(19) DANMARK

(10) DK/EP 3015679 T3

(12) Oversættelse af europæisk patentskrift

Patent- og Varemærkestyrelsen

(51) Int.Cl.: *F 02 B 75/02 (2006.01)* *F 02 B 23/10 (2006.01)* *F 02 D 19/06 (2006.01)* *F 02 M 21/02 (2006.01)*

(45) Oversættelsen bekendtgjort den: **2019-03-11**

(80) Dato for Den Europæiske Patentmyndigheds bekendtgørelse om meddelelse af patentet: **2018-12-05**

(86) Europæisk ansøgning nr.: **15190165.9**

(86) Europæisk indleveringsdag: **2015-10-16**

(87) Den europæiske ansøgnings publiceringsdag: **2016-05-04**

(30) Prioritet: **2014-10-31 EP 14191343**

(84) Designerede stater: **AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR**

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(54) Benævnelse: **Cylinder til en stempelforbrændingsmotor, stempelforbrændingsmotor samt fremgangsmåde til drift af en stempelforbrændingsmotor**

(56) Fremdragne publikationer:
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A large engine and a method of operating such a large engine

The invention relates to a large engine, configured as a uniflow-scavenged two-stroke large diesel engine or as a gas engine, having a cylinder and a gas supply system as well as to a method of operating such a large engine in accordance with the preamble of the independent claim of the respective category.

Large diesel engines are frequently used as drive units for ships or also in stationary operation, e.g. for the driving of large generators for the generation of electrical energy. In this respect, the engines as a rule run in permanent operation over considerable time periods, which make high demands on the operating security and on the availability. In particular long servicing intervals, low wear and an economic handling of fuels and operating materials are therefore central criteria for the operation of the machines for the operator. The piston running behavior of such large-bore slow running diesel engines is, among others, a determining factor for the length of the servicing intervals, the availability and, over the lubricant consumption, also directly for the operating costs and thus for the operating efficiency.

A further essential point having increasing importance for some years is the quality of the emissions, in particular the nitrogen oxide concentration in the emissions. The legal provisions and the limit values for the corresponding exhaust emission standards have been made increasingly stricter here and will also be made even stricter in the near future. This in particular has the consequence with two-stroke large diesel engines that the combustion of the classical heavy oil, which is heavily contaminated with pollutants, but also the combustion of diesel oil or other fuels

will become increasingly more problematic because the observation of the exhaust emission standards will become more and more difficult, technically more complex and thus more expensive or because ultimately their observation is even no longer sensibly possible so that the corresponding engines will have to be taken out of operation or at least converted in a complex and/or expensive manner.

In practice, there has therefore already long been a need for so-called "dual-fuel" engines, i.e. engines, which can be operated using two different fuels. In this respect, on the one hand, gas, e.g. in the form of a natural gas, for example a so-called "liquefied natural gas" (LNG), or a gas in the form of a liquefied petroleum gas or another gas suitable for driving an internal combustion engine should frequently be combusted and, on the other hand, another fuel such as petrol, diesel, heavy fuel oil or another suitable liquid fuel can be combusted in one and the same engine. In this respect, the engines can be both two-stroke engines and four-stroke engines and they can, in this respect, be small engines, medium-sized engines or also large engines, in particular also uniflow-scavenged two-stroke large diesel engines, such as are used as drive units in ships, but are also frequently used for producing electrical energy in power stations.

However, purely gas engines, i.e. engines which can only be operated with gas and not alternatively with diesel, heavy oil or another fuel, are also in increasing demand, in particular when high exhaust emission standards are required which can only be economically sensibly observed with a justifiable technical effort by the combustion of gas. Such a purely gas engine is set forth, for example, in WO 2010 147071 A1. Further prior art can e.g. be found in DE 10 2010 005814 A1.

From WO 2013/007863, a two-stroke large engine with a gas supply system is known, whereby the gas inlet into the cylinder in the cylinder wall (or in the cylinder liner) is arranged between the scavenging air openings and the cylinder cover.

Irrespective of whether it is a dual fuel engine or a purely gas engine, the process of the introduction of the fuel gas into the combustion space of the cylinder of a corresponding reciprocating piston internal combustion engine is of decisive importance for the reliable and safe operation of such an engine.

The gas supply system for supplying the fuel gas into the combustion space of the reciprocating piston internal combustion engine in this respect has a plurality of different functions, which additionally influence one another in a certain manner so that the different part functions have to be exactly coordinated with one another. In addition, the exact configuration of the gas supply system can e.g. inter alia depend on a given or preferred operating state, for example whether the engine is mostly operated in the full-load range or in the part-load range or whether a change frequently has to be made between these modes of operation. Or whether it is e.g. a dual fuel engine or a purely gas engine. Or the specific configuration of the gas supply system can also depend on the construction size of the engine, on its power, cubic capacity, preferred revolution range or on other construction variants or operating states of a reciprocating piston internal combustion engine familiar per se to the skilled person.

In this respect, in particular with two-stroke large diesel engines, the gas supply system not only has the function of metering the quantity of gas to be introduced into the combustion space of the engine as is frequently a sufficient demand with four-stroke engines. With two-stroke large diesel engines, in addition to further demands, the correct and reliable mixture of the gas introduced by the gas supply system with the scavenging air is inter alia a decisive point.

A further major point relates to the operating safety of the engine. In the event that the gas supply system fails totally or in part, it is possible that e.g. much too large a quantity of gas is introduced into the combustion space of the engine if no suitable safety measures are taken to prevent it. Such excessively large amounts

of gas in the combustion space of the engine can e.g. result in uncontrolled explosions in the exhaust gas system or in the scavenging system of a large diesel engine, which can have the result that the corresponding components overheat or are even damaged so that in the worst case the whole engine can fail, which can in particular have catastrophic consequences on the high seas and can even result in the loss of the whole ship.

But even if the consequences of such a malfunction are not so dramatic, less serious disturbances of this kind can also bring about unwanted disadvantages.

If, for example due to a defect, the gas supply system respectively introduces too large a quantity or too small a quantity of gas into the combustion space of the engine, this can have a considerably negative effect on the emission values such as the nitrogen oxide content or other emission values.

Or, however, on an operation with a gas, e.g. with the above-mentioned LNG, an incomplete combustion and thus a phenomenon known to the skilled person as CH_4 slip, which has values which are too high, can result on a malfunction of the gas supply system. If a corresponding engine is now operated with gas, that is in a gas mode, the exhaust gases arising from a malfunction of the gas supply system on the combustion can inter alia include not insubstantial quantities of methane and/or formaldehyde emissions which can, for example result from a spontaneous cooling of the combustion flame ("flame quenching") on the combustion procedure at the cylinder wall or can be released in a phase in which the scavenging of the cylinder with fresh air takes place with a simultaneously not closed outlet valve or which can be released in a different manner.

Similar effects can occur if the gas is introduced into the combustion space in an unsuitable manner, e.g. at an unsuitable pressure or in an unsuitable geometry of

the injection jet, at an unsuitable angle, at an unsuitable position or is introduced into the combustion space of the engine unsuitably in another manner.

If e.g. the methane concentration in the exhaust gas is correspondingly high due to such operating states, the safe operation of the engine can likewise thereby be impaired. There is, for example, the risk that the methane ignites in the exhaust gas system, e.g. in the exhaust gas manifold or in a region before or even in the turbo-charger. In the worst case, the ignition of the methane in the exhaust gas system can result in damage to corresponding components, which are exposed to the harmful effects of the methane gas ignition. In addition, methane is an extremely effective greenhouse gas which is, as is known, at least 25 times more effective than carbon dioxide, which can have a significant negative influence on the calculation of the so-called "energy efficiency design index" (EEDI).

The frequently used and very sensitive oxidation catalysts can also be substantially impaired by a malfunction of the exhaust gas supply system and by the consequences resulting from this and can in the worst case be irreparably damaged after only a short time, whereby the further safe and reliable operation of the engine is at least questionable and in the worst case made impossible.

It is thus the object of the invention to provide a large engine, which is configured as a uniflow-scavenged two-stroke large diesel engine or as a gas engine, having a cylinder and a gas supply system, wherein the gas supply system can be flexibly adapted to different engines or to different operating states and demands such that it can also be possible to retrofit in older engines in specific cases or such that existing engines can be converted or adapted flexibly and simply to other operating modes or operating states. In this respect, the safe introduction of the gas into the combustion space of the reciprocating piston internal combustion engine should be reliably ensured and an ideal mixture with the scavenging air should in particular, but not only be ensured in the combustion space in the case

of uniflow-scavenged two-stroke large diesel engines. In addition, the gas supply system and its components should be reliably secured against leaking.

The subject matters of the invention satisfying these objects are characterized by the features of the independent claims 1 and 14.

The dependent claims relate to particularly advantageous embodiments of the invention.

The invention thus relates to a large engine, configured as a uniflow-scavenged two-stroke large diesel engine or as a gas engine, having a cylinder with a gas supply system, wherein the gas supply system comprises a gas inlet valve having a gas inlet nozzle, which is configured such that it is arranged in a cylinder wall of a cylinder of the large engine such that a fuel gas provided as fuel can be supplied to a combustion space of the cylinder by means of the gas inlet nozzle in the installed state of the gas supply system, wherein the gas supply system is provided in a region of the cylinder wall between a top dead center and a bottom dead center of a piston of the large engine. In this respect, the gas inlet valve comprises a pressure space arranged in a valve housing, in which pressure space the fuel gas can be provided via a gas supply for storage and supply into the gas inlet nozzle in the operating state. A valve body having a valve disk arranged at a valve shaft and having a valve seat is provided in the pressure space, said valve seat sealingly cooperating with the valve disk in a closed state of the valve body such that the supply of the fuel gas from the pressure space into the gas inlet nozzle is inhibited and the valve disk can be raised from the valve seat by means of a valve drive in operative connection with the valve shaft so that the fuel gas can be supplied to the gas inlet nozzle from the pressure space past the valve disk in an open state of the valve body. The gas inlet nozzle is connected releasably to the gas inlet valve and replaceably to the cylinder wall and a nozzle axis of the gas inlet nozzle is arranged at a predefinable angle with respect to a valve axis of the

gas inlet valve such that the fuel gas can be injected into the combustion space of the cylinder in the operating and installed state of the gas supply system at a predefinable angle differing from zero with respect to a radial direction and/or with respect to an axial direction of the cylinder.

That angle is then called the injection angle which is the supplementary angle to the angle between the nozzle axis of the gas inlet nozzle and the valve axis of the gas inlet valve, that is the injection angle and the angle between the nozzle axis and the valve axis complement each other to form 180°. The injection angle preferably amounts to an angle between 10 degrees and 80 degrees, particularly preferably between 10 degrees and 35 degrees; the injection angle is specifically approximately 22.5 degrees.

A gas supply system of a large engine in accordance with the invention thus comprises a valve, which has a valve shaft with a valve disk as essential elements, wherein the desired quantity of gas being introduced into the combustion space by the usually hydraulically controlled valve via the valve disk, which sealingly cooperates with a valve seat in the closed state, and possibly by nozzles and a suitable nozzle channel.

A gas supply system is thus available for the first time for supplying the fuel gas into the combustion space of a large engine, which has a previously unknown flexibility due to its two-part design. Since the gas inlet nozzle is connected releasably to the gas inlet valve and replaceably to the cylinder wall, the gas inlet nozzle can, for example, be replaced separately from the gas inlet valve in dependence on the demands on specific combustion conditions in the combustion space of the large engine so that different injection conditions for injecting the fuel gas into the combustion space can be implemented simply and can be coordinated ideally to changing demands.

Furthermore, a nozzle axis of the gas inlet nozzle of the present invention is arranged at a predefinable angle with respect to a valve axis of the gas inlet valve such that the fuel gas can be injected into the combustion space of the cylinder in the operating and installed state of the gas supply system at a predefinable injection angle differing from zero. This is of particular importance because the two main functions of the gas inlet nozzle are thereby considered simultaneously: on the one hand, an exact metering is possible under the most varied operating conditions by using a gas inlet nozzle adapted to the conditions in the combustion space. And, on the other hand, an ideal mixture of the scavenging air and the fuel gas can be achieved by the correct selection of the injection angle and by the correct choice of the geometry of the injection jet or of a plurality of injection jets so that an ideal combustion is ensured so that in particular the required emission standards can be observed and, on the other hand, the consumption of fuel can be minimized.

This means that a plurality of different functions, which moreover influence one another in a certain manner, can be exactly coordinated with one another for the first time by the present invention. It is thus possible e.g. by the exact design of the gas supply system, specifically by the choice of a suitable gas inlet nozzle, to focus ideally inter alia on a given or preferred operating state, for example whether the engine is usually operated in the full-load range or in the part-load range or whether a change frequently has to be made between these operating modes. Or the gas inlet nozzle can be ideally selected in dependence on whether it is a dual fuel engine or a purely gas engine. Or the specific design of the gas supply system can also e.g. for the first time be flexibly set to the construction size of the engine, its power, cubic capacity, preferred revolution range or other construction variants or operating states of the reciprocating piston internal combustion engine familiar per se to the skilled person without the total gas supply system having to be adapted or replaced. Only e.g. the gas inlet nozzle thus has to be suitably selected and correspondingly replaced by using the gas supply system of the large engine

in accordance with the invention, while the gas inlet valve as such can be maintained.

In this respect, in particular with two-stroke large diesel engines, the gas supply system not only has the function of metering the quantity of gas to be introduced into the combustion space of the engine as is frequently a sufficient demand with four-stroke engines. With two-stroke large diesel engines, as already mentioned, in addition to further demands, the correct and reliable mixture of the gas introduced by the gas supply system with the scavenging air is inter alia a decisive point which is considered in full for the first time by the gas supply system in accordance with the invention.

As a further measure, in accordance with the invention, for improving the mixture of the fuel gas with the scavenging air or to further optimize the introduction of the fuel gas into the cylinder, the gas inlet nozzle can also have a plurality of nozzle openings.

It is also possible that two nozzle openings are aligned so differently that the fuel gas can be introduced into the combustion space at two different injection angles, whereby e.g. a better distribution in the combustion space can be achieved.

Or alternatively or simultaneously, however, two nozzle openings can be designed differently such that the fuel gas can be introduced into the combustion space through the two nozzle openings with two different injection quantities and/or at two different flow speeds and/or with two different jet geometries.

A further major point relates to the operating safety of the engine. In the event that the gas supply system fails totally or in part, it is possible that e.g. much too large a quantity of gas is introduced into the combustion space of the engine if no suitable safety measures are taken to prevent it. Such excessively large amounts

of fuel gas in the combustion space of the engine can e.g. result in uncontrolled explosions in the exhaust gas system or in the scavenging system of a large diesel engine, which can have the result that the corresponding components overheat or are even damaged so that in the worst case the whole engine can fail, which can in particular have catastrophic consequences on the high seas and can even result in the loss of the whole ship.

But even if the consequences of such a malfunction are not so dramatic, less serious disturbances of this kind can also bring about unwanted disadvantages.

If, for example due to a defect, the gas supply system respectively introduces too large a quantity or too small a quantity of gas into the combustion space of the engine, this can have a considerably negative effect on the emission values such as the nitrogen oxide content or other emission values.

Or, however, on an operation with a gas, e.g. with the above-mentioned LNG, an incomplete combustion and thus a phenomenon known to the skilled person as CH_4 slip, which has values which are too high, can result on a malfunction of the gas supply system. If a corresponding engine is now operated with gas, that is in a gas mode, the exhaust gases arising from a malfunction of the gas supply system on the combustion can inter alia include a not insubstantial quantity of methane and/or formaldehyde emissions which can, for example result from a spontaneous cooling of the combustion flame ("flame quenching") on the combustion procedure at the cylinder wall or can be released in a phase in which the scavenging of the cylinder with fresh air takes place with a simultaneously not closed outlet valve or which can be released in a different manner.

Similar effects can occur if the gas is introduced into the combustion space in an unsuitable manner, e.g. at an unsuitable pressure or in an unsuitable geometry of

the injection jet, at an unsuitable angle, at an unsuitable position or is introduced into the combustion space of the engine unsuitably in another manner.

If e.g. the methane concentration in the exhaust gas is correspondingly high due to such operating states, the safe operation of the engine can likewise thereby be impaired. There is, for example, the risk that the methane ignites in the exhaust gas system, e.g. in the exhaust gas manifold or in a region before or even in the turbo-charger. In the worst case, the ignition of the methane in the exhaust gas system can result in damage to corresponding components, which are exposed to the harmful effects of the methane gas ignition. In addition, methane is an extremely effective greenhouse gas which is, as is known, at least 25 times more effective than carbon dioxide, which can have a significant negative influence on the calculation of the so-called "energy efficiency design index" (EEDI).

The frequently used and very sensitive oxidation catalysts can also be substantially impaired by a malfunction of the exhaust gas supply system and by the consequences resulting from this and can in the worst case be irreparably damaged after only a short time, whereby the further safe and reliable operation of the engine is at least questionable and in the worst case made impossible.

These in part very serious problems can also be reliably avoided for the first time by the present invention.

For this purpose, a gas supply system of a large engine in accordance with the invention for monitoring a fuel gas flow into the combustion space can comprise a control system for determining a position of the valve body, wherein the control system is a path sensor, in particular an electrical or electromagnetic path sensor, specifically an inductive, capacitive or optical path sensor, so that the respective position of the valve body, in particular whether the valve body is open or closed, can be reliably detected at any time.

The control system is in this respect particularly preferably in signal communication with a monitoring unit such that the position of the valve body in the operating state can be detected by the monitoring unit and thus a fuel gas supply to the gas inlet nozzle can be inhibited in dependence on a crank angle of the large engine and/or in dependence on a position of a gas exchange valve of the large engine. It is thus possible for the first time by the invention not only to detect a malfunction of the gas supply system, but also to set a safety protocol into operation in dependence on the malfunction so that e.g. the fuel gas supply to the gas supply system is automatically inhibited and the large engine is preferably automatically switched off, particularly preferably automatically switched to an operation with an alternative fuel such as diesel oil or heavy oil.

The gas inlet system is in this respect preferably configured such that the valve drive for driving the valve shaft is a mechanical, electrical or pneumatic valve drive, in particular a hydraulic valve drive, wherein particularly preferably a restoration device, in particular in the form of a restoration spring, acting against the valve drive is provided for closing the valve body.

The valve shaft is guided in a specific embodiment particularly important for practice in a guide bore of a shaft housing. In this respect, the guide bore can be applied with a hydraulic oil which is at a sealing pressure for sealing against the gas pressure of the fuel gas in the pressure space, said sealing pressure preferably being higher than a fuel gas pressure of the fuel gas in the pressure space so that a penetration of the fuel gas into the guide bore or a flowing of the fuel gas through the guide bore can be substantially prevented.

The fact that the sealing pressure is preferably higher than a fuel gas pressure of the fuel gas in the pressure space is necessary, because no fuel gas may enter along the valve shaft in an uncontrolled manner into further components of the gas supply system, in particular not into a hydraulic drive of the valve shaft, i.e. no gas

leaks may occur, or only small, technically insignificant gas leaks which are under very good control, may occur at or along the valve shaft. The valve shaft must therefore be correspondingly reliably sealed, which was attempted with the previous system, e.g. with corresponding sealing rings, but which rather proved to be disadvantageous all in all because the sealing rings either do not show any sufficient sealing effect with respect to the pressurized fuel gas or also e.g. wear too fast due to friction phenomena which occur in the operating state and are thus ultimately too expensive and too unreliable.

These problems are also reliably avoided for the first time by the present invention.

The gas supply system is particularly preferably provided at the cylinder in a region between the top dead center and the bottom dead center of the piston of the large engine, which region is remote from the top dead center by 20% to 80%, preferably by 45% to 65%, particularly preferably by 50% to 60% of the spacing between the top dead center and the bottom dead center. It has namely been found that not only an even more ideal mixture of the fuel gas with the scavenging air can thereby be achieved. It can also thereby be avoided that fuel gas is injected into hot exhaust gases still being in the cylinder liner, whereby the fuel gas could ignite prematurely, as may be the case when the gas supply system is located, as in the prior art, in the proximity of the top dead center at the cylinder liner.

The large engine in accordance with the invention is particularly preferably a dual fuel engine for the combustion of the fuel gas and alternatively for the combustion of a further fuel, in particular for the combustion of diesel or heavy oil.

In another embodiment, a large engine in accordance with the invention can naturally also be a purely gas engine with which only fuel gas can be combusted, that is no other fuel.

The invention also relates not least to a method of operating a large engine in accordance with the invention, wherein, as already described, a malfunction of the gas supply system is detected in the operating state, the fuel gas supply to the gas supply system is inhibited and the large engine is preferably automatically switched off, particularly preferably automatically switched to an operation with an alternative fuel such as diesel oil or heavy oil.

The invention will be explained in more detail in the following with reference to the schematic drawing. There is shown:

Fig. 1 a particularly preferred embodiment of a gas supply system of a large engine in accordance with the invention with a gas inlet valve and a gas inlet nozzle.

Fig. 1 shows, in a schematic representation for explaining the interaction of the different components, the basic structure of a gas supply at a cylinder liner of an embodiment of a large engine in accordance with the invention, which is here designed by way of example as a two-stroke large diesel engine with uniflow scavenging, but which is not shown in any more detail for reasons of clarity since the basic structure of a uniflow-scavenged two-stroke large diesel engine is sufficiently familiar to the skilled person. The gas supply system in accordance with the invention will be designated as a whole by the reference numeral 1 in the following.

The specific embodiment of a gas supply system 1 in accordance with the invention in accordance with Fig. 1, installed in a uniflow-scavenged two-stroke large diesel engine, comprises a gas inlet valve 2 having a gas inlet nozzle 3, which is designed and is arranged in a cylinder wall 41 of a cylinder 4 of the reciprocating piston internal combustion engine such that a fuel gas 5 provided as a fuel can be supplied to a combustion space 42, not shown in more detail, of the

cylinder 4 by means of the gas inlet nozzle 3 in the installed state of the gas supply system 1. In this respect, the gas inlet valve 2 comprises a pressure space 22 which is arranged in a valve housing 21, in which pressure space 22 the fuel gas 5 can be provided via a gas supply 23 for storage and supply into the gas inlet nozzle 3 in the operating state.

In the pressure space 22 itself or in the region of the pressure space 22, a valve body 6 is provided having a valve disk 62 arranged at a valve shaft 61 and having a valve seat 63, said valve seat 63 sealingly cooperating with the valve disk 2 in a closed state of the valve body 6 such that the supply of the fuel gas 5 from the pressure space 22 into the gas inlet nozzle 3 is inhibited.

For this purpose, the valve disk 2 can be raised from the valve seat 3 by means of a valve drive 7 being in operative connection with the valve shaft 61 so that the fuel gas 5 can be supplied to the gas inlet nozzle 3 from the pressure space 22 past the valve disk 2 in an open state of the valve body 6.

In accordance with the present invention, the gas inlet nozzle 3 is connected releasably to the gas inlet valve 2 and replaceably to the cylinder wall 41, wherein a nozzle axis D of the gas inlet nozzle 3 is arranged at a predefinable angle α with respect to a valve axis V of the gas inlet valve 2 such that the fuel gas 5 can be injected into the combustion space 42 of the cylinder 4 in the operating and installed state of the gas supply system 1 at a predefinable angle differing from zero with respect to a radial direction R and/or with respect to an axial direction A of the cylinder 4.

In this respect, that angle is called the injection angle β with respect to the radial direction which is the supplementary angle to the angle α between the nozzle axis D of the gas inlet nozzle 3 and the valve axis V of the gas inlet valve 2, that is the injection angle β and the angle α between the nozzle axis D and the valve axis V

complement one another to form 180°, $\beta = 180° - \alpha$ thus applies. The injection angle β preferably amounts between 10 degrees and 80 degrees, particularly preferably between 10 degrees and 35 degrees, the injection angle is specifically approximately 22.5 degrees.

With respect to the axial direction, the angle β is preferably 90°, that is the fuel gas 5 is preferably injected perpendicular to the axial direction A, that is in the drawing plane of Fig. 1 or in parallel with the surface of the piston.

The relatively simple gas inlet nozzle 3 in this respect has only one nozzle opening 31. In another specific embodiment, the gas inlet nozzle 3 can, however, indeed also have a plurality of nozzle openings 31, wherein e.g. two nozzle openings 31 can be aligned differently such that the fuel gas 5 can be introduced into the combustion space 42 at two different injection angles, and/or wherein two nozzle openings 31 can also or simultaneously be configured differently such that the fuel gas 5 can be introduced into the combustion space 42 through the two nozzle openings 31 with two different injection quantities and/or at two different speeds and/or with two different jet geometries.

Furthermore, in practice, a control system not shown explicitly in Fig. 1 for monitoring a fuel gas flow into the combustion space 42 and for determining a position of the valve body 6 can be provided, said control system particularly preferably being a path sensor, in particular an electrical or an electromagnetic path sensor, specifically an inductive, capacitive or optical path sensor. The control system is in signal communication with a monitoring unit such that the position of the valve body 6 in the operating state can be detected by the monitoring unit and a fuel gas supply to the gas inlet nozzle 3 can be automatically inhibited in dependence on a crank angle of the large engine and/or in dependence on a position of a gas exchange valve of the large engine.

In the embodiment in accordance with Fig. 1, the valve drive 7 for driving the valve shaft 61 is a hydraulic valve drive 7, whose function and operation are generally familiar to the skilled person. To close the valve body 6, a restoration device 8 in the form of a restoration spring 81 acting against the valve drive 7 is provided.

The valve shaft 61 is guided in a guide bore 91 of a shaft housing 9, wherein the guide bore 91 can be applied with a hydraulic oil 10 at a sealing pressure PA, said sealing pressure PA being higher here than a fuel gas pressure BG of the fuel gas 5 in the pressure space 22 such that a penetration of the fuel gas 5 into the guide bore 91 can be substantially prevented.

The gas supply system of the large engine in accordance with the invention is preferably configured as a low-pressure gas system. This means that the injection pressure at which the fuel gas 5 is injected into the combustion space 42 of the cylinder 4 is at most 100 bar (10 MPa). The injection pressure is preferably a maximum of 50 bar (5 MPa) and is particularly preferably at most 20 bar (2 MPa). The gas pressure, that is the pressure at which the fuel gas is injected into the cylinder, is typically not constant during the operation of the internal combustion engine but can rather vary in dependence on the load or the revolution speed of the internal combustion engine, for example.

With the large engine in accordance with the invention, in a preferred embodiment, a gas pressure, which is as small as possible is sought for the injection of the fuel gas 5 into the combustion space 42. The maximum injection pressure for the fuel gas 5 can thus, for example, also be only 15 bar or even smaller.

Of course, an injection pressure of the fuel gas 5, which is as small as possible, has great advantages under safety aspects. In addition, the total gas supply system only has to be configured for such comparatively small operating pressures, which is in particular especially advantageous with respect to the

sealing of the system, the forces which occur, for example at the valve seat 63, and also with respect to the configuration of the valve drive 7 as well as with respect to the pressure load of the gas-conducting lines or of their connections. No special high-pressure compressors are also required with which the fuel gas 5 has to be compressed as in the case of high-pressure systems to an operating pressure of, for example, 350 bar or even higher, which is also advantageous under economic and cost aspects.

Particularly with respect to an injection pressure of the fuel gas 5 which is as small as possible, it is therefore particularly preferred if the gas inlet valve 2 and the gas inlet nozzle 3 are arranged in the cylinder wall 41 or in the cylinder liner and have a spacing which is as large as possible from the top dead center of the piston movement. It can namely hereby be achieved that the compression pressure in the cylinder, toward which the fuel gas 5 has to be injected, is still comparatively small. When the piston releases the scavenging air slits on the downward movement in a uniflow-scavenged two-stroke large diesel engine, the scavenging air begins to flow into the cylinder. This takes place for so long until the piston has again completely closed the scavenging air slits on its subsequent upward movement. Only when the gas exchange valve (here the outlet valve) is closed and is completely closed – as a rule after the closing of the scavenging air slits – the compression pressure in the cylinder start to rise due to the upward movement of the piston until it reaches its maximum value, for instance when the piston is at the top dead center.

The fuel gas injection therefore preferably takes place when there is not yet any particularly large compression pressure in the cylinder and the injection of the fuel gas 5 preferably takes place before the outlet valve is closed.

It is preferred for this reason if the gas inlet nozzle 3 provided in the cylinder wall 41 has a spacing from the top dead center which is as large as possible with

respect to the axial direction A. On the other hand, there are also construction restrictions as to where the gas inlet nozzle 3 can be arranged. It has proved to be a very good compromise in practice, if the gas inlet nozzle 3 of the gas supply system 1 is arranged at such a level (with respect to the axial direction A) that its spacing from the top dead center particularly preferably amounts to 50% to 60% of the spacing between the top dead center and the bottom dead center.

It has been found to be particularly preferred with respect to the position of the scavenging air slits if the gas inlet nozzle 3 is arranged at such a level (with respect to the axial direction A) that its spacing from the upper edge of the scavenging air slits ("upper edge" with respect to the typical position of use of the internal combustion engine) preferably amounts to less than 50% and particularly preferably to 30% to 40% of the spacing of the upper edge of the scavenging air slits from the top dead center.

The preferred arrangement of the gas inlet nozzle 3 also has the advantage, in addition to the making possible of a small injection pressure for the fuel gas 5, that a large spacing exists with respect to the axial direction A between the gas inlet nozzle 3 and the gas exchange valve or the outlet valve. It is hereby avoided, on the one hand, that a substantial proportion of the injected fuel gas 5 can escape through the outlet valve in a non-combusted state and, on the other hand, there is more time for the closing of the outlet valve.

The preferred arrangement of the gas inlet nozzle 3 is furthermore also advantageous with respect to an air/fuel gas mixture which is as homogeneous as possible. Due to the early injection of the fuel gas 5 with respect to the compression stroke of the piston, the scavenging air and the fuel gas have sufficient time to mix intimately before the combustion of the mixture begins. This results in a combustion process in the combustion space 42 which is optimum and in particular also as low-emission as possible.

With respect to the ideal mixing of the scavenging air and the fuel gas 5, it is also advantageous if – as already mentioned – the fuel gas 5 is injected into the combustion space 42 of the cylinder 4, with respect to the radial direction, by the injection angle β differing from zero. A particularly intense mixing hereby takes place with the scavenging air which is typically swirl-loaded in the cylinder 4 or in the combustion space 42 on the compression stroke of the piston.

Two or more such gas supply systems 1 or two or more gas inlet nozzles 3 having gas inlet valves 2 can naturally also be provided at a cylinder.

It is understood that the embodiments of the invention described in this application can also be combined in any suitable manner in dependence on the application and that in particular the special embodiments shown in the drawing are only to be understood by way of example. The skilled person immediately recognizes simple advantageous further developments of the described embodiments of the invention, as defined in the attached claims, and understands that such simple further developments are naturally also covered by the invention.

P A T E N T K R A V

1. Stor motor, udformet som længdeskyllet stor totaktsdieselmotor eller som gasmotor, med en cylinder med et gastilførselssystem, hvor gastilførselssystemet omfatter en gasindløbsventil (2) med en gasindløbsdyse (3), som er udformet således, at den er anbragt i en cylindervæg (41) i den store motors cylinder (4) således, at en som brændstof til rådighed stillet brændgas (5) i gastilførselssystemets indbyggede tilstand ved hjælp af gasindløbsdysen (3) kan tilføres til et brændkammer (42) i cylinderen (4), hvor gastilførselssystemet er tilvejebragt i et område af cylindervæggen (41) mellem et øvre dødpunkt og et nedre dødpunkt af et stempel i den store motor, **kendetegnet ved, at** gasindløbsventilen (2) omfatter et i et ventilhus (21) anbragt trykkammer (22), hvori brændgassen (5) via en gastilførsel (23) stilles til rådighed til lagring og tilførsel til gasindløbsdysen (3) i driftstilstand, og der i trykkammeret (22) er tilvejebragt et ventillegeme (6) med en på en ventilspindel (61) anbragt ventiltallerken (62) samt et ventilsæde (63), hvilket ventilsæde (63) i en lukket tilstand af ventillegemet (6) samvirker tætnende med ventiltallerkenen (62) således, at tilførslen af brændgassen (5) fra trykkammeret (22) til gasindløbsdysen (3) er standset, og ventiltallerkenen (62) kan løftes fra ventilsædet (63) ved hjælp af en med ventilspindelen (61) funktionsforbundet ventildrev (7), således at brændgassen (5) i en åbnet tilstand af ventillegemet (6) kan tilføres fra trykkammeret (22) forbi ventiltallerkenen (62) til gasindløbsdysen (3), hvor gasindløbsdysen (3) er forbundet aftageligt med gasindløbsventilen (2) og udskifteligt med cylindervæggen (41), og hvor en dyseakse (D) i gasindløbsdysen (3) er anbragt under en forudbestemmelig vinkel (α) i forhold til en ventilakse (V) i gasindløbsventilen (2) således, at brændgassen (5) i gastilførselssystemets drifts- og indbyggede tilstand kan indsprøjtes i cylinderens (4) brændkammer (42) under en forudbestemmelig vinkel, som er forskellig fra nul, i forhold til en radial retning (R) og/eller i forhold til en aksial retning (A) af cylinderen (4).

2. Stor motor ifølge krav 1, hvor gasindløbsdysen (3) udviser en flerhed af dyseåbninger (31).

3. Stor motor ifølge krav 2, hvor to dyseåbninger (31) er orienteret forskelligt således, at brændgassen (5) kan indføres i brændkammeret (42) under to forskellige indsprøjtningsvinkler.

4. Stor motor ifølge et af kravene 2 eller 3, hvor to dyseåbninger (31) er udformet forskelligt således, at brændgassen (5) kan indføres i brændkammeret (42) gennem de to dyseåbninger (31) med to forskellige indsprøjtningsmængder og/eller med to forskellige strømningshastigheder og/eller med to forskellige strålegeometrier.

5. Stor motor ifølge et af de foregående krav, hvor der til overvågning af en brændgasstrøm ind i brændkammeret (42) er tilvejebragt et kontrolsystem til bestemmelse af en position af ventillegemet (6).

6. Stor motor ifølge krav 5, hvor kontrolsystemet er en forskydningssensor, især en elektrisk eller elektromagnetisk forskydningssensor, specielt en induktiv, kapacitiv eller optisk forskydningssensor.

7. Stor motor ifølge et af kravene 5 eller 6, hvor kontrolsystemet er signalforbundet med en overvågningsenhed, og ventillegemets (6) position i driftstilstand kan registreres af overvågningsenheden, således at en brændgastilførsel til gasindløbsdysen (3) kan standses i afhængighed af en krumtapvinkel af den store motor og/eller i afhængighed af en stilling af en gasskifteventil i den store motor.

8. Stor motor ifølge et af de foregående krav, hvor ventildrevet (7) til drift af ventilspindelen (61) er et mekanisk, elektrisk eller pneumatisk ventildrev (7), især et hydraulisk ventildrev (7).

9. Stor motor ifølge et af de foregående krav, hvor der til lukning af ventillegemet (6) er tilvejebragt en mod ventildrevet (7) virkende tilbagestillingsindretning (8), især i form af en tilbagestillingsfjeder (81).

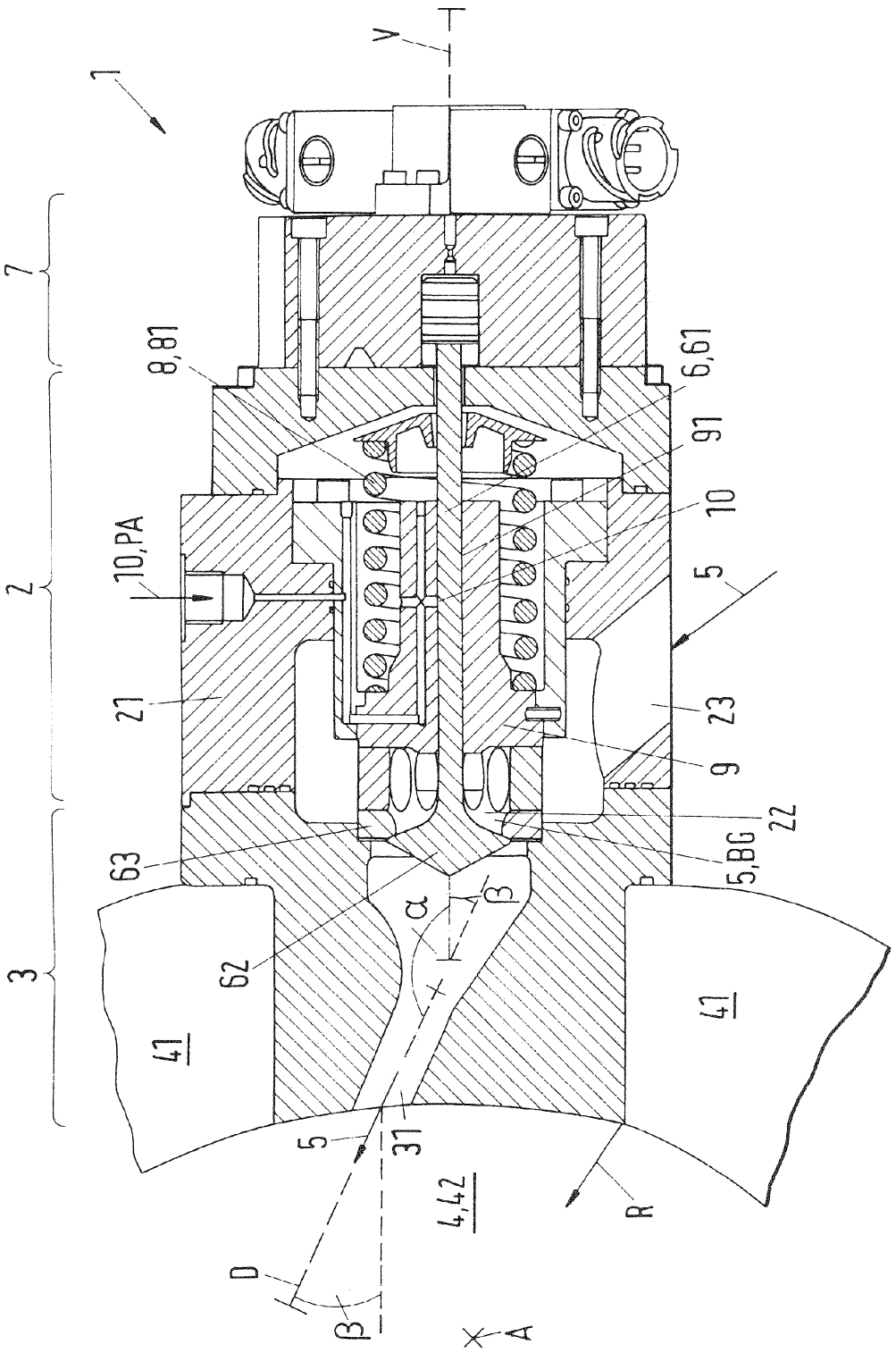
10. Stor motor ifølge et af de foregående krav, hvor ventilspindelen (61) er styret i en styreboring (91) i et spindelhus (9).

11. Stor motor ifølge krav 10, hvor styreboringen (91) kan påvirkes med en under et tætningstryk (PA) stående hydraulikolie (10), hvilket tætningstryk (PA) er højere end et brændgastryk (BG) i brændgassen (5) i trykkammeret (22), således at en indtrængen af brændgassen (5) i styreboringen (91) i det væsentlige kan forhindres.

12. Stor motor ifølge et af de foregående krav, hvor gastilførselssystemet (1) er tilvejebragt på cylinderen (4) i et område mellem det øvre dødpunkt og det nedre dødpunkt af et stempel i den store motor, som ligger med afstand fra det øvre dødpunkt med 20% til 80%, fortrinsvis 45% til 65%, særligt foretrukket 50% til 60% af afstanden mellem det øvre dødpunkt og det nedre dødpunkt.

13. Stor motor ifølge et af de foregående krav, hvor den store motor er en dual-fuel motor til forbrænding af brændgassen (5) og alternativt til forbrænding af et yderligere brændstof, især til forbrænding af diesel eller tung olie.

14. Fremgangsmåde til drift af en stor motor ifølge et af de foregående krav, hvor en fejlfunktion af gastilførselssystemet (1) detekteres, brændgastilførslen til gastilførselssystemet (1) standses, og den store motor fortrinsvis frakobles automatisk, især fortrinsvis automatisk til en drift med et alternativt brændstof, såsom eksempelvis dieselolie eller tung olie.


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