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GB 2277776 A **GB 0878278 A** **DE 003405558 A**

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INT CL[6] **F02B 3/06 3/08 3/10 7/00 7/02 7/04**
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(54) Abstract Title
Two stage fuel injection for a compression ignition engine

(57) A compression ignition combustion engine that has fuel injected into a cylinder in two stages or feeds; the first stage injects fuel at the end of the induction stroke or beginning of the compression stroke to form a premixed charge with a fuel/air ratio insufficient to cause auto-ignition after compression; and a second stage that injects fuel during the compression stroke to form a locally rich mixture cloud that auto-ignites before the end of the compression stroke and either ignites the premixed charge or increases the temperature and pressure in the cylinder so that the premixed charge auto-ignites. Embodiments are described where either one injector is used to perform both injection stages or two separate injectors are used, one for each stage. Where two injectors are used the first fuel injector may be a low pressure type for introducing a widely dispersed fuel spray and the second injector may be a high pressure type for introducing a narrow and highly atomised fuel spray. However it is preferred to use one electromagnetic injector for both injections.

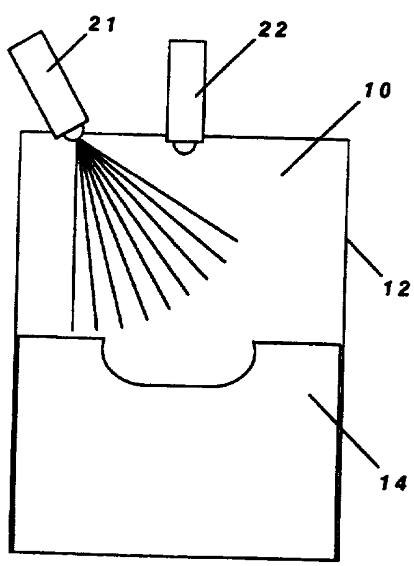

Fig.1

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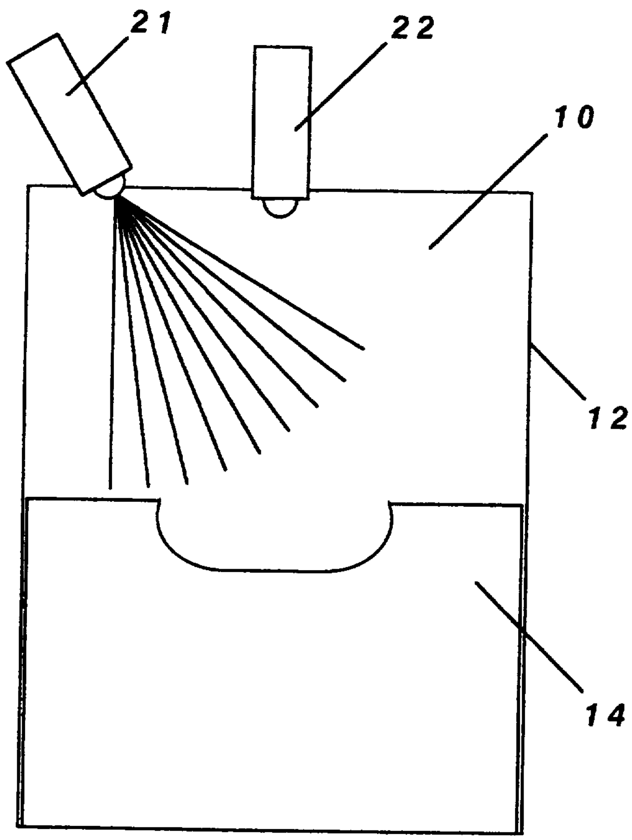

Fig.1

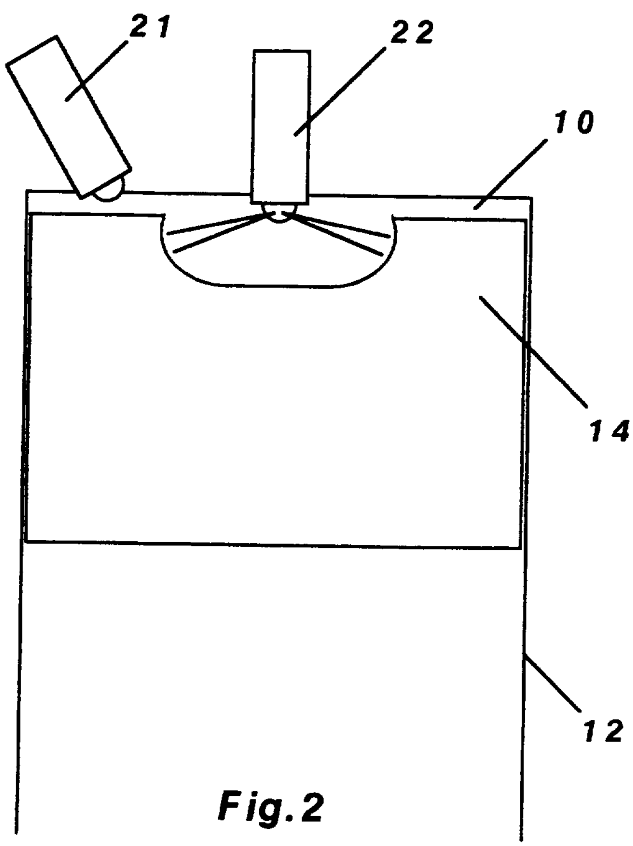

Fig.2

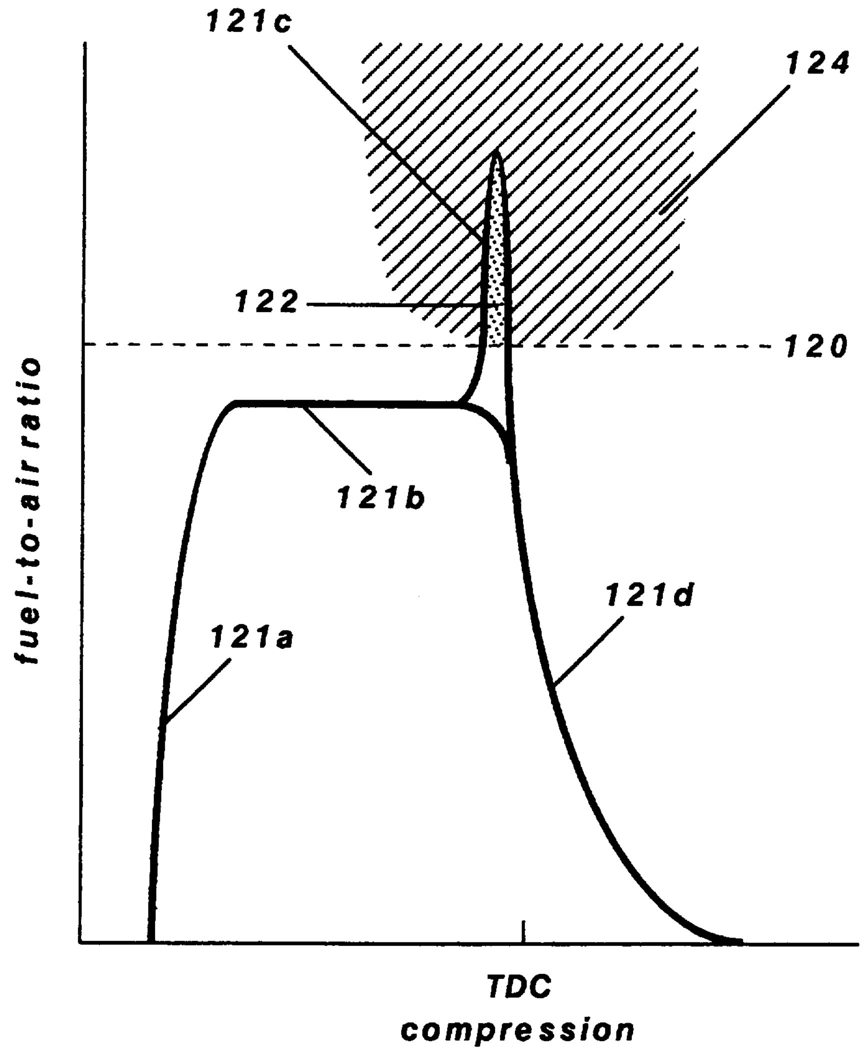
121c
124
122
120
fuel-to-air ratio
121b
121d
121a
TDC
compression

Fig.3

INTERNAL COMBUSTION ENGINE

Field of the invention

The present invention relates to a compression ignition internal combustion engine.

Background of the invention

In a known compression ignition engine having a pilot injection system, a first small quantity of fuel is injected into a substantially pure air charge to form a locally rich cloud which autoignites during the compression stroke. Subsequently a second main quantity of fuel is injected to feed into the ignited cloud and burn as a diffusion flame at a controlled rate determined by the rate of the second injection. Such a system is suitable for engines burning a low cetane fuel where, without the pilot injection, the formation of a substantial quantity of premixed charge because of the long time delay between the start of injection and the instant of ignition would cause severe diesel knock as the premixed charge autoignites. Such undesirable diesel knock can be avoided by a pilot injection which provides an ignition source for the main fuel injection without allowing time for the fuel to form a premixed charge.

A disadvantage of the compression ignition engine in which the fuel burns as a diffusion flame at a rate determined by the rate of injection without forming a sizeable premixed charge is that the NO_x emissions produced are significantly higher than that produced by burning of a premixed charge.

Technical paper SAE 970313 describes a premixed charge compression ignition engine where a premixed charge of gasoline fuel is prepared by injecting the fuel into the intake passage of the engine. In this case, because the gasoline charge cannot itself autoignite in the engine, a small quantity of diesel fuel is injected which autoignites by compression ignition to form a flame kernel, thereby providing an ignition source from which the premixed gasoline charge can burn by normal flame propagation.

This combustion system has been shown to produce lower NO_x emissions than a conventional diesel engine, though in reality this is a gasoline engine even though the ignition is not initiated by a spark plug. In this system, two fuels of distinctly different ignition characteristics are used, the gasoline fuel being easily vaporised but not readily ignited by compression, while the diesel fuel is used specifically for ignition by compression.

Technical paper SAE 971676 describes another premixed charge compression ignition engine where a premixed charge of diesel fuel is prepared by injecting the fuel into a heated intake passage of the engine. The heated fuel is vaporised and mixed homogeneously with the intake air and the whole charge is burnt by compression ignition in the engine. The spontaneous autoignition of a large premixed charge would normally produce severe diesel knock but, in this case, the charge is highly diluted so that the knock can be tolerated. This combustion system has the advantage that the NO_x emissions in the exhaust gases are extremely low, but has the disadvantage that it is difficult to vaporise the diesel fuel in the intake system, requiring a continuous supply of heat and risking accumulation of deposits.

Object of the invention

The present invention aims to induce controlled autoignition of a premixed charge in a diesel engine in order to achieve low NO_x emissions, but in a manner which mitigates at least some of the above disadvantages.

Summary of the invention

According to a first aspect of the present invention, there is provided a compression ignition internal combustion engine burning a diesel fuel and having an engine management system controlling at least one fuel injector for injecting the fuel directly into the combustion chamber of each engine cylinder, comprising means for injecting a metered first quantity of the fuel into the engine cylinder during the intake stroke or early in the compression stroke of the engine cycle to form a premixed charge that is too lean to autoignite at the pressure and temperature prevailing in the combustion chamber during the compression stroke, and means for injecting a second quantity of the fuel into the cylinder later in the compression stroke to form a locally rich mixture cloud that autoignites before the end of the compression stroke, the ignited rich cloud from the second fuel injection thereby providing a flame kernel that serves to ignite the premixed charge from the first fuel injection.

As diesel engine normally run unthrottled and have a substantially constant mass of air in the combustion chamber under all operating conditions, the mixture strength of the premixed charge will vary as a function of engine load.

At medium load, the fuel-to-air ratio of the premixed charge from the first fuel injection is sufficiently rich for the premixed charge to autoignite after the rich cloud from the second fuel injection has autoignited. The autoignition of the premixed charge is caused in this case by the increase in pressure and temperature in the combustion chamber as a result of the autoignition of the second injection.

At lighter loads, the premixed charge will be too lean to autoignite even under the higher temperature and pressure generated by the autoignition of the second injection but it will burn nevertheless by flame propagation from the ignition kernel created by this autoignition.

During idling and very low load conditions, the premixed charge would be too lean to burn, even by flame propagation, but in this case one may dispense with the first injection.

There is a limit to the amount of fuel that can be injected in the first injection if one is to avoid premature autoignition of the premixed charge. This maximum quantity of fuel in the premixed charge corresponds to a load in the medium range. For loads between this medium load and full load, it is necessary to prolong the duration of the second injection rather than the first in order to increase the total quantity of fuel burnt whereupon the first injection is substantially constant and autoignites after the autoignition of the second injection.

The invention therefore achieves complete burning of the premixed charge either by normal flame or by controlled autoignition or by a combination of the two in different parts of the premixed charge. The NO_x produced will vary depending on the mode of combustion of the premixed charge, but will always be lower than that produced in a conventional diesel engine.

In a preferred embodiment of the invention, means are provided for determining the total air charge in the engine cylinder in order that the first quantity of the fuel may be metered to the engine to achieve a desired fuel-to-air ratio in the premixed charge. Such means may be pressure and temperature sensors for measuring the density of the air charge drawn into the engine cylinder, or an air flow meter for measuring the air mass flow into the intake system of the engine.

To ensure that the first injection never autoignites prematurely, it is important to maintain the fuel-to-air ratio in the premixed charge below a predetermined maximum value. The maximum value, below which autoignition cannot occur, may be determined by prior calibration for a given fuel composition, engine compression ratio and temperature of the premixed charge at the beginning of the compression stroke.

It is preferred to use the same fuel injector for the first and the second fuel injections during each engine cycle, this being achievable with an electromagnetic fuel injector supplied with pressurised fuel from a so-called "common rail" diesel fuel injection system.

Alternatively, the first and the second fuel injections may be provided separately by two fuel injectors. In this case, advantage may be taken of the presence of two injectors to produce different fuel spray patterns that are better matched to the respective combustion requirements. For example, the first fuel injector may be a low-pressure gasoline-type fuel injector for introducing a widely dispersed fuel spray into the combustion chamber during the intake stroke to produce a premixed charge. The second fuel injector may be a high-pressure diesel-type fuel injector for introducing a number of narrow and highly atomised fuel sprays into the combustion chamber towards the end of the compression stroke to produce a compact mixture cloud which readily autoignites.

In a second aspect of the present invention, there is provided a method of operating a compression ignition internal combustion engine burning a diesel fuel and having an engine management system controlling at least one fuel injector for injecting the fuel directly into the combustion chamber of each engine cylinder, the method comprising the steps of injecting a metered first quantity of the fuel into the engine cylinder during the intake stroke or early in the compression stroke of the engine cycle to form a premixed charge that is too lean to autoignite at the pressure and temperature prevailing in the combustion chamber during the compression stroke, and injecting a second quantity of the fuel into the cylinder later in the compression stroke to form a locally rich mixture cloud that autoignites before the end of the compression stroke, the ignited rich cloud from the second fuel injection thereby providing a flame kernel that serves to ignite the premixed charge from the first fuel injection.

The present invention achieves progressive control of the engine load, covering the entire load range, by operating with a combination of premixed charge combustion and heterogeneous charge combustion. The ignition timing of the engine cycle is always controlled by the compression ignition timing of the second fuel injection. The NO_x emissions produced will be very low at low and medium loads and will increase to a level similar to that of a conventional diesel engine at higher loads.

All the known methods for increasing engine power and reducing NO_x emissions, such as turbo-charging and exhaust gas recirculation, can be applied in the present invention in the same manner as in a conventional compression ignition engine.

Brief description of the drawings

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is a schematic section of an engine cylinder with the piston near the beginning of the compression stroke,

Figure 2 is a schematic section of the engine cylinder with the piston near the end of the compression stroke, and

Figure 3 is a graph showing the variation with crankshaft angle of the fuel-to-air ratio within the engine cylinder during the compression stroke.

Detailed description of the drawings

Figure 1 shows a schematic section of an engine cylinder 12 with a piston 14 reciprocating along the cylinder and defining a variable combustion chamber 10. Intake and exhaust valves to the combustion chamber 10 are provided but are not shown in this section of the drawing. Two fuel injectors 21, 22 are provided in the combustion chamber 10, the first fuel injector 21 being designed and positioned to disperse fuel uniformly throughout the cylinder charge and the second fuel injector 22 being designed and positioned to form a compact fuel cloud close to the tip of the fuel injector 22.

In Figure 1, the piston 14 is near the beginning of the compression stroke and the first fuel injector 21 is injecting a spray of fuel into the combustion chamber to form a premixed charge. An engine management system (not shown) determines the air charge in the combustion chamber 10 and calculates the quantity of the fuel to be injected by the first fuel injector 21 so that the fuel-to-air ratio of the premixed charge is always leaner than a limit value stored in the engine management system. This ensures that the premixed charge cannot autoignite until additional fuel is injected by the second fuel injector 22 later in the engine cycle.

In Figure 2, the piston 14 is near the end of the compression stroke and the second fuel injector 22 is injecting a number of narrow fuel sprays into a bowl on the top of the piston 14. The engine management system determines the load demand on the engine and sets the quantity of the fuel to be injected by the second fuel injector 22 so that autoignition occurs and diffusion flames follows at or near the narrow fuel sprays. This provides the flame kernel which serves to ignite the premixed charge created by the first fuel injector 21.

In Figure 3, the solid line 121 represents the fuel-to-air ratio within the engine cylinder set by the first and the second fuel injections during the compression stroke. During the first fuel injection near the beginning of the compression stroke, the line 121a climbs as the fuel-to-air ratio increases and reaches a plateau 121b, which is the fuel-to-air ratio of the premixed charge at the end of the first fuel injection.

Also shown in Figure 3 is a dashed line 120 which represents the limit fuel-to-air ratio below which the premixed charge cannot autoignite at the prevailing pressure and temperature in the cylinder during the compression stroke. This limit fuel-to-air ratio 120 is determined by prior calibration and stored in the engine management system for a given fuel composition, engine compression ratio and temperature of the premixed charge at the beginning of the compression stroke.

The plateau 121b of the fuel-to-air ratio of the premixed charge is set in the present invention leaner than the limit fuel-to-air ratio 120 so that the premixed charge can never autoignite until after the second fuel injection occurs later in the compression stroke. When the second injection occurs, the fuel-to-air ratio is increased locally along the line 121c to a value above the limit fuel-to-air ratio 120. The dotted area 122, enclosed by the solid line 121c and the dashed line 120, penetrates into an autoignition region represented by the hatched area 124 lying above the dashed line 120 which indicates that the charge from the second fuel injection will autoignite.

If the fuel-to-air ratio 121b of the premixed charge is sufficiently rich, it will immediately autoignite triggered by the autoignition of the second fuel injection. If the fuel-to-air ratio 121b is too lean to get into autoignition, the premixed charge will still be ignited by the flame kernel created by the autoignition of the second fuel injection and burn by normal flame propagation. In either case the fuel-to-air ratio in the cylinder will drop rapidly to zero as the whole charge is consume, as represented by the solid line 121d.

It will be clear from the foregoing discussion that steps are taken to ensure that the fuel-to-air ratio in the cylinder charge is set at different times in the engine cycle to have clearly defined and distinctly different ignition characteristics such that the actual ignition timing is always controlled by the compression ignition timing of the second fuel injection, and not by premature ignition of the premixed charge which is prevented from getting into autoignition by careful calibration of the first fuel injection. This ability to control the ignition timing directly by the engine management system ensures that every firing cycle is repeatable and that the thermodynamic efficiency of the cycle is optimised.

Once ignition has been initiated by the second fuel injection, it is advantageous for a substantial proportion of the premixed charge to be triggered into autoignition as soon as possible so that the NO_x produced will be as low as possible. Accordingly, the engine management system should set the fuel-to-air ratio of the premixed charge as close as possible to a maximum value bordering on autoignition, but not exceeding the limit value that would cause autoignition.

CLAIMS

1. A compression ignition internal combustion engine burning a diesel fuel and having an engine management system controlling at least one fuel injector for injecting the fuel directly into the combustion chamber of each engine cylinder, comprising means for injecting a metered first quantity of the fuel into the engine cylinder during the intake stroke or early in the compression stroke of the engine cycle to form a premixed charge that is too lean to autoignite at the pressure and temperature prevailing in the combustion chamber during the compression stroke, and means for injecting a second quantity of the fuel into the cylinder later in the compression stroke to form a locally rich mixture cloud that autoignites before the end of the compression stroke, the ignited rich cloud from the second fuel injection thereby providing a flame kernel that serves to ignite the premixed charge from the first fuel injection.

2. A compression ignition internal combustion engine as claimed in claim 1, wherein means are provided for determining the total air charge in the engine cylinder.

3. A compression ignition internal combustion engine as claimed in claim 2, wherein the means for determining the air charge comprise pressure and temperature sensors for measuring the density of the air charge drawn into the engine cylinder.

4. A compression ignition internal combustion engine as claimed in claim 2, wherein the means for determining the air charge comprise an air flow meter for measuring the air mass flow into the intake system of the engine.

5. A compression ignition internal combustion engine as claimed in any preceding claim, wherein the same fuel

injector provides the first and the second fuel injections during each engine cycle.

6. A compression ignition internal combustion engine as claimed in any one of claims 1 to 4, wherein separate fuel injectors provide the first and the second fuel injections respectively.

7. A method of operating a compression ignition internal combustion engine burning a diesel fuel and having an engine management system controlling at least one fuel injector for injecting the fuel directly into the combustion chamber of each engine cylinder, the method comprising the steps of injecting a metered first quantity of the fuel into the engine cylinder during the intake stroke or early in the compression stroke of the engine cycle to form a premixed charge that is too lean to autoignite at the pressure and temperature prevailing in the combustion chamber during the compression stroke, and injecting a second quantity of the fuel into the cylinder later in the compression stroke to form a locally rich mixture cloud that autoignites before the end of the compression stroke, the ignited rich cloud from the second fuel injection thereby providing a flame kernel that serves to ignite the premixed charge from the first fuel injection.

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Application No:	GB 9718439.4	**Examiner:**	David Glover
Claims searched:	1-7	**Date of search:**	26 January 1998

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.P): F1B

Int Cl (Ed.6): F02B 3/06, 3/08, 3/10, 7/00, 7/02, 7/04

Other: Online: WPI

Documents considered to be relevant:

Category	Identity of document and relevant passage		Relevant to claims
A	GB 2277776 A	(Greenhough)	
X	GB 0878278	(Institut Francais Du Petrole) see whole document, particularly page 2 lines 45-63	1 & 5
X	DE 3405558	(Simon)	1 at least

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