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(57) Sammendrag:
A large two-stroke turbocharged uniflow scavenged internal combustion engine and a method of operating the engine by supplying a carbon-based fuel to the combustion chambers, combusting the carbon-based fuel in the combustion chambers, thereby producing a stream of exhasut gas containing carbon dioxide, recirculating a first portion of the stream of exhaust gas, and exhausting a second portion of the stream of exhaust gas, cooling the first portion of the stream of recirculated exhaust gas in the exhaust gas system using a stream of heat exchange medium thereby heating the stream of heat exchange medium, chemically absorbing carbon dioxide from the second portion of the stream of exhaust gas into a solvent by supplying a flow of carbon dioxide lean solvent to an absorber (42) and discharging a flow of carbon dioxide rich solvent from the absorber (42) to a desorber (64) and reboiler (62) assembly, and regenerating the carbon rich solvent in the desorber (64) and reboiler (62) assembly through heating by supplying at least a portion of the heated stream of heat exchange medium to the desorber (66) and reboiler (62)

Fortsættes...

assembly for heating the solvent.

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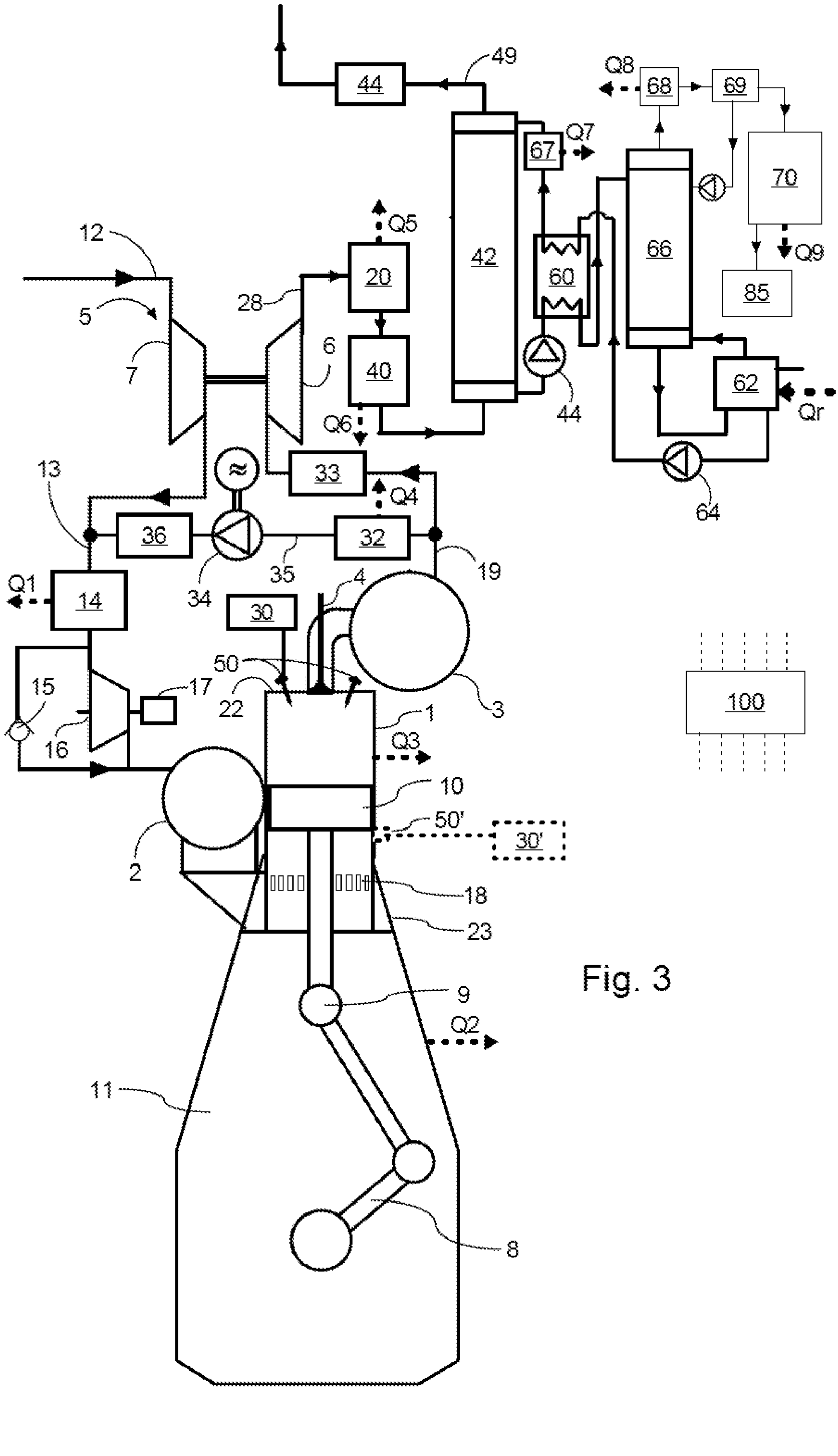
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Fig. 3

METHOD AND LARGE TWO-STROKE UNIFLOW SCAVENGED INTERNAL COMBUSTION ENGINE CONFIGURED FOR CARBON DIOXIDE CAPTURE

TECHNICAL FIELD

The disclosure relates to large two-stroke internal combustion engines, in particular, large two-stroke uniflow scavenged internal combustion engines with crossheads running on a carbon-based fuel (gaseous or liquid fuel), configured to reduce carbon dioxide emissions, and to a method of operating such a type of engine.

BACKGROUND

Large two-stroke turbocharged uniflow scavenged internal combustion engines with crossheads are for example used for propulsion of large oceangoing vessels or as the primary mover in a power plant. Not only due to their sheer size, these two-stroke diesel engines are constructed differently from any other internal combustion engine. Their exhaust valves may weigh up to 400 kg, pistons have a diameter of up to 100 cm and the maximum operating pressure in the combustion chamber is typically several hundred bar. The forces involved at these high pressure levels and piston sizes are enormous.

Large two-stroke turbocharged internal combustion engines that are operated with liquid fuel (e.g. fuel oil, marine diesel, heavy fuel oil, ethanol, dimethyl ether (DME) or with gaseous fuel (e.g. methane, natural gas (LNG), petroleum gas (LPG), methanol or ethane).

Engines that operate with a gaseous fuel may operate according to the Otto cycle in which gaseous fuel is admitted by fuel valves arranged medially along the length of the cylinder liner or in the cylinder cover, i.e. these engines admit the gaseous fuel during the upward stroke (from BDC to TDC) of the piston starting well before the exhaust valve closes, and compress a mixture of gaseous fuel and scavenging air in the combustion chamber and ignites the compressed mixture at or near TDC by timed ignition means, such as e.g. liquid fuel injection.

Engines that are operated with liquid fuel, and also engines that are operated with gaseous fuel with high-pressure injection, inject the gaseous- or liquid fuel when the piston is close to or at TDC, i.e. when the compression pressure in the combustion chamber is at or close to its maximum, and are thus operated according to the Diesel cycle, i.e. with compression ignition.

The liquid- and gaseous fuels used in known large two-stroke turbocharged unit flow scavenged internal combustion engines generally contain carbon, i.e. these are carbon-based fuels, and their combustion results in the generation of carbon dioxide that is exhausted into the atmosphere. Carbon dioxide emissions are generally considered to contribute to climate change and to be minimized or avoided.

Known Carbon Capture Technologies are typically classified into three categories: post-combustion CO2 capture, pre-combustion CO2 capture, and oxy-fuel combustion. Pre-combustion means separating and capturing the carbonaceous components before the combustion of fuel.

In pre-combustion carbon dioxide capture, the fuel is reacted first with oxygen and/or steam and then further processed in a water-gas shift reactor to produce a mixture of H_2 and CO_2. The CO_2 is captured from a high-pressure gas mixture that contains between 15% and 40% CO_2. An advantage of pre-combustion is that the gas volume required for processing is greatly reduced and the CO_2 concentration in the gas is increased. This will reduce energy consumption and equipment investment for the separation process.

In Oxy-Fuel combustion, the carbon-based fuel is combusted in re-circulated flue gas and pure O_2, rather than air. This limits its commercialization potential due to the high cost of O_2 separation. The oxy-fuel combustion technology consists of an air separation unit where the nitrogen is removed from the air. Then the carbon-based fuel is combusted in the re-circulated flue gas and pure oxygen. The flue gas now, primarily consisting of particulate matter from the combustion, CO_2, sulfur oxides from the fuel, and water is sent to a particulate matter removal unit, and sulfur removal unit before condensing the water out, leaving a stream of CO_2 that can be compressed. The main advantage is that it enables nearly 100% CO_2 capture.

In post-combustion technology, the carbon based fuels are combusted as in conventional energy generation, and the CO_2 is captured from the exhaust gas. This carbon separation technology is roughly divided into four sub-areas, namely, absorption, adsorption, membranes, and cryogenics. An amine solvent can be used to capture the CO_2 by absorption from exhaust gas. Here CO_2 is captured in the solvent, followed by a regeneration process of the amine. A drawback is the massive scale-up for power plants and the substantial energy required for the carbon dioxide capture process. In particular, a very significant amount of energy is required for amine solvent regeneration.

DK181014B1 discloses a large two-stroke turbocharged uniflow scavenged internal combustion engine with crossheads according to the preamble of claim 1.

US2013298761 discloses a method and system for on-board treatment of an exhaust stream containing CO_2 emitted by a hydrocarbon-fueled internal combustion engine (ICE) used to power a vehicle in order to reduce the amount of CO_2 discharged into the atmosphere which include: a. a treatment zone on board the vehicle containing a capture agent that is a liquid, a slurry or a finely divided flowable solid having a predetermined capacity for extracting CO_2 from the exhaust stream, the treatment zone having an inlet for admitting the exhaust gas stream and an outlet for passage of a treated exhaust stream to contact the capture agent having a reduced CO_2 content, the treatment zone including a heat exchanger with an inlet for receiving the hot exhaust gas stream from the ICE for passage in heat exchange relation with the capture agent to release CO_2 and regenerate the capture agent, and an outlet for the cooled exhaust gas stream, the treatment zone having a CO_2 discharge outlet for CO_2 released from the regenerated capture agent; b. a compression zone in fluid communication with the CO_2 discharge outlet from the treatment zone, the compression zone including one or more compressors for reducing the volume of the CO2; c. a storage zone for receiving the compressed CO2 for temporary storage on board the vehicle; d. an exhaust gas conduit in fluid communication with the treated exhaust gas stream outlet from the treatment zone; and e. at least one waste heat recovery zone for recovery of heat energy from the exhaust gas stream, the ICE cooling system and/or directly from the ICE for conversion to electrical or mechanical energy.

SUMMARY

It is an object to provide an engine and a method that overcomes or at least reduces the problems indicated above.

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

According to a first aspect, there is provided a large two-stroke turbocharged uniflow scavenged internal combustion engine with crossheads, the engine comprising:

at least one combustion chamber, delimited by a cylinder liner, a piston configured to reciprocate in the cylinder liner, and a cylinder cover,

scavenge ports arranged in the cylinder liner for admitting scavenge gas into the at least one combustion chamber,

a fuel system configured for supplying a carbon-based fuel to the at least one combustion chamber, the at least one combustion chamber being configured for combusting the carbon-based fuel thereby generating a stream of exhaust gas that contains carbon dioxide,

an exhaust gas outlet arranged in the cylinder cover and controlled by an exhaust valve,

the at least one combustion chamber being connected to a scavenge gas receiver via the scavenge ports and to an exhaust gas receiver via the exhaust gas outlet,

an exhaust gas system comprising a turbine of a turbocharger system driven by the stream of exhaust gas,

an air inlet system comprising a compressor of the turbocharger system, the compressor being configured for supplying pressurized scavenge air to the scavenge gas receiver,

an exhaust gas recirculation system configured for recirculating a portion of the exhaust gas originating from the at least one combustion chamber to the scavenge gas receiver, the exhaust gas recirculation system comprising a blower for assisting the flow of exhaust gas to the scavenge air receiver,

an absorber, preferably an absorption tower, for absorbing carbon dioxide into a solvent,

a desorber and reboiler assembly for desorbing carbon dioxide from the solvent,

the absorber having a solvent inlet receiving carbon dioxide lean solvent from the desorber and a solvent outlet supplying carbon dioxide rich solvent to the desorber,

the absorber being arranged for the stream of exhaust gas passing through the absorber for separation of carbon dioxide from the stream of exhaust gas by chemical absorption into the solvent, the desorber and reboiler assembly having an inlet receiving carbon dioxide rich solvent from the absorber and an outlet supplying carbon dioxide lean solvent to the absorber,

the desorber and reboiler assembly being configured for heating the solvent to release carbon dioxide from the solvent, and

a heat exchanging arrangement configured to exchange heat between the recirculated exhaust gas in the exhaust gas recirculation system and the solvent in the desorber and reboiler assembly.

The amount of energy required for regenerating the solvent is significant and can amount to over 60% of the engine shaft power delivered by the large two-stroke internal combustion engine. Such a penalty for the energy efficiency of the engine would render the operation with a carbon dioxide capture system significantly more expensive compared to an engine without such a carbon dioxide capture system. However, the inventors realized that a large two-stroke diesel engine that is operated with exhaust gas recirculation generates a stream of excess energy, since the recirculated exhaust gas is advantageously cooled using a heat exchange medium before it is reintroduced into the cylinder. The inventors also realized that this exchange medium (e.g. water or steam) would be heated to a temperature that is sufficient for using the heat exchange medium directly for heating and thereby regenerating the carbon dioxide rich solvent in the desorber and reboiler assembly.

In a possible implementation form of the first aspect, the engine comprises an exhaust gas recirculation heat exchanger in the exhaust gas recirculation system configured for exchanging heat between the exhaust gas in the exhaust gas recirculation system and a heat exchange medium to thereby cool the exhaust gas in the exhaust gas recirculation system and heat the heat exchange medium, and a heat exchanger configured to exchange heat between the solvent and the heat exchange medium to heat the solvent and cool the heat exchange medium.

In a possible implementation form of the first aspect, the exhaust gas recirculation system comprises a scrubber, preferably a wet scrubber, the scrubber being arranged in the exhaust gas recirculation system downstream of the exhaust gas recirculation heat exchanger.

In a possible implementation form of the first aspect, the engine comprises a controller configured to regulate the percentage by mass of recirculated exhaust gas in the scavenge gas to at least 40%, preferably between 40% and 55%.

In a possible implementation form of the first aspect, the controller is configured to control the speed of the blower to regulate the percentage of recirculated exhaust gas in the scavenge gas.

According to a second aspect, there is provided a method of operating a large two-stroke turbocharged uniflow scavenged internal combustion engine with a plurality of combustion chambers, the method comprising:

supplying a carbon-based fuel to the combustion chambers,

combusting the carbon-based fuel in the combustion chambers, thereby generating a stream of exhaust gas containing carbon dioxide,

recirculating a first portion of the stream of exhaust gas, and exhausting a second portion of the stream of exhaust gas,

supplying a stream of pressurized scavenge gas to the combustion chambers, the stream of pressurized scavenge gas containing the recirculated exhaust gas,

cooling the stream of recirculated exhaust gas in the exhaust gas system using a stream of heat exchange medium thereby heating the stream of heat exchange medium,

chemically absorbing carbon dioxide from the second portion of the stream of exhaust gas into a solvent by supplying a flow of carbon dioxide lean solvent to an absorber and discharging a flow of carbon dioxide rich solvent from the absorber to a desorber and reboiler assembly, and

regenerating the carbon rich solvent in the desorber and reboiler assembly through heating by supplying at least a portion of the heated stream of heat exchange medium to the desorber and reboiler assembly for heating the solvent.

In a possible implementation form of the second aspect, the method comprises recirculating at least 40% by mass of the stream of exhaust gas, preferably recirculating at least 40 to 55% by mass of the stream of exhaust gas.

In a possible implementation form of the second aspect, the method comprises controlling the speed of a blower in an exhaust gas recirculation system for regulating the percentage of recirculated exhaust gas in the pressurized scavenge gas.

In a possible implementation form of the second aspect, the method comprises supplying a flow of gas containing carbon dioxide and water vapor or steam generated in the desorber 66 to a separator 69 for separating the carbon dioxide and water vapor or steam, the separator preferably being a knockout drum to obtain a stream of a gas mainly containing carbon dioxide and a stream of a liquid mainly containing water.

In a possible implementation form of the second aspect, the method comprises supplying the stream of gas mainly containing carbon dioxide to a liquefaction unit and liquefying the stream of gas mainly containing carbon dioxide to obtain a stream of liquefied carbon dioxide, the method preferably comprising directing the stream of liquefied carbon dioxide into a liquefied carbon dioxide storage unit.

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

BRIEF DESCRIPTION OF THE DRAWINGS

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

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

Fig. 2 is an elevated view from another angle of the large two-stroke engine of Fig. 1,

Fig. 3 is a diagrammatic representation of the large two-stroke engine according to Figs. 1 and 2 in an embodiment,

Fig. 4a is a diagrammatic representation of a first embodiment of the heat pump used in the embodiment of Figs. 1 to 3,

Fig. 4b is a diagrammatic representation of a second embodiment of the heat pump used in the embodiment of Figs. 1 to 3, and

Fig. 5 is a diagrammatic representation in more detail of an embodiment of a heat pump used in the embodiment of Figs. 1 to 4a, and

Fig. 6 is a diagrammatic representation of the large two-stroke engine according to Figs. 1 and 2, in another embodiment.

DETAILED DESCRIPTION

In the following detailed description, an internal combustion engine will be described with reference to a large two-stroke low-speed turbocharged internal combustion crosshead engine in the example embodiments. Figs. 1, 2, and 3 show an embodiment of a large low-speed turbocharged two-stroke diesel engine with a crankshaft 8 and crossheads 9. Figs. 1 and 2 are elevated views from different angles. Fig. 3 is a diagrammatic representation of an embodiment of the large low-speed turbocharged two-stroke diesel engine of Figs. 1 and 2 with its intake and exhaust systems. In this embodiment, the engine has six cylinders in line. However, the large low-speed turbocharged two-stroke internal combustion engine may have between four and fourteen cylinders in line, with the cylinder liners carried by an engine frame 11. The engine may e.g. be used as the main engine in a marine vessel or as a stationary engine for operating a generator in a power station. The total output of the engine may, for example, range from 1,000 to 110,000 kW.

The engine is in this example embodiment an engine of the two-stroke uniflow scavenged type with scavenging ports 18 in the lower region of the cylinder liners 1 and a central exhaust valve 4 in a cylinder cover 22 at the top of the cylinder liners 1. The scavenge gas is passed from the scavenge gas receiver 2 through the scavenge ports 18 of the individual cylinder liners 1 when the piston 10 is below the scavenge ports 18.

When the engine is operated as a premix engine (Otto principle), gaseous fuel containing carbon (e.g. Methanol, petroleum gas or LPG, methane, natural gas LNG, or Ethane) is admitted from gaseous fuel admission valves 50′ under the control of an electronic controller 100 when the piston 10 is in its upward movement (from BDC to TDC) and before the piston 10 passes the fuel valves 50′ (gas admission valves). Gaseous or liquid carbon containing fuel (e.g. fuel oil) is injected at high pressure (preferably 300 bar or more) is injected into the combustion chamber fuel valves 50 when the piston 10 is at or near TDC. The fuel gas is admitted at a relatively low pressure that is below 30 bar, preferably below 25 bar, more preferably below 20 bar, and supplied by a gaseous fuel supply system 30′. The current containing fuel for injecting through the fuel valves 50 is supplied by a fuel system 30. High-pressure can either be generated by the fuel system 30 (common rail) or in the fuel valves 50. The fuel admission valves 50’ are, preferably evenly, distributed around the circumference of the cylinder liner and placed in the central region of the length of the cylinder liner 1. The admission of the gaseous fuel takes place when the compression pressure is relatively low, i.e. much lower than the compression pressure when the piston reaches TDC, hence allowing admission at relatively low pressure.

When the engine is operated as a compression ignition engine (Diesel principle) there are no gas admission valves 50’ and the carbon containing fuel (gaseous or liquid) is injected at high pressure through the fuel valves 50 when the piston 10 is at or near TDC.

A piston 10 in the cylinder liner 1 compresses the charge of gaseous fuel and scavenge gas, (or compresses the scavenge gas in case the operation is with fuel injection at TDC only) and at or near TDC ignition is triggered by injection of the fuel at high pressure from fuel valves 50 that are preferably arranged in the cylinder cover 22 or through the compression in case of liquid fuel injection at or near TDC only. Combustion follows and exhaust gas containing carbon dioxide is generated.

When the exhaust valve 4 is opened, the combustion gas flows through a combustion gas duct associated with the cylinder 1 into the combustion/exhaust gas receiver 3 and onwards through a first exhaust gas conduit 19 that includes a selectively catalytic reactor 33 for reduction of nitrous oxides (NOx) in the exhaust gas.

Through a shaft, the turbine 6 drives a compressor 7 supplied with fresh air via an air inlet 12. The compressor 7 delivers pressurized scavenge air to a scavenge air conduit 13 leading to the scavenge air receiver 2. The scavenge air in conduit 13 passes an intercooler 14 for cooling the scavenge air.

Either upstream (shown) or downstream (not shown) of the intercooler 14 the exhaust gas recirculation conduit 35 connects to the scavenge air conduit 13. At this position, recirculated exhaust gas is mixed with the scavenge air to form scavenge gas that flows to the scavenge gas receiver 2. A controller 100 (electronic control unit) is configured to adjust the ratio between the scavenge air and exhaust gas in the scavenge gas, as will be described in greater detail below.

The cooled scavenge air or gas passes via an auxiliary blower 16 driven by an electric motor 17 that pressurizes the scavenge airflow when the compressor 7 of the turbocharger 5 does not deliver sufficient pressure for the scavenge air receiver 2, i.e. in low- or partial load conditions of the engine. At higher engine loads the turbocharger compressor 7 delivers sufficient compressed scavenge air and then the auxiliary blower 16 is bypassed via a non-return valve 15. It is noted, that the examination may comprise more than one turbocharger 5, thereby forming a turbocharger system.

The controller 100, which as such may be comprised of several interconnected electronic units that comprise a processor and other hardware for performing the function of a controller), is generally in control of the operation of the engine and exerts control over e.g. gaseous fuel admission (quantity and timing), liquid fuel injection (quantity and timing), and opening and closing of the exhaust valve 4 (timing and extent of lift), recirculated exhaust gas ratio and operation of various coolers, pumps, and other equipment. Hereto, the controller 100 is in receipt of various signals from sensors that inform the controller 100 of the operating conditions of the engine (engine load, engine speed, blower speed, scavenging gas temperature, exalt gas temperature at various locations, exhaust gas temperature at various locations, pressures in the scavenging system, in the combustion chambers, in the exhaust gas system, and in the exhaust gas recirculation system. Preferably, the engine comprises a variable timing exhaust valve actuation system allowing individual control of the exhaust valve timing for each combustion chamber. The controller 100 is connected via signal lines or wireless connections to the fuel valves 50, the liquid fuel admission valves 50’, the exhaust valve actuator, an angular position sensor that detects the angle of the crankshaft and generates a signal representative of the position of the crankshaft, and a pressure sensor, preferably in the cylinder cover 22 or alternatively in the cylinder liner 1 generating a signal representative of the pressure in the combustion chamber.

Depending on the engine size, the cylinder liner 1 may be manufactured in different sizes with cylinder bores typically ranging from 250 mm to 1000 mm, and corresponding typical lengths ranging from 1000 mm to 4500 mm.

The cylinder liners 1 are mounted in a cylinder frame 23 with a cylinder cover 22 placed on the top of each cylinder liner 1 with a gas-tight interface therebetween. The piston 10 is arranged to reciprocate between Bottom Dead Center (BDC) and Top Dead Center (TDC). These two extreme positions of the piston 10 are separated by a 180 degrees revolution of the crankshaft 8. The cylinder liner 1 is provided with a plurality of circumferentially distributed cylinder lubrication holes that are connected to a cylinder lubrication line that provides a supply of cylinder lubrication oil when the piston 10 passes the cylinder lubrication holes 25, thereafter piston rings in the piston 10 (not shown) distribute the cylinder lubrication oil over the running surface (inner surface) of the cylinder liner 1. The cylinder liners are provided with a jacket (not shown) and jacket cooling water is circulated in the space between the jacket and the cylinder liner.

The liquid fuel valves 50 (typically more than one per cylinder, preferably three or four), are mounted in the cylinder cover 22 and connected to a source of pressurized carbon containing fuel 30. The liquid fuel valves 50 are preferably arranged around the exhaust valve 4, in particular around the central outlet (opening) in the cylinder cover 22, and circumferentially evenly distributed. The central outline is controlled by the exhaust valve 4. The timing and quantity of the apparition fuel injection are controlled by the controller 100. The fuel valves 50 are only used to inject a small amount of ignition liquid (pilot) if the engine is operating in the premix mode. If the engine is operating in a compression ignition mode, the amount of liquid fuel required for operating the engine with the actual engine load is injected through the liquid fuel valves 50. The cylinder 22 cover may be provided with pre-chambers (not shown) and a tip of the liquid fuel valves 50, typically a tip provided with a nozzle with one or more nozzle holes is arranged such that the pilot oil (ignition liquid) is injected and atomized into the pre-chambers to trigger ignition. The pre-chambers assist in ensuring reliable ignition.

The fuel admission vales 50′ are installed in the cylinder liner 1 (or in the cylinder cover 22), with their nozzle substantially flush with the inner surface of the cylinder liner 1 and with the rear end of the fuel valve 50′ protruding from the outer wall of the cylinder liner 1. Typically, one or two, but possibly as many as three or four fuel valves 50′ are provided in each cylinder liner 1, circumferentially distributed (preferably circumferentially evenly distributed) around the cylinder liner 1. The fuel admission valves 50′ are in an embodiment arranged substantially medial along the length of the cylinder liner 1. The fuel admission valves 50′ are connected to a pressurized source of gaseous fuel 30′ (e.g. Methanol, LPG, LNG, Ethane, or Ammonia), i.e. the fuel is in the gaseous phase when it is delivered to the fuel admission valves 50′. Since the gaseous fuel is admitted during the stroke of the piston 10 from BDC to TDC, the pressure of the source of gaseous fuel merely needs to be higher than the pressure residing in the cylinder liner 1, and typically a pressure of less than 20 bar is sufficient for the gaseous fuel delivered to the fuel admission valves 50.′ The fuel admission valves 50′ are connected to the controller 100, which determines the timing of the opening and closing of the fuel admission valves 50’, and the duration of the opening of the fuel admission valves 50’.

The liquid fuel for ignition is in an embodiment a fuel oil, marine diesel, heavy fuel oil, ethanol, or Dimethyl ether (DME).

The gaseous operation mode can be one of several operation modes of the engine. Other modes may include a liquid fuel operation mode, in which all of the fuel required for the operation of the engine is provided in liquid form through the liquid fuel valves 50. In the gaseous fuel operation mode, the engine is operated with gaseous fuel that is admitted during the stroke of the piston from BDC to TDC at relatively low pressure as the main fuel, i.e. providing for a major portion of the energy supplied to the engine, whereas the liquid fuel is, by comparison, constitutes a relatively small amount of fuel that makes only a relatively small contribution to the amount of energy supplied to the engine, the purpose of the liquid fuel being timed ignition, i.e. the liquid fuel serves as an ignition liquid.

Thus, the engine of the present embodiment can be a dual-fuel engine, i.e. the engine has a mode in which it operates exclusively on liquid fuel and a mode in which it nearly exclusively operates on gaseous fuel.

In this embodiment, the engine is shown as a premix engine operating according to the Otto principle. However, this should be understood that the engine can just as well be a compression ignition engine (operating according to the

Diesel principle), with the carbon-based fuel (gaseous or liquid) being injected at high pressure when the piston 10 is at or near TDC.

The engine is operated by supplying a carbon-based fuel to the combustion chambers (liquid and/or gaseous fuel), combusting the carbon-based fuel in the combustion chambers, thereby generating a stream of exhaust gas containing carbon dioxide, preferably recirculating a first portion of the stream of exhaust gas (or of the combustion gas in an embodiment where the recirculated gases taken directly from the combustion chambers), and exhausting another (second) portion of the stream of exhaust gas as exhaust gas, supplying pressurized scavenge gas containing exhaust gas to the combustion chambers, the pressurized scavenge gas containing in an embodiment at least 40% by mass recirculated exhaust gas, preferably 40 to 55%, separating carbon dioxide from the exhaust gas in a carbon dioxide absorption process, and storing the separated carbon dioxide.

Downstream of the turbine 6 of the turbocharger, the exhaust gas enters a second exhaust conduit 28 which leads the exhaust gas to a boiler 20 (also referred to as economizer), which is configured to generate steam. The steam is used e.g. aboard a marine vessel in which the engine is installed for various purposes or the steam can be directly used for heating a desorber 66 and regenerator 62 assembly that will be described in greater detail further below since this steam has a temperature sufficient for being supplied directly to the regenerator 66 and reboiler 62 assembly.

Downstream of the boiler 20, the second exhaust conduit 28 continues to a first heat exchanger 40 in which the exhaust gas exchanges seat with a primary medium that will be described in greater detail further below.

Downstream of the first heat exchanger 40, the second exhaust conduit 28 continues and connects to an inlet at the bottom of an absorber 42. The absorber 42 is preferably an absorbing tower, e.g. a packed absorbing tower. The exhaust gas flows through the absorber 42 to an outlet at the top of the absorber 42.

The absorber 42 is part of a system for chemically absorbing carbon dioxide using a solvent. An example of a suitable solvent is an amine solution. The amine solution may comprise primary, secondary, and/or tertiary amines. Another example of a suitable solution is a NaOH/KOH solution, preferably an aqueous amine NaOH/KOH solution.

Carbon dioxide is removed from the exhaust gas by a packed absorption tower (absorber) 42. This reaction is exothermic and increases the solvent temperature along the absorption tower 42. As an example, the carbon dioxide concentration in the exhaust gas from the engine is between 4-5% (no exhaust gas recirculation) and 9-10% (with exhaust gas recirculation) by volume and is introduced in the absorber 42 countercurrent with the solvent, which enters at the top of the absorption tower 42 and is referred to as the carbon dioxide lean solvent. This carbon dioxide lean solvent is supplied by a desorber 66 at approximately 35 °C to 55 °C and ambient pressure. At the top of the absorber 42 a wash water section consisting of a packed bed removes most of the volatile amine sorbent, that has escaped to the exhaust gas, by condensing and solubilizing it. The total height of the absorption tower 42 can be up to 50 meters. As carbon dioxide is absorbed in the absorber 42, a stream of carbon dioxide rich solvent from the bottom of the absorber 42 is fed by a pump 44 into a cross heat exchanger 60 for heat exchange with a stream of carbon dioxide lean solvent before it is introduced into the desorber 66 and reboiler 62 assembly where it is heated in the reboiler 62, in order to release the carbon dioxide from the solvent. The stripping (desorbing) temperature varies between 120 °C and 150 °C, and the operating pressure reaches up to 5 bar.

A water-saturated carbon dioxide stream is released from the top of the desorber column 66 and is cooled in a heat exchanger 68 in order to condense most of the water content, which is then separated in a knockout drum 69 and returned to the desorber column 66. The stream of carbon dioxide from the knock-out drum 69 is subsequently compressed/liquified in a liquefaction unit 70 and stored temporarily in a storage tank 88, which is an embodiment a cryogenic storage tank. From the temporary storage tank 85, the liquefied carbon dioxide can be transported to a final storage or utility site (not shown). If the engine is installed in a marine vessel, the temporary storage tank 88 will be arranged in the marine vessel and will be emptied when the marine vessel is in a harbor that is provided with utilities for receiving liquefied carbon dioxide.

The regeneration process of the amine solution does not remove all the carbon dioxide in the solution, and the regenerated carbon dioxide lean solvent is recycled to the absorption tower 42 with a lean carbon dioxide loading by the action of a pump 64. Before reaching the absorber 42, the carbon dioxide rich solvent exchanges heat with the carbon dioxide lean solvent in the cross heat exchanger 60 and in a heat exchanger 67.

The carbon dioxide loading of the solvent after it has absorbed carbon dioxide through the column is referred to as the carbon dioxide rich solvent. The difference between this lean and rich load is the amount of captured carbon dioxide from the exhaust gas.

The carbon dioxide concentration in the exhaust gas leaving the absorber 42 is up to 10 times lower than the carbon dioxide concentration of the exhaust gas that enters the absorber 42.

Some of the amines of the solvent may still be present in the exhaust gas leaving the absorber 42, and these are removed by an amine scrubber 44 that is arranged in the exhaust conduit 49 downstream of the absorber 42.

The engine produces a number of excess energy flows Q1, Q2, Qn, also referred to as waste heat flows, from various parts of the engine. In the embodiment of Fig. 3 these include:

- Q1, the primary cooling medium (e.g. water) of the scavenge air cooler 14. The cooling water from the scavenge air cooler 14 will typically have a temperature between approximately 20 and 240 °C,

- Q2, the primary medium engine lubrication oil, which will typically have a temperature between 45 and 55 °C
- Q3 the primary cooling medium (e.g. water) of the cylinder jacket cooler. The cooling water from the cylinder jacket will typically have a temperature between approximately 70 and 90 °C,
- Q4 the primary cooling medium (e.g. water) of an exhaust gas recirculation conduit cooler 32, which typically has a temperature between approximately 50 to 350 °C,
- Q5 the boiler 20, which typically will supply steam with a temperature between approximately 160 and 170 °C,
- Q6 the primary medium (e.g. water) that is used in the first heat exchanger 40, that will typically have a temperature between 160 and 170 °C,
- Q7, the primary medium (e.g. water) that is used in the second heat exchanger 67, that will typically have a temperature between 100 and 170 °C,
- Q8 the primary medium (e.g. water) that is used in the third heat exchanger 68, that will typically have a temperature between 95 and 105 °C,
- Q9 the primary medium (e.g. water) that is used to cool the liquefaction unit 70, that will have a temperature that depends on the type of technology used for liquefaction and on the type of cooling system used for the liquefaction unit 70.

It is noted that this list of excess energy flows generated by the engine is not exhaustive and merely serves to provide examples of such sources.

At least one of the above-listed sources of excess energy Q1, Q2 ... Qn, in particular, those that have a temperature below the temperature required for heating the desorber 66 and regenerator 62 assembly (which requires a secondary medium with a temperature of at least 120 °C preferably at least 110 °C) is supplied to a heat pump 80. The heat pump 80 is configured to generate a stream of energy Qr in the form of the flow of a secondary medium (e.g. water or steam) with a temperature of at least 120 °C preferably at least 130 °C. Preferably the temperature of the secondary medium supplied to the desorber 66 and reboiler 62 assembly is between 130 and 140 °C most preferred approximately 136 °C.

A first embodiment of the implementation of the pump 80 is shown in Fig. 4a. In this embodiment, a plurality of sources of excess energy Q1, Q2 ... Qn is applied to the single heat pump 80, and a stream of energy Qr that is supplied to the desorber 66 and regenerator 62 assembly is generated by the pump 80.

A second embodiment of the implementation of the pump 80 is shown in Fig. 4b. In this embodiment, one of a plurality of sources of excess energy Q1, Q2 ... Qn is applied to one of a plurality of heat pumps 80 and the stream of energy Qr that is supplied to the desorber 66 and regenerator 62 assembly is generated by the plurality of heat pumps 80 and preferably combined into one stream of energy Qr to the desorber 66 and regenerator 62 assembly.

The heat pump or pumps 80 are used to boost the temperature of the amine solution in the reboiler 62. The heat pump 80 comprises at least an evaporator, a condenser, a compressor, and a throttling valve. Within the heat pump 80 a heat pump (refrigerating) fluid is cycled in a cycle that comprises the evaporator, a condenser, a compressor, and a throttling valve, as shown in Fig. 5. The heat pump 80 functions by the evaporator receiving thermal heat from the flow of energy Q2. The heat pump fluid evaporates in the evaporator and enters the compressor. The compressor is driven, e.g. by an electric motor that receives electric power, e.g. from an alternator or generator driven by takeoff power from the crankshaft of the engine. The compressor increases the pressure and temperature of the heat pump fluid. Downstream of the compressor the heat pump fluid enters the condenser, and heat is transferred to the heat sink and the heat pump fluid condenses. Subsequently, the heat pump fluid expands in the throttling valve before it re-enters the evaporator and the cycle repeats. The secondary medium, e.g. water or steam, transports heat from the condenser to the reboiler 62, preferably in a cycle that is driven by a pump, the secondary medium having a temperature of at least 120 °C preferably at least 130 °C. Thus, the reboiler 62 forms the heat sink for the heat pump 80.

To boost the efficiency of the heat pump 80 the condenser part is in an embodiment split into three heat exchanger (HEX) regions; a super-heater, a condenser, and a sub-cooler. The heat extracted in the super-heater and condenser region is sent to the heat sink. The heat extracted in the sub-cooler is used to preheat the heat pump fluid leaving the evaporator. By having this condenser arrangement less work is needed for the compressor and the system efficiency increases. Moreover, a water loop with a steam HEX and an electrical coil is applied in between the condenser, super-heater and reboiler 62. The fluid entering the steam HEX is in an embodiment the steam generated in the boiler 20. The steam HEX and electrical coil ensure that the reboiler 62 receives sufficient energy in the whole engine load range.

In Fig. 5 several energy flows Q1, Q2 ... Qn are utilized. If only one energy flow is Q1, Q2 ... Qn applied the deaerator below the evaporator can be removed.

In an embodiment, the engine is provided with an exhaust gas recirculation system that comprises an exhaust gas recirculation conduit 35 that connects the first exhaust conduit 19 to the scavenge air conduit 13. Preferably, the exhaust gas recirculation conduit 35 connects to the first exhaust gas conduit 19 upstream of the selective catalytic reactor 33. Preferably, the exhaust gas recirculation conduit 35 connects to the scavenge air conduit 13 upstream of the scavenge air cooler 14. However, it should be understood that the exhaust gas recirculation conduit 35 can also connect to the scavenge air conduit 13 downstream of the scavenge air cooler 14. The exhaust gas recirculation conduit 35 comprises a blower 34 to force exhaust gas from the exhaust gas conduit to the scavenge air conduit, since the pressure in the scavenger conduit 13 is typically higher than the pressure in the first exhaust gas conduit 19 during engine operation. In the shown embodiment the blower 34 is driven by an electric motor, but it is understood that the blower could be powered by any other source of rotary power. In the shown embodiment, the blower 34 is arranged between the exhaust gas recirculation cooler 32 and an exhaust gas recirculation scrubber 36. However, it is understood that the position of the blower 35 could be upstream or downstream of the other elements in the exhaust gas recirculation circuit 35. The exhaust gas recirculation cooler 32 is arranged upstream of the exhaust gas recirculation scrubber 36. The main purpose of the exhaust gas recirculation scrubber 36 is to remove impurities (soot). The controller 100 is configured to control the speed of the blower 34 in the exhaust gas recirculation system for regulating the percentage of recirculated exhaust gas in the pressurized scavenge gas, preferably to a percentage by mass of at least 35% to increase the concentration of carbon dioxide in the exhaust gas and thereby increase the effectiveness of the carbon dioxide absorption system. The exhaust gas recirculation rate can also be controlled by means of valves (not shown) that are controlled by the controller 100. Thus, the controller 100 is configured to operate the engine with a percentage of recirculated exhaust gas in the pressurized scavenge gas of 40% or higher, 45% or higher, or 50% or higher depending on the operating conditions. Generally, the controller 100 is configured to operate with the highest possible percentage of recirculated exhaust/combustion gas since this facilitates the removal of current dioxide from the exhaust gas. By the “highest possible”, is meant the highest ratio that does not cause unacceptable detrimental effects, such as a reduction in the quality of the combustion process, the reliability of the combustion process, an unacceptable increase in the heat load on the engine, etc. The medium (e.g. water or steam) used to exchange heat with the exhaust gas in the exhaust gas recirculation cooler 32 leaves the exhaust gas recirculation cooler 32 with a temperature of approximately 130 to170 °C and this medium can therefore be directly used in the desorber 66 and regenerator 62 assembly, i.e. without involving heat pump 80. The recirculated exhaust gas enters the exhaust gas recirculation cooler 32 with a temperature between approximately 260 and 400°C and the desired temperature for the medium can be obtained by adjusting the flow rate of the medium through the exhaust gas recirculation cooler 32. Exhaust gas recirculation increases the carbon dioxide concentration of the exhaust gas supplied to the absorber 42 resulting in a lower energy consumption of the desorber 66 and regenerator 62 assembly. A higher exhaust gas recirculation ratio also reduces the magnitude of the flow of exhaust gas to the absorber 42 and thus, an absorber tower with a lesser diameter can be used when exhaust gas recirculation is used or the ratio is increased. Further, the energy extracted in the exhaust gas recirculation cooler 32, which is excess energy (waste heat) that is supplied to the desorber 66 and regenerator 62 assembly, thereby significantly reducing the amount of energy that needs to be supplied to operate the desorber 66 and regenerator 62 assembly. The medium coming from the exhaust gas recirculation cooler 32 has a high temperature compared to other excess heat streams of the engine (since the medium is heated by exhaust gas that has not passed through the turbine 6 of the turbocharger 5) and can therefore be used directly in the desorber 66 and regenerator 62 assembly.

Fig. 6 shows another embodiment of the engine. In this embodiment, structures and features that are the same or similar to corresponding structures and features previously described or shown herein are denoted by the same reference numeral as previously used for simplicity. In this embodiment, the engine and the operation thereof are largely identical to the previous embodiment, and hence only the differences with the previous embodiment will be described in detail.

This embodiment comprises an optional second scavenge air cooler 14a downstream of the scavenge air cooler 14. The scavenge air cooler 14 can be configured to generate a stream of heat exchange medium to the desorber 66 and regenerator 62 assembly with the temperature that is sufficient for direct use in the desorber 66 and regenerator 62 assembly. The second scavenge air cooler 14a generates an excess energy flow Q10 in the form of a stream of primary medium (e.g. water) with a temperature that requires the use of the heat pump 80 to generate a stream of a secondary medium before the stream of energy can be used in the desorber 66 and regenerator 62 assembly. The stream of energy Q10 generated in the second scavenge air cooler 14a is sent to the heat exchanger 80.

In this embodiment, there can optionally be provided an additional fourth heat exchanger 41 downstream of the first heat exchanger 40. This additional fourth heat exchanger 41 allows for the generation of another excess energy stream Q 11 that is supplied to the heat pump 80.

In this embodiment, there can also be created an additional excess energy flow Q12 from excess heat from the exhaust gas recirculation scrubber 36 that is supplied to the heat pump 80.

The various aspects and implementations have been described in conjunction with various embodiments herein. The embodiments can be combined in various ways. Further, other variations to the disclosed embodiments can be understood and effected by those skilled in the art in practicing the claimed subject-matter, from a study of the drawings, the disclosure, and the appended claims. In the claims, the word "comprising" does not exclude other elements or steps, and the indefinite article "a" or "an" does not exclude a plurality. A single processor, controller, or other unit may fulfill the functions of several items recited in the claims. The mere fact that certain measures are recited in mutually different dependent claims does not indicate that a combination of these measures cannot be used to advantage. The reference signs used in the claims shall not be construed as limiting the scope.

CLAIMS

1. A large two-stroke turbocharged uniflow scavenged internal combustion engine with crossheads, the engine comprising:

at least one combustion chamber, delimited by a cylinder liner (1), a piston (10) configured to reciprocate in the cylinder liner (1), and a cylinder cover (22),

scavenge ports (18) arranged in the cylinder liner (1) for admitting scavenge gas into the at least one combustion chamber,

a fuel system (30) configured for supplying a carbon-based fuel to the at least one combustion chamber,

the at least one combustion chamber being configured for combusting the carbon-based fuel thereby generating a stream of exhaust gas that contains carbon dioxide,

an exhaust gas outlet arranged in the cylinder cover (22) and controlled by an exhaust valve (4),

the at least one combustion chamber being connected to a scavenge gas receiver (2) via the scavenge ports (18) and to an exhaust gas receiver (3) via the exhaust gas outlet,

an exhaust gas system comprising a turbine (6) of a turbocharger system (5) driven by the stream of exhaust gas,

an air inlet system comprising a compressor (7) of the turbocharger system (5), the compressor (7) being configured for supplying pressurized scavenge air to the scavenge gas receiver (2),

an exhaust gas recirculation system configured for recirculating a portion of the exhaust gas originating from the at least one combustion chamber to the scavenge gas receiver (2), the exhaust gas recirculation system comprising a blower (34) for assisting the flow of exhaust gas to the scavenge air receiver (2), characterized by

an absorber (42), preferably an absorption tower, for absorbing carbon dioxide into a solvent,

a desorber (66) and reboiler (62) assembly for desorbing carbon dioxide from the solvent,

the absorber (42) having a solvent inlet receiving carbon dioxide lean solvent from the desorber (66) and a solvent outlet supplying carbon dioxide rich solvent to the desorber (66),

the absorber (42) being arranged for the stream of exhaust gas passing through the absorber (42) for separation of carbon dioxide from the stream of exhaust gas by chemical absorption into the solvent,

the desorber (66) and reboiler (62) assembly having an inlet receiving carbon dioxide rich solvent from the absorber (42) and an outlet supplying carbon dioxide lean solvent to the absorber (42),

the desorber (66) and reboiler (62) assembly being configured for heating the solvent to release carbon dioxide from the solvent, and

a heat exchanging arrangement configured to exchange heat between the recirculated exhaust gas in the exhaust gas recirculation system and the solvent in the desorber (66) and re-boiler (62) assembly.

2. The engine according to claim 1, comprising an exhaust gas recirculation heat exchanger (32) in the exhaust gas recirculation system configured for exchanging heat between the exhaust gas in the exhaust gas recirculation system and a heat exchange medium to thereby cool the exhaust gas in the exhaust gas recirculation system and heat the heat exchange

medium, and a heat exchanger configured to exchange heat between the solvent and the heat exchange medium to heat the solvent and cool the heat exchange medium.

3. The engine according to claim 1 or 2, wherein the exhaust gas recirculation system comprises a scrubber (36), preferably a wet scrubber, the scrubber being arranged in the exhaust gas recirculation system downstream of the exhaust gas recirculation heat exchanger (32).

4. The engine according to any one of the preceding claims, comprising a controller (100) configured to regulate the percentage by mass of recirculated exhaust gas in the scavenge gas to at least 40%, preferably between 40% and 55%.

5. The engine according to claim 4, wherein the controller (100) is configured to control the speed of the blower (75) to regulate the percentage of recirculated exhaust gas in the scavenge gas.

6. A method of operating a large two-stroke turbocharged uniflow scavenged internal combustion engine with a plurality of combustion chambers, the method comprising:

supplying a carbon-based fuel to the combustion chambers,

combusting the carbon-based fuel in the combustion chambers, thereby generating a stream of exhaust gas containing carbon dioxide,

recirculating a first portion of the stream of exhaust gas, and exhausting a second portion of the stream of exhaust gas, supplying a stream of pressurized scavenge gas to the combustion chambers, the stream of pressurized scavenge gas containing the recirculated exhaust gas,

cooling the stream of recirculated exhaust gas in the exhaust gas system using a stream of heat exchange medium thereby heating the stream of heat exchange medium,

chemically absorbing carbon dioxide from the second portion of the stream of exhaust gas into a solvent by supplying a flow of carbon dioxide lean solvent to an absorber (42) and discharging a flow of carbon dioxide rich solvent from the absorber (42) to a desorber (66) and reboiler (62) assembly, and

regenerating the carbon rich solvent in the desorber (66) and reboiler (62) assembly through heating by supplying at least a portion of the heated stream of heat exchange medium to the desorber (66) and reboiler (62) assembly for heating the solvent.

7. The method according to claim 6, comprising recirculating at least 40% by mass of the stream of exhaust gas, preferably recirculating at least 40 to 55% by mass of the stream of exhaust gas.

8. The method according to claims 6 or 7, comprising controlling the speed of a blower (36) in an exhaust gas recirculation system for regulating the percentage of recirculated exhaust gas in the pressurized scavenge gas.

9. The method according to claim 6, 7 or 8, comprising supplying a flow of gas containing carbon dioxide and water vapor or steam generated in the desorber (66) to a separator

(69) for separating the carbon dioxide and water vapor or steam, the separator preferably being a knockout drum to obtain a stream of a gas mainly containing carbon dioxide and a stream of a liquid mainly containing water.

10. The method according to claim 9, comprising supplying the stream of gas mainly containing carbon dioxide to a liquefaction unit (70) and liquefying the stream of gas mainly containing carbon dioxide to obtain a stream of liquefied carbon dioxide, the method preferably comprising directing the stream of liquefied carbon dioxide into a liquefied carbon dioxide storage unit (85).

PATENTKRAV

1. Stor, turboladet, totaktsforbrændingsmotor med længdeskylning og krydshoveder, hvilken motor omfatter:

mindst ét forbrændingskammer, der er afgrænset af en cylinderforing (1), et stempel (10), der er konfigureret til at bevæge sig frem og tilbage i cylinderforingen (1), og et cylinderdæksel (22),

skylleåbninger (18), der er anbragt i cylinderforingen (1) til at lade skyllegas trænge ind i det mindst ene forbrændingskammer,

et brændstofsystem (30), der er konfigureret til tilførsel af et carbonbaseret brændstof til det mindst ene forbrændingskammer,

hvor det mindst ene forbrændingskammer er konfigureret til forbrænding af det carbonbaserede brændstof, hvorved der genereres en strøm af udstødningsgas, som indeholder carbondioxid,

et udstødningsgasudløb, der er anbragt i cylinderdækslet (22) og styres af en udstødningsventil (4),

hvor det mindst ene forbrændingskammer er forbundet med en skyllegasmodtager (2) via skylleåbningerne (18) og med en udstødningsgasmodtager (3) via udstødningsgasudløbet,

et udstødningsgassystem, der omfatter en turbine (6) i et turboladersystem (5), som drives af udstødningsgasstrømmen,

et luftindløbssystem, der omfatter en kompressor (7) i turboladersystemet (5), hvilken kompressor (7) er konfigureret til at tilføre tryksat skylleluft til skyllegasmodtageren (2), et udstødningsgas-recirkulationssystem, der er konfigureret til recirkulation af en del af udstødningsgassen stammende det det mindst ene forbrændingskammer til skyllegasmodtageren (2), hvilket udstødningsgas-recirkulationssystem omfatter en blæser (34) til at assistere strømmen af udstødningsgas til skyllegasmodtageren (2),

kendetegnet ved,

en absorber (42), fortrinsvis et absorptionstårn, til absorbering af carbondioxid i et opløsningsmiddel,

en desorber- (66) og reboiler- (62) enhed til desorption af carbondioxid fra opløsningsmidlet,

hvilken absorber (42) har et opløsningsmiddelindløb til modtagelse af carbondioxidfattigt opløsningsmiddel fra desorberen (66) og et opløsningsmiddeludløb, der tilfører carbondioxidrigt opløsningsmiddel til desorberen (66),

hvor absorberen (42) er indrettet til strømmen af udstødningsgas, der passerer gennem absorberen (42) for separation af carbondioxid fra strømmen af udstødningsgas ved kemisk absorption i opløsningsmidlet,

hvor desorber- (66) og reboiler- (62) enheden har et indløb, der modtager carbondioxidrigt opløsningsmiddel fra absorberen (42), og et udløb, der tilfører carbondioxidfattigt opløsningsmiddel til absorberen (42),

hvor desorber- (66) og reboiler- (62) enheden er konfigureret til opvarmning af opløsningsmidlet for at frigøre carbondioxid fra opløsningsmidlet, og

en varmevekslingsanordning, der er konfigureret til at udveksle varme mellem den recirkulerede udstødningsgas i udstødningsgas-recirkulationssystemet og opløsningsmidlet i desorber- (66) og reboiler- (62) enheden.

2. Motor ifølge krav 1, og som omfatter en udstødningsgas-recirkulationsvarmeveksler (32) i udstødningsgas-recirkulationssystemet, der er konfigureret til varmeudveksling mellem udstødningsgassen i udstødningsgas-recirkulationssystemet og et varmevekslingsmedium for derved at afkøle udstødningsgassen i udstødningsgas-recirkulationssystemet og opvarme varmevekslingsmediet, og en varmeveksler, der er konfigureret til at udveksle varme mellem opløsningsmidlet og varmevekslingsmediet for at opvarme opløsningsmidlet og afkøle varmevekslingsmediet.

3. Motor ifølge krav 1 eller 2, hvor udstødningsgas-recirkulationssystemet omfatter en scrubber (36), fortrinsvis en vådscrubber, hvilken scrubber er placeret i udstødningsgas-recirkulationssystemet nedstrøms for udstødningsgas-recirkulationsvarmeveksleren (32).

4. Motor ifølge et hvilket som helst af de foregående krav, og som omfatter en styreenhed (100), der er konfigureret til at regulere masseprocentsatsen af recirkuleret udstødningsgas i skylleluften til mindst 40 %, fortrinsvis mellem 40 % og 55 %.

5. Motor ifølge krav 4, hvor styreenheden (100) er konfigureret til at styre blæserens (75) hastighed for at regulere procentsatsen af recirkuleret udstødningsgas i skyllegassen.

6. Fremgangsmåde til drift af en stor, turboladet, totaktsforbrændingsmotor med en flerhed af forbrændingskamre, hvilken fremgangsmåde omfatter:

tilførsel af et carbonbaseret brændstof til forbrændingskamrene,

forbrænding af det carbonbasede brændstof i forbrændingskamrene, hvorved der genereres en strøm af udstødningsgas indeholdende carbondioxid,

recirkulation af en første del af strømmen af udstødningsgas, og udstødning af en anden del af strømmen af udstødningsgas,

tilførsel af en strøm af tryksat skyllegas til forbrændingskamrene, hvilken strøm af tryksat skyllegas indeholder den recirkulerede udstødningsgas,

afkøling af strømmen af recirkuleret udstødningsgas i udstødningsgassystemet ved hjælp af en strøm af varmevekslingsmedium, hvorved strømmen af varmevekslingsmedium opvarmes,

kemisk absorption af carbondioxid fra den anden del af strømmen af udstødningsgas i et opløsningsmiddel ved tilførsel af en strøm af carbondioxidfattigt opløsningsmiddel til en absorber (42) og udledning af en strøm af carbondioxidrigt opløsningsmiddel fra absorberen (42) til en desorber- (66) og reboiler- (62) enhed, og

regenerering af det carbonrige opløsningsmiddel i desorber- (66) og reboiler- (62) enheden via opvarmning ved tilførsel af mindst en del af den opvarmede strøm af varmevekslingsmedium til desorber- (66) og reboiler- (62) enheden for opvarmning af opløsningsmidlet.

7. Fremgangsmåde ifølge krav 6, og som omfatter recirkulation af mindst 40 masse-% af strømmen af udstødningsgas, fortrinsvis recirkulation af mindst 40 til 55 masse-% af strømmen af udstødningsgas.

8. Fremgangsmåde ifølge krav 6 eller 7, og som omfatter styring af en blæsers (36) hastighed i et udstødningsgas-recirkulationssystem til regulering af procentsatsen af recirkuleret udstødningsgas i den tryksatte skyllegas.

9. Fremgangsmåde ifølge krav 6, 7 eller 8, og som omfatter tilførsel af en gasstrøm, der indeholder carbondioxid og vanddamp eller damp genereret i desorberen (66) til en separator (69) for separation af carbondioxiden og vanddampen eller dampen, hvilken separator fortrinsvis er en knockout-tromle til et opnå en strøm af gas, der hovedsageligt indeholder carbondioxid, og en strøm af en væske, der hovedsageligt indeholder vand.

10. Fremgangsmåde ifølge krav 9, og som omfatter tilførsel af gasstrømmen, der hovedsageligt indeholder carbondioxid, til en likvefaktionsenhed (70) og flydendegørelse af strømmen af gas, der hovedsageligt indeholder carbondioxid, for at opnå en strøm af flydende carbondioxid, hvilken fremgangsmåde fortrinsvis omfatter dirigering af strømmen af flydende carbondioxid ind i en enhed (85) til oplagring af flydende carbondioxid.

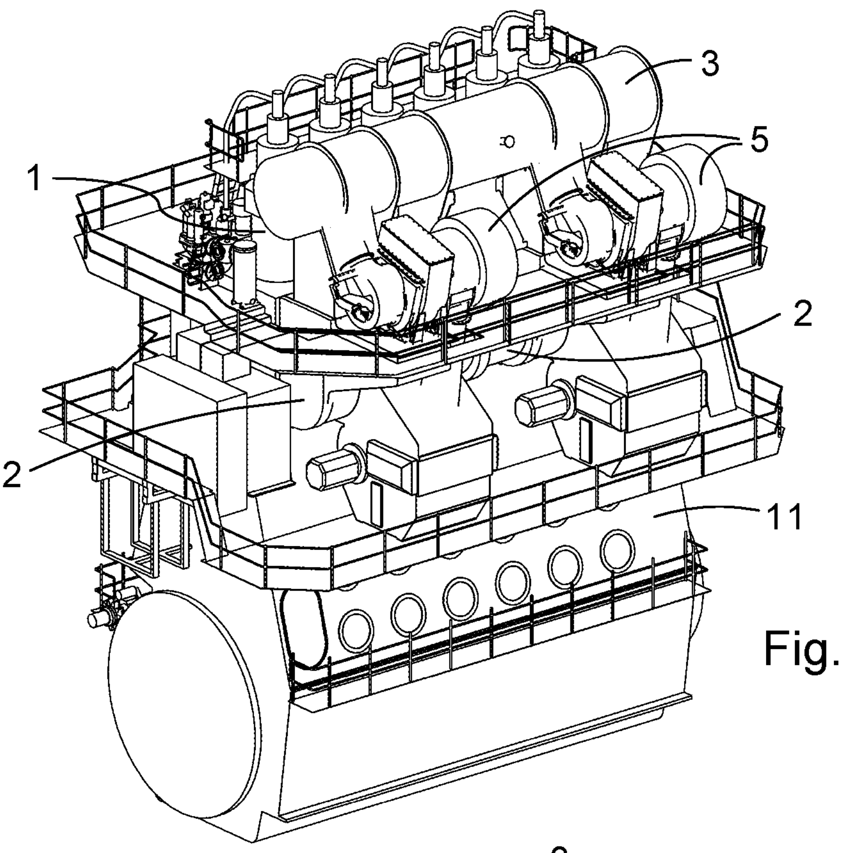


Fig. 1

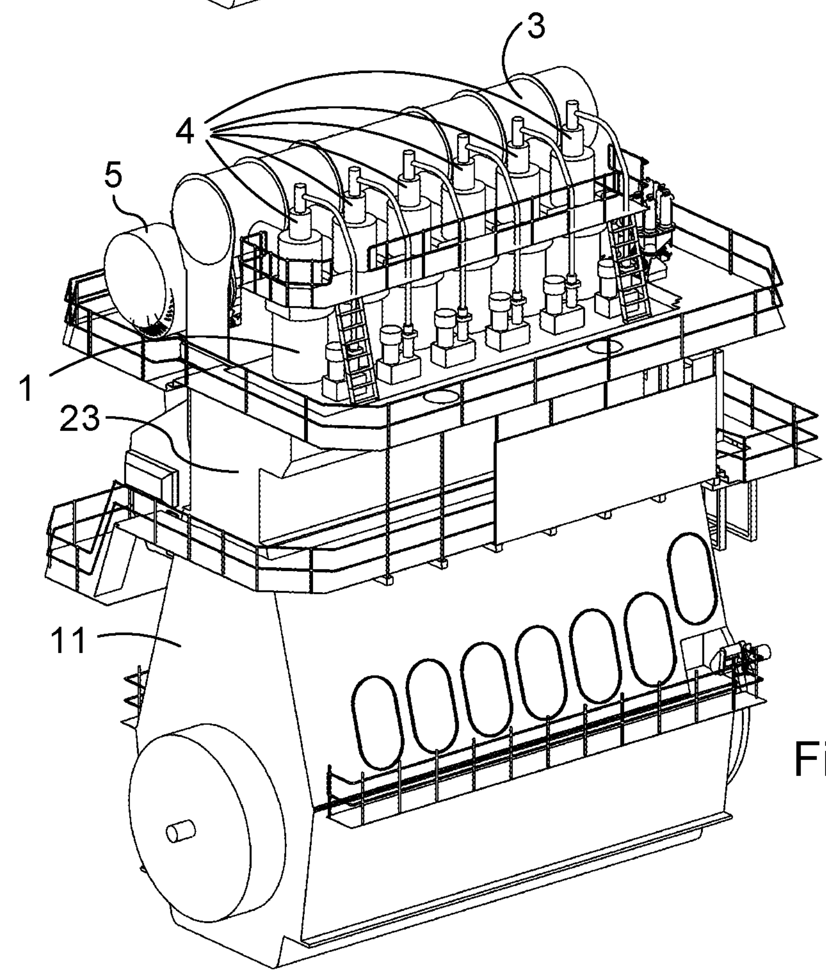
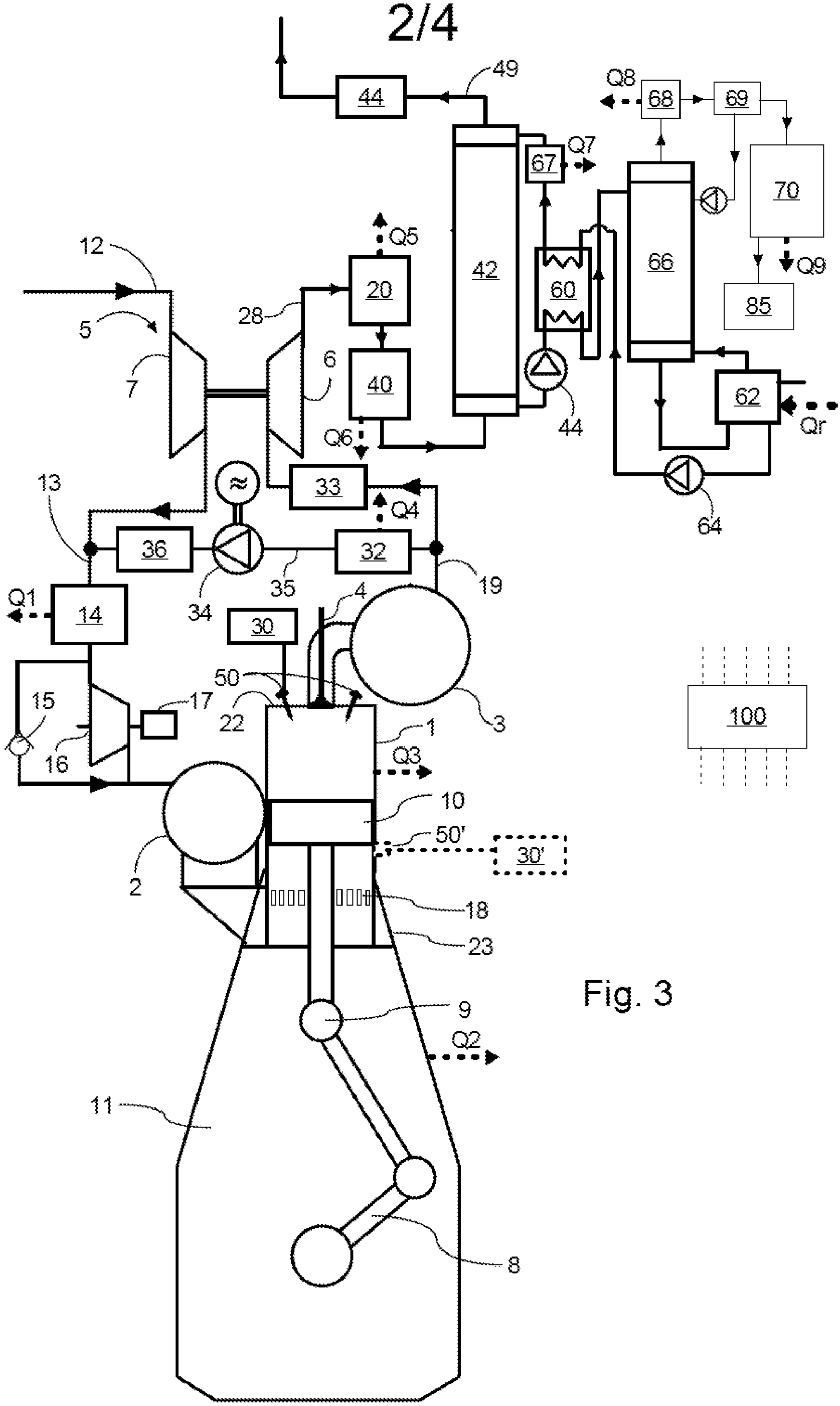


Fig. 2



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Q2
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Fig. 3

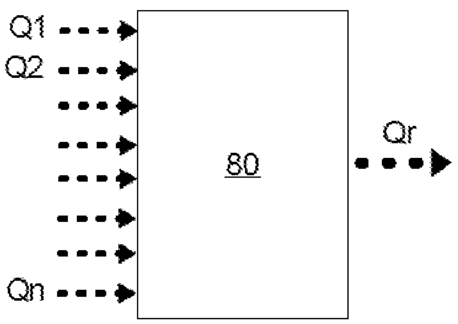


Fig. 4a

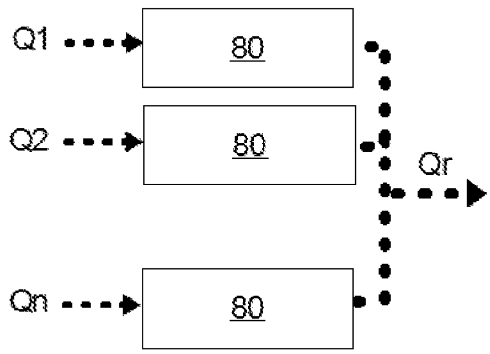


Fig. 4b

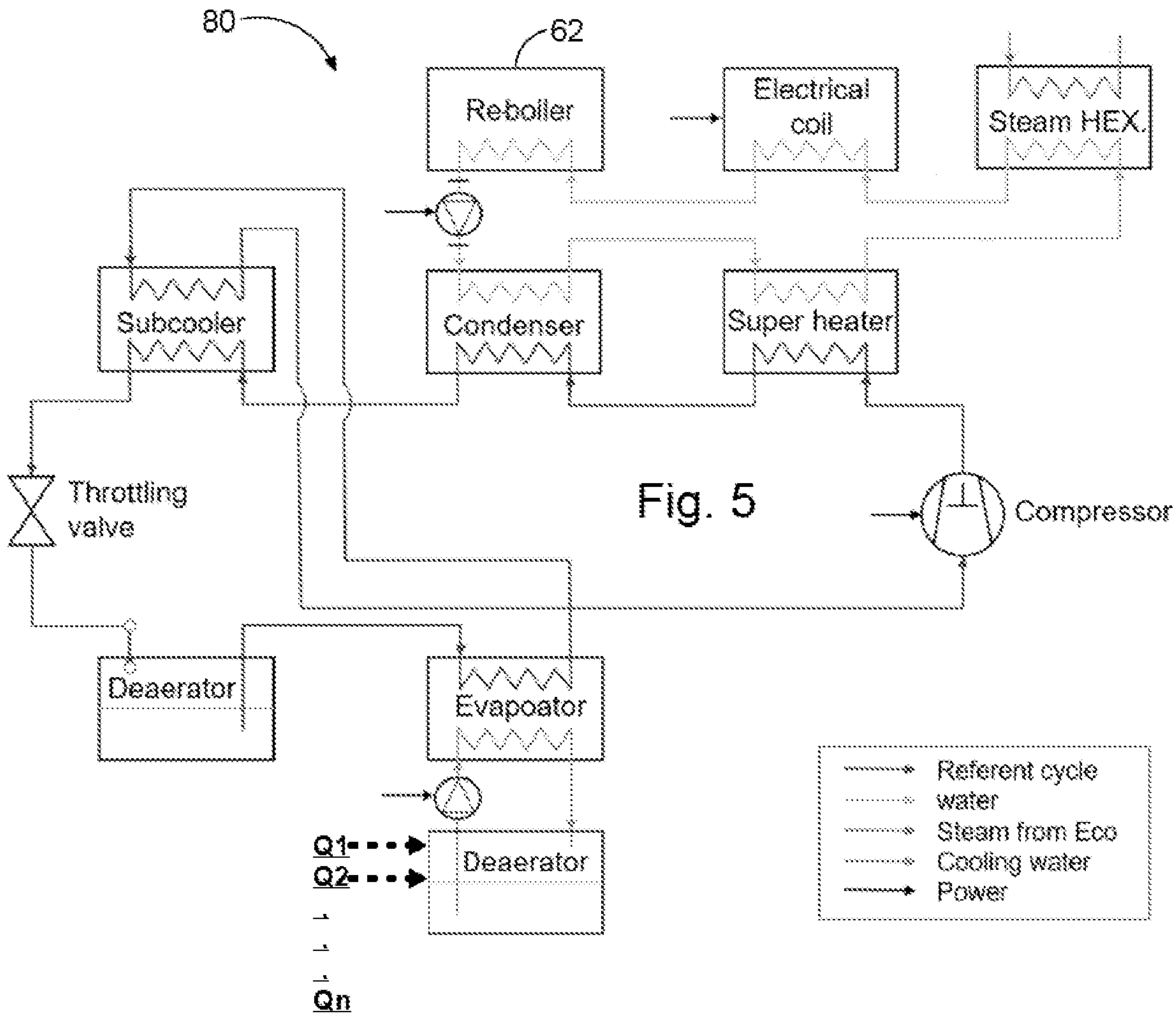
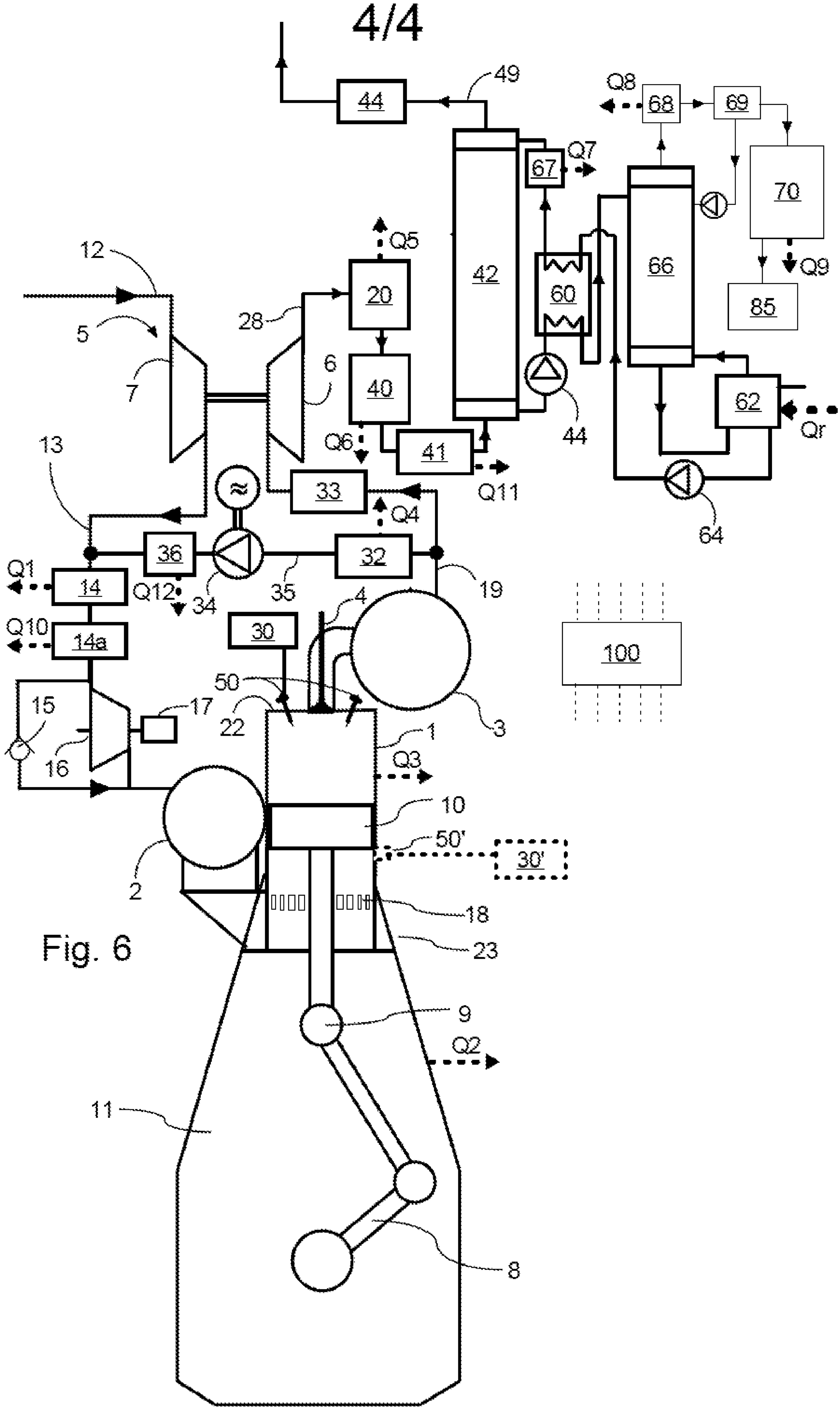


Fig. 5



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Q11
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Q4
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32
Q1
14
Q12
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Q10
14a
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Q3
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50'
30'
2
18
23
9
Q2
11
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Fig. 6

DANISH PATENT AND TRADEMARK OFFICE

Additional search report - patent

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PA 2022 70534

A. Classification

B01D 53/14 (2006.01); B01D 53/92 (2006.01), F01N 3/00 (2006.01), F02B 25/04 (2006.01)

According to International Patent Classification (IPC)

B. Fields searched

PCT-minimum documentation searched (classification system followed by classification symbols)

IPC: B01D, F02B, F02D, F02G, F01N
CPC: B01D, F02B, F02D, F02G, Y02C, Y02T

Documentation searched other than PCT-minimum documentation

DK, NO, SE, FI: IPC-classes as specified in Box A above

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

EPODOC, WPI, FULL TEXT: ENGLISH

C. Documents considered to be relevant

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant for claim No.
A	US 4899544 A (BOYD RANDALL T) 1990.02.13, see especially pg. 2, col. 4, ln. 5-42, pg. 4, col. 7, ln. 56-68, pg. 4, col. 8, ln. 17-30; fig. 1, 3A, 3B.	1-10
A	WO 2015/135685 A1 (WINTERTHUR GAS & DIESEL LTD) 2015.09.17, see especially abstract; pg. 6, ln. 24-28, pg. 21, ln. 1-3; fig. 5).	1-10
A	JP 2010088982 A (TOSHIBA CORP) 2010.04.22, see whole document.	1-10
A	US 2014/0127119 A1 (FUJIMOTO NORIKAZU et al.) 2014.05.08, see whole document.	1-10
A	WO 2021/045455 A1 (SAMSUNG HEAVY IND) 2021.03.11, see whole document.	1-10
A	JP 2019217492 A (YANMAR CO LTD) 2019.12.26, see whole document.	1-10
A	WO 2022/124464 A1 (DAEWOO SHIPBUILDING & MARINE) 2022.06.16, see whole document.	1-10

☐ Further documents are listed in the continuation of Box C

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Additional search report - patent

Application No.
PA 2022 70534

C. Documents considered to be relevant (continuation)

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant for claim No.