(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
18 May 2006 (18.05.2006)

PCT

(10) International Publication Number
WO 2006/051299 A1

(51) International Patent Classification:
F02B 33/44 (2006.01) *F02B 39/04* (2006.01)
F02B 37/02 (2006.01) *F02B 29/04* (2006.01)
F02B 37/04 (2006.01) *F02B 37/18* (2006.01)

(21) International Application Number:
PCT/GB2005/004334

(22) International Filing Date:
10 November 2005 (10.11.2005)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
0424845.6 10 November 2004 (10.11.2004) GB

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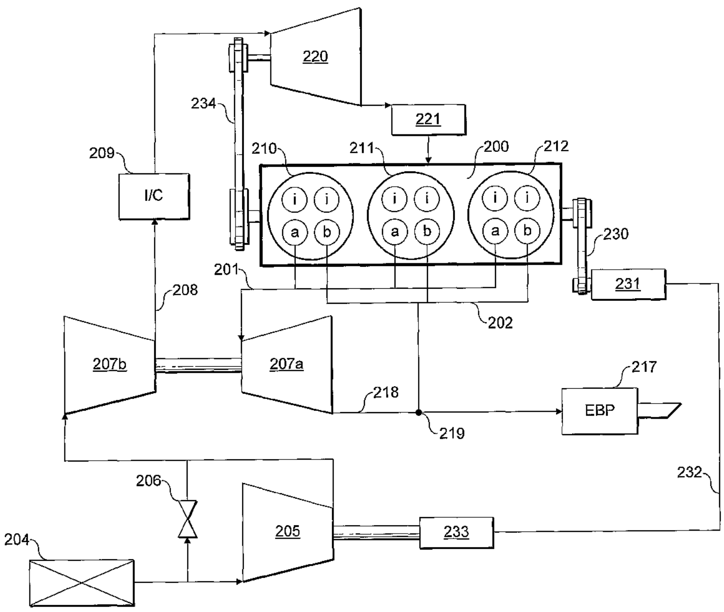
(81) Designated States *(unless otherwise indicated, for every kind of national protection available)*: AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BW, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KM, KN, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, LY, MA, MD, MG, MK, MN, MW, MX, MZ, NA, NG, NI, NO, NZ, OM, PG, PH, PL, PT, RO, RU, SC, SD, SE, SG, SK, SL, SM, SY, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, YU, ZA, ZM, ZW.

(84) Designated States *(unless otherwise indicated, for every kind of regional protection available)*: ARIPO (BW, GH, GM, KE, LS, MW, MZ, NA, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU, IE, IS, IT, LT, LU, LV, MC, NL, PL, PT, RO, SE, SI, SK, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Declarations under Rule 4.17:
— *as to applicant's entitlement to apply for and be granted a patent (Rule 4.17(ii))*
— *of inventorship (Rule 4.17(iv))*

[Continued on next page]

(54) Title: A PRESSURE-CHARGED GASOLINE INTERNAL COMBUSTION ENGINE



(57) Abstract: A pressure-charged gasoline internal combustion engine (200) comprising a combustion chamber (210, 211, 212) ; inlet valve means (i) ; exhaust valve means (a, b) ; a compressor (205) ; an intercooler (209) which receives compressed charge air from the compressor (205) ; an expander (220) which receives cooled compressed charge air from the intercooler (209) ; and an injector which injects gasoline directly into the combustion chamber (210, 211, 212) for mixing with the charge air conditioned previously by the compressor (205), the intercooler (209) and the expander (220). The compressor (205) is a supercharger.

WO 2006/051299 A1

Published:
— *with international search report*

A PRESSURE-CHARGED GASOLINE INTERNAL COMBUSTION ENGINE

The present invention relates to a pressure-charged gasoline internal combustion engine.

It is a known problem to provide improved fuel economy from gasoline internal combustion engines. One route to improve fuel economy is to reduce the swept volume of an engine and run the engine at a higher BMEP (Brake Mean Effective Pressure) for any given output. To achieve this and provide good fuel economy, particularly in part-load operating conditions, the expansion ratio of the engine should be kept as high as possible. In normal engines this also means that the compression ratio is similarly kept high. However, a high compression ratio is undesirable in a pressure-charged gasoline internal combustion engine because at high loads in such an engine the knock limit is a severe limitation. Indeed it is generally desirable to reduce the compression ratio of the engine at high loads and speeds of the engine, particularly when pressure charging is used; pressure charging being important to maintain high load engine performance. It has previously been proposed (e.g. by Saab) to provide an engine with a variable compression ratio (VCR) which can be varied with engine speed and load to meet the different requirements.

The present invention provides a pressure-charged gasoline internal combustion engine comprising:

a combustion chamber;

inlet valve means controlling admission of charge air into the combustion chamber;

exhaust valve means for controlling exhaust of combusted gases from the combustion chamber;

a compressor for compressing charge air prior to delivery of the charge air to the combustion chamber;

an intercooler which receives compressed charge air from the compressor and cools the compressed charge air;

an expander which receives cooled compressed charge air from the intercooler and expands the cooled compressed charge air to reduce both the temperature and pressure of the charge air prior to delivery of the charge air to the combustion chamber; and

an injector which injects gasoline directly into the combustion chamber for mixing with the charge air conditioned previously by the compressor, the intercooler and the expander; wherein:

the compressor is a supercharger which is driven by power output by the engine rather than by exhaust gases flowing from the engine.

The concept of using an expander to expand and cool gases previously compressed in a compressor (e.g. turbo-charger) shall be referred to as "turboexpansion". The use of turboexpansion and gasoline direct injection (GDI) together is synergistic because both act to reduce the start of compression temperature in the engine. In contrast the VCR approach of the prior art (e.g. of SAAB) has aimed to reduce the end of compression temperature in the engine by reducing the compression ratio. Turboexpansion and GDI are charge-air conditioning concepts which are complementary and synergistic since they occur in successive parts of the induction process. References in this specification to a "supercharger" should be taken to be references to an engine-driven compressor, i.e. the power output of the engine is used to drive the compressor rather than exhaust gas flow.

Preferred embodiments of the present invention will now be described with reference to the accompanying drawings, in which:

Figure 1 is a schematic illustration of a first embodiment of pressure-charged gasoline internal combustion engine modified to accord with the present invention;

Figure 2 is a schematic illustration of a combustion chamber of a first variant of the engine of Figure 1;

Figure 3 is a temperature/entropy diagram illustrating operation of the engine of Figures 1 and 2;

Figure 4 is a schematic illustration of a combustion chamber of a second variant of the engine of Figure 1; and

Figure 5 is a schematic illustration of a second embodiment of a pressure-charged gasoline internal combustion engine in accordance with the present invention.

Turning to Figure 1, there can be seen in the figure a pressure-charged internal combustion engine 10 having an induction system comprising a compressor in the form of a supercharger 11, having a compressor section 11A. The supercharger 11 is driven by a belt 50 which takes drive from an output shaft of the engine 10. The supercharger 10 is powered by power taken from power output by the engine and not by the waste energy of the exhaust gas flow (i.e. the supercharger is not a turbocharger). The compressor section 11A draws in charge air from atmosphere via an air filter 12. The air compressed in the compressor section 11A is then passed through an intercooler 13 to an expander 14.

The expander 14 expands the compressed air, delivering work back to a crankshaft of the engine 10 via a belt drive 15. The expanded air is delivered to a plenum 16 from where it is delivered to the engine 10 where it undergoes combustion.

Figure 2 shows a combustion chamber 1 of the engine 10, defined in a cylinder 2 by a piston 3 reciprocating in the cylinder. An inlet passage 4 leads charge air from the plenum 16 to the combustion chamber 1 with an inlet valve 5 operated by a cam 6 controlling flow of charge air into the chamber 1. An exhaust passage 7 allows flow of combusted gases out of the combustion chamber 1 to atmosphere, the flow being controlled by an exhaust valve 8 operated by a cam 9. A spark plug 17 can ignite fuel in the combustion chamber 1 and fuel is delivered directly to the chamber 1 by a gasoline direct injection (GDI) injector 18 (e.g. an air assist injector). The GDI injector 18 is vertically oriented to avoid bore wetting as much as possible and to give the largest possible spray angle. This maximises the amount of fuel which vaporises in the charge air and not on contact with internal surfaces of the engine 10. The GDI injector 18 produces an homogeneous fuel/air mixture rather than a stratified mixture to minimise NOx production and avoid the need for a lean NOx trap in the exhaust system.

The induction system operates (see Figure 3) by pressurising the charge air from atmospheric pressure (PAtm) to a pressure (Pupper) greater than the pressure (Pplenum) required in the plenum; the applicant refers to such compressing of the air in the supercharger 11 as "over-compressing" since the pressure of the compressed air is greater than that needed in the plenum 16. The temperature of the charge air increases from atmospheric temperature T_1, to a higher temperature T_2 after compression in the compressor 11A. The compressed air is then cooled to a temperature T_3 by the intercooler 13 and then further cooled by expansion in the expander 14 to a temperature T_4; ideally at full-load of the engine the temperature T_4 of charge air in the plenum is less than atmospheric temperature.

The expanding of the compressed air in the expander 14 removes some work from the air (which is delivered to the crankshaft via the belt drive 15) and simultaneously reduces both pressure and temperature of the charge air. This makes it possible to achieve in the plenum a chosen air density at a temperature below that which can be achieved by use of an intercooler alone; theoretically twice atmospheric density is possible. The expander 14 could of any type and could deliver output power mechanically, hydraulically or electrically.

The GDI injector 41 will inject fuel directly into the combustion chamber whilst the intake valve 24 is open and the vaporisation of the gasoline fuel in the charge air will reduce further the charge air temperature and increase inlet charge density and therefore increase volumetric efficiency. The use of a centrally located GDI injector 41 gives fewer constraints (when compared with a port-injected engine) in piston crown geometry and intake port profile since the GDI injector 41 can give a required fuel/air mixture characteristic without the need for flow control by features of the piston and/or intake port; the design of the piston and intake port can therefore be optimised for volumetric efficiency and full load performance.

Advanced injector designs currently available produce highly repeatable and controlled sprays and can be used to extend the range of lean burn operating conditions and produce improvements in hydrocarbon emissions. The GDI system will preferably be a spray-guided GDI system which generates less NOx than a port or wall guided system and thus avoids the need for a NOx trap in the exhaust system (the use of which reduces fuel efficiency by requiring the engine to run rich periodically to regenerate the NOx trap).

The reduction of the temperature of the inlet charge by use of an expander and GDI allows the engine to run at a compression ratio not possible normally without knock occurring. This is achieved in an engine with a compression ratio fixed for all speeds and loads.

Improved knock performance also provides an efficiency benefit by allowing the phasing of the combustion event to be advanced at the detonation borderline. This leads to improvements in thermal efficiency and enables a given torque level to be attained at lower boost pressures.

Advancing the combustion event also leads to lower exhaust temperatures and therefore reduces the degree of fuel enrichment necessary to limit the temperature of the exhaust gas entering a catalytic converter in the exhaust system. This can result in reduced wide open throttle fuel consumption. The ability to delay the addition of the fuel until the exhaust valve has closed eliminates, or dramatically reduces, the carry-over of fuel into the exhaust system during the valve overlap period. Such carried over fuel would give rise to exothermic reactions in the catalyst and require extra enrichment for catalyst substrate protection, which is fuel inefficient.

Rather than using conventional mechanical valve train, a fully variable valve train could be used. Turning to Figure 4, in the figure it can be seen that charge air from the plenum 16 is delivered to a combustion chamber 20 (formed by piston 21 reciprocating in a cylinder 22) via an inlet passage 23. Delivery of charge air into the chamber 20 is controlled by an inlet valve 24 which is operated by a hydraulic actuator 25 (comprising a piston 28 movable in a cylinder 29), the piston 28 being mounted on a valve stem 27 of the valve 24). A spark plug 40 is operative in the combustion chamber 20. A GDI injector 41 is axially mounted in the head of the cylinder 22 to deliver gasoline directly into the combustion chamber 20 (e.g. with the use of compressed air assistance). Combusted gases leave the combustion chamber 20 via an exhaust passage 26 under the control of an exhaust valve 34 which is operated by a hydraulic actuator 35 (comprising a piston 36 movable in a cylinder 37, the piston 36 being mounted on a valve stem 38 of the valve 34). The combusted gases flow through the exhaust passage 26 to atmosphere.

The actuators 25 and 35 are controlled respectively by electro-hydraulic servo-valves 23 and 27 which control flow of hydraulic fluid to the actuators 25,35 from a pump 30 or to a sump 31. The servo-valves 23 and 27 are controlled by an electronic controller which controls the movement of the inlet valve 24 and the exhaust valve 34 in a closed loop feedback control arrangement which uses position feedback signals provided by two position sensors 32 and 33. The electronic controller 24 also controls operation of the spark plug 40 and the direct injector 41.

The use of valves controlled by actuators as in Figure 4 can allow throttleless operation of the engine, improving efficiency by dispensing with throttling losses in part-load conditions. Also the ability to vary all three of valve lift height, valve opening time and valve closing time allows greater efficiency. The use of a fully variable valve train allows optimisation of the expansion ratio in the engine in all conditions. The use of a fully variable valves as shown in Figure 4 could permit spark-less Controlled Auto-Ignition (sometimes called Homogeneous Charge Compression Ignition) in part-load conditions. The recycling of exhaust gases in CAI operation would allow lean burn ignition without exceeding NOx limits for a 3-way oxidising catalyst.

Increased freedom to optimise valve events is permitted by the use of a GDI injector, which can be controlled to inject fuel only after the exhaust valve has closed and thus e.g. reduce carry-over of fuel into the exhaust system during valve overlap. This can allow significantly improved low-speed torque by suitable valve timing variation.

With fixed valve timing, the exhaust valve opening point is usually dictated by the trade-off between expansion work and pumping work at full-load. Again part-load operation is compromised by the full-load performance requirements. With a variable valve train, exhaust valve opening can be retarded at part-load relative to its full-load timing, increasing the effective expansion ratio.

The fully variable valve train described above also allows the engine to switch between 2 and 4 stroke operation, cylinder deactivation, conversion of one or more cylinders into compressors of air for storage in a tank and later expansion in the cylinders (such compression taking place e.g. during braking of a vehicle), or a stop-at-idle fuel economy strategy.

The controllable valves of figure 4 also allow control of the charging system as will now be described with reference to Figures 5 and 6.

In Figure 5 there can be seen a multi-cylinder engine having three cylinders 210, 211, 212. Each cylinder has a pair of inlet valves "i" and two exhaust valves "a" and "b". The exhaust valves "a" and "b" at least are each operated by a hydraulic actuator connected to the valve (as shown in Figure 4). Each exhaust valve "a" would be opened and closed independently of the exhaust valve "b" in the same cylinder.

Combusted gases flowing from the cylinders 210, 211, 212 flow through the exhaust valves "a" to a first exhaust duct 201. This exhaust duct 14 relays the combusted gases to the turbine stage 207a of a turbocharger 207.

The exhaust valves "b" are all connected to a second exhaust duct 202 through which combusted gases can flow from the cylinders 210, 211, 212 through the exhaust valves "b" directly to a catalytic converter 217, bypassing the turbocharger 207.

The combusted gases expanded in the turbine 207a are output from the turbocharger 207 via an exhaust duct 218,which is joined to the exhaust duct 202 at a joint 219. At the joint 219 the combusted gases flowing from the turbo-charger 207 combine with the combusted gases flowing through the exhaust duct 202 and then the combined flow passes through the catalytic converter 217 to atmosphere.

The electronic controller can use its control of the actuators to control the opening and closing of the exhaust valves "a" to control what proportion of the total combusted gases flowing from each cylinder flow to the exhaust duct 201 and what proportion of the combusted gases flow through the exhaust duct 202. In this way the controller can control operation of the turbocharger 207. When greater boost is required then a greater proportion of the total combusted gases expelled from the cylinders 10, 11, 12 and 13 is fed through the turbo-charger 207 and vice versa.

Charge air is drawn into the engine via an air filter 204 and initially pressurised in an electrically operated supercharger 205. The engine 200 drives via belt 230 an electrical generator 231 which is connected via electrical cable 232 to an electric motor 233 which drives the supercharger 205. The charge air pressurised in the supercharger 205 is then delivered to the turbocharger 207 to be further compressed. The air leaving the turbocharger 207 is then cooled by intercooler 209 and then expanded in expander 220 before delivery to a plenum 221 from where it is delivered into the engine via the inlet valves i. Work output by expander 220 is delivered back to an engine output shaft or engine crankshaft via a belt 234. The work output by expander 220 thus forms part of the total work output by the engine 200 e.g via a crankshaft.

A bypass valve 206 allows the supercharger 205 to be bypassed. It could be spring-loaded so that it opens automatically on the creation of a sufficient pressure differential across it. More likely it will be electrically controllable to allow bypass of the supercharger 205 under control of the engine management system; the supercharger will be used at low speeds and loads when boost provided by the supercharger alone is insufficient and not used at all at high speeds and loads.

As described above the supercharger is used on start-up and/or at low speeds and with the supercharger bypassed otherwise so that only the turbocharger is used. Alternatively the turbocharger could be set up for part-load operation and could be used with an axial flow supercharger switched on for full load operation.

The electrically-driven supercharger 203 could be replaced by a belt-driven supercharger, perhaps driven via a clutch which enables the supercharger to be driven selectively.

Above the use of fully variable valve operating mechanisms with hydraulic actuators to permit CAI operation has been described. However, it would be possible to use variable mechanical valve operating systems (e.g. cam profile switching systems, cam phasing systems, etc.) to facilitate limited CAI operation and to reduce throttling losses at part-load. CAI operation can allow diluted operation at part-load conditions without exceeding the limits for NOx of a 3-way catalyst acting as an oxicat and still avoiding the need for a NOx trap. A variable mechanical valve operating system could also control the relative amounts of exhaust gas passing along conduits 201 and 202 in the Figure 5 engine.

The applicant expects a specific power output of 135-150 bhp/litre from the engines of the invention with an improvement in fuel economy over prior art engines of equivalent combustion chamber volumes.

A supercharger can be used as described provided that the energy lost by inefficiencies in the supercharger and expander combination is more than offset by the increased efficiency of the engine arising from the use of an increased compression ratio. Use of twin superchargers is also possible, e.g. with the turbocharger 201 of Figure 5 replaced by a supercharger driven by power output by the engine.

CLAIMS:

1. A pressure-charged gasoline internal combustion engine comprising:

a combustion chamber;

inlet valve means controlling admission of charge air into the combustion chamber;

exhaust valve means controlling exhaust of combusted gases from the combustion chamber;

a compressor for compressing charge air prior to delivery to the combustion chamber;

an intercooler which receives compressed charge air from the compressor and cools the compressed charge air;

an expander which receives cooled compressed charge air from the intercooler and expands the cooled compressed charge air to reduce both the temperature and pressure of the charge air prior to delivery of the charge air to the combustion chamber; and

an injector which injects gasoline directly into the combustion chamber for mixing with the charged air conditioned previously by the compressor, the intercooler and the expander wherein:

the compressor is a supercharger which is driven by power output by the engine rather than exhaust gases flowing from the engine.

2. A pressure-charged gasoline internal combustion engine as claimed in claim 1 wherein the expander provides work output which is combined with work output of the engine.

3. A pressure-charged gasoline internal combustion engine as claimed in claim 2 wherein the expander is connected to

an output shaft of the engine to deliver the work output thereby to the output shaft.

4. A pressure-charged gasoline internal combustion engine as claimed in any one of claims 1 to 3 comprising a turbocharger in addition to the supercharger, wherein the exhaust valve means comprises a first exhaust valve controlling flow of combusted gases to the turbocharger and a second exhaust valve controlling flow of combusted gases to a bypass passage which bypasses the turbocharger, and the exhaust valve means is operated by a valve operating mechanism which controls operation of the first and second exhaust valves to control in degree boost provided by the turbocharger.

5. A pressure-charged gasoline internal combustion engine as claimed in any one of the preceding claims wherein the combustion chamber is defined by walls of a cylinder in the engine and a piston reciprocating in the cylinder and the injector is centrally located in an upper surface of the cylinder to deliver fuel axially into the cylinder.

6. A pressure-charged gasoline internal combustion engine as claimed in any one of the preceding claims wherein the injector delivers gasoline to the combustion chamber in a manner which creates a homogeneous mixture of fuel and air therein.

7. A pressure-charged gasoline internal combustion engine as claimed in any one of the preceding claims wherein the injector is a part of a spray-guided gasoline direct injection system.

8. A pressure-charged gasoline internal combustion engine as claimed in any one of the preceding claims comprising a valve operating mechanism for operating the inlet valve means which can vary operation of the inlet valve means with variation in engine speed and load, wherein in selected part load operating conditions of the engine the inlet valve means is used to control in amount the charge air delivered to the combustion chamber and the engine operates without throttling of the charge air prior to delivery of the charge air via the inlet valve means to the combustion chamber.

9. A pressure-charged gasoline internal combustion engine as claimed in any one of the preceding claims comprising a valve operating mechanism for operating the exhaust valve means which can vary operation of the exhaust valve means with variation in engine speed and load, wherein in selected part load operating conditions the exhaust valve means is closed early in each exhaust intake to trap combusted gases in the combustion chamber for mixing with the charge air next delivered to the combustion chamber and the fuel next injected into the combustion chamber to create a homogeneous mixture of air, combusted gases and fuel which is ignited by compression ignition.

10. A pressure-charged gasoline internal combustion engine as claimed in any one of the preceding claims wherein the expander outputs work via a transmission system connected also to the engine.

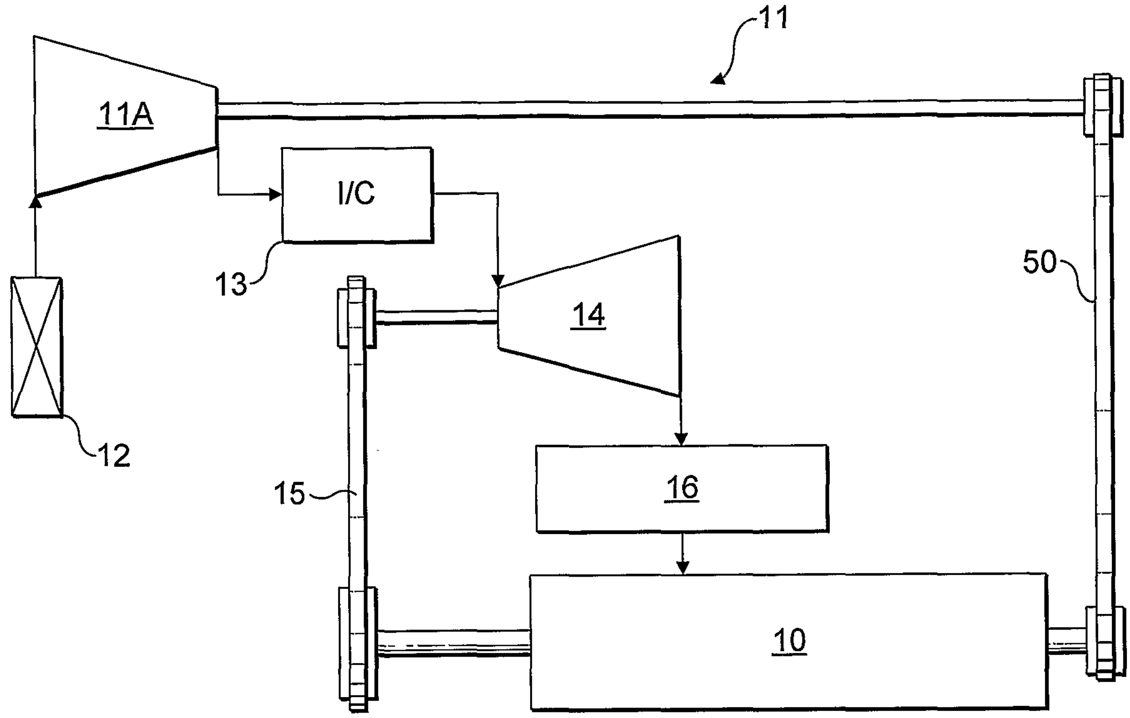

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FIG. 1

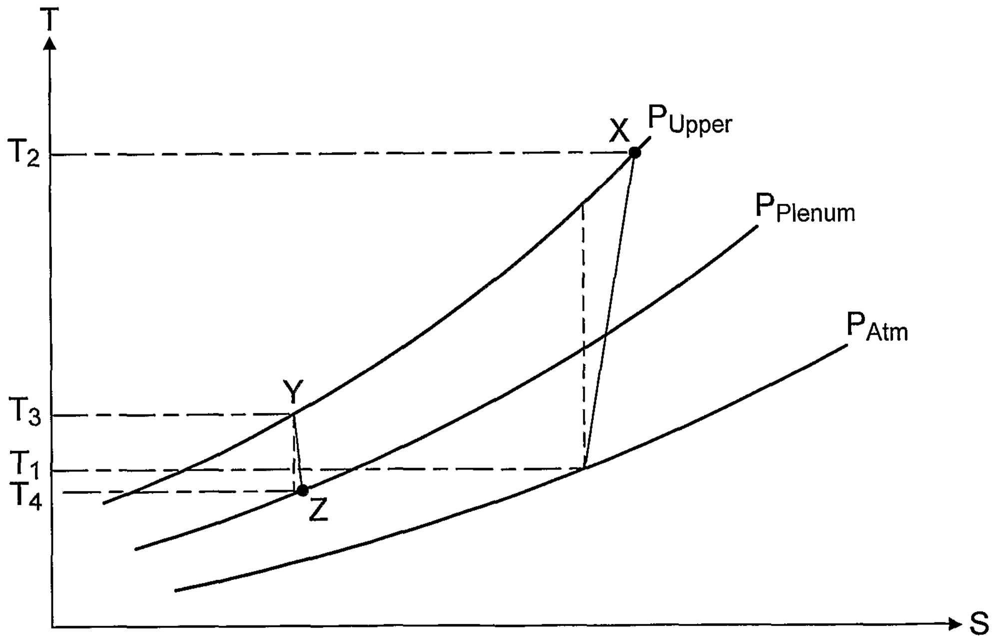
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FIG. 3

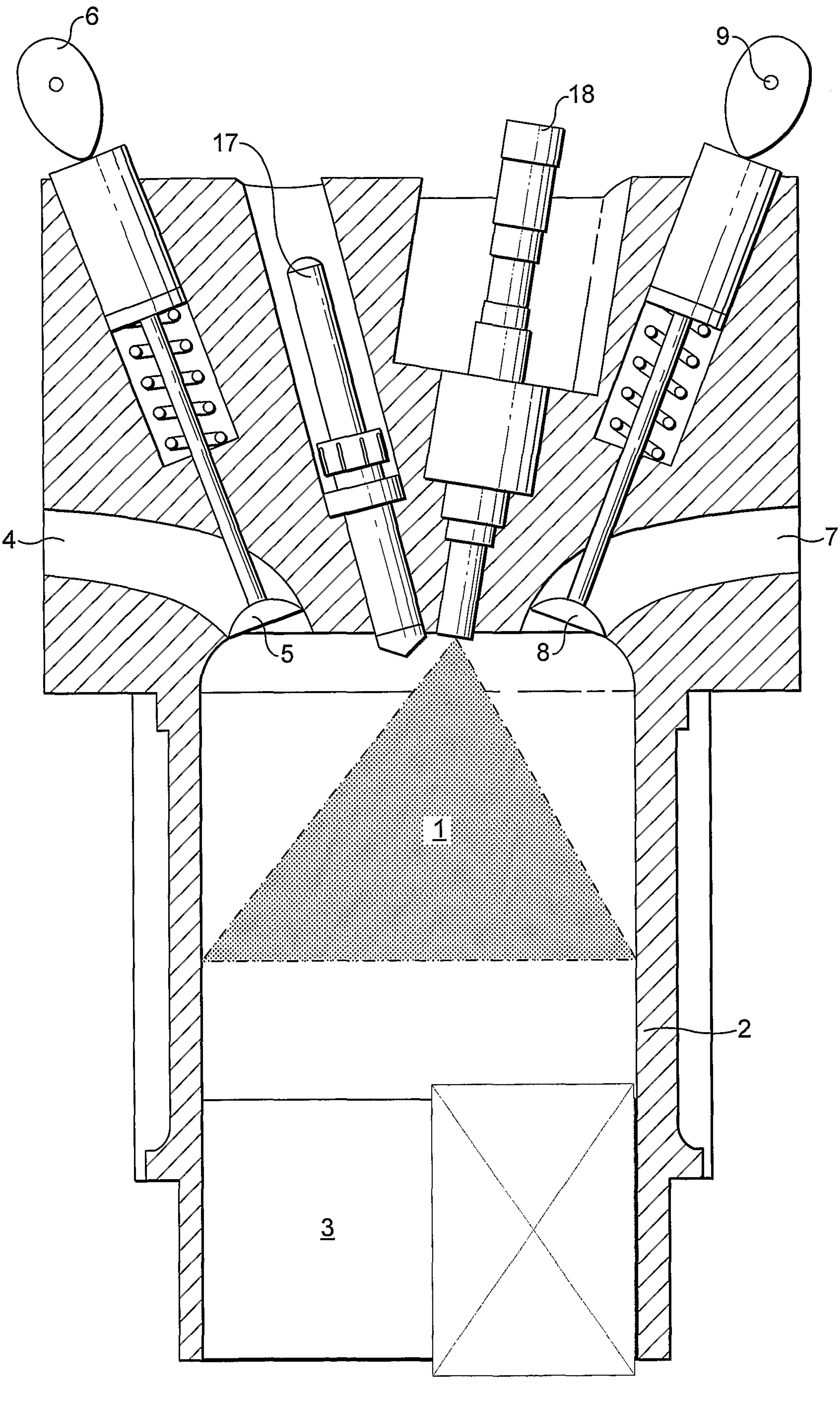

FIG. 2

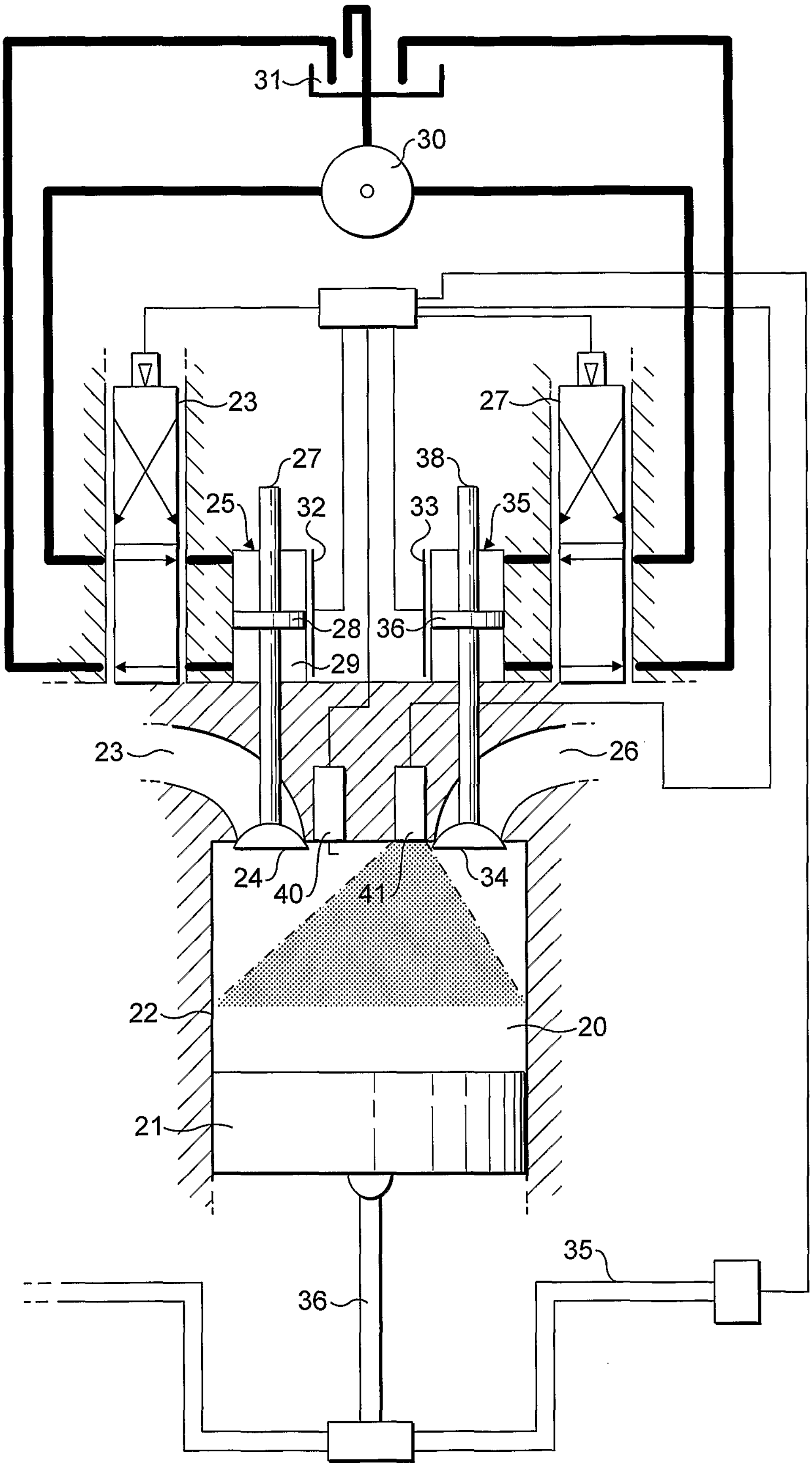

FIG. 4

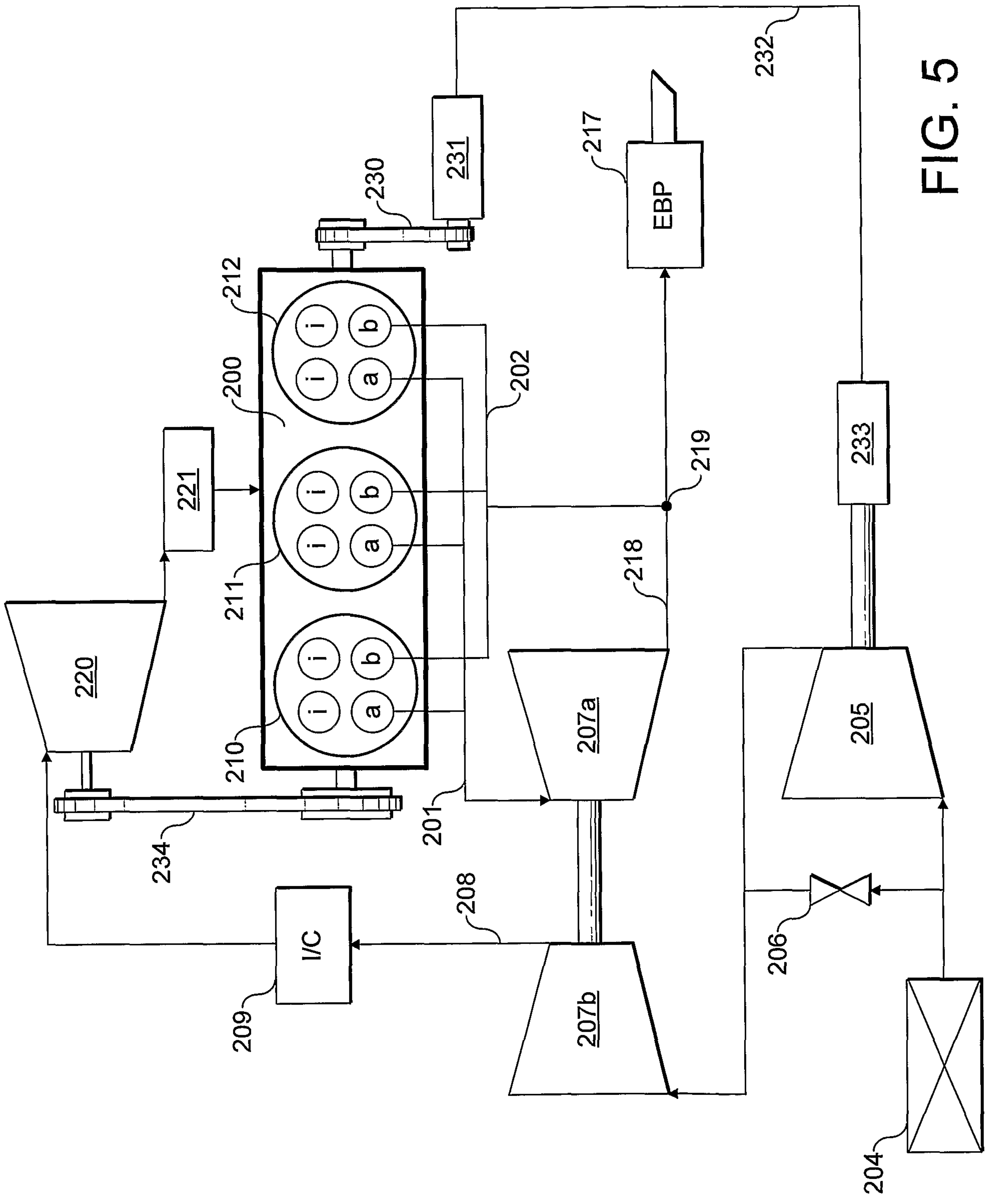

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FIG. 5

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB2005/004334

A. CLASSIFICATION OF SUBJECT MATTER
F02B33/44 F02B37/02 F02B37/04 F02B39/04 F02B29/04
F02B37/18

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F02B F02D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, WPI Data, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	WO 98/55744 A (LEE, HEON, SEOK) 10 December 1998 (1998-12-10) page 7, line 1 - page 10, line 4; figure 1 page 18, line 6 - page 19, line 14; figure 7	1-3,10 4-6,8,9
Y	WO 02/079624 A (SAAB AUTOMOBILE AB; EKENBERG, MARTIN) 10 October 2002 (2002-10-10) page 3, line 17 - page 6, line 24; figure 1	4
Y	US 2001/017123 A1 (RAAB ALOIS ET AL) 30 August 2001 (2001-08-30) the whole document	5,6,8,9

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

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Date of the actual completion of the international search	Date of mailing of the international search report
17 January 2006	24/01/2006

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31–70) 340–2040, Tx. 31 651 epo nl, Fax: (+31–70) 340–3016	Nobre, S

INTERNATIONAL SEARCH REPORT

International Application No
. . ./GB2005/004334

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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