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(54) **Title:** A LARGE TWO-STROKE DIESEL ENGINE WITH HYDRAULICALLY ACTUATED EXHAUST GAS VALVES

(57) **Abstract:** The present invention relates to a large two-stroke diesel engine (1) of the cross-head type in which the exhaust valves (11) are hydraulically actuated by providing hydraulic actuators (19) associated with each of the exhaust valves (11) with high pressure hydraulic fluid. Fuel oil or heavy fuel oil can be used as hydraulic fluid. The invention also relates to a control valve (25) for use in such an engine (1) , to a hydraulic system for such an engine (1) and to an hydraulic actuator (19) for such an engine (1).

A LARGE TWO-STROKE DIESEL ENGINE WITH HYDRAULICALLY ACTUATED EXHAUST GAS VALVES

The present invention relates to a large two-stroke diesel engine of the cross-head type in which the exhaust gas valves are hydraulically actuated by providing hydraulic actuators associated with each of the exhaust valves with high pressure hydraulic fluid.

BACKGROUND ART

Large two-stroke diesel engines of the cross-head type are typically used for marine propulsion and as prime movers in power plants. Not only due to sheer size, these combustion engines are constructed differently from any other combustion engines. The two stroke principle and the use of heavy fuel oil with a viscosity below 700cSt at 50°C (the oil does not flow at room temperatures) make them a class of their own in the engine world.

In many conventional engines of this type exhaust gas valves and the fuel injection system are driven with a rotating cam coupled directly to the engine crankshaft. Two-stroke engines use scavenge ports to control the inlet of air into the cylinders, and consequently the inlet timing is rigidly linked to crank angle. This leaves only the exhaust valve and the fuel injection to be converted to a more flexible control.

Fuel consumption, reliability and power output requirements for this type of engine are extremely high. In the recent past, environmental requirements have lead to a demand for a reduction in exhaust gas emissions. In order to fulfill these sometimes contradicting requirements it is necessary to have full and flexible control over the fuel injection timing and dosage as well as full and flexible control over the opening and closing timing and the degree of opening of the exhaust valves as opposed to the conventional rotating cam driven exhaust valves and fuel injectors.

Due to the size of this type of engine, electric actuators cannot be used to operate the exhaust valves. Such exhaust valves can weigh up to 450 Kg in the largest of these engines.

The ME engine range by MAN B&W Diesel A/S are large two stroke diesel engines of the crosshead type with electro-hydraulically controlled exhaust valves and electro-hydraulically activated fuel injection. The hydraulic system is operated with oil from the engine lubrication system. The lubrication oil system is operated with a 3 to 4 bar low pressure pump. Another pump of a high pressure type delivers lubrication oil at about 200 bar to a common rail. The lubrication oil from the common rail is directed via a hydraulic valve to a fuel oil booster that boosts the 200 bar pressure in the common rail up to the required 1000 bar in the fuel line. The fuel line is heated to about 90 to 150 °C to ensure that the fuel oil can flow and has the appropriate viscosity. The lubrication oil from the common rail is directed via a timing valve to a hydraulic exhaust valve actuator to operate the exhaust valve.

The lubrication oil from the lubrication system of these engines is however as such not sufficiently clean to be used in the common rail hydraulic system. The lubrication oil has therefore to be filtered in order to remove any particles above 5-10 μ before it can be pumped into the common rail.

The Wärtsilä/Sulzer RT-flex range of engines are a large two stroke diesel engines of the crosshead type with electro-hydraulically controlled exhaust valves and electro-hydraulically activated fuel injection. The hydraulic system for the valve actuation is operated with a dedicated hydraulic oil. The lubrication system is completely separate from the hydraulic system.

DISCLOSURE OF THE INVENTION

On this background, it is an object of the present invention to provide a large two-stroke diesel engine of the cross-head type with improved control over the fuel injection.

This object is achieved in accordance with claim 1 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, each of the injectors being operated with fuel from the common rail, and a proportional valve associated with each cylinder, whereby the proportional valves control the flow of fuel from the common fuel rail to the respective injectors.

The use of a proportional control valve allows the profile the fuel injection timing, dosage and profile to be controlled more accurately and flexibly. Further, the use of a proportional control valve allows rate shaping and pre-injection to be performed without additional equipment, e.g. the rate shaping is substantially exclusively created via the control signal to the proportional valve.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with an improved control over the exhaust valve actuation.

This object is achieved in accordance with claim 10 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, each of the injectors being operated with fuel from the common rail, and a proportional control valve associated with each cylinder, whereby the proportional control valves control the flow of fuel from the common fuel rail to the respective hydraulic valve actuators.

The use of a proportional control valve provides full and flexible control over the opening and closing timing and the degree of opening of the exhaust valves. Further, the position of the exhaust valves are be controlled in a flexible manner for each cylinder, thus allowing e.g. the exhaust valve of a particular cylinder to be slightly open during ist compression stroke to facilitate engine startup.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with a more simple and flexible overall hydraulic system.

This object is achieved in accordance with claim 18 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, and a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, each of the injectors being operated with fuel from the common rail, the hydraulic valve actuators being connected with the common rail via respective supply conduits, whereby the supply conduits are provided with heating means.

Thus, when heavy fuel oil, also known as HFO or the like is used a hydraulic medium, the HFO is kept at the correct viscosity.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with a hydraulic exhaust valve actuating system that can operate across a large temperature span.

This object is achieved in accordance with claim 30 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, and a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, each of the injectors being operated with fuel from the common rail, each of the hydraulic valve actuators being operated with fuel from the common rail, whereby the hydraulic valve actuator is provided with means to compensate for dimensional changes caused by:

operation at different temperatures,
reconditioning, i.e. grinding of the valve seat, and
production tolerances.

Thus, the hydraulic valve actuator will assume the correct position across a large temperature span and ensure that the valve head lands correctly at the valve seat at all times.

It is another object of the present invention to provide a method of controlling the temperature of fuel in a supply conduit of a large two-stroke diesel engine of the cross-head type.

This object is achieved in accordance with claim 35 by providing a method of controlling the temperature of fuel in a pressure conduit of a large two-stroke diesel engine of the cross-head type, the pressure conduit connecting a common fuel rail to a hydraulic exhaust valve actuator, and the method comprises the step of controlling the temperature of the fuel entering the pressure conduit to maintain the temperature gradient of the fuel below a predetermined threshold during changes in operating temperature of the fuel.

Thus, fuel can be used as a hydraulic medium for operating exhaust valve actuators that are sensitive to changes in operating temperature.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with a hydraulic exhaust valve actuating system that can operate with a large variety of hydraulic liquids.

This object is achieved in accordance with claim 40 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, and a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, each of said injectors being operated with fuel from said common rail, said hydraulic valve actuators being connected with said common rail via respective hydraulic lines and eventually other hydraulic components such as valves, characterized in that the static gaskets that seal the connection between conduits and other hydraulic components of the engine, and dynamic gaskets in the valve actuator are made of: cast iron, steel, Polytetraflourethylene (PTFE), Flouro Rubber, (FPM), Copolymer (NBR), Nitril Rubber, Poly(dimethylsiloxane) (SI) or combinations and/or mixtures thereof.

Selecting the gaskets from these materials allows non-dedicated hydraulic fluids such as fuel to be used in the hydraulic system without the fuel damaging the gaskets.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with a hydraulic exhaust valve actuating system that can operate with a large variety of hydraulic liquids.

This object is achieved in accordance with claim 41 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, and a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, a supply conduit and valve means associated with each cylinder for delivering fuel from the common rail to the respective injectors a supply conduit and valve means associated with each cylinder for delivering fuel from the common rail to the respective hydraulic valve actuators, and a heated return conduit for transporting fuel from the hydraulic valve actuators to a fuel tank or to a conduit leading to the intake of the high pressure pump.

Thus, HFO with a low viscosity can be used as hydraulic medium.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with a hydraulic exhaust valve actuating system that can operate across a large temperature span.

This object is achieved in accordance with claim 48 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a hydraulic valve actuator associated with each of the exhaust valves, a common fuel rail having one or more accumulators connected thereto, and a high pressure fuel pump feeding fuel under high pressure to the common fuel rail, a pressure conduit and valve means associated with each cylinder for delivering fuel from the common rail to the respective injectors a supply conduit and valve means associated with each cylinder for delivering fuel from the common rail to the respective hydraulic valve actuators, and a return conduit for transporting fuel from the hydraulic valve actuators to a fuel tank or to a conduit leading to the intake of the high pressure pump, wherein at least one of the conduits includes means for neutralizing the effects of dimensional changes of the conduits caused by changes in operating temperature.

Thus, the hydraulic system will be able to operate across a large temperature span and ensure that conduits are not mechanically stressed due to temperature induced dimensional changes.

It is another object of the present invention to provide a new use of a proportional valve.

This object is achieved in accordance with claim 51 by providing a use of a proportional valve to control the flow of fuel from a common fuel rail of a large two-stroke diesel engine of the cross-head type to the fuel injectors and/or to a fuel operated component.

It is another object of the present invention to provide an electrically controlled valve for controlling the flow of fuel from a common fuel rail of a large two stroke diesel engine of the crosshead type to one or more fuel operated or fuel consuming engine components.

This object is achieved in accordance with claim 54 by providing an electrically controlled valve for controlling the flow of fuel from a common fuel rail of a large two stroke diesel engine of the crosshead type to one or more fuel operated or fuel consuming engine components, the electronically controlled valve comprising a valve housing and a solenoid, whereby said solenoid is thermally insulated from the valve housing.

It is another object of the present invention to provide a large two-stroke diesel engine of the cross-head type with an improved circulation for the hydraulic system during engine stops.

This object is achieved in accordance with claim 58 by providing a large two-stroke diesel engine of the cross-head type, comprising: a crankcase frame supporting a crankshaft and a cylinder frame mounted on the crankcase frame, a plurality of cylinders carried by the cylinder frame, each cylinder being provided with at least one fuel injector and with at least one exhaust valve, a common fuel rail, and a high pressure fuel pump feeding fuel under high pressure to the common fuel rail during engine operation, a supply conduit and valve means associated with each cylinder for delivering fuel from said common rail to the respective injectors, said high pressure fuel pump being mechanically driven by the crankshaft during engine operation and electrically driven by an electric motor during engine stops to provide fuel at a low pressure for circulating the fuel through said supply conduit and/or the common rail and/or through other engine components operated with fuel.

By using the high pressure pump as both high pressure source and as low pressure source for circulation during engine stops the number of component is reduced, thereby rendering the overall construction and maintenance costs more competitive.

It is another object of the present invention to provide a hydraulically actuated gas exchange valve for an internal combustion engine with an improved air spring.

This object is achieved in accordance with claim 63 by providing a hydraulically actuated gas exchange valve for an internal combustion engine comprising: a stationary valve housing, a gas exchange valve that is movable between a seated position and an unseated position and includes an elongated valve stem with a valve head on one end and a free extremity on the opposite end, a hydraulic actuator, the hydraulic actuator comprising a piston acting on the free extremity of the valve stem for urging the valve to an unseated position when the hydraulic actuator is supplied with pressurized hydraulic fluid, a pneumatic spring urging the valve to the seated position, the pneumatic spring comprising: a cylinder secured to the valve stem, the cylinder being closed in the direction towards the free extremity of the valve stem and open in the direction towards the valve head, and a matching stationary piston received in the cylinder, the piston being secured to the valve housing and forming together with the cylinder a spring chamber for the pneumatic spring.

This construction of the air spring reduces the chance of hydraulic medium from the actuator entering the spring chamber.

It is another object of the present invention to provide an improved hydraulically actuated gas exchange valve for an internal combustion engine.

This object is achieved in accordance with claim 66 by providing a hydraulically actuated gas exchange valve for an internal combustion engine comprising: a stationary valve housing, a gas exchange valve that can travel between a seated position in which the valve is closed and an unseated position in which the valve is open and includes an elongated valve stem with a valve head on one end and a free extremity on the opposite end, a hydraulic actuator, the hydraulic actuator comprising a piston acting on the free extremity of the valve stem for urging the valve to an unseated position when the hydraulic actuator is supplied with pressurized hydraulic fluid, a pneumatic spring urging the valve to the seated position, whereby the length of travel of the valve in the in the opening direction is determined by the balance of the opposing forces of the hydraulic actuator and the air spring.

Thus the actuator does not need to be provided with an end of stroke limiter at the end of the opening stroke, and there is no need to suddenly cut-off the supply of high pressure hydraulic fluid at the end of the opening stroke. The lack of an end of stroke limiter reduces mechanical load and shocks, whilst the absence of a sharp cut-off of the high pressure hydraulic fluid avoids potentially damaging hydraulic shockwaves.

It is another object of the present invention to provide a hydraulic actuator for a gas exchange valve of an internal combustion engine that can operate accurately across a large temperature span.

This object is achieved in accordance with claim 69, by providing a hydraulic actuator for a gas exchange valve of an internal combustion engine comprising: a stationary cylinder with a proximate end and an open distal end and including a pressure chamber that can be alternately connected to a source of high-pressure hydraulic fluid or to a return line by valve means, a piston with a proximate end received in said primary pressure chamber and a distal end acting on the free extremity of the valve stem of said valve for urging the valve to an unseated position when the pressure chamber is connected to said source of high-pressure hydraulic fluid, said piston comprising a first part and a second part, said first part extending from the distal end towards the proximate end and said second part being disposed at the proximate end, said second part slidably engages the first part to thereby form a compensation chamber between said first part and said second part, a spring means urging said first part and second part away from one other to thereby expand said compensation chamber, a first flow path between said compensation chamber and said pressure chamber, said first flow path being open only when said second part is located in a small predetermined axial range at the proximate end of said cylinder to allow an excess of hydraulic fluid to escape from said compensation chamber, a second flow path between said compensation chamber and said pressure chamber, said second flow path allowing said compensation chamber to be refilled under the influence of said spring means.

The compensation chamber ensures that the actuator piston will always start at a position within an axial range that results in a return to a correct position with the valve head landing accurately on the valve seat. The position within the axial range depends on the change of volume of the compensation chamber during the opening period, and or during the closed position. These volume changes can during both parts of the cycle be positive or negative.

It is another object of the present invention to provide a hydraulic actuator for a gas exchange valve of an internal combustion engine that can overcome a large counterforce at the start of the opening process and deliver a controlled force in the opening direction once the gas exchange valve has opened.

This object is achieved in accordance with claim 74, by providing a hydraulic actuator for a gas exchange valve of an internal combustion engine comprising: a stationary cylinder including a pressure chamber that can be alternately connected to source of high-pressure hydraulic fluid or to a return line via a first port in said cylinder, a piston received in said pressure chamber and acting on the free extremity of the valve stem of said valve for urging the valve to an unseated position when the pressure chamber is connected to said source of high-pressure hydraulic fluid, said piston being axially movable between a retracted position in which the valve is seated to an extended position in which the valve is open, said piston having a first effective area on which said pressurized hydraulic fluid in said pressure chamber acts to urge the piston towards the extended position when said piston is located between said retracted position and a predetermined intermediate position, the piston having a second effective area, smaller than said first effective area, on which said pressurized hydraulic fluid in said pressure chamber acts to urge the piston towards the extended position when said piston is located between said intermediate position and said extended position.

The combined action of the first and second effective piston areas, i.e. the overall area of the piston on which the high pressure fluid acts, results in a large actuator force during the first part of the opening movement of the gas exchange valve, whilst the effect of the second effective piston area alone results in a well controlled course of the remainder of the opening movement of the gas exchange valve.

Further objects, features, advantages and properties of the large two-stroke diesel engine and methods of operating thereof accordingly will become apparent from the detailed description.

BRIEF DESCRIPTION OF THE DRAWINGS

In the following detailed portion of the present description, the invention will be explained in more detail with reference to the exemplary embodiments shown in the drawings, in which

Fig. 1 shows a frontal view of an outline of a cylinder in a two-stroke crosshead engine with a cylinder cover,
Fig. 2 shows a cross-sectional view of an outline of a cylinder in the engine shown in Fig. 1,
Fig. 3 is a diagrammatic representation the hydraulic and of the lubrication system of the engine shown in Fig. 1,
Fig. 4 is another embodiment of the diagrammatic representation of the hydraulic and of the lubrication system shown in Fig. 3,

Fig. 5 is a cross-sectional view of a pressure pipe useful herein,

Fig. 6 is another cross-sectional view of an alternative pressure pipe useful herein,

Fig. 7 is a longitudinal sectional view of a first embodiment of the hydraulically actuated exhaust valve in the cylinder in Fig. 2 with the valve seated and the piston in its retracted position,

Fig. 8 shows a typical opening profile of the exhaust valve actuator according to the invention,

Fig. 9 shows on a larger scale, a cross-sectional view of the actuator shown in Fig. 7 with the piston in a partially extended position,

Fig. 10 shows the same view as Fig. 9 with the piston in a retracted position and the piston cap in its highest position within its axial range,

Fig. 11 is a detailed view on the upper section of the actuator with the piston in substantially a retracted position and the piston cap positioned in the substantially highest position of its axial range,

Fig. 12 is a detailed view on the upper section of the actuator with the piston in a retracted position and the piston cap positioned in the substantially lowest position of its axial range, and

Fig. 13 is a detailed view on the upper section of the actuator with the piston in a retracted position and the piston cap positioned substantially in the middle of its axial range.

DETAILED DESCRIPTION

Fig. 1 shows an engine 1 according to the invention. The engine is a low-speed two-stroke crosshead diesel engine, which may be a propulsion engine in a ship or a prime mover in a power plant. These engines have typically from 6 up to 16 cylinders in line. The engine is built up from a bedplate 2 with the main bearings for the crankshaft 3. The bedplate is divided into sections of suitable size in accordance with production facilities available. A welded design A-shaped crankcase frame 4 is mounted on the bedplate. A cylinder frame 5 is mounted on top of the crankcase frame 4. Staybolts (not shown) connect the bedplate to the cylinder frame and keep the structure together. The cylinders 6 are carried by the cylinder frame 5.

Fig. 2 shows a cylinder 6 of an internal combustion engine. The cylinder 6 is of the uniflow type and has scavenge air ports 7 located in an airbox 8, which from a scavenge air receiver 9 (Fig. 1), is supplied with scavenge air pressurized by a turbocharger 10 (Fig. 1). A not shown crosshead connects the piston rod 14 with the crankshaft 3 (Fig. 1).

An exhaust valve 11 is mounted centrally in the top of the cylinder in a cylinder cover 12. At the end of the expansion stroke the exhaust valve 11 opens before an engine piston 13 passes down past the scavenge air ports 7, whereby the combustion gases in a combustion chamber 15 above the piston 13 flow out through an exhaust passage 16 opening into an exhaust receiver 17 and the pressure in the combustion chamber 15 is relieved. The exhaust valve 11 closes again during the upward movement of the piston 13 at an adjustable moment that may e.g. depend on the desired effective compression ratio for the subsequent combustion. During the closing movement, the exhaust valve is driven upwards by a pneumatic spring 18.

In consideration of the durability of the valve 11 and of an advantageous, accurate control of the conditions in the combustion chamber and thus of the efficiency of the engine, the exhaust valve 11 may advantageously be controlled very accurately.

The exhaust valve 11 is opened by means of a hydraulically driven actuator 19. The hydraulic fluid (fuel) is supplied through a pressure conduit 20 connecting an inlet port on the actuator 19 with a control port on the top surface of a distributor block 21 supported by a console 22. A return conduit 43 connects an outlet port on the actuator 19 to a return port on the top surface of the distributor block 21.

Each cylinder 6 is provided with two or three injectors 23 (only one is shown) connected by a ring conduit (not shown). The fuel is supplied from distributor block 21 to the injectors 23 through feed conduit 24. The injectors 23 are connected to a return port on the distributor block 21 via a return conduit 49.

The console 22 is connected to a return conduit leading to a supply line and to a common fuel rail (40 in Fig. 3, not shown in Fig. 2)

The distributor block 21 carries a proportional control valve 25 that controls the connection of the ports on top of the distributor block 21 with the return conduit (43 in Fig. 3) and common fuel rail 40 (Fig. 3) (not shown) in the console 22.

In the console 22 a channel 41 (Fig. 3) branching off from the common fuel rail 40 passes the pressurized hydraulic fluid to an inlet port on the proportional control valve 25.

The fuel in the common fuel rail 40 (Fig. 3) is used as a hydraulic fluid to both drive the valve actuator 19 and to feed the injectors 23. The pressure in the common rail 40 varies in dependence of the operating state of the engine 1 such as running speed and load condition. Typically the pressure in the common fuel rail 40 varies between 600 Bar and 2000 Bar.

Each cylinder 5 of the engine 1 is associated with an electronic control unit 26 which receives general synchronizing and control signals through wires 27 and transmits electronic control signals to the proportional control valve 25, among others, through a wire 28. There may be one control unit 26 per cylinder, or several cylinders may be associated with the same control unit (not shown). The control units 26 may also receive signals from an overall control unit (not shown) common to all the cylinders.

With reference to Fig. 3 the hydraulic system and the lubrication system of the engine 1 are diagrammatically shown. The hydraulic system serves as both a fuel injection system and as an exhaust valve actuation system.

The lubrication system comprises a lubrication tank, a filter and an electrically driven low pressure pump. The lubrication system is completely separate from the hydraulic system.

The hydraulic system is operated with fuel, typically HFO (both water emulsified and non-water emulsified). Water is often emulsified into the HFO in order to reduce NOx emissions. The emulsification takes place in a separate emulsification unit (not shown). The fuel for the operation of the engine is stored in a heated tank 29. The fuel used is typically so called heavy fuel oil (HFO) with a viscosity of 500 to 700 cSt at 50°C and cannot flow at room temperature. The HFO in the tank is kept at about 50°C at substantially all times, i.e. also during engine stops. Typically ships with the present type of engine are provided with generator sets (Genset), i.e. smaller diesel engines that provide electrical power and heat for the ship and for the main engine during stops of the main engine.

From the heated tank the HFO is lead to a filter or centrifuge 30 and to a preheater 31. The temperature of the HFO leaving the preheater 31 is controlled in accordance with the operating status and the grade of HFO. During engine stops, when the HFO is circulated at low pressure through the hydraulic system, the temperature of the HFO is kept in the range of 45 to 60 °C. During engine operation the temperature of the HFO leaving the preheater 31, is kept between 90 and 150 °C, depending on the viscosity of the HFO. A sensor (not shown) measures the viscosity of the HFO just downstream of the preheater 31 (or another suitable place). The temperature of the HFO leaving the preheater 31 is typically controlled to result in a viscosity at the measuring point in the range of 10 to 20 cSt.

A forked intermediate conduit 32 connects the preheater to both a high pressure fuel pump 33 and an auxiliary low pressure circulation pump 34. Non-return valves 35 are disposed in the conduits downstream of each pump to prevent back-suction.

During engine operation the high pressure fuel pump 33 is driven by gearwheel 36 on the crankshaft 3 via a gearwheel 37. Hereby, the high pressure fuel pump 33 produces a nominal pressure of 1000 to 1500 bar, but the pressure may fluctuate between 600 and 2000 bar in dependence of the operating conditions.

During engine stops the auxiliary low pressure circulation pump 34 is driven by an electric motor 38. Hereby, a pressure of about 3 to 10 bar is delivered for circulating the HFO through the hydraulic system during engine stops.

The common fuel rail 40 extends along all cylinders and the connections to the cylinders 6 that are not shown in Fig. 3 are symbolized by the short upward lines that extend from the common rail.

The cylinder 6 shown in Fig. 3 is provided with HFO through a supply conduit 41 that branches off from the common rail 40 and leads to an inlet port of the proportional control valve 25. The supply conduit 41 is provided with a number of fluid accumulators 42 that deliver most of the fluid volume when the proportional control valve 25 opens and are post-fed from the common rail 40 while the proportional control valve 25 is closed.

The pressure conduit 20 connects one of the two outlet ports of the proportional control valve 25 to an inlet port of the hydraulic actuator 19. The feed conduit 24 connects another of the two outlet ports to the injectors 23. Two control ports on the proportional control valve 25 are connected to respective discharge ports on the top surface of the distributor block via channels in the distributor block. The proportional control valve 25 also has two tank ports connected to the return conduit 43 for used hydraulic fluid (HFO).

The proportional control valve 25 is a solenoid driven spool valve with three positions. The solenoid 44 receives a control signal from control unit 26 (Fig. 2) via wire 28. The solenoid 44 is mounted to the housing of proportional control valve 25 with a ceramic plate 45 therebetween to thermally insulate the solenoid 44 from the proportional control valve 25, which during engine operation may reach temperatures of over 150 °C. This construction protects the sensitive solenoid 44 from overheating. According to another embodiment (not shown) the solenoid 44 is connected to the valve housing via insulating spacers.

In the center position, in which the solenoid 44 is not active, the inlet port of the proportional control valve 25 is closed and the two outlet ports of the proportional control valve 25 are connected to the return conduit 43. When the solenoid is activated to urge the valve spool to the left (left as in Fig. 3) the inlet port of the proportional control valve is connected to pressure conduit 20, and high pressure HFO is passed to the pressure conduit 20 so that the actuator 19 opens the exhaust valve 11. In this position feed conduit 24 is connected to return conduit 43. When the solenoid 44 is activated to urge the valve spool to the right (right as in Fig. 3) the inlet port of the proportional control valve 25 is connected to the feed conduit 24, and high pressure HFO is passed to the feed conduit 24 so that the injectors 23 inject fuel into combustion chamber 15. In this position pressure conduit 20 is connected to return conduit 43. The fuel injection timing, the volume of fuel injected and the shape of the injection pattern is controlled with the proportional valve. According to a further preferred embodiment, (not shown) the flow of fuel from the common fuel rail to the injectors is controlled by an on/off type valve. This on/off type valve could be a separate valve from the valve that controls the flow to and from the hydraulic actuator.

This separate valve controlling the flow to and from the actuator may also be an on/off type valve.

A conventional fuel limiter 46 is placed in pressure conduit 24, to avoid excessive amounts of HFO entering the cylinder should the proportional control valve 25 erroneously open up too long.

The pressure in the return line 43 is kept to an overpressure of a few bar to avoid penetration of air into the hydraulic system and to prevent the water contained in the water emulsified HFO from forming vapor bubbles. A pressure control valve 47 at the downstream end to the return conduit 43 ensures that a predetermined minimum overpressure is maintained in the return conduit 43. The overpressure in the return conduit 43 is preferably 3 to 10 bar. An accumulator or expansion vessel 48 is connected to the return conduit 43 to absorb pressure fluctuations that can occur when the proportional control valve 25 changes position.

A second return conduit 49 connects the outlet port of the injectors 23 to return conduit 43. Downstream of pressure control valve 47 the return conduit 43 feeds the used HFO to the preheater 31 to complete the cycle.

The conduits that transport the HFO from the outlet of the preheater 31 to the common rail 40 and from the common rail 40 via the proportional control valve 25 to the hydraulic valve actuators 19 and the injectors 23 are provided with heating means symbolized in Fig. 3 by heating coils. The conduits can be heated along their full length by e.g. steam tracing with or electric heating elements. The heating of these conduits serves to reduce heat loss of the hot HFO when it moves downstream from the preheater. During engine operation the temperature of the HFO in the conduits towards the injectors and hydraulic valve actuators is kept close to 150°C, depending however on the viscosity of the HFO used. Adjacent conduits that run parallel for part of their length, such as pressure conduit 20 and feed conduit 24 can be provided with a common heating means (not shown).

Return lines 43 and 49 are also provided with heating means of the same type as described above. The temperature of the HFO in the return lines is less critical and the heating means are calibrated to ensure that the temperature of the HFO does not fall below 50°C.

During engine stops the HFO is circulated through the hydraulic system by circulation pump 34 (at relatively low pressures of 3 to 10 bar) to avoid air being trapped in the hydraulic system and to avoid local cooling and hardening of the HFO. The temperature of the oil leaving the preheater 31 is set at about 50°C during engine stops, to avoid solidification of the HFO.

In order to reach both the injectors 23 and the hydraulic actuator 19 during circulation the proportional control valve periodically changes position. According to another embodiment a fourth bypass position (not shown) is provided for the proportional control valve. In this position the proportional control valve opens to the injectors and to the hydraulic valve actuator simultaneously. According to yet another (not shown) embodiment, a separate bypass valve is provided that allows the HFO to flow from the common rail to the injectors and to the hydraulic valve actuator simultaneously

The heated pressure conduit 20 is provided with means to enable it to operate in a temperature range from the 50°C during circulation to the about 150°C during engine operation. Heat expansion causes the length of the pressure conduit 20 to increase when the temperature of the HFO after and engine stop is increased from about 50°C to about 150°C and vice versa.

As shown in Fig. 5 pressure conduit 20 is provided with one or more U-shaped sections 50 that can absorb the difference in length at different operation temperatures through the flexibility of the U-shaped section. Alternatively, or in combination, parts of pressure conduit 20 and other conduits that need to operate at both low and high temperatures may be suspended axially freely between two brackets 51 and 52, as shown in Fig. 6. Each of the brackets includes a bushing 53 in which an end of the pressure conduit 20 is received such that it is radially secured but axially movable. An O-ring 54 or similar gasket of, cast iron, steel, Polytetra-flourethylene (PTFE), Flouro Rubber, (FPM), Copolymer (NBR), Nitril Rubber, Poly(dimethylsiloxane) (SI), or similar materials ensures a substantially hermetic seal between the end of the conduit and the bushing. The pressure exerted on the opposing free ends of the conduit 20 balance one another. Axial changes in length of conduit 20 are absorbed by its axially freely suspended conduit ends.

The gaskets in the hydraulic system are selected from the group of cast iron, steel, Polytetraflourethylene (PTFE), Flouro Rubber, (FPM), Copolymer (NBR), Nitril Rubber, Poly(dimethylsiloxane) (SI), mixtures thereof or similar materials to ensure a substantially hermetic seal between the components of the hydraulic system. A particular gasket is described below with reference to Fig. 9.

Fig. 4 shows another preferred embodiment of the hydraulic system. This embodiment is substantially identical with the embodiment shown in Fig. 3, however high pressure pump 33 also serves as a low pressure pump for circulation of the HFO during engine stops. Hereto, a clutch 56 controlled by the central control unit is disposed between the gearwheel 37 and the high pressure pump 33. During engine operation the clutch 56 is engaged and the high pressure pump 33 is driven by the crankshaft 3. During engine stops the clutch 56 is disengaged. Another clutch 55 controlled by the central control unit is disposed between the high pressure pump 33 and electrical motor 38'. The clutch 55 is disengaged during engine operation, and engaged during engine stops. The electric motor 38' drives the high pressure pump 33 during engine stops, be it much slower than the running speed during engine operation, to provide enough hydraulic pressure for circulating the HFO at 3 to 10 bar.

With reference to Figs. 7 to 11 a preferred embodiment of the actuator 19 and the pneumatic spring 18 will be described in detail.

The exhaust valve 11 has a stem 57 standing upright from the valve head 58, and the upper end of the stem 57 supports a pneumatic cylinder 59 securely mounted on the valve stem 57 so as to be pressure-sealing and longitudinally displaceable over a stationary piston 60. The stationary piston 60 is part of the spring housing 61. Above the stationary piston 60 there is a spring chamber 62 connected to a pressurized-air supply (not shown), which keeps the spring chamber 62 filled with pressurized air at a predetermined minimum pressure of, for example, an overpressure of 4.5 bar. Other air pressures can also be used, such as from 3 to 10 bar. The minimum pressure is selected according to the desired spring characteristic of the pneumatic spring. It is possible to interconnect the spring chambers on several different cylinders, but preferably each spring chamber is separately cut off by a non-return valve 63 at the pressurized-air supply. The pressurized air in the spring chamber 62 creates a persistent upward force on the pneumatic cylinder 59. The upward force increases when the pneumatic cylinder 59 is displaced downwards and compresses the air in the spring chamber 62 that is prevented from flowing out by of the non-return valve 63.

The spring housing 61 defines a cavity 64 around and above the pneumatic spring 18. The cavity 64 is connected to a drain 65 so that the cavity has atmospheric pressure. Any leak oil from the actuator 19 will enter the cavity 64 and be disposed of via drain 65. The spring construction renders it very difficult for leak oil to enter the spring chamber 62, since the pneumatic cylinder 59 forms an umbrella that forces the leak oil to flow over it and down to the bottom of cavity 64 without any risk of it to enter the spring chamber 62. This is of importance since the leak oil (HFO) could accumulate and harden inside the spring chamber or block pneumatic conduits when the leak oil manages to penetrate even further into the pneumatic system.

With reference to Figs. 7 and 9 the hydraulic valve actuator 19 is constructed from a cylinder 66 supported by the top of the housing 61. A piston 67 is received in a central bore in the cylinder 66. The central bore is closed at the top of the cylinder 66 and is open to the bottom of the cylinder 66. The central bore is coaxially arranged with a bore 68 in the housing 61. The upper (proximate) end of the piston 67 is received in the central bore, whilst the distal end of the piston 67 acts on the top of the valve stem 57.

A primary pressure chamber 69 is defined between cylinder 66 and the top of the piston 67. Hydraulic fluid (HFO) is supplied to and discharged from the hydraulic valve actuator via a port 70. Port 70 opens into an intermediate pressure chamber 71 disposed below a primary pressure chamber 69 and defined between the cylinder 66 and an intermediate section of the piston 67. Port 70 is connected alternately with the pressure conduit 20 and the return conduit 43 as controlled by the proportional control valve, which is in this drawing by way of example shown as an on/off type valve 25', although a proportional type valve could have been used instead. A secondary pressure chamber 73 is defined by an enlarged diameter section 74 of piston 67 and a corresponding enlarged diameter section of the central bore. Optionally, the a gasket 68' can be provided in between the enloarged diameter section 74 and cylinder 66 in order to reduce the amount of leak oil entering the cavity 64. The secondary pressure chamber 73 is supplied with high pressure HFO from the intermediate chamber 71 via axial channels 75 formed by recesses 75 in the piston 67 during the first part of the opening stroke of the hydraulic actuator 19. At a predetermined intermediate position during the opening stroke the axial channels 75 are closed by a control ledge 76 on the cylinder 67. At the same time, a port 77 connects the secondary pressure chamber 73 with the return line 43 since an upper edge of the enlarged diameter section 74 is now positioned below the upper edge of port 77. Thus, the enlarged diameter section 74 assists in overcoming the large force exerted on the valve head 58 by the pressure in the combustion chamber 15 during the first part of the opening stroke of the hydraulic actuator 19. At the predetermined intermediate position of the piston 67 the supply of high pressure fluid to the secondary chamber 73 is interrupted and the secondary pressure chamber is vented via port 77. The pressure in the combustion chamber 15 has now fallen and the action of the enlarged diameter section 74 is no longer required.

Fig. 8 shows a typical opening profile of the exhaust valve. The start of the opening movement in phase I requires high forces from the hydraulic actuator 19 to overcome the pressure in the combustion chamber 15 and to for providing acceleration to the relatively heavy exhaust valve 11. During this phase the hydraulic actuator 19 is required to deliver a maximum force. However, hydraulic shock waves cause by rapid opening of the control valve 25 or 25´ should be avoided. In phase II the exhaust valve 11 reaches the fully open position, and in this section the exhaust valve 11 should be gently slowed down to a halt, preferably without any objects abutting one another. In phase III the return movement of the exhaust valve 11 should start gently, hydraulic pressure waves by abrupt opening and closing of the control valve 25 or 25′ should be avoided. A gentle and accurate landing of the valve head 58 on the valve seat at the end Phase IV is most critical since metal objects abut with one another. It is therefore crucial that the exhaust valve 11 and piston 67 are gradually slowed down in order to minimize mass acceleration forces and to avoid impact of the valve head on the valve seat. The appropriate opening profile for the exhaust valve 11 can in accordance with the present invention be obtained in several ways. One way is by using a simple hydraulic actuator for exhaust valve, the such as a hydraulic cylinder (not shown), combined with appropriate control of the proportional control valve so that substantially exclusively the degree of opening of the proportional control valve ensures the correct force and resistance applied by the actuator on the exhaust valve is used to obtain the appropriate opening profile. Another way is by using a hydraulic actuator and a valve spring described here with an inherent characteristic that enables the appropriate opening profile for the exhaust valve to be obtained with an on/off type control valve. An actuator with an inherent characteristic can also be combined with a proportional valve.

When exhaust valve is to be opened and the proportional control valve 25 supplies the high-pressure fluid to the port 70 and the primary-, intermediate- and secondary pressure chambers are pressurized. The high pressure hydraulic fluid in the primary- and secondary pressure chambers causes the piston 67 to be pressed downwards.

The piston 67 (the first piston part) is provided with a piston cap 78 (the second piston part). The upper part (the proximate end) of the piston 67 slidably engages the piston cap 78 to thereby form a compensation chamber 79 between the piston 67 and the piston cap 78. According to a preferred embodiment the piston cap 78 fits over the top of the piston 67. The piston cap 78 could however also be arranged to fit inside the top of the piston 67 (not shown). A spring 80 urges the piston 67 and the piston cap 78 away from one other to thereby expand the compensation chamber 79. A first flow path is provided between the compensation chamber 79 and the primary pressure chamber 69. The first flow path includes a valve member 81 that is fitted in a reception bore in the top of the piston cap 78. The spring 80 urges the valve member 81 upwards towards the piston cap 78. According to another embodiment (not shown) separate springs may be provided for urging the piston cap 78 and the valve member 81 upwards. This allows the force applied to either element to be adjusted independently from one another.

The valve member 81 is provided with an axial bore 82 and two radial bores 83 and 84 that connect the compensation chamber 79 with the primary pressure chamber 69, unless the valve member 81 is in its upper position within the reception bore. In this upper position (Fig. 9 and 12) the openings of bore 84 are obscured by the walls of the reception bore and thus the first flow path is closed. The first flow path serves to allow an excess of hydraulic fluid to escape from the compensation chamber 79 when the piston 67 is in its upper position and the piston cap 78 is due to an excess in the amount of hydraulic fluid in the compensation chamber 79 placed closer to the top of the primary pressure chamber 69 than required. In this situation (Fig. 10 & 11) the valve member 81 abuts with the end surface of the cylinder 66 and the valve member 81 is moved down relative to the piston cap 78 thereby opening the first flow path so that the compensation chamber 79 can be evacuated until the valve head 58 rests on the valve seat. The first flow path is thus only open when the piston cap part is located in a small predetermined axial range at the upper (proximate) end of the cylinder 66.

A second flow path is present between the compensation chamber 79 and the intermediate pressure chamber 71. The second flow path is according to a preferred embodiment formed by an annular gap 85 between the piston 67 and the piston cap 78. Since the annular gap 85 is narrow, the second flow path has a relatively high flow resistance. The second flow path allows the compensation chamber 79 to be refilled under the influence of spring 80. The correct filling flow rate of the compensation chamber is obtained by selecting the appropriate characteristics for the force of spring 80 and the resistance of the flow path 85.

A venting conduit 86 with a high flow restriction is provided in the top of the cylinder 66 and connects the top of the primary pressure chamber 69 that is formed by the dampening chamber 87 with the return conduit 43.

The piston cap 78 has an axially tapered outer circumference that is increasing in diameter towards the top of the piston. The tapered section cooperates with an inwardly projecting annular flange 88 that extends from the central bore just above the position at which the port 70 opens into the central bore. The tapered section forms together with the annular flange 88 a narrow annular gap 89 of a size that varies with the position of the piston. The hydraulic fluid has to be pressed through the annular gap 89 to flow from the intermediate pressure chamber 71 to the primary pressure chamber 69. This causes a pressure drop between the intermediate pressure chamber 71 and the primary pressure chamber 69. The pressure drop increases when the size of the annular gap 89 decreases, and increases progressively with increasing flow rates, thus effectively preventing the piston 67 from reaching high speeds. The tapered section is dimensioned such that the annular gap 89 is smaller towards the end of the opening stroke. The speed of the piston 67 is therefore effectively limited towards the end of the stroke even if the supply pressure of the hydraulic liquid is relatively high. The tapered section is shown in Figs. 9 to 11 as having a slightly outwardly curved profile, but other profiles such as a truncated cone, a slightly inwardly curved profile, combinations thereof or any desired predetermined profile are possible. Such a profile could be determined through tests, computer simulations or analytical methods that indicate how large the flow restriction should be at each position along the stroke for an optimal dynamic behavior of the valve actuator. The tapered section may then be constructed accordingly.

The downward force of the actuator 19 and the upward force of the air spring 18 are balanced at the end of the outward stroke, i.e. the piston 67 and the exhaust valve 11 will come to a stop by themselves, as shown in phase II of Fig. 8. Neither cutting-off the supply of high pressure HFO nor a stroke limiter are required for stopping the piston and exhaust valve. Since the supply of HFO does not need to be suddenly cut-off there are no hydraulic shockwaves that otherwise stress the whole hydraulic system. The absence of a stroke limiter results in lower mechanical loads and shocks.

The pressure of the HFO supplied to the hydraulic actuator 19 and the pressure of air supplied to the air spring 18 are controlled to ensure that the exhaust valve 11 reaches the correct open position. The actuator 19 and the air spring 18 are dimensioned such that they easily reach balancing opposing forces in the open position.

The flow path between flange 88 and the tapered section of the piston cap 78 is narrow when the piston 67 is close to the fully open position. The narrow gap has a dampening effect on the motion of the piston 67. The piston comes therefore to a stop in the open piston with little or no overshoot and following oscillations.

The piston 67 is retuned to the retracted position under the influence of the air spring 18. The hydraulic actuator 19 is provided with an end of stroke damper in the form of a dampening chamber 87 in the top of cylinder 66 (at the proximate end). The top of the piston cap 78 is dimensioned to fit with a minute clearance into the dampening chamber 87, and when the top of the second piston part 78 plunges into the dampening chamber most of the kinetic energy of the piston 67 and exhaust valve 11 in the return stroke is absorbed by forcing the hydraulic liquid out of the dampening chamber 87 through the minute clearances formed by an annular gap 90, and the valve head 58 lands gently on the valve seat.

The flow resistance of the flow path between the port 70 and the primary pressure chamber 69 is adjusted by changing the design of the tapered section in accordance with the pressure needed in the primary pressure chamber 69 at the respective position of the piston 67. The hydraulic valve actuator 19 can therefore operate properly with a high-pressure source that has a varying pressure. Relatively low supply pressures will cause a slower acceleration of the valve. Consequently, the electronic control unit 26 continuously adapts the timing and the length to the valve opening to compensate for pressure variations in of the high pressure hydraulic fluid supply. When the supply pressure is relatively low the electronic control unit 26 will instruct the proportional control valve 25 to open relatively early and stay open for a relatively longer time, to ensure that the exhaust valve is open long enough for a proper evacuation of the gases in the combustion chamber, and vice versa when the supply pressure is relatively high.

Cylinder 66 comprises a venting and recirculation conduit 86 via which warm hydraulic fluid can be circulated through the actuator and back into the return conduit 43. This is advantageous for keeping the valve at service temperature when the engine is not running, and further this provides for an effective de-aeration.

Operation of the hydraulic valve

In the closed position of the exhaust valve 11, the piston part cap 78 is at a position with its top inside the dampening chamber 87 within the range of positions that the valve member 81 allows. Fig. 10 illustrates the highest possible position of the piston cap 87 with the first flow path is open, and Fig. 12 shows the lowest possible position of the piston cap 78 with the valve member 81 closed. Within the range of positions there is always a narrow annular gap 90 between the top of the piston cap 78 and the walls of the dampening chamber 87.

The exhaust valve 11 is opened by feeding high pressure medium (HFO or Fuel oil) from the proportional control valve 25 (according to another embodiment other types of valves such as on/off type valves 25' or servo valves can be used instead of a proportional valve) to port 70 (Fig 10). From here, the hydraulic medium passes via annular gap 89 and annular gap 90 into the primary pressure chamber 69 and dampening chamber 87 and builds up a pressure that urges the piston 67 downwards. From port 70 the hydraulic fluid flows also into the intermediate chamber 71 and via the axial channels 75 into the secondary pressure chamber 73. The pressure thus acting on the enlarged diameter section 74 adds to the force with which the piston 67 is urged downwards.

The exhaust valve 11 starts to open when the combined force on piston 67 exceeds the counterforces from the pressure in the air spring 18 and in the combustion chamber 15. During the start of the opening movement the limited flow through annual gap 90 into the dampening chamber 87 causes a slow pressure build up in the latter chamber, thus ensuring that the start of the opening movement is smooth, without violent acceleration and without hydraulic shock waves, cf. Fig. 8, phase I.

When the exhaust valve 11 is partially opened the pressure in the combustion chamber 15 and the force required to complete the opening of the exhaust valve 11 drop significantly. At this stage the downward force acting on the piston 67 is reduced by cutting off the flow of hydraulic fluid to the secondary pressure chamber 73 with the control ledge 76 and simultaneously connecting the secondary pressure chamber 73 to the return conduit 43 via port 77 to allow hydraulic fluid from the return conduit 43 to feed the further expansion of the secondary pressure chamber 73 during the remainder of the opening stroke so as to avoid cavitation in axial channels 75 and in the secondary pressure chamber 73.

The flow area of gap 89 decreases when the opening of the of the exhaust valve 11 increases. Thereby, the pressure in the primary pressure chamber 69 and in the compensation chamber 79 gradually decreases. Simultaneously, the pressure in the air spring 18 gradually increases and therefore the velocity of the exhaust valve 11 steadily decreases until there is a perfect balance between the forces exerted by the hydraulic and pneumatic media. Since the opposing fluid pressures change gradually the exhaust valve 11 and piston 67 decelerate smoothly to a full stop without any hydraulic shockwaves or mechanical abutment, cf. Fig. 8, phase II. Any oscillating movement of the exhaust valve 11 around the fully open position is reduced by the dampening effect of the strongly reduced flow area of gap 89.

The valve member 81 is closed against the under side of the piston cap 78 by the force of spring 80 during the open period of the exhaust valve 11. The amount of hydraulic fluid that is trapped inside the compensation chamber 79 secures a predefined position of the piston cap 78. The pressure difference between the intermediate pressure chamber 71 and the primary pressure chamber 69 as well as the force of the spring 80 urge the piston cap 78 upwards and thus a small amount of hydraulic fluid is sucked into the compensation chamber 79 via the annular gap 85 between the piston cap and the piston.In the completely open position of the gas exchange valve 11 the pressure in the primary pressure chamber 69 and the intermediate pressure chamber 71 is equal, and only the spring 80 urges the piston cap 78 upwards.The refilling of the compensation chamber 79 during the opening and completely open period of the exhaust valve 11 causes the piston cap 78 to move slowly upwards relative to the piston 67.

The exhaust valve 11 closes again when the proportional control valve 25 changes position and connects port 70 with return conduit 43. The thrust of the air spring 18 forces the hydraulic fluid from the primary pressure chamber 69 via annular gap 89 into the return conduit 43. The small flow area in annular gap 89 assures a soft start-up of the return stroke with a steadily increasing velocity controlled by the steadily increasing flow area of annular gap 89 during upward movement of the piston 67, cf. Fig. 8, phase III. Due to the higher pressure in the primary pressure chamber 69 than in the intermediate chamber 71 evacuation via annular gap 85 will cause the compensation chamber 79 to shrink somewhat. The hydraulic fluid in the secondary pressure chamber 73 is evacuated via port 77 and when the latter is obscured by the increased diameter section 74 via the axial channels 75, intermediate chamber 71, port 70 and return conduit 43.

In the final stage of the closing movement the piston cap 78 plunges into dampening chamber 87 and the thus formed annular gap 90 reduces the available flow area for the hydraulic fluid trapped in the dampening chamber significantly. Via annular gap 90 the hydraulic fluid trapped in dampening chamber 87 is forced out of the dampening chamber which thereby via a corresponding increase in the pressure in compensation chamber 79 acts as a braking force on the piston 67 to thereby decelerate it, cf. Fig. 8, phase IV. The increase in pressure in the compensation chamber 79 will cause some hydraulic fluid to be evacuated therefrom via annular gap 85. The landing speed of the valve head 58 on the valve seat is therefore largely determined by the flow area of annular gap 90 just prior to the closing of the exhaust valve 11. Venting conduit 86 and annular gap 85 contribute marginally to the outflow from the dampening chamber 87.

If the compensation chamber 79 has completely expanded during the opening period of the exhaust valve 11,the piston cap 78 will assume a slightly higher position when it plunges into the dampening chamber 87. Hereby, the valve member 81 will abut with the end of the cylinder 66 (the bottom of the dampening chamber) and open the first flow path to evacuate the compensation chamber 79 (Fig. 11) so that the piston cap 78 can assume a correct position (Fig. 10).

If the compensation chamber 79 has completely contracted during the return stroke of the exhaust valve 11 the piston cap 78 will assume a slightly lower position when it plunges into the damping chamber 87, and the valve member 81 will not abut with the end of the cylinder (Fig. 12). Until the next opening period the spring 80 will urge the piston cap 78 upwards. The compensation chamber 79 will thus receive the missing amount of hydraulic fluid via annular gap 85 until valve member 81 abuts with the end of the cylinder 66 (Fig. 13) and ensures that the piston cap 78 assumes a substantially central position within its axial range.

The operation of piston cap 78 in combination with the compensation chamber 79 enables the hydraulic actuator 19 to automatically compensate for dimensional changes caused by operation at different temperatures, reconditioning, i.e. grinding of the valve seat, and production tolerances. Thus, the valve head 58 will always land gently and precisely on the valve seat.

According to an embodiment of the invention, the hydraulic actuator 19 can also be realized as shown in Fig. 7 without the compensation chamber. This embodiment can be deployed in engines in which compensation for dimensional changes is less critical, e.g. when a normal hydraulic fluid is used as hydraulic fluid that is operated at 30 to 60 °C.

Although the present invention has been described in detail for purpose of illustration, it is understood that such detail is solely for that purpose, and variations can be made therein by those skilled in the art without departing from the scope of the invention.

CLAIMS:

1. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame (4),

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11),

a common fuel rail (40) having one or more accumulators (42) connected thereto,

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40),

each of said injectors (23) being operated with fuel from said common fuel rail (40), and

a proportional valve (25) associated with each cylinder (6),

whereby said proportional valves (25) control the flow of fuel from the common fuel rail (40) to the respective injectors (23).

2. An engine according to claim 1, wherein the fuel injection timing, the volume of fuel injected and the

shape of the injection pattern is controlled with said proportional valve (25).

3. An engine according to claim 1 or 2, wherein said actuators (19) are operably connected to said common fuel rail (40).

4. An engine according to claim 3, wherein a control valve (25,25') is associated with each of the cylinders (6), said control valves (25,25') controlling the flow of fuel from the common fuel rail (40) to the respective hydraulic valve actuators (19).

5. An engine according to claim 4, wherein said control valves (25') are on/off type valves.

6. An engine according to claim 4, wherein said control valves are proportional type valves (25).

7. An engine according to claim 5 or 6, wherein said proportional valve and said control valve are combined into one integral valve (25) with a single spool.

8. An engine according to claim 7, wherein said integral valve (25) comprises a valve housing and a solenoid (44) for controlling the spool, whereby said solenoid (44) is thermally insulated from said valve housing.

9. An engine according to claim 8, wherein a layer of thermally insulating material (45) is disposed between the electric solenoid (44) and the valve housing.

10. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11),

a common fuel rail (40) having one or more accumulators (42) connected thereto,

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40),

each of said injectors (23) being operated with fuel from said common rail (40), and

a proportional control valve (25) associated with each cylinder (6),

whereby said proportional control valves (25) control the flow of fuel from the common fuel rail (40) to the respective hydraulic valve actuators (19).

11. An engine according to claim 10, wherein the timing of the opening and closing of the exhaust valves (11), and the extent of opening of the exhaust valves is controlled by the respective proportional control valves (25).

12. An engine according to claim 10 or 11, wherein the opening and closing profile of the exhaust valves (11) is largely determined by the characteristic of the hydraulic valve actuators (19).

13. An engine according to claim 10 or 11, wherein the opening and closing profile of the exhaust valves (11) is largely determined by the proportional valves (25).

14. An engine according to any of claims 10 to 12, wherein the proportional control valves (25) are operated to act as on/off type valves when controlling the hydraulic valve actuators (19).

15. An engine according to any of claims 10 to 14, wherein the flow of fuel from the common fuel rail (40) to the injectors (23) is controlled by on/off type valves (25').

16. An engine according to any of claims 10 to 14, wherein the flow of fuel from the common fuel rail (40) to the injectors (23) is controlled by proportional type valves (25).

17. An engine according to claim 15 or 16, wherein said proportional control valves (25) and said valves for controlling the flow of fuel to the injectors (23) are combined into integral valves (25) with a single spool.

18. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11),

a common fuel rail (40) having one or more accumulators (42) connected thereto, and

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail,

each of said injectors (23) being operated with fuel from said common rail (40),

said hydraulic valve actuators (19) being connected with said common rail (40) via respective pressure conduits (20), whereby said pressure conduits are provided with heating means.

19. An engine according to claim 18, wherein the fuel entering supply conduits leading to the common rail (40) has a temperature in the range between 90 and 150 °C when the engine (1) is in operation, and said heating means at least reduces heat loss of the fuel in said supply conduits.

20. An engine according to claim 18 or 19, wherein the fuel entering the supply conduits has a temperature in the range between 40 and 70 °C when the fuel is circulated during engine stops, and said heating means at least reduces heat loss of the fuel in said pressure conduits (20).

21. An engine according to any of claims 18 to 20, wherein said engine further comprises return conduits (43) for transporting fuel from said hydraulic valve actuators (19) to a fuel tank (29) or to conduits leading to the intake side of the high pressure pump (33), whereby said return conduits (43) are provided with heating means.

22. An engine according to any of claims 18 to 21, wherein said heating means are electrically operated.

23. An engine according to any of claims 18 to 22, wherein heating means are steam operated.

24. An engine according to any of claims 18 to 23, wherein said heating means are configured to operate also when the engine is stopped.

25. An engine according to any of claims 18 to 24, further being provided with means (34,38',55,56) to circulate said fuel through said supply and/or return and/or pressure conduits when the engine (1) is stopped.

26. An engine according to claim 25, further provided with means to circulate said fuel through the conduits (24) and other hydraulic (25,25') components that connect said common fuel rail (40) with said injectors (23) when the engine (1) is stopped.

27. An engine according to claim 26, wherein said engine (1) comprises an integrated proportional valve (25) that controls the flow of fuel to both the hydraulic valve actuators (19) and to the injectors (23), whereby circulation of the fuel during engine stops is enabled by one or more of:

periodically opening for flow to either the injectors (23) or the hydraulic valve actuator (19), or

providing a bypass position for said integrated proportional valve (25) in which the proportional valve (19)opens to the injectors (23) and to the hydraulic valve actuators (19) simultaneously, or

providing a separate bypass valve that allows the fuel to flow from the common rail (40) to the injectors (23) and to the hydraulic valve actuators (19) simultaneously.

28. An engine according to any of claims 25 to 27, wherein said circulation is performed at low pressure, preferably at a pressure between 3 and 10 bar.

29. An engine according to claims 18 to 28, wherein the return conduit (43) is pressurized when water emulsified fuel is used.

30. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11), a common fuel rail (40) having one or more accumulators (42) connected thereto, and

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40),

each of said injectors (23) being operated with fuel from said common rail (40),

each of said hydraulic valve actuators (19) being operated with fuel from said common rail (40), whereby said hydraulic valve actuators (19) are provided with means to compensate for dimensional changes caused by operating at different temperatures.

31. An engine according to claim 30, wherein said means to compensate comprise a piston (67) with a first part (67) and a second part (78), said second part (78) slidably engages the first part (67) to thereby form a compensation chamber (79) between said first part (67) and said second part (78).

32. An engine according to claim 31, further comprising a spring means (80) urging said first part (67) and second part (78) away from one other to thereby expand said compensation chamber (79).

33. An engine according to claim 32, further comprising:

a first flow path between said compensation chamber (79) and a pressure chamber (69,87), said first flow path being open only when said second part (78) is located in a small predetermined axial range at the top of said hydraulic actuator (19) to allow an excess of hydraulic fluid to be evacuated from said compensation chamber (79), and

a second flow path between said compensation chamber (79) and said pressure chamber (69, 87), said second flow path allowing said compensation chamber (79) to be refilled under the influence of said spring means (80).

34. An engine according to claim 33, wherein the second part (78) plunges into a blind end of a stroke dampening chamber (87) just before the piston (67) reaches a retracted position in which the exhaust valve (11) is seated.

35. A method of controlling the temperature of fuel in a pressure conduit (20) of a large two-stroke diesel engine (1) of the cross-head type, said pressure conduit (20) connecting a common fuel rail (40) to a hydraulic valve actuator (19), and said method comprising the step of controlling the temperature of the fuel entering said pressure conduit (20) to maintain the temperature gradient of the fuel below a predetermined threshold during changes in operating temperature of the fuel.

36. A method according to claim 35, wherein the fuel temperature gradient is kept between 0 and -2 °C per minute during a change in operating temperature of the fuel from the fuel temperature during engine operation to the fuel temperature for circulating the fuel during engine stops.

37. A method according to claim 35 or 36, wherein the fuel temperature gradient is kept between 0 and 2 °C per minute during a change in operating temperature of the fuel from the fuel temperature for circulating the fuel

during engine stops to the fuel temperature during engine operation.

38. A method according to claim 36 or 37, wherein the temperature of the fuel during engine operation is between 90 and 150 °C, and the temperature of the fuel for circulation during engine stops is between 40 and 70 °C.

39. A method according to any of claims 35 to 38, wherein the temperature of the fuel is substantially controlled by a preheater (31) disposed in the fluid circuit upstream of the high pressure fuel pump (33).

40. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11),

a common fuel rail (40) having one or more accumulators (42) connected thereto, and

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40), each of said injectors (23) being operated with fuel from said common rail (40),

said hydraulic valve actuators (19) being connected with said common rail (40) via respective hydraulic lines and eventually other hydraulic components such as valves, characterized in that the static gaskets (68') that seal the connection between conduits and other hydraulic components of the engine, and dynamic gaskets in the valve actuator are made of cast iron, steel, Polytetraflourethylene (PTFE), Flouro Rubber, (FPM), Copolymer (NBR), Nitril Rubber, Poly(dimethylsiloxane) (SI) or combinations and/or mixtures thereof.

41. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11),

a common fuel rail (40) having one or more accumulators (42) connected thereto, and

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40), a supply conduit (24) and valve (25) means associated with each cylinder (6) for delivering fuel from said common rail (40) to the respective injectors (23),

a pressure conduit (20) and valve means (25,25′) associated with each cylinder (6) for delivering fuel from said common rail (40) to the respective hydraulic valve actuators (19), and

a heated return conduit (43) for transporting fuel from the hydraulic valve actuators (19) to a fuel tank (29) or to a conduit leading to the intake side of the high pressure pump (33).

42. An engine according to claim 41, wherein said heated return conduit (43) is pressurized.

43. An engine according to claim 42 wherein said heated return conduit (43) includes a pressure control valve (47), preferably a pressure control valve to maintain a given minimum pressure in said heated return conduit.

44. An engine according to any of claims 41 to 43, wherein one or more accumulators (48) are connected to said pressurized and heated return conduit (43).

45. An engine according to any of claims 41 to 44, wherein said heated return conduit (43) includes heating means to reduce heat loss op the fuel contained therein.

46. An engine according to claim 45, wherein the heating means is adapted to maintain the fuel contained in the return conduit (43) above 50° C.

47. An engine according to any of claims 43 to 46, wherein the given minimum pressure is about 3 to 10 bar.

48. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a hydraulic valve actuator (19) associated with each of the exhaust valves (11),

a common fuel rail (40) having one or more accumulators (42) connected thereto, and

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40),

a supply conduit (24) and valve means (25) associated with each cylinder (6) for delivering fuel from said common rail (40) to the respective injectors (23),

a pressure conduit (20) and valve means (25,25') associated with each cylinder (6) for delivering fuel from said common rail (40) to the respective hydraulic valve actuators (19), and

a return conduit (43) for transporting fuel from the hydraulic valve actuators (19) to a fuel tank (29) or to a conduit leading to the intake of the high pressure pump (33),

wherein at least one of said conduits (20,24,40,43) includes means for neutralizing (50,53,54) the effects of dimensional changes of said conduits caused by changes in operating temperature.

49. An engine according to claim 48, wherein said means for neutralizing (50,53,54) comprise conduit sections that are longitudinally freely suspended at their extremities between oppositely disposed supporting elements (51,52,53,54).

50. An engine according to claim 48 or 49, wherein said means for neutralizing comprise one or more U-shaped conduit sections (50).

51. The use of a proportional valve (25) to control the flow of fuel from a common fuel rail (40) of a large two-stroke diesel engine (1) of the cross-head type to the fuel injectors (23) and/or to a fuel operated component (19).

52. A use of a proportional valve according to claim 51, wherein the fuel operated component is a hydraulic valve actuator (19).

53. A use of a proportional valve according to claim 52, wherein the proportional valve (25) is operated to act as an on/off valve.

54. An electrically controlled valve (25,25') for controlling the flow of fuel from a common fuel rail (40) of a large two stroke diesel engine (1) of the crosshead type to one or more fuel operated or fuel consuming

engine components (19,23), said electronically controlled valve (25,25') comprising a valve housing and a solenoid (44), characterized in that said solenoid (44) is thermally insulated from said valve housing.

55. A valve according to claim 54, wherein said valve is a proportional valve (25).

56. A valve according to claim 54 or 55, wherein said valve (25,25') has at least three positions, a central position connecting two hydraulic fuel operated or fuel consuming engine components (19,23) with a return conduit (43), a first non-central position in which a first of said two hydraulic fuel operated or fuel consuming engine components is connected with a source of high pressure fuel (41) whilst a second of said two hydraulic fuel operated or fuel consuming engine components is connected with said return conduit (43), and a second non-central position in which the second of said two hydraulic fuel operated or fuel consuming engine components is connected with said source of high pressure fuel (41) whilst the first of said two hydraulic fuel operated or fuel consuming engine components is connected with said return conduit (43).

57. A valve according to any of claims 54 to 56, wherein said solenoid (44) is thermally insulated from said housing by a layer (45) of insulating material, preferably a ceramic material.

58. A large two-stroke diesel engine (1) of the cross-head type, comprising:

a crankcase frame (4) supporting a crankshaft (3) and a cylinder frame (5) mounted on the crankcase frame,

a plurality of cylinders (6) carried by the cylinder frame (5), each cylinder being provided with at least one fuel injector (23) and with at least one exhaust valve (11),

a common fuel rail (40), and

a high pressure fuel pump (33) feeding fuel under high pressure to the common fuel rail (40) during engine operation,

a supply conduit (24) and valve means (25) associated with each cylinder (6) for delivering fuel from said common rail (40) to the respective injectors (23),

said high pressure fuel pump (33) being mechanically driven by the crankshaft (3) during engine operation and electrically driven by an electric motor (38') during engine stops to provide fuel at a low pressure for circulating the fuel through said supply conduit (24) and/or the common rail (40) and/or through other engine components operated with fuel.

59. An engine according to claim 58, further comprises a clutch (56) for connecting and disconnecting said high pressure fuel pump (33) to said crankshaft (3).

60. An engine according to claim 58 or 59, further comprises a clutch (55) for connecting and disconnecting said high pressure fuel pump (33) to said electric motor (38').

61. An engine according to any of claims 58, to 60, further comprising a gear (36,37) for connecting said crankshaft (3) to said high pressure fuel pump (33).

62. An engine according to any of claims 58 to 61, wherein said high pressure pump (33) delivers fuel at a pressure of 600 to 2000 bar during engine operation and a pressure of 3 to 10 bar during engine stops.

63. A hydraulically actuated gas exchange valve (11) for an internal combustion engine (1) comprising:

- a stationary valve housing (61),
- a gas exchange valve (11) that is movable between a seated position and an unseated position and includes an elongated valve stem (57) with a valve head (58) on one end and a free extremity on the opposite end,
- a hydraulic actuator (19), said hydraulic actuator comprising a piston (67) acting on the free extremity of the valve stem (57) for urging the gas exchange valve (11) to an unseated position when the hydraulic actuator (19) is supplied with pressurized hydraulic fluid,
- a pneumatic spring (18) urging the valve to the seated position,
- said pneumatic spring (18) comprising:
    - o a cylinder (59) secured to the valve stem (57), said cylinder being closed in the direction towards the free extremity of the valve stem (57) and open in the direction towards the valve head (58), and
    - o a matching stationary piston (60) received in said cylinder (59), said piston being secured to said valve housing (61) and forming together with said cylinder (59) a spring chamber (62) for said pneumatic spring (18).

64. A hydraulically actuated gas exchange valve (11) according to claim 63, wherein a conduit for supply of pressurized air to the spring chamber (62) passes through the piston (60).

65. A hydraulically actuated gas exchange valve (11) according to claim 63 or 64, wherein valve stem (57) passes slidably and substantially airtight through a bore in the piston (60).

66. A hydraulically actuated gas exchange valve assembly for an internal combustion engine (1) comprising:
- a stationary valve housing (61),
- a gas exchange valve (11) that can travel between a seated position in which the valve is closed and an unseated position in which the valve is open and includes an elongated valve stem (57) with a valve head (58) on one end and a free extremity on the opposite end,
- a hydraulic valve actuator (19), said hydraulic valve actuator (19) comprising a piston (67) acting on the free extremity of the valve stem (57) for urging the gas exchange valve (11) to an unseated position when the hydraulic valve actuator (19) is supplied with pressurized hydraulic fluid,
- a pneumatic spring (18) urging the gas exchange valve (11) to the seated position,
- whereby the length of travel of the gas exchange valve (11) in the opening direction is determined by the balance of the opposing forces of the hydraulic valve actuator (19) and the air spring (18).

67. A gas exchange valve assembly according to claim 66, wherein said hydraulic valve actuator (19) comprises:

- a piston (67) disposed in a cylinder (66) that acts on the free end of the valve stem (57), said piston (67) being in a retracted position when the gas exchange (11) valve is seated and in an extended position in which the gas exchange valve (11) is not seated,
- a primary pressure chamber (69,87) in which pressurized fluid acts on a surface area of the piston (67) to actuate it to the extended position,
- a port (70) which can be alternately connected to a high-pressure source of hydraulic fluid (40) or to a return line (43), and
- a flow path between said port (70) and said primary pressure chamber (69,87), whereby the flow resistance of said flow path is substantially higher in the extended position than in the retracted position.

68. A gas exchange valve according to claim 65, wherein the flow path of the fluid from a source of high pressure fluid (20) into the actuator is throttled when the piston is (67) in an extended position, so that the movement of the piston (67) in the extended position is dampened.

69. A hydraulic actuator (19) for a gas exchange valve (18) of an internal combustion engine (1) comprising:

- a stationary cylinder (66) with a proximate end and an open distal end and including a pressure chamber (69,87) that can be alternately connected to a source of high-pressure hydraulic fluid (40) or to a return line (43) by valve means (25,25'),
- a piston (67) with a proximate end received in said primary pressure chamber (69,87) and a distal end acting on the free extremity of the valve stem (57) of said gas exchange valve (11) for urging the valve

to an unseated position when the pressure chamber (69, 87) is connected to said source of high-pressure hydraulic fluid (40),

- said piston (67) comprising a first part (67) and a second part (78), said first part (67) extending from the distal end towards the proximate end and said second (78) part being disposed at the proximate end,
- said second part (78) slidably engages the first part (67) to thereby form a compensation chamber (79) between said first part (67) and said second part (78),
- a spring means (80) urging said first part (67) and second part (78) away from one other to thereby expand said compensation chamber (79),
- a first flow path between said compensation chamber (79) and said pressure chamber (69,87), said first flow path being open only when said second part (78) is located in a small predetermined axial range at the proximate end of said stationary cylinder (66) to allow an excess of hydraulic fluid to be evacuated from said compensation chamber (79),
- a second flow path between said compensation chamber (79) and said pressure chamber (69,87), said second flow path allowing said compensation chamber (79) to be refilled under the influence of said spring means (80).

70. An actuator according to claim 67, wherein the second part (78) plunges into a blind end of stroke dampening chamber (87) just before the piston 67 reaches the retracted position.

71. An actuator according to claim 69 or 70, wherein said said blind stroke dampening chamber 87 is a part of said pressure chamber 69.

72. An actuator according to claim 71, wherein said second part 78 has a diameter slightly smaller than the diameter of the blind end of stroke dampening chamber (87) so as to allow a restricted flow of hydraulic fluid between said blind end of stroke dampening chamber (87) and the remainder of the pressure chamber (69) when said second part (78) is received in the blind end of stroke dampening chamber.

73. An actuator according to any of claims 69 to 72, wherein said first flow path is open when the second part (78) is in the upper end of its axial range and the gas exchange valve (11) is seated.

74. A hydraulic actuator (19) for a gas exchange valve (11) of an internal combustion engine (1) comprising:

- a stationary cylinder (66) including a pressure chamber (69,73,87) that can be alternately connected to a source of high-pressure hydraulic fluid (40) or to a return conduit (43) via a port (70) in said stationary cylinder (66),
- a piston (67) received in said pressure chamber (69,73,87) and acting on the free extremity of the valve stem (57) of said gas exchange valve (11) for urging the gas exchange valve (11) to an unseated position when the pressure chamber (69,73,87) is connected to said source of high-pressure hydraulic fluid (40),
- said piston (67) being axially movable between a retracted position in which the gas exchange valve (11) is seated to an extended position in which the gas exchange valve (11) is open,
- said piston (67) having a first effective area on which said pressurized hydraulic fluid in said

pressure chamber (69,73,87) acts to urge said piston (67) towards the extended position when said piston (67) is located between said retracted position and a predetermined intermediate position,

- said piston (67) having a second effective area, smaller than said first effective area, on which said pressurized hydraulic fluid in said pressure chamber (69,87) acts to urge said piston (67) towards the extended position when said piston (67) is located between said intermediate position and said extended position.

75. An actuator according to claim 74, wherein said piston (67) comprises an enlarged piston section (74) with a first predetermined diameter that is larger than the diameter of the remainder of said piston, said stationary cylinder (66) being provided with a section with an enlarged diameter corresponding to the enlarged piston section (74), said enlarged piston section (74) and said enlarged cylinder section forming an secondary pressure chamber (73) that is in fluid communication with said port (70) when said piston (67) is located between said retracted position and said predetermined intermediate position whilst said secondary pressure chamber (73) is connected to a return port (77) and not in fluid communication with said first port (70) when said piston is (67) located between said intermediate position and said extended position.

76. An actuator according to claim 74 or 75, wherein said stationary cylinder (66) comprises slits (75) or a reduced diameter section that provides a flow path from said first port (70) to said secondary pressure chamber (73) when said piston (67) is located between said retracted position and said predetermined intermediate position.

77. An actuator according to any of claims 74 to 76, wherein said pressure chamber (69,87) comprises a primary pressure chamber (69, 87) that is in fluid connection with said first port (70) via a primary flow path, said actuator further comprising means (78,88,89) for varying the flow resistance of said primary flow path relative to the position of the piston (67).

78. An actuator according to claim 77, wherein the flow resistance of said primary flow path increases when the piston (67) moves from the retracted position to the extended position and vice versa.

79. An actuator according to claim 78, wherein said piston (67) comprises a tapered section which together with a flange (88) protruding inwardly from said cylinder forms a flow restriction (89) that increases when the piston (67) moves from the retracted position to the extended position and vice versa.

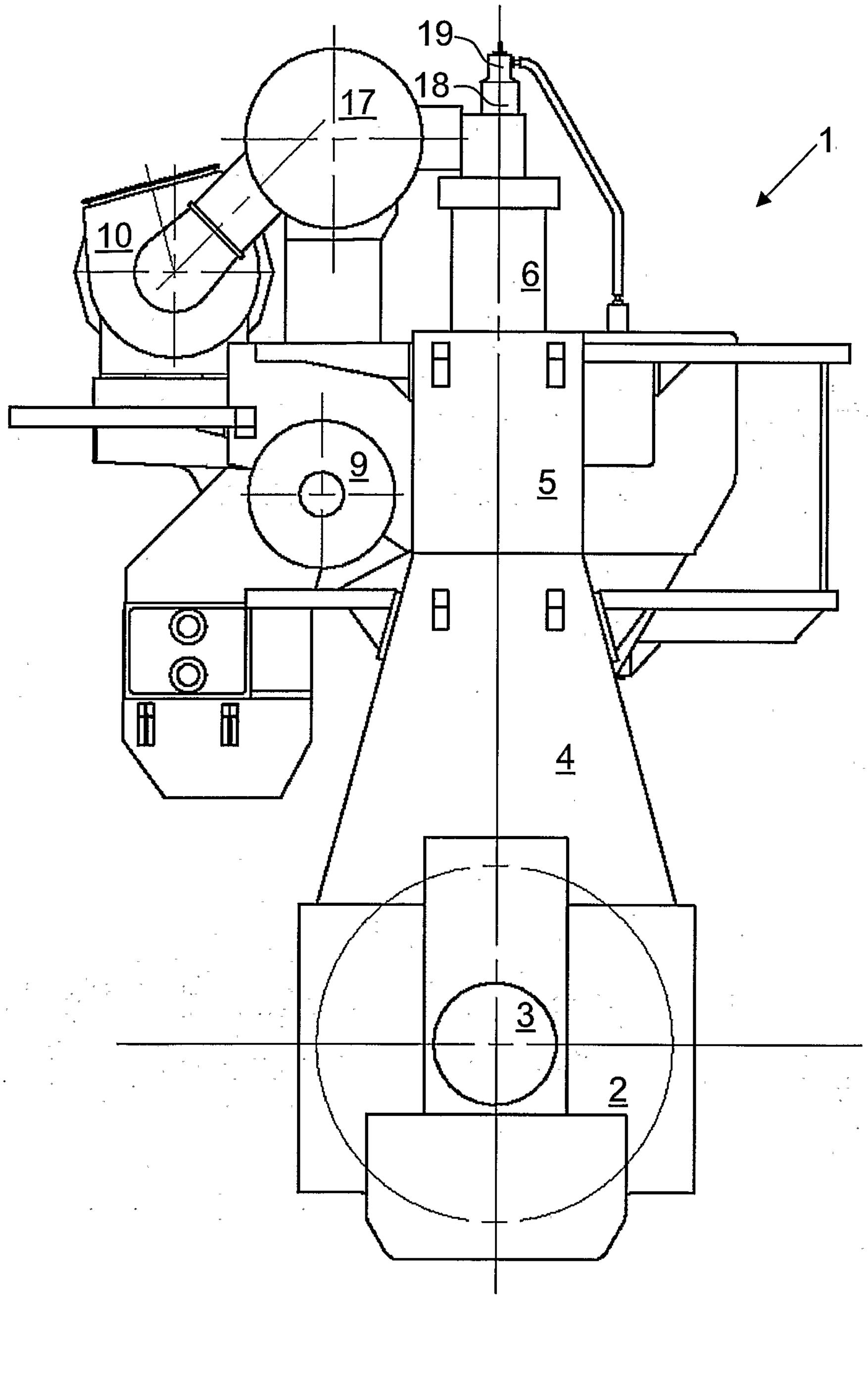

Fig. 1

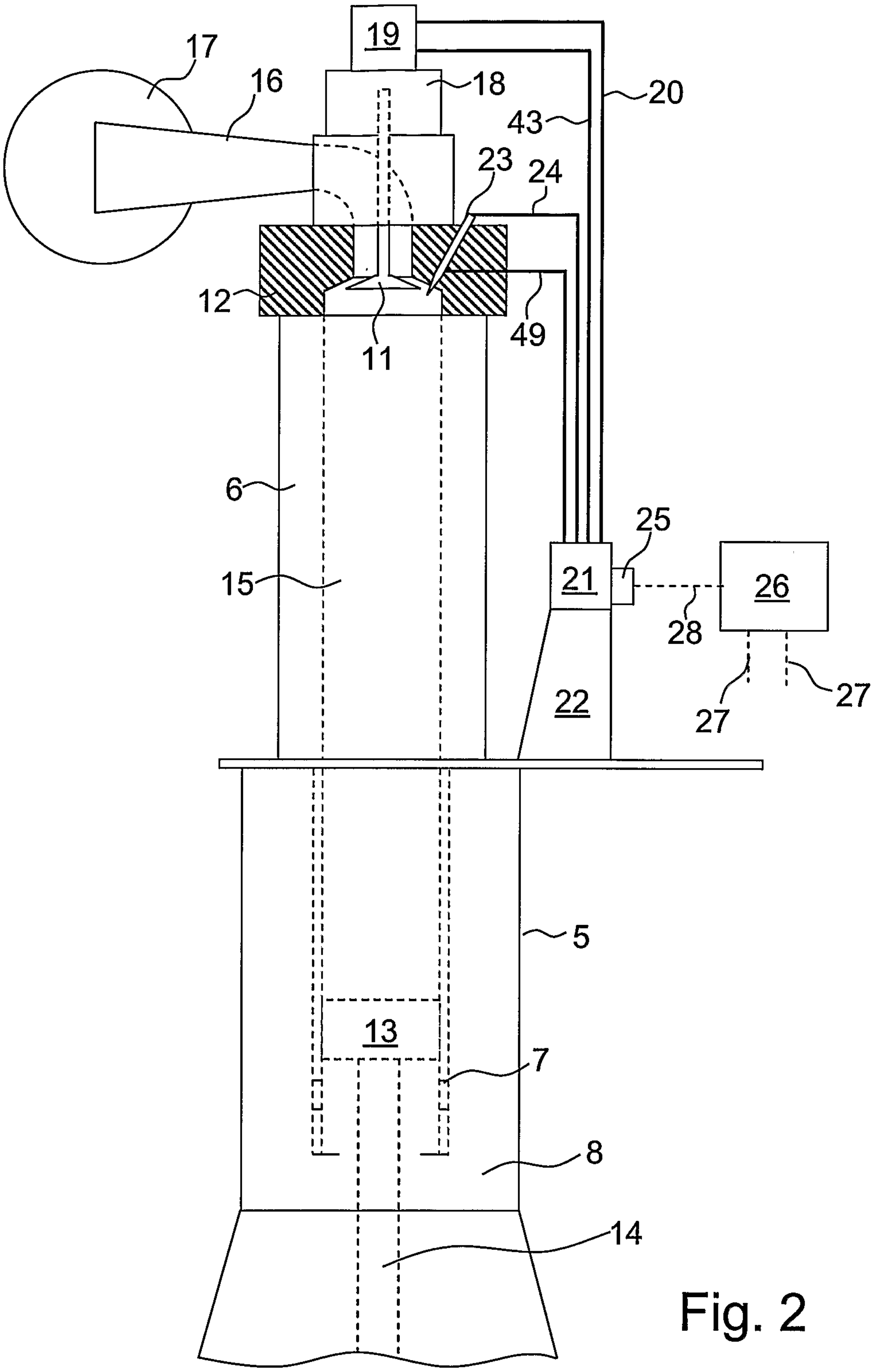

Fig. 2

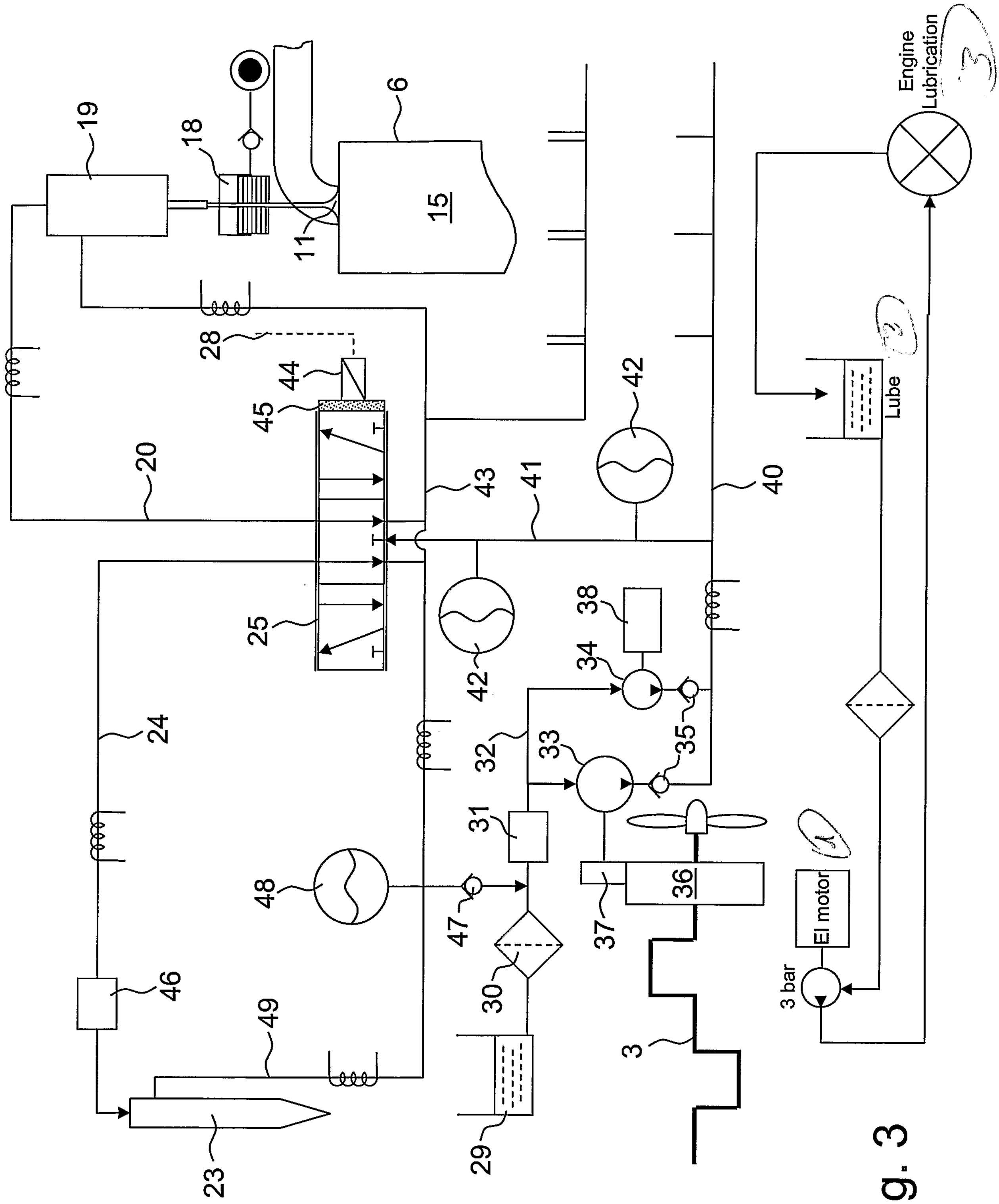
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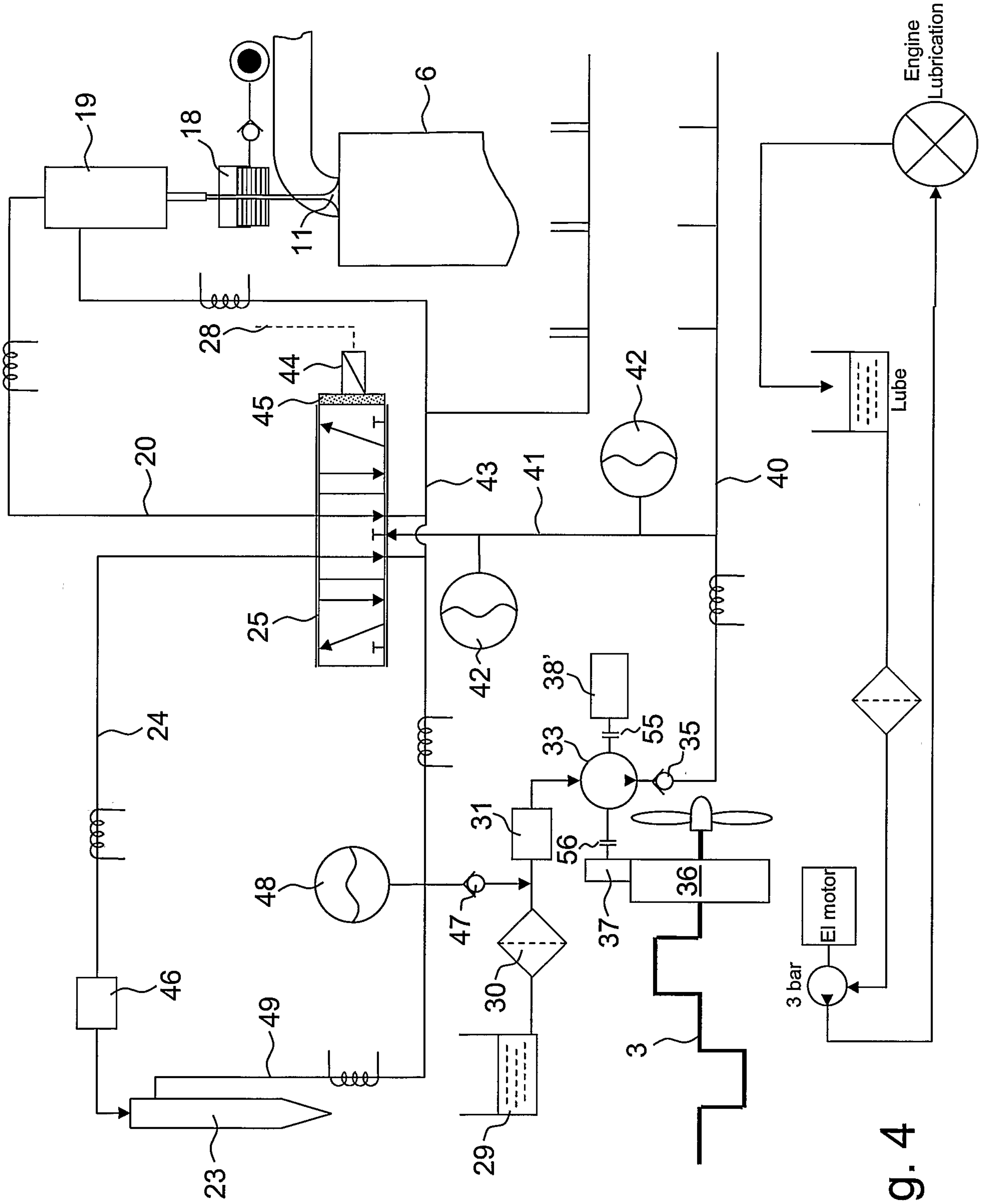
Fig. 3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El motor
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Fig. 4

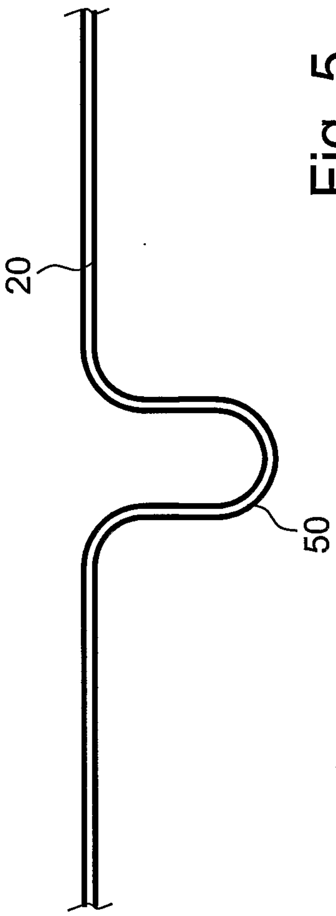

Fig. 5

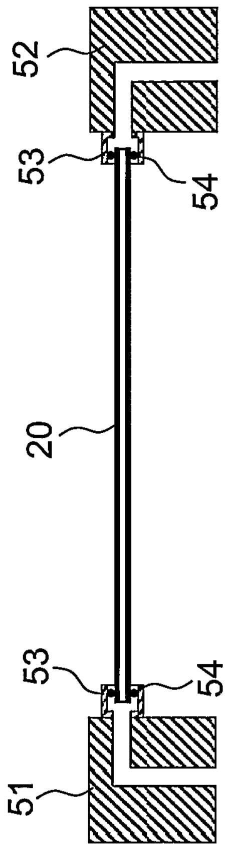

Fig. 6

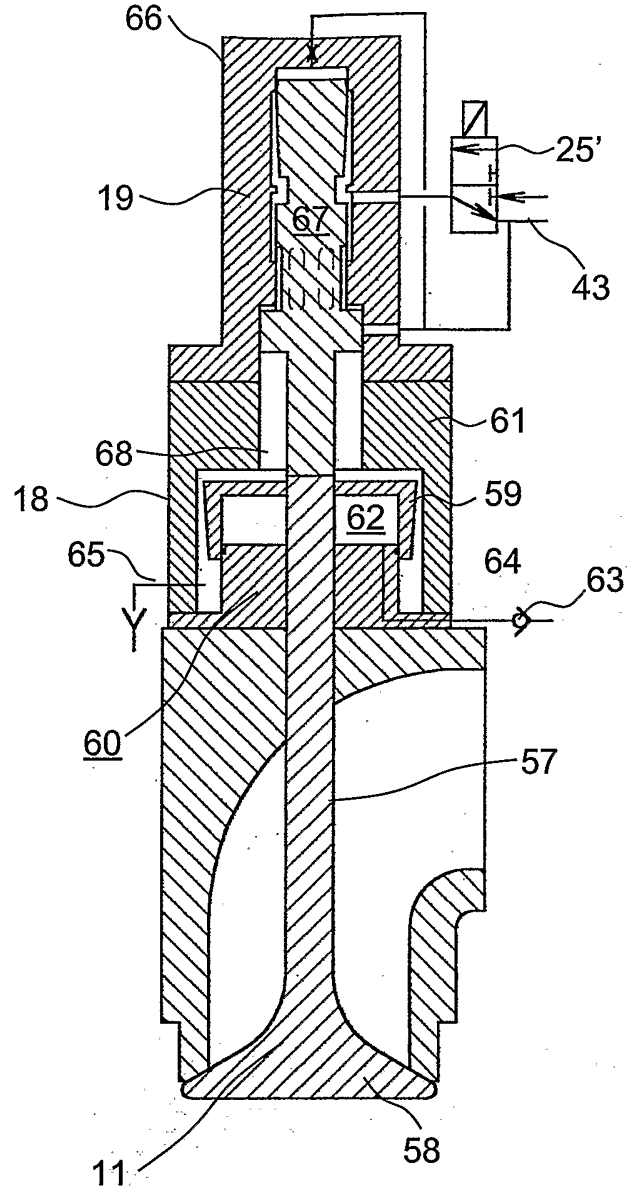

Fig. 7

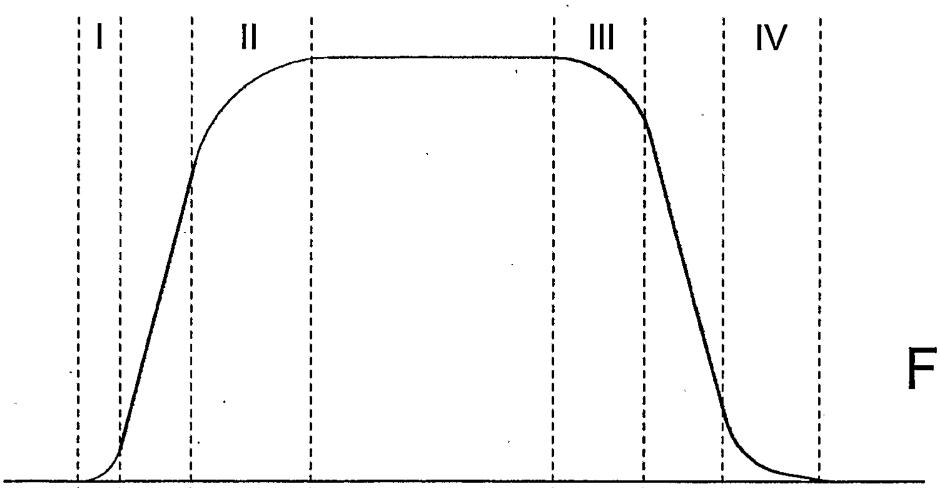

Fig. 8

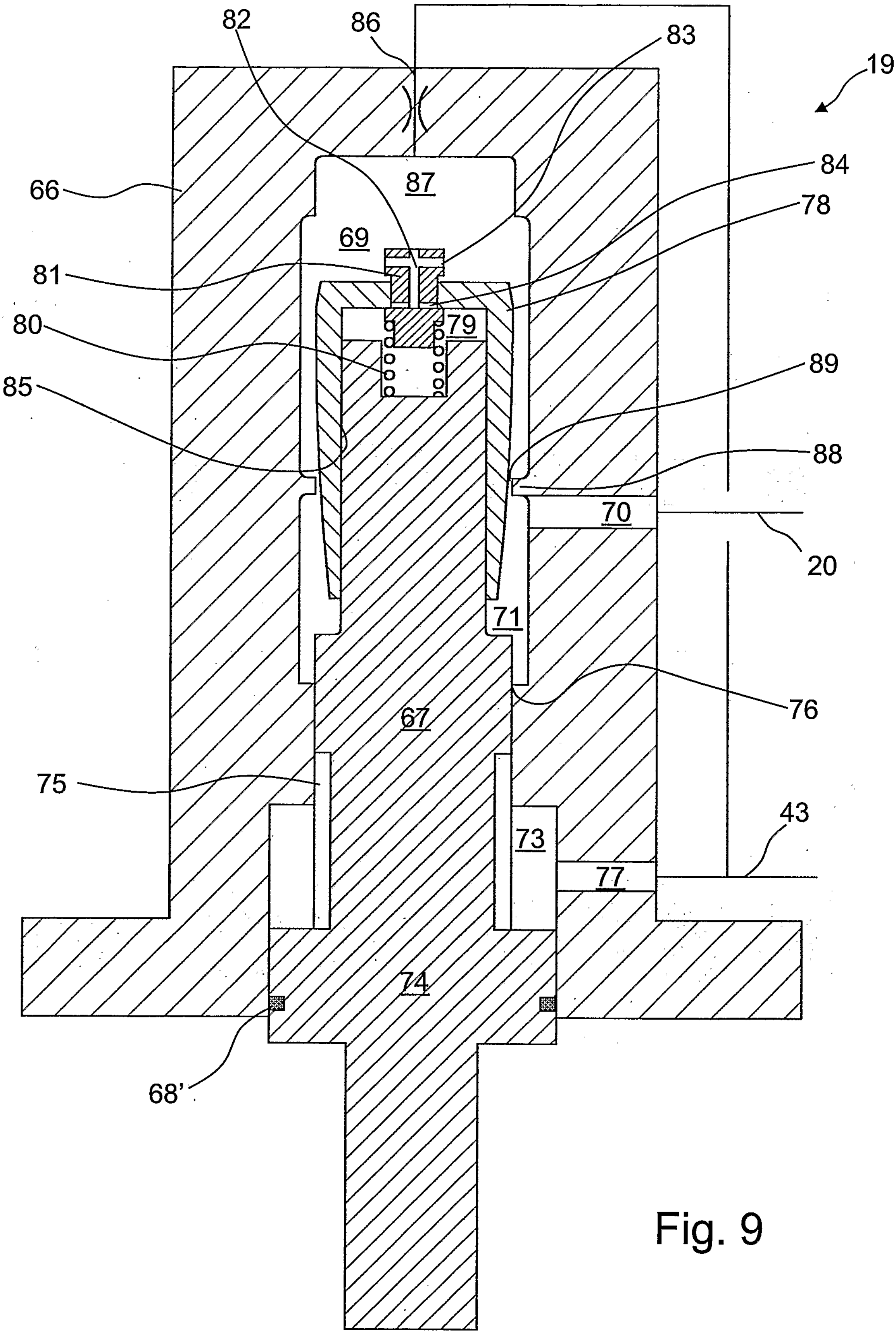
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Fig. 9

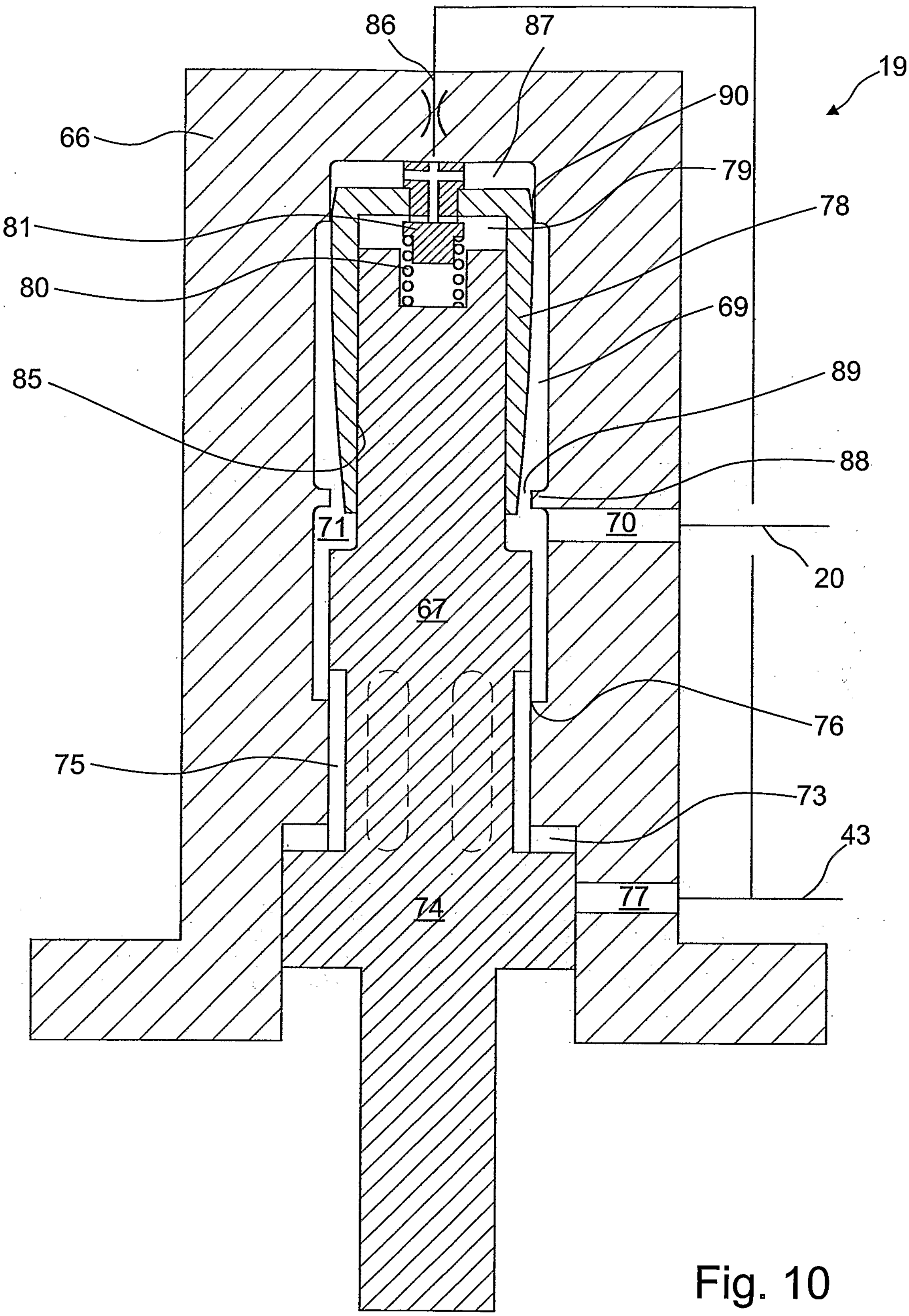

Fig. 10

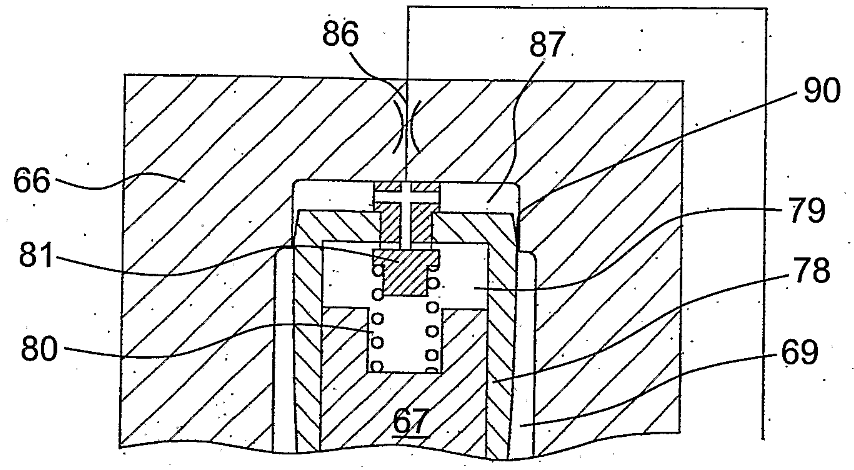

Fig. 11

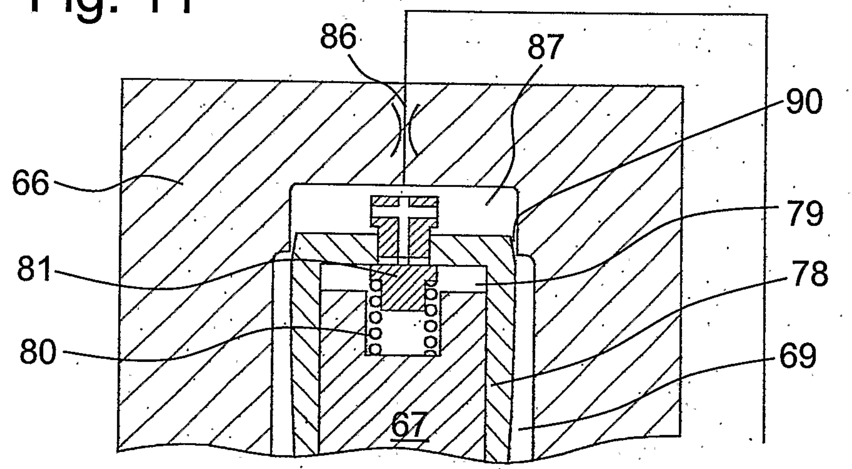

Fig. 12

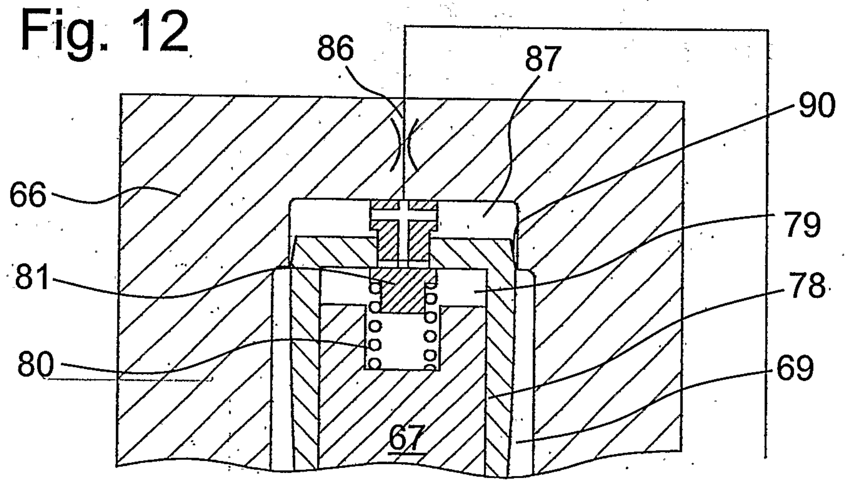

Fig. 13

INTERNATIONAL SEARCH REPORT

International Application No
PCT/EP2005/001040

A. CLASSIFICATION OF SUBJECT MATTER
F01L9/02 F02M55/02 F02M47/00

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F01L F02M

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, PAJ, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X A Y	EP 1 130 251 A (WAERTSILAE NSD SCHWEIZ AG) 5 September 2001 (2001-09-05) the whole document	1,2 3-34, 48-50 58-61
A	DE 197 47 240 A1 (MAN B & W DIESEL A/S, KOPENHAGEN/KOEBENHAVN, DK) 16 July 1998 (1998-07-16)	35-47
X	DE 103 11 493 A1 (MAN B & W DIESEL A/S, KOPENHAGEN/KOEBENHAVN) 7 October 2004 (2004-10-07) the whole document	51-53
X	US 6 561 165 B1 (HLOUSEK JAROSLAW) 13 May 2003 (2003-05-13) the whole document	54,55,57

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
10 March 2006	29 MAR 2006

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Klinger, T

INTERNATIONAL SEARCH REPORT

International Application No
PCT/EP2005/001040

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	EP 1 471 236 A (HITACHI, LTD) 27 October 2004 (2004-10-27) the whole document	58-61
X	GB 2 102 065 A (* SULZER BROTHERS LIMITED) 26 January 1983 (1983-01-26)	63-68
Y	the whole document	69-74
Y	WO 00/12895 A (DIESEL ENGINE RETARDERS, INC) 9 March 2000 (2000-03-09) the whole document	69-73
Y	US 2002/184996 A1 (LOU ZHENG) 12 December 2002 (2002-12-12)	74
A	the whole document	75-79

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
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