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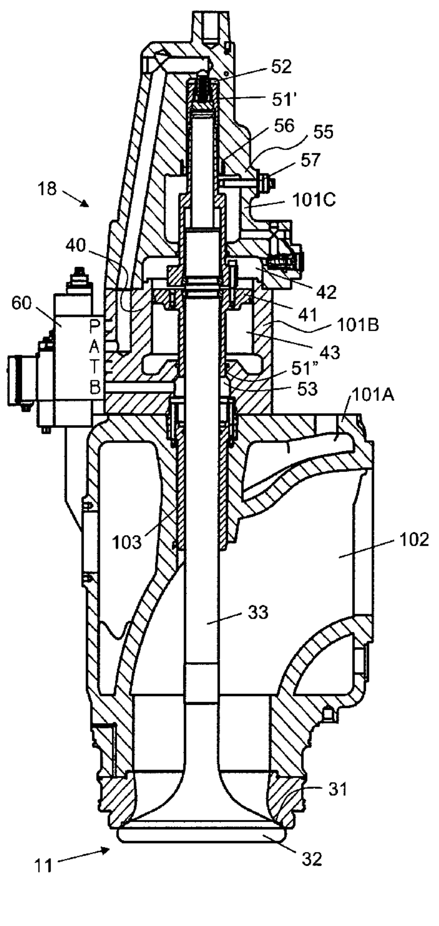
(54) **Title:** EXHAUST VALVE ASSEMBLY FOR A LARGE TWO-STROKE DIESEL ENGINE



(57) Abstract: The present invention relates to an exhaust valve assembly (18) for a large two-stroke diesel engine (1). The assembly (18) includes an exhaust valve (11) that is movable in opposite directions between a closed position and a variable open position. A double acting spring assembly (40) is operably connected to the exhaust valve (11) and forms a mass-spring system together with the exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11) . The double acting spring assembly (40) stores the exhaust valve (11) back and variable open position the exhaust valve (11) energy during translation of and forth between the closed for subsequent propulsion of in an opposite direction. Hydraulic means (50) hold the exhaust valve (11) on command from a controller (27) in the closed or in the variable open position.

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EXHAUST VALVE ASSEMBLY FOR A LARGE TWO-STROKE DIESEL ENGINE

The present invention relates to an exhaust valve assembly for an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type, in particularly to valve assembly that includes a fluidic operated and electronically controlled actuator.

BACKGROUND ART

Large two-stroke engines of the crosshead type, that are typically used in propulsion systems of large ships or as prime mover in power plants, have in the recent past evolved from camshaft controlled engines to electronically controlled engines. Electronic control offers an improvised flexibility over the timing and shaping of the fuel injection and the exhaust valve. The combustion process can thereby be better controlled resulting in a more effective combustion and lower emission values allowing smokeless operation at all running speeds, reduced part-load fuel consumption and lower minimum running speeds. CN 1485530 (CN filing), JP 2004-084670 (JP filing), KR 2004-20003 (KR filing) discloses a large electronically controlled two-stroke diesel engine. In this engine the exhaust valves are actuated by a hydraulic actuator that is powered with high pressure hydraulic fluid. The actuator urges the exhaust valve to open against the counter force of an air spring. Most of the energy delivered by the hydraulic actuator in the opening stroke of the exhaust valve is stored in the gas spring as potential energy. The stored energy is unlike cam shaft driven engines not reused in the closing stroke, but instead, wasted because there are no means to reuse it. The unused energy is transformed to heat going with the return oil to the tank of the hydraulic system. The amount of hydraulic energy that is used to open the exhaust valve on a large two-stroke diesel energy is quite significant, and a substantial part of the fuel savings obtained by the increased combustion control of the electronically controlled engine are lost in the exhaust valve actuation.

DISCLOSURE OF THE INVENTION

On this background, it is an object of the present invention to provide an exhaust valve assembly of the kind referred to initially that uses little hydraulic energy, and is flexible and accurate.

This object is achieved in accordance with claim 1 by providing an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type, said assembly comprising an exhaust valve that is movable in opposite directions between a closed position and a variable open position; a double acting spring assembly operably connected to the exhaust valve and forming a mass-spring system together with the exhaust valve and the mass of any other parts moving in unison with the exhaust valve, said double acting spring assembly storing energy during translation of the exhaust valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction, a position sensor that provides a signal indicative of the position of the exhaust valve, a controller in receipt of the signal from said position sensor, and hydraulic means for holding the exhaust valve on command from said controller.

After being released by the controller at a point in time that can be flexibly controlled by the controller the exhaust valve and other parts moving in unison therewith are propelled mainly by the potential energy stored in the double acting spring assembly and swings substantially unhindered from the closed position to the variable open position. Energy for the return stroke is stored in the double acting spring assembly when the exhaust valve is slowing down towards the variable open position. Thus, only the energy lost due to friction and flow resistance needs to be re-supplied to keep the exhaust valve moving. This is a significant energy saving compared to the valve assembly disclosed in CN 1485530 (CN filing), JP 2004-084670 (JP filing), KR 2004-20003 (KR filing). Via the signal from the position sensor the exhaust valve can be stopped by the hydraulic means for holding the exhaust valve under the command of the controller at exactly at the intended position, thereby improving the accuracy of the translative movement of the exhaust valve.

It is another object of the present invention to provide an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type that uses little hydraulic energy, and is flexible and accurate.

This object is achieved in accordance with claim 16 by providing an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type, said assembly comprising an exhaust valve that is movable between a closed position and a variable open position; a double acting spring assembly operably connected to the exhaust valve and forming a mass-spring system together with the exhaust valve and the mass of any other parts moving in unison with the exhaust valve; said double acting spring assembly storing energy during translation of the exhaust valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction; a position sensor that provides a signal indicative of the position of the exhaust valve; a controller in receipt of the signal from said position sensor; means for holding the exhaust valve in the closed position and in the variable open position; means for releasing the exhaust valve from being held in the closed position to allow the double acting spring assembly to propel the exhaust valve towards the variable open position; means for releasing the exhaust valve from being held in the variable open position to allow the double acting spring assembly to propel the exhaust valve towards the closed position; and hydraulic means for supplying additional energy to the exhaust valve on command from said controller to compensate for energy dissipation in the assembly.

The hydraulic means allow additional energy to be supplied in an efficient and flexible manner that can be varied in accordance with circumstances, such as the varying operating conditions of the engine. The controller may be configured to use the signal from the position sensor and optionally other parameters to determine the required amount of additional energy that is to be fed into the system, and configured to command said hydraulic means for supplying additional energy to deliver the determined amount of hydraulic energy.

The controller may also be configured to command said hydraulic means for supplying additional energy to provide an initial amount of energy that may be predetermined or determined by the controller on the basis of operating parameters of the engine.

The controller may be is configured to correct a deviation between the actual and expected velocity of the exhaust valve by either commanding said hydraulic means for supplying additional energy to deliver an amount of additional energy, or by commanding said hydraulic means for holding to slow down the exhaust valve.

The controller may be configured to use the signal from the position sensor and optionally other parameters to determine that the exhaust valve may overshoot or approach the valve seat too fast and is configured to command said hydraulic means for holding to slow down the exhaust valve before the exhaust valve overshoots or hits the valve seat with an excessive velocity.

It is another object of the present invention to provide an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type that uses little hydraulic energy, and is flexible and accurate.

This object is achieved by providing an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type according to claim 24 comprising an exhaust valve that is movable between a closed position and a variable open position, a double acting spring assembly operably connected to the exhaust valve and forming a mass-spring system together with the exhaust valve and any other parts moving in unison with the exhaust valve, said double acting spring assembly storing energy during translation of the exhaust valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction, a position sensor that provides a signal indicative of the position of the exhaust valve, a controller in receipt of the signal from said position sensor, means for holding the exhaust valve in the closed position and in the variable open position, means for releasing the exhaust valve from being held in the closed position to allow the double acting spring to propel the exhaust valve towards the variable open position, means for releasing the exhaust valve from being held in the variable open position to allow the double acting spring assembly to propel the exhaust valve towards the closed position, hydraulic means for supplying additional energy to the exhaust valve on command from said controller to compensate for energy dissipation in the assembly.

The amount of additional energy that is supplied, and the timing thereof is determined by the controller. The timing and amount can be adapted to the operating conditions to ensure that the exhaust valve reaches the extreme positions with a minimum need of dampening at the approach to the extreme positions.

The controller can be configured to use the signal from the position sensor and optionally other parameters to determine the required amount of additional energy that is to be supplied to the exhaust valve, and configured to command said hydraulic means for supplying additional energy to deliver the determined amount of hydraulic energy.

The controller may be configured to command said hydraulic means for supplying additional energy to provide an initial amount of energy that may be predetermined or determined by the controller on the basis of operating parameters of the engine.

The controller can be configured to use the signal from the position sensor and optionally other parameters to determine that the exhaust valve may overshoot or abut with the valve seat with an excessive velocity and is configured to command said hydraulic means for holding to slow down the exhaust valve before the exhaust valve overshoots or hits the valve seat with an excessive velocity.

It is a further object of the present invention to provide an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type that uses little hydraulic energy, and is flexible and accurate.

This object is achieved by an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type according to claim 24 and having a predetermined engine speed R_m in rounds per minute at its nominal maximum continuous rating, said assembly comprising an exhaust valve that is movable between a closed position and a variable open position, a double acting spring assembly operably connected to the exhaust valve and forming a mass-spring system together with said exhaust valve and the mass of any other parts moving in unison with the exhaust valve, said double acting spring assembly storing energy during translation of the gas exchange valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction, means for holding the exhaust valve in the closed position and in the variable open position, means for releasing the exhaust valve from being held in the closed position to allow the double acting spring to propel the exhaust valve towards the variable open position, means for releasing the exhaust valve from being held in the variable open position to allow the double acting spring to propel the exhaust valve towards the closed position, hydraulic means for supplying additional energy to the exhaust valve to compensate for energy dissipation in the assembly, and said mass-spring system being characterized by an Eigenfrequency in a range between 1 to 64 times R_m.

The Eigenfrequency determines the velocity and acceleration with which the exhaust valve will move from the one extreme position to another. When the Eigenfrequency is in the range between 1 to 64 times R_m the valve will open and close quickly enough to provide for an adequate and timely evacuation of exhaust gas in the combustion chamber.

The double acting spring assembly may use the compression of a gaseous medium to convert the kinetic energy of the exhaust valve and any parts moving in unison therewith into potential energy. The Eigenfrequency can be influenced by the base pressure of the gaseous medium in the double acting spring assembly. The Eigenfrequency may be adjusted by adapting the base pressure.

It is a further object of the present invention to provide an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type that uses little hydraulic energy, and is flexible and accurate.

This object is achieved by providing an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type according to claim 28 and having an engine speed R in rounds per minute, the assembly comprising, an exhaust valve that is movable between a closed position and a variable open position, a double acting spring assembly operably connected to the exhaust valve and forming a mass-spring system together with said exhaust valve, and the mass of any other parts moving in unison with the exhaust valve, said double acting spring assembly storing energy by during translation of the gas exchange valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction, means for holding the exhaust valve in the closed position and in the variable open position, means for releasing the exhaust valve from being held in the closed position to allow the double acting spring to propel the exhaust valve towards the variable open position, means for releasing the exhaust valve from being held in the variable open position to allow the double acting spring to propel the exhaust valve towards the closed position, hydraulic means for supplying additional energy to the exhaust valve to compensate for energy dissipation in the assembly, said double acting spring assembly being of the type in which a gaseous medium is compressed from a base pressure to a higher pressure to store the energy, and the gaseous medium expands from the higher pressure to the base pressure in order to propel the exhaust valve, a controller that is configured to determine a desired value for the Eigenfrequency of the mass-spring system and configured to control/adjust the base pressure of the gaseous medium in said double acting spring assembly accordingly.

The controller can adjust the Eigenfrequency at an appropriate level in accordance with the operating conditions. At low running speeds the Eigenfrequency can be lowered to reduce the load on the mechanical and hydraulic systems and also to reduce the amount of energy that is dissipated in the exhaust valve assembly.

The controller can determine the desired Eigenfrequency as a function of the actual engine speed. The controller may also determine the desired Eigenfrequency to be in a range of two to ten times the actual engine speed R.

It is still a further object of the invention to provide an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type that uses little hydraulic energy, and with an improved construction.

This object is achieved by providing an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type according to claim 37, said assembly comprising a valve housing, an exhaust valve that is movable between a closed position and a variable open position, a double acting air spring assembly operably connected to the exhaust valve and forming a mass-spring system together with said exhaust valve and the mass of any other parts moving in unison with the exhaust valve, said double acting air spring assembly storing energy during translation of the gas exchange valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction, two hydraulic piston cylinder arrangements acting in opposite directions for holding the exhaust valve in the closed position and in the variable open position, and for releasing the exhaust valve from being held in the closed position to allow the double acting spring to propel the exhaust valve towards the variable open position, for releasing the exhaust valve from being held in the variable open position to allow the double acting spring to propel the exhaust valve towards the closed position, and for supplying additional energy to the exhaust valve to compensate for energy dissipation in the assembly.

This exhaust valve assembly is able to operate with a minimum amount of hydraulic energy.

The double acting air spring may include a piston that is fastened to the valve stem by two oppositely disposed wedges. The wedges may be formed by wedge rings or wedge bushings.

The exhaust valve assembly may comprise a flange that is disposed over the valve stem facing the spring piston.

One of said wedge rings may be received in a conical collet bore that is disposed in the spring piston, the other wedge ring may be received in a conical collet bore disposed in said flange, said flange can be tightened to the spring piston and thereby the oppositely disposed wedges may be driven into their respective collet bore and tightened radially against the periphery of the valve stem to thereby fasten the spring piston and the wedge bushing to the valve stem.

The flange may be tightened to the spring piston by tension bolts assembled through the flange.

The hydraulic piston of the hydraulic piston cylinder arrangement that urges the exhaust valve in the closing direction can be a ring piston that is disposed around the stem of the exhaust valve.

The ring piston can be formed by a sleeve connected to the spring piston. The sleeve may be directed towards the valve head and be integral with the spring piston.

A hydraulic valve block for controlling the flow of hydraulic fluid to and from the hydraulic piston-cylinder assemblies may be attached directly to the valve housing.

The exhaust valve assembly may further include a position sensor that comprises a tapered sleeve of a Ferro-magnetic material. The tapered sleeve may be an integral part of said busing.

It is yet another object of the invention to provide an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type that uses little hydraulic energy, and has improved safety aspects.

This object is achieved by providing an exhaust valve assembly for a large two-stroke diesel engine of the crosshead type in accordance with claim 55, said assembly comprising an exhaust valve that is movable in opposite directions between a closed position and a variable open position, a double acting spring assembly operably connected to the exhaust valve and forming a mass-spring system together with the exhaust valve and any other parts moving in unison with the exhaust valve, said double acting spring assembly including two spring chambers for storing energy during translation of the exhaust valve back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve in an opposite direction, hydraulic means for holding the exhaust valve on command from said controller, means for evacuating the spring chamber that urges the exhaust valve in the opening direction when the hydraulic pressure falls away or drops below a predetermined threshold so that the exhaust valve will automatically assume the closed position when the hydraulic pressure falls away or drops below the predetermined threshold.

The means for evacuating may comprise a relief valve that is urged towards an open position by resilient means and a urged towards a closed position by a hydraulic piston, so that the resilient means will urge the relief valve to the open position when the hydraulic pressure falls away or drops below the predetermined threshold.

Further objects, features, advantages and properties of the valve assemblies according to the invention 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. 1A shows a layout diagram of a large two-stroke diesel engine,

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 of the main features of the exhaust valve assembly according to the invention,

Fig. 4 is a cross-sectional view through an embodiment of a control valve for use with the invention,

Fig. 5 is a cross-sectional view through another embodiment of a control valve for use with the invention,

Fig. 6 is diagrammatic representation of an embodiment of the exhaust valve assembly according to the invention,

Fig. 7 is diagrammatic representation of another embodiment of the exhaust valve assembly according to the invention,

Fig. 8 is diagrammatic representation of yet another embodiment of the exhaust valve assembly according to the invention,

Fig. 9 is a cross-sectional view through yet another embodiment of a control valve for use with the invention,

Fig. 10 is a system diagram of an embodiment of the exhaust valve assembly according to the invention,

Fig. 11 is an operation chart illustrating the positions of a control valve relative to the movement of the exhaust valve,

Fig. 12 is a diagrammatic representation of a further embodiment of the exhaust valve assembly according to the invention,

Fig. 13 is a diagrammatic representation of another embodiment of the exhaust valve assembly according to the invention,

Fig. 14 is a cross-sectional view through an embodiment of the exhaust valve assembly according to the invention,

Fig. 15 is a detail of Fig. 14,

Fig. 16 is the cross-sectional view of Fig. 14 with highlighted details,

Fig. 17 is a cross-sectional view through another embodiment of the exhaust valve assembly according to the invention,

Fig. 18 is a cross-sectional view through an yet another embodiment of the exhaust valve assembly according to the invention, and

Fig. 19 is a cross-sectional view through a further embodiment of the exhaust valve assembly according to the invention.

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. An exhaust valve assembly 18 is mounted on the top of each cylinder 6. The cylinder frame 5 also carries an exhaust receiver 17, a turbo charger 10 and a scavenge air receiver 9.

Fig. 1A shows a layout diagram for engine power and speed. The diagram includes four layout points L1,L2,L3 and L4. L1 designates nominal maximum continuous rating (nominal MCR), at 100% engine power and 100% engine speed R_m. The 100 % engine speed (R_m) of a large two-stroke diesel engine is typically in the range of 76 (for the largest types with a cylinder bore of up to 1080 mm) to 250 RPM (for the smaller types with a bore down to 260 mm). Engine power ranges from 100.000 Kw (for the largest types) to 1600 KW (for the smaller types). L2, designates the minimum engine power at maximum engine speed, L3 designates maximum engine power a minimum engine speed and L4 designates minimum engine power at minimum engine speed.

Fig. 2 shows a cylinder 6 of an internal combustion engine in further detail. The cylinder 6 is of the uniflow type and has scavenge air ports 7 located in an airbox 8, which from the scavenge air receiver 9 (Fig. 1), is supplied with scavenge air pressurized by the turbocharger 10 (Fig. 1). A not shown crosshead connects the piston rod 14 via a not shown crank 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 the 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. 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 and closed by means of an exhaust valve assembly 18 that includes air springs and hydraulic piston cylinder assemblies. A valve block is mounted directly to the housing of the exhaust valve assembly 18. The valve block includes a control valve 60 that controls the flow of hydraulic fluid to and from the exhaust valve assembly under command of a control computer 27 via line 30. The control computer 27 receives feedback and other signals via lines 28. The valve block is supplied with high pressure hydraulic fluid through a pressure conduit 21 connecting an inlet port on the valve block with a port on the top surface of a distributor block 22 supported by a console 23. A return conduit 20 connects an outlet port of the valve block to a return port on the top surface of the distributor block 22. The distributor block 22 is connected to a source of high pressure hydraulic fluid and to a tank (not shown in Fig. 2).

Each cylinder 6 is provided with two or three injectors 24 (only one is shown) connected by a ring conduit (not shown). The fuel is supplied from distributor block 22 to the injectors 24 through feed conduit 25. The injectors 24 are connected to a return port on the distributor block 22 via a return conduit 19.

Fig. 3 is a diagrammatic representation of an exhaust valve assembly 18 according to an embodiment of the invention. The exhaust valve 11 can move between a closed position in which the valve head 31 abuts with the valve seat 32, and an open position. The degree of opening of the exhaust valve 11 may vary depending on e.g. the operating conditions. A double acting air spring 40 is operably connected to the exhaust valve by a spring piston 41 that is secured to the valve stem 33. The double acting air spring 40 comprises two spring chambers 42 and 43. Spring chamber 42 is compressed when the exhaust valve moves towards the closed position, whilst spring chamber 43 is compressed when the exhaust valve 11 moves towards the variable open position. The spring chambers 42 and 43 function as potential energy accumulators. When the spring chamber 42 is compressed the potential energy accumulated therein can drive the exhaust valve 11 in the opening direction. When the spring chamber 43 is compressed the energy accumulated therein can drive the exhaust valve 11 in the closing direction. When the exhaust valve 11 is substantially in the middle between the closed and variable open position, as shown in Fig. 3, the pressure in the two spring chambers 42 and 43 is substantially equal and the double acting air spring 40 does in this position not effectively apply any force to the exhaust valve 11. The air spring 40 can alternately be formed by two separate cylinders with their respective spring pistons (not shown).

The air spring 40 forms in combination with the mass of the exhaust valve 11 and the mass of any parts moving in unison with the exhaust valve 11, such as the spring piston 41, a mass-spring system that - once set in motion - can oscillate between the closed position and the variable open position mainly using the energy that is stored and released in spring chambers 42 and 43 of the double acting gas spring 40. The kinetic energy of the exhaust valve 11 and any other parts moving in unison therewith is transformed to potential energy in the spring chambers of gas spring 40 and vice versa. Only the energy lost due to friction and viscous dissipation needs to be replenished in order to keep the exhaust valve 11 oscillating between the closed and open positions, i.e. to avoid that the oscillating movement dampens out.

A double acting hydraulic cylinder 50 is also connected to the valve stem 33. The hydraulic cylinder 50 includes a double acting piston 51 that is secured to the valve stem 33. The double acting piston 51 divides the hydraulic cylinder into a pressure chamber 52 and a pressure chamber 53. Alternatively, the exhaust valve 11 can be connected to two separate hydraulic cylinders instead of one double acting cylinder (described further below).

Pressure chamber 52 is connected to a port A of a control valve 60 via a conduit 66. Pressure chamber 53 is connected to a port B of a control valve 60 via a conduit 67. The control valve 60 is via a port P connected to a source of high pressure hydraulic fluid, such as a common hydraulic pressure rail of the engine (not shown) or a high pressure pump or pumping station (not shown). The control valve 60 is connected via a port T and a conduit 71 including a non-return valve 72 to a hydraulic fluid reservoir or tank 73.

When pressure chamber 52 is pressurized it urges the exhaust valve 11 in the opening direction, whilst pressure chamber 53 urges the exhaust valve 11 in the closing direction when pressurized. The pressurization of the pressure chambers 52 and 53 is controlled by the control valve 60 under command from the control computer 27 (Fig. 2). The exact operation will be described in greater detail below. In principle only the energy that is lost through viscous dissipation and by friction needs to be supplied via the double acting hydraulic cylinder 50 in order to maintain the oscillating movement of the exhaust valve 11.

For use in a large two-stroke diesel engine of the crosshead type the exhaust valve 11 must be kept closed during a part of the engine cycle and open during another part of the engine cycle, i.e. the exhaust valve 11 cannot be allowed to oscillate freely between the open and closed positions. The oscillating movement of the exhaust valve 11 can be halted at the top or bottom of the sine wave, i.e. in the closed and open positions and remain there without any significant energy loss. When the exhaust valve 11 is later released, the potential energy in the respective spring chamber 42 and43 will transform into kinetic energy in of the exhaust valve 11 and the oscillating movement will continue. Despite the stops, the oscillating movement of the exhaust valve 11 can be characterized by an Eigenfrequency that determines the velocity profile of the exhaust valve 11.

It has to be ensured that the exhaust valve 11 moves from the open to the closed position and vice versa sufficiently quick. In a system with a variable Eigenfrequency it is necessary that the Eigenfrequency of the mass-spring system expressed in Hertz is at any time at least equal two to ten times the actual crankshaft speed R expressed in rounds per minute (RPM). If the mass-spring system operates with a fixed Eigenfrequency, this fixed Eigenfrequency has to be at least equal to the crankshaft speed R_m in rounds per minute (RPM) at the nominal maximum continuous rating of the engine, and is preferably in the range between 2 to 60 times R_m.

The oscillating movement of the exhaust valve 11 is halted in either of the two extreme positions by the double acting hydraulic cylinder 50. In Fig. 3 the control valve 60 is by way of example illustrated as a 6-position four-way valve. The same hydraulic connections could also be obtained with other valve arrangements to obtain a hydraulic control system that can supply additional energy to the oscillating movement of the exhaust valve 11 and halt the exhaust valve 11 in the closed and in the open positions.

The hydraulic control valve 60 operates to create and end connections between ports A,B,P and T in patterns that are useful for controlling the exhaust valve 11. These patterns are illustrated in Fig. 3 by listing the modes of the hydraulic control system:

Mode 1: port A is connected to port T and port P is connected to port B. In this mode pressurized fluid is led to pressure chamber 53, whilst pressure chamber 52 is connected to the tank or drain. This mode is used to halt the exhaust valve in the closed position, and the potential energy stored in spring chamber 42 is not released. This mode can also be used for forced closing of the exhaust valve 11, and for supplying additional energy to the exhaust valve 11 during the closing stroke.

Mode 2: port P is connected to port A and port B is connected to port T. In this mode pressurized fluid is led to pressure chamber 52, whilst pressure chamber 53 is connected to the tank or drain. This mode can be used for forced opening of the exhaust valve 11 (if needed), and for supplying additional energy to the exhaust valve during the opening stroke.

Mode 3: both port A and B are connected to port T. Pressure chambers 52 and 53 are connected to one another and to tank. This will allow fluid to flow from pressure chamber 52 to pressure chamber 53, and vice versa. The non-return valve 72 supports this effect. This mode is used to allow the exhaust valve 11 to move freely from the closed position to the open position driven by the potential energy in the spring chamber 42 and from the open position to the closed position driven by the potential energy in the spring chamber 43. In this mode the movement of the exhaust valve is substantially unhindered and unsupported by the double acting hydraulic cylinder 50, and in this mode the exhaust valve 11 will substantially behave as a freely oscillating mass-spring system.

Mode 4a: all ports as closed, thereby entrapping the hydraulic fluid in both pressure chambers 52 and 53 and preventing movement of the double acting hydraulic piston 51.

Mode 4b: port A and T are closed. Port P is connected to port B. The hydraulic fluid in pressure chamber 52 is entrapped, whilst high pressure will be maintained in pressure chamber 53.

Mode 4c: port B and T are closed. Port P is connected to port A. The hydraulic fluid in pressure chamber 53 is entrapped, whilst high pressure will be maintained in pressure chamber 52.

When mode 4a, 4b or 4c is engaged the movement of the exhaust valve 11 will come to an end. Only one of the modes 4a,4b and 4c is needed in a particular design. This mode is used to halt the exhaust valve 11 in the open position, or at any other desired point. Which mode is used depends on the configuration of the hydraulic system.

The control valve 60 is preferably a proportional valve, so that the transitions between the respective modes can be smooth, with a little throttling, thereby minimizing pressure peaks in the system due to abrupt breaking of masses and fluid hammering effects. Mechanical loads due to high accelerative forces are thereby also reduced.

Fig. 4 shows in greater detail an embodiment of the control valve 60. In this embodiment the control valve 60 is a spool valve with internal connections illustrated by the symbolic representation of the valve on the right side of Fig. 4. The control valve includes a housing 61 in which a main spool 62 is disposed, an electrically driven pilot valve 63, an electronic regulator 64 and a linear position transmitter 65. The regulator 64 receives a command signal from the control computer 27 and a feedback spool position signal from the linear position transmitter. The regulator 64 controls the position of the spool 62 in a well known closed loop manner.

Fig. 5 shows an alternative embodiment of the control valve 60, where ports A and B have been interchanged. All other features of this valve are equal to the embodiment shown in Fig. 4 and this valve has the same symbolic representation as the control valve of Fig. 4.

The hydraulic control system can, as illustrated in Fig. 3, be composed of a single hydraulic valve, or could be formed by a combination of one or more hydraulic valves, possibly combined with one or more non return valves. An example of such an alternative embodiment is the combination of two three-position three-way valves 60A and 60B shown in Fig. 6. A low pressure source 75 keeps the system filled at any time during operation.

Fig. 7 shows an alternative embodiment of the hydraulic system with a relatively less complicated spool construction than the embodiment of Figs. 4 and 5. The control valve 60 according to this embodiment is a five position four-way spool valve that is combined with non-return valves 68 and 69.

Fig. 8 shows another preferred embodiment of the hydraulic control valve and system, which is substantially identical to the embodiment of Fig. 7, however, the non-return valves 68 and 69 have been incorporated into the spool valve.

Fig. 9 shows the mechanical outline of the control valve 60 of the embodiment of Fig. 8 and its symbolic representation. The complete system using the control valve 60 of Figs. 8 and 9 is shown in Fig. 10. A feed conduit 91 branches out to pressure conduits 92 and 93 to provide high pressure air from a source of high pressure air to the spring chambers 42 and 43. The pressure in the supply conduit 91 is kept a base pressure which is typically in the range of 3 to 10 bar. Branch 92 leads to spring chamber 42 and includes a non-return valve 94 and a relief valve 95. Branch 93 leads to spring chamber 43 and includes a non-return valve 96 and a relief valve 97. The non-return valves 94 and 96 allow the respective spring chambers 42 and 43 to be supplied with additional air at base pressure to compensate for any air that leaks from the spring chambers 42 and 43 during operation. The pressure relief valves 95 and 97 allow the spring chambers 42 and 43 to be evacuated when the pressure in the spring chambers exceeds a predetermined threshold.

A position sensor 55 measures the position of the exhaust valve 11. In a preferred embodiment the position sensor 55 is an eddy current type sensor that includes a conical element 56 of a ferromagnetic material moving in unison with the valve stem 33 and a stationary pick up element 57 that is secured to the housing of the exhaust valve assembly 18. Other sensor types such as sensors that include a linear variable differential transformer or sensors that include magnetostrictive rod can be used instead. The signal of the position sensor 55 is transmitted to the control computer 27 (controller) via line 28.

In this embodiment two optional hydraulic end of stroke dampers 58 and 59 are included in the hydraulic cylinder 50. End of stroke damper 59 secures the exhaust valve 11 from damage in case of overshooting of the exhaust valve in the opening stroke. End of stroke damper 58 secures the exhaust valve 11 from damage in case of an eminent landing of the valve head 31 of the valve seat 32 with an excessive velocity. End of stroke damper 58 can be of the self adjusting type.

The control computer 27 provides an electric control signal to the control valve 60 via line 30. The hydraulic control system is identical to the control system of Fig. 8, however, an air release valve 82 and an orifice 83 have been added. The air release valve 82 serves to de-aerate the hydraulic system, whilst the orifice 82 allows the oil in the hydraulic control system to be refreshed to prevent overheating or other damaging circumstances.

<u>Operation</u>

With reference to Fig. 11 the operation of the exhaust valve assembly 18 is described in further detail. In the chart of Fig. 11 the position of the exhaust valve 11 and the position of the control valve 60 are plotted against time for one opening and closing movement of the exhaust valve 11, which occurs once per revolution of the crankshaft 3. At the start of the chart the hydraulic control valve 60 is kept by the control computer 27 in position "1" and the exhaust valve 11 is forced by the double acting hydraulic cylinder 50 to stay in the closed position. The control computer 27 determines on the basis the actual crankshaft angular position, crankshaft rotational speed (the sensors that provide these signals are not shown), and perhaps also on the basis of other parameters such as ambient pressure and temperature, etc., and in dependence of the operator selected engine program (fuel economy program, low emission program, etc.) when it is time to open the exhaust valve 11. Then, the control computer 27 commands the control valve 60 to move to position "5", thereby sending pressurized oil into pressure chamber 52, and connecting pressure chamber 53 to tank. Thus, the double acting hydraulic piston 50 urges the exhaust valve 11 in the opening direction to give a power pulse of length "a". The length the power pulse is determined by the control computer 27 on the basis of feedback signals including the crank shaft angular position and rotational speed and the position and velocity of the exhaust valve spindle 33. The measured pressure in the hydraulic pressure supply line may also be measured and used to fine tune the length "a" of the power pulse. Based on these feedback signals, the impulse length is calculated by the control computer 27 to be no longer than required to overcome the opening forces of the pressurized exhaust gas in the combustion chamber pressing on the spindle. The length of "a" can therefore be adapted to the load of the engine, to use only the amount of pressurized hydraulic fluid as needed and therefore minimizing the energy consumption: for instance, at low load, the impulse "a" may not be needed as the force of air spring 40 will be sufficient to open the exhaust valve 11.

When the power pulse has been executed, the control computer 27 commands the control valve 60 to take position "4", thereby letting the mass-spring system moving unhindered towards the open position.

An additional power pulse with a length "b" can optionally be added during the opening stroke of the exhaust valve 11. However, this is optional and will only be performed if the control computer 27, based on the feedback signals, determines the opening slope to be not steep enough to open the exhaust valve 11 to the desired opening height. This optional power pulse can be useful in the overall performance of the exhaust valve 11 with minimum energy consumption. After optional pulse "b" the exhaust valve mass-spring system returns to an unhindered movement with the control valve commanded by the control computer 27 to position 4.

When the exhaust valve 11 is reaching the fully open position its velocity will be decreased as the kinetic energy of the exhaust valve and the masses moving in unison therewith is converted to potential energy in spring chamber 43. Power pulses with length "a" and "b" correct any deviance from the ideal lifting height.

When the signal from the position and velocity sensor 55 indicates that the exhaust valve 11 has reached its fully open position, the control computer 27 commends the control valve 60 to assume position "3", thereby preventing the exhaust valve 11 to commence its return stroke. The control valve 60 is moved gently, i.e. with a little throttling, to position "3" when the control computer 27 detects a potential overshoot of the exhaust valve 11 in order to avoid shockwaves in the hydraulic system.

The control computer 27 determines when it is time to close the exhaust valve 11, based on angular crankshaft position and rotational speed, other parameters, and on the program selected by the engine operator. At this point in time the control computer 27 commands the control valve 60 to assume position 2, and the potential energy stored in spring chamber 43 drives the exhaust valve 11 to the closed position.

Just before the exhaust valve reaches the fully closed position, the control computer commands the control valve 60 to assume position "1" giving a closing pulse to the exhaust valve 11. Based on the control computer's input, the closing pulse will be profiled to close the exhaust valve 11 with the desired closing velocity. In this embodiment, the hydraulic end of stroke dampener 58 ensures a soft landing of the valve head 31 on the valve seat 32.

When the exhaust valve 11 has reached the closed position the control computer 27 commands the control valve 60 to assume position "1" to keep the exhaust valve 11 closed, and this concludes the cycle.

Fig. 12 shows another preferred embodiment of the exhaust valve assembly 18. This embodiment differs from the embodiment of Figs. 9 and 10 mainly in that the Eigenfrequency of the mass-spring system can be varied. Hereto, the base pressure in the spring chamber 42 and 43 is adjusted to a level that corresponds to the desired Eigenfrequency. The control computer 27 determines the desired Eigenfrequency for the mass-spring system on the basis of the rotational speed of the crankshaft 3, and optionally also on the basis of other parameters, such as the operation program selected by the operator. As described above, the control computer 27 will typically keep the Eigenfrequency of the mass-spring system in the range of two to ten times the actual crankshaft speed R in rounds per minute (RPM).

The feed conduit 91 branches out to pressure conduits 92 and 93 to provide high pressure air from a source of high pressure air to the spring chambers 42 and 43. The pressure in the feed conduit 91 is regulated by a pressure reducing valve 90 that is commanded by the control computer 27 with the assistance of a pressure feedback signal from a pressure sensor 99 that senses up the actual pressure in pressure conduit 91.

A pressure conduit 92 leads to spring chamber 42 and includes a non-return valve 94 and a pressure controlled relief valve 95'. A pressure conduit 93 leads to spring chamber 43 and includes a non-return valve 96 and a relief valve 97'. The non-return valves 94 and 96 allow the pressure chambers 42 and 43 to be supplied with additional air at the regulated pressure in the pressure conduit 91 to increase the pressure in the spring chambers 42 and 43 when the control computer 27 has determined that the Eigenfrequency needs to be risen.

The relief valves 95' and 97' are pilot operated by a pneumatic piston instead of a mechanical spring. The controlled pressure in the feed conduit 91 acts on the pneumatic piston. The opening pressure of the relief valves 95' and 97' is therefore an amplification of the controlled pressure in the feed conduit 91. The amplification factor of the relief valves 95' and 97' is given by the difference in area of the valve seating and the pneumatic piston, and is selected to be substantially equal to the compression ratio of the spring chambers 42 and 43. The pressure relief valves 95' and 97' allow the spring chambers 42 and 43 to evacuate some air when the controlled pressure in the feed conduit 91 is lowered under command of the control computer 27.

In this embodiment the pressure reducing valve 90 (or pressure regulating valve) is common for both spring chambers 42 and 43, so only a single pressure reducing valve 90 is needed for the engine 1.

Fig. 13 shows another preferred embodiment that includes two pressure reducing valves 90 and 90', where pressure reducing valve 90 regulates the airflow to all spring chambers 42 of the engine, and pressure reducing valve 90' regulates the airflow to all spring chambers 43 of the engine. This embodiment allows for different pressures in upper and under air cylinders. This embodiment allows for different pressures in the upper spring chamber 42 and the lower spring chamber 43. Thereby, the amplitude of the free vibration can be influenced. For example, by lowering the feed pressure in the upper spring chamber 42, the natural lifting height of the exhaust valve 11 can be reduced. With a reduced lifting height the amount energy required to move the exhaust valve can be further reduced at low engine loads.

The system illustrated with reference to Figs. 12 and 13 allows the characteristics of the air spring to be changed by altering the air pressure in the spring chambers 42 and 43 smoothly when the engine is running.

Figs. 14,15 and 16 show the mechanical construction of a preferred embodiment of the exhaust valve assembly 18 in greater detail. The exhaust valve assembly 18 includes a three part housing 101A, 101b and 101C. The lower housing 101A includes the valve seat 31, the first part 102 of the exhaust passage 16 and includes a bushing 103 for guiding and sealing the valve stem 33. The lower housing part 101A is fitted into the cylinder cover 12 (Fig. 2). The middle housing 101B includes the main part of the double acting spring cylinder spring 40 and of the pressure chamber 53. The control valve 60 is directly mounted to the middle housing part 101B for minimizing the length of the hydraulic conduits between the control valve 60 and pressure chambers 52 and 53. The upper housing part 101C includes a minor part of the double acting air cylinder 40, and the pressure chamber 52. The stationary pick up element 57 of the position sensor 55 is mounted in the upper housing 101C.

The spring piston 41 is provided with an upwardly opening conical collet bore in with a wedge ring 110 received therein (Fig. 16Z). The wedge ring 110 secures the spring piston 41 to the valve stem 33 by a wedge effect. The wedge effect is augmented by an inwardly projecting protrusion on the wedge ring that fits into a circumferential groove 112 in the valve stem 33. This construction is as such very suited to withstand upwardly directed forces that are applied to the spring piston 41. However, the compressed air in spring chamber 42 exerts significant downwardly directed forces on the spring piston 41. Therefore, a flange 114 facing the spring piston 41 with a downwardly directed collet bore is placed above the spring piston 41. A wedge ring 116 engaging a circumferential groove 113 with an inward protrusion is received in the collet bore. The flange 114 is tightened to the spring piston 41 by several tension bolts 117 that are assembled through the flange, thereby driving the wedge rings 110 and 114 into their respective collet bore. The wedge rings 110 and 114 are thereby radially tightened to against the periphery of the valve stem 33, thus creating a secure engagement with the valve stem. The two oppositely disposed wedge rings 110 and 116 ensure that the spring piston 41 to valve stem 33 connection can withstand both upwardly and downwardly directed forces.

The spring piston 41 includes a downwardly extending sleeve 47, in sealing engagement with a bore in the middle housing part 101B. The pressure chamber 53 is formed below the bore. The downwardly extending sleeve 47 plunges into the pressure chamber 53 when the exhaust valve 11 moves downwards, and the downwardly extending sleeve 47 thus acts as a hydraulic piston 51'' that urges the exhaust valve 11 in the closing direction when the pressure chamber 53 is pressurized with hydraulic fluid.

A hydraulic piston 51' is placed on top of the valve stem 33. The hydraulic piston 51' is received in a bore in the upper housing part 101C that thereby forms the pressure chamber 52. As shown in Fig. 16W, an as such conventional self adjusting hydraulic end of stroke damper 58 is disposed in the top of the hydraulic piston 51'.

The flange 114 includes an upwardly extending sleeve 115. The upwardly extending sleeve 115 includes a conical part 56 with a tapered cross-sectional shape of a ferromagnetic material. As shown in Fig. 16X, the conical part 56 cooperates with the pick-up element 57 of the exhaust valve position sensor 55. The pick-up element 57 includes an eddy current sensor that measures the distance to a ferromagnetic body, in this case the conical part 56.

Fig. 16Y shows a security valve 125 that includes a spring biased piston 126. The security valve is disposed in a venting conduit 124 that connects the upper spring chamber 42 to the surrounding atmosphere. The piston 126 is urged by the hydraulic system pressure towards its closed position. The air pressure inside urges the piston 126 to its open position. When the hydraulic system pressure is within a range of normal values, the piston 126 overcomes the force from the air pressure and assumes its closed position. Should the hydraulic system pressure fall away, due to a breakdown or other cause, the air pressure biased piston 126 will assume the open position, and the air in the spring chamber 42 will be evacuated. Thus, the exhaust valve 11 will assume the closed position if the hydraulic system pressure falls away or drops below a predetermined threshold.

Fig. 17 shows another preferred embodiment of the exhaust valve assembly 18. This embodiment is substantially identical to the embodiment described with reference to Fig. 16 with the following changes. The flange 114 is not provided with an upwardly extending sleeve. The position sensor 55' is a magnetostrictive rod-sensor. A Teflon® ring with magnets inside (not shown) is be placed inside a recess in the top of the valve stem 33. A sensor rod 59 is attached to the top of the upper housing part 101C, and extends into a bore (not shown) in the valve stem 33. The Teflon® ring moves without contact along the sensor rod 59. The sensor signal corresponds to the position (and calculated speed) of the valve stem 33. Any "pumping effect" caused by the rod entering the oil filled bore in the moving valve stem 33 is avoided by overdimensioning the diameter of the bore relative to the diameter of the rod. The top valve stem 33 itself forms the hydraulic piston 51', and there is no hydraulic end of stroke damper to prevent the valve head 32 from abutting the valve seat 31 with excessive speed. The end of stroke dampening function is taken over by the control computer 27. Hereto, the control computer 27 increases the pressure in pressure chamber 52 when the control computer detects - based on the feedback signal from position sensor 55' - that the valve head 32 is about to abut with the valve seat 31 with an excessive velocity. The permissible landing speed for the valve head 32 on the valve seat 31 is in the range of 0,05 m/s to 0,4 m/s.

Fig. 18 shows another embodiment of the exhaust valve assembly 18. This embodiment is substantially identical to the embodiment described with reference to Fig. 16 with the following changes: The spring piston 41 is fitted over a slim or reduced diameter part of the valve stem 33, and abuts with a shoulder formed by a transition to a normal diameter part of the valve stem. The spring piston 41 includes an upwardly extending sleeve 48. The hydraulic piston 51 is also fitted over the slim valve stem part. The hydraulic piston 51 is double acting and the lower surface of the hydraulic piston 51 rests on the upper surface of the sleeve 48. The hydraulic piston 51 includes an upwardly extending sleeve 54. A nut 119 in threaded engagement with the upper part of the slim valve spindle part is placed on top of the sleeve 54 and tightens the parts that are fitted over the slim valve spindle part, and the conical part 56 is an integral part of the top of the valve stem 33.

Fig. 19 shows another embodiment of the exhaust valve assembly 18. This embodiment is substantially identical to the embodiment described with reference to Fig. 16 with the following changes: The hydraulic piston 51 is double acting. The hydraulic piston 51 is fitted over a slim or reduced diameter part of the valve stem 33 and abuts with a shoulder formed by a transition to a normal diameter part of the valve stem 33. The hydraulic piston 51 includes an upwardly extending sleeve 54. The spring piston 41 is also fitted over the slim valve stem part, and the lower surface of the spring piston 41 rests on the upper surface of the sleeve 54. A nut 119' in threaded engagement with the upper part of the slim valve spindle part is placed on top of the spring piston 41 and tightens the parts that are fitted over the slim valve spindle part, and the conical part 56 is an integral part nut 119'.

The above described embodiments of the exhaust valve assembly 18 allow full control the exhaust valve 11 of a two-stroke diesel engine 1 without a camshaft at any load point and at varying hydraulic feed pressure requiring only a minimum amount of energy. By measuring the exhaust valve position, the hydraulic feed pressure and the crank shaft angle position, the control computer 27 will be able to dose only the required amount of pressurized hydraulic fluid.

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. An exhaust valve assembly (18) for a large two-stroke diesel engine (1) of the crosshead type, said assembly (18) comprising:

- an exhaust valve (11) that is movable in opposite directions between a closed position and a variable open position;
- a double acting spring assembly (40) operably connected to the exhaust valve (11) and forming a mass-spring system together with the exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11);
- said double acting spring assembly (40) storing energy during translation of the exhaust valve (11) back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve (11) in an opposite direction;
- a position sensor (55) that provides a signal indicative of the position of the exhaust valve (11);
- a controller (27) in receipt of the signal from said position sensor (55); and
- hydraulic means (50) for holding the exhaust valve (11) on command from said controller (27).

2. A valve assembly (18) according to claim 1, characterized in that said hydraulic means (50) for holding the exhaust valve (11) comprises two cylinder-piston arrangements (50,51,51′,51′′) each including one pressure chamber (52,53) and acting in opposite directions.

3. A valve assembly (18) according to claim 1, characterized in that said hydraulic means (50) for

holding the exhaust valve (11) comprises a double acting cylinder piston (51) arrangement type including two pressure chambers (52,53).

4. A valve assembly (18) according to claim 2 or 3, wherein one or more hydraulic valves (60) control the flow of hydraulic fluid to and from the pressure chambers (52,53).

5. A valve assembly (18) according to claim 4, wherein said one or more valves (60) receive a control signal from said controller unit (27).

6. A valve assembly (18) according to claim 4, wherein said one or more valves (60) can in at least one of their positions close for fluid flow to and from either one or both of said pressure chambers (52,53) for holding the exhaust valve (11) in the closed or in the variable open position.

7. A valve assembly (18) according to claim 5 or 6, wherein said hydraulic valve or valves (60) can in at least one of their positions create a fluidic communication path between the two pressure chambers (52,53) for releasing the exhaust valve (11) and allowing the spring assembly (40) to propel the exhaust valve (11).

8. A valve assembly (18) according to any of claims 4 to 7, wherein said hydraulic valve or valves (60) can in at least one of their positions connect one of the pressure chambers (52) to a source (P) of high pressure hydraulic fluid for supplying additional energy to the exhaust valve (11) to compensate for energy dissipation in the assembly (18).

9. A valve assembly (18) according to claim 8, wherein said hydraulic valve or valves (60) can in at least one of their positions connect the other pressure chamber (53) to a source (P) of high pressure hydraulic fluid for supplying additional energy to the exhaust valve to compensate for energy dissipation in the assembly.

10. A valve assembly (18) according to any of claims 4 to 9, wherein said hydraulic valve or valves (60) can in at least one of their positions throttle the hydraulic fluid going into or coming out of one or both of said pressure chambers (52,53) for slowing down said exhaust valve (11).

11. A valve assembly (18) according to any of claims 1 to 10, further including an end of stroke dampener (58,59) for slowing down the exhaust valve (11) in the final part of the closing- and/or opening stroke.

12. A valve assembly (18) according to claim 11, wherein the one or more end of stroke dampeners (58,59) are an integral part of said pressure chambers (52,53).

13. A valve assembly according to any of claims 1 to 12, wherein the double acting spring arrangement (40) is an air spring arrangement including at least two spring chambers (42,43), the air in one spring chamber (43) being compressed when the exhaust valve moves (11) in the opening direction and the air in the other spring chamber (42) being compressed when the exhaust valve (11) moves in the closing direction.

14. A valve assembly (18) according to claim 13, wherein said double acting spring arrangement (40) is formed by a double acting spring piston (41) operably connected to the exhaust valve (11), said spring piston (41) dividing

a cylinder (40) in the valve assembly (18) into two opposite spring chambers (42,43).

15. A valve assembly (18) according to claim 13, wherein said hydraulic piston cylinder arrangements (51',52.51'',53)) are, seen in the direction of translation, disposed the on opposite sides of said air chamber

16. An exhaust valve assembly (18) for a large two-stroke diesel engine (1) of the crosshead type, said assembly (18) comprising:

- an exhaust valve (11) that is movable between a closed position and a variable open position;
- a double acting spring assembly (40) operably connected to the exhaust valve (11) and forming a mass-spring system together with the exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11);
- said double acting spring assembly (40) storing energy during translation of the exhaust valve (11) back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve (11) in an opposite direction;
- a position sensor (55) that provides a signal indicative of the position of the exhaust valve (11);
- a controller (27) in receipt of the signal from said position sensor (55);
- means (50,60) for holding the exhaust valve (11) in the closed position and in the variable open position;
- means (50,60) for releasing the exhaust valve (11) from being held in the closed position to allow the double acting spring assembly (40) to propel the

exhaust valve (11) towards the variable open position;

- means for releasing the exhaust valve (11) from being held in the variable open position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the closed position; and
- hydraulic means (50,60) for supplying additional energy to the exhaust valve (11) on command from said controller (27) to compensate for energy dissipation in the assembly (18).

17. A valve assembly (18) according to claim 16, wherein said controller (27) is configured to use the signal from the position sensor (55) and optionally other parameters to determine the required amount of additional energy that is to be fed into the system, and configured to command said hydraulic means (50,60) for supplying additional energy to deliver the determined amount of hydraulic energy.

18. A valve assembly (18) according to claim 16 or 17, wherein said controller (27) is configured to command said hydraulic means (50,60) for supplying additional energy to provide an initial amount of energy that may be predetermined or determined by the controller on the basis of operating parameters of the engine (1).

19. A valve assembly (18) according to claim 18, wherein said controller (27) is configured to periodically or continuously monitor the signal from said position sensor (55) during at least a part of the stroke of the exhaust valve (11).

20. A valve assembly (18) according to claim 19, wherein said controller (27) is configured to compare the actual

position and/or velocity with an expected position and/or velocity.

21. A valve assembly (18) according to claim 20, wherein said controller (27) is configured to correct a deviation between the actual and expected velocity of the exhaust valve (11) by either commanding said hydraulic means (50) for supplying additional energy to deliver an amount of additional energy, or by commanding said hydraulic means for holding (50) to slow down the exhaust valve (11).

22. A valve assembly (18) according to any of claims 16 to 20, wherein said controller (27) is configured to use the signal from the position sensor (55) and optionally other parameters to determine that the exhaust valve (11) may overshoot or approach the valve seat (32) too fast and is configured to command said hydraulic means (50) for holding to slow down the exhaust valve (11) before the exhaust valve (11) overshoots or hits the valve seat (32) with an excessive velocity.

23. A valve assembly (18) according to any of claims 16 to 22, wherein said means for (50) for holding the exhaust valve (11) in the closed position and in the variable open position; said means (50) for releasing the exhaust valve (11) from being held in the closed position; said hydraulic means (50) for releasing the exhaust valve (11) from being held in the variable open position; and said hydraulic means (50) for supplying additional energy to the exhaust valve (11) are formed by a pair of oppositely acting hydraulic cylinder piston assemblies (50,51,51',51'',52).

24. An exhaust valve assembly (18) for a large two-stroke diesel engine (1) of the crosshead type and having a predetermined engine speed R_m in rounds per minute at its

nominal maximum continuous rating, said assembly (18) comprising:

- an exhaust valve (11) that is movable between a closed position and a variable open position;
- a double acting spring assembly (40) operably connected to the exhaust valve (11) and forming a mass-spring system together with said exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11);
- said double acting spring assembly (40) storing energy during translation of the gas exchange valve (11) back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve (11) in an opposite direction;
- means (50) for holding the exhaust valve (11) in the closed position and in the variable open position;
- means (50) for releasing the exhaust valve (11) from being held in the closed position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the variable open position;
- means (50) for releasing the exhaust valve (11) from being held in the variable open position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the closed position;
- hydraulic means (50) for supplying additional energy to the exhaust valve to compensate for energy dissipation in the assembly (11); and
- said mass-spring system being characterized by an Eigenfrequency in a range between 1 to 64 times R_m.

25. A valve assembly (18) according to claim 24, wherein the double acting spring assembly (40) comprises air cylinders with pistons (41).

26. A valve assembly (18) according to claim 25, including a double acting air piston (41).

27. A valve assembly (18) according to claim 24, wherein the double acting spring assembly comprises gas accumulators.

28. An exhaust valve assembly (18) for a large two-stroke diesel engine (1) of the crosshead type and having an engine speed R in rounds per minute, said assembly comprising:

- an exhaust valve (11) that is movable between a closed position and a variable open position;
- a double acting spring assembly (40) operably connected to the exhaust valve (11) and forming a mass-spring system together with said exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11);
- said double acting spring assembly storing (40) energy by during translation of the gas exchange valve back (11) and forth between the closed and variable open position for subsequent propulsion of the exhaust valve (11) in an opposite direction;
- means (50) for holding the exhaust valve (11) in the closed position and in the variable open position,
- means (50) for releasing the exhaust valve (11) from being held in the closed position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the variable open position;
- means (50) for releasing the exhaust valve (11) from being held in the variable open position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the closed position;

- hydraulic means (50) for supplying additional energy to the exhaust valve (11) to compensate for energy dissipation in the assembly;
- said double acting spring assembly (40) being of the type in which a gaseous medium is compressed from a base pressure to a higher pressure to store the energy, and the gaseous medium expands from the higher pressure to the base pressure in order to propel the exhaust valve (11);
- a controller (27) that is configured to determine a desired value for the Eigenfrequency of the mass-spring system and configured to control/adjust the base pressure of the gaseous medium in said double acting spring assembly accordingly.

29. A valve assembly (18) according to claim 28, wherein the controller (27) determines the desired Eigenfrequency as a function of the actual engine speed.

30. A valve assembly (18) according to claims 28 or 29, wherein the controller (27) determines the desired Eigenfrequency to be in a range of two to ten times the actual engine speed.

31. A valve assembly (18) according to any of claims 28 to 30, wherein the Eigenfrequency is adjusted by adapting the pressure in the air cylinder or cylinders (40).

32. A valve assembly (18) according to claim 31, wherein each of said spring chambers (42,43) is connected to a pressure line (92,93) with a pressure controlled by the controller unit (27).

33. A valve assembly (18) according to claim 32, wherein the pressure in each pressure line (92,93) can be controlled individually for each spring chamber (42,43).

34. A valve assembly (18) according to claim 33, wherein a non-return valve (94,96) is disposed in each pressure line (92,93) for preventing return flow from the respective spring chambers (42,43), a pressure controlled relief valve (95,97) is disposed in each pressure line (92,93) between the non-return valve (94,96) and the spring chamber (42,43), whereby the control pressure for the relief valve (95,97) is the pressure upstream of the respective non-return valve (94,96).

35. A valve assembly (18) according to claim 34, wherein the opening pressure of the relief valve (95,97) is a factor K times the control pressure, whereby K is substantially equal to the compression ratio of the respective spring chamber (42,43) in the closed or variable open position respectively

36. A valve assembly (18) according to any of claims 32 to 35, further including a pressure sensor (99,99') upstream of the non-return valves (94,96) for providing a feedback signal to the controller (27).

37. An exhaust valve assembly (18) for a large two-stroke diesel engine (1) of the crosshead type, said assembly (18) comprising:

- a valve housing (101A,101B,101C);
- an exhaust valve (11) that is movable between a closed position and a variable open position;
- a double acting air spring assembly (40) operably connected to the exhaust valve (11) and forming a mass-spring system together with said exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11);

- said double acting air spring assembly (40) storing energy during translation of the gas exchange valve (11) back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve (11) in an opposite direction;
- two hydraulic piston cylinder arrangements (50,51,51',51'',52,53) acting in opposite directions for:
  - o holding the exhaust valve (11) in the closed position and in the variable open position, and for releasing the exhaust valve (11) from being held in the closed position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the variable open position;
  - o for releasing the exhaust valve (11) from being held in the variable open position to allow the double acting spring assembly (40) to propel the exhaust valve (11) towards the closed position; and
  - o for supplying additional energy to the exhaust valve (11) to compensate for energy dissipation in the assembly (18).

38. A valve assembly (18) according to claim 37, wherein the double acting air spring assembly (40) comprises a piston (41) that is fastened to the valve stem (33) by two oppositely disposed wedges.

39. A valve assembly (18) according to claim 38, wherein said wedges are formed by wedge rings (110,116) or wedge bushings.

40. A valve assembly (18) according to claim 39, wherein said assembly (18) comprises a flange (114) that is

disposed over the valve stem (33) facing the spring piston (41).

41. A valve assembly (18) according to claim 40, wherein one of said wedge rings (110,116) is received in a conical collet bore that is disposed in the spring piston (41), the other wedge ring (110,116) being received in a conical collet bore disposed in said flange (114), said flange (114) being tightened to the spring piston (41) and thereby the oppositely disposed wedge rings (110,116)are driven into their respective collet bores and tightened radially against the periphery of the valve stem (33) to thereby fasten the spring piston (41) and the flange (114) to the valve stem (33).

42. A valve assembly (18) according to claim 41, wherein the flange (114) is tightened to the spring piston (41) by tension bolts (117) assembled through the flange (114).

43. A valve assembly (18) according to any of claims 37 to 42, wherein at least the hydraulic piston (51) of the hydraulic piston cylinder arrangement (50,51,51'',53) that urges the exhaust valve (11) in the closing direction includes a ring piston (51'') that is disposed around the stem (33) of the exhaust valve (11).

44. A valve assembly (18) according to claim 43, wherein said ring piston (51'') is formed by a sleeve (47) connected to the spring piston (41).

45. A valve assembly (18) according to claim 44, wherein said sleeve (47) is directed towards the valve head (32) and is integral with the spring piston (41).

46. A valve assembly (18) according to any of claims 37 to 45, wherein a hydraulic valve block for controlling the flow of hydraulic fluid to and from the hydraulic piston-cylinder assembly (50) is attached directly to the valve housing (101A,101B,101C).

47. A valve assembly (18) according to any of claims 37 to 46, further including a position sensor (55) that comprises a tapered part (56) of a Ferro-magnetic material.

48. A valve assembly (18) according to claim 47, wherein said tapered part (56) is an integral part of a bushing (115) extending from the flange (114).

49. A valve assembly (18) according to any of claims 37 to 48, wherein the hydraulic piston (52) that urges the exhaust valve (11) in the opening direction is formed by the top of the valve stem (33).

50. A valve assembly (18) according to any of claims 37 to 49, wherein the top of the valve stem (33) is provided with a longitudinal bore that is used for sensing the position of the exhaust valve (11).

51. A valve assembly (18) according to claim 37, wherein the air spring assembly (40) comprises a double acting spring piston (41) and the hydraulic piston cylinder arrangements (50) include one common double acting hydraulic piston (51), whereby the spring piston (41) and the double acting hydraulic piston (51) are fitted over a slim upper part of the valve stem (33) and are sandwiched between the lower fat part of the valve stem (33) and a nut (119,119') in threaded engagement with the upper part of the valve stem (33).

52. A valve assembly (18) according to claim 51, wherein a bushing (54) is fitted over said slim upper part of the valve stem (33) and disposed in between said double acting hydraulic piston (51) and said double acting spring piston (41).

53. A valve assembly (18) according to claim 51 or 51, wherein at least a part of the nut (119') is tapered for use in a position sensor.

54. A valve assembly (18) according to any of claims 37 to 53, wherein at least a part of the valve stem (33) is tapered for use in a position sensor (55).

55. An exhaust valve assembly (18) for a large two-stroke diesel engine (1) of the crosshead type, said assembly (18) comprising:

- an exhaust valve (11) that is movable in opposite directions between a closed position and a variable open position;
- a double acting air spring assembly (40), operably connected to the exhaust valve (11) and forming a mass-spring system together with the exhaust valve (11) and the mass of any other parts moving in unison with the exhaust valve (11);
- said double acting spring assembly (40)) including two spring chambers (42,43) for storing energy during translation of the exhaust valve (11) back and forth between the closed and variable open position for subsequent propulsion of the exhaust valve (11) in an opposite direction;
- hydraulic means (50) for holding the exhaust valve on command from said controller (27);
- means for evacuating the spring chamber (42) that urges the exhaust valve (11) in the opening

direction when the hydraulic pressure falls away so that the exhaust valve (11) will automatically assume the closed position when the hydraulic pressure falls away or drops below a predetermined threshold.

56. An exhaust valve assembly (18) according to claim 55, wherein said means for evacuating comprises a relief valve (125) that is urged towards an open position by resilient means of by the air pressure in the spring chamber (42) and a urged towards a closed position by the hydraulic pressure, so that the resilient means or the air pressure will urge the relief valve (125) to the open position when the hydraulic pressure falls away or drops below a predetermined threshold.

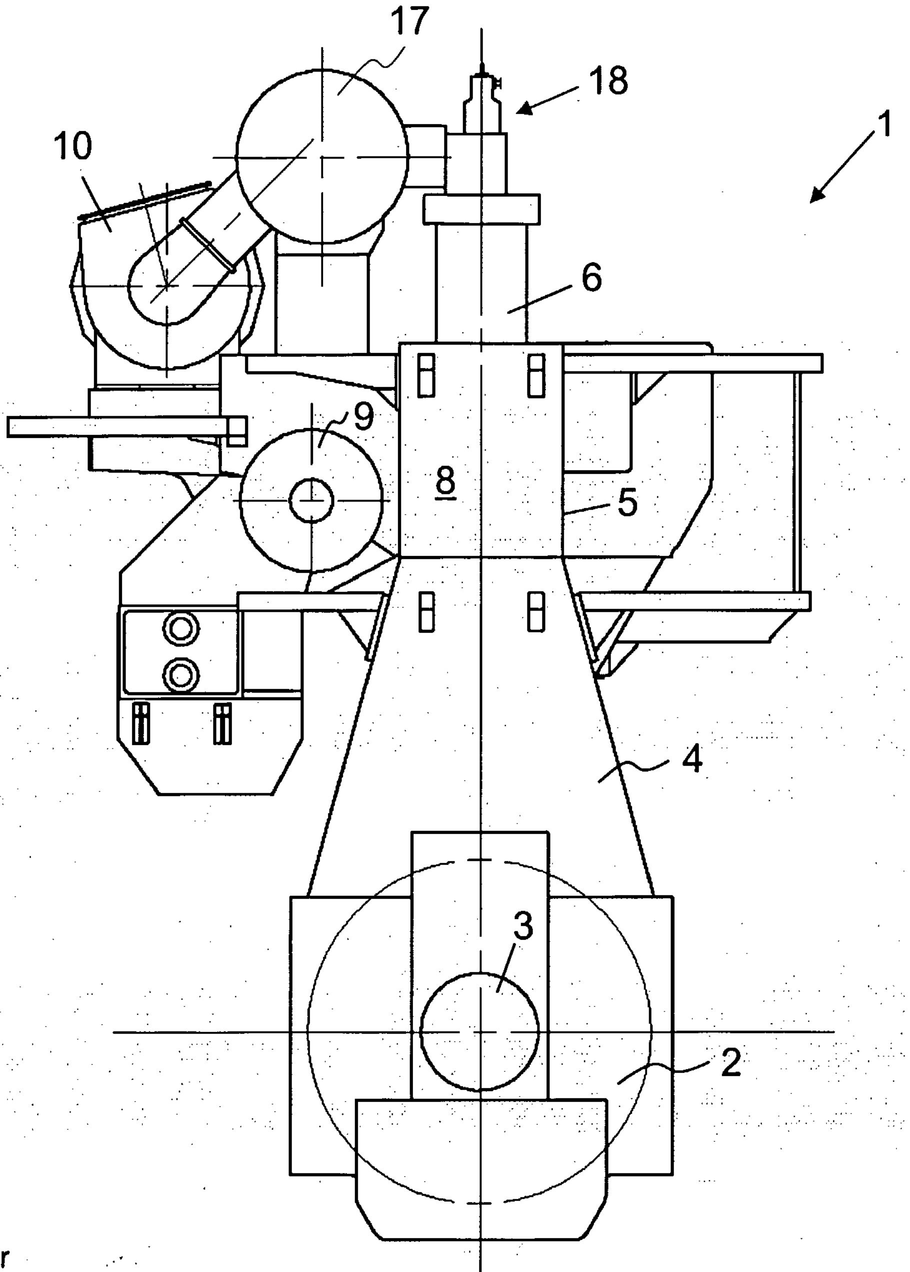


Fig. 1

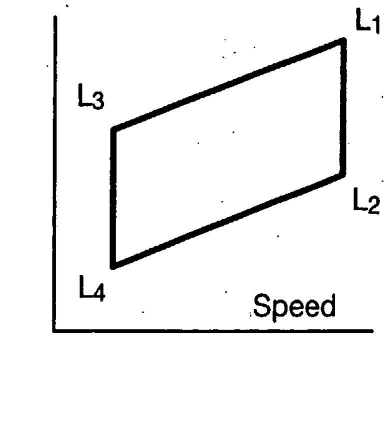


Fig. 1A

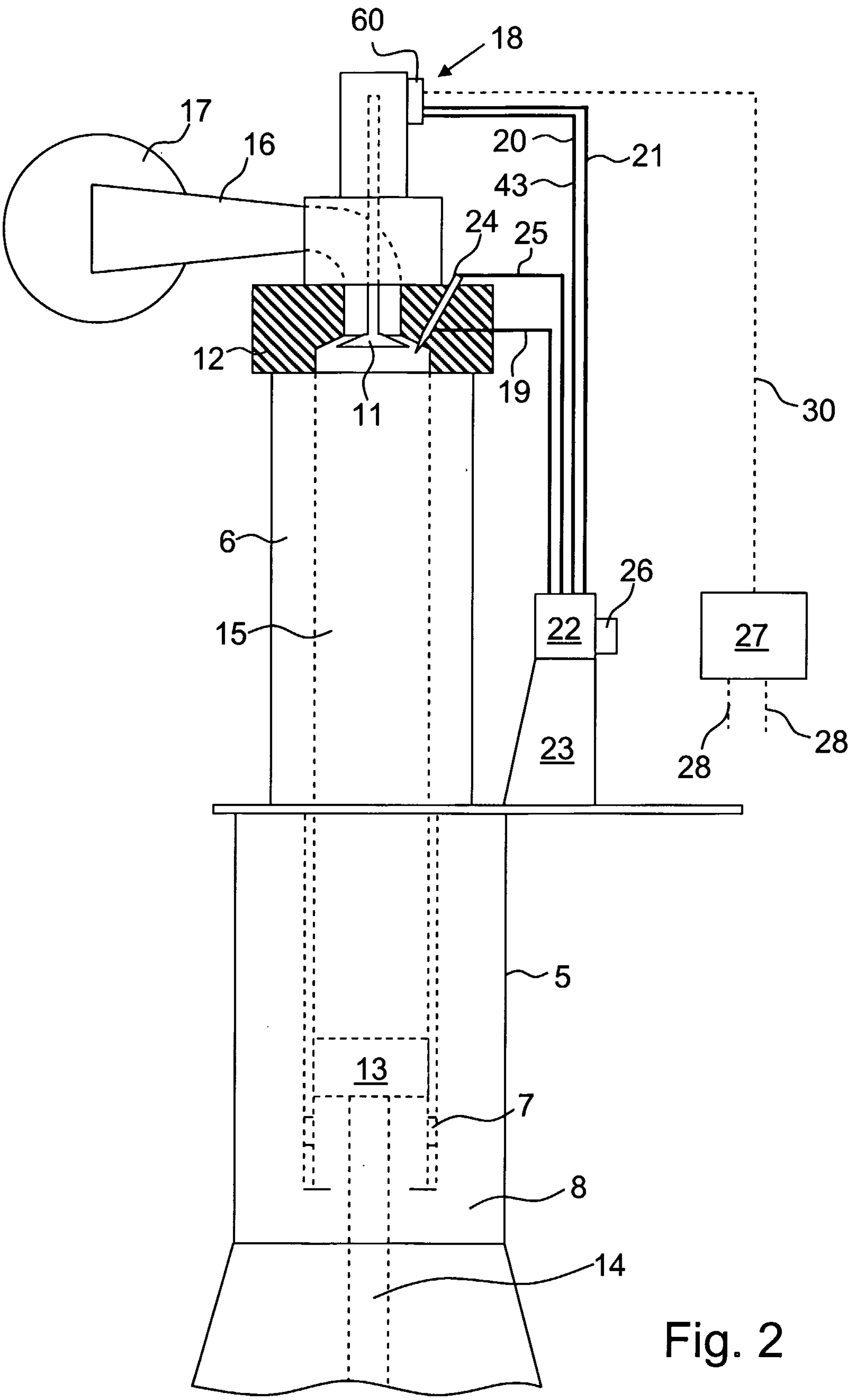
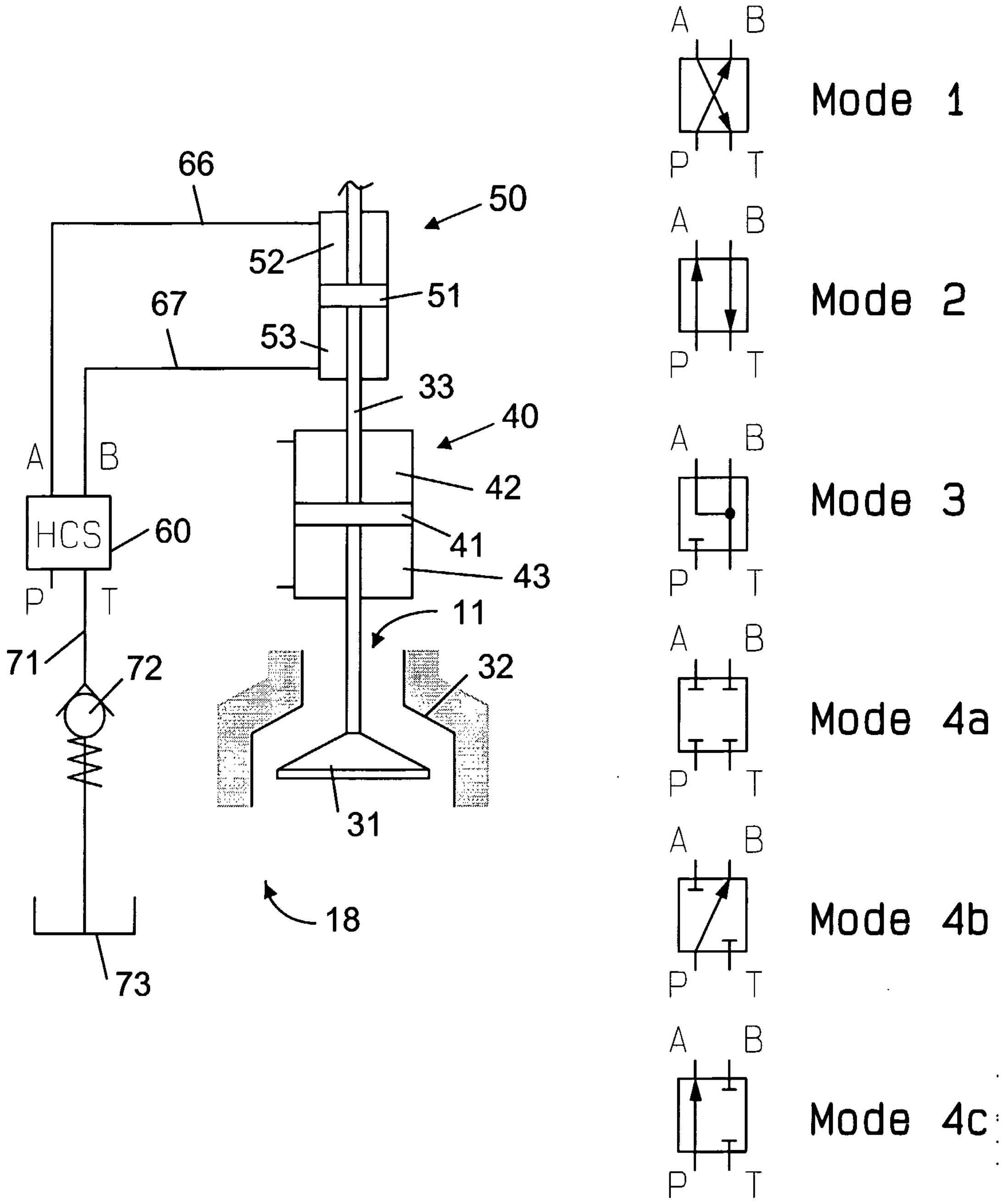


Fig. 2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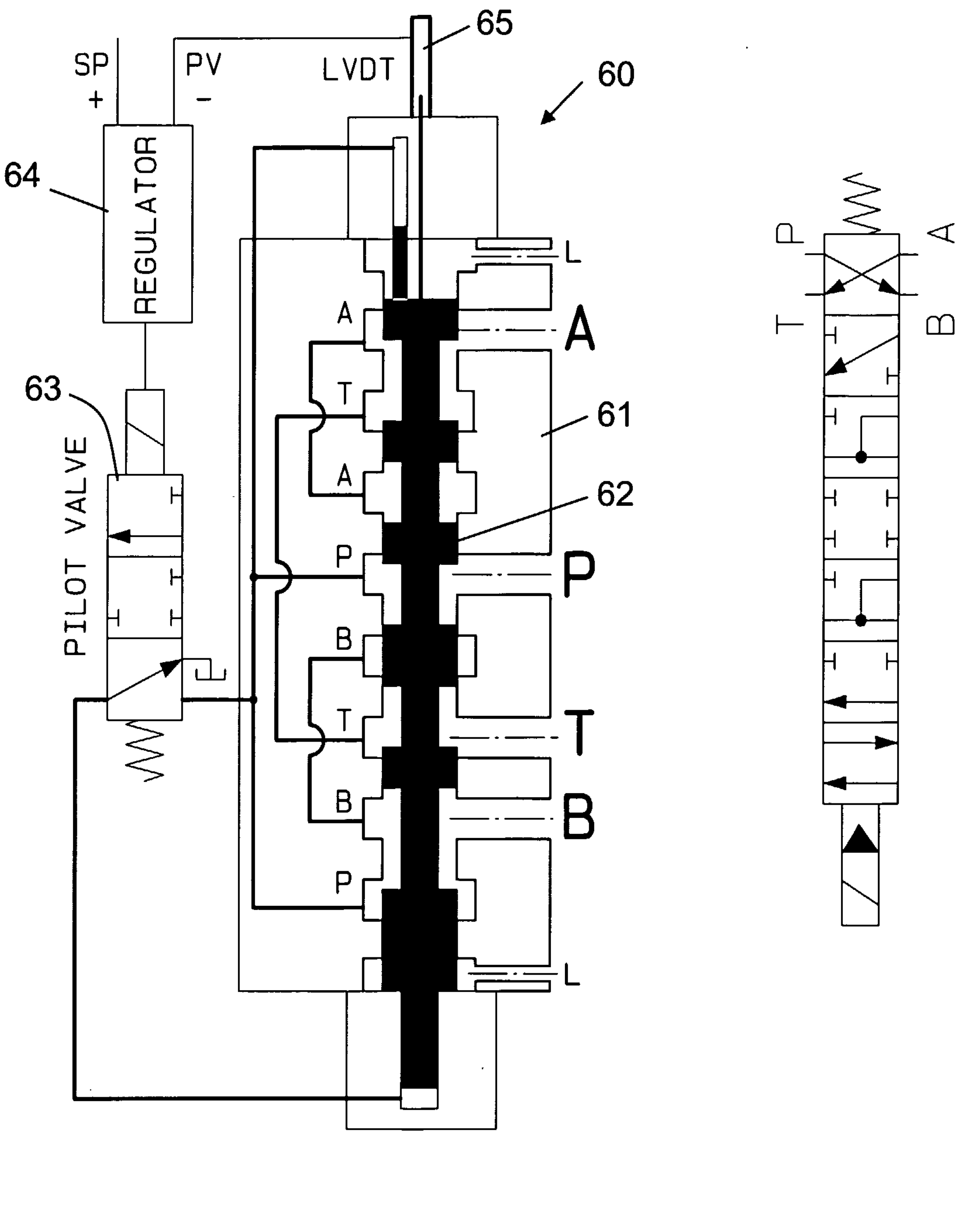
50
52
51
67
53
33
40
42
41
43
A
B
HCS
60
P
T
11
71
72
32
31
18
73
A B
P T
Mode 1
A B
P T
Mode 2
A B
P T
Mode 3
A B
P T
Mode 4a
A B
P T
Mode 4b
A B
P T
Mode 4c

Fig. 3



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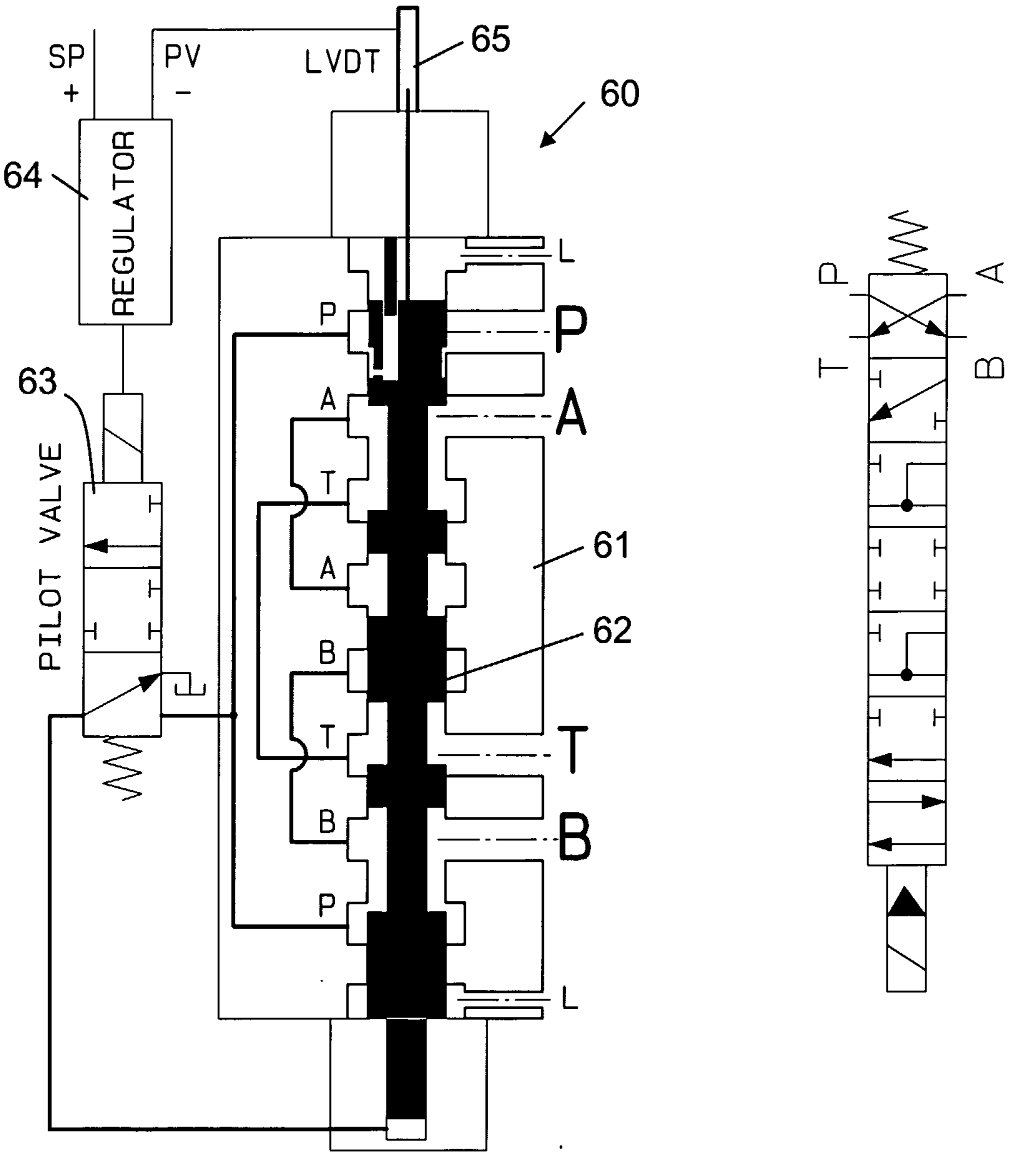
60
SP
+
PV
−
LVDT
64
REGULATOR
63
PILOT VALVE
L
A
T
A
61
62
P
P
B
T
T
B
B
P
L
P
A
T
B

Fig. 4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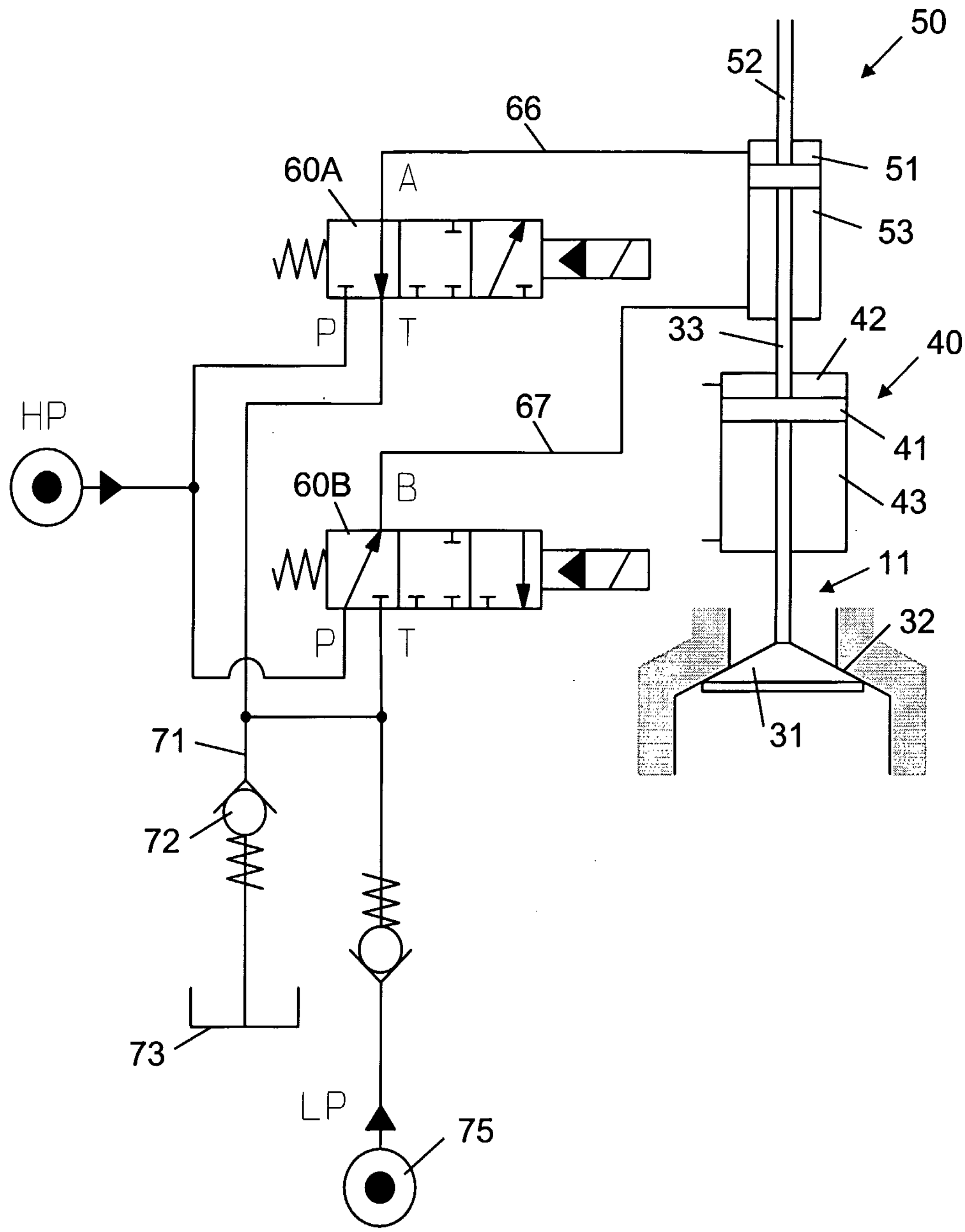
+
PV
–
LVDT
65
60
64
REGULATOR
63
PILOT VALVE
L
P
P
A
A
T
A
61
62
B
B
T
T
B
B
P
L
P
A
T
B

Fig. 5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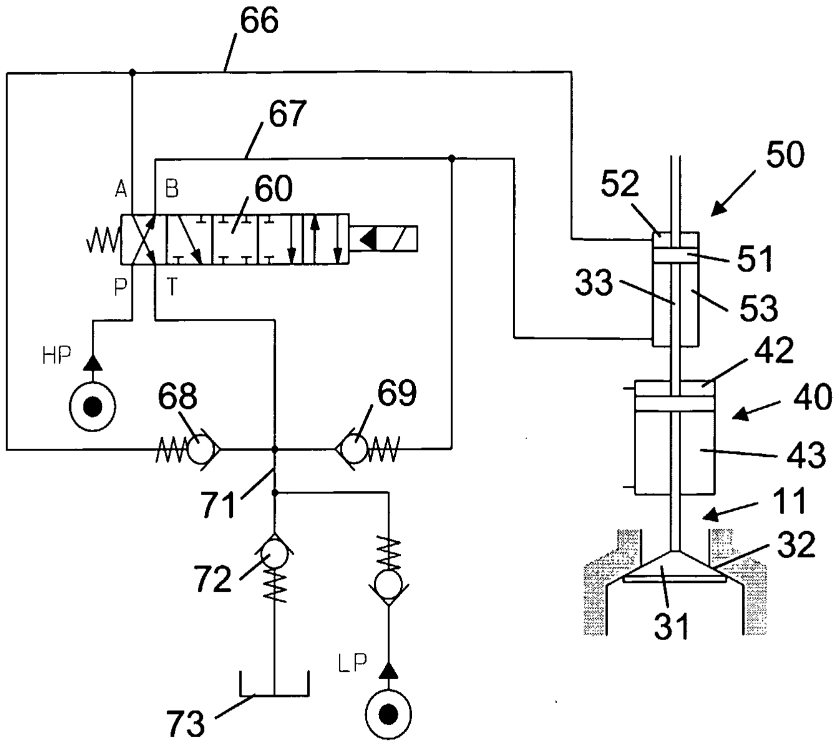
52
66
51
60A
A
53
P
T
33
42
40
HP
67
41
60B
B
43
11
P
T
32
71
31
72
73
LP
75

Fig. 6


66
67
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A
B
60
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P
T
HP
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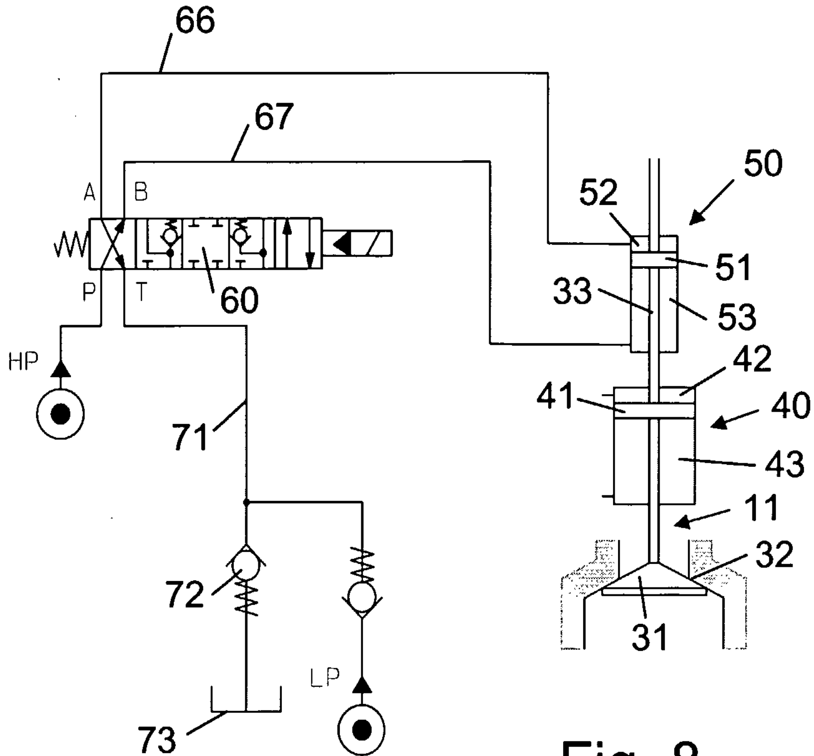
11
32
72
31
LP
73

Fig. 7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67
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A
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P
T
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41
40
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11
32
72
31
LP
73

Fig. 8

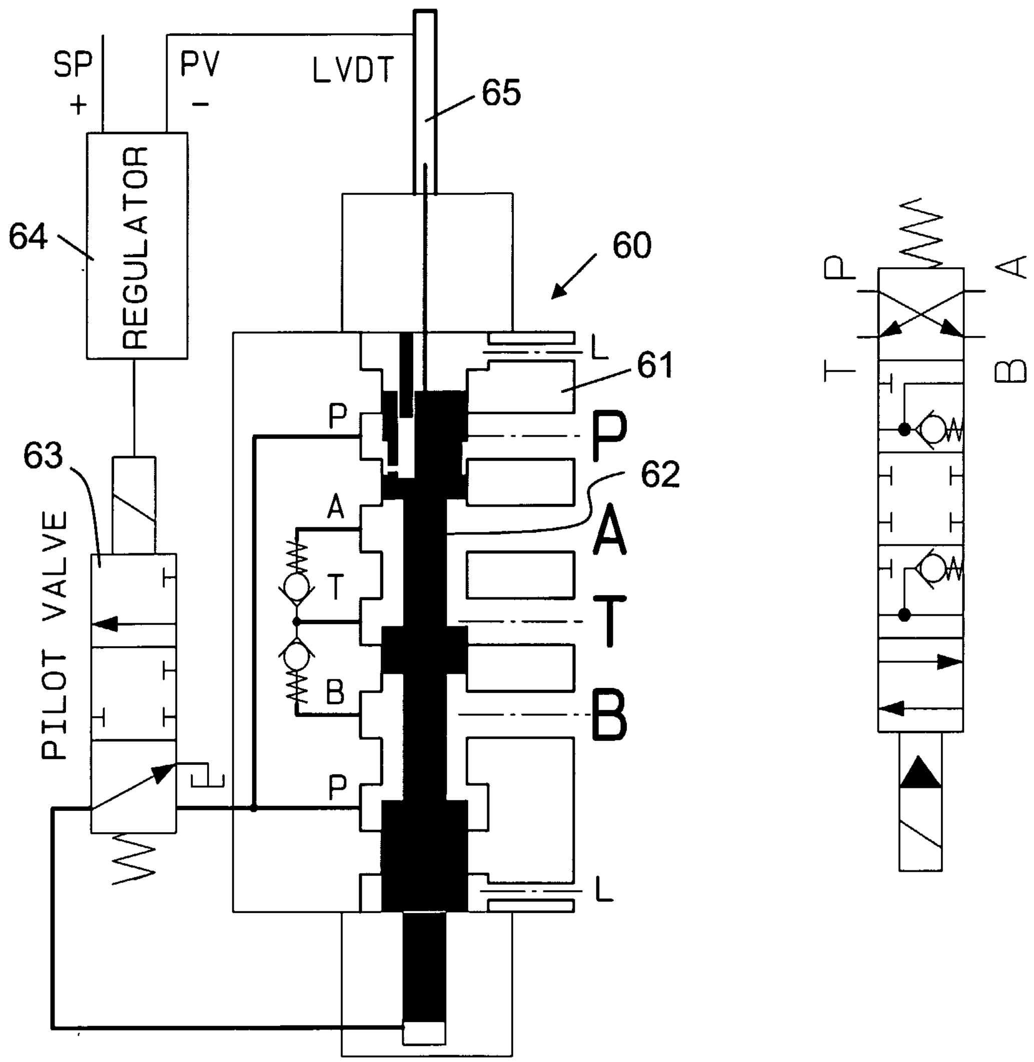

SP
+
PV
−
LVDT
65
64
REGULATOR
60
61
L
P
62
A
T
B
P
L
63
PILOT VALVE
P
A
T
B
P
A
T
B

Fig. 9

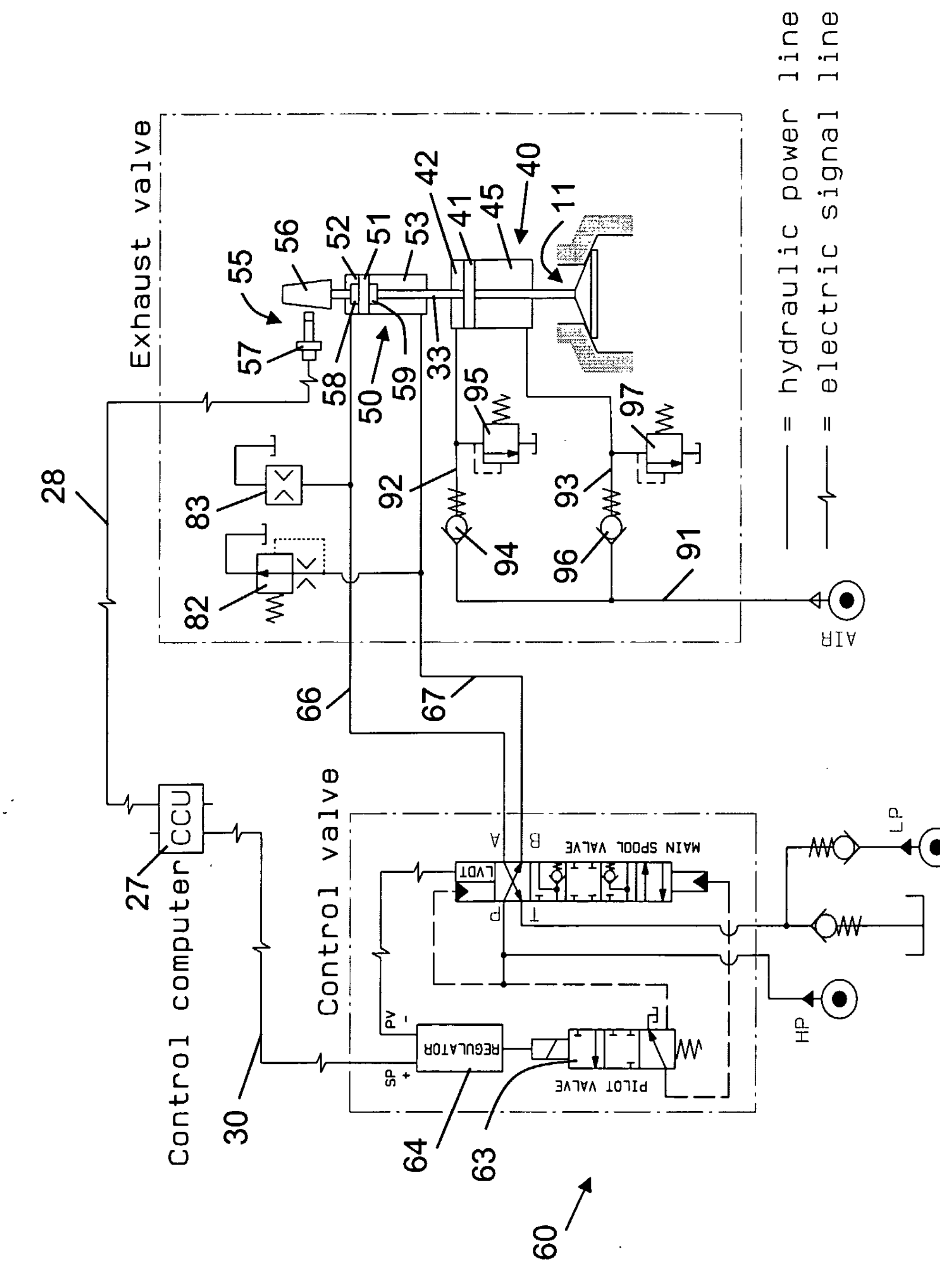

Exhaust valve
Control valve
Control computer
CCU
MAIN SPOOL VALVE
PILOT VALVE
REGULATOR
LVDT
SP
PV
A
B
P
T
AIR
HP
LP
= hydraulic power line
= electric signal line
11
27
28
30
33
40
41
42
45
50
51
52
53
55
56
57
58
59
60
63
64
66
67
82
83
91
92
93
94
95
96
97

Fig. 10

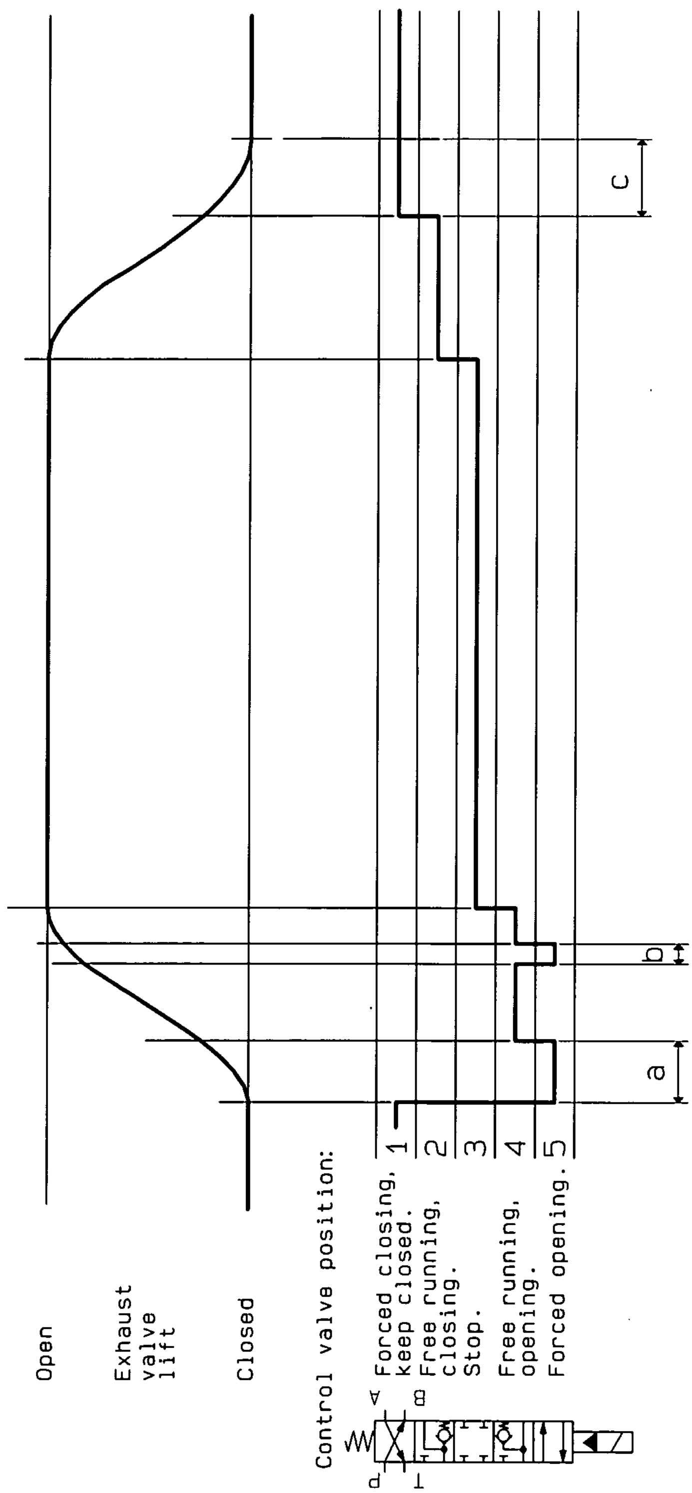

Open
Exhaust valve lift
Closed
Control valve position:
1 Forced closing, keep closed.
2 Free running, closing.
3 Stop.
4 Free running, opening.
5 Forced opening.
a
b
c
A
B
P
T

FIG. 11

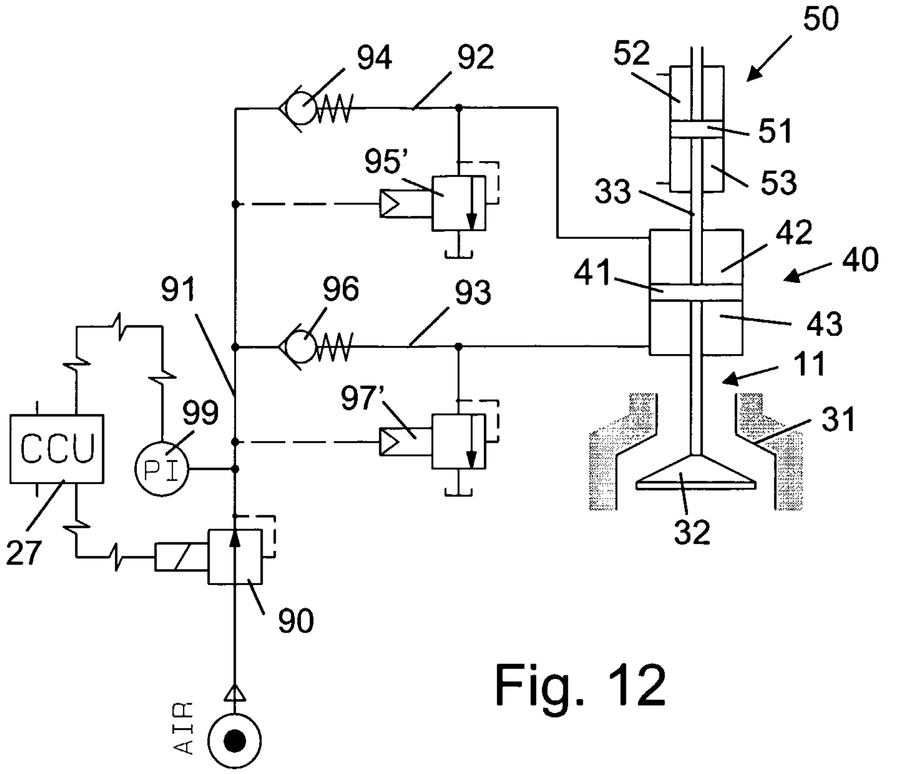
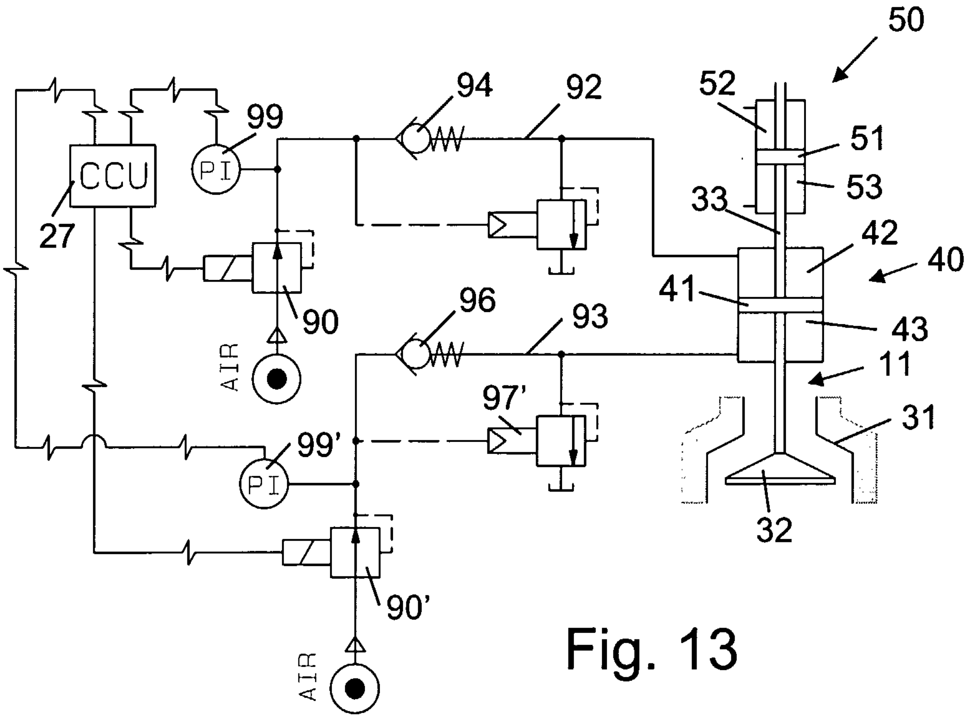

50
52
94
92
51
95'
53
33
42
40
41
91
96
93
43
CCU
11
99
97'
PI
31
27
32
90
AIR

Fig. 12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99
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CCU
PI
53
33
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43
AIR
11
97'
31
99'
PI
32
90'
AIR

Fig. 13

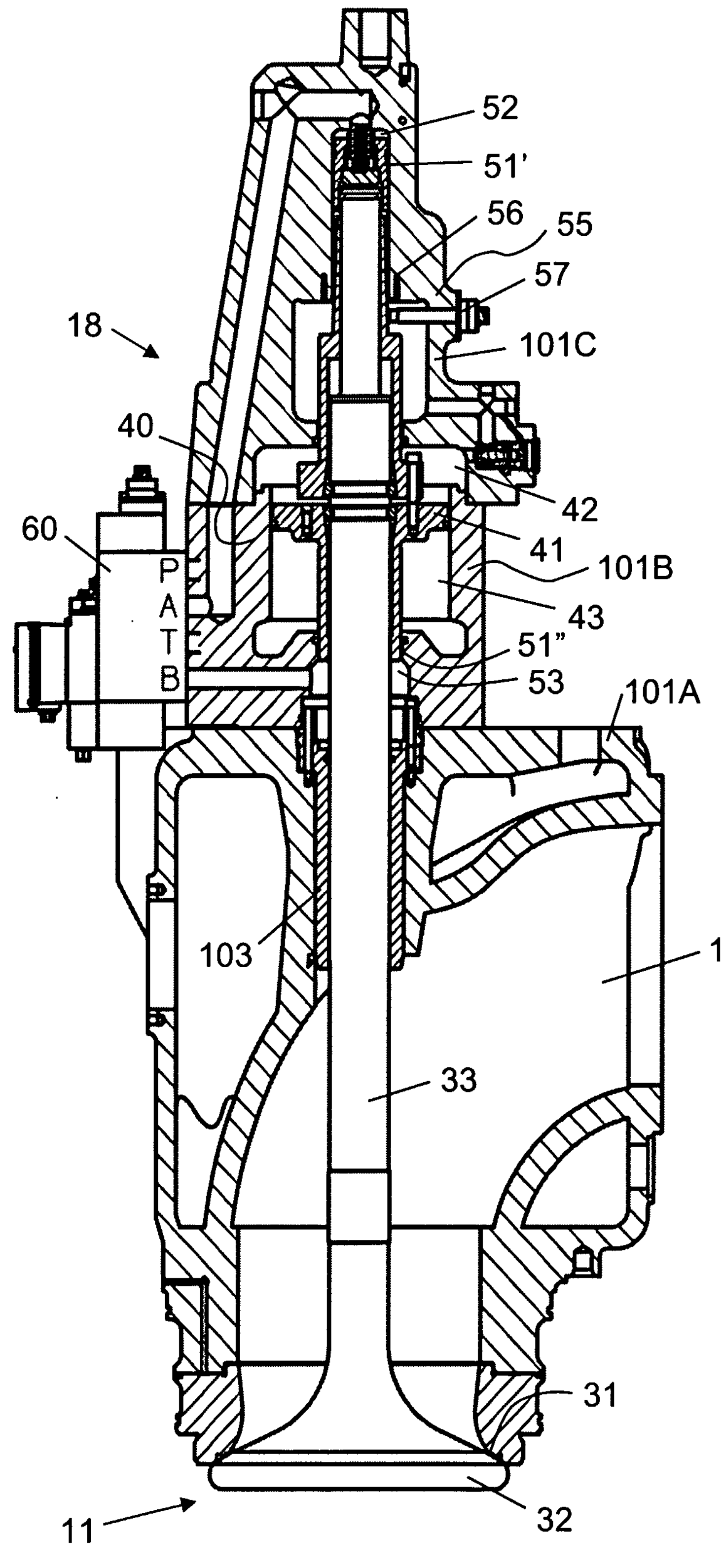

52
51'
56
55
57
18
101C
40
42
60
41
101B
P
A
T
B
43
51"
53
101A
103
102
33
31
11
32

FIG. 14

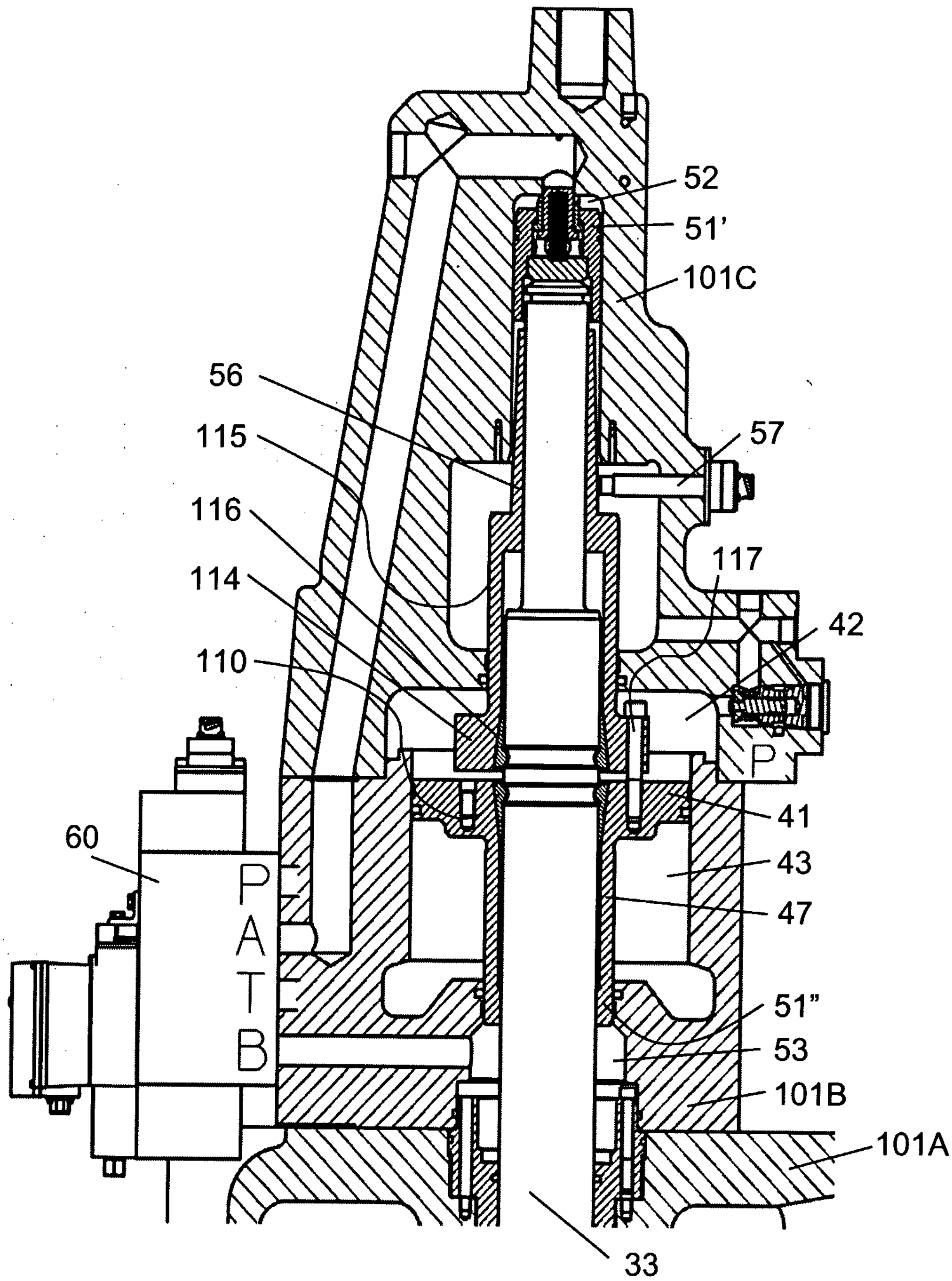

52
51'
101C
56
115
57
116
117
114
42
110
P
41
60
43
P
A
47
T
B
51"
53
101B
101A
33

FIG. 15

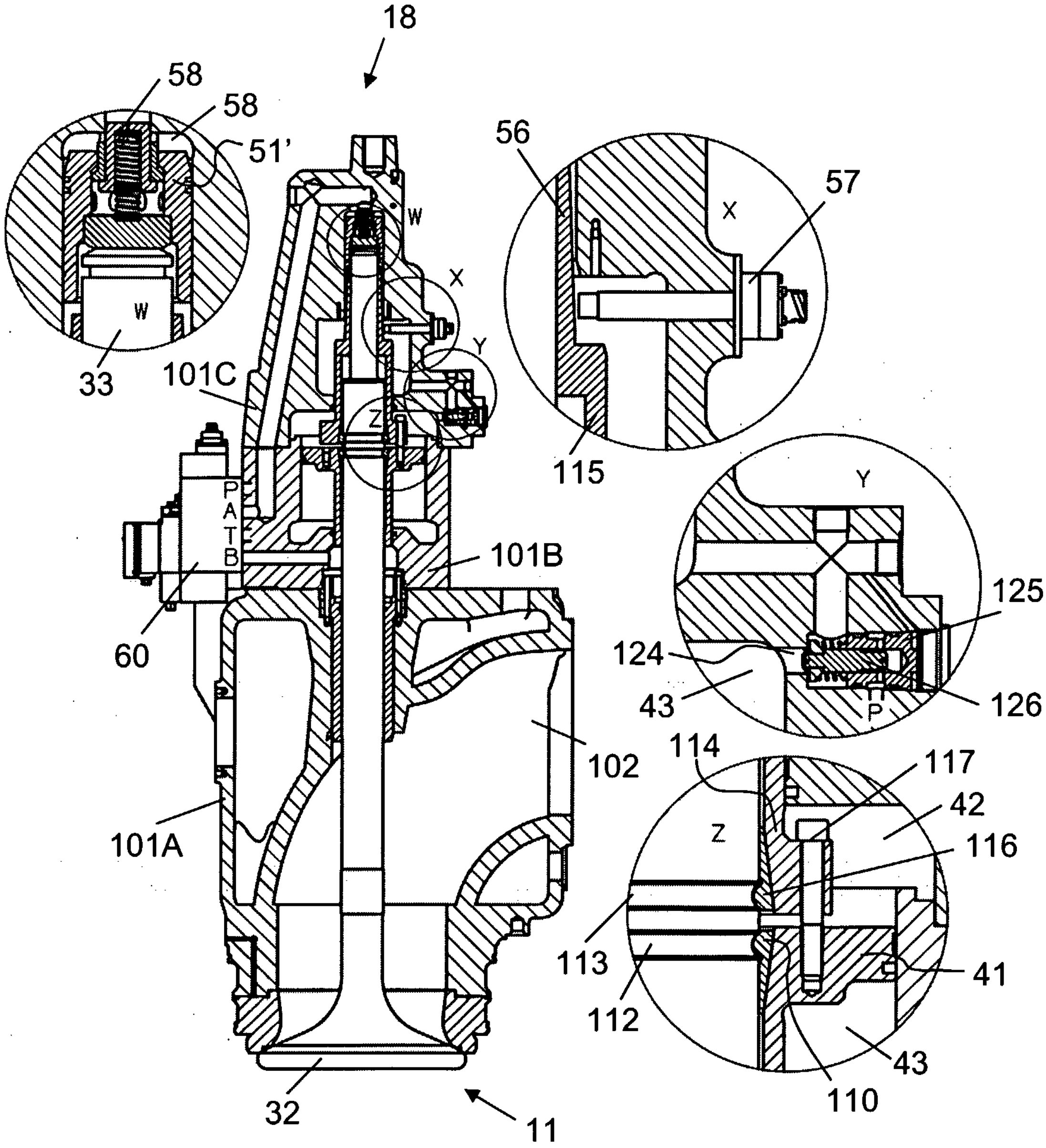

18
58
58
51'
W
W
33
101C
X
Y
Z
P
A
T
B
60
101B
101A
102
32
11
56
57
X
115
Y
125
124
43
P
126
114
117
42
Z
116
113
112
41
43
110

FIG. 16

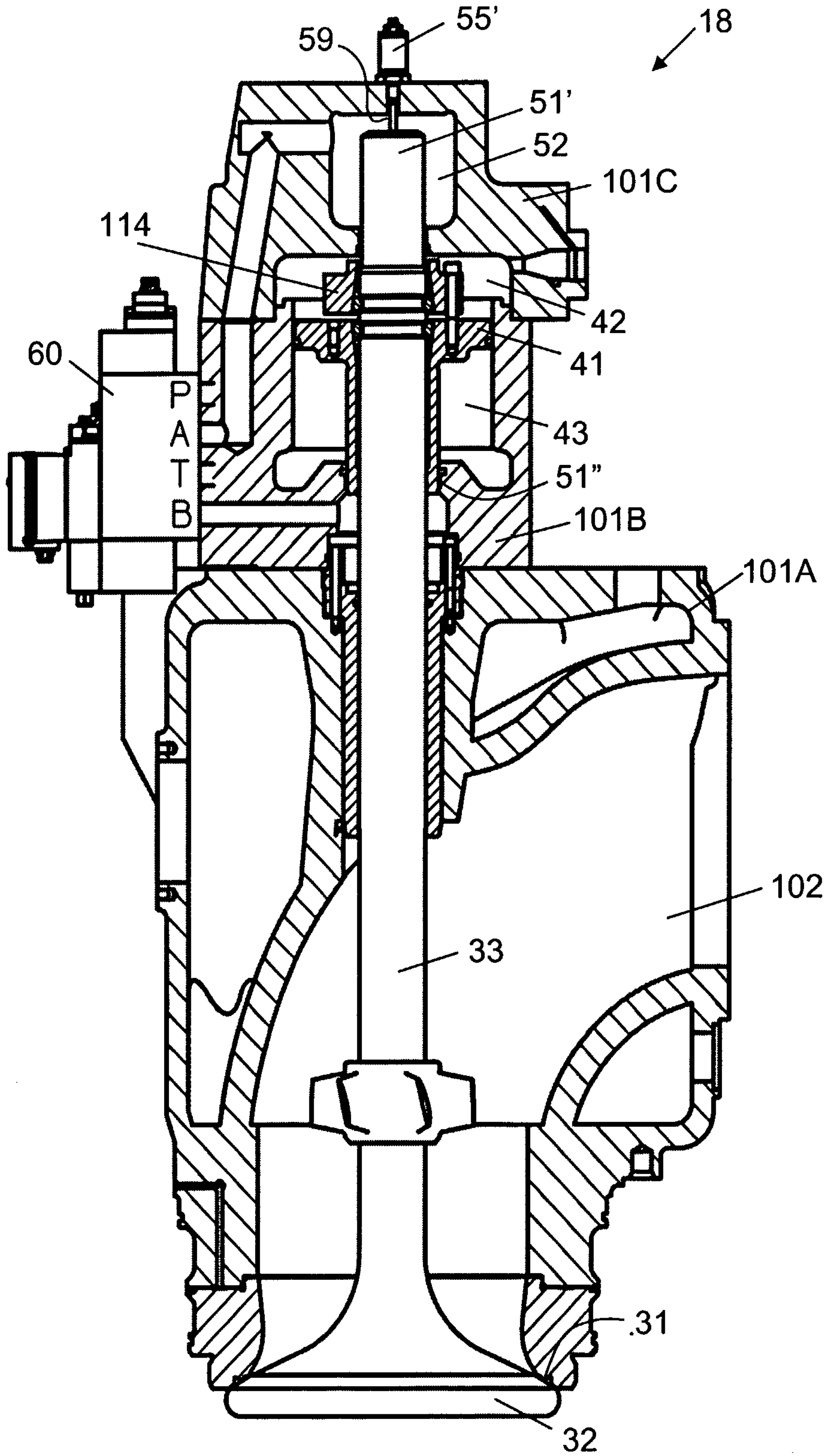
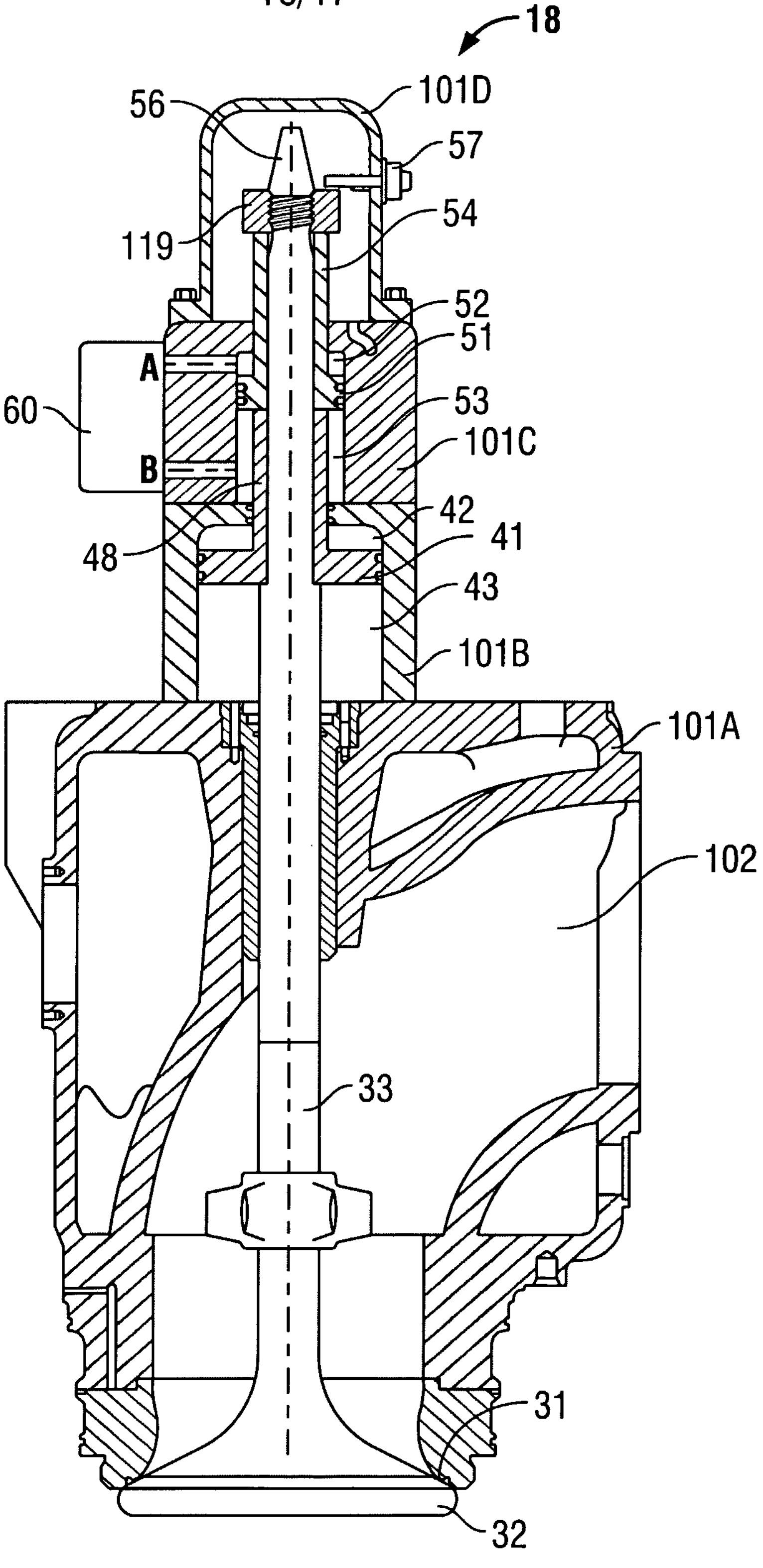


FIG. 17


18
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101D
57
119
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52
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A
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101C
B
42
48
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43
101B
101A
102
33
31
32

FIG. 18

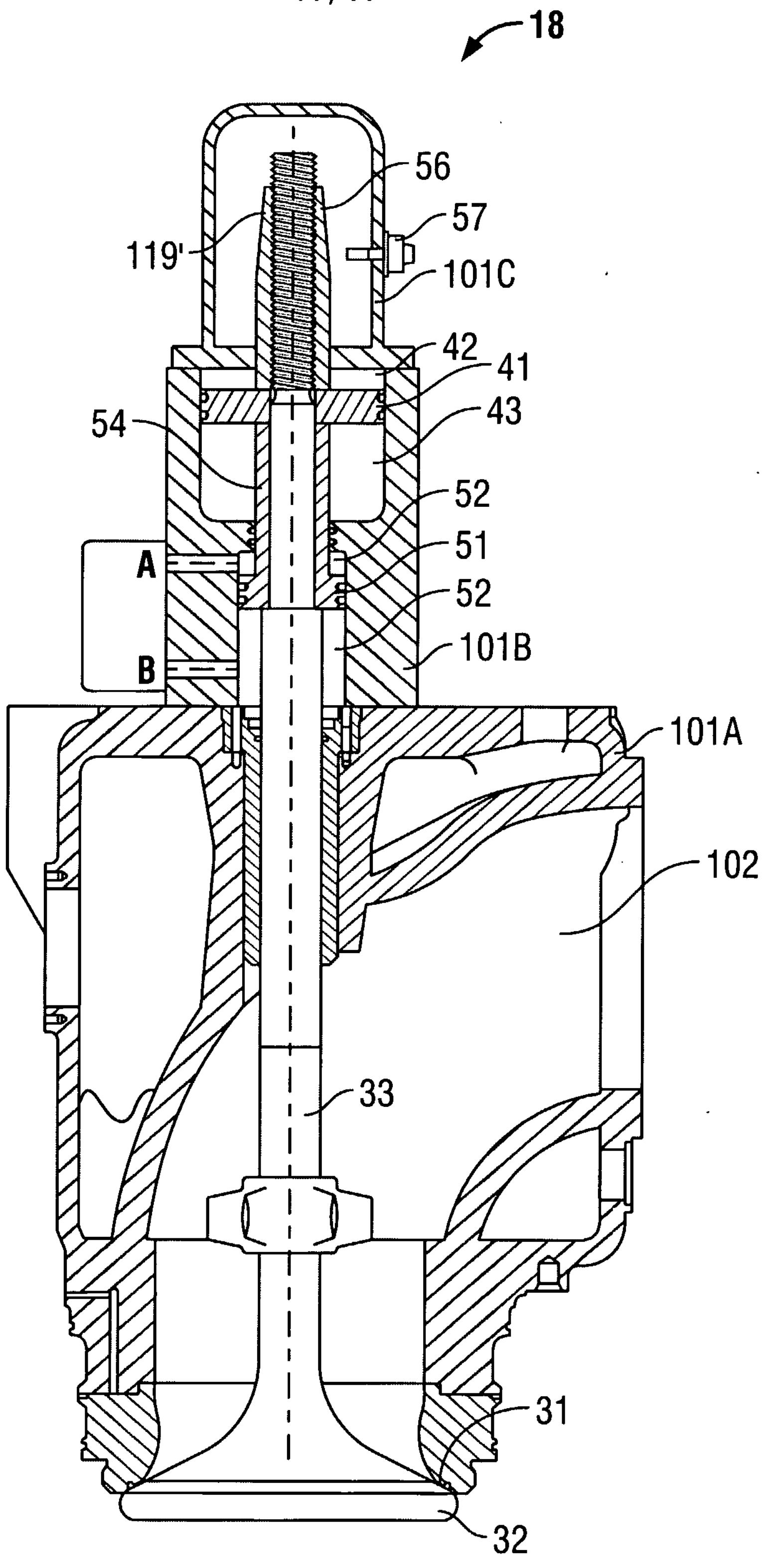

FIG. 19

INTERNATIONAL SEARCH REPORT

International Application No
PCT/EP2005/003934

A. CLASSIFICATION OF SUBJECT MATTER
F02B1/12 F01L9/02 F02B75/00 F01L1/46

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)
F02B F01L F02D

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	WO 98/57048 A (MAN B & W DIESEL A/S; LINDQUIST, HENNING; CENKER, POUL) 17 December 1998 (1998-12-17) the whole document	1-54
Y	WO 99/17003 A (KVAERNER ASA; HILDRE, HANS, PETTER) 8 April 1999 (1999-04-08) the whole document	1-54
Y	WO 02/29216 A (RENAULT SPORT; MASSE, SERGE) 11 April 2002 (2002-04-11) page 9, lines 20-31; claims 1,2; figure 1 page 10, lines 21-30	1-54
A	DE 101 61 438 A1 (MAN B&W DIESEL A/S, COPENHAGEN SV) 3 July 2003 (2003-07-03) the whole document	1-56

-/--

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

[X] Patent family members are listed in annex.

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"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

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
7 December 2005	15/12/2005

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	Clot, P

INTERNATIONAL SEARCH REPORT

International Application No
PCT/EP2005/003934

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	DE 42 20 689 A1 (MAN B&W DIESEL A/S, HVIDOVRE, DK; MAN B & W DIESEL A/S, KOPENHAGEN/KOE) 7 January 1993 (1993-01-07) the whole document	1-56
A	WO 94/29585 A (MAN B & W DIESEL A/S; LINDQUIST, HENNING) 22 December 1994 (1994-12-22) the whole document	1-56
A	EP 0 601 639 A (PHILIPS ELECTRONICS N.V) 15 June 1994 (1994-06-15) claim 1; figure 1	1-56
A	US 4 930 464 A (LETSCHE ET AL) 5 June 1990 (1990-06-05) claim 1; figure 1	1-27
A	US 5 809 950 A (LETSCHE ET AL) 22 September 1998 (1998-09-22) column 6, line 61 - column 7, line 10; figure 1	1,16,24, 28
A	US 6 315 265 B1 (ADLER SCOTT PETER ET AL) 13 November 2001 (2001-11-13) claim 1; figure 1	1-27

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/EP2005/003934

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
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		GR 1001229 B	30-06-1993
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		KR 251393 B1	15-04-2000
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		JP 2118325 C	06-12-1996
		JP 2153209 A	12-06-1990
		JP 8006567 B	24-01-1996
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		FR 2749346 A1	05-12-1997
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US 6315265 B1	13-11-2001	NONE	