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(54) Benævnelse: **FUEL SUPPLY SYSTEM FOR A LARGE TWO-STROKE COMPRESSION-IGNITED HIGH-PRESSURE GAS INJECTION INTERNAL COMBUSTION ENGINE**

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(57) Sammendrag:
A fuel supply system for supplying high-pressure gas to a large two-stroke compression-ignited internal combustion engine. The engine is provided with a fuel injection system for injecting the supplied high-pressure gas into the combustion chambers of the engine. The fuel supply system comprises a feed conduit (9) connecting an outlet of a liquefied gas storage tank (8) to the inlet of a high-pressure pump (40) for transporting liquefied gas from the liquefied gas storage tank (8) to the high-pressure pump (40), a transfer conduit (50) connecting the outlet of the highpressure pump (40) to the inlet of a high-pressure vaporizer(14) for transporting high-pressure liquefied gas from the high-pressure pump (40) to the high-pressure vaporizer (14), a supply conduit (18) connecting the outlet of the highpressure vaporizer (14) to an inlet of the fuel injection system of the engine for transporting high-pressure vaporized gas to the fuel injection system of the engine. The highpressure pump (40) comprises two or more pump units (41,42,43). Each pump unit (41,42,43) comprises a pump piston (62) slidably disposed in a pump cylinder (61) and a hydraulically powered drive piston (46) slidably disposed in a drive cylinder (45) with the drive piston (46) coupled to the pump piston (62) for driving the pump piston (62).

Fortsættes ...

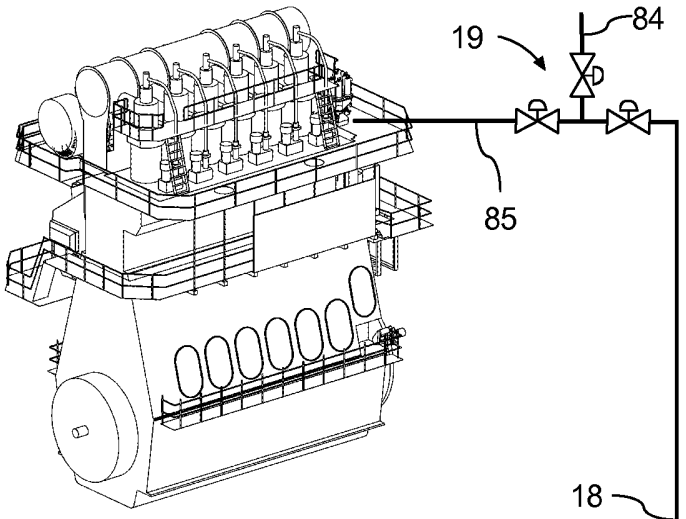
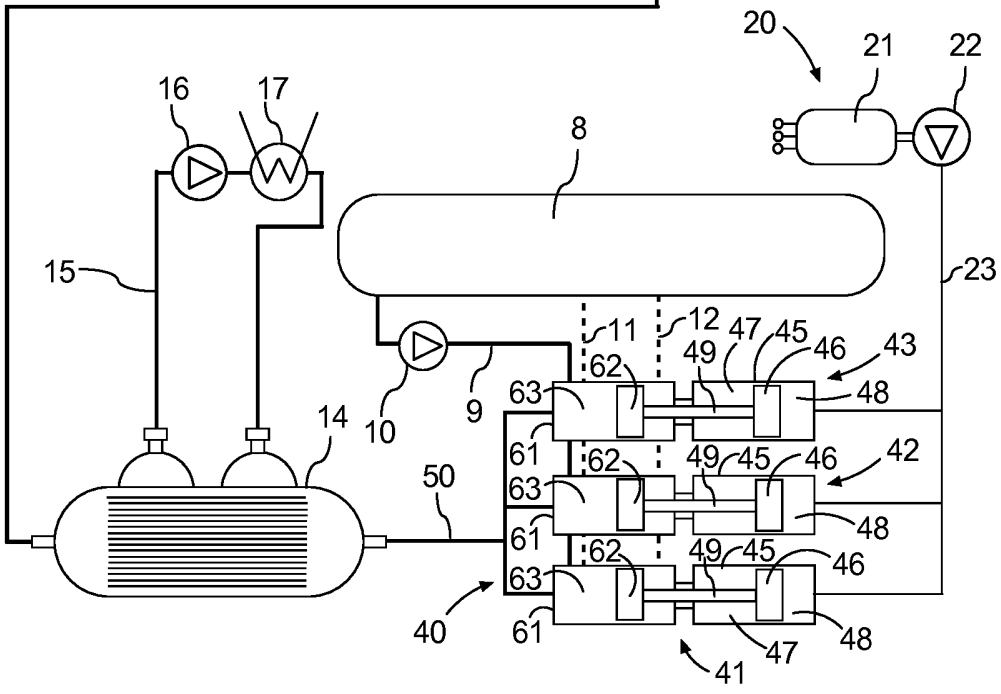

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Fig. 2


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FUEL SUPPLY SYSTEM FOR A LARGE TWO-STROKE COMPRESSION-IGNITED HIGH-PRESSURE GAS INJECTION INTERNAL COMBUSTION ENGINE

TECHNICAL FIELD

The disclosure relates to a fuel supply system for a large slow-running two-stroke uniflow compression-ignited internal combustion engines, in particular, to a fuel supply system for large slow-running two-stroke compression-ignited internal combustion engines that are supplied with high-pressure gas for injecting the gas at high-pressure into the combustion chambers of the engine.

BACKGROUND

Large two-stroke uniflow turbocharged compression-ignited internal combustion crosshead engines are typically used as propulsion systems for large ships or as prime mover in power plants. The sheer size, weight and power output renders them quite different from common combustion engines and places large two-stroke turbocharged compression-ignited internal combustion engines in a class for themselves.

Large two-stroke compression-ignited internal combustion engines are conventionally operated with a liquid fuel such as e.g. fuel oil or heavy fuel oil. However, increased focus on environmental aspects has led to the development towards using alternative types of fuel such as gas, methanol, coal slurry, petroleum coke and the like. One group of fuels that is in increasing demand is liquefied gas, in particular liquefied natural gas (LNG). Natural gas is converted to liquid form at a cryogenic temperature in a liquefaction

plant. LNG is transported over long distances to a destination by specially designed cryogenic sea vessels (LNG carriers). The LNG carrier is provided with one or more LNG storage tanks. An LNG storage tank has the ability to store LNG at the very low temperature of -162 °C (-260 °F). LNG storage tanks normally have double containers, where the inner contains LNG and the outer container contains insulation materials. The most common tank type is the full containment tank. Tanks vary greatly in size, depending on usage. Despite substantial thermal insulation heat is continually transmitted from the outside to the LNG in the LNG storage tank, causing LNG to evaporate in the LNG storage tank. If these LNG vapors are not released from the LNG storage tank, the pressure and temperature within the LNG storage tank will continue to rise, which is not acceptable and dangerous. LNG is a cryogen, and is kept in its liquid state at very low temperatures. The temperature within the tank will remain constant if the pressure is kept constant by allowing boil off gas to escape from the storage tank. This process known as auto-refrigeration. Thus, LNG is continually vaporized and boil-off gas is generated in the LNG storage tank during transporter of LNG by the LNG carrier.

The boil-off gas generated in the LNG storage tank is used as a fuel for a ship propulsion engine or burned in a gas combustor.

In case where a high-pressure gas injection engine, for example, large two-stroke compression-ignited internal combustion engine is used as a ship propulsion engine of an LNG carrier, a high-pressure cryogenic pump are used to pump

high-pressure liquefied natural gas from the LNG storage tank to a high-pressure vaporizer. Cryogenic pumps typically have two or more pump cylinders with pump pistons slidably arranged therein. Known cryogenic pumps use a crankshaft to drive the pump pistons. The crankshaft is driven by an electric drive motor via a belt drive.

The pressure of the LNG delivered by the high-pressure cryogenic pump to the vaporizer is regulated by the operation of the electric drive motor and by using control valves. However, this known control system is relatively slow and difficult to control, especially in transient operation with relatively rapidly changing the demand for fuel from the large two-stroke diesel engine.

EP2832972 discloses a fuel gas supply apparatus comprising a reciprocating piston cryogenic pump that is driven by a rotary hydraulic motor. The inertia of the components of the reciprocating piston cryogenic pump, such as the crankshaft and pistons combined with the inertia of the rotary hydraulic motor result in the cryogenic pump assembly to have a significant combined inertia. Consequently, changes in the supply pressure to the rotary hydraulic motor will not be instantaneously translated into a change in pressure at the outlet of the reciprocating piston cryogenic pump since the energy accumulated in the moving mass will need time to be converted into energy at the outlet of the pump. Thus, dynamic control of the pressure delivered by the reciprocating piston cryogenic pump will be hampered by this delay and thus be inaccurate in situations where the delivered pressure needs to be dynamically adapted.

There is therefore a need to provide an improved fuel supply system for supplying high-pressure gas to a large two-stroke compression ignited internal combustion engine.

SUMMARY

It is an object of the invention to provide a fuel supply system that overcomes or at least reduces the problems indicated above.

The foregoing and other objects are achieved by the features of the independent claims. Further implementation forms are apparent from the dependent claims, the description and the figures.

According to a first aspect there is provided a fuel supply system for supplying high-pressure gas to a large two-stroke compression-ignited internal combustion engine, the engine being provided with a fuel injection system for injecting the supplied high-pressure gas into the combustion chambers of the engine, the fuel supply system comprising a feed conduit connecting an outlet of a liquefied gas storage tank to the inlet of a high-pressure pump for transporting liquefied gas from the liquefied gas storage tank to the high-pressure pump, a transfer conduit connecting the outlet of the high-pressure pump to the inlet of a high-pressure vaporizer for transporting high-pressure liquefied gas from the high-pressure pump to the high-pressure vaporizer, a supply conduit connecting the outlet of the high-pressure vaporizer to an inlet of the fuel injection system of the engine for transporting high-pressure vaporized gas to the fuel

injection system of the engine, the high-pressure pump comprising two or more separately operated pump units, each pump unit comprising a pump piston slidably disposed in a pump cylinder and a hydraulically powered drive piston slidably disposed in a drive cylinder with the drive piston coupled to the pump piston for driving the pump piston.

By providing a fuel supply system with a high-pressure pump in which each of the pump pistons is activated by linear hydraulic actuator, the pressure of the high-pressure liquefied gas delivered to the vaporizer can be controlled accurately by controlling the pressure of the hydraulic fluid supply to the linear actuators. This is possible because a drive system based on hydraulic linear actuators has practically no inertia when compared to other types of drives and the drive system based on hydraulic linear actuators responds immediately to changes in pressure in the hydraulic fluid delivered to the hydraulic linear actuators. Thus, a change in the pressure of the hydraulic fluid delivered to the linear actuators reflects immediately in the pressure of the liquefied gas supplied to the vaporizer. Controlling the hydraulic supply pressure is relatively easy and straightforward. Thus, the gas pressure can be controlled with a significantly faster response time and less overshoot.

According to a first possible implementation of the first aspect the fuel supply system further comprising at least one hydraulic control valve connected to a source of high-pressure hydraulic fluid and to tank for controlling the flow of hydraulic fluid to and from the drive cylinder of one or more of the pump units, the source of high-pressure hydraulic fluid

preferably being a source with a variable and controllable pressure level.

According to a second possible implementation of the first aspect the drive cylinder comprises a drive chamber and a return chamber.

According to a third possible implementation of the first aspect the drive chamber is connected to the hydraulic control valve and the return chamber is, preferably permanently, connected to a source of hydraulic fluid with a pressure that is lower than the pressure of the source of high-pressure hydraulic fluid.

According to a fourth possible implementation of the first aspect the drive cylinder is provided with a position sensor for sensing the position of the drive piston in the drive cylinder concerned.

According to a fifth possible implementation of the first aspect the fuel supply system further comprises an electronic control unit in receipt of a signal from the position sensor, wherein the at least one hydraulic control valve is an electronic control valve coupled to the electronic control unit.

According to a sixth possible implementation of the first aspect the electronic control unit is configured for selectively connecting the drive chambers of the pump units to the source of high-pressure hydraulic fluid or to tank.

According to a seventh possible implementation of the first aspect the electronic control unit is configured to start a pump stroke of a drive piston when the pump stroke of another drive piston is nearing its end so that there is a small overlap between the ending pump stroke and the starting pump stroke. Thus, a substantially stable flow of LNG to the vaporizer without significant pressure fluctuations can be achieved

According to an eighth possible implementation of the first aspect the electronic control unit is configured to take into account the dynamics of the ending of a pump stroke and the dynamics of the starting pump stroke in order to obtain a substantially constant flow of high-pressure liquefied gas from the high-pressure pump to the high-pressure vaporizer.

According to a ninth possible implementation of the first aspect the electronic control unit is configured to determine when the pump stroke of one of the drive cylinders/units has to start and to determine when a pump stroke of any of the drive cylinders has to end. Thus, the point at which the pump stroke starts and especially where a pump stroke ends can be accurately controlled.

According to a tenth possible implementation of the first aspect the electronic control unit is configured to activate the respective drive cylinders substantially consecutively, preferably with a small overlap.

According to an eleventh possible implementation of the first aspect the electronic control unit is configured to operate

the drive pistons of the remaining functioning pump units when one of the pump units is out of order. Thus, redundancy is obtained and pumping action can be continued if one of the pump units is out of order.

According to a twelfth possible implementation of the first aspect the electronic control unit is configured to operate the drive pistons of the remaining functioning pump units such that the drive cylinders of the remaining functioning pump units are activated substantially consecutively, preferably with a small overlap.

According to a thirteenth possible implementation of the first aspect the electronic control unit is configured to adjust the position of the drive piston at which the drive chamber is disconnected from the source of high-pressure hydraulic fluid in relation to the magnitude of the flow of liquefied gas from the high-pressure pump to the high-pressure vaporizer. Thus, the position in which the pump stroke reverses can be kept the same, regardless of the speed and the resulting inertia of the pump piston and drive piston.

According to a fourteenth possible implementation of the first aspect the electronic control unit is configured to adjust the position of the drive piston at which the drive chamber of the drive piston concerned is disconnected from the source of high-pressure fluid in the direction opposite to the direction of the drive stroke when the flow of liquefied gas from the high-pressure pump to the high-pressure vaporizer increases.

According to a fifteen possible implementation of the first aspect the electronic control unit is configured to adjust the position of the drive piston at which the drive chamber of the drive piston concerned is disconnected from the source of high-pressure fluid in the direction of the direction of the drive stroke when the flow of liquefied gas from the high-pressure pump to the high-pressure vaporizer decreases.

According to a sixteenth possible implementation of the first aspect the electronic control unit is configured to adjust the position of a drive piston at which the drive chamber of the drive piston concerned is disconnected from the source of high-pressure fluid according to an algorithm, plan or randomly in order to distribute the position at which the pump piston reverses over an area of the stroke of the pump piston in order to reduce wear of the pump cylinder.

According to a seventeenth possible implementation of the first aspect the electronic control unit is configured to control the pressure of the liquefied gas in the transfer conduit by controlling the pressure of the hydraulic fluid supplied to the drive chambers. Thus, an effective and immediately responsive control of the pressure of the liquefied gas in the transfer conduit is achieved.

According to an eighteen possible implementation of the first aspect the electronic control unit is configured to use a desired pressure of the liquefied gas in the transfer conduit in a feed-forward function for controlling the pressure of the hydraulic fluid supplied to the drive chambers. By using feed-forward control of the liquefied gas pressure via the

hydraulic pressure, an even faster and stable control of the pressure in the liquefied gas can be achieved.

According to a nineteenth possible implementation of the first aspect the electronic control unit is configured to use a measured pressure of the liquefied gas in the transfer conduit in a feedback function for controlling the pressure of the hydraulic fluid supplied to the drive chambers. Thus, nonlinearities and transients can be accommodated by the control system.

According to a twentieth possible implementation of the first aspect the electronic control unit is configured to control the activation and deactivation of the respective drive pistons independently from the control the pressure of the hydraulic fluid supplied to the drive chambers. Thus, the control strategy for the activation of the drive pistons can be optimized by the electronic control unit independently of the pressure control.

According to a twenty-first possible implementation of the first aspect the electronic control unit is configured to use a signal representative of the position of the drive pistons to control the activation and deactivation of the drive pistons.

According to a second aspect, there is provided a large, two-stroke turbocharged compression ignited internal combustion engine with a high-pressure gas injection system and a fuel supply system according to the first aspect and any possible implementation thereof.

According to a third aspect there is provided an LNG carrier or a cargo vessel with a liquefied gas tank comprising an engine according to the second aspect.

According to a fourth aspect, there is provided a method for supplying vaporized gas at high pressure to an internal combustion engine for high pressure gas injection into the engine, the method comprising:
storing liquefied gas in a liquefied gas storage tank,
pumping the liquefied gas with a high pressure pump to a high-pressure vaporizer,
vaporizing the high pressure liquefied gas in the high-pressure vaporizer,
and supplying the vaporized high pressure gas to the engine,
the high pressure pump comprising two or more separately operated pump units, each pump unit comprising a pump piston slidably disposed in a pump cylinder, and a hydraulically powered with the drive piston coupled to the pump piston for driving the pump piston, the method further comprising:
individually supplying hydraulic fluid at high-pressure to the drive cylinders for driving the drive pistons separately, and
controlling the pressure of the liquefied gas leaving the high pressure pump by individually controlling the pressure of the hydraulic fluid supplied to the drive cylinders.

According to a first possible implementation of the fourth aspect the method further comprises activating one of the drive piston for a drive stroke and thereafter, deactivating the one drive piston for a return stroke.

According to a second possible implementation of the fourth aspect the pump piston and the drive piston are connected to one another to move in unison.

According to a third possible implementation of the fourth aspect the method further comprises starting a pump stroke of a drive piston when the pump stroke of another drive piston is nearing its end so that there is a small overlap between the ending pump stroke and the starting pump stroke.

According to a fourth possible implementation of the fourth aspect the method further comprises to take into account the dynamics of the ending of a pump stroke and the dynamics of the starting pump stroke in order to obtain a substantially constant flow of high-pressure liquefied gas from the high-pressure pump to the high-pressure vaporizer.

According to a fifth possible implementation of the fourth aspect the method further comprises activating the respective drive cylinders (substantially consecutively, preferably with a small overlap.

These and other aspects of the invention will be apparent from the embodiments described below.

BRIEF DESCRIPTION OF THE DRAWINGS

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

Fig. 1 is an elevated front view of a large two-stroke diesel engine according to an example embodiment,

Fig. 2 is a diagrammatic representation a fuel supply system for supplying high-pressure natural gas from an LNG storage tank to the large two-stroke diesel engine according to Fig. 1,

Fig. 3 is an elevated view of the high-pressure pump in the fuel supply system of Fig. 2,

Fig. 4 is a diagrammatic representation of high-pressure pump of Fig. 3,

Fig. 5 is a detailed sectional view of a pump unit of the high-pressure pump of Fig. 3,

Figs. 6 to 8 are graphs illustrating the operation of the high-pressure pump of Fig. 3,

Fig. 9 is a diagrammatic representation of a control system for controlling the high-pressure pump of Fig. 3, and

Figs. 10 and 11 are graphs illustrating the piston movements of the high-pressure pump of Fig. 3 at various speeds.

DETAILED DESCRIPTION

In the following detailed description, a fuel supply system for a large two-stroke low-speed turbocharged compression-ignited internal combustion engine with crossheads will be described with reference to the example embodiments, but it is understood that the internal combustion engine could be of another type, such as a two-stroke Otto, a four-stoke Otto or Diesel, with or without turbocharging, with or without exhaust gas recirculation or selective catalytic reduction.

Fig. 1 shows a large low-speed turbocharged two-stroke diesel engine with a turning wheel and crossheads. In this example

embodiment the engine has six cylinders in line. Large low-speed turbocharged two-stroke diesel engines have typically between four and fourteen cylinders in line, carried by a cylinder frame that is carried by an engine frame 6. The engine may e.g. be used as the main engine in a marine vessel or as a stationary engine for operating a generator in a power station. The total output of the engine may, for example, range from 1,000 to 110,000 kW.

The engine is in this example embodiment a compression-ignited engine of the two-stroke uniflow type with scavenge ports at the lower region of the cylinder 1 and a central exhaust valve 4 at the top of the cylinder liners 1. The scavenge air is passed from the scavenge air receiver 2 to the scavenge ports of the individual cylinders 1. A piston in the cylinder liner 1 compresses the scavenge air, high-pressure gaseous fuel is injected through fuel valves in the cylinder cover, combustion follows and exhaust gas is generated.

When an exhaust valve 4 is opened, the exhaust gas flows through an exhaust duct associated with the cylinder 1 into the exhaust gas receiver 3 and onwards to a turbine of the turbocharger 5, from which the exhaust gas flows away through an exhaust conduit to the atmosphere. The turbine of the turbocharger 5 drives a compressor supplied with fresh air via an air inlet. The compressor delivers pressurized scavenge air to a scavenge air conduit leading to the scavenge air receiver 2. The scavenge air in the scavenge air conduit passes an intercooler 7 for cooling the scavenge air.

Fig. 2 is a schematic view of a fuel supply system for the engine. The fuel supply system can be installed on a marine vessel, such as e.g. an LNG carrier or a cargo vessel with a liquefied gas tank, such as e.g. a container ship with a liquefied gas tank.

The fuel supply system comprises an LNG storage tank 8 in which natural gas is stored under cryogenic conditions. The pressure in the LNG storage tank 8 is relatively low and kept constant by allowing boil off gas to escape from the tank, e.g. for use in a boiler or a low pressure gas injection engine, such as an auxiliary engine of the marine vessel. The boil off process also keeps the LNG in the storage tank cold. The liquefied gas in storage tank 8 can be of another type than natural gas, such as e.g. ethane or methane.

A feed conduit 9 connects an outlet of the LNG storage tank 8 to the inlet of a high-pressure pump 40. A low-pressure feed pump 10 assists the transport of the liquefied gas from the LNG storage tank 8 to the inlet of the high-pressure pump 40. Alternatively, the LNG storage tank 8 can be pressurized so that the low pressure supply pump 10 can be dispensed with. A transfer conduit 50 connects the outlet of the high-pressure pump 40 to the inlet of a high-pressure vaporizer 14 for transporting high-pressure liquefied gas from said high-pressure pump 40 to said high-pressure vaporizer 14. The high-pressure pump 40 pumps liquefied gas via said transfer conduit 50 to the high-pressure vaporizer 14. The high-pressure vaporizer 14 receives the high-pressure liquefied gas and vaporizes the gas using a heat exchanger in the high-pressure vaporizer 14. The-pressure vaporizer 14 exchanges heat

between a heat exchange medium, such as e.g. glycol, that circulates through a circulation circuit 15 and the liquefied gas. The circulation circuit 15 includes a circulation pump 16 and a heater 17. High-pressure vaporized gas leaves the high-pressure vaporizer 14 via the outlet of the high-pressure vaporizer 14 that is connected to a supply conduit 18.

A supply conduit 18 connects the outlet of the high-pressure vaporizer 14 to an inlet of the fuel injection system of the engine and allows for transporting high-pressure vaporized gas to the fuel injection system of the engine. A valve arrangement 19 controls the connection between the fuel supply system and the large two-stroke diesel engine.

The high-pressure pump 40 is provided with two or more pump units 41,42,43, (in the present embodiment 3 pump units are shown). Each pump unit 41,42,43 includes a pump piston 62 slidably disposed in a pump cylinder 61 and a hydraulically powered drive piston 46 slidably disposed in a drive cylinder 45 with the drive piston 46 coupled to the pump piston 62 for driving said pump piston 62.

The pump piston 62 and the pump cylinder 61 form a cryogenic positive displacement pump. The pump piston 62 and the pump cylinder 61 form the so-called cold end of a pump unit with a pump chamber 63. The cold end is kept cool by a circulation circuit that includes a liquefied gas circulation supply conduit 11 and a liquefied gas circulation return conduit 12. The circulated liquefied gas serves to cool the cold end of the pump units 41,42,43.

The pump cylinder 61 is connected to the drive piston of the pump unit 41,42,43 concerned via a piston rod 49. The drive piston 46 divides the interior of the drive cylinder 45 into a drive chamber 48 and a return chamber 47.

The drive cylinders 45 are connected to a source of high-pressure hydraulic liquid 20, e.g. a pump or pump station via a high-pressure hydraulic liquid supply conduit 23. In the shown embodiment the source of high-pressure hydraulic fluid 20 includes an electric drive motor 21 that drives a high-pressure pump 22. The high-pressure pump 22 can e.g. be a positive displacement pump, preferably a variable displacement positive displacement pump. For redundancy purposes the source of high-pressure hydraulic fluid includes in an embodiment two high-pressure hydraulic pumps 22, each driven by its own electric drive motor 21.

Fig. 3 is an elevated view of the high-pressure pump 40, with three pump units 41, 42, 43 with their pump cylinders 61, drive cylinders 45 and control valves 24, supported by a frame 35 together with accumulators 53 for equalizing the upper pressure of the high-pressure pump 40, and for equalizing the lower pressure for the return chamber The pump units 41, 42, 43 are arranged in a compact way on the frame 35 and the components on the frame 35 have no spark generating components and only ATEX approved electrical components, thereby allowing the unit to be installed without problems in an ATEX environment.

Fig. 4 is a diagrammatic representation of high-pressure pump 40 with its pump units 41, 42 and 43. Each pump unit 41, 42,

43 is connected to tank via a hydraulic liquid return line 26 and to the source of high pressure hydraulic liquid that includes a variable displacement positive displacement pump 22 that connects to the respective pump units 41, 42 and 43 via a hydraulic liquid supply conduit 23. Each pump unit 41, 42 and 43 is connected to the transfer conduit 50.

Each pump unit 41, 42, 43 comprises hydraulic control valve 24 configured to selectively connect the respective drive chamber 48 to the source of high pressure hydraulic liquid or to tank via a control conduit 25.

Each pump unit 41, 42, 43 comprises a drive unit 44 in the form of a linear hydraulic actuator that is formed by the drive cylinder 45 with the drive piston 46 slidably arranged therein. The return chamber 47 is permanently connected to a source of hydraulic pressure, that includes a hydraulic pump 30, e.g. a variable displacement positive displacement pump, via a return chamber supply line 31, that preferably includes a flow restriction 33 and is coupled to an accumulator 32 for ensuring stable supply of pressurized hydraulic liquid to the return chamber 47. Alternatively, the source of low pressure is obtained from the high pressure hydraulic system via a pressure reduction valve. In an embodiment the pressure of the hydraulic liquid supply to the return chamber is significantly less than the pressure of the hydraulic liquid supplied to the drive chamber 48. Alternatively, the effective pressure surface of the side of the drive piston 46 facing the return chamber 47 can be arranged to be significantly smaller than the effective pressure surface of the drive piston facing the drive chamber 48. In the latter case the

pressure of the hydraulic fluid in the return chamber 47 can be substantially equal to the pressure of the hydraulic fluid supply to the drive chamber 48.

Each pump unit 41, 42, 43 comprises a pump 60 in the form of a linear positive displacement pump formed by pump cylinder 61 with the pump piston 62 received therein to form a pump chamber 63. The pump chamber 63 is connected to the feed conduit 9 via a first one-way valve 51 that only allows flow to the pressure chamber 63. The pump chamber 63 is connected to the transfer conduit 50 via a second one-way valve 52 that only allows flow from the pressure chamber 63.

Fig. 5 is a detailed sectional view of a pump unit 41, 42, 43 of the high-pressure pump 40. The pump units 41,42,43 comprises a hydraulic linear actuator 44 that includes a cylinder 45 with a drive piston 46 arranged therein. The drive piston 46 is connected to a piston shaft 47, preferably formed as one unit therewith. The piston rod 49 and the drive piston 46 are provided with a bore 58 for receiving a rod 57 of a position sensor 56. The signal of the position sensor 56 is transmitted to an electronic control unit 70. The drive piston 46 divides the interior of the drive cylinder 45 in a drive chamber 48 and a return chamber 47. In Fig. 5 the return chamber is not recognizable because the drive piston 46 as reached the end of its drive stroke. The drive chamber 48 is connected to the hydraulic control valve 24 via a bore 25. The return chamber 47 is permanently connected to a source of hydraulic pressure via a bore 31.

The piston rod 47 of the linear hydraulic actuator 44 is connected to the piston rod 62 of the cryogenic pump 60. The connection between piston rod 47 and piston rod 62 is established by a connector piece 54, in a way that causes the piston rod 47 and the piston rod 62 to move in unison. The drive cylinder 45 is connected to the pump cylinder 61 by a bolt connection 55. The cryogenic pump 60 is provided with an outlet that connects the pump chamber 63 to the transfer conduit 50.

Fig. 9 is a diagrammatic representation of a control system in the form of an electronic control unit 70 for controlling the operation of the high-pressure pump 40.

The electronic control unit 70 receives a gas pressure set point 71. The gas pressure set point 71 is transmitted to a summation point 72. At first summation point 72 the measured gas pressure is subtracted and the difference between the setpoint and the measured gas pressure is transmitted to a PI controller 74 is part of a feedback control loop.

The gas pressure set point is transmitted to a feed-forward piston ratio gain unit 78. The signal from the feed-forward piston ratio gain unit 78 is compared with the signal from the PI controller 74 at a second summation point 76.

The measured gas pressure that is fed to the first summation point 72 is based on a measurement of the gas pressure in the pipe volumes 85 at the engine, i.e. downstream of the valve arrangement 19. The valve arrangement 19 is double block and bleed valve arrangement that receives flow of vaporized gas

from supply conduit 18. The measured gas pressure is filtered in a filter 86.

The result of the comparison at the second summation point 76 is transmitted to the source of high-pressure hydraulic liquid 20. Based on the signal, the source of high-pressure hydraulic liquid 20 delivers hydraulic liquid with the correct pressure to the high-pressure pumping unit 40.

The electronic control unit 70 receives a signal representative of the position of the drive pistons and processes this position signal in a piston supervision unit 92. The piston supervision unit 92 is coupled to a piston activation strategy unit 90. The details of the operation of the piston supervision unit 92 and the piston activation strategy unit 90 will be displayed explained in greater detail further below. The signal of the piston activation strategy unit 90 is transmitted to the control valves 24 of the high-pressure pump 40 for activating the drive pistons 46.

Activation of the drive pistons 46 causes liquefied high-pressure gas to be pumped through the high-pressure vaporizer 14 to the supply conduit 18.

The primary pressure control of the electronic control unit 70 is feed-forward. The PI (proportional integral) controller compensates for nonlinearities and assists with transients.

The gas pressure is self-controlled by setting the hydraulic feed pressure to the pump units 41, 42, 43. The pressure control is on the hydraulic side, and there is no need to act

on the gas side. This system makes it impossible to arrive at too high gas pressures when the hydraulic pressure is controlled properly.

The drive pistons 46 are controlled via a control strategy, that is not an active part of the pressure control.

Each pump unit 41,42,43 is individually controllable. Thus, it is possible to run different piston strategies and various operating conditions. Further, the possibility to run pump units 41,42,43 individually provides for redundancy since it is possible to change from three to two pump units 41,42,43 between two strokes.

The return speed can be greater than the forward (pump) speed, thereby making it possible to have overlap when running with two pump units only. The overlap between the pump units 41, 42, 43 can be adjusted in accordance to needs in order to reduce pressure spikes.

The end position of the pump stroke can be varied over time to distribute wear over an area of the pump cylinder 61, as opposed to having high wear at a fixed and position on the cylinder.

The system allows for little or no pressure overshoot, even at sudden shutdown (piston stop) since there is very low inertia and other factors that negatively affect the dynamic response.

The control valves 24 can be hydraulically controlled valves or electronically controlled valves. In the embodiment the control valve 24 is a hydraulically controlled valve there is provided an electronically controlled solenoid valve (not shown) that controls the hydraulic control signal to the control valve 24. The electronically controlled solenoid valve receives electronic control signals from the electronic control unit 70.

The electronic control unit 70, in particular the piston activation strategy unit 90, is configured for selectively connecting the drive chamber 48 of the pump units 41,42,43 to the source of high-pressure hydraulic fluid 20 or to tank.

The electronic control unit 70, in particular the piston activation strategy unit 90, is configured to start a pump stroke of a drive piston 47 when the pump stroke of another drive piston 47 is nearing its end so that there is a small overlap between the ending pump stroke and the starting pump stroke. In an embodiment the electronic control unit 70 is configured to activate the respective drive cylinders substantially consecutively, preferably with a small overlap. Thus, a substantially stable flow of LNG to the high-pressure vaporizer 14 without significant pressure fluctuations can be achieved, as illustrated in Figs. 6 and 7.

Figs. 6,7 and 8 illustrate a typical operation of the high-pressure pump 40. The thin continuous line represents pump unit 41 the thick continuous line represents pump unit 42 and the dotted line represents pump unit 43. Fig. 6 is a graph illustrating the movement of the drive pistons 46/pump pistons

62. As can be seen from the graph, the start of the pump stroke of the next pump unit starts just ahead of the end of the pump stroke of the presently active pump unit. Figure 7 illustrates the resulting pressure composed from the pressure output from the three pump units 41,42,43 transfer conduit 50. The resulting pressure is substantially constant and without fluctuations.

Fig. 8 shows the speed profile of the pump units in which can be clearly seen that the speed of the return stroke is significantly higher than the speed of the pump stroke, thus allowing for overlap between the pump units, even if only two out of three or more pump units are in use.

In an embodiment, the electronic control unit 70, in particular the piston activation strategy unit 90, is configured to take into account the dynamics of the ending of a pump stroke and the dynamics of the starting pump stroke in order to obtain a substantially constant flow of high-pressure liquefied gas from the high-pressure pump to the high-pressure vaporizer 14.

In an embodiment the electronic control unit 70, in particular the piston activation strategy unit 90, is configured to determine when the pump stroke of one of the pump units 41, 42, 43 has to start and to determine when a pump stroke of any of the drive units 41, 42, 43 has to end. Thus, the point at which the pump stroke starts and especially where it ends can be accurately controlled by the piston strategy unit 90, preferably together with the piston super vision unit 92.

In an embodiment the electronic control unit 70 is configured to operate the drive pistons of the remaining functioning pump units 41, 42, 43 when one of the pump units 41, 42, 43 is out of order. Thus, redundancy is obtained and pumping action can be continued if one of the pump units 41, 42, 43 is out of order.

In an embodiment the electronic control unit 70 is configured to adjust the position of the drive piston 46 at which the drive chamber 48 is disconnected from the source of high-pressure hydraulic fluid in relation to the magnitude of the flow of liquefied gas from the high-pressure pump 40 to the high-pressure vaporizer. Thus, the position in which the pump stroke reverses can be controlled, regardless of the speed and the resulting inertia of the drive piston 46 and pump piston 62.

According to an embodiment the electronic control unit 70 is configured to adjust the position of the drive piston at which the drive chamber 48 of the drive piston 46 concerned is disconnected from the source of high-pressure fluid 20 in the direction opposite to the direction of the drive stroke when the flow of liquefied gas from the high-pressure pump to the high-pressure vaporizer increases and the electronic control unit 70 is configured to adjust the position of the drive piston 46 at which the drive chamber 48 of the drive piston 46 concerned is disconnected from the source of high-pressure fluid 20 in the direction of the direction of the drive stroke when the flow of liquefied gas from the high-pressure pump to the high-pressure vaporizer decreases. This is illustrated in Figs. 10 and 11.

Fig. 10 shows the effect of the increased speed of the drive piston 46 and pump piston 62 on the end position of the drive/pump stroke. The thin continuous line represents pump unit 41 the thick continuous line represents pump unit 42 and the dotted line represents pump unit 43. The electronic control unit 70 signals the hydraulic control valve 24 to connect the drive chamber 48 to tank when the drive piston has reached a stroke of 80 mm, regardless of the load/magnitude of the flow of liquefied gas delivered by the high-pressure pump 40. Due to the inertia and the higher-speed the stop/reverse position of the drive piston 46 changes from 85 mm at 25% load to 89 mm at 50% load to 98 mm at hundred percent load.

Fig. 11 is a graph illustrating the effect of the electronic control unit 70 compensating for the increased speed of the drive piston 46/pump piston 62 by connecting the drive chamber 48 to tank at shorter stroke when the load is high and at a longer stroke when the load is low. As can be seen in the graph, the electronic control unit 70 can in this way accurately control the end position of the drive/pump stroke.

In the example in the graph the signal for connecting the drive chamber 48 to tank for 25% load (i.e. 25% of the maximum capacity of the high-pressure pump 40) for the next drive cylinder is issued when the previous cylinder is 75 mm into the drive chamber. The drive chamber of the “previous” drive cylinder is connected to tank when it is 93 mm into the drive chamber. The connection to the source of high pressure of the next drive cylinder "signal on” and the connection to tank of

the "previous" cylinder "signal off" is illustrated in table 1 below.

	25% load	45% load	70% load	100% Load
Signal on	75 mm	75 mm	75 mm	75 mm
Signal off	93 mm	86 mm	83 mm	80 mm
Stop position	97 mm	97 mm	97 mm	97 mm

Table 1

Of course, it is also still possible to program the electronic control unit 72 deliberately varied the start position in order to reduce wear of the pump cylinder 61.

In an embodiment the electronic control unit 70 is configured to adjust the position of a drive piston 46 at which the drive chamber 48 of the drive piston 46 concerned is disconnected from the source of high-pressure fluid 20 according to an algorithm, plan or randomly in order to distribute the position at which the pump piston 62 reverses over an area of the stroke of the pump piston 62 in order to reduce wear of the pump cylinder 61. It is known that wear on the pump cylinder 61 is highest at the position where the pump stroke ends. By varying the position in which the pump stroke ends the wear of the pump cylinder 61 can be spread over a larger area and thus the life span of the pump cylinder 61 can be significantly increased.

In an embodiment the electronic control unit 70 is configured to control the activation and deactivation of the respective drive pistons 46 independently from the control pressure of the hydraulic fluid supplied to the drive chambers 48. Thus,

the control strategy for the activation of the drive pistons can be optimized by the electronic control unit 70 independently of the pressure control.

The invention has been described in conjunction with various embodiments herein. However, other variations to the disclosed embodiments can be understood and effected by those skilled in the art in practicing the claimed invention, from a study of the drawings, the disclosure, and the appended claims. In the claims, the word "comprising" does not exclude other elements or steps, and the indefinite article "a" or "an" does not exclude a plurality. The electronic control unit can be formed by a combination of separate electronic control units. The mere fact that certain measures are recited in mutually different dependent claims does not indicate that a combination of these measured cannot be used to advantage. The reference signs used in the claims shall not be construed as limiting the scope.

PATENTKRAV

1. Brændstoftilførselssystem til tilførsel af højtryksgas til en stor, totakts intern forbrændingsmotor med kompressionstænding, hvilken motor er forsynet med et brændstofindsprøjtningssystem til indsprøjtning af den tilførte højtryksgas i forbrændingskammeret i motoren,
hvilket brændstoftilførselssystem omfatter:
en forsyningsledning (9), der forbinder et udløb på en tank til lagring af flydende gas (8) med indløbet på en højtrykspumpe (40) til transport af flydende gas fra lagertanken (8) til flydende gas og hen til højtrykspumpen (40),
en overførselsledning (50), der forbinder udløbet på højtrykspumpen (40) med indløbet på en højtryksfordamper (14) til transport af den flydende højtryksgas fra højtrykspumpen (40) til højtryksfordamperen (14),
en forsyningsledning (18), der forbinder udløbet på højtryksfordamperen (14) med et indløb på brændstofindsprøjtningssystemet på motoren til transport af den fordampede højtryksgas til brændstofindsprøjtningssystemet i motoren,
kendetegnet ved, at højtrykspumpen (40) omfatter to eller flere separat aktiverede pumpeenheder (41, 42, 43), hvor hver enkelt pumpeenhed (41, 42, 43) omfatter et pumpestempel (62), der er placeret forskydeligt i en pumpecylinder (61), og en hydraulisk drevet drivstempel (46), der er placeret forskydeligt i en drivcylinder (45), hvori drivstemplet (46) er forbundet med pumpestemplet (62) for at drive pumpestemplet (62).

2. Brændstoftilførselssystem ifølge krav 1, som desuden omfatter i det mindste en reguleringsventil (24), som er forbundet med en kilde af hydraulisk væske under højt tryk (20) og med en tank til regulering af flowet af den hydrauliske væske til og fra drivcylinderen (45) på en eller

flere af pumpeenhederne (41, 42, 43), hvilken kilde af hydraulisk væske under højt tryk (20) fortrinsvis er en kilde med et variabelt og et regulerbart trykniveau.

3. Brændstoftilførselssystem ifølge krav 1 eller 2, hvori drivcylinderen (45) omfatter et drivkammer (48) og et returkammer (47).

4. Brændstoftilførselssystem ifølge krav 3, hvori drivkammeret (48) er forbundet med reguleringsventilen (24) og returkammeret fortrinsvis er forbundet med en kilde af en hydraulisk væske (30) med et tryk, der er lavere end trykket af kilden (20) af den hydrauliske væske under højt tryk.

5. Brændstoftilførselssystem ifølge et hvilket som helst af kravene 1 til 4, hvori drivcylinderen (45) er forsynet med en positionssensor (56) til registrering af positionen af drivstemplet (46) i den involverede drivcylinder (45).

6. Brændstoftilførselssystem ifølge krav 5, som desuden omfatter en elektronisk styreenhed (70), der modtager et signal fra positionssensoren (56), og hvori den i det mindste ene reguleringsventil (24) er en elektronisk reguleringsventil, der er forbundet med den elektroniske styreenhed (70).

7. Brændstoftilførselssystem ifølge krav 6, hvori den elektroniske styreenhed (70) er udformet til efter valg at forbinde drivkamrene (48) i pumpeenhederne (41, 42, 43) med kilden af hydraulisk væske under højt tryk (20) eller med tanken.

8. Brændstoftilførselssystem ifølge krav 6 eller 7, hvori den elektroniske styreenhed (70) er udformet til at i gangsætte et pumpeslag på et drivstempel (46), når pumpeslaget fra et andet drivstempel (46) nærmer sig sin afslutning, således at der er en lille overlapning mellem det afsluttende pumpeslag og det begyndende pumpeslag.

9. Brændstoftilførselssystem ifølge krav 8, hvori den elektroniske styreenhed (70) er udformet til at tage hensyn til dynamikken af afslutningen af et pumpeslag og dynamikken af det begyndende pumpeslag for at opnå et i det væsentlige konstant flow af flydende højtryksgas fra højtrykspumpen (40) til højtryksfordamperen (14).

10. Brændstoftilførselssystem ifølge et hvilket som helst af kravene 6 til 9, hvori den elektroniske styreenhed (70) er udformet til at justere den position af drivstemplet (46), ved hvilken drivkammeret (48) kobles fra kilden af hydraulisk væske under højt tryk (20) i forhold til størrelsen af flowet af flydende gas fra højtrykspumpen (40) til højtryksfordamperen (14).

11. Brændstoftilførselssystem ifølge krav 10, hvori den elektroniske styreenhed (70) er udformet til at justere den position af drivstemplet (46), ved hvilken drivkammeret (48) til den involverede drivstempel kobles fra kilden af højtryksvæske (20) i retningen, der er modsat af retningen af drivslaget, når flowet af flydende gas fra højtrykspumpen (40) til højtryksfordamperen (14) forøges.

12. Brændstoftilførselssystem ifølge krav 10 eller 11, hvori den elektroniske styreenhed (70) er udformet til at justere den position af drivstemplet (46), ved hvilken drivkammeret (48) til det involverede drivstempel (46)

kobles fra kilden af højtryksvæske (20) i retning af retningen af drivslaget, når flowet af flydende gas fra højtrykspumpen (40) til højtryksfordamperen (14) forminskes.

13. Brændstoftilførselssystem ifølge et hvilket som helst af kravene 6 til 12, hvori den elektroniske styreenhed (70) er udformet til at justere positionen, ved hvilken drivkammeret (48) til det involverede drivstempel (46) kobles fra kilden af hydraulisk væske under højt tryk (20) i overensstemmelse med en algoritme, plan eller tilfældigt for at fordele positionen, ved hvilken pumpestemplet (62) vender om over et område af slaget af pumpestemplet (62) for at reducere slitagen af pumpecylinderen (61).

14. Brændstoftilførselssystem ifølge et hvilket som helst af kravene 6 til 13, hvori den elektroniske styreenhed (70) er udformet til at regulere trykket af den flydende gas i overførselsledningen (50) ved regulering af trykket af den hydrauliske væske, der tilføres til drivkammeret (48).

15. Brændstoftilførselssystem ifølge krav 14, hvori den elektroniske styreenhed (70) er udformet til at anvende et ønsket tryk af den flydende gas i overførselsledningen (50) i en feed-forward-funktion til regulering af trykket af den hydrauliske væske, der tilføres til drivkamrene (48).

16. Brændstoftilførselssystem ifølge krav 14 eller 15, hvori den elektroniske styreenhed (70) er udformet til at anvende et målt tryk af den flydende gas eller af en fordampet gas i en feedback-funktion til regulering af trykket af den hydrauliske væske, der tilføres til drivkamrene (48).

17. Brændstoftilførselssystem ifølge et hvilket som helst af kravene 14 til 16, hvori den elektroniske styreenhed (70) er udformet til at regulere aktiveringen og deaktiveringen af de respektive drivstempler (46) uafhængigt af reguleringen af trykket af den hydrauliske væske, der tilføres til drivkamrene (48).

18. Stor, totakts turboladet intern forbrændingsmotor med kompressionstænding og med et indsprøjtningssystem til højtryksgas og et brændstoftilførselssystem ifølge et hvilket som helst af kravene 1 til 17.

19. LNG-tankskib eller fragtskib med en tank til flydende gas, der omfatter en motor ifølge krav 18.

20. Fremgangsmåde til tilførsel af fordampet gas under højt tryk til en intern forbrændingsmotor til indsprøjtning af højtryksgas i motoren, hvilken fremgangsmåde omfatter:
opbevaring af flydende gas i en lagertank (8) til flydende gas,
pumpning af den flydende gas med en højtrykspumpe (40) til en højtryksfordamper (14),
fordampning af den flydende højtryksgas i højtryksfordamperen (14)
og tilførsel af den fordampede højtryksgas til motoren,
kendetegnet ved, at højtrykspumpen (40) omfatter to eller flere separat aktiverede pumpeenheder (41, 42, 43), hvor hver pumpeenhed omfatter et pumpestempel (62), der er placeret forskydeligt i en pumpecylinder (61), og en hydraulisk drevet (45), hvor drivstemplet (46) er forbundet med pumpestemplet (62) for at drive pumpestemplet (62), hvilken fremgangsmåde desuden omfatter:

individuel tilførsel af hydraulisk væske under højt tryk til drivcylindrene (45) for at drive drivstemplerne (46) separat og

regulering af trykket af den flydende gas, der forlader højtrykspumpen (40), ved en individuel regulering af trykket af den hydrauliske væske, der tilføres til drivcylindrene.

21. Fremgangsmåde ifølge krav 20, der omfatter igangsætning af et pumpeslag på et drivstempel (46), når pumpeslaget på et andet drivstempel (46) nærmer sig sin afslutning, således at der er en lille overlapning mellem det afsluttende pumpeslag og det begyndende pumpeslag.

22. Fremgangsmåde ifølge krav 21, der omfatter aktivering af de respektive drivcylindre (45) i det væsentlige efter hinanden, fortrinsvis med en lille overlapning.

23. Fremgangsmåde ifølge krav 21, der omfatter hensyntagen til dynamikken af afslutningen af et pumpeslag og dynamikken af det begyndende pumpeslag for at opnå et i det væsentlige konstant flow af flydende højtryksgas fra højtrykspumpen (40) til højtryksfordamperen (14).

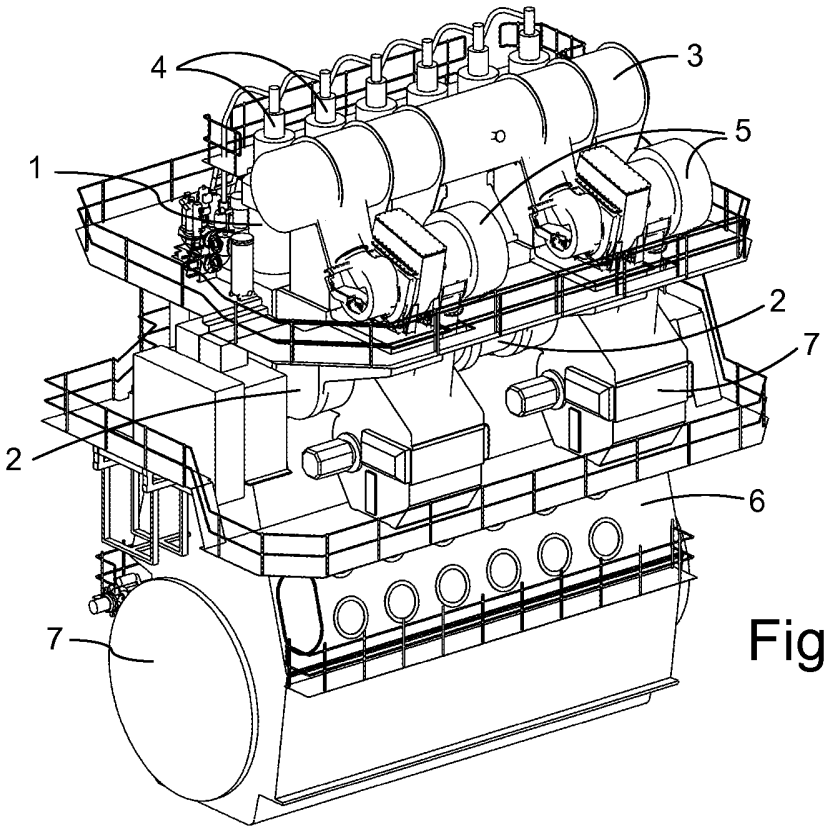


Fig. 1

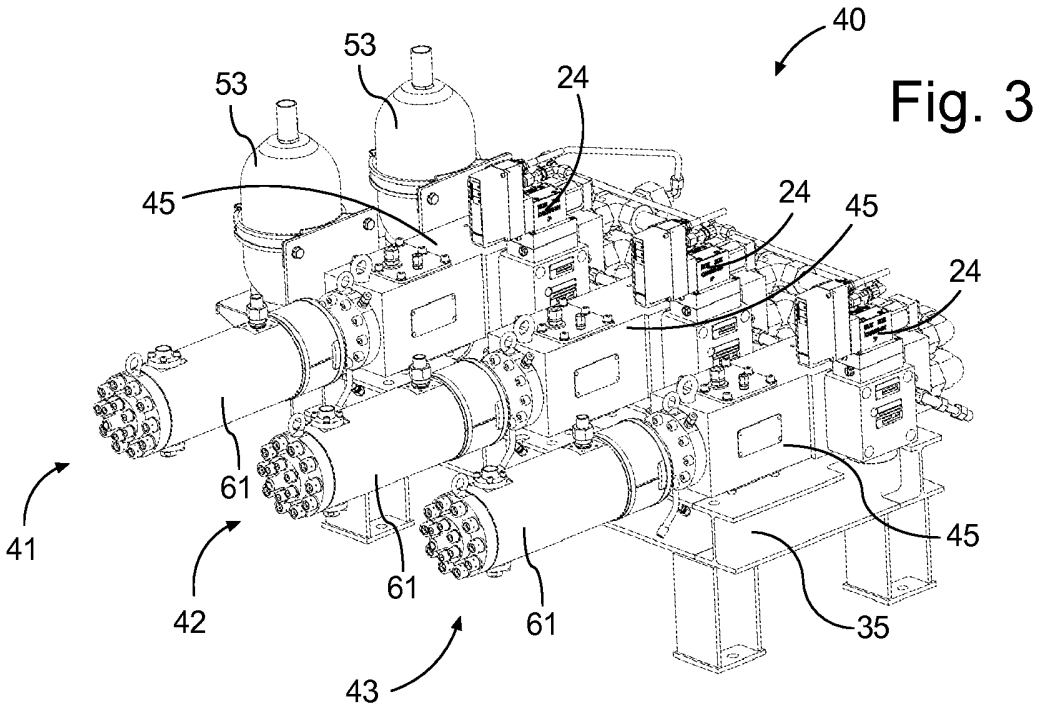


Fig. 3

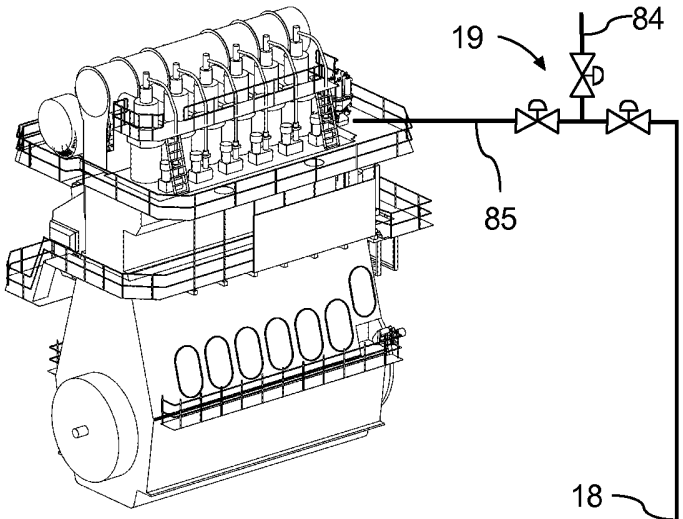

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Fig. 2

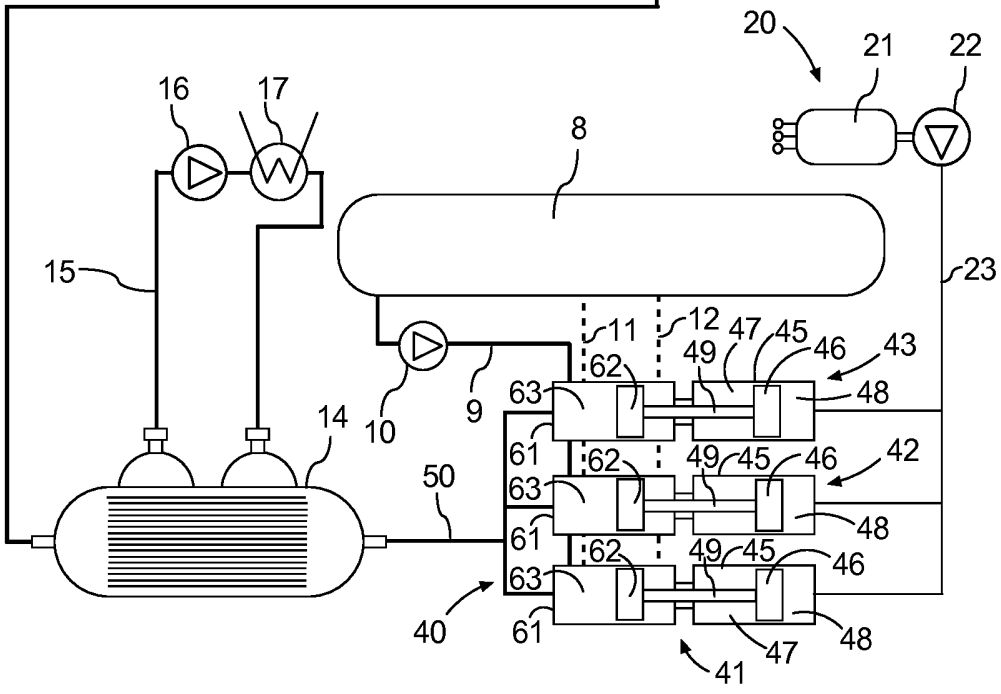

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Fig. 4

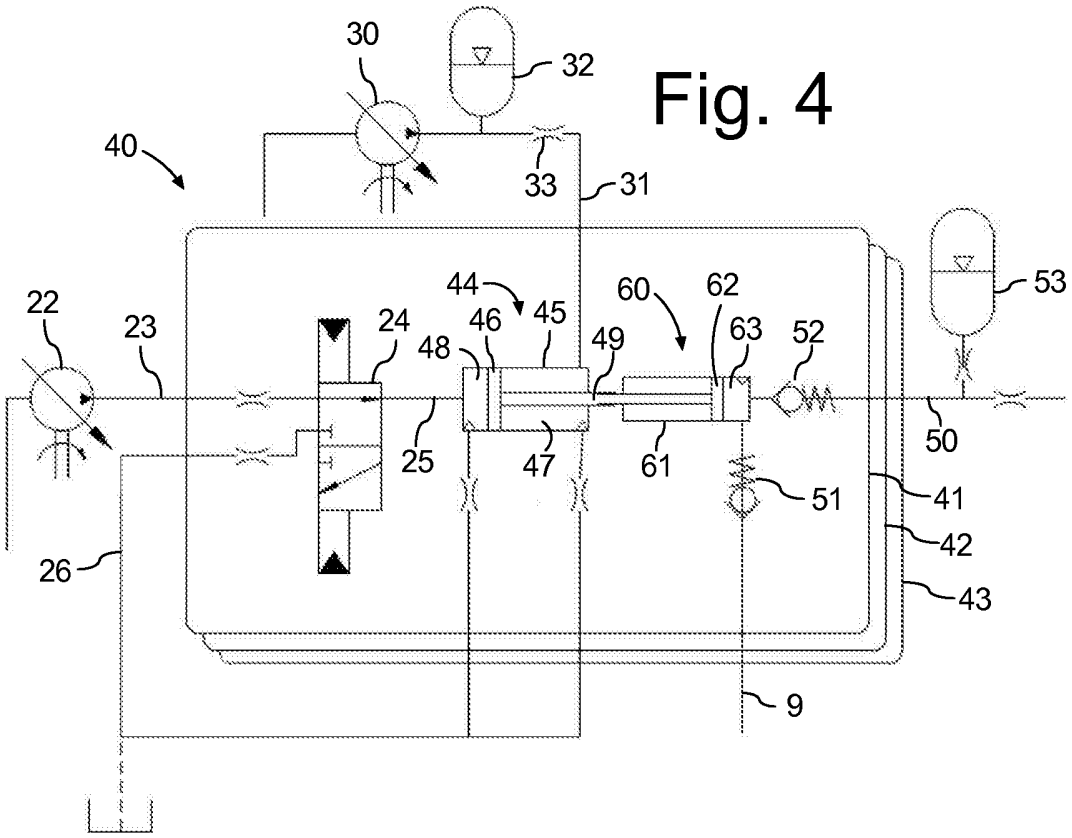
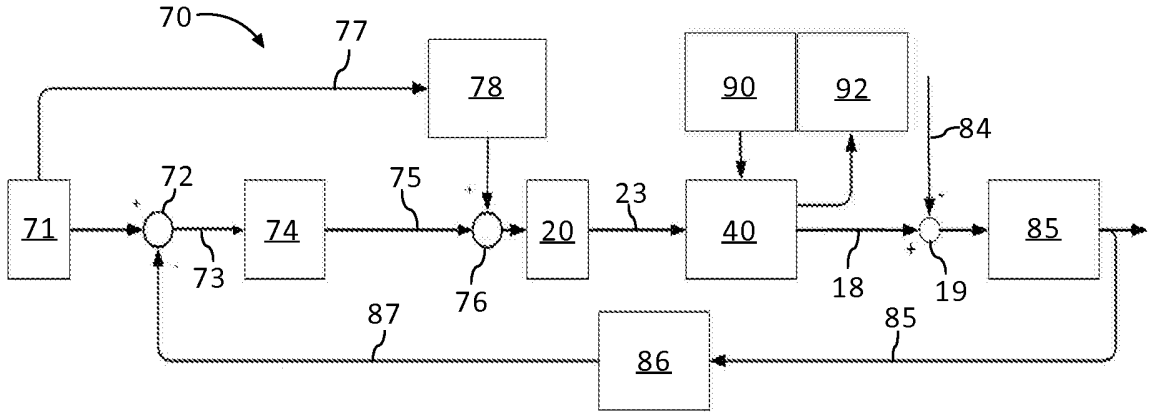

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Fig. 9


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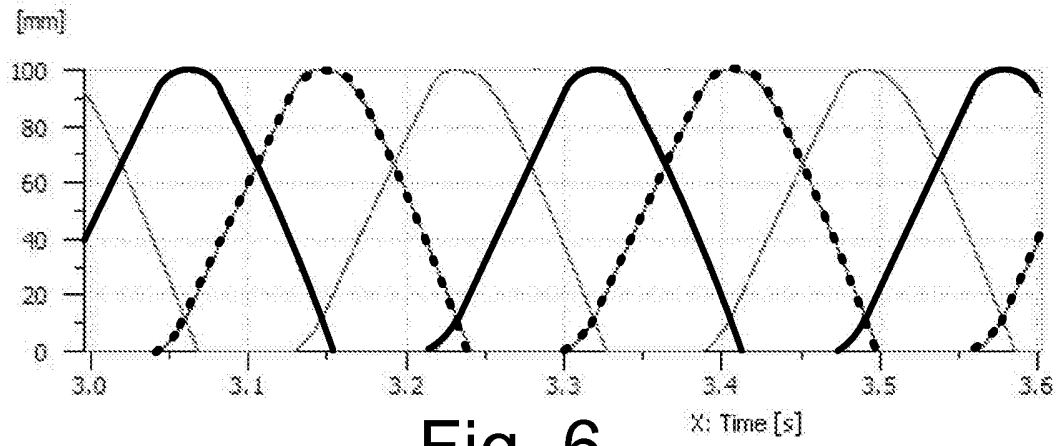


Fig. 6

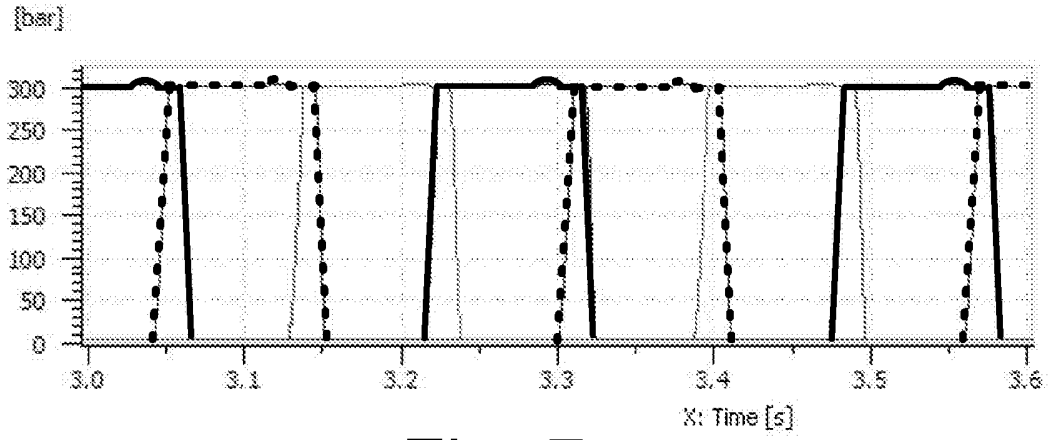


Fig. 7

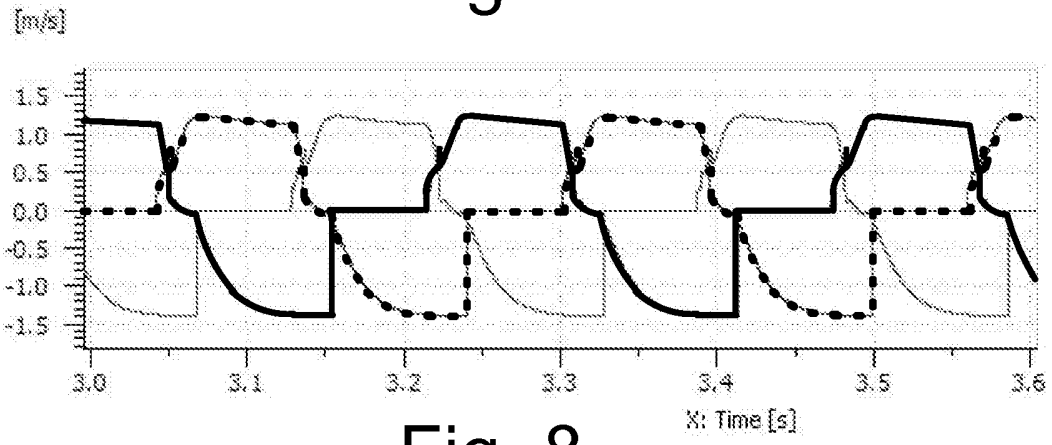


Fig. 8

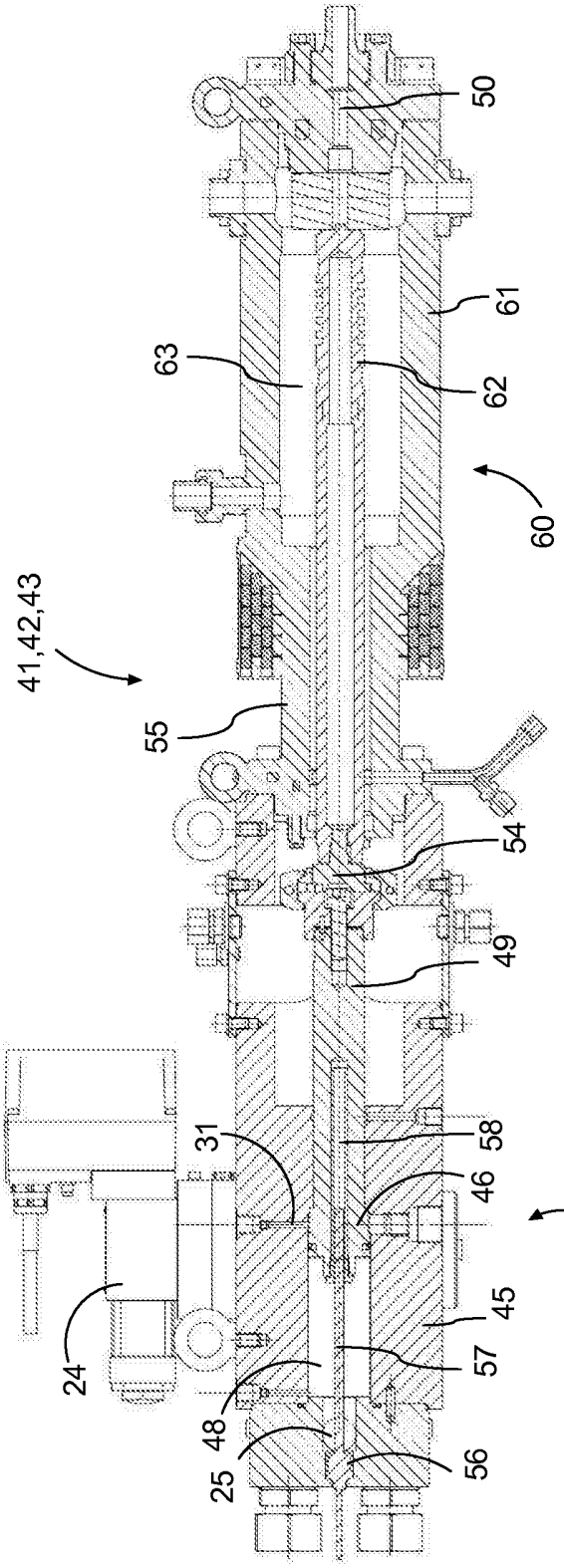


Fig. 5

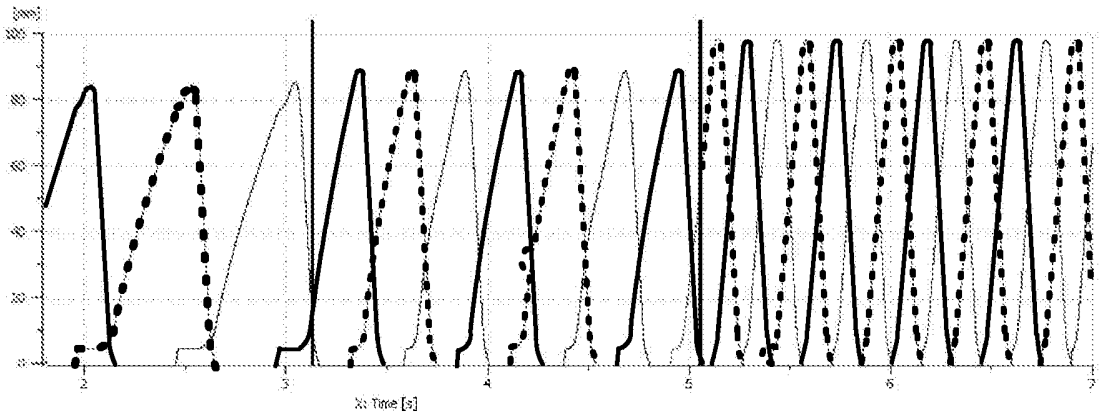


Fig. 10

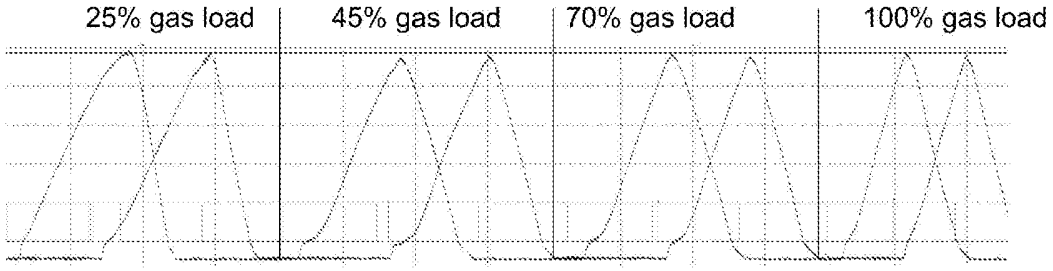


Fig. 11

DANISH PATENT AND TRADEMARK OFFICE

SEARCH REPORT - PATENT	Application No. PA 2016 70360

1. ☐ Certain claims were found unsearchable (See Box No. I).

2. ☐ Unity of invention is lacking prior to search (See Box No. II).

A. CLASSIFICATION OF SUBJECT MATTER

F02M 21/02 (2006.01); F02B 43/00 (2006.01)

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)

IPC&CPC: F02B, F02M, B63B

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

DK, NO, SE, FI: IPC-classes as above.

Electronic database consulted during the search (name of database and, where practicable, search terms used)

EPODOC, WPI, FULLTEXT:ENGLISH

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant for claim No.
Y A	EP 2832972 A1 (MITSUBISHI HEAVY IND LTD) 2015-02-04. See paras. 0001-0003, 0013-0015, 0040-0049, and fig. 1.	1-7, 18-20 8-17, 21-23
Y	WO 2016/058100 A1 (WESTPORT POWER INC) 2016-04-21. See paras. 0025, 0027, and figs. 1-2.	1-7, 18-20
A	US 2003/0051486 A1 (URSAN et al.) 2003-03-20. See paras. 0045, 0054, and fig. 1.	1-23

☒ Further documents are listed in the continuation of Box C.

* Special categories of cited documents:	"P" Document published prior to the filing date but later than the priority date claimed.
"A" Document defining the general state of the art which is not considered to be of particular relevance.	"T" Document not in conflict with the application but cited to understand the principle or theory underlying the invention.
"D" Document cited in the application.	"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.
"E" Earlier application or patent but published on or after the filing date.	"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.
"L" Document which may throw doubt on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified).	"&" Document member of the same patent family.
"O" Document referring to an oral disclosure, use, exhibition or other means.	

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Date of completion of the search report

5 December 2016

Authorized officer

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Telephone No. +45 4350 8532

SEARCH REPORT - PATENT		Application No. PA 2016 70360
C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant for claim No.
A	US 2014/0137572 A1 (STEFFEN) 2014-05-22. See paras. 0015-0018, and figs. 1-2.	1-23
A	US 2016/0017835 A1 (GARNER et al.) 2016-01-21. See paras. 0017, 0038, 0051, and figs. 1, 4.	1-23
A	EP2933183 A1 (DAEWOO SHIPBUILDING & MARINE) 2015-10-21. See paras. 0015-0019, 0130-0134, and fig. 5.	1-23

SEARCH REPORT - PATENT	Application No. PA 2016 70360

Box No. I Observations where certain claims were found unsearchable

This search report has not been established in respect of certain claims for the following reasons:

1. ☐ Claims Nos.:
because they relate to subject matter not required to be searched, namely:

2. ☐ Claims Nos.:
because they relate to parts of the patent application that do not comply with the prescribed requirements to such an extent that no meaningful search can be carried out, specifically:

3. ☐ Claims Nos.:
because of other matters.

Box No. II Observations where unity of invention is lacking prior to the search

The Danish Patent and Trademark Office found multiple inventions in this patent application, as follows:

SEARCH REPORT - PATENT	Application No. PA 2016 70360

SUPPLEMENTAL BOX

Continuation of Box [.]