

A system for lubricating a large two-stroke engine using controlled mass flow in an injector nozzle

FIELD OF THE INVENTION

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

10 More specific the invention relates to a method for lubricating a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a system comprising

- a lubricant supply,
- 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,
- 15 - 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
- 20 - 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 includes the step of providing for the engine
- 25 - an adjustable valve comprising a valve member and a valve seat at the nozzle for opening and closing for flow of lubricant through a sac hole to the nozzle aperture from a pressure chamber in the injector during an injection-cycle,
- wherein the nozzle aperture has a cross section area A_3 , and wherein the sac hole has a cross section area A_2 and wherein a cross section area A_1 is provided between the valve
- 30 member and the valve seat, and
- wherein the method comprises in cyclic operation

- in the injection phase, providing pressure-liquid to the lubricant supply conduit

The invention further relates to a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a system comprising

- 5 - a lubricant supply,
- 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,
- 10 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 to which the controller is connected,
- wherein each injector comprises
- 15 - 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 comprising a valve member and a valve seat at the nozzle for
- 20 opening and closing for flow of lubricant through a sac hole to the nozzle aperture from a pressure chamber in the injector during an injection-cycle,
- wherein the nozzle aperture has a cross section area A_3 , and wherein the sac hole has a cross section area A_2 and wherein a cross section area A_1 is provided between the valve member and the valve seat,
- 25 wherein a use of the engine comprises in cyclic operation
- in the injection phase, providing pressure-liquid to the lubricant supply conduit.

Furthermore, the invention relates to an injector for a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a lubricant system

30 comprising a controller and which 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 comprising a valve member and a valve seat at the nozzle for opening and closing for flow of lubricant through a sac hole to the nozzle aperture from a pressure chamber in the injector during an injection-cycle, wherein the nozzle aperture has a cross section area A_3 , and wherein the sac hole has a cross section area A_2 and wherein a cross section area A_1 is provided between the valve member and the valve seat, and wherein the injector is arranged for in cyclic operation
- in the injection phase, providing pressure-liquid through a common rail system where all injectors are connected to the lubricant supply conduit.

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

Moreover, it is noted that the engine may further comprise

- a computer 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.

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.

5 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
10 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
15 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
20 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
25 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
30 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.

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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.

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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.

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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.

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One of the important factors for spray formation is cavitation in the nozzle, which is formation of vapour or gas cavities inside the liquid due to low static pressure. 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.

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WO 2018/215645 discloses a system mentioned in the introductory paragraph and as defined in the preamble of the independent claims. An extensive explanation regarding formation of vapour or gas cavities is disclosed in WO 2018/215645 having the same

applicant as the present application. WO 2018/215645 disclose an explanation regarding the cavitation.

5 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
10 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, in particular
15 for SIP injection.

As it was described above, the prior art on lubricant injection draws 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
20 has not been used in practice for lubricant injection in large marine engines, neither for jet injection nor for 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,
25 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 SIP lubrication.

In studies leading to the invention disclosed in WO 2018/215645, not only cavitation as
30 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 it has been found beneficial to control the increase of mass flow of the lubricant through the nozzle during opening of the injection.

5 It has shown important that the opening of the valve effects how fast it can reach its designed operation. When opening the valve, it is important to achieve cavitation in as short a time as possible since the valve requires cavitation in the nozzle tip for a spray to develop.

10 It is therefore desirable to provide a system which ensures that cavitation is achieved quickly and that a large amount of the lubricant is delivered in form of a spray.

The prior art documents do not disclose a method or a system for controlling the injector for obtaining that cavitation is achieved quickly and that a large amount of the lubricant is delivered in form of a spray.

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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 providing that cavitation is achieved quickly and that a large amount of the lubricant is delivered in form of a spray.

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

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One may also say that it is the objective to improve lubrication with SIP injectors or lubrication with a common rail system by accelerating the mass flow in the injection in order to achieve cavitation in as short a time as possible for establishing cavitation in the nozzle tip for a spray to develop.

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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 method according to the invention for lubricating a large combustion engine, for example slow-running two-stroke engine, with a plurality of injectors and wherein the method is arranged for operating within a set of parameters as set forth in the following. The method is defined in claims 1 and 9.

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These objectives are also achieved by a large combustion engine, for example slow-running two-stroke engine, with a plurality of injectors and wherein the engine is arranged for operating within a set of parameters as set forth in the following. The large slow-running combustion engine is defined in claim 2.

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These objectives are also achieved by an injector defined in claim 8.

The lubricant supply conduit is preferably a common rail.

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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.

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The invention may be used in a lubricating principle where the injectors receive pressurised lubrication oil from a lubricator through lubricant supply conduits, one for each injector.

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The invention may also be used in a lubricating principle where the multiple lubricant supply conduits are substituted by a single common lubricant supply conduit. Then the conduit connections 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 lubricant supply conduit in common and simultaneously.

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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.

- 5 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.
- 10 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
- 15 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 $\pi \approx 3.14$. The nozzle has one or more, typically not
- 20 more than two, nozzle exits.

In 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

25 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.

30 Optionally, a controller is provided as an add-on system for upgrade. 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 state 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. Optionally, the engine comprises a controller. As will become more apparent in the following, in advantageous embodiments, the controller is configured to also control the lubricant pressure and optionally also the temperature of the lubricant.

As it was discussed in the introduction, the prior art on lubricant injection draws 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 SIP injection. The parameters for spray injection, in particular SIP injection, have been outside the range where cavitation is achieved.

Only after the studies leading to the invention disclosed in WO 2018/215645, further studies have been performed to find out how to optimize the lubricant injection involving cavitation in the nozzle of the injector.

Thus, in contrast to the conclusions in the prior art and against the trend in the field, the 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 SIP lubrication.

In studies leading to the invention, not only cavitation as such has been found beneficial for spray formation of lubricants, but spray quality, control, and stability is even improved when the cavitation extends to the nozzle exit. It has surprisingly been found that it is possible to improve spray conditions for lubricant injection.

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 according to ISO 5167-1:2022 for pipes with a circular cross-section as shown in (1) below:

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

In (1) $\dot{m}$ is the mass flow, C_d is the discharge coefficient, A_0 is the area of the restriction (D2 in Fig. 3), Y is the expansion coefficient (=1 for incompressible fluids, which is a fair assumption for cylinder lubricant oil under the given conditions) [Ravendran R, Jensen P, De Claville Christiansen J et al. Rheological behaviour 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 before the restriction and β is the diameter ratio, which is the ratio of the diameter of the restriction to the diameter of the pipe $\beta = D2/D1$, as shown for a cross-section of a circular pipe illustrated in Fig. 3.

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When β is small in (1) the resulting flow is small, and when β is large the resulting flow is large. This is true as long as $\beta < 1$ is true. In the case where $\beta \geq 1$ there is not a restriction anymore. By rearranging (1) to what is shown in (2) it can also be seen that the pressure drop over the restriction decreases with increasing β and vice versa.

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$$\Delta P = \frac{1}{2\rho} (1 - \beta^4) \left(\frac{\dot{m}}{C_d A_0 Y} \right)^2 \quad (2)$$

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In the beginning of the nozzle aperture the primary pressure drop is over A1 as shown in Fig. 9. Ideally for the nozzle to operate with a spray, the primary pressure drop must be over A3. The primary pressure drop in the nozzle tip is over A3 if there is no restriction in A1. This is the case when the area of A1 is equal to A2.

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The injectors can “relatively quickly” rise in mass flow. In Fig. 12, it takes approx. 1 ms. to rise to 2.5, 3.5 and 4 bar respectively which can be converted to mass flow as shown in (2). The advantage of rapidly increasing mass flow is that the velocity of the flow also increases. As shown in the Fig. 12, the cavitation number becomes lower at higher speeds.

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In fig. 12, the cavitation number is read on the axis to the right. In general, it may be expected that with a cavitation number of 1.5 and below, the liquid will cavitate and will break up more, i.e. more will be atomized. In some cases, it will be the difference between the oil breaking up or not.

What is important according to the present invention is the time it takes the injector to reach cavitation, i.e. a cavitation number below 1.5.

5 It is an advantage that the oil is atomized, so it is important to control how quickly the mass flow must increase, then you can control the cavitation and thereby also the atomization. It is advantageous not to stay in the area where there is no spray and therefore not a desired effect. In summary it is advantageous to have most of the injected lubricant oil injected as a spray by controlling the mass flow out of the injector.

10 For the oil to cavitate, the cavitation number must be below 1.5. The cavitation number is shown in (3).

$$\sigma = \frac{P_a - P_v}{\frac{1}{2} \rho V^2} \quad (3)$$

15 In (3) 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 (3) 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 $\dot{m} = A \cdot V \cdot \rho$ can be seen from (1) that the cavitation number must decrease
20 with increasing ΔP .

This means that for the nozzle to cavitate it is a requirement, that the needle is pulled back enough for the primary pressure drop to be over the spray hole A3. Otherwise, the velocity of the fluid would not be large enough.

25 The effect of the present invention is to ensure that the nozzle opens quickly. For the nozzle to be able to reach cavitation in a fast manner it requires the needle to be pulled out of its seat fast. To do this, the valve is operated with the following parameters:

Parameter	Max	Min
Mass of needle	0.15 kg	0.02 kg

Resulting force pulling in the needle	10000 N	5 N
Temperature of the oil	200 °C	60°C
Oil dynamic viscosity	0.07302 Pa*s	0.0126818 Pa*s
Oil delivery pressure to valve	300 Bar	20 Bar
Area ratio A1/A3	0.8-85	0

Table 1

Here newtons second law applies:

$$F = m \cdot a \quad (4)$$

- 5 Where F is the sum of forces acting on the needle, which is also known as the resulting force, m is the mass of the object moving and a is the acceleration of the object moving.

Alternatively, the resulting maximum force pulling in the needle may be in the size of 15-20 N. This will especially be the case in situations where piezo-electric actuators are not used.

The limitation of the mass of the needle is found at the largest needle in iron that there is room for, and the smallest needle which has been made in aluminum.

- 15 The temperature is set based on how hot the oil nozzle becomes during operation, moreover the viscosity of the oil is calculated for the two temperatures with the correlation below:

$$\mu [Pa \cdot s] = 0,006 * e^{\frac{1246}{8,31 * T [^{\circ}C]}}$$

- 20 The area ratio is based on a nozzle which has a ratio of 0 when it is closed, and in the open state it is between 0.8 and 85.

- As mentioned above it is wanted to increase the flow as quickly as possible in the valve to achieve cavitation. To have this effected with a greater mass it will be necessary to use a greater force in order for other conditions to be met. There is therefore a limit to how small a force can be used together with the largest mass, and vice versa.

A combination of the parameters given in table 1 must be selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 as defined in equation (6) for the intended operation. This is independent of the mass of the needle. Accordingly, the combination of mass and force shall be combined to fulfill the equation (6).

The rise of the flow rate was also experimentally measured using the Bosch rate of injection.

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.

This can also be shown by using the cavitation number defined in (3) and the equation from the Bosch rate of injection method as explained in “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” shown in (5).

$$\dot{m} = \frac{\Delta P \cdot A_{pipe}}{\alpha} \quad (5)$$

In (5) $\dot{m}$ is the mass flow rate, ΔP Is the pressure difference over the injector, A_{pipe} is the cross-sectional area of the measuring pipe employed in the Bosch rate of injection principle and α is the speed of sound in the fluid. By using the fact that ΔP in (5) can be measured, and both the tube area A and the speed of sound α are constants the mass flow can be calculated. By using that the mass flow is defined as $\dot{m} = V_{pipe} \cdot A_{pipe} \cdot \rho$ where A is the flow area and ρ is the density the velocity in the measuring tube can be calculated. By using incompressibility and continuity: $A_{pipe} \cdot V_{pipe} = A_3 \cdot V_3$ The velocity in the spray hole A_3 can be calculated and used for calculating the cavitation number (3).

This is done and shown in Fig. 11. As can be seen from this Figure, the nozzle is able to reach cavitation after about 1 ms. since cavitation happens when the cavitation number is close to or below 1.5. This is advantageous because the faster the nozzle can cavitate, the faster it reaches its intended operation, which means atomizing the

lubrication oil. The period from 0 to 1 ms. the needle is pulling back, and the mass flow is accelerating. From 1 ms. and onwards the nozzle begins to choke, and the mass flow is becoming steadier until it drops when the nozzle closes.

- 5 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.

Furthermore, it is also important that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s² for the intended operation, as shown in (6),
10 which is a rewritten form of (4).

$$\frac{F}{m} > 50 \left[\frac{m}{s^2} \right] \quad (6)$$

It is also important for the intended operation that for an area ratio equal to or greater than 0.8 the cavitation parameter C defined in WO 2018/215645 is greater than 0, shown
15 in (7).

$$A1 \cdot e^{\left(\frac{B1 \cdot G1}{\ln(G2 \cdot \mu)}\right)} + C1 \cdot e^{\left(\frac{F1 \cdot G1}{\ln(G2 \cdot \mu)}\right)} + A2 \cdot e^{(B2 \cdot P)} + C2 \cdot e^{(F2 \cdot P)} + A3 \cdot D + B3 > 0 \quad (7)$$

20 Where μ is dynamic viscosity, P is pressure and D is diameter of the spray hole. This is consistent with the definition in WO 2018/215645 which is also true for all constants in (7).

Furthermore, the cavitation number depends on the diameter ratio from (1), as shown
25 in (8).

$$\sigma = \frac{P_a - P_v}{\frac{1}{2} \rho \left(\frac{\left(C_d A_0 Y \sqrt{\frac{2 \Delta P \rho}{(1 - \beta^4)}} \right)}{A \cdot \rho} \right)^2} \quad (8)$$

Showing that for a small diameter ratio the cavitation number is large and so no cavitation takes place for a small diameter ratio. The diameter ratio can be replaced by another

geometrical ratio to better describe the geometry, for a nozzle an area ratio could for example be used. For the intended operation it is important that the cavitation number in (8) is also below 1.5.

5 Therefore, the values of the six parameters shown in table 1 is selected so that the cavitation number given by equation (8) is lower than 1.5 and selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.

10 In experimental tests the pressure variation measured by a gauge at the nozzle of the injector is used to calculate the volume flow through the nozzle.

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

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

20 • A rise in mass flow in 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 for lubrication.

25 • The injected amount as a function of ramp time is approximately a straight line, making it 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.

30 • 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.

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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,

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- 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,
- at least one flowmeter for measuring the lubricant flow,

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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,

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- 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,

- and preferably an adjustable valve at the nozzle for opening and closing for flow of lubricant to the nozzle aperture from a pressure chamber in the injector during an injection-cycle,

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wherein the method comprises in cyclic operation.

It is noted that “adjustable” means to the extent that it can be controlled how long time it must be open. However, it is also possible to use other types of valves. Also, how much the needle is pulled may be controlled.

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Optionally the system may also comprise a lubricant return line connecting the lubricant supply with the lubricant injectors.

In a specific embodiment the method of lubricating a large slow-running two-stroke engine 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.

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In a specific embodiment the method of lubricating a large slow-running two-stroke engine includes the step of providing pressure-liquid through a common rail system where all injectors are connected to a common rail.

10 Moreover, the system may comprise:

- providing one flowmeter for all injectors in the engine, one flowmeter for all injectors in a cylinder or one flowmeter for each injector in the engine,
- measuring with the at least one flowmeter the lubricant flow,
- transforming in the controller the measured actual flow into an actual amount,

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Moreover, the system may comprise:

- providing one pressure gauge in the lubricant supply conduit,
- measuring with the at least one pressure gauge the pressure in the lubricant in lubricant supply conduit,

20 - transforming in the controller the measured pressure to a mass flow.

The measurement of the pressure in lubricant supply conduit may be used in the controller to ensure that a desired pressure is established by regulating the pump.

25 However, experiments shows that pressure variation in the cylinder is very limited when the lubricant is injected. In operation the pressure in the cylinder will follow a so-called banana formed curve. The increase in pressure during initial stroke and the time of the injection is limited. Accordingly, the need for controlling and regulating the pressure is limited.

30

Alternatively, a pressure relief valve may be used to ensure that the pressure is maintained at the desired level in the lubricant supply conduit. Such pressure relief valve will be calibrated and adjusted before operation to the desired pressure for the injectors to be used.

So, more methods may be used for keeping a desired constant pressure in the lubricant supply conduit.

Moreover, the system may comprise:

- 5 - operating within the defined parameters to ensure that the nozzle opens quickly for accelerating the mass flow in the injection.

Hereby it is possible to achieve cavitation in a short time as possible for establishing cavitation in the nozzle tip for a spray to develop quickly.

10

Such time may be less than 2 ms. Or preferably less than 1 ms.

The controller may be built into the injector or may be connected to the injector.

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

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

20

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.

25

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.

30

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;
- 5 - 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
- 10 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.

Moreover, the engine may comprises

- 15 - a pressure-control port flow-connected to a 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;
- 20 - 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
- 25 its motion in the injection phase.

The engine may further comprise 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.

5 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.

10

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.

15

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.

20

The engine and the method may comprise the hydraulically driven inlet-valve system known from WO 2019/114905.

25

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

30

- 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.

5 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
10 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
15 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.

20 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
25 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
30 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
5 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
10 lubricant supply from the lubricant inlet port to the conduit.

The engine and the method comprising the electrically driven inlet-valve system is known from WO 2019/114903. However, the engine differs as it is arranged to operate within the parameters given in table 1.

15 Where, the values of the six parameters are selected so that the cavitation number given by equation (8) is lower than 1.5 and where a combination of the parameters given in table 1 is selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.

20 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.

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

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
30 more sophisticated model-based regulation.

The method and the engine according to the invention is especially suitable for use for 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 SIP injection and injection into the ring pack.

The invention also relates to an injector for a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a lubricant system comprising a controller and which 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 injector is arranged for in cyclic operation

- in the injection phase, providing pressure-liquid through a common rail system where all injectors are connected to the lubricant supply conduit and wherein the injector is arranged for operating within the parameters given in table 1.

Where, the values of the six parameters are selected so that the cavitation number given by equation (8) is lower than 1.5 and where a combination of the parameters given in table 1 is selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.

The injector may further comprise

- an adjustable valve at the nozzle for opening and closing for flow of lubricant to the nozzle aperture from a pressure chamber in the injector during an injection-cycle.

The injector is intended for use in the method or in the engine according to the invention.

Especially the injector is for use for 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.

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.

5 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 - however is embedded in the cylinder wall to ensure free movement of the piston - through the cylinder wall, when the injector is properly mounted, the nozzle itself, optionally, has more than a single aperture. For
10 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.

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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.

20 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.

25 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
30 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.

5 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.

10 The term “pressurized lubricant” is used for lubricant provided at a pressure high enough that it can be used for injection as an atomized spray into the cylinder. For 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.

15 The term “pressure gauge” is used for a component which is able to measure a pressure in lubricant oil independent of the method used, e.g. also for determining of pressure difference.

20 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.

25 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 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 lubricant oils, and one is used for an additive. The injectors comprise a nozzle having one or more nozzle apertures for injecting lubricant oil into the cylinder in form of a spray.

The engine further comprises a controller. The controller is configured for controlling the pressure in the lubricant oil and mass flow of lubricant oil.

The controller may also be 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 state 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 through the lubricant supply conduit.

5

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

10 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.

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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
20 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
25 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.

30 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 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
5 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
10 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.

15 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
20 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
30 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,
- 5 - 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,

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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

15 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.

20

Optionally, 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

25 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.

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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
5 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
10 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
15 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
20 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.
25 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
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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.

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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

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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.

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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.

20

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

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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-

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valve system assists in a short closing time after injection, adding to precision in timing and volume of injected lubricant.

5 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.

10

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.

20

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.

25

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

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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
5 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
10 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
15 from the valve system to the outlet-valve system.

Optional details of valve and actuator

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
20 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.

25 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
30 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.

5 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.

10

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
15 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.

20 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. With piezo-electric actuators very high forces may be obtained. For example, in the size of 10000 N. The force may also be higher as this will result in a shorter time for obtaining the operational situation
25 with cavitation in the injection and thereby a very quick start of the injected lubricant oil to be in form of a spray.

In some concrete embodiments, the valve member is cylindrical and comprises a stationary valve member, which, in turn, comprises a corresponding cylindrical bushing
30 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 an 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.

5 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.

10 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 in 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.

15 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
20 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.

25 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.

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Optional parameters

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 π 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. When injecting the lubricant oil a cavitation is obtained in the lubricant oil.

- 5 A spray of atomized droplets is 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
- 10 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
- 15 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.
- 20 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
- 25 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
- 30 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>.

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 sketch of a circular pipe with an area of restriction,
- 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 partially enlarged sketch of the front part of an injector,
- Fig. 10 is a graph illustrating the relation between time and mass flow,
- Fig. 11 is a graph illustrating the relation between mass flow, cavitation number and time,
- Fig. 12 is a graph illustrating the relation between pressure, time and cavitation number,
- Fig. 13 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,
- Fig. 14 is a sketch of the injector in Fig 13, however illustrated in open position,
- Fig. 15 is a first graph illustrating limit for relation force to mass, and
- Fig. 16 is a second graph illustrating limit for relation force to mass.

30 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.

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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 bars, for example 25 to 100 bars, optionally 30 to 80 bars or even 50 to 80 bars, 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 directed towards the cylinder liner.

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 the lubricant supply conduit 9, 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.

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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.

5 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.

10 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
15 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.

20 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.
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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
30 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”.

5 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.

15 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.

20 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.

25 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.

30 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
5 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.

10 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
15 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 re-
20 turning 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 in turn is filled by lubricant from the rear chamber 16A that received the lubricant through the lubricant inlet port 4A.

25 For proper functioning, the lubricant inlet port 4a is provided with lubricant from the lubricant supply 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.

30 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 line 9 or even lower.

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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.

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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.

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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 lubricant supply conduit 9. In case of the controller 11 being an add-on unit, the controller 11 would have at least 4 conduit connectors.

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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 lubricant 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 lubricant supply conduit 9. In this case, the return conduit 13 in extension of the

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lubricant 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.

FIG. 4 illustrates a second, alternative embodiment of an outlet-valve system 15. The generalised principle of the outlet-valve system 15 is 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. A nozzle aperture 5' is provided in the nozzle 5 tip for ejection of lubrication oil. Inside a cavity 40 of the nozzle 5, an outlet-valve member 18 is provided, the outlet-valve member 18 comprising a stem 41 and a cylindrical sealing head 42 which is arranged slidingly in a cylindrical cavity part 43 at the nozzle tip 44. The position of the valve member 18 is pre-stressed backwards away from the nozzle tip 44 by a spring 45 and is offset forwards by oil pressure acting through a channel 46 upon the back part 47 of the stem 41, the oil pressure acting against the spring force. The nozzle aperture 5' is sealingly covered by the sealing head 42 which abuts the cylindrical cavity part 43 at the nozzle tip 44, unless the valve member 18 is pushed forward such that the sealing head 43 slides pass and away from the nozzle aperture 5' to allow lubricant oil to flow from the inner cavity 46 through the nozzle aperture 5' for ejection.

The following values are non-limiting illustrative examples of possible working pressures. The pressure in the return conduit 13 and the lubricant supply conduit 9 is 10 bar. The pressure in the supply conduit 12 is 40 bars. The outlet-valve 15 opens at 37 bar, such that the lubricant is injected at 37 bars. The springs 28A and 29B are configured for pressing the plunger-member 29 and the actuator-member 28 fully back to the rear-most 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 bars, 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 lubricant supply conduit 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.

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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.

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The above-described embodiments are known from WO 2019/114905. However, the engine differs as it is arranged to operate within the parameters given in table 1.

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Where, the values of the six parameters are selected so that the cavitation number given by equation (8) is lower than 1.5 and where a combination of the parameters given in table 1 is selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.

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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.

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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 bars, for example 25 to 100 bars, optionally 30 to 80 bars or even 50 to 80 bars, 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 directed towards the cylinder liner.

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, typically through a common lubricant supply conduit 9, from a lubricant supply 25, for example oil circuit, of the engine including a potential 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 25 to 100 bars, optionally 30 to 80 bars, which is a typical range of pressure for 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.

5 A flow meter 35 is connected with the injector 4 and the electrical cables 110 and is arranged for measuring the flow. This is as explained above in connection with Fig. 1

10 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.

15 The above-described embodiment is known from WO 2019/114903. However, the engine differs as it is arranged to operate within the parameters given in table 1.

20 Where, the values of the six parameters are selected so that the cavitation number given by equation (8) is lower than 1.5 and where a combination of the parameters given in table 1 is selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.

25 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.

30 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.

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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.

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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.

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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 Figs 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 supply 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.

5

The injector is of a type generally described in WO 2012/126473. The injector can be actuated electromechanically, for example in the form of a solenoid valve or piezomechanical element.

10 Fig. 13 and Fig 14 show a further embodiment of an injector 4 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. 13 and 14 is actuated electromechanically in the form
15 of a solenoid operated opening /closing valve 213.

The injector is made as a unit. The opening/closing valve 213 is an electromechanical valve integrated in the injector 4 for dosing the lubricating oil. The electromechanical opening/closing valve 213 includes a spring-biased push member 231 acting on an outlet valve member 18. The outlet valve member 18 cooperates with a valve seat.
20

Dosing of lubricating oil is performed by activating the opening/closing valve 213 in the injector 4 for dosing the lubricating oil. The activation moves the push member 231 of the opening/closing valve 4 and controls the injection of the lubricating oil.
25

Figs. 15 and 16 illustrate two graphs of the relation between the resulting pulling force and the mass of the needle. The resulting pulling force is shown along the x-axis and the mass of the needle is shown along the y-axis.

30 As mentioned earlier it is an object to increase the flow as quickly as possible in the valve to achieve cavitation. However, there is a limit to how small a force can be used together with the largest mass, and vice versa. The graphs in Figs 15 and 16 is used to illustrate the possible combination of parameters given in Table 1.

The graphs show the same. However, Fig.15 shows only the lower values of force with associated mass, and Fig 16 shows the whole range of forces from 5N to 10.000N with logarithmic division of the x-axis.

- 5 From table 1 the force used to move the needle with the largest mass could be seen to be at least 7.5N to fulfill equation (6) when selecting the needle with the largest mass. However, if using a smaller needle and thus a lower mass, a smaller force will also fulfill equation (6). The relation is illustrated by the sloping part of the graph from 5N to 7.5N

10

Therefore, the minimum force in table 1 is set to be 5N. And the combination of the parameters given in table 1 must be selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.

- 15 Figures 3 and 9-12 are used for explanation in the section:” DESCRIPTION / SUMMARY OF THE INVENTION”.

Numbering

- 1 cylinder
- 20 2 cylinder liner
- 3 cylinder wall
- 4 injector
- 4' injector housing
- 4A inlet port of oil injector 4
- 25 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
- 30 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
- 5 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
- 10 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
- 15 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
- 20 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
- 25 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
- 30 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
- 5 30C toggle-valve return port
- 31 pressure-valve
- 31A pressure-valve inlet port
- 31B pressure-valve return port
- 31C pressure regulator, ex. Spring-loaded pressure adjustment member injection-phase
- 10 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
- 15 34 return exit line from controller 11 to return conduit 13
- 35 flow meter
- 112 lubricant inlet port of injector 4
- 113 inlet-valve system of injector 4
- 114 swirl in cylinder
- 20 115 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
- 121 inlet-valve housing of inlet-valve system 115
- 122 O-ring at end of flow chamber 116
- 25 123 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
- 30 128 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
- 5 203 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
- 10 212 injector signal bus
- 213 opening/closing valve
- 231 push member

CLAIMS

1. A method for lubricating a large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a system comprising
- 5 - a lubricant supply,
- 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,
- 10 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
- 15 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 includes the step of providing for the engine
- an adjustable valve comprising a valve member and a valve seat at the nozzle (5) for
- 20 opening and closing for flow of lubricant through a sac hole to the nozzle aperture (5') from a pressure chamber in the injector during an injection-cycle,
- wherein the nozzle aperture has a cross section area A3, and wherein the sac hole has a cross section area A2 and wherein a cross section area A1 is provided between the valve member and the valve seat, and
- 25 wherein the method comprises in cyclic operation
- in the injection phase, providing pressure-liquid to the lubricant supply conduit and **characterised in that** the method is arranged for operating within the following parameters:

Parameter	Max	Min
Mass of needle	0.15 kg	0.02 kg
Resulting force pulling in the needle	10000 N	5 N
Temperature of the oil	200 °C	60°C

Oil dynamic viscosity	0.07302 Pa*s	0.0126818 Pa*s
Oil delivery pressure to valve	300 Bar	20 Bar
Area ratio A1/A3	0.8-85	0

and selecting the values of the six parameters so that the cavitation number

$$\sigma = \frac{P_a - P_v}{\frac{1}{2} \rho \left(\frac{\left(C_d A_0 Y \sqrt{\frac{2 \Delta P \rho}{(1 - \beta^4)}} \right)}{A \cdot \rho} \right)^2} < 1.5$$

5 and selecting a combination of the parameters given in the above table so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s² for the intended operation.

2. A large slow-running two-stroke engine comprising a cylinder with a reciprocal piston inside and with a system comprising
- 10 - a lubricant supply,
- 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,
- 15 - 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 to which the controller is connected,
- 20 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 comprising a valve member and a valve seat at the nozzle (5) for opening and closing for flow of lubricant through a sac hole to the nozzle aperture (5') from a pressure chamber in the injector during an injection-cycle,
- wherein the nozzle aperture has a cross section area A3, and wherein the sac hole has a cross section area A2 and wherein a cross section area A1 is provided between the valve member and the valve seat,
- wherein a use of the engine comprises in cyclic operation
- in the injection phase, providing pressure-liquid to the lubricant supply conduit and **characterised in that** the engine is arranged for operating within the following parameters:

Parameter	Max	Min
Mass of needle	0.15 kg	0.02 kg
Resulting force pulling in the needle	10000 N	5 N
Temperature of the oil	200 °C	60°C
Oil dynamic viscosity	0.07302 Pa*s	0.0126818 Pa*s
Oil delivery pressure to valve	300 Bar	20 Bar
Area ratio A1/A3	0.8-85	0

and wherein the values of the six parameters are selected so that the cavitation number

$$\sigma = \frac{P_a - P_v}{\frac{1}{2} \rho \left(\frac{\left(C_d A_0 Y \sqrt{\frac{2 \Delta P \rho}{(1 - \beta^4)}} \right)}{A \cdot \rho} \right)^2} < 1.5$$

- and wherein a combination of the parameters given in the above table is selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s² for the intended operation.

3. The large slow-running two-stroke engine according to claim 2, **characterised in that** the engine further comprises
- a computer (11') to which the controller is connected, or alternatively

- a mobile phone arranged to communicate with the controller.

4. The large slow-running two-stroke engine according to claim 2 or 3, **characterised in that** the engine includes a common rail system where all injectors are connected to a common rail.

5. The large slow-running two-stroke engine according to any one of claims 2 – 4, **characterised in that** the lubricant system is chosen from mechanically driven systems, hydraulically driven systems and common-rail systems.

6. The large slow-running two-stroke engine according to any one of claims 2 – 5, **characterised in that** the engine comprises a hydraulically driven inlet-valve system.

7. The large slow-running two-stroke engine according to any one of claims 2 – 5, **characterised in that** the engine comprises an electrically driven inlet-valve system.

8. An injector for a large slow-running two-stroke engine comprising a cylinder (1) with a reciprocal piston inside and with a lubricant system comprising a controller and which 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 comprising a valve member and a valve seat at the nozzle (5) for opening and closing for flow of lubricant through a sac hole to the nozzle aperture (5') from a pressure chamber in the injector during an injection-cycle,

wherein the nozzle aperture has a cross section area A_3 , and wherein the sac hole has a cross section area A_2 and wherein a cross section area A_1 is provided between the valve member and the valve seat, and

wherein the injector is arranged for in cyclic operation

- in the injection phase, providing pressure-liquid through a common rail system where all injectors are connected to the lubricant supply conduit and **characterised in that** the injector is arranged for operating within the following parameters:

Parameter	Max	Min
Mass of needle	0.15 kg	0.02 kg
Resulting force pulling in the needle	10000 N	5 N
Temperature of the oil	200 °C	60°C
Dynamic viscosity	0.07302 Pa*s	0.0126818 Pa*s
Oil delivery pressure to valve	300 Bar	20 Bar
Area ratio A1/A3	0.8-85	0

and wherein values of the six parameters are selected so that the cavitation number

$$\sigma = \frac{P_a - P_v}{\frac{1}{2} \rho \left(\frac{C_d A_0 Y \sqrt{\frac{2 \Delta P \rho}{(1 - \beta^4)}}}{A \cdot \rho} \right)^2} < 1.5$$

5 and wherein a combination of the parameters given in the above table is selected so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s² for the intended operation.

9. 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
- 10 - a lubricant supply (25),
- 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 (12) connecting the lubricant supply (25) with the lubricant
- 15 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) is an electromechanical activated injector and comprises

- an inlet port flow-connected to the lubricant supply conduit (12) 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,
- 5 wherein the method includes the step of providing
- an adjustable valve comprising a valve member and a valve seat at the nozzle (5) for opening and closing for flow of lubricant through a sac hole to the nozzle aperture (5') from a pressure chamber in the injector,
- wherein the nozzle aperture has a cross section area A3, and wherein the sac hole has a
- 10 cross section area A2 and wherein a cross section area A1 is provided between the valve member and the valve seat, and
- wherein the method comprises in cyclic operation
- in the injection phase, providing pressure-liquid through a common rail system where all injectors are connected to the lubricant supply conduit and **characterised in that**
- 15 the injector is arranged for operating within the following parameters:

Parameter	Max	Min
Mass of needle	0.15 kg	0.02 kg
Resulting force pulling in the needle	10000 N	5 N
Temperature of the oil	200 °C	60°C
Oil dynamic viscosity	0.07302 Pa*s	0.0126818 Pa*s
Oil delivery pressure to valve	300 Bar	20 Bar
Area ratio A1/A3	0.8-85	0

and selecting the values of the six parameters so that the cavitation number

$$\sigma = \frac{P_a - P_v}{\left(\frac{1}{2} \rho \left(\frac{C_d A_0 Y \sqrt{\frac{2 \Delta P \rho}{(1 - \beta^4)}}}{A \cdot \rho} \right)^2 \right)} < 1.5$$

and selecting a combination of the parameters given in the above table so that the ratio of the resulting force acting on the needle to the mass of the needle is above 50 m/s^2 for the intended operation.