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INT CL **F02D, F02M**
Other: **EPODOC, WPI**

(54) Abstract Title**: Dual combustion mode engine**

(57) An engine 10 is disclosed in which a first group of engine cylinders 14 operate in HCCI combustion mode only and a second group of engine cylinders 12 operate in SI combustion mode only. A first EGR duct is arranged to provide flow from an exhaust 18 of the first group of cylinders 14 to an intake of the first group of cylinders 14 and a first EGR valve 72 is disposed in the first EGR duct to control this flow. A second EGR duct is arranged to provide flow from an exhaust 16 of the second group of cylinders 12 to the intake of the first group of cylinders 14 via a second EGR valve 70 disposed in the second EGR duct. By adjusting the EGR valves 70, 72 the temperature of the gas flowing to the first group of cylinders 14 can be controlled. During a cold start, the second EGR valve 70 is opened to pull exhaust gas from the second group of cylinders 12 through the first group of cylinders 14 so as to preheat them.

Figure 5

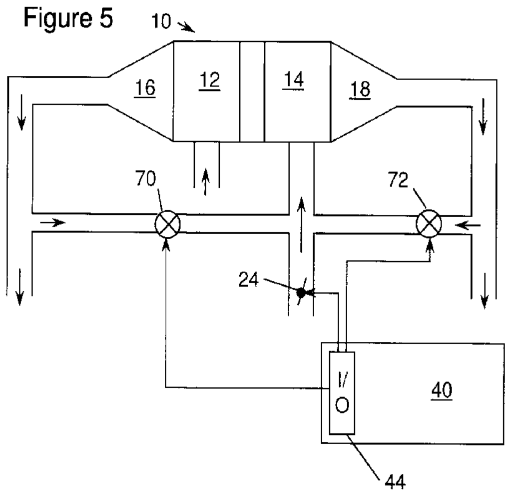

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Figure 1

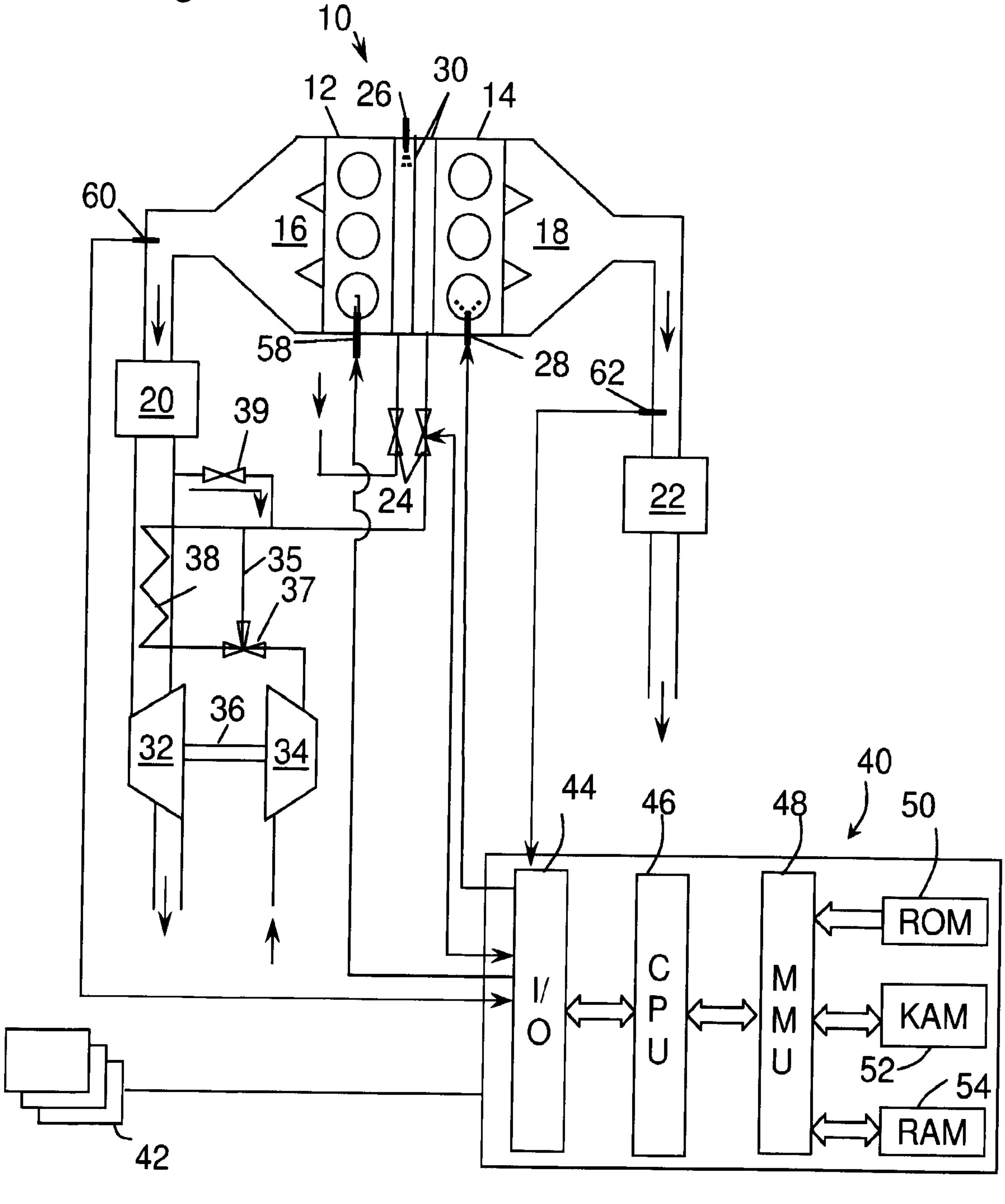

10
12
26
30
14
60
16
18
58
28
62
20
39
24
22
38
35
37
36
32
34
40
44
46
48
50
ROM
I/O
CPU
MMU
KAM
52
54
RAM
42

Figure 2a

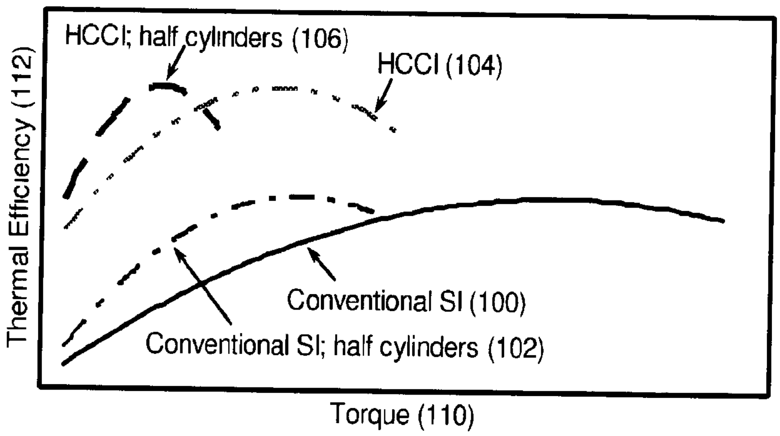


Figure 2b

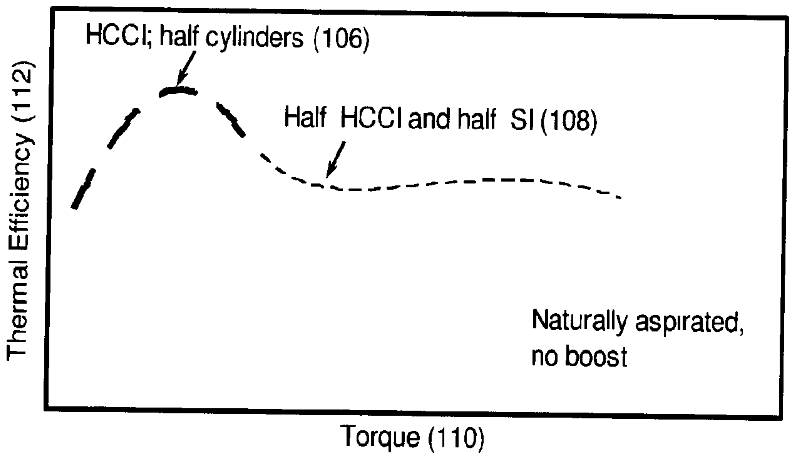


Figure 2c

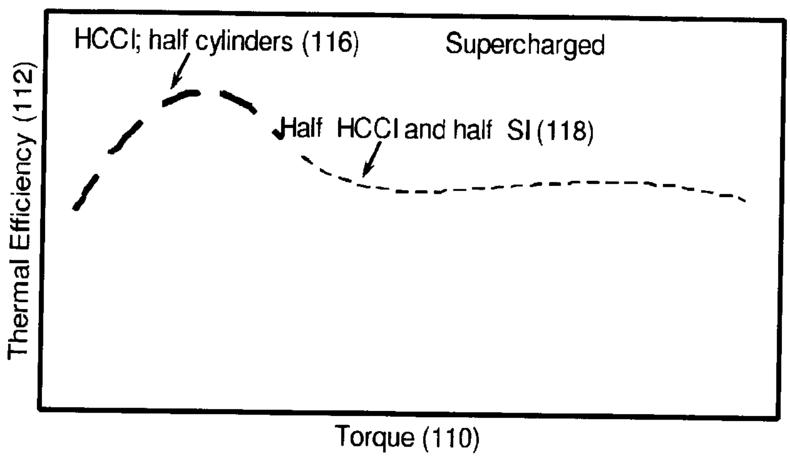


Figure 3

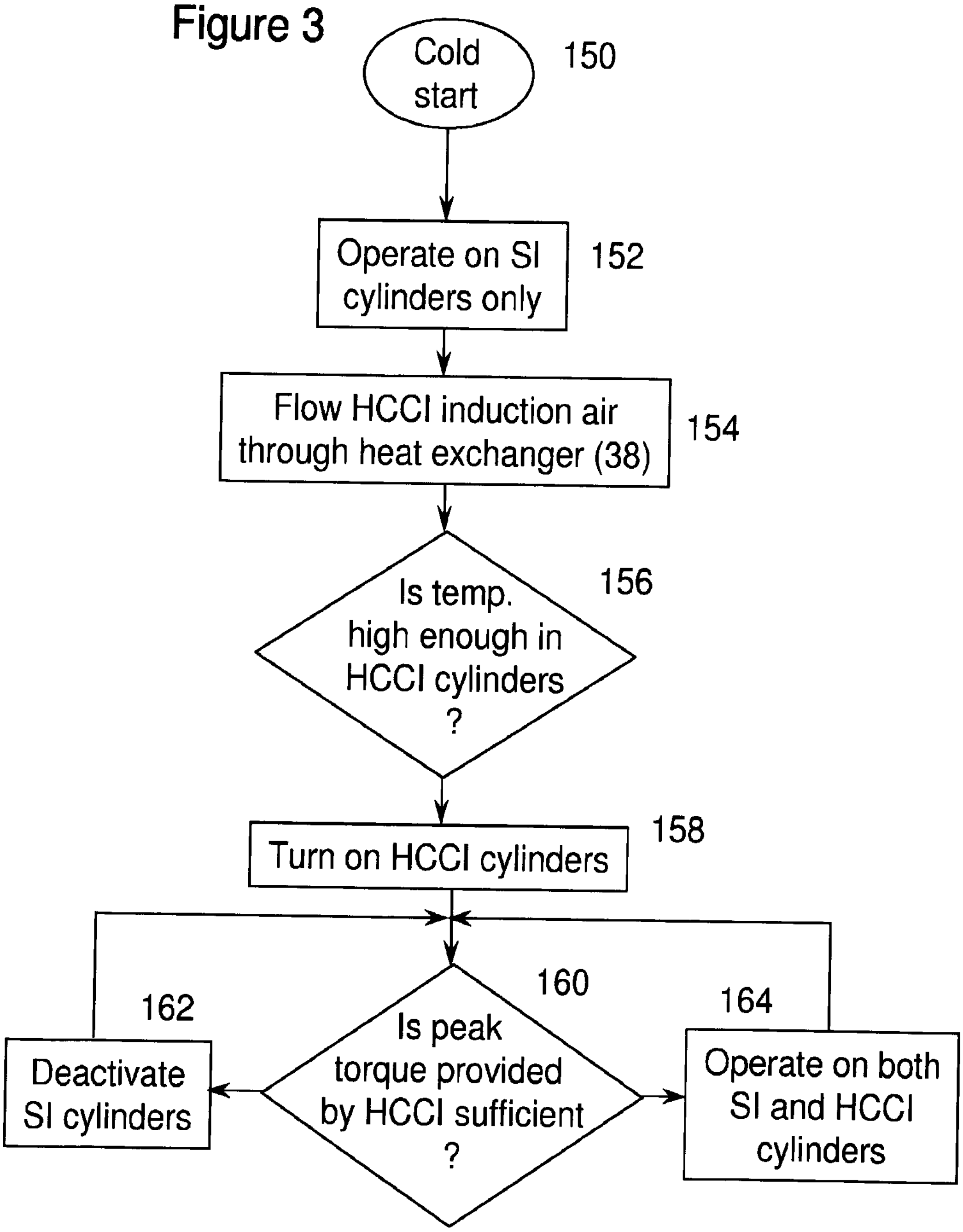

Cold start
150
Operate on SI cylinders only
152
Flow HCCI induction air through heat exchanger (38)
154
Is temp. high enough in HCCI cylinders ?
156
Turn on HCCI cylinders
158
Is peak torque provided by HCCI sufficient ?
160
162
Deactivate SI cylinders
164
Operate on both SI and HCCI cylinders

Figure 4a

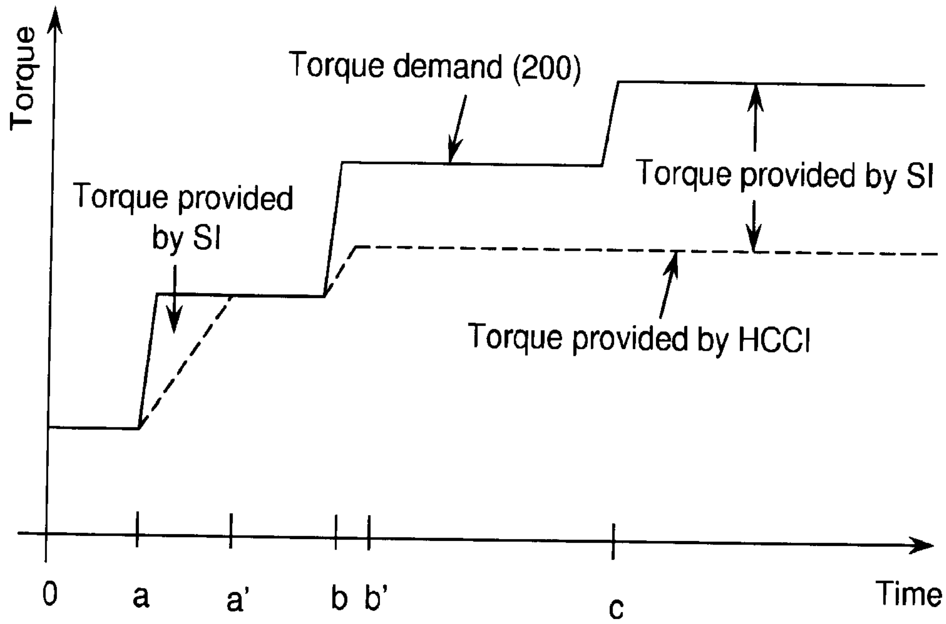


Figure 4b

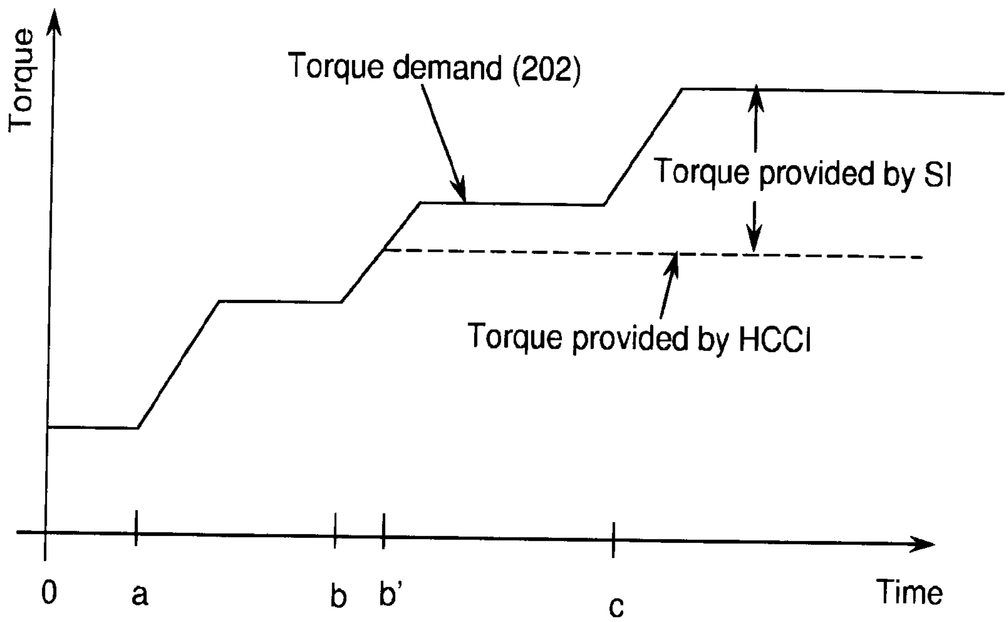
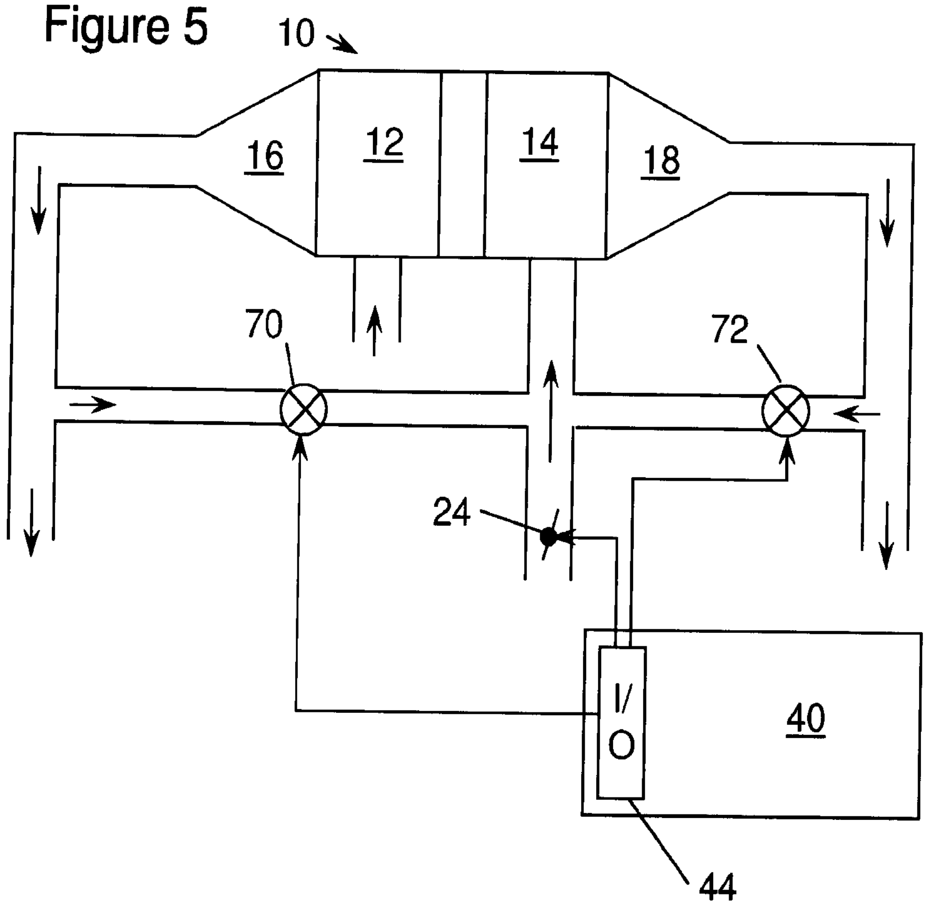


Figure 5



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A Dual Combustion Mode Engine

The present invention relates to an internal combustion engine in which a portion of cylinders exclusively operate under a spark-ignition combustion mode and remaining cylinders exclusively operate under a homogeneous-charge compression-ignition mode.

Homogeneous-charge, compression-ignition combustion is known to those skilled in the art to provide high fuel efficiency and low emission operation in internal combustion engines. However, HCCI operation is feasible in a narrow range in engine torque, approximately one-third of the torque range of a conventional spark-ignited engine. Thus, most HCCI engines being developed are dual mode engines in which HCCI is used at low torque conditions. When a higher torque is desired, operation is transitioned to an alternative combustion mode, such as spark-ignition combustion or heterogeneous, compression-ignition combustion (diesel). Challenges accompanying such transitions include: torque matching (providing driver demanded torque during the transition interval), maintaining emission control, and robustly returning to HCCI combustion, to name a few. Another difficulty encountered in engines which transition from one combustion mode to another is that the combustion system geometry cannot be optimized for either combustion mode, but is instead a compromise. For example, a desirable compression ratio for HCCI combustion is about 15:1 and about 10.5:1 for spark-ignition combustion.

A disadvantage of HCCI combustion is its inferior transient response to a demand for a change in torque, orders of magnitude slower than SI combustion. The inventors of the present invention have recognized that HCCI operation cannot provide a vehicle operator with the responsiveness that they have come to expect from a SI engine.

In U.S. patent application 2004/0182359, an 8-cylinder HCCI/SI engine is described in which HCCI to SI transitions are made one cylinder at a time, i.e., at a lower torque demand all 8 cylinders are operating in HCCI combustion mode and as torque demand exceeds what HCCI combustion can provide, cylinders are individually switched to SI operation. The inventors of the present invention have recognized that it would be desirable to have an engine which provides the desired range in output torque at the high efficiency of HCCI combustion without having to undergo a combustion mode transition in any given cylinder because of the compromises inherent in designing a cylinder to operate robustly and efficiently in both HCCI and SI combustion modes over a wide operating range.

It is an object of this invention to provided an improved internal combustion engine utilising two different forms of combustion process.

According to a first aspect of the invention there is provided an internal combustion engine comprising a first group of engine cylinders, a second group of engine cylinders, a first exhaust gas recirculation duct adapted to provide flow from an exhaust of the first group of engine cylinders to an intake of the first group of engine cylinders, a first exhaust gas recirculation valve disposed in the first exhaust gas recirculation duct, a second exhaust gas recirculation duct adapted to provide flow from an exhaust of the second group of engine cylinders to an intake of the first group of engine cylinders via a first exhaust gas recirculation valve and a second exhaust gas recirculation valve disposed in the second exhaust gas recirculation duct.

The first group of engine cylinders may operate in HCCI combustion mode only and the second group of engine cylinders may operate in SI combustion mode only.

A compression ratio of the first group of engine cylinders may be at least two ratios higher than the second group of engine cylinders.

The engine may further comprise an electronic control unit coupled to the first and second groups of engine cylinders and the first and second exhaust gas recirculation valves, the electronic control unit may command a first position to the first exhaust gas recirculation valve and a second position to the second exhaust gas recirculation valve based on a desired intake temperature in the first group of engine cylinders.

The first group of engine cylinders and the second group of engine cylinders may be mutually exclusive. The exhaust of the first group of engine cylinders may be separated from an exhaust of the second group of engine cylinders. The intake of the first group of engine cylinders may be separated from an intake of the second group of engine cylinders.

The number of cylinders in the first group of engine cylinders may be greater than the number of cylinders in the second group of engine cylinders. The engine may have more than one bank of cylinders and the first group of engine cylinders may comprise all but one cylinder on each bank of cylinders of the engine.

According to a second aspect of the invention there is provided a method for controlling an internal combustion engine comprising adjusting a valve position of a crossover exhaust gas recirculation valve, the crossover exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an intake of a first group of engine cylinders and an exhaust of a second group of engine cylinders wherein the first and second groups are mutually exclusive.

The adjustment may be based on a desired intake temperature of gases provided to the first group of engine cylinders.

The method may further comprise adjusting a valve position of a second exhaust gas recirculation valve, the second exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an exhaust of the first group of engine cylinders and the intake of the first group of engine cylinders.

The valve may be adjusted to a fully open position during starting of the engine.

The exhaust gas recirculation duct may be coupled to the exhaust of the second group of engine cylinders downstream of an exhaust treatment device coupled to the exhaust of the second group of engine cylinders.

According to a third aspect of the invention there is provided a computer readable storage medium having stored data representing instructions executable by a computer comprising instructions to adjust a valve position of a crossover exhaust gas recirculation valve, the crossover exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an intake of a first group of engine cylinders and an exhaust of a second group of engine cylinders wherein the first and second groups are mutually exclusive.

Intakes of the first and second groups of engine cylinders may be separated and exhausts of the first and second groups of engine cylinders may be separated.

The medium may further comprise instructions to determine a desired intake temperature for the first group of engine cylinders wherein the adjustment of the crossover exhaust gas recirculation valve is based on providing the desired intake temperature.

The medium may further comprise instructions to adjust a valve position of a second exhaust gas recirculation valve, the second exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an exhaust of the first group of engine cylinders and the intake of the first group of engine cylinders.

The medium may further comprise instructions to fully open the crossover exhaust gas recirculation valve during starting of the engine.

The medium may further comprise instructions to provide fuel to the second group of engine cylinders and to provide no fuel to the first group of engine cylinders during the starting.

The first group of engine cylinders may comprise all, less one, engine cylinders in each bank of the engine.

The medium may further comprise instructions to determine crank angle of peak pressure for the first group of engine cylinders wherein the adjustment of the crossover exhaust gas recirculation valve is based on providing a desired crank angle of peak pressure.

The desired crank angle of peak pressure may occur between 5 and 20 degrees after top centre.

An advantage of the present method is that control of the EGR valves can be feedback controlled on combustion in the HCCI cylinder. By doing this, efficient HCCI combustion is ensured.

A further advantage is that by providing two controllable exhaust gas streams of different temperatures to the intake of HCCI cylinders, intake temperature to HCCI cylinders can be controlled. Intake temperature is a key factor in controlling ignition timing with HCCI combustion.

The invention will now be described by way of example with reference to the accompanying drawing of which:-

Figure 1 is a schematic representation of an internal combustion engine according to one aspect of the present invention;

Figures 2a-2c are graphs of torque versus fuel efficiency for prior art SI and HCCI engines and an engine according to said one aspect of the present invention;

Figure 3 is a flowchart of a cold start strategy;

Figures 4a and 4b are graphs of driver demanded torque over time for an engine according to the present invention; and

Figure 5 is a schematic of the exhaust gas recirculation arrangement according to an aspect of the present invention.

In Figure 1, a multi-cylinder internal combustion engine 10 is shown. By way of example, engine 10 is shown to have six cylinders, two banks of 3 cylinders each. One bank of cylinders 12 is adapted to operate in a conventional spark-ignition (SI) mode all cylinders operating in this combustion mode are referred to as a 'group' of cylinders.

The other bank of cylinders 14 is adapted to operate in a homogeneous-charge, compression-ignition (HCCI) mode all cylinders operating in this combustion mode are referred to as a 'group' of cylinders. Therefore, in this example, all of the cylinders in the first bank of cylinders 12 form a group of cylinders and all of the cylinders of the other bank of cylinders form an alternative group of cylinders. However, the term 'group' as meant herein means one or more cylinders.

Combustion air is provided to the cylinder banks via an intake manifold 30, which is separated such that air for cylinder bank 12 does not mix with air for cylinder bank 14. Each bank is provided with a throttle valve 24, or other means to control flow.

SI combustion characteristically occurs at stoichiometric proportions of air and fuel, meaning that if combustion were complete, all the fuel and oxygen would be completely combusted to H_2O and CO_2. To control the amount of torque produced in SI combustion, the amount of air is controlled by the respective throttle valve 24. The amount of fuel added to the air, via injector 26, is metered to provide a stoichiometric mixture. For clarity, only one fuel injector 26 is shown in Figure 1 for the 3 cylinders of bank 12. However, each cylinder is provided with a fuel injector. Similarly, each cylinder in bank 14 is provided with a fuel injector like fuel injector 28. In Figure 1, bank 12 is shown with a port injector in which fuel is sprayed outside of the cylinder and is brought into the cylinder with the combustion air and bank 14 is shown with a direct fuel injector in which fuel is sprayed directly into the combustion air which has been inducted into the cylinder. These types of fuel injection systems are shown by way of example. Both banks could be provided with port injection or both with direct injection according to the present invention. I.e., both HCCI and SI combustion can be accomplished with port, direct, or a combination of port and direct injection. Alternatively, fuel is provided by central body injection for either combustion mode.

Again, for clarity, only one of the three spark plugs for each of the cylinders of bank 12 is shown. Bank 14 cylinders may also have spark plugs. Although the bank 14 cylinders are HCCI cylinders, which indicates that combustion is initiated by compression ignition, it is known to those skilled in the art that at some operating conditions, it is useful to employ spark assist to initiate combustion. Alternatively, another ignition assist device such as glow plugs, plasma jet igniters, catalytic assisted glow plugs, as examples, could be used for ignition assist in HCCI. In SI combustion, a spark initiates a flame kernel and a flame front travels throughout the cylinder. In spark assisted HCCI, a spark initiates a flame kernel at the location of the spark plug. However, the mixture in the cylinder is too weak (not enough fuel or too much burned gases in the mixture) to sustain a flame front traveling through the cylinder gases. The flame kernel combusts the fuel-air mixture near the spark plug. The release of energy by the combustion of the mixture near the spark plug increases the pressure in the cylinder thereby causing the gases away from the spark plug to attain their ignition temperature and to self-ignite. When spark assist HCCI is contemplated, all SI cylinders are provided with a spark plug 58.

Engine 10 is shown to be a 6-cylinder with bank 12 being SI and bank 14 being HCCI by way of example. This is not intended to be limiting. Engine 10 has any number of cylinders greater than one and in any configuration such as 'in-line', 'V', 'W', 'radial', 'opposed' or any other cylinder arrangement. The HCCI and SI cylinders need not be segregated by banks. There could be HCCI and SI cylinders on any given bank. However, as mentioned above, the intake gases to the HCCI cylinders and SI cylinders remain separated and exhaust gases coming from HCCI cylinders and SI cylinders also remain separated. Thus, such arrangements may require complicated manifolds to maintain the separation. An expected arrangement is that every other cylinder in the firing order is alternately HCCI and SI.

SI engines are typically produced with a 9.5-10.5:1 compression ratio, which is the ratio of the volume in the cylinder above the piston when the piston is at the top of its travel divided by the volume in the cylinder above the piston when the piston is at the bottom of its travel. HCCI combustion occurs more favorably with a higher compression ratio: 13-15:1. In prior art engines in which combustion mode is transitioned, the compression ratio selected is a compromise between the two compression ratios.

According to the present invention, however, because each cylinder is optimized for a single combustion mode, the engine is produced with the compression ratio appropriate for the particular combustion mode. Thus, unlike prior art engines, the engine according to the present invention has some cylinders with a substantially higher compression ratio than other cylinders.

HCCI combustion occurs in a dilute mixture, either very lean of stoichiometric with excess air and/or with a very high level of exhaust dilution. It is well known to those skilled in the art to provide exhaust dilution by either recirculating exhaust gases into the engine intake, known as exhaust gas recirculation (EGR) sometimes referred to as external EGR, or to retain exhaust gases in the cylinder from a prior combustion event to mix with the combustion gases of an upcoming combustion event, commonly known as internal EGR. The latter is often achieved by valve timing adjustments. Typically exhaust gases are routed from an exhaust duct to an intake duct via a control valve (EGR valve). The present invention provides for an alternative configuration for EGR in which gases exhausted from the SI cylinder bank 12 are routed to the intake of the HCCI cylinder bank 14 via valve 39.

In Figure 1, the exhaust gases are collected downstream of exhaust gas treatment device 20. This is shown by way of example and not intended to be limiting. The exhaust gases can be taken from any position in the exhaust duct. There are two advantages for circulating exhaust gases from the SI bank 12 to the HCCI bank 14. Typically, SI combustion occurs with a stoichiometric mixture, which provides combustion gases containing primarily CO_2, H_2O, and N_2. In contrast, HCCI cylinders combust lean mixtures which have excess air. Thus, HCCI exhaust gas has significant levels of O_2 and more N_2 than a SI exhaust gas. To obtain a desired dilutent fraction, a greater amount of HCCI exhaust gas is recycled compared with the SI exhaust gas quantity.

It is well known to those skilled in the art that to achieve ignition in HCCI, it is common to heat the intake air. Because the exhaust gas temperature is higher with SI combustion, less intake heating is required when the EGR employed comes from SI combusting cylinders. In particular, with reference to Figure 1, gas from the exhaust duct coupled to cylinder bank 12 is drawn through EGR system 39 and supplied to the intake of cylinder bank 14 via a control valve.

Continuing with Figure 1, each cylinder bank is provided with an exhaust gas treatment device, 20 and 22. In one embodiment, device 20 is a three-way catalyst, which efficiently oxidizes CO and hydrocarbons and reduces nitrogen oxides (NOx) when provided a stoichiometric exhaust gas. As mentioned above, obtaining higher fuel efficiency motivates HCCI development. Another advantage of HCCI combustion, which occurs in a very lean or dilute mixture, is that it produces very low levels of NOx, particularly compared to SI operation. In one embodiment, the HCCI cylinders require no NOx treatment and treatment device 22 is an oxidation catalyst to process unburned fuel and CO. In another embodiment, a lean NOx treatment device is employed to process the low levels of NOx when very low NOx levels are required or when HCCI operation is extended into regions at which the NOx produced is somewhat higher than typical HCCI combustion. In this embodiment, the lean NOx treatment device is either a lean NOx trap or a lean NOx catalyst. A lean NOx trap stores NOx during lean operation. When the trap is no longer able to store additional NOx, the trap is purged by operating lean for a period of time. During the lean operation, the NOx is desorbed from the trap and reacted to N2 and O2. To accomplish a rich excursion with HCCI, one alternative is to operate with a very high level of EGR to displace excess air. A lean NOx catalyst processes NOx in the presence of a reductant, either fuel or urea.

In Figure 1, an indication of exhaust gas constituents is provided by an exhaust gas sensor, 60 and 62, situated in the exhaust ducts exiting each cylinder bank. Only a single sensor is shown in Figure 1. Exhaust gas sensor 60 is an oxygen sensor, either heated or unheated, which provides an indication of whether the exhaust gas is near stoichiometric. In another embodiment, sensor 60 is a wide-range oxygen sensor provides a measure of exhaust gas stoichiometric.

Exhaust gas sensor 62 measures NOx concentration. Alternatively, sensor 62 can be a wide-range oxygen sensor.

Although only one exhaust gas sensor is shown in each of the exhaust ducts of engine 10, it is known to use multiple exhaust gas sensors. In one embodiment, both a wide-range oxygen sensor and a NOx sensor are placed in place of sensor 62. Furthermore, it is common practice to provide a sensor both upstream and downstream of an exhaust treatment device. The inventors of the present invention contemplate any known exhaust gas sensor type in any location in the exhaust ducts.

The signal from the exhaust gas oxygen sensor 60 is commonly used for air-fuel ratio feedback control of SI combustion. Analogously, HCCI combustion timing is controlled by adjusting intake temperature, according to one alternative embodiment. Adjustment of intake temperature is feedback controlled based on a combustion parameter such as crank angle of peak pressure. Examples of sensors from which crank angle of peak pressure can be ascertained include: head bolt strain gauge, in-cylinder pressure sensor, ionization sensor, a head gasket sensor, sensor measuring instantaneous flywheel speed, etc. For stoichiometric SI combustion, it is well known by those skilled in the art that the crank angle of peak pressure corresponding to peak efficiency operation (at a given speed/torque condition) occurs roughly at 15 degrees after top dead center. Alternative combustion systems, particular lean burn, tend to have the crank angle of peak pressure occur at a somewhat earlier time, e.g., 12 degrees after top dead center to achieve peak efficiency. Furthermore, there are other objectives, besides achieving peak efficiency, such as emission control, which cause the desired crank angle of peak pressure to be other than that providing peak efficiency. It is expected that a desired crank angle of peak pressure is in a range of 5 to 20 degrees after top center. Various combustion control parameters, such as: intake temperature, EGR valve position, throttle valve position, flow through an intake heat exchanger, and pressure charging, can be feedback controlled based on crank angle of peak pressure, particularly for the HCCI cylinders.

Because HCCI combustion is dilute, the peak torque capable from a given cylinder is much less than peak torque from a SI cylinder. To increase the amount of torque from a HCCI cylinder, compressor 34 increases the intake manifold pressure on the HCCI cylinders, allowing for an increased amount of fuel delivery while maintaining a high dilution.

As shown in Figure 1, compressor 34 is connected by a shaft to turbine 32, a device known as a turbocharger. The unconventional ducting of Figure 1 has turbine 32 extracting work from SI cylinder exhaust gases which compress intake gases of HCCI cylinders via compressor 34. HCCI combustion is known to provide superior fuel efficiency to SI combustion. Thus, HCCI exhaust gases have a lower enthalpy than SI exhaust gases because HCCI allows more of the energy release of combustion to be extracted. Thus, it is desirable to extract the SI exhaust gas energy to pressure charge the HCCI inlet. In another embodiment, exhaust turbine 32 is coupled to the HCCI exhaust duct. The turbocharger of Figure 1 is a variable geometry turbocharger. In yet another embodiment, a supercharger is provided in place of turbocharger (comprising elements 32, 34, and 36). A supercharger is a compressor, like compressor 34 of Figure 1, which is driven by engine 10. A supercharger is not coupled to a turbine.

In Figure 1, an intake gas heat exchanger 38 is contained with the exhaust duct coupled to the SI cylinders. It is known in the art that one of the methods to control ignition timing of HCCI combustion is by controlling the intake temperature. Diverter valve 37 allows adjustment of the quantity of HCCI intake gases passing through heat exchanger 38 and the quantity passing through bypass duct 35, thereby providing control of HCCI intake temperature.

Continuing to refer to Figure 1, electronic control unit (ECU) 40 is provided to control engine 10. ECU 40 has a microprocessor 46, called a central processing unit (CPU), in communication with memory management unit (MMU) 48. MMU 48 controls the movement of data among the various computer readable storage media and communicates data to and from CPU 46. The computer readable storage media preferably include volatile and nonvolatile storage in read-only memory (ROM) 50, random-access memory (RAM) 54, and keep-alive memory (KAM) 52, for example. KAM 52 may be used to store various operating variables while CPU 46 is powered down. The computer-readable storage media may be implemented using any of a number of known memory devices such as PROMs (programmable read-only memory), EPROMs (electrically PROM), EEPROMs (electrically erasable PROM), flash memory, or any other electric, magnetic, optical, or combination memory devices capable of storing data, some of which represent executable instructions, used by CPU 46 in controlling the engine or vehicle into which the engine is mounted. The computer-readable storage media may also include floppy disks, CD-ROMs, hard disks, and the like. CPU 46 communicates with various sensors and actuators via an input/output (I/O) interface 44.

Examples of items that are actuated under control by CPU 46, through I/O interface 44, are commands to fuel injectors 26 and 28 such as fuel injection timing, fuel injection rate, and fuel injection duration. Additional parameters under CPU 46 control are position of throttle valves 24, timing of spark plug 58, position of control EGR valves 39, other control valves 37, variable geometry turbocharger nozzle position, intake and exhaust valve timing, and others.

Sensors 42 communicating input through I/O interface 44 may include piston position, engine rotational speed, vehicle speed, coolant temperature, intake manifold pressure, accelerator pedal position, throttle valve position, air temperature, exhaust temperature, exhaust stoichiometry, exhaust component concentration, and air flow. Some ECU 40 architectures do not contain MMU 48. If no MMU 48 is employed, CPU 46 manages data and connects directly to ROM 50, RAM 54, and KAM 52.

Of course, the present invention could utilize more than one CPU 46 to provide engine control and ECU 40 may contain multiple ROM 50, RAM 54, and KAM 52 coupled to MMU 48 or CPU 46 depending upon the particular application.

Referring to Figure 2a, a graph of thermal efficiency as a function of torque is shown for a conventional SI engine as curve 100. Curve 104 indicates the higher thermal efficiency that is possible when operating the same displacement engine in the HCCI combustion mode. The thermal efficiency is markedly improved. However, HCCI does not provide the same range in torque as a SI engine of the same displacement. To provide the same torque level, either the displacement of the engine has to be roughly doubled or combustion mode is changed from HCCI at low torque and then SI when higher torque is demanded.

Also shown in Figure 2a are curves 102 and 106, which is only half of the engine cylinders operating on SI and HCCI respectively. The peak thermal efficiency is the same as the corresponding curves 100 and 104 when the engine is operated with all cylinders on SI and HCCI, respectively. The range in torque, is half that of the running the whole engine.

According to an aspect of the present invention, half of the cylinders are operated with HCCI combustion and half of the cylinders are operated with SI combustion, the effect of such operation on torque and thermal efficiency being shown in Figure 2b. Note that Figure 2b relates to a naturally aspirated engine in which there is no supercharger or turbocharger to pressurize intake gases. Because of the high efficiency of HCCI combustion, it is desirable to operate only the HCCI cylinders at low torque demands.

Thus, curve 106 of Figures 2a and 2b are identical, i.e., half of the cylinders operating on HCCI and the other cylinders being deactivated. When a higher demand in torque is desired, SI cylinders are activated and torque is being provided by both the SI and HCCI cylinders, shown as curve 108 in Figure 2b. Because HCCI cylinders cannot provide the same torque range as SI, the peak torque in Figure 2b is less than what is shown in Figure 2a. The efficiency shown in Figure 2b exceeds that of the SI engine efficiency, curve 102, of Figure 2a at all values of torque.

To make up for the lesser torque of the engine illustrated in Figure 2b, either the displacement of the engine can be increased or boosting applied. Boosting can be applied to the HCCI cylinders or all engine cylinders. However, as discussed above, since HCCI provides less torque, an approach for HCCI cylinders to match torque is to boost only HCCI cylinders. Curve 116 of Figure 2c shows thermal efficiency where boost is provided by a supercharger. The torque range of curve 116 is wider than that of curve 106 (of Figure 2b) due to the increased amount of air delivered to the HCCI cylinders by the supercharger. The torque level at which half HCCI and half SI (curve 118 of Figure 2c) is invoked it higher than in Figure 2b. The maximum torque in Figure 2c is about that of Figure 2a.

If a turbocharger were employed in place of a supercharger, no increase in torque range with HCCI only operation is possible because the SI cylinders are deactivated, thus no exhaust to drive the turbocharger.

Because achieving a sufficiently high temperature to cause auto-ignition is paramount in HCCI combustion, providing a robust cold start presents a serious hurdle for

HCCI combustion. Those skilled in the art normally consider it necessary to start on SI combustion and transitioning to HCCI combustion after the engine has achieved a suitable operating temperature. However, with the present invention, the cylinders are adapted to operate in only one combustion mode. To overcome the problem of cold starting, the inventors propose starting the engine using only the SI cylinders. During the period of SI combustion, air can be delivered to HCCI cylinders through heat exchanger 38. By blowing warm air through the HCCI cylinder bank 14, the engine surfaces can be preheated and ready for HCCI combustion. In addition, the engine coolant is heated by the SI cylinders and preheats the HCCI cylinders.

According to an aspect of the present invention, the SI cylinders are equipped with valve deactivators (not shown). During HCCI only operation, the SI cylinders are deactivated by closing off the intake and exhaust valves. The piston continues to reciprocate, but the gas in the cylinder at the last combustion event remains trapped in the cylinder. If the valves were allowed to remain active, the flow of air through SI cylinder bank 12 would flow into exhaust treatment device 20. If device 20 is a three-way catalyst, oxygen would be absorbed onto the surfaces and when the SI cylinders were reactivated, the three-way catalyst would be unable to reduce NOx until such oxygen is removed from device 20. Furthermore, the flow of air through SI cylinder bank 20 cools the engine down, thereby making restart more difficult.

In one embodiment, valve deactivators are provided for the HCCI cylinders (not shown in Figure 1). However, because HCCI combustion is more efficient than SI combustion, it is desirable to operate the HCCI cylinders whenever possible. When the HCCI cylinders have not attained a suitable operating temperature, the HCCI cylinders can be deactivated. However, according to an aspect of the invention discussed above, exhaust waste energy from SI cylinders is transferred to HCCI induction air via heat exchanger 38. Thus, it may be preferable to allow valves in HCCI cylinders to operate as normal to allow heating.

Referring to Figure 3, a cold start is initiated in block 150. The engine starts operation on SI cylinders only, block 152. Induction air flows through heat exchanger 38 and then through HCCI cylinders to warm up the HCCI cylinders, block 154. In block 156, it is determined whether a sufficient temperature has been achieved. If so, operation in the HCCI cylinders is initiated in block 158. If the torque that can be produced by the HCCI cylinders is sufficient to meet torque demand (determined in block 160), control is passed to block 162 in which the SI cylinders are deactivated. If not enough torque can be produced in the HCCI cylinders, the engine is operated with all cylinders active, block 164.

In Figure 4a, a hypothetical driver torque demand as a function of time is shown as line 200. Between time 0 and time a, the driver demands a relatively low torque, at time a, the driver pushes down the accelerator pedal, which indicates demand for higher torque. The driver again demands increased torque at times b and c. Between times 0 and a, only the HCCI cylinders are active. Since the HCCI combustion mode provides superior fuel economy and the HCCI cylinders are capable of providing the desired torque, the SI cylinders are deactivated. As the torque demand isn't changing, there is no concern about the slow transient response of HCCI combustion. When the transition to a higher torque is demanded at time a, the HCCI cylinders cannot respond quickly enough to attain the new, higher torque. At time a', the HCCI cylinders have attained the desired torque level. However, between time a and time a', if only HCCI cylinders were active, the torque response of the vehicle would be unacceptable to the operator of the vehicle. Thus, according to the present invention, the SI cylinders are activated at time a so that the operator demanded torque can be more closely followed than with HCCI cylinders alone. As the HCCI cylinders ramp up their torque production, the SI cylinders are ramped down and eventually deactivated at time a'. At time b, another rapid torque increase is demanded. Again, the SI cylinders are activated to fill in and the torque produced by the HCCI cylinders is ramped up. As at time a when a rapid torque increase is demanded, the HCCI cylinders are too slow to provide the transient response requested. Thus, the SI cylinders are reactivated at time b. At time b', the HCCI cylinders have attained their maximum torque output condition. Thus at time b', the SI cylinders remain active. At time c, the further increase in demanded torque is provided solely by the SI cylinders because the HCCI cylinders are already operating at their peak capacity.

In Figure 4b, a similar increase in torque is shown, however with a much slower demand for transient torque response. From time 0 to time b', the driver demand is slow enough that the HCCI cylinders can provide the desired torque demand. At time b', the HCCI cylinders have reached their capacity and the SI cylinders are activated. Further increases in torque are provided by the SI cylinders.

In another embodiment, the SI cylinders remain active at all times. In one example of this embodiment, an 8-cylinder engine is started on 2 SI cylinders. The remaining 6 cylinders are HCCI cylinders, which are turned on when they reach a suitable temperature which supports robust HCCI combustion. In this embodiment, the SI cylinders remain operational even after HCCI combustion has been achieved in the 6 HCCI cylinders.

Referring now to Figure 5, a HCCI bank 14 and a SI bank 12 are incorporated in engine 10. The arrangement of Figure 5 is provided simply for ease of illustration. As discussed above, various arrangements are contemplated by the inventors of the present invention, in one example each bank has one HCCI cylinder and three SI cylinders with complicated manifolds to keep their intake and exhaust gases separated.) Bank 12 is provided with exhaust 16 and bank 14 is provided with exhaust 18. A portion of exhaust gases from bank 12 may be drawn off by an EGR system through EGR valve 70. Similarly, exhaust gases are drawn from an exhaust from bank 14 through EGR valve 72. Both EGR ducts flow into the intake to bank 14. The intake to bank 14 is also supplied with throttle valve 24. Throttle valve 24, EGR valve 70 and EGR valve 72 are controlled by electronic control unit 40.

Because HCCI combustion is very dilute, HCCI combustion gases are at a much lower temperature than SI combustion gases. By controlling the proportion of EGR gases coming from bank 12 and from bank 14, the temperature in HCCI cylinders is controlled. As mentioned above, one of the ways, known by those skilled in the art, for controlling HCCI combustion timing is by adjusting the temperature of the gases in the HCCI cylinder. By continuing to operate SI cylinders while HCCI cylinders are operating, the exhaust gases from SI cylinders is available for recycle to the HCCI cylinders for controlling temperature in HCCI cylinders.

Therefore in summary there is provided an engine in which a first group of engine cylinders operate in HCCI combustion mode only and a second group of engine cylinders operate in SI combustion mode only, two exhaust gas recirculation ducts and valves are provided, a first exhaust gas recirculation duct adapted to provide flow from an exhaust of the first portion of engine cylinders to an intake of the first portion of engine cylinders with a first exhaust gas recirculation valve disposed in the first exhaust gas recirculation duct and a second exhaust gas recirculation duct adapted to provide flow from an exhaust of the second portion of engine cylinders to an intake of the first portion of engine cylinders via a first exhaust gas recirculation valve with a second exhaust gas recirculation valve disposed in the second exhaust gas recirculation duct.

An electronic control unit coupled to the first and second portions of engine cylinders and the first and second exhaust gas recirculation valves commands a first position to the first exhaust gas recirculation valve and a second position to the second exhaust gas recirculation valve based on a desired intake temperature.

Alternatively, the electronic control unit coupled to the first and second portions of engine cylinders and the first and second exhaust gas recirculation valves commands a first position to the first exhaust gas recirculation valve and a second position to the second exhaust gas recirculation valve based on a signal from a combustion sensor. Feedback control of valve position can be based on crank angle of peak pressure in the first portion of engine cylinders.

CLAIMS

1. An internal combustion engine comprising a first group of engine cylinders, a second group of engine cylinders, a first exhaust gas recirculation duct adapted to provide flow from an exhaust of the first group of engine cylinders to an intake of the first group of engine cylinders, a first exhaust gas recirculation valve disposed in the first exhaust gas recirculation duct, a second exhaust gas recirculation duct adapted to provide flow from an exhaust of the second group of engine cylinders to an intake of the first group of engine cylinders via a first exhaust gas recirculation valve and a second exhaust gas recirculation valve disposed in the second exhaust gas recirculation duct.

2. An engine as claimed in claim 1 wherein a compression ratio of the first group of engine cylinders is at least two ratios higher than the second group of engine cylinders.

3. An engine as claimed in claim 1 or in claim 2 wherein the engine further comprises an electronic control unit coupled to the first and second groups of engine cylinders and the first and second exhaust gas recirculation valves, the electronic control unit commanding a first position to the first exhaust gas recirculation valve and a second position to the second exhaust gas recirculation valve based on a desired intake temperature in the first group of engine cylinders.

4. An engine as claimed in any of claims 1 to 3 wherein the first group of engine cylinders and the second group of engine cylinders are mutually exclusive.

5. An engine as claimed in any of claims 1 to 4 wherein the exhaust of the first group of engine cylinders

is separated from an exhaust of the second group of engine cylinders.

6. An engine as claimed in any of claims 1 to 5 wherein the intake of the first group of engine cylinders is separated from an intake of the second group of engine cylinders.

7. An engine as claimed in any of claims 1 to 6 wherein the number of cylinders in the first group of engine cylinders is greater than the number of cylinders in the second group of engine cylinders.

8. An engine as claimed in claim 7 wherein the engine has more than one bank of cylinders and the first group of engine cylinders comprises all but one cylinder on each bank of cylinders of the engine.

9. A method for controlling an internal combustion engine comprising adjusting a valve position of a crossover exhaust gas recirculation valve, the crossover exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an intake of a first group of engine cylinders and an exhaust of a second group of engine cylinders wherein the first and second groups are mutually exclusive.

10. A method as claimed in claim 9 wherein the adjustment is based on a desired intake temperature of gases provided to the first group of engine cylinders.

11. A method as claimed in claim 9 or in claim 10 wherein the method further comprises adjusting a valve position of a second exhaust gas recirculation valve, the second exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an exhaust of the

first group of engine cylinders and the intake of the first group of engine cylinders.

12. A method as claimed in any of claims 9 to 11 wherein the valve is adjusted to a fully open position during starting of the engine.

13. A method as claimed in any of claims 9 to 12 wherein the exhaust gas recirculation duct is coupled to the exhaust of the second group of engine cylinders downstream of an exhaust treatment device coupled to the exhaust of the second group of engine cylinders.

14. A computer readable storage medium having stored data representing instructions executable by a computer comprising instructions to adjust a valve position of a crossover exhaust gas recirculation valve, the crossover exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an intake of a first group of engine cylinders and an exhaust of a second group of engine cylinders wherein the first and second groups are mutually exclusive.

15. A medium as claimed in claim 14 wherein intakes of the first and second groups of engine cylinders are separated and exhausts of the first and second groups of engine cylinders are separated.

16. A medium as claimed in claim 14 or in claim 15 wherein the medium further comprises instructions to determine a desired intake temperature for the first group of engine cylinders wherein the adjustment of the crossover exhaust gas recirculation valve is based on providing the desired intake temperature.

17. A medium as claimed in claim 16 wherein the medium further comprises instructions to adjust a valve position of

a second exhaust gas recirculation valve, the second exhaust gas recirculation valve being disposed in an exhaust gas recirculation duct between an exhaust of the first group of engine cylinders and the intake of the first group of engine cylinders.

18. A medium as claimed in any of claims 14 to 17 wherein the medium further comprises instructions to fully open the crossover exhaust gas recirculation valve during starting of the engine.

19. A medium as claimed in claim 18 wherein the medium further comprises instructions to provide fuel to the second group of engine cylinders and to provide no fuel to the first group of engine cylinders during the starting.

20. A medium as claimed in any of claims 14 to 19 wherein the first group of engine cylinders comprise all, less one, engine cylinders in each bank of the engine.

21. A medium as claimed in any of claims 14 to 20 wherein the medium further comprises instructions to determine crank angle of peak pressure for the first group of engine cylinders wherein the adjustment of the crossover exhaust gas recirculation valve is based on providing a desired crank angle of peak pressure.

22. A medium as claimed in claim 21 wherein the desired crank angle of peak pressure occurs between 5 and 20 degrees after top centre.

23. An internal combustion engine substantially as described herein with reference to the accompanying drawing.

24. A method for controlling an internal combustion engine substantially as described herein with reference to the accompanying drawing.

25. A computer readable storage medium substantially as described herein with reference to the accompanying drawing.

Application No: GB0618010.3 **Examiner:** John Twin

Claims searched: 1 to 25 **Date of search:** 9 January 2007

Patents Act 1977: Search Report under Section 17

Documents considered to be relevant:

Category	Relevant to claims	Identity of document and passage or figure of particular relevance
X	9,13 at least	DE 19500761 A (Daimler Benz) - see eg WPI abstract accession no.1996-334549[34], fig.1
X	9,13 at least	WO 2006/032886 A (Lotus Car et al.) - see eg fig.2
X	9 at least	JP 2005054710 A (Nissan Diesel) - see eg EPODOC abstract, fig.1
X	9 at least	DE 4421258 A (BMW) - see eg WPI abstract accession no.1996-040970[05]
X	9 at least	US 4303053 A (Nissan) - see eg fig.2
X	9 at least	JP 09112256 A (Mitsubishi) - see eg WPI abstract accession no.1997-296152[27]; the fig.

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X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category	P	Document published on or after the declared priority date but before the filing date of this invention.
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Field of Search:

Search of GB, EP, WO & US patent documents classified in the following areas of the UKC^{X} :

Worldwide search of patent documents classified in the following areas of the IPC

F02D; F02M

The following online and other databases have been used in the preparation of this search report

EPODOC, WPI