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(54) **Title:** A METHOD FOR LUBRICATING A LARGE TWO-STROKE ENGINE USING CONTROLLED PRESSURE VARIATIONS IN COMMON RAIL

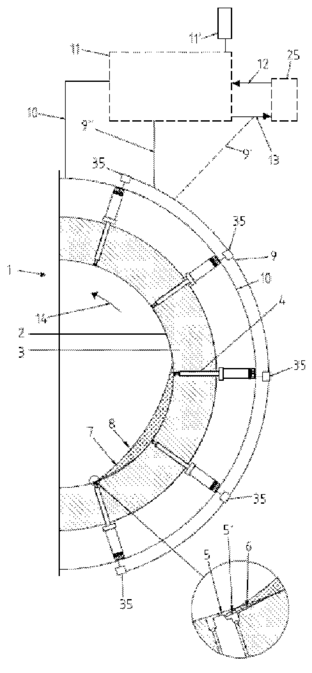

Fig. 1

(57) **Abstract:** A large engine, in particular a marine engine or engine for power plants, and a method of lubricating the engine with spray injectors, in particular SIP injectors. According to the method there is provided a controller arranged for varying and con- trolling the pressure in the lubricant supply conduit within a desired pressure range and basing the desired pressure on the engine load and a measurement of the actual pressure in the lubricant supply conduit.

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A method for lubricating a large two-stroke engine using controlled pressure variations in common rail

FIELD OF THE INVENTION

The present invention relates to a large combustion engine, for example large slow-running two-stroke engine, and a method of lubricating such engine and use thereof.

More specific the invention relates to a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a lubricant system comprising
- a lubricant supply including a lubricant pump that raises the pressure of the lubricant to a lubricant pressure which is a typical range of pressure for spray injectors,
- a plurality of lubricant injectors distributed along a perimeter of the cylinder for injection of lubricant into the cylinder at various positions on the perimeter during injection phases,
- a lubricant supply conduit connecting the lubricant supply with the lubricant injectors,
the engine further comprising- a controller for controlling the amount and timing of the lubricant injection by at least one of the lubricant injectors,
wherein each injector comprises
- an inlet port flow-connected to the lubricant supply conduit for receiving lubricant from it,
- a nozzle with a nozzle aperture extending into the cylinder configured for injecting lubricant from the inlet port into the cylinder in the injection phase.

Preferably each injector comprises an adjustable valve at the nozzle for opening and closing for flow of lubricant to the nozzle aperture in the injection phase.

The method relates to the lubricating of such large slow-running two-stroke engine. More specific to a method of lubricating a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a system comprising
- a lubricant supply including a lubricant pump,

- a plurality of lubricant injectors distributed along a perimeter of the cylinder for injection of lubricant into the cylinder at various positions on the perimeter during injection phases,
- a lubricant supply conduit connecting the lubricant supply with the lubricant injectors, the engine further comprising
- a controller for controlling the amount and timing of the lubricant injection by at least one of the lubricant injectors,
wherein each injector comprises
- an inlet port flow-connected to the lubricant supply conduit for receiving lubricant from it,
- a nozzle with a nozzle aperture extending into the cylinder configured for injecting lubricant from the inlet port into the cylinder in the injection phase,
wherein the method comprises
- raising the pressure of the lubricant to a lubricant pressure which is a typical range of pressure for spray injectors, and further in cyclic operation
- in the injection phase, providing pressure-liquid at inlet port of the injector and exerting force on the valve and by the force driving a valve body into motion inside the injector and upon pressure rise above the predetermined limit, pumping a predetermined lubricant volume through the nozzle aperture into the cylinder.

It is noted that "adjustable" means to the extent that it can be controlled how much we pull the needle and how long it must be open. However, it is also possible to use other types of valves.

Moreover, the engine may further comprise
- a computer (11’) to which the controller is connected, or alternatively
- a mobile phone to communicate with the controller.

Moreover, the engine may further comprise
-at least one flowmeter for measuring the lubricant flow.

Moreover, it is noted that “extends into the cylinder” means that the injector is embedded in the cylinder wall to ensure free movement of the piston, but extends through the cylinder wall.

BACKGROUND OF THE INVENTION

Due to the focus on environmental protection, efforts are on-going with respect reduction of emissions from marine engines. This also involves the steady optimization of lubrication systems for such engines, especially due to increased competition. One of the economic aspects gaining increased attention is a reduction of oil consumption, not only because of environmental protection but also because this is a significant part of the operational costs of ships. A further concern is proper lubrication despite reduced lubricant volume because the longevity of engines should not be compromised by the reduction of oil consumption. Thus, there is a need for steady improvements with respect to lubrication.

For lubricating of large slow-running two-stroke marine diesel engines, several different systems exist, including injection of lubrication oil directly onto the cylinder liner or injection of oil quills to the piston rings.

The lubrication may be effected as jet injection which is an injection where lubricant in metered quantity is injected as a compact jet into the liner of a cylinder or into the piston ring pack. This is also called pulse lubrication.

Alternatively, the lubrication may be effected as a spray injection where lubricant under hight pressure is injected into the combustion chamber. Hereby a fine mist of the lubricant is injected into the cylinder, preferably in the combustion chamber. The lubricant is provided as a mist of atomized droplets. The injector for spray injection is called spray injector. A specific spray injection is SIP injection. The injector for SIP injection is called SIP injector. The SIP injection is explained in further detail below.

An example of a lubricant injector for a marine engine is disclosed in EP1767751, in which a non-return valve is used to provide the lubricant access to the nozzle passage inside the cylinder liner. The non-return valve comprises a reciprocating spring-pressed ball in a valve seat just upstream of the nozzle passage, where the ball is displaced by pressurised lubricant. The ball valve is a traditional technical solution, based on a principle dating back to the start of the previous century, for example as disclosed in GB214922 from 1923.

An alternative and relatively new lubrication method, compared to traditional lubrication, is commercially called Swirl Injection Principle (SIP). It is based on injection of a spray of atomized droplets of lubricant into the scavenging air swirl inside the cylinder. The helically upwards directed swirl results in the lubricant being pulled towards the Top Dead Centre (TDC) of the cylinder and pressed outwards against the cylinder wall as a thin and even layer. This is explained in detail in international patent applications WO2010/149162 and WO2016/173601. The injectors comprise an injector housing inside which a reciprocating valve member is provided, typically a valve needle. The valve member, for example with a needle tip, closes and opens the lubricant's access to a nozzle aperture according to a precise timing. In current SIP systems, a spray with atomized droplets is achieved at a pressure of, typically, 35-40 bar. In comparison, the oil pressure is less than 30 bar and often less than 10 bar in systems working with compact oil jets that are introduced into the cylinder. In some types of SIP injectors, the high pressure of the lubricant is also used to move a spring-loaded valve member against the spring force away from the nozzle aperture such that the highly pressurised oil is released therefrom as atomized droplets. The ejection of oil leads to a lowering of the pressure of the oil on the valve member, resulting in the valve member returning to its origin and remaining there until the next lubricant cycle where highly pressurized lubricant is supplied to the lubricant injector again.

In such large marine engines, a number of injectors are arranged in the circumference of the cylinder, and each injector comprises one or more nozzle apertures for delivering lubricant jets or sprays into the cylinder from each injector. Examples of SIP lubricant injector systems in marine engines are disclosed in international patent applications WO2002/35068, WO2004/038189, WO2005/124112, WO2010/149162, WO2012/126480, WO2012/126473, WO2014/048438, and WO2016/173601.

Optimization of the spray in SIP lubrication is undergoing steady development. Although, lubrication injectors have some similarities with fuel injectors, comparison also shows different behaviour and different effects. This is mainly attributed to the different working conditions for the injectors, which leads to different effects such as viscosity, surface tension and liquid pressure. Accordingly, results from studies of fuel injection are not automatically transferable to lubricant injection, and the difference in behaviour is in some cases surprising.

For SIP injection, a precisely controlled timing is essential in addition to the objective of minimizing oil consumption. For this reason, SIP systems are specially designed for quick reactive response during injection-cycles.

A spray injection need not to be precisely controlled in the same extent as SIP injection. A spray injection need not to use the scavenging air swirl inside the cylinder for distributing the atomized droplets of lubricant on the cylinder liner.

With the introduction of the HJL Smartlube 4.0 system the performance of the lubrication systems is increased. however, there is a desire to obtain a method which in automatic way ensures that the system provides a desired amount of lubricant independent from external disturbances like change in lubricant type, pressure and temperature and mechanical wear of components in the lubrication system and which ensures that a large part of delivered lubricant oil is delivered in form of a spray.

The HJL Smartlube 4.0 system is a unique combination of compact design and advanced functionality. The system comprises one or two high pressure units for delivering the lubricant oil to the injectors. The high-pressure unit comprise a pump which through a common rail is connected with all injectors in the engine. A cylinder manifold connecting all injectors in a cylinder with the common rail. In the system each injector is controlled independently by control signals received from the controller.

One of the important factors for spray formation is cavitation in the nozzle, which is formation of vapour cavities inside the liquid due to evaporation. Cavitation in the nozzle influences the atomization of the liquid because it introduces pronounced disturbances in the liquid stream, which destabilizes the jet. In the field of fuel injection, such cavitation in the nozzle has been intensively studied, however, there are also a few studies on cavitation in lubricant injectors.

An extensive explanation regarding formation of vapour cavities is disclosed in WO 2018/215645 having the same applicant as the present application. The content of WO 2018/215645 is hereby incorporated by reference instead of repeating the explanation regarding the cavitation.

As far as the conclusions of these findings can be applied to lubricant injection, which behaves differently from fuel injection due to different viscosities, cavitation at the nozzle exit is disadvantageous. In other words, parameters, such as nozzle dimensions, as well as pressure and viscosity of the lubricant, should be chosen such that cavitation, especially at the nozzle exit, is avoided. This is also in agreement with current commercial SIP injection systems for marine engines, which are operated at parameters that do not cause cavitation in the nozzle.

As there is a steady motivation for improvement of lubrication in large two-stroke gas and diesel engines, for example marine engines or engines for power plants, cavitation design advantageously is part of the considerations for optimising the spray injection, in particular for SIP injection.

As it was described above, the prior art on lubricant injection draw the conclusion that cavitation, especially near the nozzle tip, is disadvantageous for the spray stability, distribution of lubrication oil, and leads to a less controlled spray. Accordingly, cavitation has not been used in practice for lubricant injection in large marine engines, neither for jet injection nor for spray injection, e.g. SIP injection. The parameters for spray injection, in particular SIP injection, have been outside the range where cavitation is achieved.

However, in contrast to the conclusions in the prior art and against the trend in the field, further detailed studies have surprisingly revealed that cavitation for lubricant in the nozzle can be used to provide stable, controlled injection of sprays, and uniform distribution of lubrication oil, which is a crucial factor for optimized spray injection, in particular SIP lubrication.

In studies leading to the invention disclosed in WO 2018/215645, not only cavitation as such has been found beneficial for spray formation of lubricants, but spray quality, control, and stability is even improved if the cavitation extends to the nozzle exit.

In further studies leading to the present invention is has been found beneficial to control the pressure of lubricant oil delivered through the common rail. It has shown that controlling the pressure in the common rail will affect how the lubrication oil droplets move in the scavenging air swirl of a marine two-stroke engine.

It has shown that the distribution of the oil droplets on the cylinder wall depends on the engine load.

When the load is low it has been shown that smaller droplets does not reach the surface of the cylinder wall. Moreover, is has been shown that when the load increases, the size of the small droplets that do not reach the cylinder wall also increases. This is believed to be due to increase of the air swirl speed and density. Therefore, more drops are "grabbed" and there is more droplet break-up, which captures even more oil in the air.

It has also shown that the common-rail pressure affects where the droplets end up on the cylinder wall.

It has shown important to control of the pressure of lubricant oil delivered through the common rail in order to influence on the size of droplets in the spray injected into the cylinder and accordingly also to influence on the distribution of the lubricant oil on the cylinder wall.

EP 0049603 A2 describes a system for providing lubricant to a low speed ignition machine. However, this document teaches the use of lubrication timed in order to be arranged between the piston rings. Moreover, the document only relates to control the amount of lubricant based on the power developed. There is no mention of controlling and varying the pressure depending on the load on the engine. Moreover, the document does not specify the use of a pressure in the range for SIP injection.

WO 2023/088526 describes a large combustion engine and a method described in the introductory paragraphs. It teaches that this system uses the lubricant with a pressure in the range for SIP injection. However, this document does not teach that a controller arranged for controlling and varying the pressure in the lubricant supply conduit.

The prior art documents do not disclose a method or a system for controlling the pressure in the common rail for affects where the droplets end up on the cylinder wall.

DESCRIPTION / SUMMARY OF THE INVENTION

It is the objective of the invention to provide an improvement in the prior art systems in order to obtain the desired effect in form of ensuring that a large part of the lubricant oil hits the surface of the cylinder wall by adjusting the pressure in the lubricant supply conduit which is preferably a common rail.

Especially, it is the objective to improve spray lubrication with spray injectors used in a common rail system in large combustion engines, for example in a large slow-running two-stroke engine. Further, the objective is to improve SIP lubrication with spray injectors in form of SIP injectors.

A further objective of the invention is to provide a system in which a variation in the pressure is used to control the degree of atomisation and thus the size of the lubricant droplets in the spray.

However, the method according to the invention may also be used in large four-stroke combustion engines, for example marine engines or combustion engines for power plants.

These objectives are achieved by a large combustion engine, for example slow-running two-stroke engine, described by way of introduction and as defined in the preamble of claim 1 and which large combustion engine is peculiar in that he controller is arranged for controlling and varying the pressure in the lubricant supply conduit within a desired pressure range in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar, optionally 30 to 80 bar.

These objectives are also achieved by a method according to the invention for lubricating a large combustion engine, described by way of introduction and as defined in the preamble of claim 7 and which method is peculiar in that the method comprises the step of:

with the controller varying and controlling the pressure in the lubricant supply conduit within a desired pressure range in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar, optionally 30 to 80 bar.

The lubricant supply conduit is preferably a common rail.

The lubricant supply is preferably a high-pressure unit. The high-pressure unit preferably comprises a pump.

The lubricating supply system is preferably the HJL Smartlube 4.0 system.

The invention may also be used in a lubricating principle in engines being modified in that multiple lubricant supply conduits are substituted by a single common lubricant supply conduit. Then the high-pressure unit feeds lubricant to the injectors by a "common rail" system in which all injectors of an engine cylinder, or a subgroup of injectors for a single engine cylinder, are receiving lubricant through the single common lubricant supply conduit in common and simultaneously.

Optionally, there is provided a return line for back flow of lubricant from the injectors.

The large two-stroke engine comprises a cylinder with a reciprocal piston inside and with a plurality of lubricant injectors distributed along a perimeter of the cylinder for injection of lubricant into the cylinder at various positions on the perimeter during injection phases. For example, the engine is a marine engine or a large engine on a power plant. Typically, the engine is burning fuel oil.

The term injection phase is used for the time during which lubricant is injected into the cylinder by the injector. The term injection cycle is used for the time it takes to inject lubricant by the injector into the cylinder and until the next injection. This terminology is in line with the above-mentioned prior art.

The term injector is herein used for a lubricant injection valve system comprising a housing with a lubricant inlet and one single injection nozzle with a nozzle exit from which the lubricant leaves the nozzle into the cylinder as a spray, the nozzle exit having an exit aperture with an exit size S. For example, the exit aperture is circular with a diameter D, in which case the diameter D is a measure for the size S. If the exit aperture deviates from a circular shape, a potential measure for the size S is the aperture area or an averaged diameter; the latter being useful in case of slight oval or elliptical deviation from a circle. For example, for a noncircular exit aperture, the cross-sectional dimension is an equivalent diameter calculated as twice the square root of the ratio between the cross-sectional area and the number Pi≈3.14. The nozzle has one or more, typically not more than two, nozzle exits.

In spray injectors, e.g. SIP injectors, the nozzle comprises a spray hole, formed as a channel with a length L, for example between 0.5 and 1 mm, one end of which forms the nozzle exit. In typical injectors, nozzle comprises a sac hole for flow of lubricant to the spray hole that extends from the sac hole to the nozzle exit. Typically, the central longitudinal axis of the spray hole has an angle with a central longitudinal axis of the sac hole, for example in the range of 30 to 90 degrees. The cross-sectional area of the sac hole perpendicular to its central longitudinal axis is often larger than the cross-sectional area of the spray hole perpendicular to its central longitudinal axis.

When varying the pressure, it is possible to control the degree of atomisation and thus the size of the lubricant droplets in the spray. This is of importance for controlling the mist of lubricant and where the lubricant droplets will hit the cylinder liner under different operational conditions of the engine.

Optionally, a controller is provided. The controller comprises a computer or is electronically or wirelessly connected to a computer. Advantageously, the computer is configured for monitoring parameters for the actual load and motion of the engine. In cooperation with the computer, on the basis of the parameters, the controller controls the amount and timing of the lubricant injection by the injectors during an injection phase. In advantageous embodiments, the controller is configured to control the lubricant pressure and optionally also the temperature of the lubricant in the common rail.

It is preferred that cavitation is established in the lubricant oil in the injector. In studies leading to the invention cavitation has been found beneficial for spray formation of lubricants. It has surprisingly been found that it is possible to improve the distribution of droplets in the spray for lubricating the cylinder wall.

For a spray to develop the primary pressure drop must be over the very last hole in the nozzle. This can be explained by the orifice equation [1] as shown in (1) below:

$$Q = C_d A_0 Y \sqrt{\frac{2\Delta P}{\rho(1-\beta^4)}} \qquad (1)$$

In (1) Q is the volume flow, C_d is the discharge coefficient, A_0 is the area of the restriction, Y is the expansion coefficient (=1 for incompressible fluids, which is a fair assumption for cylinder lubrication oil) [Ravendran R, Jensen P, De Claville Christiansen J et al. Rheological behavior of lubrication oils used in two-stroke marine engines. Industrial Lubrication and Tribology 2017; 69(5): 750–753. DOI:10.1108/ILT-03-2016-0075], ΔP is the pressure difference over the restriction, ρ is the density of the fluid and β is the area ratio, which is the ratio of the diameter of the restriction to the diameter of the pipe $\beta = D2/D1$.

For the oil to cavitate, the cavitation number must be close to or below 1. The cavitation number is shown in (2).

$$\sigma = \frac{P_a - P_v}{\frac{1}{2}\rho V^2} \qquad (2)$$

In (2) P_a is the ambient pressure (pressure of the cylinder when injection happens), P_v is the vapour pressure of the fluid, ρ is the density of the fluid and V is the velocity of the fluid. Most of the variables in (2) are constants, P_a Changes a bit with the load of the engine, ρ changes a bit with the temperature, so since these influences are quite small, the largest influence on the cavitation number is the velocity of the oil. Remembering that $Q = A \cdot V$ it can be seen from (1) that the cavitation number must decrease with increasing ΔP.

Below is explained numerical simulation effected to document the effect obtained according to the present invention.

When the lubrication oil is injected into the scavenging air of the two-stroke marine engine the droplets are affected by scavenging air. The way these droplets move is governed by the Stokes number shown in (3).

$$Stk = \frac{t \cdot \boldsymbol{U}}{l} \qquad (3)$$

In (3), t is the time constant in the exponential decay of the velocity of the particle due to drag, $\boldsymbol{U}$ is the velocity of the particle and l is the characteristic length of the shape the particle has, which for a sphere would be the diameter. From the Stokes number it follows that a particle with a low Stokes number follows the motion of the fluid well. In the case of oil droplets in the scavenging air it would mean the oil droplets follow the motion of the scavenging air well. A particle with a large Stokes number is dominated by its own inertia and follows along its initial trajectory without being affected much by the scavenging air flow.

How much the oil droplets are affected by the scavenging air flow will influence their trajectory and ultimately where they end up in cylinder.

Since the time constant of the Stokes number cannot easily be determined analytically the droplets are simulated in the scavenging air to determine how much they are affected by the scavenging flow.

The present invention is based on a simulation-based investigation of how the cylinder oil feed rate can be reduced. The model development is based on the only common-rail cylinder lubrication system for large two-stroke marine diesel engines explained in "N. Kristensen, 2019, *"HJ Common Rail Lubrication System*", CIMAC Congress".

It uses solenoid actuated valves to atomise the lubrication oil. The injectors are operated on the swirl injection principle, for example as explained in S. Lauritsen, J. Dragsted, and B. Buchholz, 2001, "Swirl Injection Lubrication - a New Technology To Obtain Low Cylinder Oil Consumption Without Sacrificing Wear Rates,", CIMAC congress pp. 921–932, where oil is injected in an upwards direction along the cylinder liner before the piston passes in the compression stroke. The aim for the current study is therefore to develop and validate a model for investigating the swirl injection principle.

To demonstrate the simulation-based investigation, a numerical framework is required, which is explained below.

The simulation performed is a computational fluid dynamics model. The governing equations of the problem are continuity (4), momentum (5), energy (6) and species (7):

$$0 = \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \boldsymbol{U}) \tag{4}$$

$$0 = \frac{\partial (\rho \boldsymbol{U})}{\partial t} + \nabla \cdot \big(\boldsymbol{U}(\rho \boldsymbol{U})\big) + \nabla p + \nabla \cdot \tau - \rho g \tag{5}$$

$$0 = \frac{\partial \rho h}{\partial t} + \nabla \cdot (\rho \boldsymbol{U} h) + \frac{\partial \mathrm{p}}{\partial t} + \nabla \cdot q - \rho g \cdot \boldsymbol{U} \tag{6}$$

$$0 = \frac{\partial \rho Y_i}{\partial t} + \nabla \cdot (\rho \boldsymbol{U} Y_i) - \nabla \mu_{\mathrm{eff}} \nabla Y_i \tag{7}$$

where

$$\tau = \nu \left(\nabla \boldsymbol{U} + (\nabla \boldsymbol{U})^T - \frac{2}{3} (\nabla \cdot \boldsymbol{U}) I \right) \tag{8}$$

In Equation 3-7 ρ is density, $\boldsymbol{U}$ is the velocity vector, p is pressure, g is the acceleration of gravity, h is enthalpy, q is heat flux, Y_i is species mass fraction, μ_{eff} is the effective dynamic viscosity, ν is the kinematic viscosity I is the identity matrix and τ is the shear stress tensor. The turbulence model employed is the $k - \epsilon$ model disclosed in "Launder B and Spalding D. The numerical computation of turbulent flows. Computer Methods in Applied Mechanics and Engineering 1974; 3(2): 269–289. DOI:10.1016/0045-7825(74)90029-2" governed by the following equations:

$$\frac{\partial (\rho k)}{\partial t} = \nabla \cdot (\rho D_{\mathrm{k}} \nabla k) + P - \rho \epsilon \tag{9}$$

$$\frac{\partial (\rho \epsilon)}{\partial t} = \nabla \cdot (\rho D_{\epsilon} \nabla \epsilon) + \frac{C_1 \epsilon}{k} \left(P + C_3 \frac{2}{3} k \nabla \cdot \boldsymbol{U} \right) - C_2 \rho \frac{\epsilon^2}{k} \tag{10}$$

where k is the turbulent kinetic energy, ϵ is the turbulent kinetic energy dissipation rate, P is the turbulent kinetic energy production rate, D is the effective diffusivity and C are different model constants. The simulations are performed as Unsteady Reynolds Averaged Navier-Stokes (URANS) with a maximum Courant–Friedrichs–Lewy (CFL) number of less than one.

The system of equations is solved to obtain the scavenging flow, which are then compared with result from literature to show the accuracy. This is illustrated in Fig. 3 and Fig. 9

Both the experimental data, and the comparison to literature from the two figures above are from a study: ”Nemati A, Cai J, Vincent M et al. Numerical Study of the Scavenging Process in a Large Two-Stroke Marine Engine Using URANS and LES Turbulence Models. SAE International 2020; 1– DOI:10.4271/2020-01-2012“. As can be seen in the figures 3 and 9 the accuracy of the scavenging flow should be good enough to determine the scavenging airs influence on the particle trajectories.

Particles are injected into the scavenging flow in the simulation and the way they move are determined by the Newton's second law.

$$F = m \cdot a \qquad (11)$$

Where F is the sum of forces acting on the particle, m is the mass of the particle and a is the acceleration. (11) can be described in greater detail as

$$m_p * \frac{d\boldsymbol{U_p}}{dt} = F_{drag} + F_{buoyancy} + F_{others} \qquad (12)$$

Where subscript p refers to the particle, $\boldsymbol{U}$ is the velocity, t is time, m is mass and F is force.

The buoyancy force is calculated as shown in (13)

$$F_{buoyancy} = (\rho_p - \rho_f)gV_p \qquad (13)$$

Where ρ_p is the density of the particle, ρ_f is the density of the scavenging air, g is the gravitational acceleration and V_p is the volume of the particle. The drag can be calculated as shown in (14).

$$F_{drag} = \frac{1}{2} C_D \rho_f |\boldsymbol{U} - \boldsymbol{U}_p| (\boldsymbol{U} - \boldsymbol{U}_p) A_p \qquad (14)$$

Here C_D is the drag coefficient, U is the velocity of the flow, U_p is the velocity of the particle and A_p is the cross-sectional area of the particle in the direction of the flow. Equation (12) is solved for each time step in the simulation and for every particle and thus having a small CFL number will give a good approximation of the exponential decay of particle velocity due to drag.

The initial velocity of the particles when they enter the cylinder was determined experimentally using the Bosch rate of injection method: “Bosch W. The fuel rate indicator: A new measuring instrument for display of the characteristics of individual injection. SAE Technical Papers 1966; DOI:10.4271/660749” A result of those measurements is shown in Fig. 10.

As seen from Fig. 10 the mass flow does not instantaneously rise when the valve is opened. This is mainly due to the speed that the needle moves with. Because the mass flow does not instantaneously rise, the first oil that is injected, is not injected as a spray.

The velocity of the particles be calculated from the mass flow by using $A \cdot U \cdot \rho = \dot{m}$. The simulations are performed with two different initial droplet size distributions shown in Fig. 11 and Fig. 12.

The two distributions have quite a different shape, but the key difference is how many small droplets they each contain.

The one in Fig. 12 has many very small drops, and the one in Fig. 11 is normally distributed, but with a median of approx. 150 µm, which is significantly higher than the one in Fig. 12.

The two droplet size distributions are used as input in a simulation model of the scavenging air in a cylinder. The simulation model can predict how large a proportion of the drops hit the wall of the cylinder.

If the distribution in Fig. 12 is used, approx. 42% of the mass injected reaches the cylinder wall and for the distribution in Fig. 11 it is about 99.5% of the mass injected that reaches the cylinder wall. Thus, droplet size distribution has a large influence on how much of the oil hits the wall.
In connection with the development of mathematical models for the oil spray in the scavenging air, it has emerged that the smallest droplets in the spray do not hit the cylinder wall at all when the load is low in the simulated engine. This also applies to high load, but the limit of which drops that reaches the cylinder wall moves with load. This

is illustrated in Fig. 13 and 14 where Fig. 13 is a distribution and where the result of using this distribution is shown in Figure 14.

How small the drops must be in order not to hit the cylinder wall is observed to change when the load changes. The actual distribution of the oil droplets that are formed in the engine depends on the scavenging air, but to a large extent also on the pressure in the nozzles. An example of a distribution of oil droplets is shown in Fig. 13. In Fig 13 there is an illustration of the distribution of droplets and Fig. 14 is an illustration of the of the droplets which do not reach the cylinder wall. The blue particles illustrate droplets in the scavenging air, and which do not reach the cylinder wall.

It is a general knowledge that the droplet size is inversely proportional to the pressure behind the needle in the injector. So, it will be advantageous to vary the pressure in the common rail when there is a change in the load on the engine and thereby achieve greater adhesion of the droplets to the cylinder wall.

The difference between Fig. 11 and Fig. 12 distribution is to illustrate through simulation how the small droplets follow the streamlines closely and therefore never end up on the wall. Where the larger droplets that initially have a trajectory towards the cylinder wall are affected, but not to a degree where they never hit the wall.

This shows that for the spray technology, in particular SIP technology to work efficiently, the droplets should not have a stokes number that is low.

To affect the Stokes number of the droplets initially the inertia of the droplets must be affected or the flow in the cylinder must be changed. The flow in the cylinder changes with the change in load of the engine both in terms of velocity and in terms of density both present in (14).

As demonstrated by a study: "Chaker, Mustapha. (2007). Key Parameters for the Performance of Impaction-Pin Nozzles Used in Inlet Fogging of Gas Turbine Engines. Journal of Engineering for Gas Turbines and Power-transactions of The Asme - J ENG GAS TURB POWER-T ASME. 129. 10.1115/1.2364006" the injection pressure is inversely proportional to droplet size of the injected fluid, which is also true for the velocity of the bulk fluid.

The injection pressure can be used to affect the initial Stokes number of the droplets and as the simulations show.

Thus, the experiments and studies show that controlling the injection pressure can be an aid in optimising the efficiency of the system as the load of the engine changes.

In practise a method according to the invention is used for controlling the common-rail pressure in for the given parameters.

Parameter	Max	Min
Common-rail pressure	300 bar	20 bar
Engine load	100 %	0 %

For the injector to be able to quickly operate for the intended operation of the injector, it is important that the injector can be operated in the given parameter span.

The method according to the invention involves lubricating a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a system comprising

- a lubricant supply comprising a high-pressure pump,
- a plurality of lubricant injectors distributed along a perimeter of the cylinder for injection of lubricant into the cylinder at various positions on the perimeter during injection phases,
- a lubricant supply conduit connecting the lubricant supply with the lubricant injectors,

the engine further comprising

- a controller for controlling the amount and timing of the lubricant injection by at least one of the lubricant injectors,
- a computer or a mobile phone to which the controller is connected,

wherein each injector comprises

- an inlet port flow-connected to the lubricant supply conduit for receiving lubricant from it,

- a nozzle with a nozzle aperture extending into the cylinder configured for injecting lubricant from the inlet port into the cylinder in the injection phase,
- an adjustable valve at the nozzle for opening and closing for flow of lubricant to the nozzle aperture during an injection-cycle,

wherein the method comprises in cyclic operation
- in the injection phase, providing pressure-liquid at inlet port of the injector and exerting force on the valve and by the force driving a valve body into motion inside the injector and upon pressure rise above the predetermined limit, pumping a predetermined lubricant volume through the nozzle aperture into the cylinder,
- providing at least one pressure gauge for measuring the pressure in the lubricant supply conduit,
- measuring with the at least one pressure gauge the pressure in the lubricant supply conduit,
- transforming in the controller the measured pressure into a control signal forwarded to the lubricant supply in order to obtain a desired pressure in the lubricant supply conduit taking into account the actual engine load.

Alternatively, in the cyclic operation the method may comprise
- in the injection phase, providing pressure-liquid at inlet port of the injector and exerting force on the valve and by the force driving a valve body into motion inside the injector and upon pressure rise above the predetermined limit, pumping a predetermined lubricant volume through the nozzle aperture into the cylinder,
- after the injection phase, retracting the valve body by draining pressure-liquid from the injector,
- during retraction re-filling a pressure chamber in the injector with lubricant for a subsequent injection phase.

The control signal to the lubricant supply may be used
- for regulating the output of high-pressure pump and thereby obtaining the desired pressure in the lubricant supply conduit. In this situation it is the rotation speed of the pump which is regulated.

or
- for regulating a valve arranged at the high-pressure pump and connecting the high-pressure pump with the lubricant supply conduit. In this situation the output of high-

pressure pump is kept constant, and the regulation of the valve is used for regulating the pressure in the lubricant supply conduit to the desired pressure. Preferably the valve is connected to the pressure side and suction side of the high-pressure pump. Thus the valve may relieve the pressure as it is connection to the suction side of the high-pressure pump or to a return line to the lubricant supply.

With the present invention the control of the pressure in the lubricant supply conduit makes is possible to obtain the desired effect in form of ensuring that a large part of the lubricant oil hits the surface of the cylinder wall by adjusting the pressure in the common rail in order to influence on the size of the droplets.

The controller controls and adjust the pressure in the lubricant supply conduit based on measurement of engine load and actual pressure in the lubricant supply conduit.

Moreover, the controller may control the adjustable valve and regulates the stroke length of a valve member in form of a plunger, whereby the lubricant amount is variably adjustable between injection phases by a stroke length adjustment mechanism. When the plunger is not fully retracted to the maximum possible retraction position adjustment is obtained. As the stroke of the plunger is always to the same position, for example the same most-forward position, varying the retraction position of the plunger after retraction regulates the stroke length.

The controller may be built into the injector or may be connected to the injector and the flowmeter by wiring or in a wireless way.

The controller may also be arranged for controlling the lubricant type used.

In some embodiments, the maximum possible retraction position of the plunger is the most rearward possible position at maximum distance from the nozzle aperture but it is possible that the plunger is held at a distance from the most rearward possible position. By regulating the distance, the injection amount for the next injection is regulated, as the stroke length is reduced relatively to the maximum possible retraction position. The effect is similar to the screw-adjustable end-stop in WO02/35068, however, the stroke length adjustment mechanism can be provided centrally and remotely from the injector, which is in contrast to the injector of WO02/35068.

The engine according to the present invention may comprise a hydraulically driven inlet-valve system wherein each injector comprises

- a lubricant inlet port flow-connected to the lubricant supply conduit for receiving lubricant from it;
- a nozzle with a nozzle aperture extending into the cylinder configured for injecting lubricant from the inlet port into the cylinder in the injection phase;
- an outlet-valve system at the nozzle for opening and closing for flow of lubricant to the nozzle aperture during an injection-cycle;
- a front chamber inside the injector between the lubricant inlet port and the outlet-valve system for receiving and accumulating a predetermined volume of lubricant from the inlet port prior to an injection phase;
- wherein the outlet-valve system is configured for opening for flow of lubricant from the front chamber through the outlet-valve system and to the nozzle aperture during an injection phase upon pressure rise above a predetermined pressure limit in the front chamber and at the outlet-valve system and configured for closing the outlet-valve system after the injection phase;
- a pressure-control port flow-connected to the pressure-control conduit for receiving pressure-liquid therefrom in the injection phase;
- a pressure chamber in communication with the pressure-control port for periodically receiving the pressure-liquid from the pressure-control port in the injection phase and for drain thereof after the injection phase;
- a reciprocal hydraulic-driven actuator-plunger in contact with the pressure chamber and pre-stressed by a spring-load from an actuator-plunger spring and being configured for motion that is driven by the pressure-liquid in the pressure chamber in the injection phase and configured for causing the pressure rise in the lubricant in the front chamber above the predetermined pressure limit by its motion in the injection phase;

wherein the engine further comprises a stroke length adjustment mechanism for variably adjusting the stroke length of the reciprocal hydraulic-driven actuator-plunger. In such embodiment the stroke length adjustment mechanism is configured for variable adjustment of the amount of pressure-liquid that is drained from the pressure chamber between injection phases.

The effect of the variable adjustability is similar to the screw-adjustable end-stop in WO02/35068, however, the stroke length adjustment mechanism can be provided centrally and remotely from the injector, which is in contrast to the injector of WO02/35068.

In some practical embodiments, the stroke length adjustment mechanism comprises a pressure regulator for variably regulating an idle-pressure in the pressure chamber between the injection phases, wherein the idle-pressure is lower than the predetermined limit for only partly counteracting the spring-load from the actuator-plunger spring on the actuator-plunger and thereby variably adjusting the retracted position for the actuator-plunger.

The outlet-valve system closes off for back-pressure from the cylinder and also prevents lubricant to enter the cylinder unless the outlet-valve is open. In addition, the outlet-valve system assists in a short closing time after injection, adding to precision in timing and volume of injected lubricant.

The engine comprising a hydraulically driven inlet-valve system may be operated according to a method comprising the stroke length adjustment mechanism is configured for variable adjustment of the amount of pressure-liquid that is drained from the pressure chamber between injection phases and wherein the method comprises adjusting the stroke length between injection cycles by adjusting the amount of pressure-liquid that is drained from the pressure chamber after the injection phase.

The engine and the method comprising the hydraulically driven inlet-valve system known from WO 2019/114905. However, the engine differs as it comprises a pressure gauge. The pressure gauge shall be used for measuring the pressure in the lubricant supply conduit.

The engine according to the present invention may also comprise an electrically driven inlet-valve system wherein each_injector comprises

- a lubricant inlet port for receiving lubricant from a lubricant supply conduit;
- a nozzle with a nozzle aperture extending into the cylinder for injecting lubricant from the inlet port into a cylinder;

- an outlet-valve system at the nozzle for opening and closing for flow of lubricant to the nozzle aperture during an injection cycle; the outlet-valve system being configured for opening for flow of lubricant to the nozzle aperture during an injection phase upon pressure rise above a predetermined limit at the outlet-valve system and for closing the outlet-valve system after the injection phase. In such embodiment each injector may comprise an electrically-driven inlet-valve system electrically connected to the controller and arranged between the lubricant inlet port and the nozzle for regulating the lubricant that is dispensed through the nozzle aperture by opening or closing for lubricant flow from the lubricant inlet port to the nozzle in dependence of an electrical control-signal received from the controller; wherein the inlet-valve system is arranged upstream of and remotely from the nozzle and upstream of and remotely from the outlet-valve system.

According to a specific embodiment of this type the inlet-valve system may comprise an inlet non-return valve with an inlet-valve member pre-stressed against an inlet-valve seat by an inlet-valve spring and arranged for throughput of lubricant from the lubricant inlet port to the outlet-valve system upon displacement of the inlet-valve member from the inlet-valve seat against force from the inlet-valve spring; wherein the inlet-valve system further comprises an electrically-driven rigid displacement-member for displacing the inlet-valve member from the inlet-valve seat during lubricant injection.

In order to provide a better speed and volume control of lubricant ejection by the injector, each of the injectors comprises an electrically-driven inlet-valve system electrically connected to the controller and arranged between the lubricant inlet port and the nozzle for regulating the lubricant that is dispensed through the nozzle aperture by opening or closing for lubricant flow from the lubricant inlet port to the nozzle in dependence of an electrical control-signal received from the controller. The inlet-valve system is arranged upstream of and remotely from the nozzle, and it is also arranged upstream of and remotely from the outlet-valve system.

The inlet-valve system of the injector doses the amount of lubricant for injection by the time the inlet-valve system stays open for the injection phase. The time is determined by the controller.

The engine comprising an electrically-driven inlet-valve system may be operated according to a method comprising sending an electrical control signal from the controller to the electrically-driven inlet-valve system for starting an injection phase, and as a consequence thereof causing flow of lubricant from the lubricant supply conduit through the lubricant inlet port, through the inlet-valve system, and into a conduit that flow-connects the inlet-valve system with the outlet-valve system, by the lubricant flow into the conduit increasing pressure in the conduit and at the outlet-valve system, by the pressure rise causing the outlet-valve system to open for flow of lubricant from the conduit to the nozzle aperture and injecting lubricant into the cylinder through the nozzle aperture; at the end of the injection phase, changing the electrical control signal from the controller to the inlet valve system and causing the inlet-valve system to close for lubricant supply from the lubricant inlet port to the conduit.

The engine and the method comprising the electrically driven inlet-valve system is generally known from WO 2019/114903.

According to a further embodiment the engine is peculiar in that the lubricant system is chosen from mechanically driven systems, hydraulically driven systems and common-rail systems.

The principle according to the invention is flexible and may be used in different lubrication systems.

According to a further embodiment the engine is peculiar in that the pressure gauge is provided in the lubricant supply conduit connected to the injector.

Hereby the pressure in the actual flow into the injector is measured. The pressure gauge may be arranged immediately in front of the inlet port of a single injector or may alternatively be arranged in a common supply conduit for more injectors, e.g., injectors in a single cylinder.

In some embodiments the method comprises that the regulation of the amount of lubricant is controlled by a feedback control/regulation, for example a PID regulation or a more sophisticated model-based regulation.

In some embodiments the method comprises that the method involves storing regulation values in a database in the controller.

The method and the engine according to the invention is especially suitable for use for spray injection, in particular SIP injection into the cylinder of a large marine engine or combustion engine for a power plant at a lubricant pressure in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar. The method is also suitable for lubrication with a combination of spray lubrication, in particular SIP injection and injection into the ring pack. The injection into the ring pack may be a lubricant spray or a compact lubricant jet.

Especially the method and the engine according to the invention is especially suitable for use for SIP injection with an injector of a type generally described in WO 2012/126473 and which is also known as Hans Jensen Lubricators E-sip injector.

Definitions

The term “regulate” refers to a situation where the lubricant amount is amended in order to correspond to a desired amount when the engine is in service.

The term “adjust” refers to the situation where the lubricant amount is amended when the injectors are calibrated.

The term “injector” is used for an injection valve system comprising a housing with a lubricant inlet and one single or more injection nozzles with a nozzle aperture as a lubricant outlet and with a movable valve member inside the housing, which opens and closes access for the lubricant to the nozzle aperture. Although, the injector has a single nozzle that extends into the cylinder through the cylinder wall, when the injector is properly mounted, the nozzle itself, optionally, has more than a single aperture. For example, nozzles with multiple apertures are disclosed in WO2012/126480.

The term “injection-phase” is used for the time during which lubricant is injected into the cylinder by an injector.

The term “idle-phase” is used for the time between injection-phases.

The term “idle state” is used for the state of a component in the idle-phase.

The term “idle-phase position or orientation” is used for the position or orientation of a movable component when in the idle state during the idle-phase, which is in contrast to an injection-phase position.

The term “injection cycle” is used for the time it takes to start an injection sequence and until the next injection sequence starts. For example, the injection sequence comprises a single injection, in which case the injection cycle is measured from the start of the injection-phase to the start of the next injection-phase. Alternatively, the injection sequence comprises multiple injections, for example multiple injections above the piston before the piston passes the injectors on its way to the TDC, for example a first injection with one lubricant followed by another injection of another lubricant, and potentially further lubricants and/or additives. Such double or multiple injections leads to oil mixing in the cylinder before the piston reaches the TDC. For example, there is one injection cycle for each revolution of the engine. However, it is also possible to have one injection cycle after a number of engine revolutions.

The term “timing” of the injection is used for the adjustment of the start of the injection-phase by the injector relatively to a specific position of the piston inside the cylinder.

The term “frequency” of the injection is used for the number of repeated injections by an injector per revolution of the engine. If the frequency is unity, there is one injection per revolution. If the frequency is 1/2, there is one injection per every two revolutions. This terminology is in line with the above-mentioned prior art.

The term “pressurized lubricant” is used for lubricant provided at a pressure high enough that it can be used for injection as a jet or spray into the cylinder. The latter is in contrast to oil injection by quills between piston rings. The pressure depends on the purpose and form of injection and is typically above 10 bar. For spray injection, in particular SIP injection, the pressure is typically higher, for example above 25 bar.

The term “flowmeter” is used for a component which is able to measure a flow independent of the method used, e.g. pressure difference, viscosity, temperature, amount.

Practical embodiments

The large engine, for example slow-running two-stroke engine, optionally a marine engine or engine for a power plant, comprises a cylinder with a reciprocal piston inside and with a plurality of lubricant injectors fixed to a wall of the cylinder and extending through the cylinder wall. The injectors are distributed along a perimeter of the cylinder and configured for injection of lubricant into the cylinder at various positions on the perimeter during injection-phases. For example, the large engine, such as slow-running two-stroke engine, is a marine engine or a large engine in power plants. Typically, the engine is burning diesel or gas fuel, for example natural gas fuel.

The engine comprises also a lubricant supply with a pressurized lubricant, typically pressurized by a lubricant feed pump. Optionally, the engine comprises more than one lubricant supply with correspondingly more than one type of lubricant, and correspondingly more than one lubricant feed pump.

Each of the plurality of injectors is connected with each of its lubricant inlets to lubricant supply through a corresponding lubricant supply conduit. Each lubricant supply includes a potential pressure source, typically a lubricant pump, which raises the pressure of the corresponding lubricant to an adequate level. For the described system, it suffices to provide a constant lubricant pressure at the corresponding lubricant inlet of the injector.

The injectors are configured for the type of lubricant to be injected. The inlet can be used to provide and add not only lubricants but also potential additives. For example, the injector optionally has more inlets of which one is used for lubricants, such as lubrication oils, and one is used for an additive.

The engine further comprises a controller. The controller is configured for controlling the amount and timing of the injection of the lubricant by the plurality of injectors. Optionally, also the injection frequency is controlled by the controller. For precise injection, it is an advantage if the controller is electronically connected to a computer or comprises a computer, where the computer is monitoring parameters for the actual load and motion of the engine. Such parameters are useful for the control of optimized injection.

Optionally, the controller is provided as an add-on system for upgrade of already existing engines. A further advantageous option is a connection of the controller to a Human Machine Interface (HMI) which comprises a display for surveillance and input panel for adjustment and/or programming of parameters for injection profiles and optionally the state of the engine. Electronic data connections are optionally wired or wireless or a combination thereof.

In a concrete embodiment, the injector comprises a lubricant inlet for receiving the lubricant from a lubricant supply conduit for injection of the lubricant into the cylinder. The lubricant inlet of the injector is connected to the lubricant supply though the lubricant supply conduit.

The injector has a lubricant flow path from the lubricant inlet to the at least one nozzle for lubricant flow from the lubricant inlet through the at least one nozzle into the cylinder.

The injector comprises one nozzle or more than one nozzle, for example two nozzles. Each nozzle has a nozzle aperture, extending into the cylinder for lubricant injection in an injection-phase. Optionally, a nozzle has more than a single aperture. For example, nozzles with multiple apertures are disclosed in WO2012/126480. In some embodiments, the injector comprises a single nozzle with a single nozzle aperture.

In particular, each injector comprises an internal actuator-driven valve system in the lubricant flow path, wherein the valve system is configured for selectively switching from the idle state without injection to an injection state with injection of the lubricant, into the cylinder through the at least one nozzle in the injection-phase in dependence of the received injection-phase signals.

Each injector comprises an actuator for driving the valve system. The actuator is functionally connected to the controller and configured for being activated by the controller for selectively driving the valve system and causing injection of the lubricant under control by the controller as a consequence of the activation of the actuator by the controller. The valve system, under control by the controller, is used for selecting which amount and timing to be used for injections and in which sequence.

In operation, the actuator is activated by the controller for starting an injection-phase with the lubricant. As a consequence thereof, the valve system is caused to open for flow of the lubricant through the flow path and injecting the lubricant into the cylinder. At the end of the injection-phase, the actuator is caused to close the valve system and stop lubricant supply.

Example with pressure control

In a specific embodiment the engine comprises a lubricant supply conduit containing lubricant at a desired pressure and being connected to the abovementioned E sip injectors.

The injector is provided with integrated opening/closing valve, preferably a solenoid valve, such that both piping and drawing of cables are appreciably simplified by only having one common supply line of pressurised lubricating oil (without need for a return line) provides that the dosing becomes proportional with the time where the opening/closing solenoid valve is open. Preferably, there is a separate local control box which is used for opening/closing the injector based on signals from the engine/control unit of the ship.

An electromechanically regulated injector designed for cylinder lubrication of large diesel engines entail advantages in relation to prior art lubricating systems. System wise it can regulate individually with regard to lubricating oil amount and timing.

The function is only dependent on a control box which can control each single injector separately or together with regard to timing and opening time. This can occur independently of other opening/closing valves and is only limited by the speed at which the opening/closing valve in an injector can execute the opening/closing cycle.

The measured flow is used for controlling the delivered amount in relation to the planned amount. By deviations of a given size for a given period of time, an associated local control box can correct the opening time for magnet valve(s) for the associated injector or injectors.

In another embodiment the engine comprises a lubricant supply conduit containing lubricant at a first pressure and a pressure-control conduit containing pressure-liquid at a pressure higher than the first pressure. In such case the injector comprises an internal hydraulic-driven pumping system where the pressure-liquid is used for driving the pumping system inside the injector housing by which the lubricant is pressurized in the injector and ejected therefrom. The injector comprises a lubricant inlet port, flow-connected to the lubricant supply conduit for receiving lubricant from it for injection into the cylinder. The injector also comprises a pressure-control port, flow-connected to the pressure-control conduit for receiving pressure-liquid therefrom in the injection phase.

The injector comprises the front chamber inside the injector between the lubricant inlet port and the outlet-valve system for receiving and accumulating a pre-determined volume of lubricant from the inlet port prior to an injection phase.

A pressure chamber in the injector is in communication with the pressure-control port for receiving the pressure-liquid from the pressure-control port in the injection phase. The pressure-liquid in the pressure chamber drives a pumping system in the injector. The pumping system comprises a reciprocal hydraulic-driven actuator-plunger in contact with the pressure chamber and pre-stressed by a spring-load from an actuator-plunger spring and configured for being driven, for example in a direction towards the nozzle, by the pressure-liquid in the pressure chamber in the injection phase, by which it is causing pressure rise in the lubricant in the front chamber above the predetermined limit and causes pumping of this predetermined lubricant volume through the non-return valve and the nozzle aperture into the cylinder.

Injection

Optionally, the injection phase comprises multiple injections, for example multiple injections above the piston before the piston passes the injectors on its way to the TDC, for example a first injection with lubricant followed by another injection of the lubricant, and potentially further lubricants and/or additives. Such double or multiple injections, especially when in SIP operation, leads to oil mixing in the cylinder before the piston reaches the TDC. The variety of selecting lubricant for injection, its amount and its timing as controlled by the controller makes a high variety of injection sequences possible, for example combinations of at least two of:

- one or more injections under the piston,
- one or more injections onto the piston,
- one or more injections above the piston during a single injection cycle.

For the various injections, also the selections for the lubricant or lubricants, potentially with additives, can be varied.

For example, the actuator is an electrically controlled actuator and is electrically connected to the controller by an electrical connection for receiving injection-phase signals from the controller, the injection-phase signals indicating the timing for the injection. For the injection-phase, an electrical control signal is sent from the controller to each of the injectors for starting an injection-phase with the lubricant. As a consequence thereof, the valve system opens for flow of the corresponding lubricant through the flow path and injects it into the cylinder. At the end of the injection-phase, the electrical control signal from the controller to the injector is changed, causing the valve system to close for lubricant injection and return to the idle state.

In a specific embodiment, the actuator comprises an electrical solenoid arrangement with a stationary solenoid part and a movable solenoid part. The valve system is connected to the movable solenoid part for being driven by the actuator upon electrical excitation of the solenoid, wherein the solenoid is configured for excitation by the injection-phase signals from the controller.

The term “a solenoid coil” should be understood as “at least one solenoid coil”, as it is possible and, in some cases, advantageous to use more than one coil, for example two or three coils.

The term “signal” from the controller is used here for an electrical current that flows from the controller to the injector. In some embodiments, the signal itself can be used for driving the actuator, for example an electromechanical actuator, if the current is sufficiently strong. For example, for switching the driving direction of an electromechanical actuator, the direction of the current is switched to an opposite direction. However, alternatively, the injector could comprise an electro-switch where the signal from the controller opens for flow of a current sufficiently strong to drive the actuator. In the latter case, the signal lines from the controller to the electro-switch can be accomplished by very thin wiring. Alternatively, the term “signal” also is used for a wireless signal.

Alternatively, the actuator is a hydraulic or pneumatic actuator. Such hydraulic or pneumatic actuator in the injector is, optionally, also electrically controlled. For example, an electrical signal from the controller to the injector causes an electromechanical actuator-valve of the injector to open for hydraulic or pneumatic flow into the actuator for driving the valve system hydraulically or pneumatically. Optionally, an electrical signal from the controller to the injector causes an electromechanical actuator-valve to open for hydraulic or pneumatic flow into the actuator for driving the actuator itself, which then, in turn, by a mechanical connection drives the valve system.

In some embodiments, the valve system is configured to select only one injector at a time among multiple injectors for supply of lubricant and for injection thereof. In some embodiment, alternatively or in addition, the valve system is configured to select more than one injector at a time among multiple injectors for supply of lubricant and for injection thereof in order to inject simultaneously multiple lubricants or lubricant in combination with additive.

In some embodiments, the injector has more than one nozzle, and multiple lubricants and additives can be injected into the cylinder through separate nozzles of the injector. In other embodiments, multiple lubricants and additives are injected into the cylinder through a single nozzle and potentially mixed inside the injector prior to ejection from the nozzle aperture.

In practical embodiments, the injector comprises a base and a rigid, optionally cylindrical, flow chamber, which is rigidly connecting the base with the nozzle for fixing the nozzle inside the cylinder wall when the base is fixed to the cylinder wall. Due to the base being provided at the opposite end of the flow chamber relatively to the nozzle, it is typically located on or at the outer side of the cylinder wall. For example, the injector comprises a flange at the base for mounting onto the outer cylinder wall. Alternatively, in order to mount the injector in the cylinder wall, the injector comprises a flange provided around the flow chamber. For example, the flange is bolted against the cylinder wall.

Advantageously, the base comprises the first and second inlet and the potential further inlets. The flow chamber is hollow and contains the flow path for lubricant flow from the lubricant inlet through the flow chamber and to the nozzle for injection of the lubricant into the cylinder. Optionally, the valve member is located in the flow chamber or in the base.

For example, the actuator is provided outside the cylinder wall when the injector is mounted at the cylinder wall. Optionally, it is fixed to the base.

In practice, the injection-phase signal is received by the actuator, causing the actuator to move the valve member in dependence of the injection-phase signal to an injection-phase position or orientation, which leads correspondingly to opening of the flow path for injection of the lubricant into the cylinder.

For example, the actuator is mechanically connected to the selection-valve member by an actuator extension for driving the valve member by the actuator extension. This is advantageous if the valve member is located in the flow chamber and, thus, inside the cylinder wall, whereas the actuator is located outside the cylinder wall. In this embodiment, the operation comprises moving the valve member by the actuator by using the actuator extension.

Outlet-valve system

Optionally, each of the injectors comprises an outlet-valve system at the nozzle configured for opening for flow of lubricant to the nozzle aperture during an injection-phase upon pressure rise above a predetermined limit at the outlet-valve system and for closing the outlet-valve system after the injection-phase when the pressure drops. The outlet-valve system closes off for back-pressure from the cylinder and also prevents lubricant to enter the cylinder in the idle-phase between injection-phases. In addition, the outlet-valve system assists in a short closing time after injection, adding to precision in timing and volume of injected lubricant.

In these embodiments, the injector comprises a lubricant flow path from the lubricant inlet through the valve system and to the outlet-valve system for flow of the lubricant from the lubricant inlet, through the valve system and the outlet-valve system and out of the injector at the nozzle aperture. The valve system is arranged as part of the injector upstream of and optionally spaced from the nozzle. Optionally, the valve system is arranged upstream of and spaced from the outlet-valve system.

For example, the outlet-valve system comprises an outlet non-return valve. In the outlet non-return valve, the outlet-valve member, for example a ball, ellipsoid, plate, or cylinder, is pre-stressed against an outlet-valve seat by an outlet-valve spring. Upon provision of pressurised lubricant in a flow chamber upstream of the outlet-valve system, the pre-stressed force of the spring is counteracted by the lubricant pressure, and if the pressure is higher than the spring force, the outlet-valve member is displaced from its outlet-valve seat, and the outlet non-return valve opens for injection of lubricant through the nozzle aperture into the cylinder. For example, the outlet-valve spring acts on the outlet-valve member in a direction away from the nozzle aperture, although, an opposite movement is also possible.

For example, for lubricating the engine, the method comprises sending an electrical control signal from the controller to the injector and by the control signal causing the injector to open the valve system for flow of lubricant from the lubricant supply conduit through the lubricant inlet, through the valve system, and into a conduit that flow-connects the valve system with the outlet-valve system.

It is noted that the pressure of the lubricant in the lubricant supply conduit is above the predetermined limit that determines the opening of the outlet-valve system in order for the lubricant supply conduit to provide lubricant through the valve system with a pressure sufficiently high to open the outlet-valve system in the injection-phase. Accordingly, the lubricant flow through the valve system and into the conduit between the valve system and the outlet-valve system causes a pressure rise at the outlet-valve system, causing the outlet-valve system to open for flow of lubricant from the conduit to the nozzle aperture by which lubricant is injected into the cylinder through the nozzle aperture. At the end of the lubrication period, the electrical control signal from the controller is changed, causing the valve system to close again for lubricant supply from the lubricant inlet to the nozzle aperture. The pressure in the conduit decreases again, and the outlet-valve system closes.

In these embodiments, there are at least two valve systems in the injector. The valve system is regulated under control by the controller, for example by electrical signals from the controller, and the outlet-valve system is activated only by the elevated pressure of the lubricant at the outlet-valve system, once the valve system has opened and caused flow of lubricant at elevated pressure from the lubricant supply conduit to the outlet-valve system. There is no mechanical connection that couples the movable parts of the valve system with movable parts of the outlet-valve system. Coupling between the opening and closing of these two systems is done only by the lubricant that flows from the valve system to the outlet-valve system.

<u>Optional details of valve and actuator</u>

In some practical embodiments, the valve system comprises a movable, actuator-driven valve member arranged for moving from an idle-phase position, in which the valve member in the idle-phase blocks the flow path, to an injection-phase position, in which the valve member opens the flow path for flow of lubricant through the flow path in the injection-phase. Advantageously, the valve member is pre-stressed towards an idle-phase position by a valve spring.

In some embodiments, for driving the movable valve member, the injector comprises a movable and actuator-driven rigid actuator-extension that is connecting the actuator with the valve system and which is used by the actuator for displacing or rotating the valve member from an idle-phase position, in which the valve member blocks the flow path, to an injection-phase position, in which the valve member opens the flow path for flow of a lubricant through the flow path for injection of the lubricant into the cylinder in an injection-phase.

Optionally, the actuator-driven rigid actuator-extension is a pull-push-member for selectively pulling or pushing the valve member in the injection. Alternatively, the actuator extension is a rotational member transferring the driving force from a rotational actuator for example selectively in one direction or the other.

In some embodiment, the actuator is an electrically controlled actuator, for example an electromechanical actuator. Optionally, the actuator comprises an electrical solenoid arrangement with a stationary solenoid part and a movable solenoid part and wherein the actuator-extension is connected to the movable solenoid part for being driven by electrical excitation of the solenoid, wherein the solenoid is configured for excitation by the injection-phase signals from the controller.

For example, the valve system comprises a linear actuator for driving the actuator-extension. In this case, the actuator-extension is connected to the actuator, for example to an arrangement of a solenoid-plunger and solenoid coil, for upon electrical activation of the actuator to drive the actuator-extension, for example pull-push-member for open for flow from the lubricant inlet. Optionally, the actuator-extension is connected to the solenoid-plunger, whereas the solenoid coil is stationary in the injector. Alternatively, the actuator-extension is connected to the solenoid coil, which is movable together with the actuator-extension.

As alternative, piezo-electric elements can be used for driving the valve member. Such elements are electrically or wireless connected to the controller for being controlled by the controller with respect to when to contract or expand.

In some concrete embodiments, the valve member is cylindrical and comprises a stationary valve member, which, in turn, comprises a corresponding cylindrical bushing inside which the cylindrical valve member is arranged for displacement along a longitudinal axis of the bushing or arranged for rotation about longitudinal axis of the bushing.

The term cylindrical bushing is used for describing that the bushing has a cylindrical hollow, typically but not necessarily with a circular cross section. The cylindrical valve member fits tightly into the cylindrical hollow of the bushing so that no lubricant can flow between the cylindrical valve member and the cylindrical bushing apart from a potential minimal amount that is only lubricating the valve member inside the bushing and which is negligible as compared to the amount of lubricant injected into the cylinder.

System advantages

The system as described herein has a number of advantages.

By providing a valve system, and optionally and outlet-valve system, inside the injector, the mass of movable members that has to be moved during operation is low. The low mass reduces reaction time of the movable objects as compared to prior art systems, why the system implies an increased reaction speed and corresponding precision with respect to timing and amount.

As the injector has a length that is typically in the order of less than a few times, for example two times, the thickness of the cylinder wall of such large engine and is extending through an opening in the cylinder wall, the distance from the valve system to the nozzle aperture is typically in the order of the thickness of the cylinder wall or even less. For example, the distance from the valve system to the nozzle aperture is less than 20 cm or even less than 10 cm, which is much shorter than the distance of several meters between a valve and the nozzle in the prior art. This implies that the distance from the valve system to the nozzle exit is extremely short in comparison and the valve system has correspondingly short reaction times and precision.

Due to the valve system being inside the injector housing and close to the nozzle, the injector has a short reaction time such that high precision can be achieved for injection timing and time length, the latter translating into volume for injection. Due to the high timing precision and the fast reaction time, the lubricant injection in a single injection-cycle can be done with multiple partial injections. As the injector as described above only has a short and rigid flow channel from the valve system to the nozzle, for example including an outlet-valve system, uncertainties and imprecisions of the injection amount and the timing are minimized in that minute compression and expansion of the oil in the relatively long conduits are avoided as well as expansion of the conduits themselves.

For example, in the time span before the piston of the engine passes the injectors, a double injection can be made so that two types of lubricant may be mixed in the cylinder, especially when using the SIP principle.

The system may only need a single lubricant line to the injectors for lubricant, as there is no need for a return line, which minimizes costs and efforts for installation and minimizes the risk for faults. This is especially so because the engines are large and would require return lines of several meters length. Also, imprecision in time a volume when closing the valve due to dead volume in a potentially return pipe for the lubricant is avoided.

Due to the outlet-valve system with the no-return valve, injectors are stable against high pressure from the cylinder. In case that the outlet-valve system comprises a non-return valve in or at the nozzle, and the non-return valve comprises a valve member, for example ball, which is spring pressed against a valve seat, a high degree of robustness against failure has been observed. These systems are simple, and the risk for clogging is minimal. Also, the valve seats tend to be self-cleaning and subject to little uneven wear, especially for valve members being balls, why a high and long-term reliability is provided. Accordingly, the injector is simple and reliable, quick and precise, and easy to construct from standard components with low production costs.

As a conclusion, the specific valve system is acting fast due to its light-weight components. Furthermore, the components are relatively simple in construction and imply low production costs. In addition to these advantages, the valve system is reliable, robust, and has a low risk for clogging. As the components are subject to relatively small pressure load, the valve system also has a long lifetime.

<u>Optional parameters</u>

For example, the injectors comprise a nozzle with a nozzle aperture that has a diameter D if the nozzle is circular, or wherein the nozzle has an equivalent diameter D which is two times the square root of the nozzle aperture area divided by pi if the nozzle is not circular; wherein the diameter D is at least 0.1 mm, and are configured for ejecting a spray of atomized droplets, which is also called a mist of oil.

A spray of atomized droplets is important in spray lubrication. Such spray of atomized droplets is in particular important in SIP lubrication, where the sprays of lubricant are repeatedly injected by the injectors into the scavenging air inside the cylinder prior to the piston passing the injectors in its movement towards the TDC. In the scavenging air, the atomized droplets are diffused and distributed onto the cylinder wall, as they are transported in a direction towards the TDC due to a swirling motion of the scavenging air towards the TDC. The atomization of the spray is due to highly pressurized lubricant in the lubricant injector at the nozzle. The pressure is higher than 10 bar, typically between 25 bar and 100 bar for this high-pressure injection. An example is an interval of between 30 and 80 bar, optionally between 35 and 60 bar. The injection time is short, typically in the order of 5-30 milliseconds (msec). However, the injection time can be adjusted to 1 msec or even less than 1 msec, for example down to 0.1 msec. Therefore, imprecisions of only a few msec may alter the injection profile detrimentally, why high precision is required, as already mentioned above, for example a precision of 0.1 msec.

Also, the viscosity influences the atomization. Lubricants used in marine engines, typically, have a typical kinematic viscosity of about 220 cSt at 40˚C and 20 cSt at 100˚C, which translates into a dynamic viscosity of between 202 and 37 mPa·s. An example of a useful lubricant is the high performance, marine diesel engine cylinder oil ExxonMobil® Mobilgard™ 570VS (or 560VS which is being phased out). Other lubricants useful for marine engines are other Mobilgard™ oils as well as Castrol® Cyltech oils. Commonly used lubricants for marine engines have largely identical viscosity profiles in the range of 40-100˚C and are all useful for atomization, for example when having a nozzle aperture diameter of 0.1-0.8 mm, and the lubricant has a pressure of 30-80 bar at the aperture and a temperature in the region of 30-100˚C or 40-100˚C. See also, the published article on this subject by Rathesan Ravendran, Peter Jensen, Jesper de Claville Christiansen, Benny Endelt, Erik Appel Jensen, (2017) "Rheological behaviour of lubrication oils used in two-stroke marine engines", Industrial Lubrication and Tribology, Vol. 69 Issue: 5, pp.750-753, https://doi.org/10.1108/ILT-03-2016-0075.

The movement of the oil droplets is governed by the relationship between the momentum of the droplets and the forces affecting the droplets.
The drops' momentum is controlled by their speed and mass, that is

$$P = m \cdot \boldsymbol{U}$$

Where P is the momentum, m is the mass of the droplet and U is the velocity of the droplet.

The forces (F) affecting the drops are described as:

$$F_{total} = F_{drag} + F_{buoyancy} + F_{others}$$

The two most important forces are buoyancy:

$$F_{buoyancy} = (\rho_p - \rho_f) g V_p$$

and drag:

$$F_{drag} = \frac{1}{2} C_D \rho_f |\boldsymbol{U} - \boldsymbol{U}_p| (\boldsymbol{U} - \boldsymbol{U}_p) A_p$$

Here ρ is density, g is the acceleration of gravity, V is volume, C_D is the drag coefficient, $\boldsymbol{U}$ is velocity and A is the cross-sectional area of the drop. The subscript f refers to the fluid around the droplet and the subscript p refers to the droplet. Drops will behave after Newton's second low:

$$m_p \cdot \frac{d\boldsymbol{U}_p}{dt} = F_{drag} + F_{buoyancy} + F_{others} \qquad (15)$$

From (15) it appears that the influence of force changes the speed of the drops and therefore their momentum. When the ship's engine changes load, it changes ρ_f in both the buoyancy force and the drag force. A change in engine load also changes U in the thrust force. It shifts the relationship between the forces affecting the drops and the drops' momentum, and therefore changes how the drops move cf. (15).

In order to obtain a favorable relationship between the momentum of the droplets and the forces affecting them, the momentum of the droplets must be affected. This can be done by changing either:

• The temperature of the oil (i.e. the viscosity)

• The pressure in the common rail

• The geometric conditions in the nozzle, e.g. by rounding the corners at the inlet from the sac hole to the spray hole.

In addition, it appears from equation (15) above that smaller droplets will be affected more by the flushing air in the cylinder than larger droplets will, if they have the same initial speed out of the nozzle and are roughly round.

In addition to influencing the momentum of the droplets, where the oil droplets hit the cylinder wall can also be controlled by changing the injection angles in the cylinder.

With that information, it is possible to influence how the droplets move in the scavenging air and ultimately influence where they end up hitting the cylinder wall.

A pressure variation measured by the pressure gauge in the injector is used to calculate the volume flow through the nozzle in experimental tests.

These experimental tests have illustrated that a good agreement between the calculated amount from the pressure measurements and what is actually injected.

The injection rate of SIP lubrication for cylinder lubrication as tested by the Bosch rate of injection method gives the following the results:

- A rise in mass flow of approximately 1 ms. Until cavitation in the nozzle starts choking the flow.
  When the choking begins, the cavitation is violent enough to atomize the oil. This results in a fast spray development, which is a strength when using the SIP principle for lubrication.
- The injected amount as a function of ramp time is approximately a straight line, making is easier to accurately dose the correct amount of oil together with the rise and fall times close to 1 ms. For the mass flow of the lubricant oil.
- The Bosch rate of injection method is capable of predicting the delivered amount of an injection within approximately 5% of the weighted amount for almost the

entire tested range of ramp times for the highly viscous fluid used called HydraWay HVXA15, which has similar properties to heated cylinder lubrication oil [Ravendran R, Jensen P, De Claville Christiansen J et al. Rheological behavior of lubrication oils used in two-stroke marine engines. Industrial Lubrication and Tribology 2017; 69(5): 750–753. DOI:10.1108/ILT-03-2016-0075].

Being able to accurately dose lubrication oil at from mg per injection and up, together with the fast rise and fall times for the mass flow, paves the way for new possibilities of lubrication strategies. These include being able to inject several times in a piston stroke and deliver the needed amount every time.

SHORT DESCRIPTION OF THE DRAWINGS

The invention will be explained in more detail with reference to the drawing, where

Fig. 1 is a sketch of part of a cylinder in a first embodiment of an engine according to the present invention,

Fig. 2 is a drawing of an embodiment of the injector illustrated in Fig. 1

Fig. 3 is a graph illustrating experimental data, and comparison to literature,

Fig. 4 is a sketch illustrating a further embodiment of a nozzle for the injector illustrated in Fig. 2,

Fig. 5 is a sketch corresponding to Fig. 1 of part of a cylinder in a further embodiment of an engine according to the present invention,

Fig. 6 is a sketch of an embodiment of the injector illustrated in Fig. 5,

Fig. 7 is an enlarged section of the inlet-valve housing of the injector illustrated in Fig. 6,

Fig. 8 is a sketch corresponding to Fig. 1 of part of a cylinder in a further embodiment of an engine according to the present invention,

Fig. 9 is a graph illustrating experimental data, and comparison to literature,

Fig. 10 is a graph illustrating the relation between time and mass flow,

Fig. 11 illustrates a first initial droplet size distribution,

Fig. 12 illustrates a second initial droplet size distribution,

Fig. 13 illustrates a third initial droplet size distribution,

Fig. 14 is an illustration of the of the droplets which do not reach the cylinder wall for the distribution illustrated in Fig. 13,

Fig. 15 is a partially sketch of a common rail system comprising a common rail, a high-pressure unit with a pump and a valve,

Fig. 16 is a sketch of a further embodiment of an injector to be used in a system according to the present invention and illustrated in closed position and

Fig. 17 is a sketch of the injector in Fig. 13, however illustrated in open position.

DETAILED DESCRIPTION / PREFERRED EMBODIMENT

FIG. 1 illustrates one half of a cylinder 1 of a large slow-running two-stroke engine, for example marine diesel engine. The cylinder 1 comprises a cylinder liner 2 on the inner side of the cylinder wall 3. Inside the cylinder wall 3, there are provided a plurality of injectors 4 for injection of lubricant into the cylinder 1. As illustrated, the injectors 4 are distributed along a circle with the same angular distance between adjacent injectors 4, although this is not strictly necessary. Also, the arrangement along a circle is not necessary, seeing that an arrangement with axially shifted injectors is also possible, for example every second injector shifted towards the piston’s top dead centre (TDC) relatively to a neighbouring injector.

Each of the injectors 4 has a nozzle 5 with a nozzle aperture 5’ from which a fine atomized spray 8 with miniature droplets 7 is ejected under high pressure into the cylinder 1.

For example, the nozzle aperture 5’ has a diameter of between 0.1 and 0.8 mm, such as between 0.2 and 0.5 mm, which at a pressure of 10-100 bar, for example 25 to 100 bar, optionally 30 to 80 bar or even 50 to 80 bar, atomizes the lubricant into a fine spray 8, which is in contrast to a compact jet of lubricant. The swirl 14 of the scavenging air in the cylinder 1 transports and presses the spray 8 against the cylinder liner 2 such that an even distribution of lubrication oil on the cylinder liner 2 is achieved. This lubrication system is known in the field as Swirl Injection Principle, SIP.

However, also other principles are envisaged in connection with the improved lubrication system, for example injectors that have jets or sprays directed towards the cylinder liner or towards the piston ring pack.

Optionally, the cylinder liner 2 is provided with free outs 6 for providing adequate space for the spray 8 or jet from the injector 4.

In addition to a lubricant supply conduit 9 – being a common supply conduit - the injectors 4 are connected to the controller 11 by a pressure-control conduit 10. The lubricant supply conduit 9 is used for providing lubricant for injection. The pressure-control conduit 10 provides oil at high pressure to activate an internal pumping system inside the injector 4, which will be explained in more detail in the following.

The pressure in the pressure-control conduit 10 is higher than the pressure in the lubricant supply conduit 9. Typically, the lubricant pressure in the lubricant supply conduit 9 is in the range of 1-15 bar, for example in the range of 5-15 bar, and the oil pressure in the pressure-control conduit 10 is in the range 20-100 bar, for example in the range 30-80 bar, optionally 50-80 bar.

The controller 11 is connected to a supply conduit 12 for receiving lubricant from a lubricant supply 25, including an oil pump, and a return conduit 13 for return of lubricant, typically to an oil reservoir, optionally for recirculation of lubricant. The lubricant pressure in the supply conduit 12 is higher than the pressure in the return conduit 13, for example at least two times higher.

The controller 11 supplies lubrication oil to the injectors 4 in precisely timed pulses, synchronised with the piston motion in the cylinder 1 of the engine. For example, for the synchronisation, the controller system 11 is electronically by wires or wireless connected to a computer 11’ which controls components in the controller 11 for the lubrication supply. Potentially, the computer 11’ is part of the controller 11, for example provided inside a single casing with the other components of the controller 11. Optionally, the computer monitors parameters for the actual state and motion of the engine, for example speed, load, and position of the crankshaft, the latter revealing the position of the pistons in the cylinders.

FIG. 2 illustrates an injector 4.

The injector 4 comprises an injector housing 4’ with an injector base 21 having a lubricant inlet port 4A for receiving lubricant from the lubricant supply conduit 9 and a pressure port 4B connected to the pressure-control conduit 10 for causing ejection of lubricant by the injector 4.

A flow chamber 16, as part of the injector housing 4’, holds the nozzle 5 relatively to the injector base 21. In the shown embodiment, the flow chamber 16 is provided as a hollow rigid rod. The flow chamber 16 is sealed against the injector base 21 by an O-ring 22 and held tightly against the injector base 21. A conduit 16’ is provided as a hollow channel inside the flow chamber 16 from the rear to the front of the flow chamber 16. The conduit 16’ communicates with the lubricant inlet port 4A and with the nozzle 5 through a rear chamber 16A, a first intermediate chamber 16B, a second intermediate chamber 16C, and front chamber 16D.

The injector 4 also comprises an outlet-valve system 15 for regulating the lubricant that is dispensed through the nozzle aperture 5’. Only if the pressure exceeds a predetermined pressure at the outlet-valve system 15, the outlet-valve system 15 opens for ejection of the lubricant into the cylinder 1 of the engine. In the embodiment of FIG. 2, the outlet-valve system 15 is exemplified as being part of the nozzle 5, although, this is not strictly necessary.

The outlet-valve system 15 comprises an outlet non-return outlet-valve 17. In the outlet non-return outlet-valve 17, an outlet-valve member 18, exemplified as a ball, is pre-stressed by a spring-load against an outlet-valve seat 19 by an outlet-valve spring 20. Upon provision of pressurised lubricant in the front chamber 16D, the pre-stressed force of the outlet-valve spring 20 is counteracted by the lubricant pressure, and when the pressure gets higher than the spring force, the outlet-valve member 18 is displaced from its outlet-valve seat 19, and the outlet non-return outlet-valve 17 opens for injection of lubricant through the nozzle aperture 5’ into the cylinder 1.

As exemplified, the outlet-valve spring 20 acts on the valve member 18 in a direction away from the nozzle aperture 5’. However, the configuration could be different with respect to the direction of the force of the outlet-valve spring 20 on the outlet-valve member 18 relatively to the nozzle aperture 5’, as long as the non-return outlet-valve

17 is closing for the supply of lubricant to the nozzle aperture 5' when in an idle state between injection phases. The closing of the non-return outlet-valve 17 in an idle state prevents unintended flow of lubricant from the front chamber 16D through the nozzle aperture 5' into the cylinder 1 between injection phases.

The rear chamber 16A communicates with the inlet port 4A for receiving lubricant from the lubricant supply conduit 9. The rear chamber 16A communicates with the first intermediate chamber 16B through rear channel 23A. The first intermediate chamber 16B communicates with the second intermediate chamber 16C through intermediate channel 23B, which is a cylindrical opening around an actuator-member 28, which will be explained below. The second intermediate chamber 16C communicates with the front chamber 16D through front channel 23C.

For sake of convenience, the term "forward motion" is used for motion towards the nozzle aperture 5' and the oppositely directed motion away from the nozzle aperture 5' is called "rearward motion".

The front chamber 16D is emptied through the nozzle aperture 5' by forward motion of a reciprocal plunger-member 29, which is spring-loaded against the forward motion by a helical plunger spring 29B in the second intermediate chamber 16C. The plunger-member 29 comprises a channel inlet 24 that leads into front channel 23C, which is an internal channel in the plunger-member 29, for example centrally in the plunger-member 29, as indicated. During forward motion of the plunger-member 29, the front channel 23C is closed by a non-return plunger-valve 26. In the shown embodiment, the non-return plunger-valve 26 is exemplified as comprising a plunger-valve ball 26A in a plunger-valve seat 26B, against which the plunger-valve ball 26A is pre-stressed by a plunger-valve spring 26C.

Forward motion of the plunger-member 29 is achieved by forward motion of an actuator-member 28, which presses against the head 29A of the plunger-member 29. The actuator-member 28 is pre-stressed in a rearward direction by a helical actuator-spring 28A in the first intermediate chamber 16B.

In this illustrated exemplary embodiment, the actuator-member 28 and the plunger-member 29 are separate elements, however, they could also be combined as a single actuator-plunger, for example by having the actuator-member 28 at one end of the single element and the plunger-member 29 at the opposite end.

Forward motion of the actuator-member 28 is achieved by pressurized lubricant from the pressure-control port 4B, which presses on the rear part 28B of the actuator-member 28 in pressure chamber 27 so that they move together.

The functioning of the injector 4 is explained in the following in greater detail. When the pressure-control port 4B is provided with pressurized oil, for example in the pressure range of 20-100 bar, the pressurized oil expands the volume of the pressure chamber 27 by pushing on the rear part 28B of the actuator-member 28 and moving the actuator-member 28 forward. As the actuator-member 28 presses against the head 29A of the plunger-member 29, the plunger-member 29 moves forward together with the actuator-member 28 against the force of the actuator-spring 28A and the plunger spring 29B. The forward motion of the plunger-member acts on the lubricant in the front chamber 16D. As the non-return valve 26 prevents the lubricant in the front chamber 16D from escaping backwards, the lubricant in the front chamber 16D is pressurised to the predetermined pressure limit at which the outlet-valve-system system 15 with the non-return outlet-valve 17 opens for ejection of the lubricant from the front chamber 16D through the nozzle aperture 5’ into the cylinder 1.

At the end of the injection phase, the oil at the pressure-control port 4B is drained, which causes the actuator-spring 28A and the plunger spring 29B to press back the actuator-member 28 and the plunger-member 29 in a direction away from the nozzle 5. The rearward motion of the plunger-member 29 reduces the pressure in the front chamber 16D, which, in turn, closes the non-return outlet-valve 17 and draws lubricant from the second intermediate chamber 16C through front channel 23C into the front chamber 16D, as the plunger non-return valve 26 is opened by the pressure reduction in the front chamber 16D. This way, the non-return plunger-valve 26 acts as a suction valve because the pressure reduction in the front chamber 16D causes refilling of the front chamber 16D with lubricant by suction through the non-return plunger-valve 26. During this returning motion of the actuator-member 28 and plunger-member 29, lubricant in the second intermediate chamber 16C is replenished from the first intermediate chamber 16B, which is turn is filled by lubricant from the rear chamber 16A that received the lubricant through the lubricant inlet port 4A.

For proper functioning, the lubricant inlet port 4a is provided with lubricant from the lubricant feed conduit 9 at a constant pressure, and the pressure-control port 4B is provided with pressure-oil from the pressure-control conduit 10 intermittently for each injection-cycle. The pressure at the pressure-control port 4B is raised in the injection phase and reduced in the idle state between injection phases.

If the forward force on the actuator-member 28 by the oil pressure in the pressure chamber 27 in the idle state is less than the combined rearward forces from the actuator spring 28A and the plunger spring 29B, the actuator-member 28 and the plunger-member 29 are returned fully to their most rearward possible position, which is the one indicated in FIG. 2. Thus, a full stroke of the plunger-member is achieved by intermittently changing the oil pressure at the pressure-control port 4B between a full pressure and a lower pressure, for example at the pressure of the lubricant in the lubricant supply conduit 9 or even lower.

However, the actuator-member 28 and the plunger-member 29 can be held offset from the most rearward position by adjusting the pressure at the pressure-control port 4B and in the pressure chamber 27 at an offset pressure level that creates a force on the actuator such that the springs 28A and 29B do not fully elongate during rearward motion of the actuator-member 28 and the plunger-member 29 but are kept slightly compressed. This is possible because the force of the springs 28A and 29B is varying in dependence on the compression length, largely following a linear dependence on the displacement of the actuator-member from the most rearward position. The offset pressure level is smaller than the pressure level necessary to cause the non-return outlet-valve 17 to open for injection.

In principle, the injector 4 can be provided with lubricant at the inlet port 4A from one lubricant source and with pressure-oil or other pressure-liquid from an entirely different source. However, typically, for sake of simplification and convenience, the pressure-oil at the pressure-control port 4B is provided from the same source as the lubricant at the inlet port 4A, however, with increased pressure, for example by use of a pressure intensifier.

As it appears from the above example, the lubricant supply conduit 9 is communicating with the return line 13. This is also illustrated in FIG. 1 by the solid line 9'' which in extension is connected to the supply conduit 9. In case of the controller 11 being an add-on unit, the controller 11 would have at least 4 conduit connectors.

However, this need not be so. Optionally, the controller 11 comprises a return exit line which is connected to the return conduit 13. In the latter case, the return conduit 13 is directly communicating with the supply conduit 9' and 9. This embodiment is illustrated in FIG. 1 by the dotted alternative line 9', which in extension is connected to the supply conduit 9. In this case, the return conduit 13 in extension of the supply conduit 9 and 9' provides lubricant directly to the lubricant inlet port 4A of the injectors 4 for injection into the cylinder, and the controller 11 is bypassed.

The following values are non-limiting illustrative examples of possible working pressures. The pressure in the return conduit 13 and the supply conduit 9 is 10 bar. The pressure in the supply conduit 12 is 40 bar. The outlet-valve 15 opens at 37 bar, such that the lubricant is injected at 37 bar. The springs 28A and 29B are configured for pressing the plunger-member 29 and the actuator-member 28 fully back to the rearmost position if the pressure at the pressure-control port 4B in the idle state between injection phases is 10 bar. The pressure-valve is adjustable to a pressure of between 10 and 30 bar, for example 20 bar, well below the 37 bar injection pressure but high enough to provide high enough pressure in the pressure chamber 27 for the actuator-member 28 not returning fully to the most rearward position but keeping a distance from the most rearward. By adjusting this distance through adjustment of the pressure in the range of 10-30 bar, the injection volume in the front chamber 16D is adjusted, because the forward motion in the injection phase is smaller the more the plunger-member 29 is offset from the most rearward position at the start of the injection phase.

Optionally the injection volume is controlled by a flowmeter inserted in the supply line 9, either for the group of injectors or for each single injector 4. The flowmeter measures flow (mass and/or volume) and is then used for control that the injector(s) is/are properly working.

The injection system with the injector 4 as described above and the controller 11 are simple to install and replace. It is a relatively low-cost technical solution, albeit robust and stable. Especially, the injection volume is precisely adjustable. Also, the system does not comprise electrical wires to and from the injector 4, which makes it robust against heat, whereas electrical wires are likely to have an insulation layer that melts in heat.

The above-described embodiments are known from WO 2019/114905. However, the engine differs as it comprises a pressure gauge 35. The pressure gauge may also be called a pressure sensor. The pressure gauge 35 shall be used for measuring the pressure in the lubricant supply conduit 9 – the common rail for supplying lubricant to the injectors 4.

In FIG. 2 a pressure gauge 35 for each injector is illustrated, however in some practical embodiments of the system fewer pressure gauges may be provided. In some embodiment only one pressure gauge is used for the lubricant supply conduit 9.

The signal from the pressure gauge for the actual pressure in the lubricant supply conduit 9 is then transmitted to the controller and used in calculation of a desired pressure depending on the load condition of the engine which pressure is obtained by controlling a high-pressure pump in the lubricant supply.

Alternatively, the high-pressure pump may not be adjustable and in this situation the desired pressure in the lubricant supply conduit 9 is obtained by controlling valves provided in connection with high-pressure pump and used for connecting the high-pressure pump to the lubricant supply conduit 9.

FIG. 5 illustrates one half of a cylinder 1 of a large slow-running two-stroke engine, for example marine diesel engine. The cylinder 1 comprises a cylinder liner 2 on the inner side of the cylinder wall 3. Inside the cylinder wall 3, there are provided a plurality of injectors 4 for injection of lubricant into the cylinder 1. As illustrated, the injectors 4 are distributed along a circle with the same angular distance between adjacent injectors 4, although this is not strictly necessary. Also, the arrangement along a circle is not necessary, seeing that an arrangement with axially shifted injectors is also possible, for example every second injector shifted towards the piston's top dead centre (TDC) relatively to a neighbouring injector.

Each of the injectors 4 has a nozzle 5 with a nozzle aperture 5' from which a fine atomized spray 8 is ejected under high pressure into the cylinder 1.

For example, the nozzle aperture 5' has a diameter of between 0.1 and 0.8 mm, such as between 0.2 and 0.5 mm, which at a pressure of 10-100 bar, for example 25 to 100 bar, optionally 30 to 80 bar or even 50 to 80 bar, atomizes the lubricant into a fine spray 8, which is in contrast to a compact jet of lubricant. The swirl 14 of the scavenging air in the cylinder 1 transports and presses the spray 8 against the cylinder liner 2 such that an even distribution of lubrication oil on the cylinder liner 2 is achieved. This lubrication system is known in the field as Swirl Injection Principle, SIP.

However, also other principles are envisaged in connection with the improved lubrication system, for example injectors that have jets or sprays directed towards the cylinder liner or towards the piston ring pack.

Optionally, the cylinder liner 2 is provided with free outs 6 for providing adequate space for the spray 8 or jet from the injector 4.

The injectors 4 receive lubrication oil through a lubricant supply conduit 9 – being a common supply conduit, from a lubricant supply 25, including a lubricant pump that raises the pressure of the lubricant to an adequate level. For example, the pressure in the supply conduit 9 is in the range of 10 bar to 400 bar, preferably a range of 25 to 100 bar, optionally 30 to 80 bar, which is a typical range of pressure for spray injectors, e.g. SIP injectors.
The injectors 4 are provided with electrical connectors 110' that are electrically communicating with a controller 11 through electrical cables 110. As mentioned earlier the injectors may alternatively be wireless communicating with the controller 11. The controller 11 sends electrical control signals to the injectors 4 for controlling injection of

lubricant by the injector 4 through the nozzle 5. As it is illustrated, one cable 110 is provided for each injector 4, which allows individual control of injection by the respective injector. However, it is also possible to provide one electrical cable 110 from the controller 11 to all injectors 4 such that all injectors 4 are injecting simultaneously upon receiving an electrical control signal through one single electrical cable. Alternatively, it is also possible to provide one electrical cable 110 from the controller 11 to a subgroup of injectors, for example a subgroup of 2, 3, 4, 5 or 6 injectors, such that a first subgroup is controlled by the controller through a first cable 10 and a second subgroup is controlled through a second cable 110. The number of cables and subgroups are selective dependent on preferred configurations.

The above-described embodiments are known from WO 2019/114905. However, the engine differs as it comprises a pressure gauge 35. The pressure gauge may also be called a pressure sensor. The pressure gauge 35 shall be used for measuring the pressure in the lubricant supply conduit 9 – the common rail for supplying lubricant to the injectors 4.

In FIG. 5 a pressure gauge 35 for each injector is illustrated, however in some practical embodiments of the system fewer pressure gauges may be provided. In some embodiment only one pressure gauge is used for the lubricant supply conduit 9.

The signal from the pressure gauge for the actual pressure in the lubricant supply conduit 9 is then transmitted to the controller and used in calculation of a desired pressure depending on the load condition of the engine which pressure is obtained by controlling a high-pressure pump in the lubricant supply.

The electrical control signals from the controller 11 to the injectors 4 are provided in precisely timed pulses, synchronised with the piston motion in the cylinder 1 of the engine. For example, for the synchronisation, the controller system 11 comprises a computer 11’ or is electronically connected a computer 11’, by wires or wireless, where the computer 11’ monitors parameters for the actual state and motion of the engine, for example speed, load, and position of the crankshaft, where the latter reveals the position of the pistons in the cylinders.

Fig. 6 illustrates principal sketches of an injector 4. Fig. 6 is an overview sketch with three different views of the exemplified injector, top view, end view and cross-sectional side view.

The injector 4 comprises a lubricant inlet port 112 for receiving lubricant from the lubricant supply conduit 9. The inlet port 112 is provided in an inlet-valve housing 121 comprising an inlet-valve system 113 communicating with the inlet port 112 for regulating the amount of lubricant received from the lubricant supply conduit 9 during a lubrication phase. The injector 4 also comprising an outlet-valve system 115 for regulating the lubricant that is dispensed through the nozzle aperture 5'. A rigid flow chamber 116 connects the inlet-valve system 113 with the outlet-valve system 115 for flow of lubricant to the nozzle 5. In the shown embodiment, the flow chamber 116 is provided as a hollow rigid rod. The flow chamber 116 is sealed against the inlet-valve housing 121 of the inlet-valve system 113 by an O-ring 122 and held tightly against the inlet-valve housing 121 by a flange 123 that is bolted by bolts 124 against the inlet-valve housing 121.

Fig. 7 is an enlarged portion of the inlet-valve system.

Fig. 7 illustrates the inlet-valve system 113 in greater detail. Inside the inlet-valve housing 121, a non-return inlet-valve 125 is provided with an inlet-valve member 126 that is pre-stressed against an inlet-valve seat 127 by an inlet-valve spring 128. The inlet-valve member 126 is exemplified as a ball, however, a different shape, for example oval, conical, plane, or cylindrical, would also work. When the inlet-valve member 126 is displaced from the inlet-valve seat 127 against the force of the inlet-valve spring 128, lubricant flows from the inlet port 112 along the inlet-valve spring 128, passes the inlet-valve member 126 and the inlet-valve seat 127, and enters a channel 129 on an opposite side of the inlet-valve member 126. From the channel 129, the lubricant flows through passage 130 and enters the hollow part 116' of the flow chamber 116, for flow to an outlet-valve system, which have a generalised principle similar to the one disclosed in WO2014/048438. This reference also provides additional technical details as well as explanations to the functioning of the injector presented here, which are not repeated here, for convenience.

In order to displace the inlet-valve member 126 (ball), a push-member 131, exemplified as a push-rod, is provided reciprocal in the channel 129. The push-member 131 is not fastened to the inlet-valve member 126 but is fastened to a reciprocal solenoid-plunger 133 that is driven by a solenoid coil 132. The solenoid-plunger 133 is retracted by a plunger spring 134 when in idle condition. When the solenoid coil 132 is excited by electrical current, the solenoid-plunger 133 is moved forward against the force of the plunger spring 134 until it comes to a halt against a plunger stop 135. Due to the movement of the solenoid-plunger 133, the push-member (push-rod) 131 pushes the inlet-valve member (ball) 126 away from the inlet-valve seat 127, allowing lubricant to flow through the inlet non-return valve 125 and into the flow chamber 116.

In advantageous embodiments, the push-member (push-rod) 131 is withdrawn a distance from the inlet-valve member (ball) 126 when in idle state, such that there is a free range distance in between the push-member 131 and the inlet-valve member 126. When the solenoid coil 132 is excited, the push-member 131 is accelerated by the solenoid coil 132 over the free-range distance before it impacts the inlet-valve member 126 after initial acceleration. This results in the inlet-valve member 126 being displaced abruptly from the inlet-valve seat 127, as compared to a situation where the inlet-valve member 126 moves together with push-member 131 during the first part of the acceleration. The quick displacement of the inlet-valve member 126, in turn, is advantageous for a precise timing of the start of the lubricant injection into the cylinder 1. Optionally, the free-range distance is adjustable by an adjustment screw 136 at the end of the solenoid-plunger 133.

After the injection phase, the lubricant supply from the inlet port 112 to the nozzle 5 is stopped by cutting the current to the solenoid coil 132, which results in the solenoid-plunger 133 being pushed back by the plunger spring 134, and the inlet-valve member 126 returns to the tight inlet-valve seat 127 for an idle phase in the injection cycle.

The amount of lubricant oil is controlled by the flowmeter and the controller/computer makes it possible to regulate the lubricant amount to achieve cavitation in as short time as possible.

Fig. 8 corresponds to Fig. 1 and illustrates a further embodiment of one half of a cylinder 1 of a large slow-running two-stroke engine, for example marine diesel engine. This embodiment comprises oil injectors. The injectors 4 may be HJ Smartlube 4.0 E-injectors. The injectors 4 are connected with a cylinder-manifold with flowmeter 203. The cylinder-manifold with flowmeter is connected with a controller 11 via a communication line 211 for flowmeter feedback signals. The controller 11 may be a local cylinder controller which is connected with a central controller 208 via a communication line 210.

The cylinder-manifold with flowmeter 203 is connected with a pump unit 205. The pump unit 205 is via the supply conduit 12 connected with the lubricant supply 25.

The pump unit 205 is via a pressurized oil feed line 214 connected to cylinder manifolds (common rail) for providing lubricant oil to the injectors 4.

An injector signal bus 212 connects the controller 11 with the injector for regulating the lubricant amount and to effect calibration of the injectors.

Fig. 15 shows a partially sketch of a common rail system comprising a common rail, a high-pressure unit with a pump and a valve.

The valve may be omitted in case the pump is variable.

The pump may be variable for regulating the rotation speed of the pump and thereby the output of high-pressure pump for obtaining the desired pressure in the lubricant supply conduit.

Alternatively, the pump is not variable. Instead, the valve arranged at the high-pressure pump and connecting the high- pressure pump with the lubricant supply conduit is used. In this situation the regulation of the valve is used for regulating the pressure in the lubricant supply conduit to the desired pressure.

The valve is connected to the pressure side and suction side of the high-pressure pump.

Thus, the valve may relieve the pressure as it is connection to the suction side of the high-pressure pump or to a return line to the lubricant supply.

The injector is preferably of a type generally described in WO 2012/126473. This injector is also known as Hans Jensen Lubricators E-sip injector. The injector can be actuated electromechanically, for example in the form of a solenoid valve or piezo-mechanical element.

Fig. 16 and Fig. 17 show a further embodiment of an injector to be used in a system according to the present invention and illustrated in closed position and in open position respectively.

The injector illustrated in Figs. 16 and 17 is actuated electromechanically in the form of a solenoid valve.

The injector is made as a unit, that the opening/closing valve is an electromechanical valve integrated in the injector for dosing the lubricating oil. The electromechanical opening/closing valve includes a spring-biased valve stem.

Dosing of lubricating oil is performed by activating the opening/closing valve in in the injector for dosing the lubricating oil. The activation moves the valve stem of the opening/closing valve and the injection of the lubricating oil.

Numbering

1 cylinder
2 cylinder liner
3 cylinder wall
4 injector
4’ injector housing
4A inlet port of oil injector 4
4B pressure-control port of oil injector 4
5 nozzle
5’ nozzle aperture
6 free cut in liner

7 atomised spray from a single injector 4

8 swirling spray

9 lubricant supply conduit

10 pressure-control conduit

11 controller

11’ computer

12 supply conduit

13 return conduit

14 swirl in cylinder

15 outlet-valve system of injector 4

16 flow chamber connecting inlet-valve system 13 with outlet-valve system 15

16’ to hollow part of flow chamber 16

16A rear chamber

16B first intermediate chamber

16C second intermediate chamber

16D front chamber

17 non-return outlet-valve, exemplified as outlet ball valve

18 outlet-valve member, exemplified as ball

19 outlet-valve seat

20 outlet-valve spring

21 injector base

22 O-ring at end of flow chamber 16

23A rear channel in actuator-member 28 connecting rear chamber 16A with first intermediate chamber 16B

23B intermediate channel between first and second intermediate chamber 16A, 16B

23C front channel in plunger-member 29 between second intermediate chamber 16B and front chamber 16D

24 channel inlet into front channel 23C

25 lubricant supply

26 non-return plunger-valve

26A plunger-valve ball

26B plunger-valve seat, against which the plunger-valve ball 26A is pre-stressed

26C plunger-valve spring pre-stressing plunger-valve ball 26A against plunger-valve seat 26B

27 pressure chamber at rear part 28B

28 actuator-member for pushing head 29’ of plunger-member 29

28A actuator-spring acting rearwards on actuator-member 28

28B rear part of actuator-member 28

29 plunger-member

29A head of plunger-member 29

29B plunger spring in second intermediate chamber 16C

30 toggle-valve

30A toggle-valve inlet port

30B toggle-valve outlet port

30C toggle-valve return port

31pressure-valve

31A pressure-valve inlet port

31B pressure-valve return port

31C pressure regulator, ex. spring-loaded pressure adjustment member injection-phase

31D pre-tensioner in pressure-valve 31

32 toggle-member

32A first toggle closure element of toggle-member 32

32B second toggle closure element of toggle-member 32

33 arrow illustrating reciprocal movement of toggle member 32

34 return exit line from controller 11 to return conduit 13

35 flowmeter

36 desired lubricant amount

37 calculated actual lubricant amount

38 signal from flowmeter

39 control signal

40 uncalibrated injector

41 calibrated injector

112 lubricant inlet port of injector 4

113 inlet-valve system of injector 4

114 swirl in cylinder

115 outlet-valve system of injector 4

116 flow chamber connecting inlet-valve system 113 with outlet-valve system 115

116’ to hollow part of flow chamber 16

117 outlet non-return valve, exemplified as outlet ball valve

118 outlet-valve member

119 outlet-valve seat

120 outlet-valve spring

121 inlet-valve housing of inlet-valve system 115

122 O-ring at end of flow chamber 116

123 flange for holding flow chamber

124 bolts for holding flange 123 and flow chamber 16 against inlet-valve housing 121

125 inlet non-return valve, exemplified as inlet ball valve

126 inlet-valve member, exemplified as ball

127 inlet-valve seat

128 inlet-valve spring

129 channel in inlet-valve system

130 passage from channel 129 to hollow part 116’ of flow chamber 16

131 push-member fastened to solenoid-plunger 133, push-member exemplified as rod

132 solenoid coil

133 solenoid-plunger in solenoid coil 131

134 plunger spring

135 plunger stop

136 adjustment screw for adjustment of the free range distance

203 cylinder manifold with flowmeter

205 pump unit

208 central controller

210 communication line between local cylinder controller and central controller

211 communication line for flowmeter feedback signals

212 injector signal bus

214 pressurized oil feed line

215 feedback control algorithm

CLAIMS

1. A large slow-running two-stroke engine comprising a cylinder (1) with a reciprocal piston inside and with a lubricant system comprising
- a lubricant supply (25) including a lubricant pump that raises the pressure of the lubricant to a lubricant pressure which is a typical range of pressure for spray injectors,
- a plurality of lubricant injectors (4) distributed along a perimeter of the cylinder (1) for injection of lubricant into the cylinder (1) at various positions on the perimeter during injection phases, and
- a lubricant supply conduit (9) connecting the lubricant supply (25) with the lubricant injectors (4),

the engine further comprising
- a controller (11) for controlling the amount and timing of the lubricant injection by at least one of the lubricant injectors (4),

wherein each injector (4) comprises
- an inlet port flow-connected to the lubricant supply conduit (9) for receiving lubricant from it,
- a nozzle (5) with a nozzle aperture (5’) extending into the cylinder (1) configured for injecting lubricant from the inlet port (4A) into the cylinder (1) in the injection phase,

characterized in that

the controller is arranged for controlling and varying the pressure in the lubricant supply conduit (9) within a desired pressure range in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar, optionally 30 to 80 bar.

2. The large slow-running two-stroke engine according to claim 1, wherein the desired pressure is based on one or more engine operating parameters.

3. The large slow-running two-stroke engine according to claim 1 or 2 wherein the desired pressure is based on the engine load.

4. The large slow-running two-stroke engine according to any one of claims 1-3 wherein the desired pressure is also based on a measurement of the actual pressure in the lubricant supply conduit (9).

5. The large slow-running two-stroke engine according to any one of claims 1-4, wherein the engine further comprises
- a computer (11') to which the controller is connected, or alternatively
- a mobile phone arranged to communicate with the controller.

6. The large slow-running two-stroke engine according to any one of claims 1-5, wherein each injector further comprises
- an adjustable valve at the nozzle for opening and closing for flow of lubricant to the nozzle aperture in the injection phase.

7. The large slow-running two-stroke engine according to any one of claims 1-6, wherein each injector is a SIP injector.

8. A method of lubricating a large slow-running two-stroke engine comprising a cylinder (1) with a reciprocal piston inside and with a system comprising
- a lubricant supply (25) including a lubricant pump,
- a plurality of lubricant injectors (4) distributed along a perimeter of the cylinder (1) for injection of lubricant into the cylinder (1) at various positions on the perimeter during injection phases,
- a lubricant supply conduit (9) connecting the lubricant supply (25) with the lubricant injectors (4),
the engine further comprising
- a controller (11) for controlling the amount and timing of the lubricant injection by at least one of the lubricant injectors (4),
wherein each injector (4) comprises
- an inlet port flow-connected to the lubricant supply conduit (9) for receiving lubricant from it,
- a nozzle (5) with a nozzle aperture (5') extending into the cylinder (1) configured for injecting lubricant from the inlet port (4A) into the cylinder (1) in the injection phase,
wherein the method comprises
- raising the pressure of the lubricant to a lubricant pressure which is a typical range of pressure for spray injectors, and further in cyclic operation
- in the injection phase, providing pressure-liquid at inlet port of the injector and exerting force on the valve and by the force driving a valve body into motion inside the

injector (4) and upon pressure rise above the predetermined limit, pumping a predetermined lubricant volume through the nozzle aperture (5’) into the cylinder (1), and wherein the method is **characterized in that** the method comprises the step of: with the controller varying and controlling the pressure in the lubricant supply conduit (9) within a desired pressure range in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar, optionally 30 to 80 bar.

9. The method of lubricating a large slow-running two-stroke engine according to claim 8, basing the desired pressure on the engine load and a measurement of the actual pressure in the lubricant supply conduit (9).

10. The method of lubricating a large slow-running two-stroke engine according to claim 8 or 9, wherein the method includes the steps of providing for the engine
- a computer (11’) to which the controller is connected, or alternatively
- a mobile phone arranged to communicate with the controller.

11. The method of lubricating a large slow-running two-stroke engine according to any one of claims 8-10, wherein the method includes the step of providing for the engine
- an adjustable valve at the nozzle (5) for opening and closing for flow of lubricant to the nozzle aperture (5’) from a pressure chamber in the injector during an injection-cycle.

12. The method of lubricating a large slow-running two-stroke engine according to any one of claims 8-10, wherein the method includes the step of providing each injector in form of a SIP injector.

13. Use of an engine according to any one of claims 1-6 for spray injection into the cylinder of a large marine engine or combustion engine for a power plant at a lubricant pressure in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar.

14. Use of an engine according to claim 13 when using the engine according to claim 7 wherein the spray injection is a SIP injection.

15. Use of a method according to any one of claims 8-11 for spray injection into the cylinder of a large marine engine or combustion engine for a power plant at a lubricant pressure in the range of 10 bar to 400 bar, preferably a range of 25 bar to 100 bar.

16. Use of a method according to claim 15 when using the method according to claim 12 wherein the spray injection is a SIP injection.

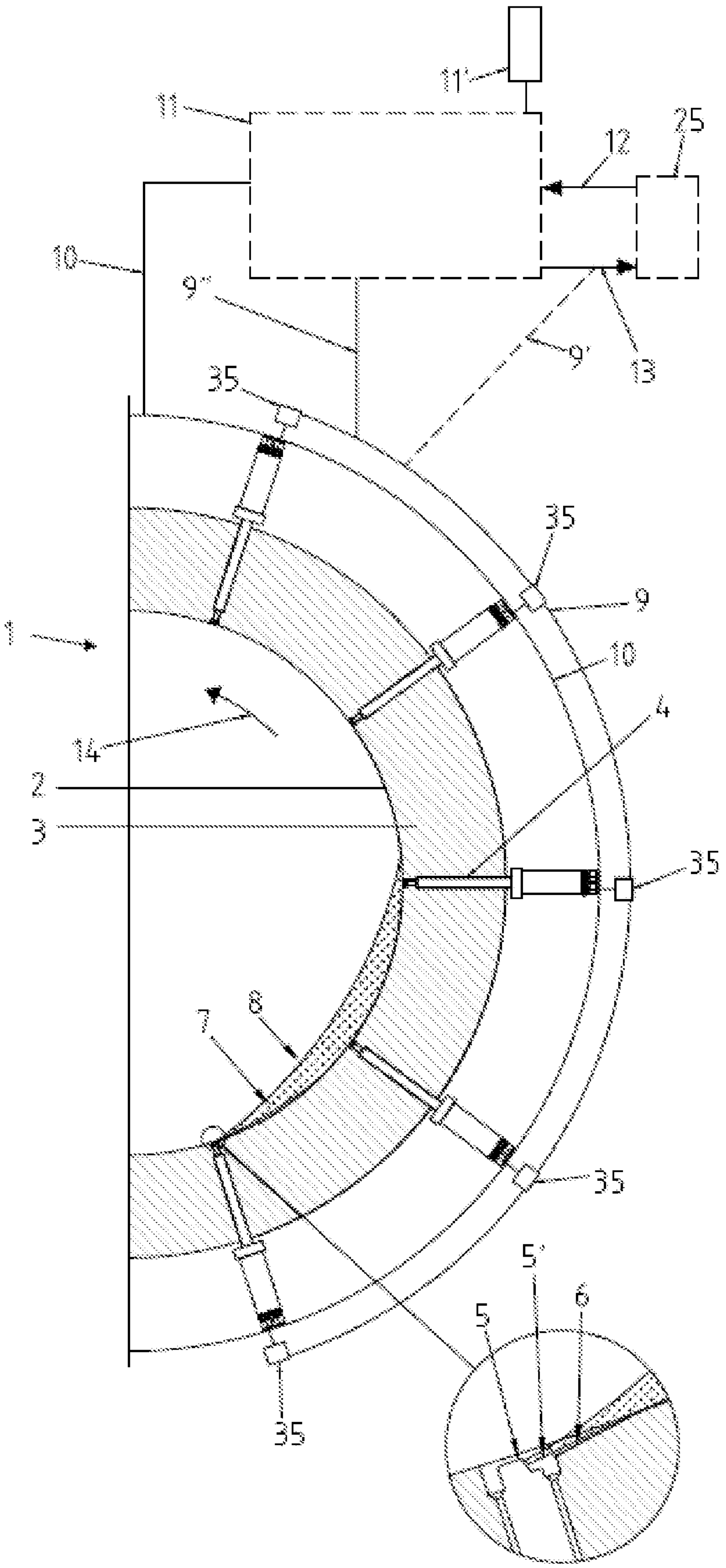

Fig. 1

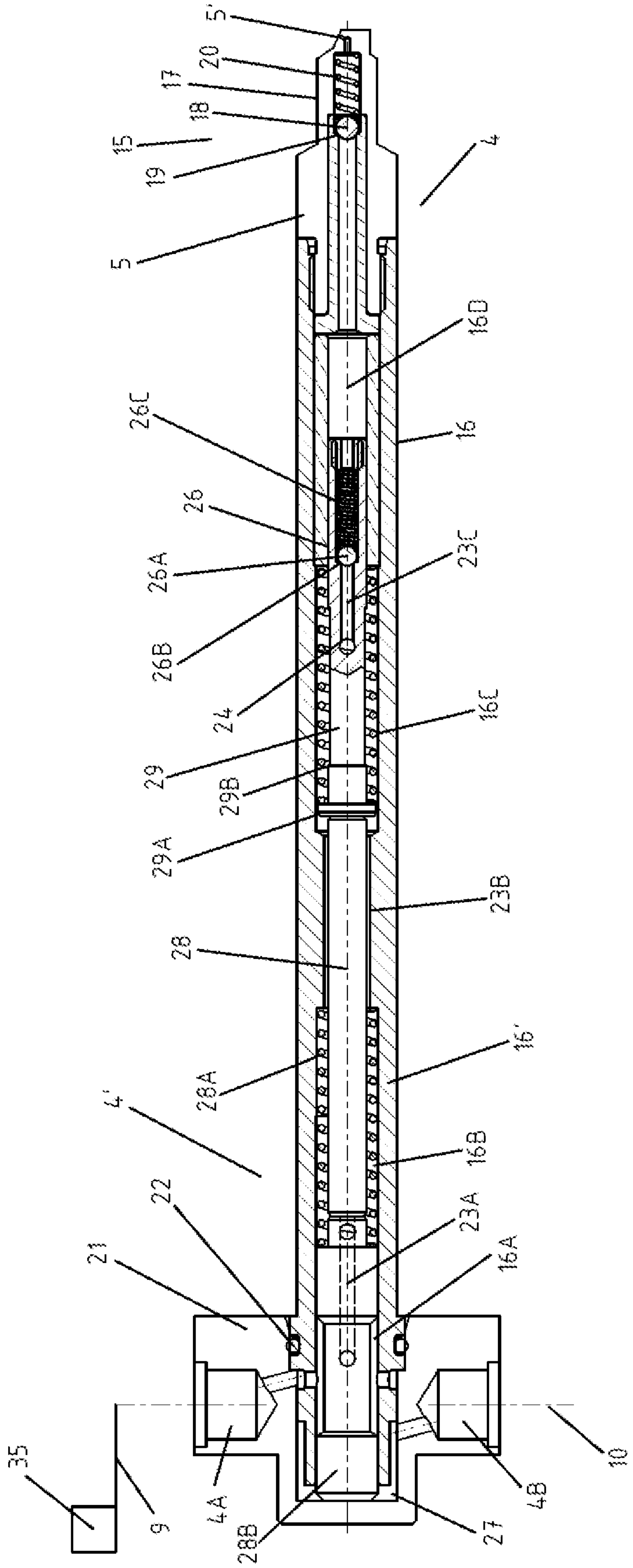
5'
20
17
18
15
19
4
5
16D
26C
16
26
26A
23C
26B
24
16C
29
29B
29A
23B
28
16'
28A
4'
16B
22
23A
21
16A
35
9
4A
28B
27
4B
10

Fig. 2

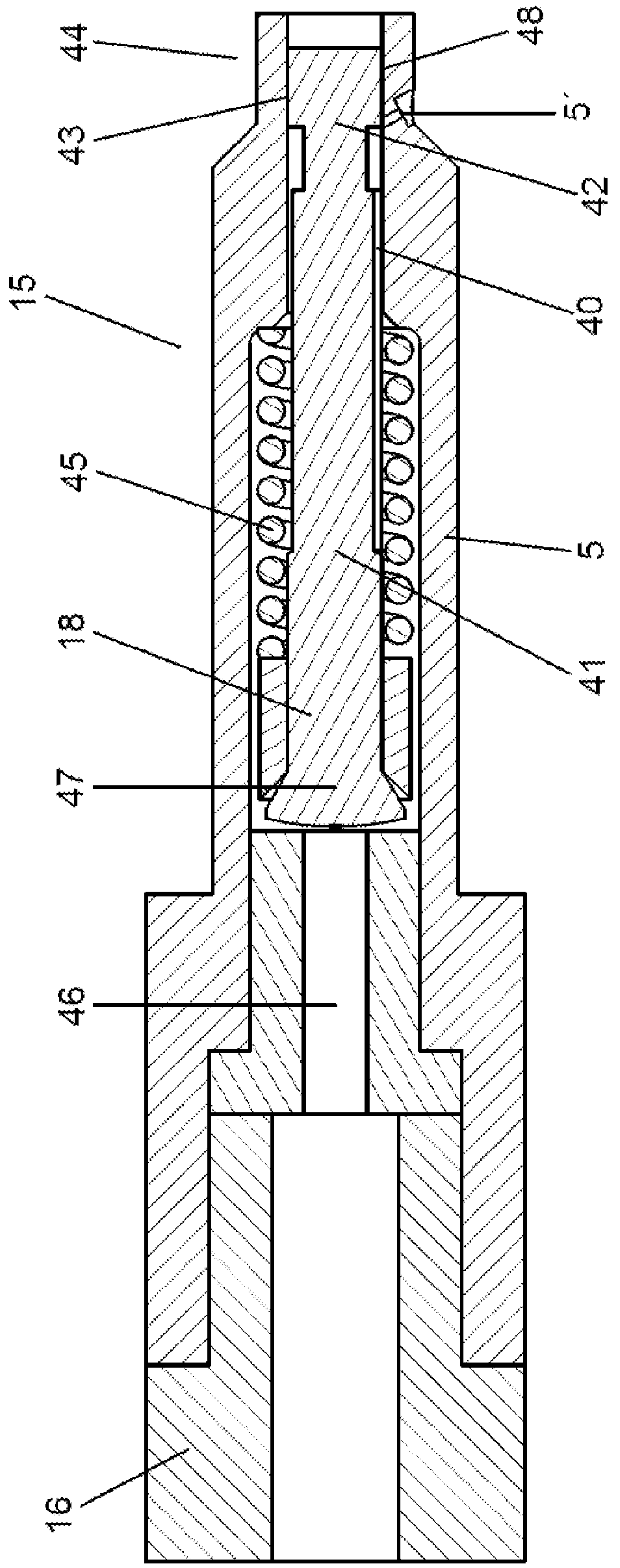

Fig. 4

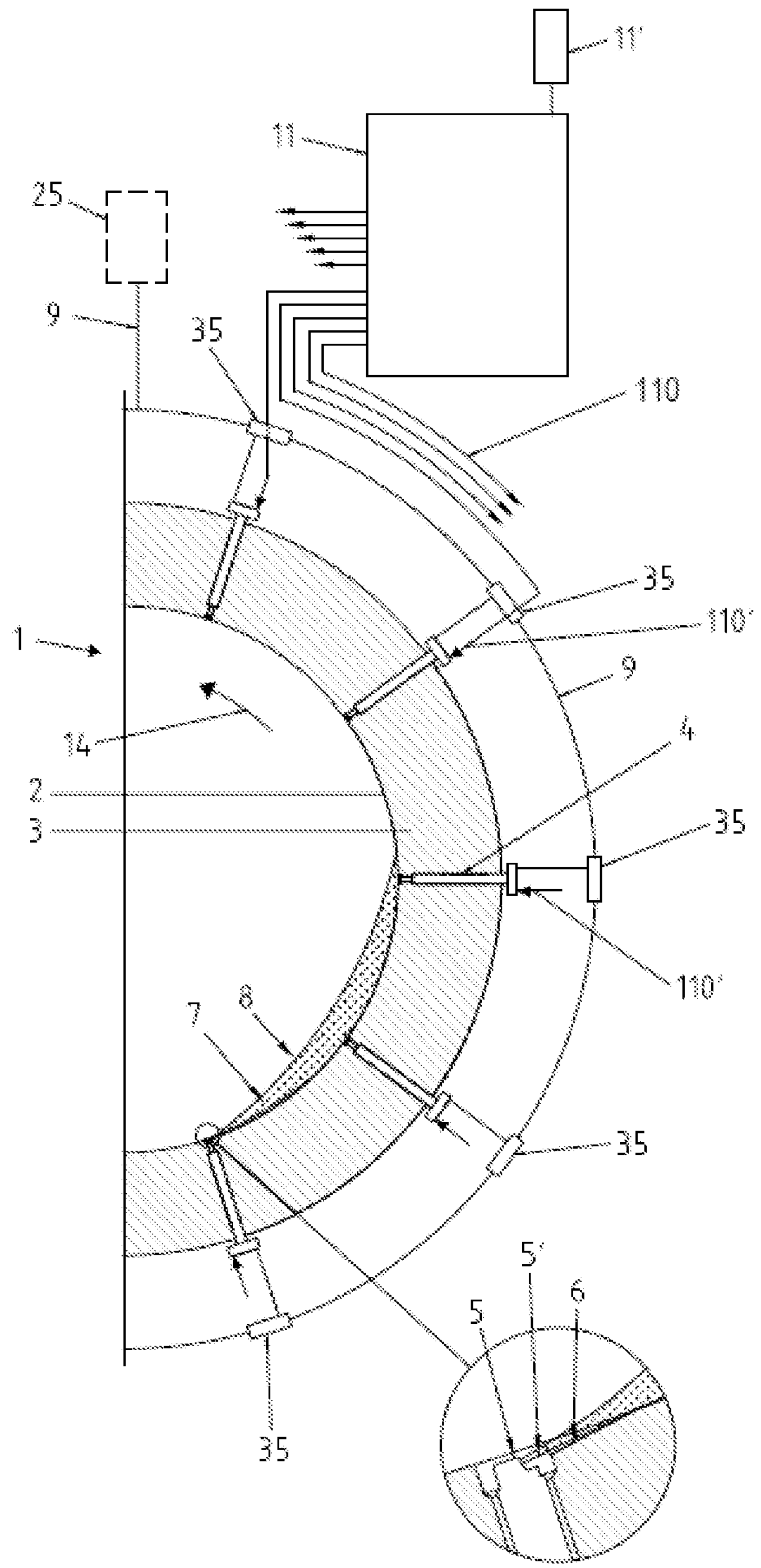

Fig. 5

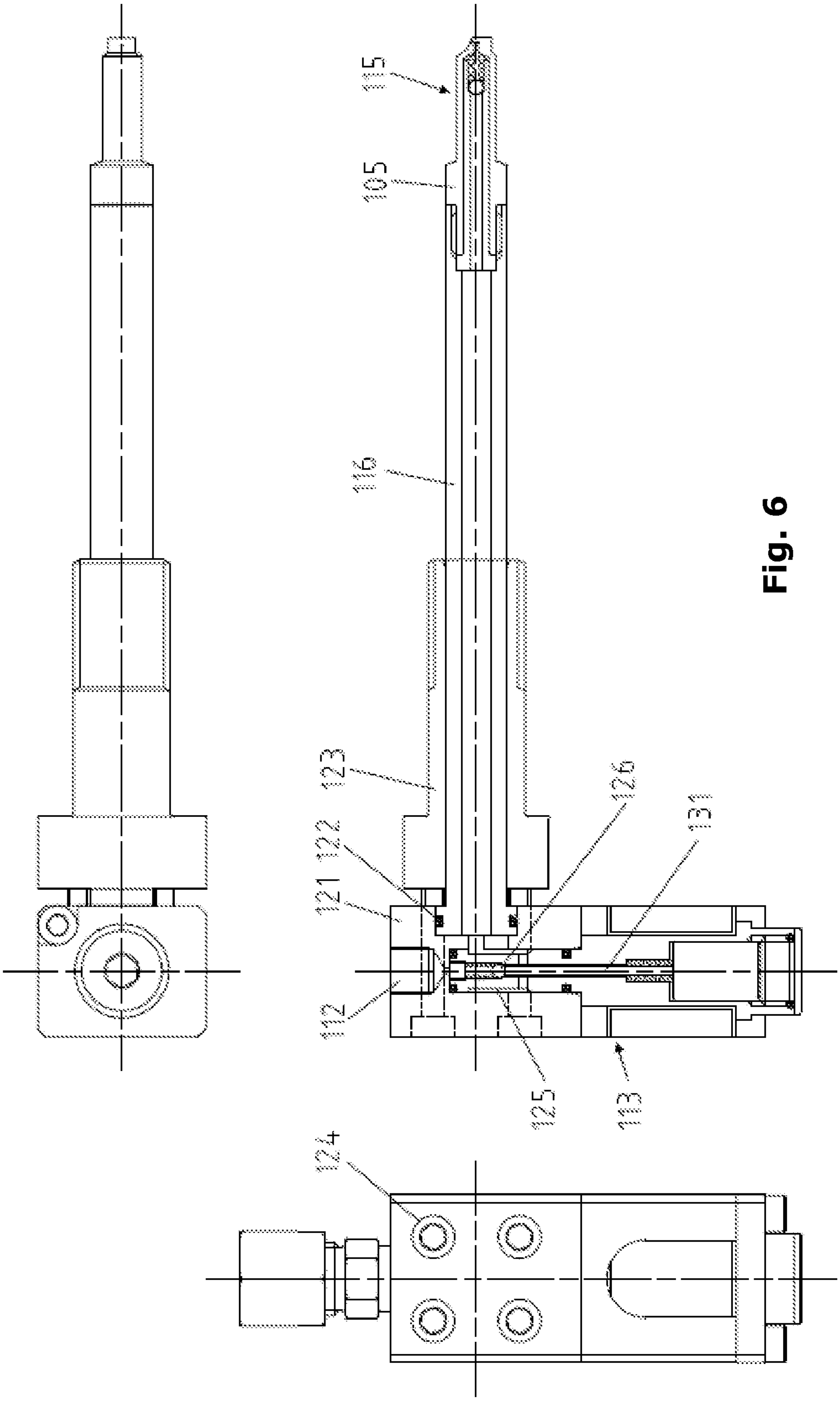

Fig. 6

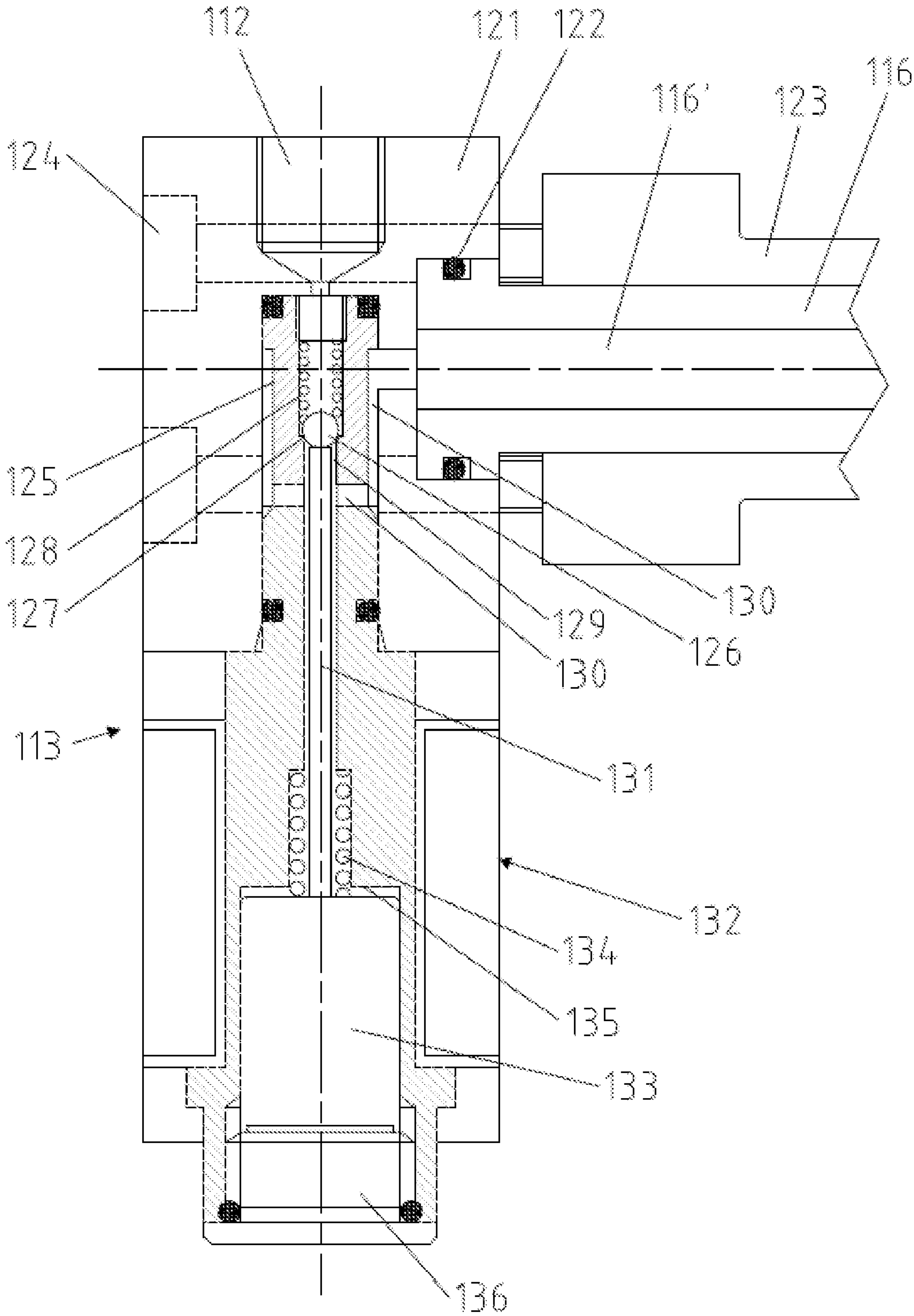
112
121
122
116'
123
116
124
125
128
127
130
129
126
130
113
131
132
134
135
133
136

Fig. 7

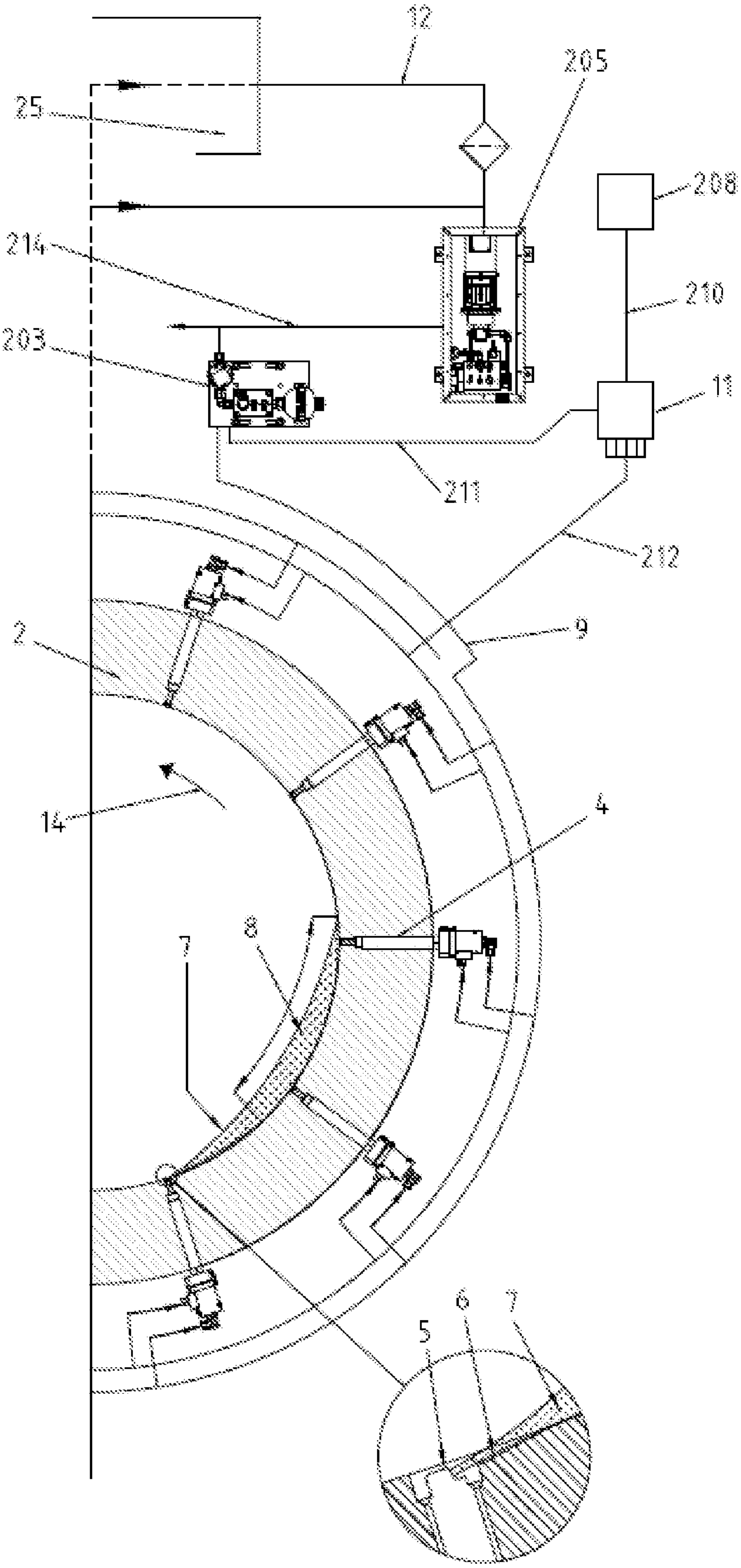

Fig. 8

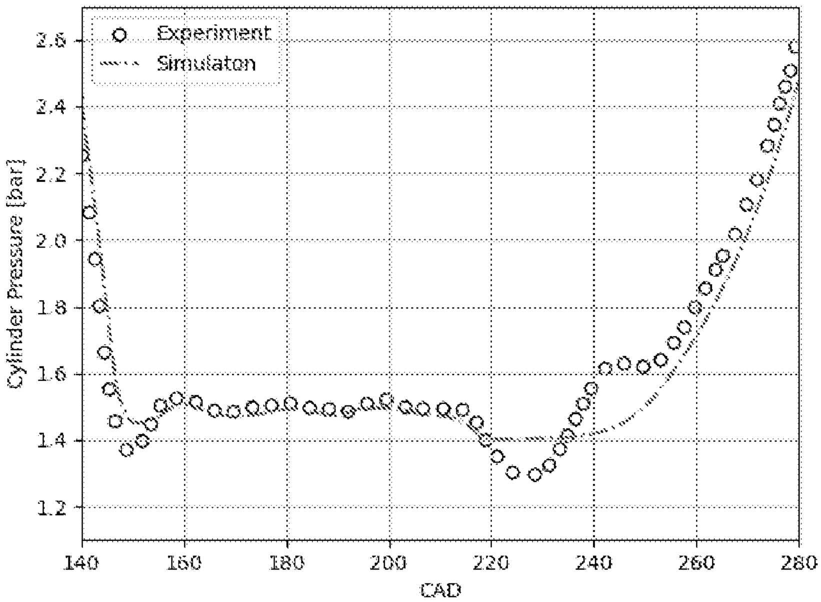
Experiment
Simulaton
Cylinder Pressure [bar]
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
140
160
180
200
220
240
260
280
CAD

Fig. 3

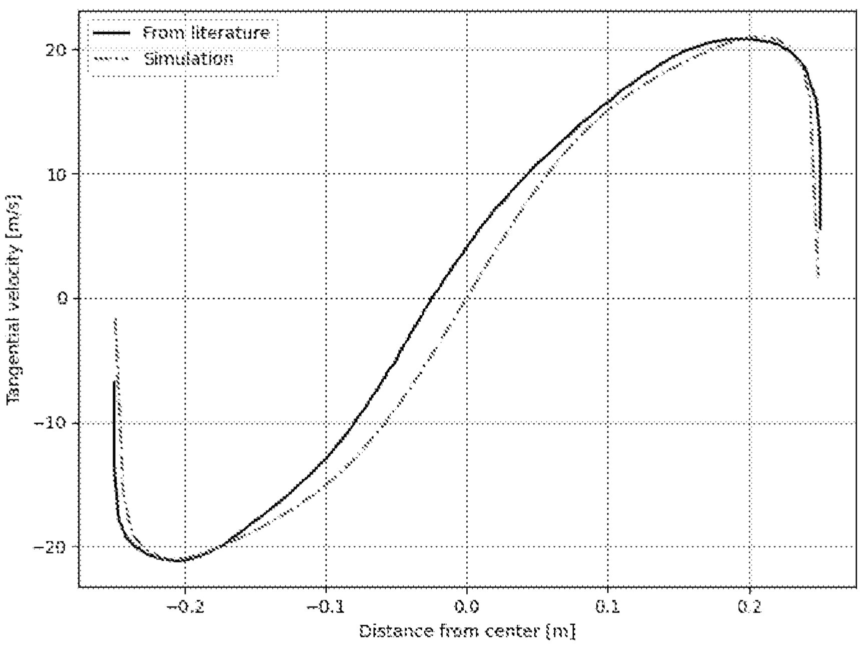
From literature
Simulation
Tangental velocity [m/s]
20
10
0
−10
−20
−0.2
−0.1
0.0
0.1
0.2
Distance from center [m]

Fig. 9

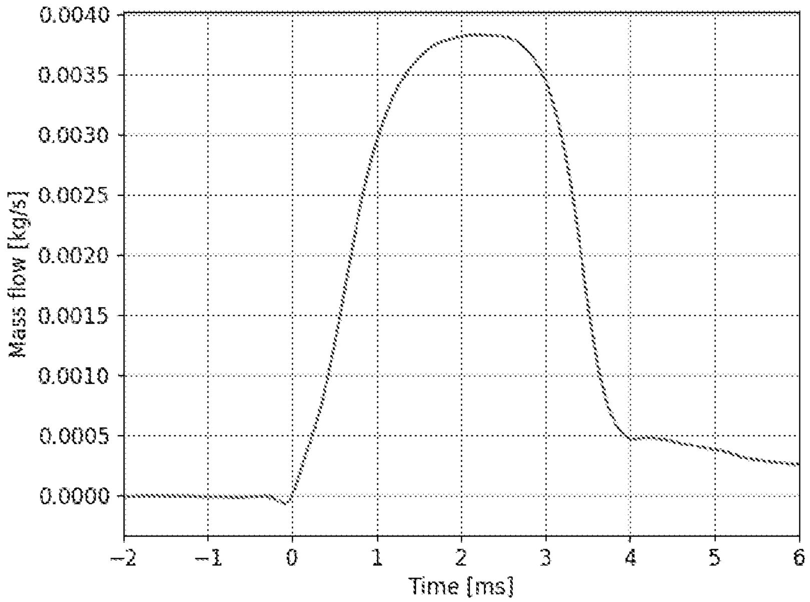
0.0040
0.0035
0.0030
0.0025
0.0020
0.0015
0.0010
0.0005
0.0000
Mass flow [kg/s]
−2
−1
0
1
2
3
4
5
6
Time [ms]

Fig. 10

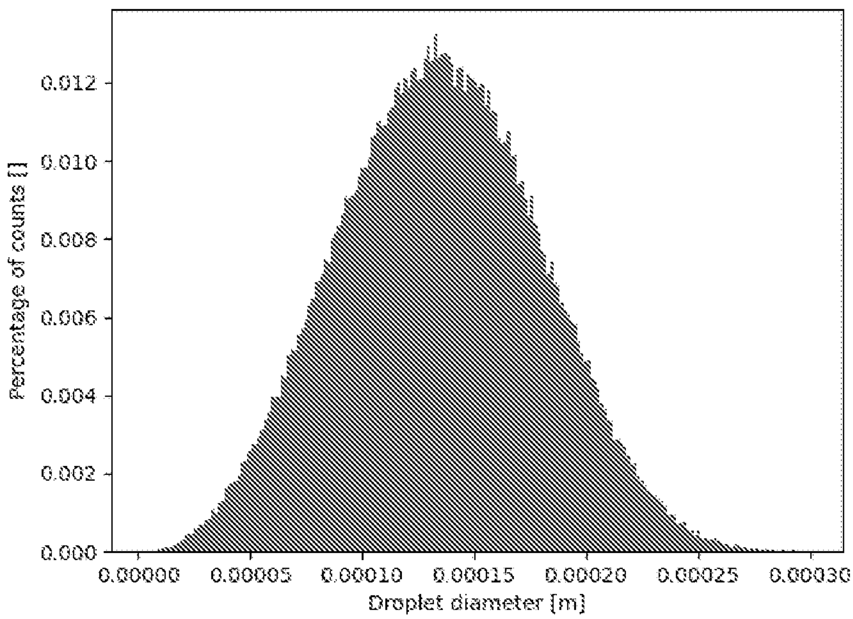
0.012
0.010
0.008
0.006
0.004
0.002
0.000
Percentage of counts []
0.00000
0.00005
0.00010
0.00015
0.00020
0.00025
0.00030
Droplet diameter [m]

Fig. 11

10/12

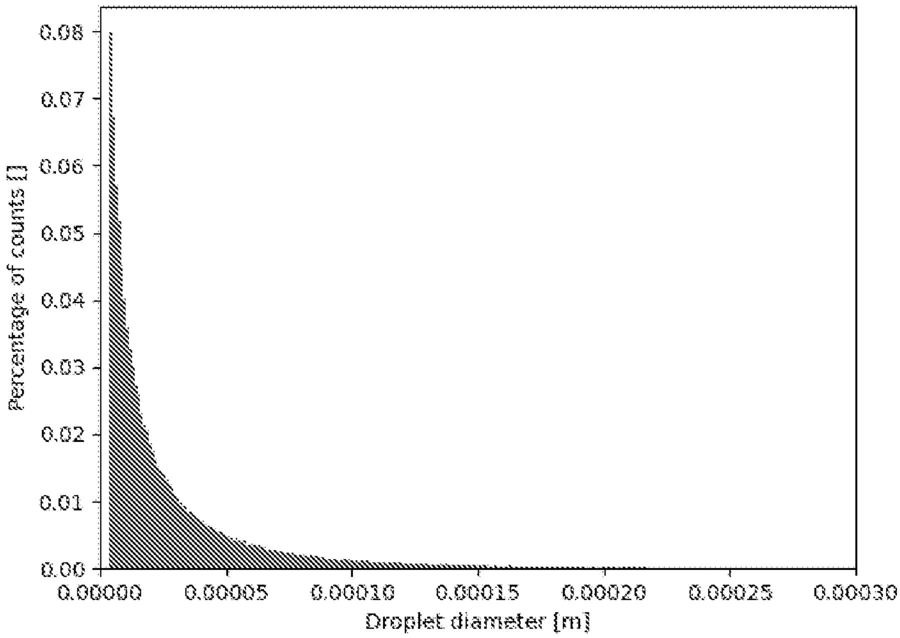

Fig. 12

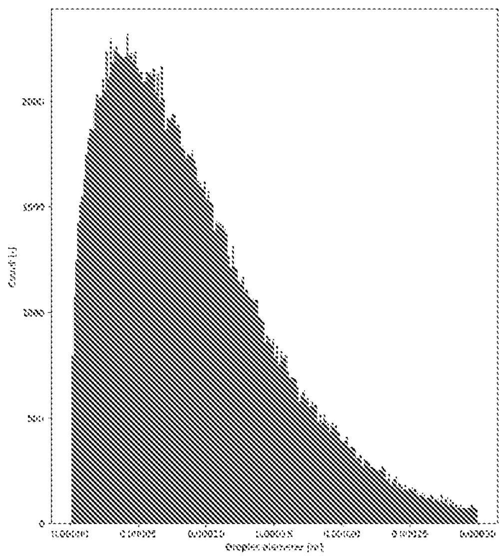

Fig. 13

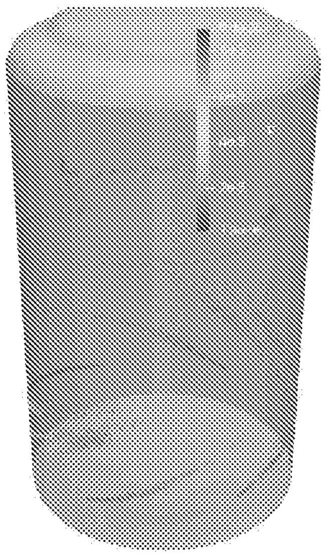

Fig. 14

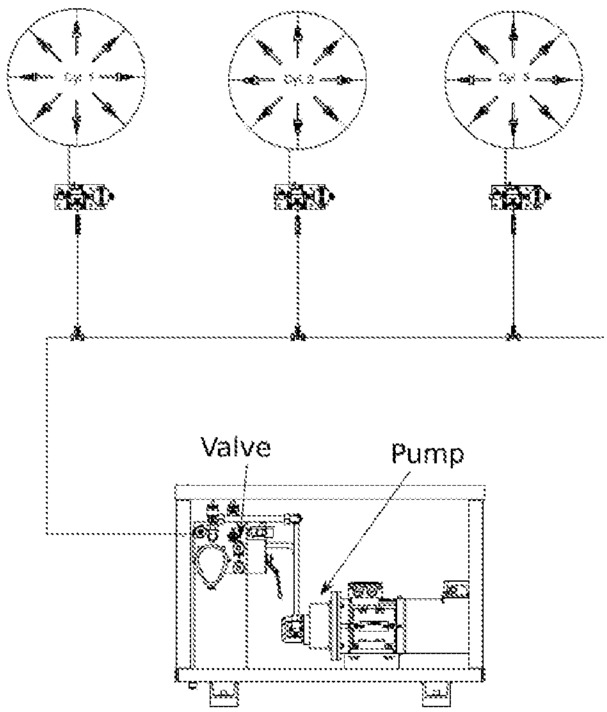

Fig. 15

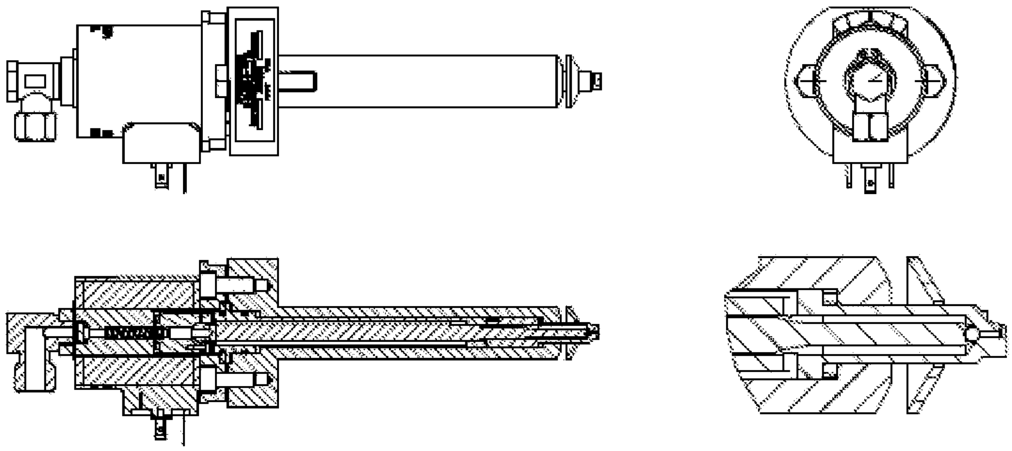

Fig. 16

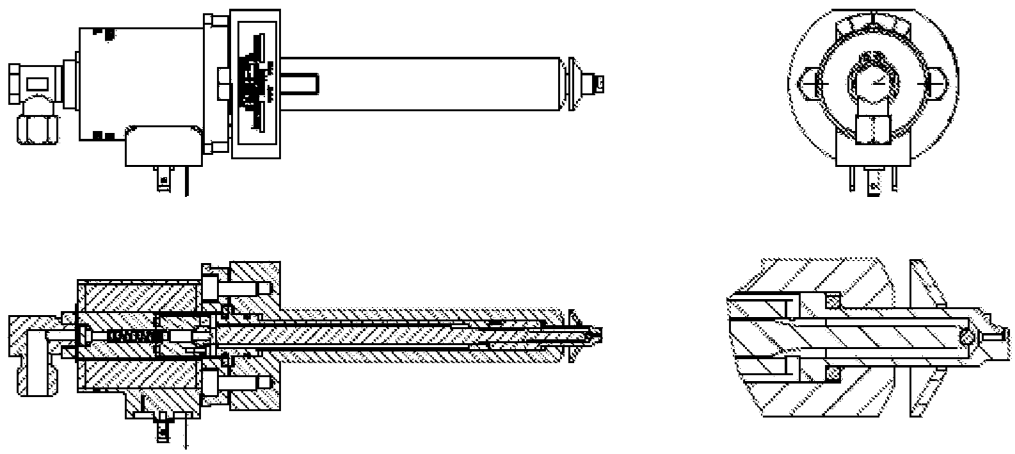

Fig. 17

INTERNATIONAL SEARCH REPORT	International application No. PCT/DK2024/050040

A. CLASSIFICATION OF SUBJECT MATTER

F01M 1/08(2006.01)i

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)

F01M 1/08; F01N; F01M 1/16; F16N 7/38
IPC & CPC: F01M, F01N, F16N

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

EPODOC

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	WO 2023088526 A1 (HANS JENSEN LUBRICATORS AS [DK]) 25 May 2023 (2023-05-25) See p. 29, ln. 2-10, p. 31, ln. 1-7, 15-31.	1-16
A	EP 0049603 A2 (BRITISH PETROLEUM CO PLC [GB]) 14 April 1982 (1982-04-14) Whole document	1-16
D,A	WO 2018215645 A1 (HANS JENSEN LUBRICATORS AS [DK]) 29 November 2018 (2018-11-29) Whole document	1-16
A	JP 2004239157 A (HITACHI SHIPBUILDING ENG CO) 26 August 2004 (2004-08-26) Whole document	1-16
A	US 2008314686 A1 (CRR DEVELOPMENTS PTY LTD [AU]) 25 December 2008 (2008-12-25) Whole document	1-16

☑ Further documents are listed in the continuation of Box C. ☑ See patent family annex.

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"D" document cited by the applicant in the international application
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Date of the actual completion of the international search	Date of mailing of the international search report
17 April 2024	**24 April 2024**

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