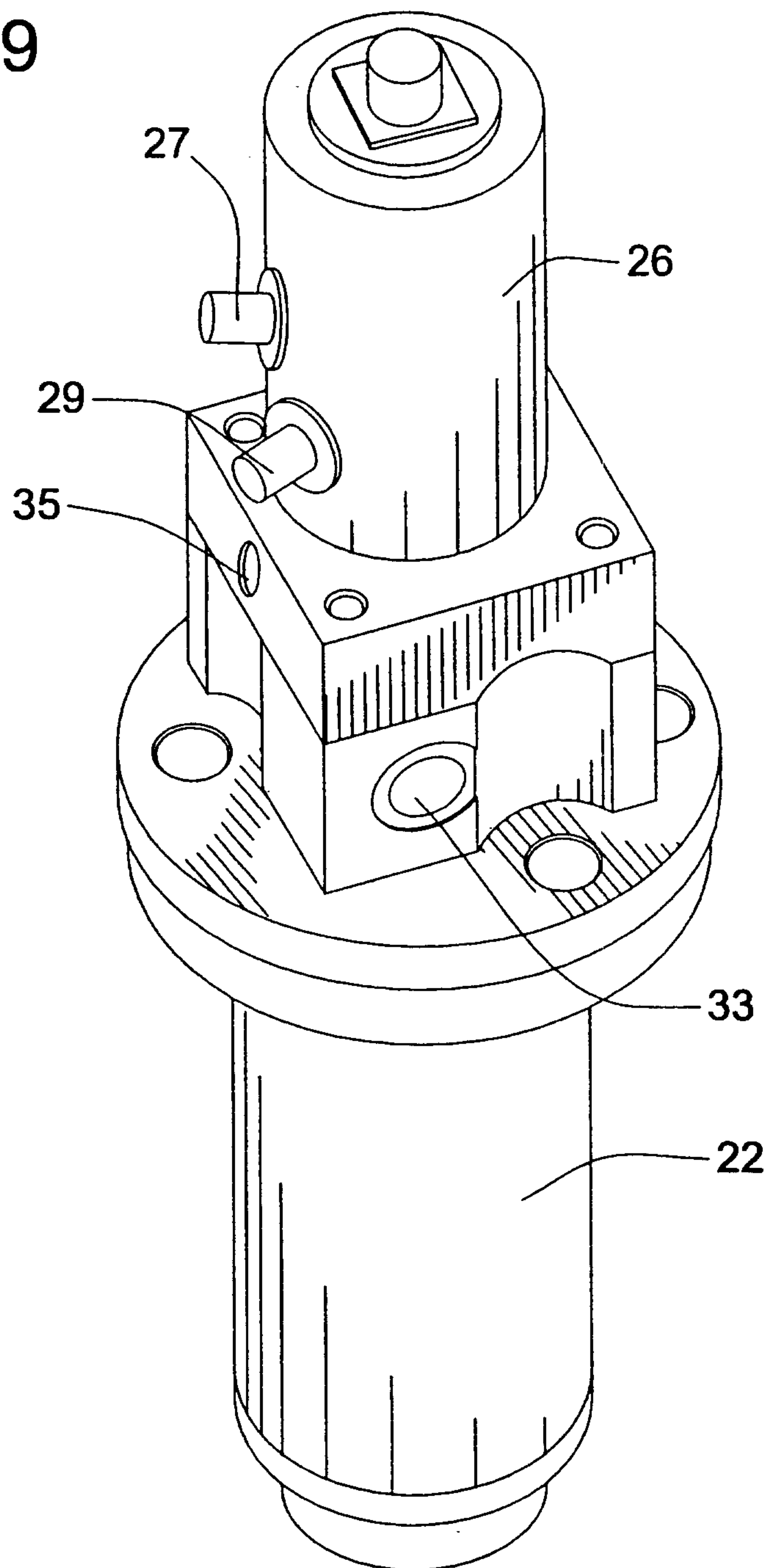
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FIG. 8



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FIG. 9



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