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(54) Benævnelse: **Stor turboladet to-takts-dieselmotor med recirkulation af udstødsgas**

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The invention relates to a large turbocharged two¿stroke diesel engine (1) of the crosshead type that is provided with an exhaust gas recirculation system (30,32) that can be activated or deactivated or can be operated with variable exhaust gas recirculation rates. In order to ensure proper matching of the turbocharger (16) , and in particular the compressor (18) of the turbocharger (16) in all operating conditions, a cylinder bypass flow path (40) that includes a controllable valve (42) enables matching the turbocharger 16 to the engine (1) in operation modes with or without exhaust gas recirculation.

Fortsættes ...

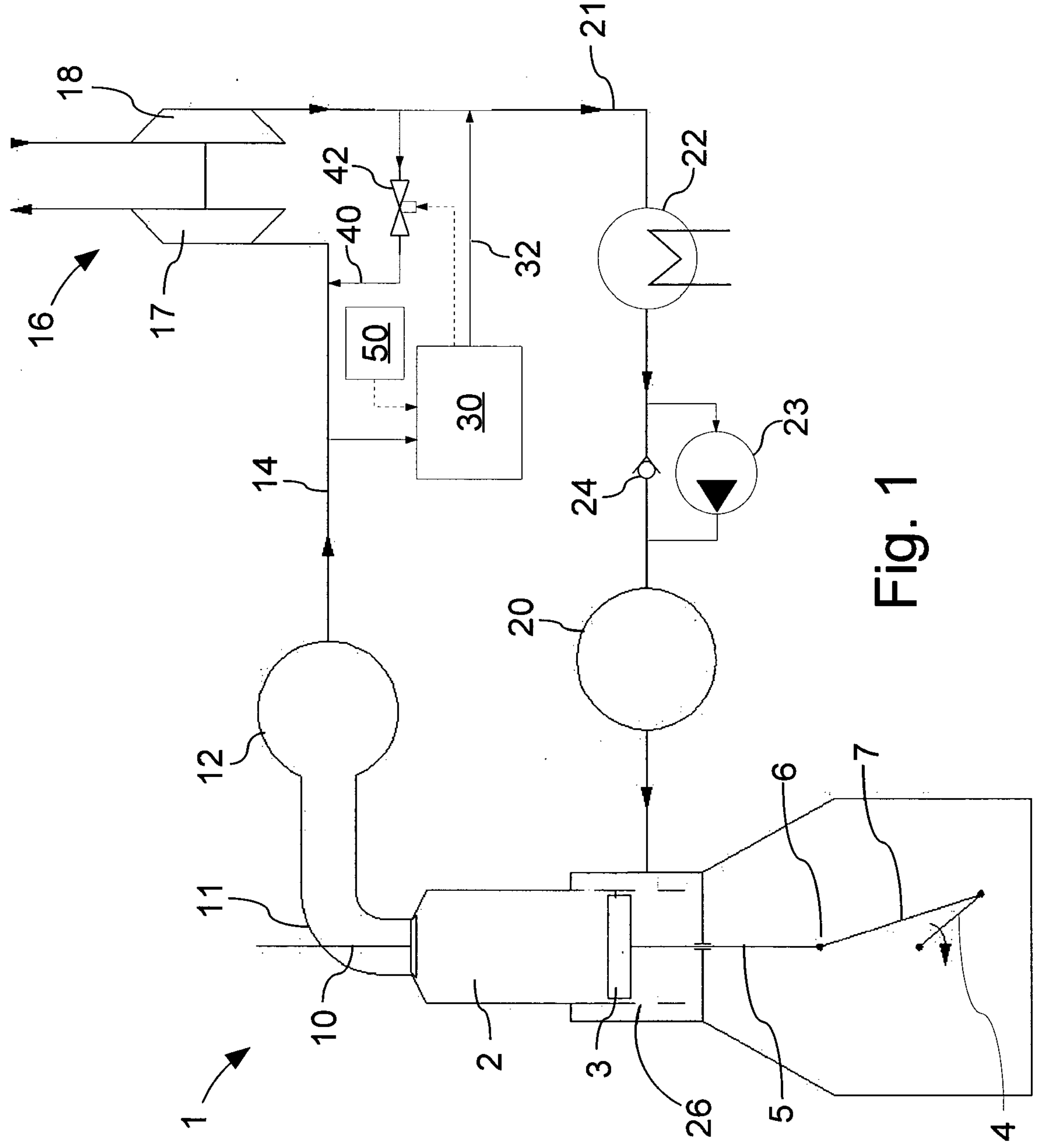

Fig. 1

LARGE TURBOCHARGED TWO-STROKE DIESEL ENGINE WITH EXHAUST GAS RECIRCULATION

The present invention relates to a large turbocharged two-stroke internal combustion piston engine of crosshead type, preferably a diesel engine with an exhaust gas purification system, in particular to a large two-stroke diesel engine of the crosshead type with an exhaust gas recirculation system.

BACKGROUND ART

Large two-stroke engines of the crosshead type are typically used in propulsion systems of large ships or as engine in power plants. Emission requirements have been and will be increasingly difficult to meet, in particular with respect to oxides of nitrogen (NO_x) levels.

Exhaust gas recirculation is a measure that is known to assist in combustion engines to reduce NO_x.

In order to meet various emission requirements such as the International Maritime Organization (IMO) Tier II and Tier III emission standards it would be advantageous to be able to operate with an exhaust gas recirculation system in large two-stroke diesel engines of the crosshead type that can be switched on and off or where the exhaust gas recirculation rate is variable.

The turbocharger of a large two-stroke engine of the crosshead type needs to be matched to the engine, for ensuring that the turbocharger does not choke or surge under any engine operating conditions and to ensure that the turbocharger is operated so that it operates with sufficient efficiency. The compressor characteristic is used to determine if the turbocharger is operating close to its maximum efficient, but with enough surge margin to ensure compressor stability. The surge margin is needed as the turbocharger operating point could approach the surge line in the compressor map during a transient, e.g. a fast engine load reduction, or during an abnormal situation.

However, when the exhaust gas recirculation rate in a large two-stroke diesel engine is changed from e.g. 40% to 0%, the massflow in the exhaust gas at the inlet to the turbine and compressor of the turbocharger will increase by approximately 30%. Thus, if the turbocharger is matched to the engine at 40% exhaust gas recirculation rate, there will be a mismatch when the exhaust gas recirculation is deactivated, since the flow capacity of the compressor would be insufficient. At this point the flow becomes sonic at the point of minimum area and is said to be chocked. The resulting efficiency and pressure both drops rapidly. The vaned compressors of the turbochargers that are in use nowadays do simply not have the full range that is necessary in order to handle the variation in flow capacity that occur when the operation is switched from operation with exhaust gas recirculation to operation without exhaust gas recirculation.

DE 10331187 discloses a large turbocharged two-stroke combustion engine that operates with a fixes EGR ratio. The engine is of the crosshead type comprising a plurality of cylinders arranged in line, an exhaust gas receiver with a large volume for equalizing pressure pulses from the exhaust of the individual cylinders to provide a substantially constant pressure at an outlet of

the exhaust gas receiver, an exhaust conduit connecting an outlet of the exhaust gas receiver to a turbine of a turbocharger, a compressor of the turbocharger driven by the turbine, the compressor delivering scavenge air via a scavenge air path that includes a scavenge air cooler to a scavenge air receiver, an exhaust gas recirculation flow path for feeding a portion of the exhaust gas into the scavenge air a cylinder bypass flow path for feeding a portion of the scavenge air into the exhaust. A low pressure turbo charger has blower in the exhaust gas recirculation flow path and a turbine in the bypass flow path. Thus, the blower is rigidly coupled to the turbine and consequently the two flows are rigidly connected by the low-pressure turbocharger 18, i.e. the flow rate in the EGR string is directly dependent on the flow rate in the bypass string.
Thus, there is a need for turbocharged two-stroke diesel engine that can be operated with varying exhaust gas recirculation rates without problems with the matching of the turbocharger to the engine.

DISCLOSURE OF THE INVENTION

On this background, it is an object of the present invention to provide a large turbocharged two-stroke diesel engine that can operate with a variable exhaust gas recirculation rate and/or with an exhaust gas recirculation system that is active or inactive.

This object is achieved by providing a large turbocharged two-stroke combustion engine of the crosshead type comprising, a plurality of cylinders arranged in line, an exhaust gas receiver with a large volume for equalizing pressure pulses from the exhaust of the individual cylinders to provide a substantially constant pressure at an outlet of the exhaust gas receiver, an exhaust conduit connecting an outlet of the exhaust gas receiver to a turbine of a turbocharger, a compressor of the turbocharger driven by the turbine, the compressor delivers scavenge air via a scavenge air path that includes a scavenge air cooler to a scavenge air receiver, an auxiliary blower associated with the scavenge air path for assisting the turbine at low engine load conditions, the scavenge air receiver is connected to the cylinders and has a large volume for reducing pressure surges caused by the inlet flow to the individual cylinders, an exhaust gas recirculation flow path including a controllable blower or compressor for feeding a portion of the exhaust gas into the scavenge air, a cylinder bypass flow path for feeding a portion of the scavenge air into the exhaust, the cylinder bypass flow path includes a controllable valve controlling the flow through the cylinder bypass flow path and an electronic control unit configured to control the exhaust gas recirculation ratio by controlling said blower or compressor in the exhaust gas recirculation flow path, and to control the opening of the valve in relation to the exhaust gas recirculation ratio.

By providing a cylinder bypass flow path and by controlling a flow of scavenge air from the compressor of the turbocharger directly to the turbine of the turbocharger during operation with exhaust gas recirculation to thereby a shift/control the operation point of the turbocharger and by controlling the exhaust gas recirculation rate it is possible to match the turbocharger well with the engine both in an operation mode with exhaust gas recirculation and in an operation mode without exhaust gas recirculation.

Only when these two flows can be controlled independently from one another it becomes possible to control the flow through the bypass in a way that ensures that the turbocharger is perfectly matched for different EGR ratios, thus achieving the advantages of the present invention.

In an embodiment the exhaust gas recirculation flow path can be activated and deactivated.

Preferably, the valve in the cylinder bypass flow path is open or partially open when the exhaust gas recirculation path is active.

In an embodiment the valve in the cylinder bypass flow path is open or partially open when the exhaust gas recirculation path is active.
In another embodiment the large turbocharged two-stroke combustion engine of the crosshead type comprises an electronic control unit configured to control the exhaust gas recirculation ratio so that it can be varied by controlling a blower in the exhaust gas recirculation flow path, and wherein the electronic control unit is configured to control the opening of the valve in relation to the exhaust gas recirculation ratio.

Preferably, the electronic control unit is configured to increase the opening of the valve with increasing exhaust gas recirculation ratio and vice versa.

In an embodiment the large turbocharged two-stroke combustion engine of the crosshead type further comprises an electronic control unit configured to control the exhaust gas recirculation ratio and wherein the electronic control unit (50) is configured to control the flow through the cylinder bypass flow path (40) in relation to the exhaust gas circulation rate and in relation to the engine load.

In an embodiment the exhaust gas recirculation path and the cylinder bypass flow path exchange heat via a heat exchanger.

The object above is also achieved by providing a method for operating a large two-stroke turbocharged diesel engine of the crosshead type with a cylinder bypass conduit and an exhaust gas recirculation system, comprising; allowing scavenge air to bypass the cylinders when exhaust gas is recycled and not allowing air to bypass the cylinders when exhaust gas is not recycled.

The object above is also achieved by providing a method for operating a large two-stroke turbocharged diesel engine of the crosshead type with a cylinder bypass conduit and an exhaust gas recirculation system, comprising controlling flow of scavenge air to bypass the cylinders in relation to the exhaust gas recirculation ratio.

Further objects, features, advantages and properties of the large two-stroke internal combustion diesel engine and method for operating a large two-stroke turbocharged diesel engine according to the invention will become apparent from the detailed description.

BRIEF DESCRIPTION OF THE DRAWINGS

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

Fig. 1 is a diagrammatic view of an exemplary embodiment of the invention,
Fig. 2 is another diagrammatic view of the engine according to the embodiment of figure 1, and
Fig. 3 is another exemplary embodiment of the engine according to the invention.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

In the following detailed description of the large turbocharged two-stroke diesel engine of the crosshead type and the method for operating a large turbocharged two-stroke diesel engine of the crosshead type according to the invention will be described by the exemplary embodiments.

Figs. 1 and 2 show a first exemplary embodiment of a large two-stroke diesel engine 1. The engine 1 may e.g. be used as the main engine in an ocean going 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 2,000 to 110,000 kW.

The engine 1 is provided with a plurality (typically between 5 and 14) of cylinders 2 arranged besides one another in line. Each cylinder 2 is provided with a reciprocating piston 3. The pistons 3 are connected to the crankshaft 4 via piston rods 5 crossheads 6 and connecting rods 7. The crossheads 6 include a crosshead bearing that is guided between guide planes.

Each cylinder 2 is provided with an exhaust valve 10 associated with their cylinder cover. The exhaust channels can be opened and closed by the exhaust valve 10. Exhaust bends 11 connect to an exhaust gas receiver 12. The exhaust gas receiver 12 is a large elongated cylindrical container disposed parallel and in close proximity to the top of the row of cylinders 2. The exhaust gas receiver 12 has a large volume enabling the exhaust gas receiver to equalize the pressure pulses that are caused by the periodic inflow of the exhaust gas from the individual cylinders 2 at the opening of the exhaust valves 12. The equalization effect of the exhaust gas receiver 12 provides for a substantially constant pressure at the outlet of the exhaust gas receiver 12. A constant pressure at the outlet of the exhaust gas receiver 12 is required since the exhaust gas driven turbocharger or turbochargers 16 that are used in large two-stroke diesel engines benefit from a constant feed pressure.

From the exhaust gas receiver 12 the exhaust gases are guided towards the turbine 17 of the turbocharger 16 via an exhaust conduit 14 (there can be a plurality of turbochargers 16). The exhaust gases are disposed into the atmosphere downstream of the turbine 17. The turbocharger 16 is a constant pressure turbocharger, i.e. the turbocharger 16 is not configured for operation with pressure pulses in the exhaust gas. The turbocharger 16 has an axial or radial turbine and is configured for exhaust gas temperatures of up to approximately 500 to 550 °C.

The turbocharger 6 also includes a compressor 18 driven by the turbine 17. The compressor 18 is connected to an air intake. The compressor 18 delivers high pressure scavenge air to a scavenge air receiver 20 via a scavenge air conduit 21 that includes a scavenge air cooler 22 and an auxiliary blower 23 with associated non-return valve 24. The auxiliary blower 23 is typically driven by an electric motor (could also be driven by a hydraulic motor) and kicks in at low load conditions (typically below 40% of the maximum continuous engine rating) to assist the compressor 18 in maintaining sufficient scavenging. When the auxiliary blower 23 is not used (typically above 40% of the maximum continuous engine rating), it is bypassed via the non return valve 24.

The scavenge air receiver 20 is a large elongated cylindrical container disposed parallel and in close proximity to the bottom of the row of cylinders 2. The scavenge air receiver 20 has a large volume enabling the scavenge air receiver 20 to compensate for the pressure drops that are caused by the periodic outflow of the scavenge air to the individual cylinders 2 at the opening of the scavenge ports 26. The compensation effect of the scavenge air receiver 26 provides for a substantially constant pressure in the scavenge air receiver so that substantially the same scavenge air pressure is available for each cylinder 2. A constant pressure in the scavenge air receiver 26 is required since the turbocharger or turbochargers 16 that are used in large two-stroke diesel engines are operated with constant feed pressure and deliver a constant feed pressure, i.e. there is no pressure pulse available for scavenging the individual cylinders 2.

The scavenge air is passed from the scavenge air receiver 20 to the scavenge air ports 26 of the individual cylinders 2.

The engine 1 is provided with an exhaust gas recirculation system. The exhaust gas recirculation system is configured to transport a portion of the exhaust gases into the scavenge air for reducing NO_x emissions. The exhaust gas recirculation system can be active or inactive or of a type that can operate with varying exhaust gas recirculation rates. The exhaust gas recirculation system includes a flow path from the exhaust gas receiver 12 or from the exhaust gas conduit 14 to the scavenge air conduit 21 or to the scavenge air receiver 20. Alternatively, the exhaust gases may be taken from the cylinders 2 directly via a valve or port (not shown).

In the exemplary embodiment of Figs. 1 and 2 the exhaust gas recirculation conduit 32 connects the exhaust gas conduit 14 to the scavenge air conduit 21.

In the embodiment shown in Figs. 1 and 2 the exhaust gas recirculation conduit 32 branches off from the exhaust gas conduit 14 at a position downstream of the exhaust gas receiver and connects to the scavenge air conduit 21 at a position downstream of the scavenge air cooler 22.

The exhaust gas recirculation conduit 32 includes various components of the exhaust gas system. These components can include cleaning equipment such as a scrubber or filter, a suction blower (driven by an electric motor or by a hydraulic motor), a cooler and one or more valves.

The blower and the valves, i.e. the components of the exhaust gas recirculation unit 30, are connected to an electronic control unit 50. The electronic control unit 50 controls the activity of the exhaust gas recirculation system on the basis of operating conditions and input from a human operator. The electronic control unit 50 can activate and deactivate the exhaust gas recirculation system and if needed variably control the exhaust gas recirculation rate, i.e. the ratio between air and exhaust gas.

The engine 1 is provided with a cylinder bypass conduit 40 that connects the scavenge air conduit 21 to the exhaust gas conduit 14.

One end of the cylinder bypass conduit 40 is connected to the scavenge air conduit 21 at a position downstream of the compressor 18 and upstream of the position where the exhaust gas recirculation conduit 32 connects to the scavenge air conduit 21. Alternatively, the one end of the cylinder bypass conduit 40 connects to the scavenge air receiver 20, as indicated by the interrupted line in Fig. 2. Other connection positions along the scavenge air conduit 21 are also possible.

The other end of the cylinder bypass conduit 40 is connected to the exhaust gas conduit 14 at a position downstream of the position where the exhaust gas recirculation conduit 32 connects to the exhaust gas conduit 14 and upstream of the inlet of the turbine 12.

Other connection positions along the exhaust gas conduit 14 or at the exhaust gas receiver 12 are also possible.

The cylinder bypass conduit 40 includes an electronically controlled valve 42 that regulates the flow of scavenge air from the scavenge air flow path 21 to the exhaust conduit 14 under command of an electronic control unit 50.

In an embodiment the valve 42 is an on/off type that is controlled by the electronic control unit 50. In this embodiment the electronic control unit 50 is configured to open the valve 42 when exhaust gas recirculation system is active, and configured to close the valve 42 when the exhaust gas recirculation system is inactive or when the exhaust gas recirculation system is operating at a low exhaust gas recirculation rate.

In another embodiment the electronically controlled valve 42 is a proportional valve. In this embodiment the electronic control unit is configured to control the opening of the electronically controlled valve 42 in relation to the exhaust gas recirculation rate and in relation to the engine load.

In an embodiment the degree of opening of the electronically controlled valve 42 is inverse to the level of the exhaust gas recirculation rate.

The exhaust gas recirculation system may be inactive for various reasons. One of the reasons can be a defect or malfunction of the exhaust gas recirculation system. Another reason for inactivity of the exhaust gas recirculation system could be the Opportunity to fuel optimize the engine with respect to a Tier II NOx emission level. The exhaust gas recirculation rate may vary, e.g. between 0% and approximately 45%.

The turbocharger 16 does not operate well or does not operate at all when it is not well matched to the engine 1 due to surging or choking. A compressor map is used to match a turbocharger to an engine. In the compressor map the pressure ratio is plotted as a function of the mass flow rate and the speed of rotation and contours of efficiency are superimposed. When matching a turbo charger to an engine, the aim is to place the operating points of the engine near or within the contours of highest efficiency but with a safe margin to the surge line.

When the exhaust gas recirculation system changes from an active state to an inactive state, the operating conditions for the turbocharger change substantially. The turbocharger 16 is namely matched to the engine 1 for operation with the exhaust gas recirculation system active (i.e. operation with an exhaust gas recirculation rate between e.g. approximately 20 and 45 % and a good match with the turbocharger 16. Without countermeasures the turbocharger 16 will not be matched well when the exhaust gas recirculation system is deactivated, since the scavenge air pressure and flow would increase by approximately 25%, which is unacceptable at high engine load and may lead to choking and overspeeding of the turbocharger and low efficiency.

Matching a turbocharger 16 for an exhaust gas recirculation engine fulfilling the IMO Tier III emission legislation or a Tier II engine running without exhaust gas recirculation (or a small amount of EGR) is a compromise between compressor stability (surge margin) and compressor/turbocharger efficiency/Fuel consumption of the engine 1. If a compressor of the turbocharger is matched with a optimal layout when running with no exhaust gas recirculation, there is an unnecessary large surge margin as exhaust gas recirculation reduces the flow rate through the compressor 18 (engine operating point moves towards the surge line). Conventional turbochargers or variable turbine area turbochargers do not have the flow range that is required to handle the variations in flow when switching between these two modes without compromising scavenge air pressure (boost pressure) and engine efficiency.

In an embodiment, the compressor 18 of the turbocharger 16 is matched for exhaust gas recirculation operation and open cylinder by-pass flow path 40. When switching to non-exhaust gas recirculation mode the cylinder by-pass flow path 40 is closed ensuring that the increase in flow and scavenge air pressure is reduced for avoiding that the compressor 18 of the turbocharger 16 chokes, and optimal running conditions in the compressor characteristic (map) are obtained. Another effect is that a lower absolute exhaust gas recirculation mass flow is required for achieving the envisaged NOx reduction, since the air flow through the cylinders 2 is reduced when the cylinder by-pass flow path 40 is open. Yet another effect is that the capacity of the exhaust gas recirculation system itself can be reduced since less suction blower power and amount of circulated exhaust gas is needed.

A negative effect of the cylinder bypass flow is an increased heat load on the engine caused by a reduced amount of scavenge gas passing through the cylinders 2. According to an embodiment the matching of the turbocharger 16 and the engine operation is as follows:

When the engine operation mode is changed to IMO Tier II, the cylinder by-pass flow path 40 is closed by means of the valve 42 under command of the electronic control unit 50, and the exhaust gas recirculation path is deactivated for all engine loads under approximately 90% of the continuous maximum rating.

The scavenge air pressure in IMO Tier II mode will increase substantially in relation to the IMO Tier III mode with active exhaust gas recirculation, but the scavenge air pressure will not exceed the matching pressure. The increase scavenging pressure at partial load conditions will itself result in a lower emissions, and allow for a vast range of possibilities to optimize the specific fuel oil consumption (SFOC).

At approximately 90% engine load the scavenging air pressure will be close to the matching pressure. At this point the turbocharger 16 has reached the matching scavenge air pressure and with further increase of the engine load the exhaust gas recirculation rate is increased gradually thus keeping the scavenging scavenge air pressure constant at 100% of the matching pressure. Thus, in Tier II mode the engine 1 is operated with a high scavenging pressure on their partial load and a bend in the scavenge air pressure at approximately 90% engine load. At 100% engine load approximately 30% of the exhaust gas recirculation rate that was used for the matching is required to keep the scavenge air pressure at the matching level. Thus, at 90% engine load there will be a small amount of exhaust gas recirculation which will provide for further and sufficient reduction in NOx emissions.

Thus, in the unlikely event that the exhaust gas recirculation flow path 32 is inoperative, there will still be 90% of the maximum continuous power rating of the engine 1 available without problems with compressor stability. Often, the turbocharger 16 will be able to handle increased pressure in emergency operation and thus, even 100% of the maximum continuous power rating of the engine 1 will typically be possible.

Alternatively, a high-pressure boiler could be installed in the exhaust gas conduit. In IMO Tier II operation mode. In this case a matching scavenging pressure is obtained at 100% engine load without using exhaust gas recirculation. Alternatively, a turbine by-pass could be installed around the turbine. The by-pass is open at 100% engine load in order to obtain the matching scavenge air pressure

The embodiment shown in figure 3 is essentially identical to the embodiment described with reference to figures 1 and 2, except that a heat exchanger 44 has been added. The heat exchanger 44 allows for heat exchange between the recirculated exhaust gases and the bypassed scavenging air. Thus, the recirculated exhaust gas is cooled and the bypass scavenging air gets even more energy that can be delivered to the turbine 17 of the turbocharger 16.

By using an cylinder by-pass when running in EGR mode the reduction in flow rate and scavenge air pressure caused by the mass and energy flow that is missed by the turbine of the turbocharger is largely compensated, i.e. the operating position of the compressor of the turbocharger on the compressor map moves away from the surge line towards safe and high efficiency areas in the compressor map.

Other advantages of the invention are significantly increased propulsion reliability in case of failure of the exhaust gas recirculation system and its components, and the same compressor stability in both IMO Tier II and III operation. Relatively high scavenging pressure in part load in IMO Tier 2 mode which will ensure low specific fuel oil consumption despite the fact that there is no exhaust gas recirculation.

Although the teaching of this application has been described in detail for purpose of illustration, it is understood that such detail is solely for that purpose, and variations can be made therein by those skilled in the art without departing from the scope of the teaching of this application.

The embodiments described above may be combined in every possible way to improve the function of the engine.

It should also be noted that there are many alternative ways of implementing the apparatuses of the teaching of this invention.

The term "comprising" as used in the claims does not exclude other elements or steps. The term "a" or "an" as used in the claims does not exclude a plurality. The single processor or other unit may fulfill the functions of several means recited in the claims.

STOR TURBOLADET TO-TAKTS-DIESELMOTOR MED RECIRKULATION AF UDSTØDSGAS

PATENTKRAV

1. Stor turboladet totakts-forbrændingsmotor (1) af typen med krydshoved omfattende:

et antal cylindre (2), der er anbragt på linje,

en udstødsgasreceiver (12) med et stort volumen til udligning af trykimpulser fra de individuelle cylindres (2) udstødning for at tilvejebringe ét i det væsentlige konstant tryk ved en udgang af udstødsgasreceiveren (12),

en udstødningskanal (10), der forbinder en udgang af udstødsgasreceiveren (12) med en turboladers (16) turbine (17),

en kompressor (18) til turboladeren (12) drevet af turbinen (17), hvilken kompressor (18) leverer skylleluft via en bane (21) til skylleluft, der indbefatter en afkøler (22) til skylleluft, til en skylleluftreceiver (20),

en hjælpeblæser (23), der er forbundet med skylleluftbanen (16) for at assistere turbinen (17) ved lav belastningstilstand for motoren,

skylleluftreceiveren (20) er forbundet med cylindrene (2) og har et stort volumen til reduktion af trykstød, der forårsages af indløbsstrømmen til de individuelle cylindre, en strømningsbane (32) til recirkulation af udstødsgas indbefattende en styrbar blæser eller kompressor til tilførsel af en del af udstødsgassen ind i skylleluften,

en strømningsbane for cylinderomledning (40) til tilførsel af en del af skylleluften ind i udstødningen,

en styrbar ventil (42), der styrer strømmen gennem strømningsbanen for cylinderomledning (40), og

en elektronisk styringsenhed (50), der er konfigureret til:

at styre recirkulationsforholdet for udstødsgas ved styring af blæseren eller kompressoren i strømningsbanen for recirkulation af udstødsgas, og til at styre ventilens (42) åbning i forhold til recirkulationsforholdet for udstødsgas.

2. Stor turboladet totakts-forbrændingsmotor (1) af typen med krydshoved ifølge krav 1, hvor strømningsbanen (40) for recirkulation af udstødsgas kan aktiveres og deaktiveres.

3. Stor turboladet totakts-forbrændingsmotor (1) af typen med krydshoved ifølge krav 2, hvor ventilen (42) i strømningsbanen for cylinderomledning (40) er åben eller delvis åben, når recirkulationsbanen (32) for udstødsgas er aktiv.

4. Stor turboladet totakts-forbrændingsmotor af typen med krydshoved ifølge et hvilket som helst af kravene 1 til 3, hvor den elektroniske styringsenhed (50) er konfigureret til at øge ventilens (42) åbning ved øgning af recirkulationsforholdet for udstødsgas og omvendt.

5. Stor turboladet totakts-forbrændingsmotor af typen med krydshoved ifølge krav 1, der endvidere omfatter en elektronisk styringsenhed, der er konfigureret til at styre recirkulationsforholdet for udstødsgas, og hvor den elektroniske styringsenhed er konfigureret til at styre strømningen gennem strømningsbanen for cylinderomledning i forhold til udstødsgassens cirkulationshastighed og i forhold til motorens belastning.

6. Stor turboladet totakts-forbrændingsmotor af typen med krydshoved ifølge krav 1, hvor recirkulationsbanen (32) for udstødsgas og strømningsbanen (40) for omledning i forhold til cylinder udveksler varme via en varmeveksler (44).

7. Fremgangsmåde til anvendelse af en stor turboladet totakts-dieselmotor af typen med krydshoved med en turbolader, en cylinderomledningskanal og et recirkulationssystem for udstødsgas, kendetegnet ved, at styring af strømmen af skylleluft, der skal gå uden om cylindrene, i forhold til recirkulationsforholdet for udstødsgas ved styrbart at tillade, at skylleluft går uden om cylindrene, når udstødsgassen recirkulerer, og ikke tillade, at luft går uden om cylindrene, når udstødsgassen ikke recirkulerer, således at turboladeren er velafstemt, når udstødsgassen recirkulerer, og når udstødsgassen ikke recirkulerer.

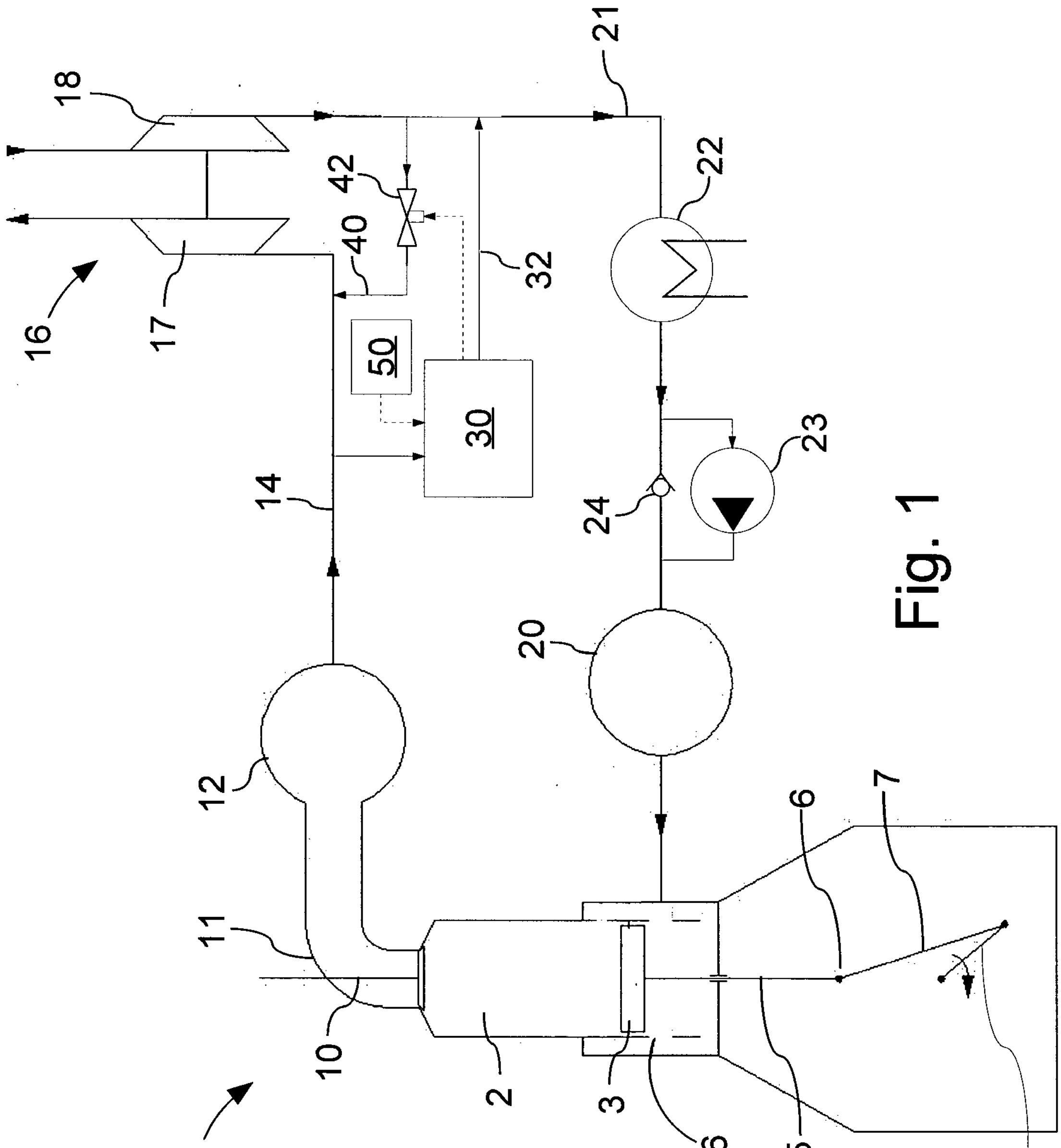

Fig. 1

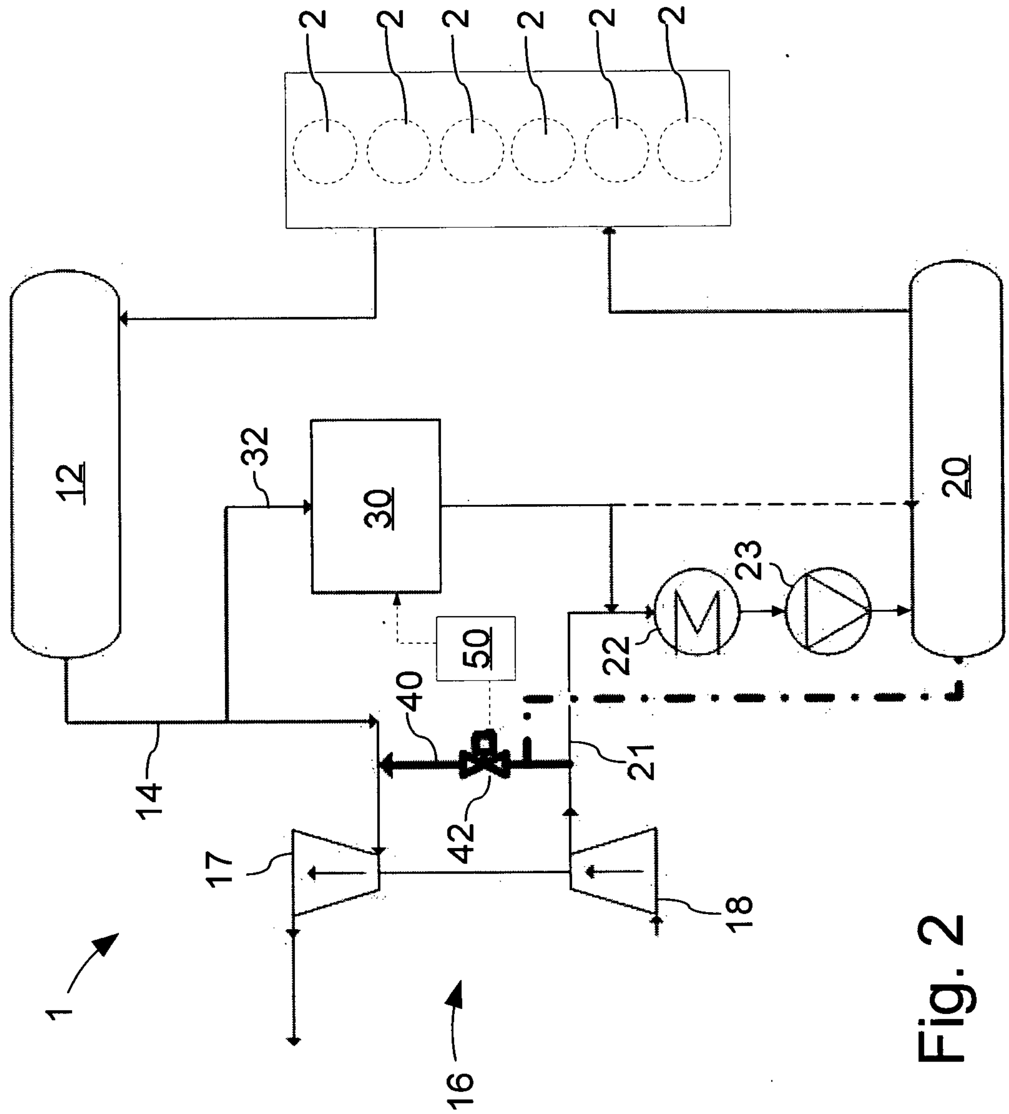

Fig. 2

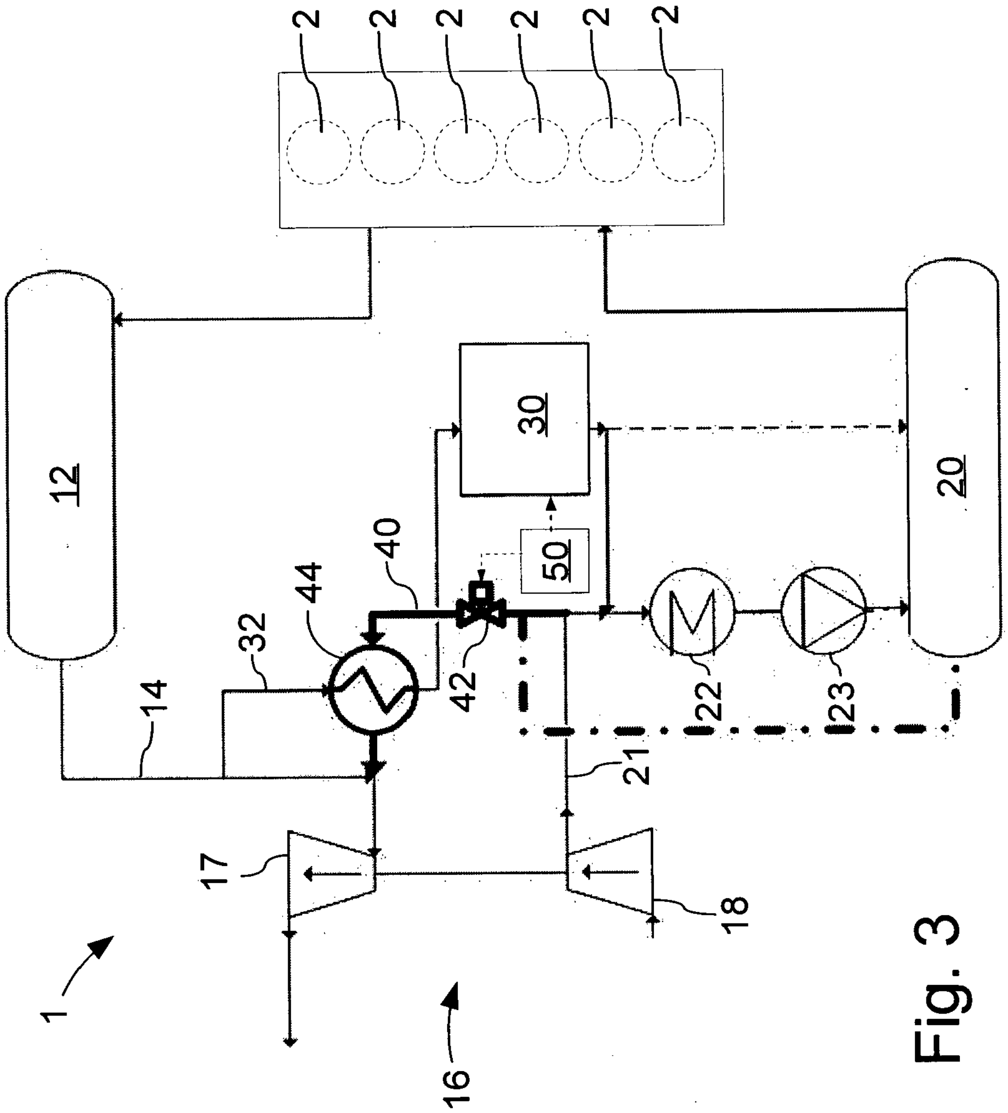

Fig. 3