



(11)

EP 2 572 089 B1

(12)

EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention
of the grant of the patent:

18.03.2015 Bulletin 2015/12

(51) Int Cl.:

F02B 25/08 ^(2006.01) **F02B 29/04** ^(2006.01)
F02B 37/013 ^(2006.01) **F02B 75/28** ^(2006.01)
F02M 25/07 ^(2006.01) **F02B 37/16** ^(2006.01)
F02B 37/18 ^(2006.01)

(21) Application number: **11721839.6**

(22) Date of filing: **16.05.2011**

(86) International application number:

PCT/US2011/000864

(87) International publication number:

WO 2011/146111 (24.11.2011 Gazette 2011/47)

(54) **EGR CONSTRUCTION FOR OPPOSED-PISTON ENGINES**

AGR-KONSTRUKTION FÜR MOTOREN MIT ENTGEGENGESETZTEN KOLBEN

CONSTRUCTION EGR POUR MOTEUR À PISTONS OPPOSÉS

(84) Designated Contracting States:

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

(30) Priority: **16.08.2010 US 401598 P**

18.05.2010 US 395845 P

(43) Date of publication of application:

27.03.2013 Bulletin 2013/13

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Description

BACKGROUND

[0001] The field is internal combustion engines. Particularly, the field relates to ported, uniflow-scavenged, opposed-piston engines with exhaust gas recirculation. More particularly, the field includes two-stroke, opposed-piston engines with one or more ported cylinders and uniflow scavenging in which an exhaust gas recirculation (EGR) construction provides a portion of the exhaust gases produced by the engine for mixture with charge air to control the production of NO_x during combustion.

[0002] As seen in FIG. 1, an internal combustion engine is illustrated by way of an opposed-piston engine that includes at least one cylinder 10 with a bore 12 and longitudinally-displaced exhaust and intake ports 14 and 16 machined or formed therein. Fuel injector nozzles 17 are located in or adjacent injector ports that open through the side of the cylinder, at or near the longitudinal center of the cylinder. Two pistons 20, 22 are disposed in the bore 12 with their end surfaces 20e, 22e in opposition to each other. For convenience, the piston 20 is referred as the "exhaust" piston because of its proximity to the exhaust port 14; and, the end of the cylinder wherein the exhaust port is formed is referred to as the "exhaust end". Similarly, the piston 22 is referred as the "intake" piston because of its proximity to the intake port 16, and the corresponding end of the cylinder is the "intake end".

[0003] Operation of an opposed-piston engine with one or more cylinders such as the cylinder 10 is well understood. In this regard, and with reference to FIG. 2, in response to combustion occurring between the end surfaces 20e, 22e the opposed pistons move away from respective top dead center (TDC) positions where they are at their closest positions relative to one another in the cylinder. While moving from TDC, the pistons keep their associated ports closed until they approach respective bottom dead center (BDC) positions in which they are furthest apart from each other. The pistons may move in phase so that the exhaust and intake ports 14, 16 open and close in unison. Alternatively, one piston may lead the other in phase, in which case the intake and exhaust ports have different opening and closing times.

[0004] In many opposed-piston constructions, a phase offset is introduced into the piston movements. As shown in FIG. 1, for example, the exhaust piston leads the intake piston and the phase offset causes the pistons to move around their BDC positions in a sequence in which the exhaust port 14 opens as the exhaust piston 20 moves through BDC while the intake port 16 is still closed so that combustion gasses start to flow out of the exhaust port 14. As the pistons continue moving away from each other, the intake port 16 opens while the exhaust port 14 is still open and a charge of pressurized air ("charge air") is forced into the cylinder 10, driving exhaust gasses out of the exhaust port 14. The displacement of exhaust gas from the cylinder through the exhaust port while admitting

charge air through the intake port is referred to as "scavenging". Because the charge air entering the cylinder flows in the same direction as the outflow of exhaust gas (toward the exhaust port), the scavenging process is referred to as "uniflow scavenging".

[0005] As the pistons move through their BDC locations and reverse direction, the exhaust port 14 is closed by the exhaust piston 20 and scavenging ceases. The intake port 16 remains open while the intake piston 22 continues to move away from BDC. As the pistons continue moving toward TDC (FIG. 2), the intake port 16 is closed and the charge air in the cylinder is compressed between the end surfaces 20e and 22e. Typically, the charge air is swirled as it passes through the intake port 16 to promote good scavenging while the ports are open and, after the ports close, to mix the air with the injected fuel. Typically, the fuel is diesel which is injected into the cylinder by high pressure injectors. With reference to FIG. 1 as an example, the swirling air (or simply, "swirl") 30 has a generally helical motion that forms a vortex in the bore which circulates around the longitudinal axis of the cylinder. As best seen in FIG. 2, as the pistons advance toward their respective TDC locations in the cylinder bore, fuel 40 is injected through the nozzles 17 directly into the swirling charge air 30 in the bore 12, between the end surfaces 20e, 22e of the pistons. The swirling mixture of charge air and fuel is compressed in a combustion chamber 32 defined between the end surfaces 20e and 22e when the pistons 20 and 22 are near their respective TDC locations. When the mixture reaches an ignition temperature, the fuel ignites in the combustion chamber, driving the pistons apart toward their respective BDC locations.

[0006] As illustrated in FIG. 2, fuel is directly injected through the side of the cylinder ("direct side injection") into the cylinder bore and the movement of the fuel interacts with the residual swirling motion of the charge air in the bore. As the engine operating level increases and the heat of combustion rises, an increasing amount of nitrogen oxide (NO_x) is produced. However, increasingly stringent emission requirements indicate the need for a significant degree of NO_x reduction. One technique reduces NO_x emission by exhaust gas recirculation ("EGR"). EGR has been incorporated into spark-ignited 4-stroke engine constructions and 2-stroke, compression-ignition engines with a single piston operating in each cylinder. However, such EGR constructions are not immediately applicable to 2-stroke, opposed-piston engines with uniflow scavenging because of the need to generate a pressure differential that pumps the exhaust gas into the inflowing stream of air in an opposed-piston engine. Therefore, there is a need for effective EGR constructions that are adapted to the designs and operations of 2-stroke opposed-piston engines with uniflow scavenging. The state of the art is represented by US-6,925,971 and by US-2009/159022.

SUMMARY

[0007] The present invention provides a ported, uniflow-scaverged, opposed piston engine, in accordance with claims which follow.

[0008] A solution to the problem is to reduce the NOx emissions of a two-stroke opposed-piston engine with uniflow scavenging by exhaust gas recirculation through one or more ported cylinders of the engine. The engine includes at least one cylinder with piston-controlled exhaust and intake ports and a charge air channel to provide charge air to at least one intake port of the engine.

[0009] In one aspect, EGR is provided by an EGR loop having an input coupled to an exhaust port of the cylinder and a loop output coupled to the charge air channel. A pressure differential provided between the exhaust gas and the charge air channel causes the exhaust gas to flow through the EGR loop to the charge air channel where exhaust gas and air are mixed and provided to the at least one intake port.

[0010] In another aspect, EGR is provided by retention of residual exhaust gasses in the ported cylinder when scavenging ceases.

BRIEF DESCRIPTION OF THE DRAWINGS

[0011]

FIG. 1 is a side sectional partially schematic drawing of a cylinder of a prior art opposed-piston engine with opposed pistons near respective bottom dead center locations, and is appropriately labeled "prior Art".

FIG. 2 is a side sectional partially schematic drawing of the cylinder of FIG. 1 with the opposed pistons near respective top dead center locations where end surfaces of the pistons define a combustion chamber, and is appropriately labeled, "Prior Art".

FIG. 3 is a conceptual schematic diagram of an Internal combustion engine of the opposed-piston type in which aspects of an air management system with EGR are illustrated.

FIG. 4 is a conceptual schematic drawing illustrating preferred constructions for EGR in the ported, uniflow scavenging, internal combustion engine of FIG. 3.

FIG. 5 is a schematic drawing of a preferred EGR construction for the ported, uniflow scavenging, internal combustion engine of FIG. 3.

FIG. 6 is a schematic drawing of an alternate EGR construction for a ported, uniflow scavenging, opposed-piston engine without a turbocharger.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0012] The EGR constructions described in this specification are presented in an explanatory context that includes a ported, uniflow-scavenging internal combustion engine having at least one cylinder in which a pair of pistons is disposed with their end surfaces in opposition. A "ported" cylinder includes one or more of intake and exhaust ports formed or machined in a sidewall thereof. This explanatory context is intended to provide a basis for understanding various EGR construction embodiments by way of illustrative examples.

[0013] In FIG. 3, an internal combustion engine 49 is embodied by an opposed-piston engine having at least one ported cylinder 50. For example, the engine may have one ported cylinder, two ported cylinders, three ported cylinders, or four or more ported cylinders. For purposes of illustration, in the examples to be illustrated and described the engine is an engine of the opposed-piston type that is presumed to have a plurality of ported cylinders. In this regard, each cylinder 50 has a bore 52 and exhaust and intake ports 54 and 56 formed or machined in respective ends thereof. The exhaust and intake ports 54 and 56 each include one or more circumferential arrays of openings in which adjacent openings are separated by a solid bridge. (In some descriptions, each opening is referred to as a "port"; however, the construction of a circumferential array of such "ports" is no different than the port constructions shown in FIG. 3.) Exhaust and intake pistons 60 and 62 are slidably disposed in the bore 52 with their end surfaces 61 and 63 opposing one another. The exhaust pistons 60 are coupled to a crankshaft 71, the intake pistons are coupled to the crankshaft 72.

[0014] When the pistons 60 and 62 of a cylinder 50 are at or near their TDC positions, a combustion chamber is defined in the bore 52 between the end surfaces 61 and 63 of the pistons. Fuel is injected directly into the combustion chamber through at least one fuel injector nozzle 100 positioned in an opening through the sidewall of a cylinder 50.

[0015] With further reference to FIG. 3, the engine 49 includes an air management system 51 that manages the transport of charge air provided to, and exhaust gas produced by, the engine 49. A representative air management system construction includes a charge air subsystem and an exhaust subsystem. In the air management system 51, the charge air subsystem includes a charge air source that receives intake air and processes it into charge air, a charge air channel coupled to the charge air source through which charge air is transported to the at least one intake port of the engine, and at least one air cooler in the charge air channel that is coupled to receive and cool the charge air (or a mixture of gasses including charge air) before delivery to the intake port or ports of the engine. Such a cooler can comprise an air-to-liquid and/or an air-to-air device, or another cooling

device. Hereinafter, such a cooler is denoted as a "charge air cooler". The charge air subsystem also includes a supercharger that pumps charge air in the charge air channel to intake ports of the engine. The exhaust subsystem includes an exhaust channel that transports exhaust products from exhaust ports of the engine to an exhaust pipe.

[0016] With reference to FIG. 3, the preferred charge air subsystem includes a supercharger 110, which can be driven by an electrical motor, or by a gear, chain, or belt apparatus coupled to a crankshaft. For example, but without limitation, in FIGS. 4, 5, and 6 the supercharger 110 is coupled by a belt linkage to the crankshaft 72 to be driven thereby. The supercharger 110 can be a single-speed or multiple-speed device, or a fully variable-speed device. Preferably, but not necessarily, the air management system 51 includes a turbo-charger 120 with a turbine 121 and a compressor that rotate on a common shaft 123. The turbine 121 is coupled to the exhaust subsystem and the compressor 122 is coupled to the charge air subsystem. The turbine 121 can be a fixed-geometry or a variable-geometry device. The turbo-charger 120 extracts energy from exhaust gas that exits the exhaust ports 54 and flows into the exhaust channel 124 directly from the exhaust ports 54, or from an exhaust manifold 125. In this regard, the turbine 121 is rotated by exhaust gas passing through it. This rotates the compressor 122, causing it to generate charge air by compressing intake air. The charge air output by the compressor 122 flows through a conduit 126 to a charge air cooler 127, whence it is pumped by the supercharger 110 to the intake ports. Air compressed by the supercharger 110 is output from the supercharger through a charge air cooler 129 to an intake manifold 130. The intake ports 56 receive charge air pumped by the supercharger 110, through the intake manifold 130. Preferably, but not necessarily, in multi-cylinder opposed-piston engines, the intake manifold 130 is constituted of an intake plenum that communicates with the intake ports 56 of all cylinders 50.

[0017] Exhaust Gas Management: It is desirable to modify or adapt an air management construction for an internal combustion engine of the ported-cylinder type in order to reduce NOx emissions produced by combustion. It is particularly desirable to control such emissions by recirculating exhaust gas through the ported cylinders of an opposed-piston engine. The recirculated exhaust gas is mixed with charge air to lower peak combustion temperatures, which lowers NOx emissions. This process is referred to as exhaust gas recirculation ("EGR"). An EGR construction can utilize exhaust gasses transported in an EGR channel external to the cylinder into the incoming stream of fresh intake air as per the valve-controlled recirculation channel 131 in FIG. 3. Alternatively, or additionally, an EGR process can utilize residual exhaust gasses that are retained in the cylinders 50 when scavenging ceases. In the case of external EGR, the exhaust gas is pumped into the inflowing stream of air. A source of pressure in communication with the EGR channel creates a

pressure differential that causes exhaust gas to flow through the EGR channel into the charge air subsystem. In some aspects, a virtual pump exists when the exhaust gas to be recirculated is obtained from a source guaranteed to be at a higher pressure than the point where it is fed into the intake stream of charge air. In other aspects, an active pump, such as the supercharger 110, is used to pump the exhaust gas to be recirculated into the charge air that the supercharger is pumping to the intake ports. In these aspects, use of a supercharger provides an additional variable for controlling EGR operations in an opposed-piston engine. In some aspects recirculated exhaust gas is cooled by way of one or more EGR coolers, which can comprise air-to-liquid and/or air-to-air devices. In other aspects, recirculated exhaust gas is cooled by one or more charge air coolers alone or in combination with one or more EGR coolers.

[0018] First EGR loop construction: In some aspects, the internal combustion engine seen in FIG. 3 includes a first EGR loop construction. With reference to FIG. 4, a first EGR loop construction for a uniflow scavenging, ported, opposed-piston application circulates exhaust gas from any source of exhaust gas exiting from one or more cylinders. For example, but without excluding other exhaust gas sources, the first EGR loop construction includes an EGR port 55 positioned inboard of exhaust port 54; that is to say, the EGR port 55 is positioned between the exhaust port 54 and the longitudinal midpoint of the cylinder 50. The EGR port construction includes one or more port openings as needed by any particular design. In response to combustion, while moving toward BDC, the exhaust piston 60 moves past the EGR port 55, opening the EGR port to a cylinder bore pressure that is guaranteed to be higher than the pressure in the intake manifold 130. This pressure differential pumps a portion of exhaust gas from the EGR port 55 through a plenum or manifold (not shown) into a conduit 133 controlled by a one-way check valve 134, and then into the intake manifold 130 where it is mixed with the charge air and recirculated therewith into the cylinder bore. Preferably, but not necessarily, the exhaust gas enters the charge air output by the supercharger 110 prior to the inlet of the charge air cooler 129. It is desirable that the ratio of the cylinder pressure when the EGR port opens ("EGR open") to the intake manifold pressure not exceed a threshold value beyond which choked flow conditions may occur in the one-way check valve 134. This pressure ratio is affected by the size of the EGR port and its location with respect to the longitudinal center of the cylinder (the closer to the center, the higher the pressure) as well as the state of the air charge system (boost, turbine back pressure, etc.).

[0019] With further reference to FIG. 4, in a variant of the first loop EGR construction, the exhaust gas flowing in the conduit 133 is mixed with charge air output by the supercharger 110 through a mixer 135 that can be constituted, for example, as a venturi. The exhaust gas is input to the mixer 135 through a valve 136; pressurized

charge air output by the supercharger 110 is provided to a mixing input of the mixer 135. The mixture of pressurized charge air and exhaust gas produced by the mixer 135 is provided to the input of the charge air cooler 129 (or, alternately, to the input of the charge air cooler 127). The valve 136 is operated by a signal output by an engine control unit (ECU) 149.

[0020] In some aspects it is desirable to dampen fluctuations in the flow of exhaust gas. In such a case and with reference to FIG. 4, an accumulator 145 is provided in the first loop in series between the EGR port 55 and the input to the valve 136. In some other aspects, it is desirable to cool the exhaust gas before mixing it with charge air. In such a case, an EGR cooler 146 is provided in the first loop in series between the check valve 134 and the input to the valve 136. Alternatively, the loop construction can be 134, 136, 146. In the case where both the EGR accumulator 145 and the cooler 146 are used, it is preferred, but not required, that the EGR cooler 146 be positioned in series between the output of the EGR accumulator 145 and the input to the valve 136.

[0021] Second EGR loop construction: In some aspects, the internal combustion engine 49 seen in FIG. 3 can include another EGR loop construction. With reference to FIGS. 3 and 4, a second EGR loop construction includes the conduit 131 and a valve 138 to shunt a portion of the exhaust gas from the exhaust manifold 54 to the input of a charge air cooler so that the portion is cooled. In order to promote optimal exhaust gas/charge air mixing, it is desirable to add a device through which the exhaust gasses and the charge air flow together and mix. When it is desirable to introduce the exhaust gas into the charge air at a location that is remote from the intake manifold 56, the exhaust gas portion is shunted to the input of the charge air cooler 127. This loop subjects the exhaust gas to the cooling effects of two charge air coolers (127 and 129). If less cooling is merited, the valve 138 can be constituted as a three-way valve (as best seen in FIG. 4), and the exhaust gas portion can be shunted around the cooler 127 to the input of the supercharger 110. This alternative subjects the exhaust gas portion to cooling by only the charge air cooler 129. A dedicated EGR cooler that cools only exhaust gas can be incorporated into the second loop, if needed. For example, an EGR cooler can be placed in the conduit 131, in series with the valve 138, or in series with the output port of the valve 138 and the input to the supercharger 110. In some aspects, the valve 138 is constituted as a single three-way device. Alternatively, the valve 138 is constituted as a pair of valves, each in a respective branch of a Y-connection from the conduit 131, in which one valve controls the provision of exhaust gas to the input of the cooler 127 and the other controls the provision of exhaust gas to the input of the supercharger 110.

[0022] EGR using retained exhaust gas: In a uniflow or loop-scavenged internal combustion engine, it is sometimes desirable to trap or retain a residual amount of exhaust gas in a cylinder after scavenging ceases.

The residual exhaust gas can be used to adjust the initial conditions for combustion to a point advantageous for reducing NOx emissions. Depending on the configuration of the turbo-machinery, at low and medium speeds and loads, uniflow-scavenged engines may exhibit incomplete scavenging. Since the residual exhaust gas inside the cylinder is hot, the resulting temperature of the new charge of air may be substantially elevated, therefore this method is best suited for reducing NOx under partial engine load conditions.

[0023] The amount of charge air that is fed into a cylinder each cycle can be used to alter the amount of residual exhaust gas left in the cylinder. In this regard, adjusting the amount of charge air that is fed into the cylinder in any given cycle of operation can be used to "tune" the amount of exhaust gas retained in the cylinder for the next combustion occurrence. In one aspect of retained exhaust gas EGR, seen in FIG. 4, a bypass conduit loop 148 including a valve 139 is placed in parallel with the supercharger 110. The valve 139 is operated to control the amount of charge air pumped into the engine by the supercharger 110. Setting the amount of charge air pumped allows control of the amount of exhaust gas scavenged, and, consequently, the amount of exhaust gas retained in any cylinder following scavenging. In this regard, if a high manifold pressure is desired (as would be indicated for high engine load conditions) the valve 139 is fully shut and charge air is delivered to the engine at a high rate. As the valve 139 is increasingly opened, an increasing amount of charge air pumped by the supercharger 110 is returned to the inlet of the supercharger, which proportionately reduces the amount of charge air delivered to the engine. Thus, the charge air/fuel ratio is reduced and the amount of exhaust gas retained in any cylinder is increased. Among the benefits realized by this aspect of retained exhaust gas EGR are NOx reduction and reduction of pumping load imposed on the engine by the supercharger 110.

[0024] An increase in the pressure felt by exhaust gas flowing to the turbine ("backpressure") can also be used to alter the amount of residual exhaust gas left in the cylinder. In this regard, adjusting the amount of backpressure in any given cycle of operation can be used to "tune" the amount of residual exhaust gas for the next combustion occurrence. Therefore, in another aspect of retained exhaust gas EGR, seen in FIG. 4, a variable valve 140 is placed in series with exhaust gas output. The setting of the valve 140 directly influences the backpressure felt upstream of the valve and, consequently, the amount of exhaust gas retained in any cylinder after scavenging. In FIG. 4, the valve 140 is placed in series with the output of the turbine 121. In this case, any backpressure resulting from the setting of the valve is distributed over all cylinders of the engine. In an alternative aspect, an equivalent valve 140a can be placed in series between the input to the turbine 121 and an exhaust manifold that collects the exhaust output of one or more cylinders. In yet another alternative aspect, the equivalent

valve 140a can be placed in series with an exhaust manifold or exhaust port of each of a plurality of cylinders.

[0025] Turbine bypass construction: Referring again to FIG. 4, a bypass conduit loop 143 including a valve 144 is placed in parallel with the turbine 121. The valve 144 is operated to control the amount of exhaust gas flowing from the engine into the turbine 121. Setting the valve 144 to bypass the turbine 121 allows exhaust energy to be dumped into the exhaust pipe 128 without operating the turbine 121 and compressor 122. This keeps the exhaust gas at a higher temperature level and increases after-treatment conversion efficiencies (for particulate filters and catalytic devices, for example) at engine warm-up during partial engine load conditions such as from a cold start. Further, setting the valve 144 to bypass the turbine 121 during engine operation under partial engine load conditions reduces turbo-charger operation, and allows more exhaust gas to be driven over the supercharger 110 (via valve 138, for example) while also delivering exhaust gas at a higher temperature to the exhaust pipe 128 to increase after-treatment conversion efficiencies. Another construction for varying the amount of exhaust gas flowing from the engine into the turbine 121 includes a turbine with a variable geometry construction to control pressure in the exhaust conduit 124, upstream of the valve 144. Using a variable geometry turbine (VGT) instead of a fixed geometry turbine does not necessarily eliminate the need for a turbine bypass valve such as the valve 144. A VGT has only a limited mass flow range where it works at acceptable efficiencies. Outside this range, a turbine bypass valve can be used to control the mass flow and intake pressure of the engine 49.

[0026] Preferred EGR Embodiment: A preferred EGR construction for a ported opposed-piston engine with uniflow scavenging is shown in FIG. 5. In the preferred construction, exhaust gas flows from the exhaust port or ports 54 of the engine, through the conduit 124 to the turbine 121, whence it passes through after-treatment conversion (not shown) and flows out the exhaust pipe 128. Preceding the input to the turbine 121, a portion of the exhaust gas is shunted from the conduit 124 via 131 and from there through the valve 138' to the input of the charge air cooler, 127 where it mixes with the incoming stream of fresh air. The exhaust gas and air are mixed and cooled in the charge air cooler 127, and the cooled gas/air mixture is input to the supercharger 110. The supercharger 110 compresses the gas/air mixture, and the compressed mixture is input to the charge air cooler 129. The cooled, compressed mixture then enters the cylinder 50 via the intake port 56. Optionally, the intake throttle valve 141 and the turbine, bypass valve 144 are included for high precision control of the ratio of recirculated exhaust gas to fresh air.

[0027] Exhaust configuration and control: The EGR and turbine bypass constructions illustrated in FIGS. 4 and 5 can be implemented in a ported engine of the uniflow scavenging type singly, or in any combination of two

or more constructions, or portions thereof, as required for a specific design. One example is an EGR configuration in which uncooled exhaust gas retained in a cylinder following scavenging is combined or mixed with recirculated exhaust gas that is cooled and mixed with charge air provided to the cylinder. The relative amounts of retained and recirculated exhaust gas can be varied in order to precisely control the EGR rate and temperature. An intake throttle valve 141 can be placed in the stream of fresh air flowing into the compressor 122 in order to more precisely control the ratio of recirculated exhaust gas to fresh air. If implemented on a per-cylinder basis, a high-speed individual EGR and charge air/fuel trim is provided to correct cylinder-to-cylinder variations caused by flow dynamics and/or manufacturing tolerances.

[0028] An EGR control process for an EGR system that utilizes one or more of the constructions illustrated in FIGS. 4 and 5, or any combination thereof, is executed by the ECU 149 in response to specified engine operating conditions by automatically operating any one or more of the valves 136, 138, 139, 140, 140a, and 144, the intake throttle valve 141, and the supercharger 110, if a multi-speed or variable speed device is used, and the turbo-charger 120, if a variable-geometry device is used. Of course, operation of valves, throttles, and associated elements used for EGR can include any one or more of electrical, pneumatic, mechanical, and hydraulic actuating operations. For fast, precise automatic operation, it is preferred that the valves be high-speed, computer-controlled devices with continuously-variable settings. Each valve has a, first state in which it is open (to some setting controlled by the ECU 149) to allow gas to flow through it, and a second state in which it is closed to block gas from flowing through it.

[0029] Preferably an EGR control process automatically operates an EGR system incorporating one or more constructions described and illustrated herein based upon one or more parameters relating to recirculated exhaust gas and to a mixture of recirculated exhaust gas and charge air. Parameter values are determined by a combination of one or more of sensors, calculations, and table lookup so as to manage the values of individual parameters and one or more ratios of EGR and mixture parameters in one or more cylinders.

[0030] Alternate EGR Embodiment: An alternate EGR construction is shown in FIG. 6 in a two-stroke opposed-piston engine with ported cylinders and uniflow scavenging in which only a supercharger provides scavenging pressure. Presume that after-treatment conversion is implemented by emission control devices that include a diesel oxidation catalyst (DOC) to reduce CO and hydrocarbons, a diesel particulate filter (DPF) to reduce soot emissions and a selective catalytic reduction device to reduce NOx emissions. All of these devices require addition of heat for operation, and the absence of a turbocharger reduces the competition for heat derived from exhaust gas, while also lowering the power density of the engine.

Moreover, the DPF and DOC now can be closely coupled at an exhaust manifold, where a turbocharger typically is mounted. Furthermore, elimination of a turbocharger and its required ducting reduces the size of the opposed-piston engine and also reduces the loss of exhaust heat by convection from the turbocharger housing and ducting. Preferably, although not necessarily, exhaust gas for recirculation is extracted from the outlet of the DPF where it is free of particulates and can be cooled and plumbed to the inlet of the supercharger. Although the exhaust gas is cooler after the DPF, it can be cooled further with an EGR cooler.

[0031] The alternate EGR embodiment for a two-stroke, ported, uniflow-scavenging, opposed-piston engine is illustrated in FIG. 6. Preferably, the engine does not include a turbocharger. Exhaust gas flows from the exhaust manifold 125, through the conduit 124, through the DOC 150 and the DPF 151, then through the valve 140 and out the exhaust pipe 128. A portion of the exhaust gas is diverted by a pressure change determined by the setting of the valve 140 into the input of an EGR cooler 142. Cooled exhaust gas output by the EGR cooler 142 is metered through a valve 147 into the air stream entering the supercharger 110. The intake throttle valve 141 can be placed in the airstream flowing to the supercharger, upstream of the output of the valves 147, in order to more precisely control the ratio of recirculated exhaust gas to air by creating a slight vacuum. Since the alternate EGR loop is being drawn through the supercharger 110, the time heeded to empty the exhaust gas from the charge air cooler 129 is much reduced, thereby improving the transient response. If the supercharger 110 is driven directly from the engine, it will achieve high flow and high speed together with the engine. The supercharger capacity enables the needed exhaust gas to be pumped as required at high engine speed and load as required to meet stringent emission requirements. The supercharger bypass valve 139 permits the pressure produced by the supercharger to be continuously varied.

[0032] Although EGR constructions have been described with reference to a ported opposed engine construction with two crankshafts, it should be understood that various aspects of these constructions can be applied to opposed-piston engines with one or more crankshafts. Moreover, various aspects of these EGR constructions can be applied to opposed-piston engines with ported cylinders disposed in opposition, and/or on either side of one or more crankshafts. Accordingly, the protection afforded to these constructions is limited only by the following claims.

[0033] The present invention also provides for arrangements defined in the following paragraphs numbered 13-15 and 21-28 not falling under the scope of the claims.

13. A ported, uniflow-scavenged, opposed-piston engine (49) including at least one cylinder (50) with piston-controlled exhaust and intake ports (54, 56)

and a charge air channel coupled to at least one intake port (56) of the engine, in which an exhaust gas recirculation (EGR) loop has a loop input coupled to an EGR port (55) of the cylinder located between the exhaust port (54) and the longitudinal center of the cylinder, and a loop output coupled to the charge air channel.

14. The ported, uniflow-scavenged, opposed-piston engine of paragraph 13, in which the EGR loop includes a valve (136) having an input coupled to the loop input, and an output coupled to a mixer (135) in the charge air channel.

15. The ported, uniflow-scavenged, opposed-piston engine of paragraph 14, in which the EGR loop further includes an EGR cooler (146) in series with the valve.

21. A ported, uniflow-scavenged, opposed-piston engine (49) including at least one cylinder (50) with piston-controlled exhaust and intake ports (54 and 56) and a charge air channel to provide charge air to at least one intake port (56) of the engine, in which the charge air channel includes a supercharger (110) with an output and an input coupled to a source of air, a charge air cooler (129) with an input coupled to the supercharger output and an output coupled to at least one intake port (56), an exhaust channel includes at least one heat-actuated exhaust treatment device (150, 151) with an input coupled to at least one exhaust port (54) and an output, and an exhaust gas recirculation (EGR) loop with a loop input coupled to an output of the at least one heat-actuated exhaust treatment device and a loop output coupled to the input of the supercharger.

22. The ported, uniflow-scavenged, opposed-piston engine of paragraph 21, in which the EGR loop includes an EGR cooler (142) with an input coupled to the output of the at least one heat-actuated exhaust treatment device (150, 151) and an output, and a variable valve (147) settable to a first state or to a second state, wherein the first state decouples the loop output from the supercharger input and the second state couples the loop output to the supercharger input.

23. The ported, uniflow-scavenged, opposed-piston engine of paragraph 22, in which the source of air includes an intake throttle valve (141) for controlling a ratio of recirculated exhaust gas provided by the EGR loop to air provided by the source of air.

24. The ported, uniflow-scavenged, opposed-piston engine of paragraph 23 without a turbocharger.

25. A method of operating a ported, uniflow-scav-

enged, opposed-piston engine (49) with one or more cylinders (50) in which a stream of intake air is supercharged (110) and then cooled in a charge air cooler (129), by providing cooled, supercharged air to at least one intake port (56), recirculating exhaust gas output from a heat-operated exhaust treatment device (150, 151), cooling the recirculated exhaust gas, introducing the cooled, recirculated exhaust gas into the stream of intake air, and controlling a ratio of cooled, recirculated exhaust gas to fresh air provided by the source of fresh air.

28. The method of paragraph 25, in which controlling a ratio includes automatically throttling the amount of fresh air in the stream of fresh air.

27. In a uniflow-scavenged, opposed-piston engine (49) with one or more ported cylinders (50) and without a turbocharger, an exhaust gas recirculation loop characterized by:

a first cooler (142) having an input coupled to an output of a heat-operated exhaust treatment device (151);

a supercharger (110) having an input coupled to a source of intake air and an output;

a second cooler (129) having an input coupled to the output of the supercharger and an output coupled to at least one intake port (56); and,

a valve (147) coupled between the output of the first cooler (142) and the input of the supercharger (110).

28. The exhaust gas recirculation loop of paragraph 27, further including a throttle valve (141) having an input coupled to the source of intake air and an output coupled to the input of the supercharger (110), in which the valve (147) has an input coupled to the output of the first cooler (142) and an output coupled in common with the output of the throttle valve (141) and the input to the supercharger (110).

Claims

1. A ported, uniflow-scavenged, opposed-piston engine (49) including one or more cylinders (50) with piston-controlled exhaust and intake ports (54, 56), a pair of pistons (60, 62) disposed in opposition in the bore of a cylinder, each of the pistons being coupled to a crankshaft (71, 72), and a charge air channel to provide charge air to at least one intake port of the engine, in which an exhaust gas recirculation (EGR) loop has a loop input (124) coupled to an exhaust port (54) of the cylinder and a loop output coupled to the charge air channel, **characterized in that:**

the engine includes a pump in communication with the EGR loop to pump exhaust gas through the EGR loop into the charge air channel.

2. The ported, uniflow-scavenged, opposed-piston engine of claim 1, in which the charge air channel includes at least one charge air cooler (127, 129), wherein the loop output is coupled in series with the at least one charge air cooler.

3. The ported, uniflow-scavenged, opposed-piston engine of claim 2, in which the EGR loop includes a variable valve (138) settable to a first state or to a second state, wherein the first state couples the loop output to a charge air input of the at least one charge air cooler (127) and the second state couples the loop output to a charge air output of the at least one charge air cooler.

4. The ported, uniflow-scavenged, opposed-piston engine of claim 3, in which the pump includes a supercharger (110) and the at least one charge air cooler includes a charge air cooler (127) with an input coupled to the compressor output of a turbo-charger (120) and an output coupled to the input of the supercharger, in which the first state couples the loop output to the input of the charge air cooler (127) and the second state couples the loop output to the input of the supercharger (110).

5. The ported, uniflow-scavenged, opposed-piston engine of claim 2, further including a charge air source (122) with a charge air output coupled to the input of the at least one charge air cooler (129), in which the loop output includes a valve (136) settable to a first state or to a second state and a mixer (135) with an exhaust gas input, a charge air input, and a mixer output coupled to the input of the at least one charge air cooler (129), in which the first state of the valve couples the loop output to the exhaust gas input of the mixer and the second state of the valve uncouples the loop output from the exhaust gas input of the mixer.

6. The ported, uniflow-scavenged, opposed-piston engine of claim 2, further including a supercharger (110) with a supercharger input and a charge air output coupled to the charge air input of the at least one charge air cooler (129) and a valve (139) in parallel with the supercharger, in which the valve is settable to a first state in which the supercharger input is coupled through the valve to the charge air output of the supercharger and to a second state in which the supercharger input is uncoupled through the valve from the charge air output of the supercharger.

7. The ported, uniflow-scavenged, opposed-piston engine of claim 1, further including a turbo-charger

- (120) with a charge air output coupled to the charge air channel and a turbine input coupled to the exhaust port (54), and a back pressure valve (140a) in series between the turbine input and the exhaust port, in which the backpressure valve is settable to a state causing a back pressure acting upon the exhaust port.
8. The ported, uniflow-scavenged, opposed-piston engine of claim 7, in which the turbo-charger includes a variable-geometry turbine (121).
9. The ported, uniflow-scavenged, opposed-piston engine of claim 1, further including a turbo-charger (120) with a charge air output coupled to the charge air channel, a turbine input coupled to the exhaust port (54), a turbine output coupled to an exhaust output (128), and a back pressure valve (140) in series between the turbine output and the exhaust output, in which the backpressure valve is settable to a state causing a back pressure acting upon the exhaust port.
10. The ported, uniflow-scavenged, opposed-piston engine of claim 9, in which the turbo-charger includes a variable-geometry turbine (121).
11. The ported, uniflow-scavenged, opposed-piston engine of claim 1, further including a turbo-charger (120) with a charge air output coupled to the charge air channel and a turbine (121) having an input coupled to the exhaust port (54) and a turbine bypass valve (144) in parallel with the turbine, in which the turbine bypass valve is settable to a first state in which the turbine input is coupled through the valve to a turbine output and to a second state in which the turbine input is uncoupled through the valve from the turbine output.
12. The ported, uniflow-scavenged, opposed-piston engine of claim 11, in which the turbine (121) is a variable-geometry turbine.
13. A method of operating a ported, uniflow-scavenged, opposed-piston engine according to any one of claims 1-12, in which charge air is pressurized and then cooled in at least one charge air cooler (127, 129) and provided to an intake port (56) of each of the one or more cylinders, by providing engine exhaust gas through an exhaust gas recirculation (EGR) loop to an input of the at least one air charge cooler (127, 129).
14. The method of operating a ported, uniflow-scavenged, opposed-piston engine of claim 13, further including retaining exhaust gas in the one or more cylinders (50) during low engine load conditions.

15. The method of operating a ported, uniflow-scavenged, opposed-piston engine of claim 14, in which exhaust gas is retained in the one or more cylinders by decreasing charge air pressure.
16. The method of operating a ported, uniflow-scavenged, opposed-piston engine of claim 14, in which exhaust gas is retained in the one or more cylinders by increasing exhaust gas back pressure.
17. The method of operating a ported, uniflow-scavenged, opposed-piston engine of claim 14, in which exhaust gas is further retained in the one or more cylinders by a combination of increasing exhaust gas back pressure and decreasing charge air pressure.

Patentansprüche

1. Motor (49) mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung, wobei der Motor einen oder mehrere Zylinder mit kolbengesteuertem Auslass und Einlass (54, 56) hat, ein Paar Kolben (60, 62), die entgegengesetzt in der Bohrung eines Zylinders angeordnet sind, wobei jeder der Kolben an eine Kurbelwelle (71, 72) und einen Ladeluft-Kanal gekoppelt ist, um zumindest einen Einlass des Motors mit einer Ladeluft zu versehen, in welchem Motor eine Abgasrückführungs (AGR)-Umführung einen Zirkulations-Einlass (124) hat, der an einem Auslass (54) des Zylinders gekoppelt ist, sowie einen Zirkulations-Auslass, der an den Ladeluft-Kanal gekoppelt ist, **dadurch gekennzeichnet, dass:**
- der Motor eine Pumpe in Kommunikation mit der AGR-Umführung hat, um Abgas durch die AGR-Umführung in den Ladeluft-Kanal zu pumpen.
2. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 1, bei dem der Ladeluft-Kanal mindestens einen Ladeluft-Kühler (127, 129) aufweist, wobei der Zirkulations-Auslass mit dem mindestens einem Ladeluft-Kühler in Serie gekoppelt ist.
3. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 2, bei dem die AGR-Umführung ein variables Ventil (138) aufweist, das in einen ersten Zustand oder in einen zweiten Zustand setzbar ist, wobei der erste Zustand den Zirkulations-Auslass an einen Ladeluft-Einlass des mindestens einen Ladeluft-Kühlers (127) koppelt, und der zweite Zustand den Zirkulations-Auslass an einen Ladeluft-Auslass des mindestens einen Ladeluft-Kühlers koppelt.
4. Motor mit entgegengesetzten Kolben mit Ein- und

Auslass und gleichlaufender Durchströmung gemäß Anspruch 3, bei dem die Pumpe einen Auflader (110) aufweist, und der mindestens eine Ladeluft-Kühler einen Ladeluft-Kühler (127) mit einem Einlass hat, der an den Kompressor-Ausgang eines Turboladers (120) gekoppelt ist, sowie einen Auslass, der an den Eingang des Aufladers gekoppelt ist, wobei der erste Zustand den Zirkulations-Auslass an den Einlass des Ladeluft-Kühlers (127) koppelt, und der zweite Zustand den Zirkulations-Auslass an den Eingang des Aufladers (110) koppelt.

5. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 2, ferner aufweisend eine Ladeluft-Quelle (122) mit einem Ladeluft-Ausgang, der an den Einlass des mindestens einen Ladeluft-Kühlers (129) gekoppelt ist, wobei der Zirkulations-Auslass ein Ventil (136) aufweist, das in einen ersten Zustand oder in einen zweiten Zustand setzbar ist, sowie einen Mischer (135) mit einem Abgas-Einlass, einem Ladeluft-Einlass, und einem Mischer-Auslass, der an den Einlass des mindestens einen Ladeluft-Kühlers (129) gekoppelt ist, wobei der erste Zustand des Ventils den Zirkulations-Auslass an den Abgas-Einlass des Mixers koppelt und der zweite Zustand des Ventils den Zirkulations-Auslass von dem Abgas-Einlass des Mixers entkoppelt.
6. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung nach Anspruch 2, ferner aufweisend einen Auflader (110) mit einem Auflader-Einlass und einem Ladeluft-Auslass, der an den Ladeluft-Einlass des mindestens einen Ladeluft-Kühlers (129) gekoppelt ist, sowie ein Ventil (139) parallel zu dem Auflader, wobei das Ventil in einen ersten Zustand setzbar ist, bei dem der Auflader durch das Ventil an den Ladeluft-Auslass des Aufladers gekoppelt ist, und in einem zweiten Zustand, bei dem der Auflader-Einlass durch das Ventil von dem Ladeluft-Auslass des Aufladers entkoppelt ist.
7. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung nach Anspruch 1, ferner aufweisend einen Turbolader (120) mit einem Ladeluft-Auslass, der an den Ladeluftkanal gekoppelt ist und einem Turbinen-Einlass, der an den Abgas-Auslass (54) gekoppelt ist, sowie ein Absperrventil (140a), das zwischen dem Turbinen-Einlass und dem Abgas-Auslass in Serie vorliegt, wobei das Absperrventil in einen Zustand setzbar ist, der einen Gegendruck verursacht, der auf den Abgas-Auslass wirkt.
8. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 7, wobei der Turbolader eine Turbine (121)

mit variabler Geometrie aufweist.

9. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung nach Anspruch 1, ferner aufweisend einen Turbolader (120) mit einem Ladeluft-Auslass, der an den Ladeluftkanal gekoppelt ist, einen Turbinen-Einlass, der an dem Abgas-Auslass (54) gekoppelt ist, einen Abgas-Auslass (128) gekoppelten Turbinen-Auslass, sowie ein Absperrventil (140), das zwischen dem Turbinen-Auslass und dem Abgas-Auslass in Serie vorliegt, wobei das Absperrventil in einen Zustand setzbar ist, bei dem ein Gegendruck verursacht wird, der auf den Abgas-Auslass wirkt.
10. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung nach Anspruch 9, wobei der Turbo-Auflader eine Turbine (121) mit variabler Geometrie aufweist.
11. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung nach Anspruch 1, ferner aufweisend einen Turbolader (120) mit einem Ladeluft-Auslass, der an den Ladeluft-Kanal gekoppelt ist, und einer Turbine (121) mit einem Einlass, der an den Abgas-Auslass (54) gekoppelt ist, sowie einem Turbinen-Beipass-Ventil (144), das mit der Turbine parallel ist, wobei das Turbinen-Beipass-Ventil in einen ersten Zustand setzbar ist, bei dem der Turbinen-Einlass durch das Ventil an einen Turbinen-Auslass gekoppelt ist, und in einen zweiten Zustand, bei dem der Turbinen-Einlass durch das Ventil von dem Turbinen-Auslass entkoppelt ist.
12. Motor mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung nach Anspruch 11, wobei die Turbine (121) eine Turbine mit variabler Geometrie ist.
13. Verfahren zum Betreiben eines Motors mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß einem der Ansprüche 1 - 12, wobei Ladeluft unter Druck gesetzt wird und dann in mindestens einem Ladeluft-Kühler (127, 129) gekühlt wird und an einen Einlass (56) eines jeden des einen oder der mehreren Zylindern geführt wird, indem das Motoren-Abgas durch eine Abgasrückführungs (AGR)-Umführung an einem Einlass des mindestens einen Ladeluft-Kühlers (127, 129) geführt wird.
14. Verfahren zum Betreiben eines Motors mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 13, ferner umfassend ein Rückhalten von Abgas in dem einen oder der mehreren Zylindern (50) während Kleinlast-Zuständen des Motors.

15. Verfahren zum Betreiben eines Motors mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 14, wobei Abgas in dem einen oder der mehreren Zylindern durch Verminderung des Ladeluft-Druckes zurückgehalten wird. 5
16. Verfahren zum Betreiben eines Motors mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 14, wobei Abgas in dem einen oder der mehreren Zylindern durch Erhöhen des Abgas-Rückdruckes zurückgehalten wird. 10
17. Verfahren zum Betreiben eines Motors mit entgegengesetzten Kolben mit Ein- und Auslass und gleichlaufender Durchströmung gemäß Anspruch 14, wobei Abgas in dem einen oder der mehreren Zylindern ferner rückgehalten wird durch eine Kombination eines Erhöehens des Abgas-Rückdruckes und einem Vermindern des Ladeluft-Druckes. 15 20

Revendications

1. Moteur à pistons opposés à orifice et à balayage longitudinal (49) comprenant un ou plusieurs cylindres (50) avec des orifices d'échappement et d'admission commandés par piston (54, 56), une paire de pistons (60, 62) disposés en opposition dans l'alésage d'un cylindre, chacun des pistons étant couplé à un vilebrequin (71, 72) et un canal d'air de suralimentation pour fournir l'air de suralimentation à au moins un orifice d'admission du moteur, dans lequel une boucle de recirculation de gaz d'échappement (EGR) a une entrée de boucle (124) couplée à un orifice d'échappement (54) du cylindre et une sortie de boucle couplée au canal d'air de suralimentation, **caractérisé en ce que :** 25 30 35
- le moteur comprend une pompe en communication avec la boucle EGR afin de pomper le gaz d'échappement par la boucle EGR dans le canal d'air de suralimentation. 40
2. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 1, dans lequel le canal d'air de suralimentation comprend au moins un refroidisseur d'air de suralimentation (127, 129), dans lequel la sortie de boucle est couplée en série avec le au moins un refroidisseur d'air de suralimentation. 45 50
3. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 2, dans lequel la boucle EGR comprend une soupape variable (138) réglable sur un premier état ou un second état, dans lequel le premier état couple la sortie de boucle à 55

une entrée d'air de suralimentation du au moins un refroidisseur d'air de suralimentation (127) et le second état couple la sortie de boucle à une sortie d'air de suralimentation du au moins un refroidisseur d'air de suralimentation.

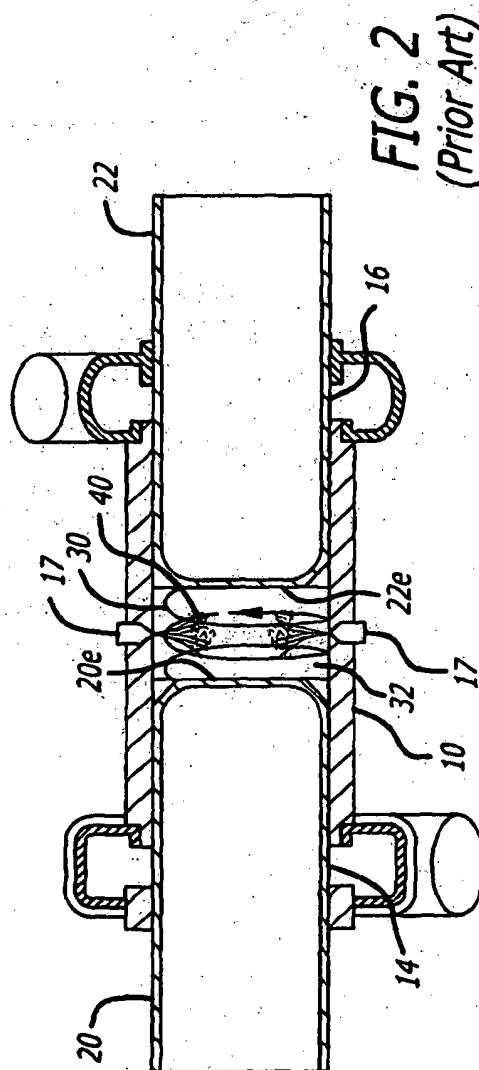
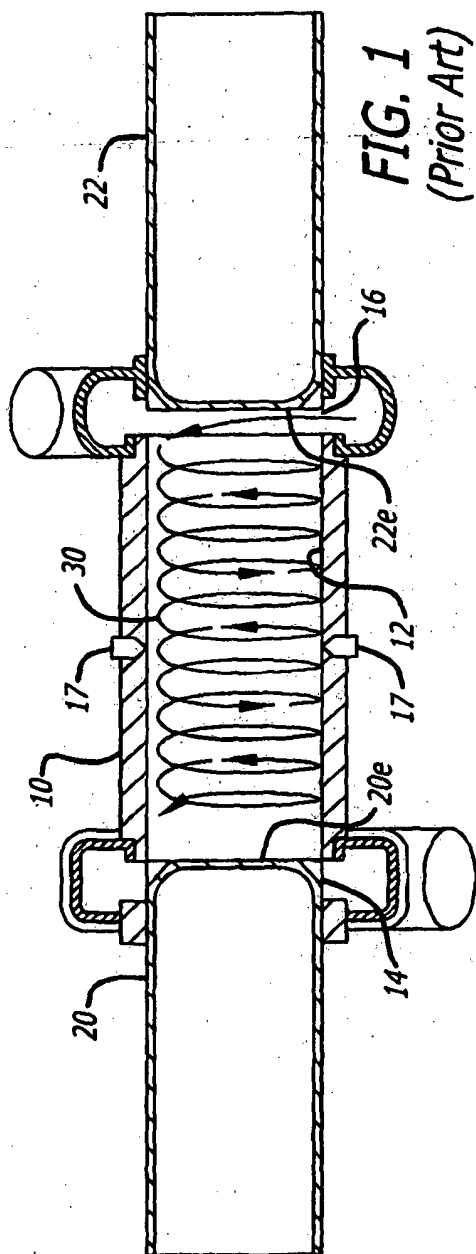
4. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 3, dans lequel la pompe comprend un compresseur à suralimentation (110) et le au moins un refroidisseur d'air de suralimentation comprend un refroidisseur d'air de suralimentation (127) avec une entrée couplée à la sortie de compresseur d'un turbocompresseur (120) et une sortie couplée à l'entrée du compresseur de suralimentation, dans lequel le premier état couple la sortie de boucle à l'entrée du refroidisseur d'air de suralimentation (127) et le second état couple la sortie de boucle à l'entrée du compresseur de suralimentation (110).
5. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 2, comprenant en outre une source d'air de suralimentation (122) avec une sortie d'air de suralimentation couplée à l'entrée du au moins un refroidisseur d'air de suralimentation (129), dans lequel la sortie de boucle comprend une soupape (136) réglable sur un premier état ou un second état et un mélangeur (135) avec une entrée de gaz d'échappement, une entrée d'air de suralimentation et une sortie de mélangeur couplée à l'entrée du au moins un refroidisseur d'air de suralimentation (129), dans lequel le premier état de la soupape couple la sortie de boucle à l'entrée de gaz d'échappement du mélangeur et le second état de la soupape découple la sortie de boucle de l'entrée de gaz d'échappement du mélangeur. 25 30 35 40
6. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 2, comprenant en outre un compresseur de suralimentation (110) avec une entrée de compresseur de suralimentation et une sortie d'air de suralimentation couplée à l'entrée d'air de suralimentation du au moins un refroidisseur d'air de suralimentation (129) et une soupape (139) en parallèle avec le compresseur de suralimentation, dans lequel la soupape est réglable sur un premier état dans lequel l'entrée de compresseur de suralimentation est couplée par le biais de la soupape à la sortie d'air de suralimentation du compresseur de suralimentation et un second état dans lequel l'entrée de compresseur de suralimentation est découplée par le biais de la soupape, de la sortie d'air de suralimentation du compresseur de suralimentation. 45 50
7. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 1, comprenant en outre un turbocompresseur (120) avec une sortie 55

d'air de suralimentation couplée au canal d'air de suralimentation et une entrée de turbine couplée à l'orifice d'échappement (54) et une soupape de contre-pression (140a) en série entre l'entrée de turbine et l'orifice d'échappement, dans lequel la soupape de contre-pression est réglable dans un état provoquant une contre-pression agissant sur l'orifice d'échappement.

8. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 7, dans lequel le turbocompresseur comprend une turbine à géométrie variable (121). 10
9. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 1, comprenant en outre un turbocompresseur (120) avec une sortie d'air de suralimentation couplée au canal d'air de suralimentation, une entrée de turbine couplée à l'orifice d'échappement (54), une sortie de turbine couplée à une sortie d'échappement (128), et une soupape de contre-pression (140) en série entre la sortie de turbine et la sortie d'échappement, dans lequel la soupape de contre-pression est réglable sur un état provoquant une contre-pression qui agit sur l'orifice d'échappement. 15
10. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 9, dans lequel le turbocompresseur comprend une turbine à géométrie variable (121). 20
11. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 1, comprenant en outre un turbocompresseur (120) avec une sortie d'air de suralimentation couplée au canal d'air de suralimentation et une turbine (121) ayant une entrée couplée à l'orifice d'échappement (54) et une soupape de dérivation de turbine (144) en parallèle avec la turbine, dans lequel la soupape de dérivation de turbine est réglable sur un premier état dans lequel l'entrée de turbine est couplée par le biais de la soupape à une sortie de turbine et un second état dans lequel l'entrée de turbine est découplée par le biais de la soupape de la sortie de turbine. 25
12. Moteur à pistons opposés à orifice et à balayage longitudinal selon la revendication 11, dans lequel la turbine (121) est une turbine à géométrie variable. 30
13. Procédé pour actionner un moteur à pistons opposés, à orifice et à balayage longitudinal selon l'une quelconque des revendications 1 à 12, dans lequel l'air de suralimentation est mis sous pression et ensuite refroidi dans au moins un refroidisseur d'air de suralimentation (127, 129) et amené à un orifice d'admission (56) de chacun des un ou plusieurs cylindres, en amenant le gaz d'échappement de mo- 35

teur par le biais d'une boucle de recirculation de gaz d'échappement (EGR) à une entrée du au moins un refroidisseur d'air de suralimentation (127, 129).

14. Procédé pour actionner un moteur à pistons opposés, à orifice et à balayage longitudinal selon la revendication 13, comprenant en outre l'étape consistant à retenir le gaz d'échappement dans les un ou plusieurs cylindres (50) pendant des conditions de charge faible de moteur. 40
15. Procédé pour actionner un moteur à pistons opposés, à orifice et à balayage longitudinal selon la revendication 14, dans lequel le gaz d'échappement est retenu dans les un ou plusieurs cylindres en diminuant la pression d'air de suralimentation. 45
16. Procédé pour actionner un moteur à pistons opposés, à orifice et à balayage longitudinal selon la revendication 14, dans lequel le gaz d'échappement est retenu dans les un ou plusieurs cylindres en augmentant la contre-pression de gaz d'échappement. 50
17. Procédé pour actionner un moteur à pistons opposés, à orifice et à balayage longitudinal selon la revendication 14, dans lequel le gaz d'échappement est en outre retenu dans les un ou plusieurs cylindres par une combinaison consistant à augmenter la contre-pression de gaz d'échappement et à diminuer la pression d'air de suralimentation. 55



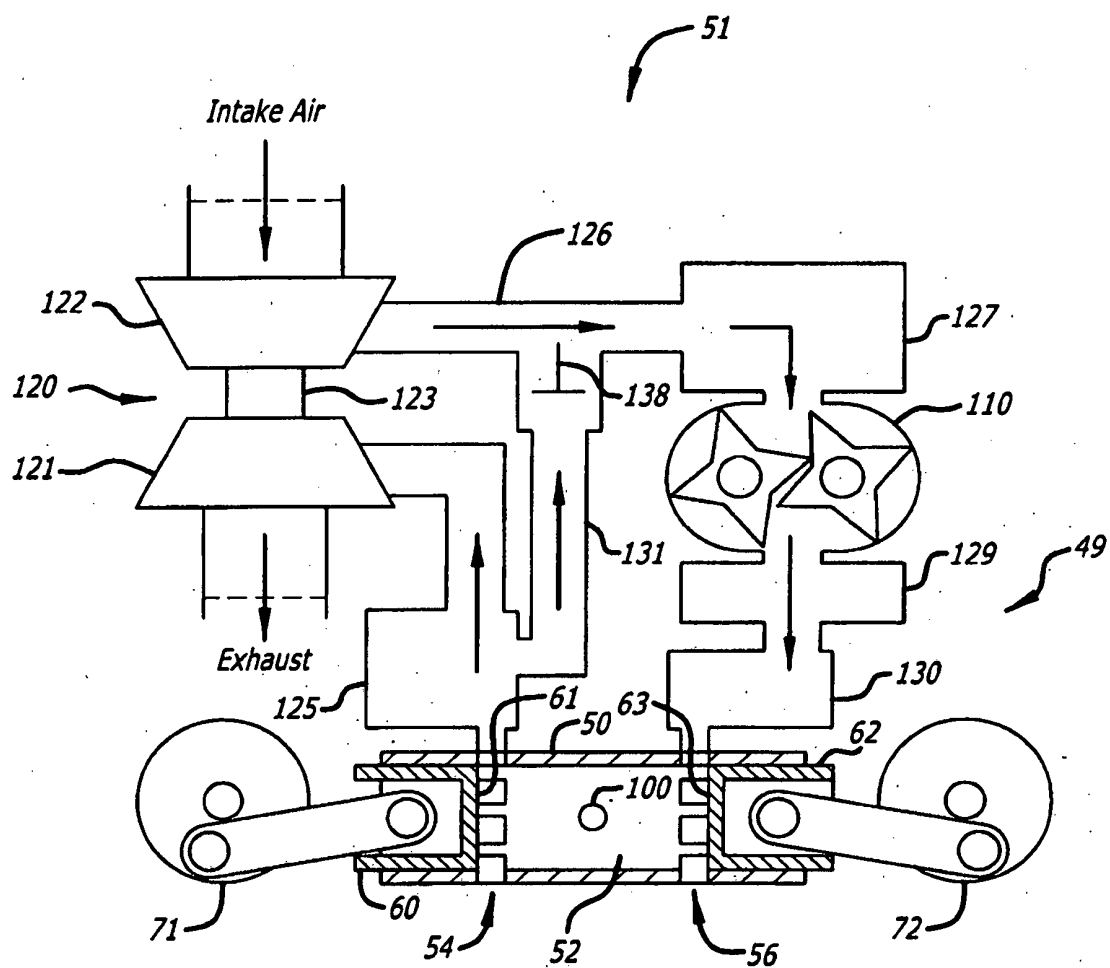


FIG. 3

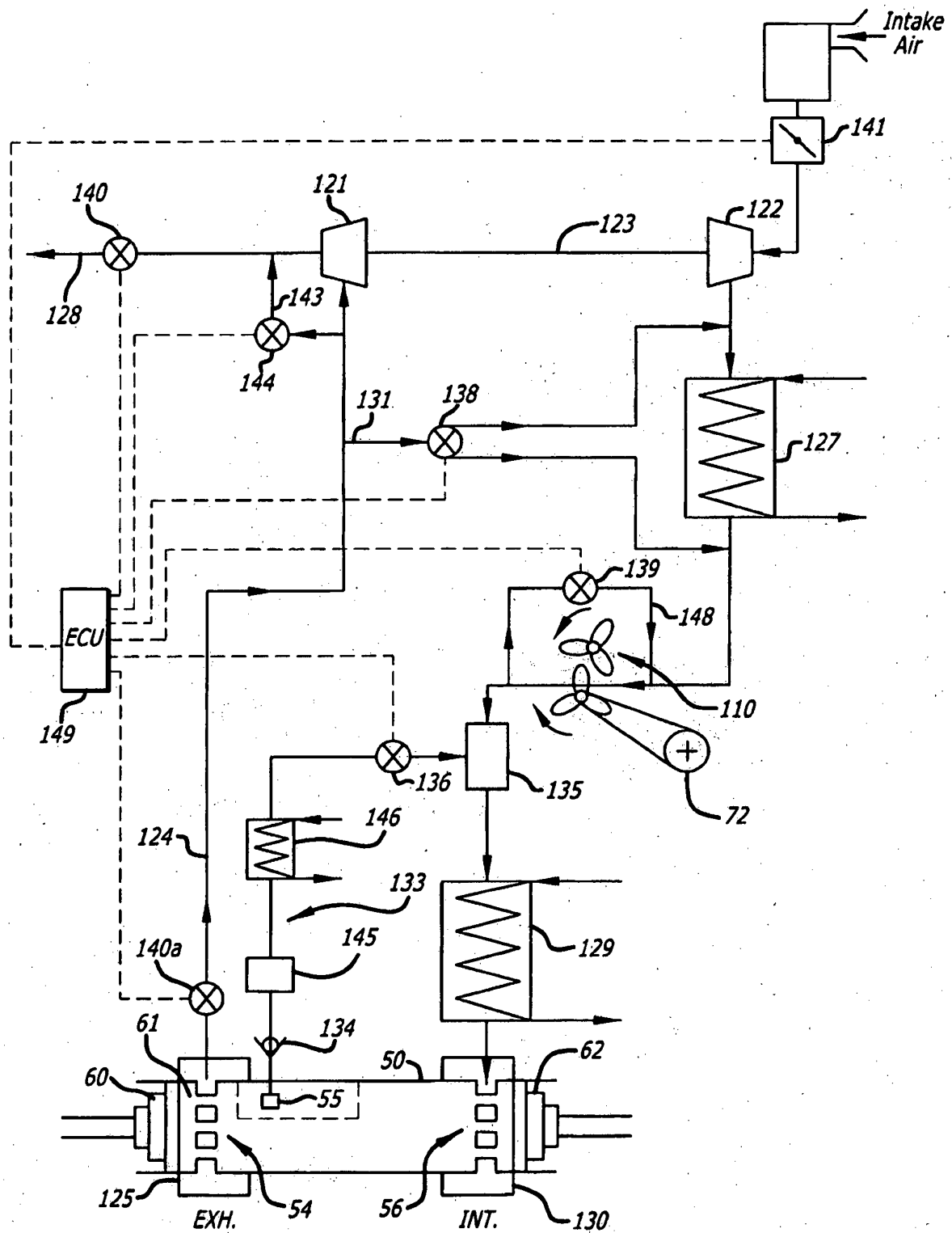


FIG. 4

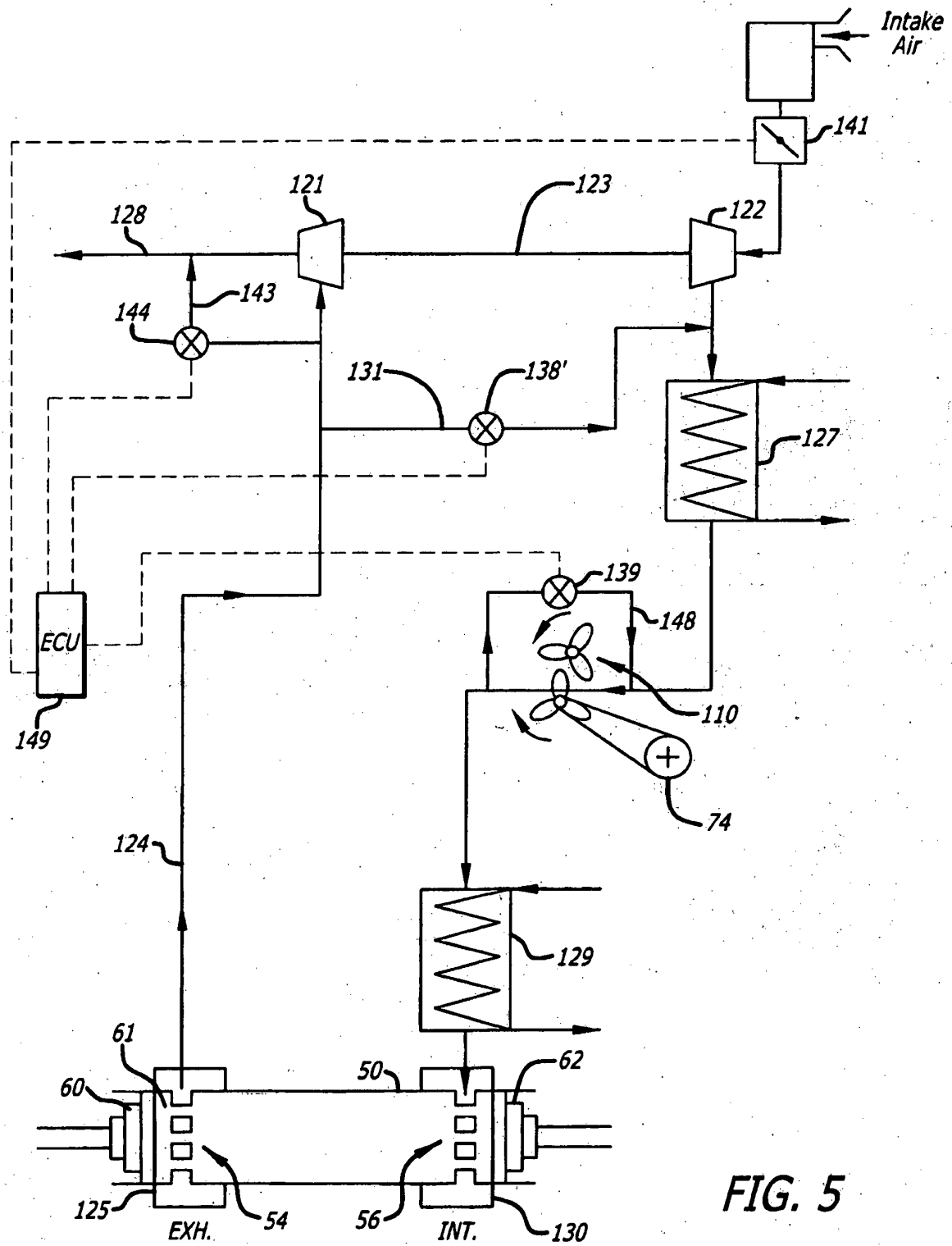


FIG. 5

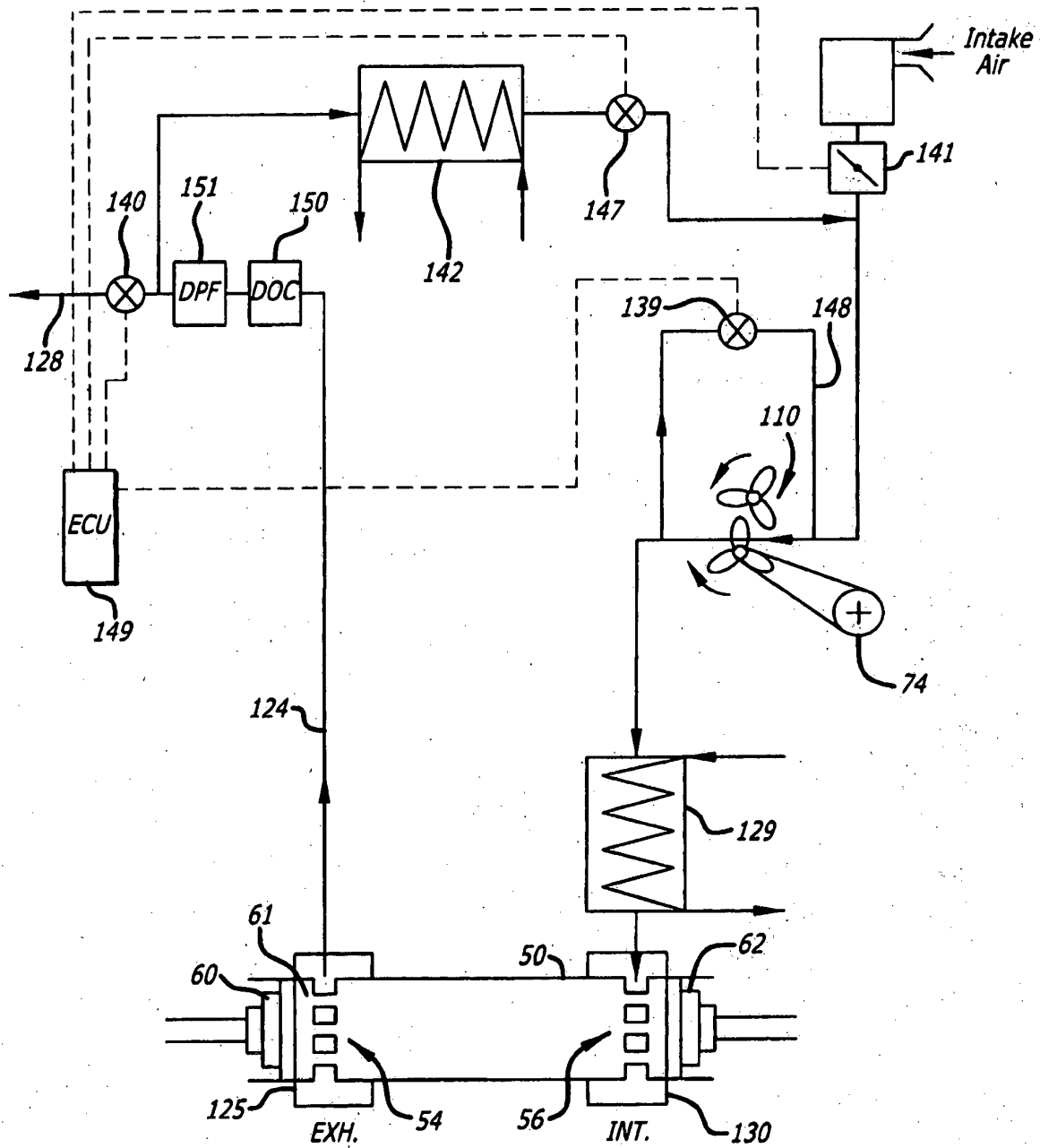


FIG. 6

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- US 6925971 B [0006]
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