



(11) **EP 2 610 469 A1**

(12) **EUROPEAN PATENT APPLICATION**
published in accordance with Art. 153(4) EPC

(43) Date of publication:
03.07.2013 Bulletin 2013/27

(51) Int Cl.:
F02D 41/40 (2006.01) **F02D 41/38** (2006.01)
F02D 45/00 (2006.01)

(21) Application number: **10856411.3**

(86) International application number:
PCT/JP2010/064397

(22) Date of filing: **25.08.2010**

(87) International publication number:
WO 2012/026005 (01.03.2012 Gazette 2012/09)

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

(72) Inventor: **HASEGAWA, Ryo**
Toyota-shi
Aichi 471-8571 (JP)

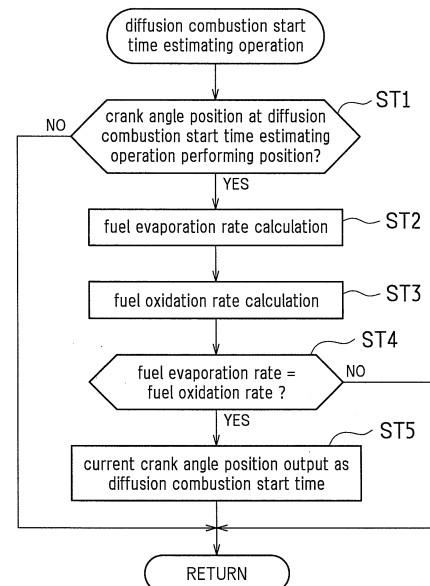
(71) Applicant: **TOYOTA JIDOSHA KABUSHIKI**
KAISHA
Aichi-ken,
471-8571 (JP)

(74) Representative: **TBK**
Bavariaring 4-6
80336 München (DE)

(54) **DEVICE FOR ESTIMATING DIFFUSE COMBUSTION START TIME AND DEVICE FOR CONTROLLING DIFFUSE COMBUSTION START TIME FOR INTERNAL COMBUSTION ENGINE**

(57) The evaporation rate and the oxidation rate of fuel injected into a combustion chamber are calculated, and a diffusion combustion start time is estimated to be the time when this fuel evaporation rate and fuel oxidation rate arrive in a state where the fuel evaporation rate and the fuel oxidation rate match from a state where there is a disparity between the fuel evaporation rate and the fuel oxidation rate. In the case where there is a disparity between this estimated diffusion combustion start time and an appropriate time, a fuel injection pressure adjustment and a swirl control valve opening adjustment are performed such that the diffusion combustion start time matches the appropriate time.

FIG.8



Description

[Technical Field]

[0001] The present invention relates to an apparatus that estimates a diffusion combustion start time and an apparatus that controls a diffusion combustion start time using a result of estimating the diffusion combustion start time of a compression self-ignition internal combustion engine represented by a diesel engine.

[Background Art]

[0002] Combustion of a diesel engine mounted in an automobile or the like is known to be mainly executed in the form of premixed combustion and diffusion combustion. Specifically, when a fuel injection from an injector into a combustion chamber is started, first, a combustible air-fuel mixture is generated due to evaporative diffusion of fuel (an ignition delay period). Next, this combustible air-fuel mixture undergoes self-ignition nearly simultaneously at several locations in the combustion chamber, and combustion progresses rapidly (premixed combustion). A fuel injection is continued, or a fuel injection is performed after a predetermined interval (a fuel injection suspending period) into the combustion chamber whose temperature has been sufficiently increased by this premixed combustion, thereby executing diffusion combustion. Thereafter, since unburnt fuel is still present after the fuel injection is terminated, heat generation continues for a while (an afterburning period).

[0003] Recently, as tougher automobile exhaust emissions regulations (for example, Euro 6) are enforced, keeping the amount of generated toxic substance, such as NO_x, within the regulation limit by appropriately maintaining the combustion start time of an air-fuel mixture is required even in situations where the pressure, temperature, heat transfer conditions, and the like in a combustion chamber change due to environmental changes, operation transients, etc.

[0004] Conventional techniques for controlling the combustion start time of an air-fuel mixture are as proposed in Patent Literatures 1 to 3.

[0005] In Patent Literatures 1 and 2, an ignition time is estimated using an ignition time estimation model that uses parameters such as a fuel injection time, intake oxygen concentration, and engine rotational speed as arguments.

[0006] In Patent Literature 3, the temperature in a cylinder at a fuel ignition time of a main injection is estimated on the assumption that no pilot injection is performed, and the form of pilot injection is set based on this estimated temperature in a cylinder so as to appropriately set main injection fuel ignition.

[Citation List]

[Patent Literature]

[0007]

[PTL 1] JP 2007-92583A

[PTL 2] JP 2008-261312A

[PTL 3] JP 2001-254645A

[Disclosure of Invention]

[Technical Problems]

[0008] As disclosed in the aforementioned patent literatures, conventional techniques for controlling the combustion start time of an air-fuel mixture is to appropriately set a premixed combustion ignition time to stabilize combustion in a combustion chamber.

[0009] However, in the aforementioned premixed combustion, the ignition time and the combustion amount (amount of heat generated) greatly vary depending on the quantity of state, such as the temperature, pressure, and oxygen concentration in a combustion chamber, and therefore it is difficult to appropriately control the ignition time particularly when an environment changes or during operation transients. Accordingly, a diffusion combustion start time, which greatly influences exhaust emissions, may not be obtained appropriately. Therefore, control of the premixed combustion ignition time provides limited controllability in terms of combustion stability and exhaust emissions.

[0010] In particular, in a situation where a fuel injection amount for premixed combustion (so-called a pilot injection amount) is increased or in a situation where an exhaust gas recirculation (EGR) amount is increased (to suppress the amount of NO_x generated by keeping the combustion temperature low) in order to meet tougher exhaust emissions regulations, it is highly likely that, according to the change in quantity of state in a combustion chamber (the aforementioned

change in quantity of state resulting from environmental changes and operation transients), the amount of ignition delay of premixed combustion is greatly changed, and the combustion amount (the amount of heat generated) is greatly changed, and therefore the controllability tends to be increasingly deteriorated. Likewise, in a situation where the ignitability of an air-fuel mixture is deteriorated when an engine operates at high altitudes or when low-cetane fuel is used, it is highly likely that the amount of ignition delay of premixed combustion is greatly changed, and the combustion amount is greatly changed.

[0011] The inventors of the present invention, in view of this point, noted that, compared with controlling the ignition time of premixed combustion, a highly accurate estimation of the start time of diffusion combustion can help greatly improve combustion stability and exhaust emissions.

[0012] It is generally believed that, in diffusion combustion, injection of fuel after the temperature in a combustion chamber has reached an ignitable temperature (for example, 1000 K) causes combustion to start no later than this fuel injection timing. However, actually, even if the temperature in the combustion chamber has reached an ignitable temperature, diffusion combustion does not start until the time when fuel injected into a combustion chamber evaporates and forms a combustible air-fuel mixture. And, this time is changed according to the evaporation rate of fuel. That is, the diffusion combustion start time is influenced by the fuel evaporation rate in a combustion chamber. Therefore, the inventors of the present invention found that it is necessary to estimate a diffusion combustion start time in consideration of this and to control a diffusion combustion start time based thereon.

[0013] The present invention has been achieved in view of the above, and an object thereof is to provide an internal combustion engine diffusion combustion start time estimating apparatus that can highly accurately estimate a diffusion combustion start time in a compression self-igniting internal-combustion engine, and a diffusion combustion start time control apparatus that controls a diffusion combustion start time using the result of estimating the diffusion combustion start time.

[Means for Solving the Problems]

- Principles of Solution -

[0014] A principle of the solution provided by the present invention for achieving the above object calculates the evaporation rate and the oxidation rate of fuel injected into a combustion chamber and estimates that a diffusion combustion start time is a time when this fuel evaporation rate and fuel oxidation rate match, i.e., a time when the fuel oxidation rate reaches the same level as the fuel evaporation rate. In the case where there is a disparity between this estimated diffusion combustion start time and an appropriate time, a control for matching the diffusion combustion start time with the appropriate time (a fuel evaporation rate correction control or the like) is performed.

- Solving Means -

[0015] Specifically, the present invention is directed to a diffusion combustion start time estimating apparatus for a compression self igniting internal combustion engine that estimates a diffusion combustion start time when fuel injected from a fuel injection valve into a combustion chamber after premixed combustion of an air-fuel mixture has started starts diffusion combustion. The diffusion combustion start time estimating apparatus is provided with a fuel evaporation rate calculating means, a fuel oxidation rate calculating means, and a diffusion combustion start time estimating means. The fuel evaporation rate calculating means calculates the evaporation rate of fuel injected from a fuel injection valve into a combustion chamber by performing a correction on a reference evaporation rate, which is specified in advance, according to at least one of the environmental conditions and operating conditions. The fuel oxidation rate calculating means calculates the oxidation rate of fuel forming an air-fuel mixture due to evaporation in the combustion chamber by performing a correction on a reference oxidation rate, which is specified in advance, according to at least one of the environmental conditions and operating conditions. The diffusion combustion start time estimating means estimates that the diffusion combustion start time of the air-fuel mixture is a time when the fuel evaporation rate calculated by the fuel evaporation rate calculating means and the fuel oxidation rate calculated by the fuel oxidation rate calculating means arrive in a state where the fuel evaporation rate and the fuel oxidation rate match from a state where there is a disparity between the fuel evaporation rate and the fuel oxidation rate.

[0016] According to the aforementioned specific configuration, the diffusion combustion start time of an air-fuel mixture is estimated to be a time when the fuel evaporation rate calculated by the fuel evaporation rate calculating means and the fuel oxidation rate calculated by the fuel oxidation rate calculating means arrive in a state where the fuel evaporation rate and the fuel oxidation rate match from a state where there is a disparity between the fuel evaporation rate and the fuel oxidation rate, and it is possible to highly accurately verify whether this estimated diffusion combustion start time is appropriately obtained or not. That is, in the case where the combustion start time of an air-fuel mixture is to be appropriately set by controlling a premixed combustion ignition time as in conventional techniques, the ignition time and the

combustion amount (amount of heat generated) of premixed combustion greatly vary depending on the quantity of state, such as the temperature, pressure, and oxygen concentration in a combustion chamber, and therefore it is difficult to appropriately control the ignition time, and controllability is limited in terms of combustion stability and exhaust emissions. In contrast, according to the present solving means, it is possible to directly estimate a diffusion combustion start time, which greatly influences exhaust emissions, and to perform a correcting operation thereon, thereby improving controllability and enabling exhaust emissions to be improved. In addition, a fuel evaporation rate is calculated by performing a correction according to the environmental conditions and the operating conditions on a reference evaporation rate that is specified in advance, and a fuel oxidation rate is calculated by performing a correction according to the environmental conditions and the operating conditions on a reference oxidation rate that is specified in advance. That is, since it is possible to obtain this reference evaporation rate and reference oxidation rate in an experimental stage of an internal combustion engine, it is not necessary to mount sensors (such as a cylinder pressure sensor) for obtaining this reference evaporation rate and reference oxidation rate in an internal combustion engine (actual machine), and it is thus possibly to simplify the configuration of the internal combustion engine and reduce costs.

[0017] Specifically, in the configuration of the aforementioned fuel evaporation rate calculating means, a fuel evaporation rate is calculated by multiplying by correction coefficients according to the actual pressure and temperature in a cylinder a map value on a steady-state fuel evaporation rate map that shows a fuel evaporation rate at each crank angle and that is prepared on the assumption that the inside of a cylinder is under a reference pressure and at a reference temperature.

[0018] Specifically, in the configuration of the aforementioned fuel oxidation rate calculating means, a fuel oxidation rate is calculated by multiplying by correction coefficients according to the actual pressure, temperature, and oxygen concentration in a cylinder a map value on a steady-state fuel oxidation rate map that shows the fuel oxidation rate at each crank angle and that is prepared on the assumption that the inside of a cylinder is under a reference pressure, at a reference temperature, and at a reference oxygen concentration.

[0019] According to these specific configurations, the actual fuel evaporation rate and fuel oxidation rate can be calculated highly accurately, and a diffusion combustion start time can be estimated highly accurately. Moreover, since a fuel evaporation rate and a fuel oxidation rate can be obtained through relatively simple calculations, e.g., multiplying a map value retrieved from a map by correction coefficients, the timings to calculate these rates can be more frequent (the interval between calculation timings can be shortened), and this also allows a diffusion combustion start time to be estimated highly accurately.

[0020] The aforementioned steady-state fuel oxidation rate map, with the fuel evaporation rate at the time when fuel is injected into a combustion chamber from a fuel injection valve being set to be "0", is prepared by straight line approximation on the assumption that the fuel oxidation rate increases at a constant acceleration from the time when fuel is injected to a fuel oxidation rate that equals to a fuel evaporation rate on the steady-state evaporation rate map at a diffusion combustion start time that is specified based on the change in pressure in a cylinder in a combustion stroke.

[0021] This makes it easy to prepare a steady-state fuel oxidation rate map because a steady-state fuel oxidation rate map is prepared by performing straight line approximation on a fuel oxidation rate whose acceleration actually changes as the crank angle changes (i.e., on the assumption that fuel is oxidized at a constant acceleration).

[0022] A configuration of a diffusion combustion start time control apparatus that controls a diffusion combustion start time estimated by any one internal combustion engine diffusion combustion start time estimating apparatus among the above-described solving means is as follows. That is, a diffusion combustion start time correcting means is provided that calculates a deviation of the estimated diffusion combustion start time relative to the target diffusion combustion start time, and performs based on this deviation a diffusion combustion start time correcting operation such that the estimated diffusion combustion start time matches the target diffusion combustion start time.

[0023] Specifically, in the configuration of this diffusion combustion start time correcting means, the diffusion combustion start time correcting operation is performed by changing a control parameter that varies the evaporation rate of fuel injected into a combustion chamber. Alternatively, the diffusion combustion start time correcting operation is performed by changing, according to the deviation between the target fuel evaporation rate with which the target diffusion combustion start time is obtained and the actual fuel evaporation rate that corresponds to the estimated diffusion combustion start time, a control parameter that varies a fuel evaporation rate.

[0024] More specifically, the control parameter that varies the evaporation rate of fuel injected into a combustion chamber is a fuel injection pressure, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the fuel injection pressure is set. Alternatively, in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the higher the fuel injection pressure is set.

[0025] In the case where the fuel injection pressure is set to be high, the particle size of fuel injected from a fuel injection valve is small, resulting in an increased fuel evaporation rate. Therefore, in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the fuel injection pressure is set so as to increase the fuel evaporation

rate and shift the timing at which the fuel oxidation rate reaches the fuel evaporation rate toward the angle of delay side. Thereby, it is possible to bring the diffusion combustion start time close to the target diffusion combustion start time.

[0026] Another example of the control parameter that varies the evaporation rate of fuel injected into a combustion chamber may be a swirl rate in a combustion chamber. In this case, in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the swirl rate is set. Alternatively, in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the higher the swirl rate is set.

[0027] In the case where the swirl rate is set to be high, the fuel evaporation rate in a combustion chamber is increased. Therefore, in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the swirl rate is set so as to increase the fuel evaporation rate and shift the timing at which the fuel oxidation rate reaches the fuel evaporation rate toward the angle of delay side. Thereby, it is possible to bring the diffusion combustion start time close to the target diffusion combustion start time.

[0028] A specific example of the operation performed in the case where a fuel injection pressure and a swirl rate are concomitantly used as the aforementioned control parameter that varies the evaporation rate of fuel injected into a combustion chamber is as follows: in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the fuel injection pressure is set, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time even when the fuel injection pressure is corrected to the correction limit, the greater the deviation therebetween, the higher the swirl rate is set. Alternatively in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the higher the fuel injection pressure is set, and in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate even when the fuel injection pressure is corrected to the correction limit, the greater the deviation therebetween, the higher the swirl rate is set.

[0029] In this way, the controllable range of a diffusion combustion start time is broadened by making the fuel evaporation rate variable by a plurality of control parameters, and it is thus possible to attain the target diffusion combustion start time.

[0030] The valve timing of an intake valve may be used as the aforementioned control parameter that varies the evaporation rate of fuel injected into a combustion chamber. In this case, in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the farther the valve timing at which an intake valve opens is set toward the angle of delay side, or in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the farther the valve timing at which an intake valve opens is set toward the angle of delay side.

[Effects of the Invention]

[0031] In the present invention, the evaporation rate and the oxidation rate of fuel injected into a combustion chamber are calculated, and a diffusion combustion start time is estimated to be the time when this fuel evaporation rate and fuel oxidation rate match, i.e., the time when the fuel oxidation rate reaches the same level as the fuel evaporation rate. In the case where there is a disparity between this estimated diffusion combustion start time and an appropriate time, a control for matching the diffusion combustion start time with the appropriate time is performed. Therefore, it is possible to directly estimate a diffusion combustion start time, which greatly influences exhaust emissions, and to perform a correcting operation thereon, thereby improving controllability and enabling exhaust emissions to be improved.

[Brief Description of Drawings]

[0032]

[FIG. 1]

FIG. 1 is a schematic configuration diagram showing an engine and the control system thereof according to an embodiment.

[FIG. 2]

FIG. 2 is a cross-sectional diagram showing the combustion chamber of a diesel engine and its surroundings.

[FIG. 3]

FIG. 3 is a block diagram showing the configuration of a control system such as an ECU.

[FIG. 4]

FIG. 4 is a schematic diagram of an intake/exhaust system and the combustion chamber for illustrating an overview of a combustion form in the combustion chamber

[FIG. 5]

FIG. 5 is a cross-sectional diagram showing the combustion chamber and its surroundings during fuel injection.

[FIG. 6]

FIG. 6 is a plan view of the combustion chamber during fuel injection.

[FIG. 7]

FIG. 7 shows waveform charts showing a change in heat generation rate (heat generation amount per unit rotation angle of a crankshaft) and a change in fuel injection rate (fuel injection amount per unit rotation angle of a crankshaft) during an expansion stroke.

[FIG. 8]

FIG. 8 is a flowchart showing a procedure of a diffusion combustion start time estimating operation.

[FIG. 9]

FIG. 9 shows diagrams to depict evaporation states of fuel injected from an injector and the respective regions showing the state of the fuel.

[FIG. 10]

FIG. 10(a) is a chart showing a diffusion combustion start time on a fuel evaporation rate map, and FIG. 10(b) is a chart showing a relationship between a diffusion combustion start time, and a fuel evaporation rate and fuel oxidation rate.

[FIG. 11]

FIG. 11 shows maps for obtaining a correction coefficient for calculating a fuel oxidation rate, FIG. 11(a) is a temperature correction coefficient map, FIG. 11(b) is a pressure correction coefficient map, and FIG. 11(c) is an oxygen concentration correction coefficient map.

[FIG. 12]

FIG. 12(a) is a chart showing a relationship between an engine crank angle and a fuel evaporation rate and fuel oxidation rate in the case where a fuel evaporation rate is lower than a reference evaporation rate, and FIG. 12(b) is a chart showing a relationship between an engine crank angle and a fuel evaporation rate and fuel oxidation rate in the case where a fuel evaporation rate is higher than a reference evaporation rate.

[FIG. 13]

FIG. 13 is a flowchart showing a procedure of a diffusion combustion start time controlling operation.

[FIG. 14]

FIG. 14 is a chart showing a relationship between a diffusion combustion start time and a fuel evaporation rate and fuel oxidation rate in the case where there is a disparity between the actual fuel evaporation rate and the target fuel evaporation rate.

[FIG. 15]

FIG. 15 is an operation amount setting map showing a relationship between a swirl amount and an amount of swirl control valve operation.

[Mode for Carrying Out the Invention]

[0033] An embodiment of the invention is described below with reference to the drawings. In the present embodiment, a case will be described in which the invention is applied to a common rail in-cylinder direct injection multi-cylinder (for example, inline four-cylinder) diesel engine (compression self-igniting internal combustion engine) mounted in an automobile.

- Engine configuration -

[0034] First, the overall configuration of a diesel engine (referred to below as simply the engine) according to the present embodiment will be described. Fig. 1 is a schematic configuration diagram of an engine 1 and a control system of the engine 1 according to the present embodiment. Fig. 2 is a cross-sectional view showing a combustion chamber 3 of the diesel engine and parts in the vicinity of the combustion chamber 3.

[0035] As shown in Fig. 1, the engine 1 according to the present embodiment is configured as a diesel engine system having a fuel supply system 2, combustion chambers 3, an intake system 6, an exhaust system 7, and the like as its main portions.

[0036] The fuel supply system 2 is provided with a supply pump 21, a common rail 22, injectors (fuel injection valves) 23, a cutoff valve 24, a fuel addition valve 26, an engine fuel path 27, an added fuel path 28, and the like.

[0037] The supply pump 21 draws fuel from a fuel tank, and after putting the drawn fuel under high pressure, supplies the fuel to the common rail 22 via the engine fuel path 27. The common rail 22 has a function as an accumulation chamber where the high pressure fuel supplied from the supply pump 21 is held (accumulated) at a specific pressure, and this accumulated fuel is distributed to each injector 23. The injectors 23 are configured from piezo injectors within

which a piezoelectric element (piezo element) is provided, and supply fuel by injection into the combustion chambers 3 by appropriately opening a valve. The details of control of fuel injection from the injectors 23 will be described later.

[0038] Also, the supply pump 21 supplies part of the fuel drawn from the fuel tank to the fuel addition valve 26 via the added fuel path 28. In the added fuel path 28, the cutoff valve 24 is provided in order to stop fuel addition by cutting off the added fuel path 28 during an emergency.

[0039] The fuel addition valve 26 is configured from an electronically controlled opening/closing valve whose valve opening period is controlled with an addition controlling operation by an ECU 100 such that the amount of fuel added to the exhaust system 7 becomes a target addition amount (an addition amount such that exhaust A/F becomes target A/F), or such that a fuel addition timing becomes a specific timing. In other words, a desired amount of fuel is supplied from the fuel addition valve 26 by injection to the exhaust system 7 (to an exhaust manifold 72 from exhaust ports 71) at a suitable timing.

[0040] The intake system 6 is provided with an intake manifold 63 connected to an intake port 15a formed in a cylinder head 15 (see Fig. 2), and an intake pipe 64 that constitutes an intake path is connected to the intake manifold 63. Also, in this intake path, an air cleaner 65, an airflow meter 43, and a throttle valve (intake throttling valve) 62 are disposed in order from the upstream side. The airflow meter 43 outputs an electrical signal according to the amount of air that flows into the intake path via the air cleaner 65.

[0041] Also, in the intake system 6, a swirl control valve 66 is provided in order to vary swirl flow (horizontal swirl flow) in the combustion chambers 3 (see FIG. 2). Specifically, each cylinder is provided with two ports, namely a normal port and a swirl port, as the aforementioned intake port 15a, and the swirl control valve 66, which is constituted by a butterfly valve whose opening is adjustable, is disposed in the normal port 15a shown in FIG. 2. The swirl control valve 66 is linked to an actuator (not shown), and the flow rate of air passing through the normal port 15a can be changed according to the opening degree of the swirl control valve 66, which is adjusted by driving the actuator. The greater the opening of the swirl control valve 66 is, the greater the amount of air that flows from the normal port 15a into the cylinder is. For this reason, swirl generated by the swirl port (not shown in FIG. 2) becomes relatively weak, and a low swirl condition (low swirl rate state) is achieved in the cylinder. On the contrary, the smaller the opening of the swirl control valve 66 is, the lower the amount of air that flows from the normal port 15a into the cylinder is. For this reason, swirl generated by the swirl port does not become relatively strong, and a high swirl condition (high swirl rate state) is achieved in the cylinder.

[0042] The exhaust system 7 is provided with the exhaust manifold 72 connected to the exhaust ports 71 formed in the cylinder head 15, and exhaust pipes 73 and 74 that constitute an exhaust path are connected to the exhaust manifold 72. Also, in this exhaust path a maniverter (exhaust purification apparatus) 77 is disposed that is provided with a NOx storage catalyst (NSR catalyst: NOx storage reduction catalyst) 75 and a diesel particulate-NOx reduction catalyst (DPNR catalyst) 76. Hereinafter, this NSR catalyst 75 and DPNR catalyst 76 will be described.

[0043] The NSR catalyst 75 is a storage reduction NOx catalyst and is composed using, for example, alumina (Al_2O_3) as a support, with, for example, an alkali metal such as potassium (K), sodium (Na), lithium (Li), or cesium (Cs), an alkaline earth element such as barium (Ba) or calcium (Ca), a rare earth element such as lanthanum (La) or yttrium (Y), and a precious metal such as platinum (Pt) supported on this support.

[0044] The NSR catalyst 75, in a state in which a large amount of oxygen is present in exhaust gas, stores NOx, and in a state in which the oxygen concentration in exhaust gas is low and a large amount of reduction component (for example, an unburned component of fuel (HC)) is present, reduces NOx to NO_2 or NO and releases the resulting NO_2 or NO. NOx that has been released as NO_2 or NO is further reduced due to quickly reacting with HC or CO in exhaust gas and becomes N_2 . Also, by reducing NO_2 or NO, HC and CO themselves are oxidized and thus become H_2O and CO_2 . In other words, by suitably adjusting the oxygen concentration or the HC component in exhaust gas introduced into the NSR catalyst 75, it is possible to purify HC, CO, and NOx in the exhaust gas. In the configuration of the present embodiment, adjustment of the oxygen concentration or the HC component in exhaust gas can be performed with an operation to add fuel from the fuel addition valve 26.

[0045] On the other hand, in the DPNR catalyst 76, a NOx storage reduction catalyst is supported on a porous ceramic structure, for example, and PM in exhaust gas is captured while passing through a porous wall. When the air-fuel ratio of the exhaust gas is lean, NOx in the exhaust gas is stored in the NOx storage reduction catalyst, and when the air-fuel ratio is rich, the stored NOx is reduced and released. Furthermore, a catalyst that oxidizes/burns the captured PM (for example, an oxidization catalyst whose main component is a precious metal such as platinum) is supported on the DPNR catalyst 76.

[0046] Here, the combustion chamber 3 of the diesel engine and parts in the vicinity of the combustion chamber 3 will be described with reference to Fig. 2. As shown in Fig. 2, in a cylinder block 11 that constitutes part of the engine, a cylindrical cylinder bore 12 is formed in each cylinder (each of four cylinders), and a piston 13 is housed within each cylinder bore 12 such that the piston 13 can slide in the vertical directions.

[0047] The combustion chamber 3 is formed on the top side of a top face 13a of the piston 13. In other words, the combustion chamber 3 is defined by a lower face of the cylinder head 15 installed on top of the cylinder block 11 via a gasket 14, an inner wall face of the cylinder bore 12, and the top face 13a of the piston 13. A cavity (recess) 13b is

concavely provided in substantially the center of the top face 13a of the piston 13, and this cavity 13b also constitutes part of the combustion chamber 3.

[0048] The shape of this cavity 13b is such that the recess size of its center portion (on a cylinder centerline P) is small, and the recess size is increased toward the peripheral side. That is, as shown in FIG. 2, when the piston 13 is near the compression top dead center, the combustion chamber 3 formed by this cavity 13b is configured such that the combustion chamber is a narrow space having a relatively small volume at the center portion, and the space is gradually increased toward the peripheral side (has an enlarged space).

[0049] A small end 18a of a connecting rod 18 is linked to the piston 13 by a piston pin 13c, and a large end of the connecting rod 18 is linked to a crankshaft that is an engine output shaft. Thus, back and forth movement of the piston 13 within the cylinder bore 12 is transmitted to the crankshaft via the connecting rod 18, and engine output is obtained due to rotation of this crank shaft. Also, a glow plug 19 is disposed facing the combustion chamber 3. The glow plug 19 glows due to the flow of electrical current immediately before the engine 1 is started, and functions as a starting assistance apparatus whereby ignition and combustion are promoted due to part of a fuel spray being blown onto the glow plug.

[0050] In the cylinder head 15, the intake port 15a that introduces air into the combustion chamber 3 and the exhaust port 71 that discharges exhaust gas from the combustion chamber 3 are formed, and an intake valve 16 that opens/closes the intake port 15a and an exhaust valve 17 that opens/closes the exhaust port 71 are disposed. The intake valve 16 and the exhaust valve 17 are disposed facing each other on either side of a cylinder center line P. That is, this engine 1 is configured as a cross flow-type engine. Also, the injector 23 that injects fuel directly into the combustion chamber 3 is installed in the cylinder head 15. The injector 23 is disposed substantially in the center above the combustion chamber 3, in an erect orientation along the cylinder center line P, and injects fuel introduced from the common rail 22 toward the combustion chamber 3 at a specific timing.

[0051] Furthermore, as shown in Fig. 1, the engine 1 is provided with a turbocharger 5. This turbocharger 5 is equipped with a turbine wheel 52 and a compressor wheel 53 that are linked via a turbine shaft 51. The compressor wheel 53 is disposed facing the inside of the intake pipe 64, and the turbine wheel 52 is disposed facing the inside of the exhaust pipe 73. Thus the turbocharger 5 uses exhaust flow (exhaust pressure) received by the turbine wheel 52 to rotate the compressor wheel 53, thereby performing a so-called supercharging operation that increases the intake pressure. In this embodiment, the turbocharger 5 is a variable nozzle-type turbocharger, in which a variable nozzle vane mechanism (not shown) is provided on the turbine wheel 52 side, and by adjusting the opening of this variable nozzle vane mechanism it is possible to adjust the supercharging pressure of the engine 1.

[0052] An intercooler 61 for forcibly cooling intake air heated due to supercharging with the turbocharger 5 is provided in the intake pipe 64 of the intake system 6.

[0053] The throttle valve 62 provided on the downstream side from the intercooler 61 is an electronically controlled opening/closing valve whose opening is capable of stepless adjustment, and has a function to constrict the area of the channel of intake air under specific conditions, and thus adjust (reduce) the amount of intake air supplied.

[0054] Also, the engine 1 is provided with an exhaust gas recirculation path (EGR path) 8 that connects the intake system 6 and the exhaust system 7. The EGR path 8 decreases the combustion temperature by appropriately directing part of the exhaust gas back to the intake system 6 and resupplying that exhaust gas to the combustion chamber 3, thus reducing the amount of NOx generated. Also, provided in the EGR path 8 are an EGR valve 81 that by being opened/closed steplessly under electronic control is capable of freely adjusting the flow rate of exhaust gas that flows through the EGR path 8, and an EGR cooler 82 for cooling exhaust that passes through (recirculates through) the EGR path 8. The EGR device (exhaust gas recirculating device) is configured with this EGR path 8, EGR valve 81, EGR cooler 82, and the like.

- Sensors -

[0055] Various sensors are installed at respective sites of the engine 1, and these sensors output signals related to environmental conditions at the respective sites and the operating state of the engine 1.

[0056] For example, the airflow meter 43 outputs a detection signal according to the flow rate of intake air (the amount of intake air) on the upstream side of the throttle valve 62 within the intake system 6. An intake temperature sensor 49 is disposed in the intake manifold 63 and outputs a detection signal according to the temperature of intake air. An intake pressure sensor 48 is disposed in the intake manifold 63 and outputs a detection signal according to the intake air pressure. An A/F (air-fuel ratio) sensor 44 outputs a detection signal that continuously changes according to the oxygen concentration in exhaust gas on the downstream side of the maniverter 77 of the exhaust system 7. An exhaust temperature sensor 45 likewise outputs a detection signal according to the temperature of exhaust gas (exhaust temperature) on the downstream side of the maniverter 77 of the exhaust system 7. A rail pressure sensor 41 outputs a detection signal according to the pressure of fuel accumulated in the common rail 22. A throttle opening sensor 42 detects the opening of the throttle valve 62.

-ECU-

[0057] As shown in Fig. 3, the ECU 100 is provided with a CPU 101, a ROM 102, a RAM 103, a backup RAM 104, and the like. In the ROM 102, various control programs, maps that are referred to when executing those various control programs, and the like are stored. The CPU 101 executes various computational processes based on the various control programs and maps stored in the ROM 102. The RAM 103 is a memory that temporarily stores data resulting from computation with the CPU 101 or data that has been input from the respective sensors. The backup RAM 104 is a nonvolatile memory that stores that data or the like to be saved when the engine 1 is stopped, for example.

[0058] The CPU 101, the ROM 102, the RAM 103, and the backup RAM 104 are connected to each other via a bus 107, and are connected to an input interface 105 and an output interface 106 via the bus 107.

[0059] The input interface 105 is connected to the rail pressure sensor 41, the throttle opening sensor 42, the airflow meter 43, the A/F sensor 44, the exhaust temperature sensor 45, the intake pressure sensor 48, and the intake temperature sensor 49. Furthermore, the input interface 105 is connected to a water temperature sensor 46 that outputs a detection signal according to the coolant temperature of the engine 1, an accelerator opening sensor 47 that outputs a detection signal according to the amount of accelerator pedal depression, a crank position sensor 40 that outputs a detection signal (pulse) each time the output shaft (crankshaft) of the engine 1 rotates a specific angle, and the like.

[0060] On the other hand, the output interface 106 is connected to the supply pump 21, the injectors 23, the fuel addition valve 26, the throttle valve 62, the swirl control valve 66, the EGR valve 81, and the like. In addition, the output interface 106 is connected to an actuator provided in the variable nozzle vane mechanism of the turbocharger 5 (not shown).

[0061] The ECU 100 executes various types of control of the engine 1 based on output from the various types of sensors described above, calculation values obtained by an arithmetic expression using such output values, or the various types of maps stored in the ROM 102.

[0062] For example, the ECU 100 performs a pilot injection (auxiliary injection) and a main injection as the fuel injection control of the injector 23.

[0063] The aforementioned pilot injection is an operation in which a small amount of fuel is injected from the injector 23 prior to the main injection. This pilot injection is an injection operation for suppressing fuel ignition delay in the main injection and for leading to stable diffusion combustion, and is also called an auxiliary injection. Also, the pilot injection in the present embodiment not only has the function to slow the initial combustion rate in the above-described main injection, but also has the preheating function to increase the temperature in a cylinder. That is, after this pilot injection is performed, a fuel injection is suspended, and the temperature of compressed gas (the temperature in a cylinder) is sufficiently increased until the main injection is started so as to reach the self-ignition temperature of fuel (for example, 1000 K), and thereby favorable ignitability of fuel injected in the main injection is secured.

[0064] The aforementioned main injection is an injection operation for generating torque of the engine 1 (operation of supplying fuel for torque generation). The injection amount in this main injection is basically determined such that the required torque is obtained determined according to operating conditions such as engine speed, accelerator operation amount, coolant temperature, and intake air temperature. For example, the greater the engine speed (engine speed calculated based on the detection value from the crank position sensor 40) or the greater the accelerator operation amount (the accelerator pedal depression amount detected by the accelerator opening sensor 47) (i.e., the greater the accelerator opening degree), the greater the resulting torque requirement value of the engine 1, and the greater the fuel injection amount in the main injection is accordingly set.

[0065] In addition to the above-described pilot injection and main injection, an after-injection and a post-injection are performed as necessary. The after-injection is an injection operation for increasing the exhaust gas temperature. Specifically, the after-injection is executed at a timing such that the majority of the combustion energy of supplied fuel is obtained as exhaust heat energy instead of being converted into the torque of engine 1. The post-injection is an injection operation for increasing the temperature of the maniverter 77 by directly introducing fuel into the exhaust system 7. For example, when the amount of PM captured by and deposited in the DPNR catalyst 76 has exceeded a specific amount (for example, indicated from detection of a before/after pressure difference of the maniverter 77), the post-injection is executed.

[0066] The ECU 100 also controls the opening of the EGR valve 81 in accordance with the operating state of the engine 1 to adjust the amount of exhaust gas recirculated towards the intake manifold 63 (EGR amount). The EGR amount is set in accordance with an EGR map that is stored in the ROM 102 in advance. Specifically, this EGR map is a map for determining the EGR amount (EGR rate) using the engine speed and the engine load as parameters. Note that this EGR map is created in advance through experimentation, simulation, or the like. In other words, the EGR amount (opening of the EGR valve 81) is obtained by applying, to the EGR map, the engine speed calculated based on the detection value from the crank position sensor 40 and the opening of the throttle valve 62 (corresponding to the engine load) detected by the aforementioned throttle opening sensor 42.

[0067] The ECU 100 furthermore executes opening degree control on the swirl control valve 66. This opening control

executed on the swirl control valve 66 is performed so as to change the amount of circumferential movement in a cylinder per unit time (or per unit of crank rotation angle) of a spray of fuel injected into the combustion chamber 3.

[0068] Also, as will be described later, the opening of the swirl control valve 66 is changed along with the execution of the fuel evaporation rate control as well. Details of the control of the opening of the swirl control valve 66 in this fuel evaporation rate control will be described later.

- Fuel injection pressure -

[0069] The fuel injection pressure when performing a fuel injection is determined based on the internal pressure of the common rail 22. In regard to the internal pressure of the common rail, normally, the higher the engine load and the greater the engine speed, the greater the target value for the pressure of fuel supplied from the common rail 22 to the injectors 23 (i.e., the target rail pressure). In other words, when the engine load is high, a large amount of air is drawn into the combustion chamber 3, making it necessary to inject a large amount of fuel into the combustion chamber 3 from the injectors 23, and therefore the pressure of injection from the injectors 23 needs to be high. Also, when the engine speed is high, the period during which injection is possible is short, making it necessary to inject a large amount of fuel per unit time, and therefore the pressure of injection from the injectors 23 needs to be high. In this way, the target rail pressure is normally set based on the engine load and the engine speed. This target rail pressure is set according to a fuel pressure setting map stored in, for example, the ROM 102. That is, determining a fuel pressure according to this fuel pressure setting map allows the valve opening period (injection rate waveform) of the injector 23 to be controlled, and it is thus possible to specify the fuel injection amount during that valve opening period.

[0070] As will be described later, the target rail pressure is changed according to the target fuel evaporation rate when the fuel evaporation rate control is performed. Details of a target rail pressure changing operation in this fuel evaporation rate control will be described later.

[0071] In this embodiment, the fuel pressure is adjusted according to the engine load or the like so as to be between 30 MPa and 200 MPa. That is, as for a fuel pressure control range, the lower limit is 30 MPa, and the upper limit is 200 MPa.

[0072] The optimum values of fuel injection parameters for the aforementioned pilot injection, main injection, and the like are different according to the temperature conditions of the engine 1, intake air, and the like.

[0073] For example, the ECU 100 adjusts the amount of fuel discharged by the supply pump 21 such that the common rail pressure becomes the same as the target rail pressure set based on the engine operating state, i.e., such that the fuel injection pressure matches the target injection pressure. Also, the ECU 100 determines the amount of fuel to be injected and the form of fuel injection based on the engine operating state. Specifically, the ECU 100 calculates the engine speed based on the value detected by the crank position sensor 40 and obtains the amount of accelerator pedal depression (accelerator opening) based on the value detected by the accelerator opening sensor 47, and then determines the total amount of fuel to be injected (the sum of the injection amount in pilot injection and the injection amount in main injection) based on the engine speed and the accelerator opening.

- Brief description of combustion forms -

[0074] Next is a brief description of forms of combustion within the combustion chamber 3 in the engine 1 of the present embodiment.

[0075] FIG. 4 schematically shows how gas (air) is drawn into one of the cylinders of the engine 1 through the intake manifold 63 and the intake port 15a, combustion is performed using fuel injected from the injector 23 into the combustion chamber 3, and the combusted gas is discharged to the exhaust manifold 72 via the exhaust port 71.

[0076] As shown in FIG. 4, the gas drawn into the cylinder includes fresh air drawn in from the intake pipe 64 through the throttle valve 62, and EGR gas drawn in from the EGR path 8 in the case where the EGR valve 81 has been opened. The proportion (i.e., the EGR rate) of the amount (mass) of EGR gas drawn in to the sum of the amount (mass) of fresh air drawn in and the EGR gas amount changes according to the opening of the EGR valve 81, which is appropriately controlled by the aforementioned ECU 100 in accordance with the operating state.

[0077] In this way, the fresh air and the EGR gas drawn into the cylinder is in-cylinder gas that is drawn into the cylinder via the intake valve 16 that is open in the intake stroke, along with the descent of the piston 13 (not shown in FIG. 4). Due to the intake valve 16 closing at the valve closing time, which is determined according to the operating state of the engine 1, the in-cylinder gas is sealed inside the cylinder (an in-cylinder gas confined state), and is compressed along with the ascent of the piston 13 in the subsequent compression stroke. When the piston 13 then reaches the vicinity of top dead center, fuel is injected directly into the combustion chamber 3 by the injector 23 being opened for only a predetermined time according to the injection amount control executed by the ECU 100 described above. Specifically, the aforementioned pilot injection is performed before the piston 13 reaches top dead center, a fuel injection is suspended, and after a predetermined interval, the aforementioned main injection is performed when the piston 13 reaches the vicinity of top dead center.

[0078] FIG. 5 is a cross-sectional diagram showing the combustion chamber 3 and its surroundings during this fuel injection, and FIG. 6 is a plan view (diagram showing the upper face of the piston 13) of the combustion chamber 3 during this fuel injection. As shown in FIG. 6, the injector 23 of the engine 1 of the present embodiment is provided with eight holes at equal intervals along the circumferential direction, and fuel is injected from the holes in a uniform manner.

[0079] Sprays A of fuel injected from each of the holes disperse in a substantially conical manner. Also, since the injection (in particular, main injection) of fuel from the holes is performed at the point in time when the piston 13 reaches the vicinity of compression top dead center, the fuel sprays A disperse inside the cavity 13b as shown in FIG. 5.

[0080] In this way, the sprays A of fuel injected from the holes formed in the injector 23 form an air-fuel mixture as they mix with in-cylinder gas over time, and then respectively disperse in a conical manner inside the cylinder and combust due to self-ignition. In other words, the fuel sprays A each form a substantially conical combustion field along with in-cylinder gas, and combustion respectively starts in each combustion field (combustion fields at eight places in the present embodiment).

[0081] The energy generated by this combustion then becomes kinetic energy for pressing the piston 13 down toward bottom dead center (energy that is to serve as engine output), thermal energy for raising the temperature in the combustion chamber 3, and thermal energy that is dissipated to the outside (e.g., coolant) via the cylinder block 11 and the cylinder head 15.

[0082] The combusted in-cylinder gas then becomes exhaust gas that is discharged to the exhaust port 71 and the exhaust manifold 72 via the exhaust valve 17 that opens in the exhaust stroke, along with the ascent of the piston 13.

- Heat generation rate waveform -

[0083] In the diesel engine 1, it is important to simultaneously satisfy demands such as improving exhaust emissions by decreasing the amount of NOx generated, reducing combustion noise during a combustion stroke, and ensuring sufficient engine torque. As a technique to concurrently satisfy such demands, appropriately controlling the changing state of the heat generation rate (changing state expressed as a heat generation rate waveform) in a cylinder during a combustion stroke is effective.

[0084] In the upper waveform presented in FIG. 7, the horizontal axis represents the crank angle, the vertical axis represents the heat generation rate, and FIG. 7 shows an ideal heat generation rate waveform regarding combustion of fuel injected in both the pilot injection and the main injection (a heat generation rate waveform that can be used to comply with strict exhaust emissions regulations as represented by Euro 6). In the waveforms, "TDC" indicates a crank angle position corresponding to the compression top dead center of the piston 13. The lower waveform presented in FIG. 7 shows an injection rate (fuel injection amount per unit rotation angle of a crankshaft) waveform of fuel injected from the injector 23.

[0085] As shown in this heat generation rate waveform and fuel injection rate waveform, in this embodiment, to comply with strict exhaust emissions regulations, the premixed combustion amount is increased by setting a relatively large pilot injection amount (for example, the pilot injection amount is set to be about 30% of the total fuel injection amount, which is a sum of the pilot injection amount and the main injection amount), and the combustion temperature is kept low by setting a high EGR, thus enabling the NOx generation amount to be suppressed.

[0086] For easier understanding of a change in heat generation rate, the heat generation rate waveform shown in FIG. 7 is of a case where combustion (diffusion combustion) of fuel injected in a main injection is started at the compression top dead center (TDC) of the piston 13. The dashed double-dotted line in FIG. 7 shows part of a change in heat generation rate by combustion of fuel injected in a pilot injection (the latter half of the heat generation rate waveform of a pilot injection). The dashed-dotted line shows part of a change in heat generation rate by combustion of fuel injected in a main injection (the first half of the heat generation rate waveform of a main injection). That is, the terminal portion of the heat generation rate waveform indicated by this dashed-dotted line (the intersection with the crank angle axis at which the heat generation rate is "0") is the diffusion combustion start time.

[0087] In the heat generation rate waveform shown in FIG. 7, the heat generation rate reaches its maximum value (peak value) at a specific piston position after the compression top dead center (e.g., a point 10° after the compression top dead center (10° ATDC)) of the piston 13, and furthermore, combustion of fuel injected in a main injection ends at another specific piston position after the compression top dead center (e.g., a point 25° after the compression top dead center (25° ATDC)). By causing combustion of an air-fuel mixture to be performed in this kind of heat production rate changing state, the engine 1 can be operated with high thermal efficiency. The values are not limited to those given above and can be suitably set.

[0088] Combustion of fuel injected in the aforementioned pilot injection exhibits a specific heat generation rate (for example, 30 [J/° CA]) at the compression top dead center (TDC) of the piston 13, and thereby, the temperature in a combustion chamber is at the ignitable temperature of an air-fuel mixture (for example, 1000 K) or greater at the injection timing of the main injection, and stable diffusion combustion of fuel injected in the main injection is performed. The values

are not limited to those given above. There may be a case where a plurality of pilot injections are performed, and in such a case, the temperature in a cylinder is increased even higher so that ignition of fuel injected in a main injection can be well-ensured.

[0089] As described above, sufficient preheating in a cylinder is performed in this embodiment by a pilot injection. Due to this preheating, in the case where a main injection is started, fuel injected in this main injection undergoes thermal decomposition after the injection thereof by being exposed to a temperature environment having a temperature no less than the self-igniting temperature, thus initiating diffusion combustion.

- Diffusion combustion start time estimating operation -

[0090] Next, a diffusion combustion start time estimating operation, which is a feature of this embodiment, will now be described. This diffusion combustion start time estimating operation estimates a timing (a crankshaft rotation angle position at that timing) at which the state of combustion of an air-fuel mixture generated by fuel injected in the aforementioned main injection shifts from a so-called low-temperature oxidation reaction combustion state (a combustion state in which the fuel oxidation rate is lower than the fuel evaporation rate) to a diffusion combustion state.

[0091] An outline of the procedure of this diffusion combustion start time estimating operation will now be described using the flowchart of FIG. 8. The diffusion combustion start time estimating operation shown in this flowchart is performed when a main injection in a cylinder that has reached a combustion stroke is started, and the routine of FIG. 8 is repeated until diffusion combustion start time information in that cylinder is obtained (until diffusion combustion start time estimation is completed) and is suspended when diffusion combustion start time information is obtained (a diffusion combustion start time estimating operation on the subject cylinder is terminated). Thereafter, when a main injection in a next cylinder that has reached a combustion stroke is started, a diffusion combustion start time estimation on that cylinder is performed, the aforementioned routine is repeated again until diffusion combustion start time information is obtained, and a diffusion combustion start time estimating operation on that cylinder is terminated when diffusion combustion start time information is obtained. Such an operation is sequentially performed on cylinders that have reached a combustion stroke, and thus a diffusion combustion start time is estimated for every combustion stroke. Hereinbelow, a specific operational procedure will now be described.

[0092] First, as a crankshaft rotates, whether or not the crank angle position (crank angle position, with "0° CA" being a crank angle corresponding to the compression top dead center of the piston 13) has reached a position where a diffusion combustion start time estimating operation is performed is determined (step ST1). This performing position is set at, for example, a crank angle position at which a main injection is performed from the injector 23 and at crank angle positions reached every time the crankshaft moves a specific rotation angle (for example, 1° CA) from that crank angle position. That is, a YES is given in the step ST1 every time the crankshaft moves a specific rotation angle after a main injection is performed and this routine is repeated until diffusion combustion start time information is obtained (until a YES is given in step ST4, which will be described later, and a diffusion combustion start time is estimated). Specifically, in an operation for determining whether the crank angle position has reached the aforementioned diffusion combustion start time estimating operation performing position or not, a time necessary for the crankshaft to move a specific rotation angle (for example, 1° CA) is calculated based on the current engine speed (an engine speed obtained immediately before a main injection is performed), a YES is given in the step ST1 every time that calculated time lapses, and a diffusion combustion start time estimating operation, which will be described below, is performed. Note that the operation for determining whether the crank angle position has arrived at a diffusion combustion start time estimating operation performing position or not is not limited to this.

[0093] When a YES is given in the step ST1 due to the crank angle position reaching the diffusion combustion start time estimating operation performing position, the procedure advances to step ST2 and the fuel evaporation rate at that time is calculated (step ST2). This fuel evaporation rate is an amount (mass) of injected combustion evaporated per unit time. Details of a fuel evaporation rate calculating operation will be described later.

[0094] Also, the oxidation rate of this evaporated fuel is calculated (step ST3). This oxidation rate is a rate of combustion due to a chemical reaction (oxidation reaction) between a fuel spray evaporated in the combustion chamber 3 and oxygen present in the combustion chamber 3, and is the amount (mass) of fuel that has undergone an oxidation reaction (contributing to combustion) per unit time. Details of a fuel oxidation rate calculating operation will also be described later.

[0095] Whether the fuel evaporation rate and the fuel oxidation rate calculated in such manners match or not is determined (step ST4). Fuel injected into the combustion chamber 3 receives heat of the combustion chamber 3 and evaporates and then undergoes combustion (oxidation reaction), and therefore during an early period after fuel injection from the injector 23, the fuel oxidation rate is lower than the fuel evaporation rate. Then, fuel evaporation progresses, and evaporated fuel is exposed to a high-temperature environment, and thus the oxidation rate is rapidly increased. That is, the oxidation rate approaches the evaporation rate. In the case where the oxidation rate does not yet match the evaporation rate and a NO is given in the step ST4, thus determining that diffusion combustion of an air-fuel mixture has not been started, the procedure advances directly to RETURN. That is, the combustion state of fuel injected in a

main injection is determined as being the aforementioned low-temperature oxidation reaction combustion state, and the procedure advances to RETURN. In this case, until the crank angle reaches the next diffusion combustion start time estimating operation performing position (until a YES is given in the step ST1), a diffusion combustion start time estimating operation is suspended (a diffusion combustion start time estimating operation is suspended during a period when the step ST1 gives a NO).

[0096] The above-described fuel evaporation rate calculating operation (step ST2), fuel oxidation rate calculating operation (step ST3), and fuel evaporation rate and fuel oxidation rate comparing operation (step ST4) are performed repetitively at specific intervals (in this embodiment, every time a crankshaft moves a specific rotation angle) until the oxidation rate and the evaporation rate match.

[0097] In the case where the oxidation rate and the evaporation rate match and a YES is given in the step ST4, it is judged that diffusion combustion of an air-fuel mixture has started, and the procedure advances to step ST5, and the current crank angle position is output as a diffusion combustion start time, this meaning that an estimation of the diffusion combustion start time of this combustion stroke is accomplished.

[0098] Hereinafter, the state of fuel evaporation and oxidation in the combustion chamber 3 will now be described using FIG. 9. FIG. 9 shows schematic diagrams for depicting evaporated states of fuel injected from the injector 23 and changes in the respective regions showing the state of the fuel. Here, a description is provided of fuel injected from one nozzle of the injector 23. Among the respective regions showing the state of fuel presented in FIG. 9, the regions indicated by a solid line (regions inside the solid line) are droplet regions where fuel droplets are present (fuel droplets are present in air), and the regions indicated by a broken line (regions between the solid line and the broken line) are evaporated fuel regions where evaporated fuel evaporated from the aforementioned droplet regions is present (air and evaporated fuel are both present). When the temperature of these evaporated fuel regions reaches a specific ignitable temperature (for example, 1000 K), diffusion combustion of an air-fuel mixture in these regions starts.

[0099] First, at the start of fuel injection shown in FIG. 9(a), substantially the entire fuel injection region is the aforementioned droplet region. The temperature of this droplet region is about, for example, 350 K, and this region is in a state in which diffusion combustion is not yet performed.

[0100] Fuel injections are performed, and when the state shown in FIG. 9(b) is reached, part of droplet fuel in the droplet region (droplet fuel present along the peripheral portion of the droplet region) receives heat in a cylinder and starts evaporating, thus generating an evaporated fuel region on the outside of this droplet region. Although the temperature of the evaporated fuel region in this state is about, for example, 600 K, and diffusion combustion is not yet performed, combustion by a so-called low-temperature oxidation reaction starts, thus contributing to an increase of the reaction field temperature. In this state, although the evaporation rate of fuel (mass per unit time of fuel that moves into the evaporated fuel region from the droplet region) is relatively high, the oxidation rate thereof (mass per unit time of fuel that contributes to combustion in the evaporated fuel region) is lower than the evaporation rate.

[0101] Fuel injections are further performed, and when the state shown in FIG. 9(c) is reached, evaporation of droplet fuel present near the peripheral portion of the droplet region progresses, and combustion due to the aforementioned low-temperature oxidation reaction also progresses, and therefore the evaporation fuel region generated on the outside of this droplet region expands, and the temperature of the evaporated fuel region in this state reaches, for example, 1000 K. Thereby, fuel (air-fuel mixture) present in this evaporated fuel region starts diffusion combustion. In this state, the fuel oxidation rate matches the fuel evaporation rate. That is, in this state, the mass per unit time of fuel that moves into the evaporated fuel region from the droplet region matches the mass per unit time of fuel that contributes to combustion among all the fuel present in the evaporated fuel region.

[0102] Hereinafter, the fuel evaporation rate calculating operation, the fuel oxidation rate calculating operation, and the fuel evaporation rate and fuel oxidation rate comparing operation performed in the above-described diffusion combustion start time estimating operation will now be specifically described.

(Fuel evaporation rate calculating operation)

[0103] First, the fuel evaporation rate calculating operation (the operation in the step ST2 of the flowchart shown in FIG. 8) is described. This fuel evaporation rate calculating operation is performed by reading the map value of the current crank angle position on a steady-state fuel evaporation rate map prepared in advance (a reference evaporation rate as referred to herein), and multiplying the map value by correction coefficients according to the environmental conditions, operating conditions, or the like of the engine 1 (a fuel evaporation rate calculating operation by the fuel evaporation rate calculating means).

[0104] Specifically, for example, by an experiment with a performance testing device on the engine 1, a fuel evaporation rate (dm_v/dt) is calculated according to the following formulas (1) and (2) at every crank angle of the engine 1 in a combustion stroke (for example, every time the crank angle moves 1° CA), and the calculated fuel evaporation rate at each crank angle is mapped to prepare a steady-state fuel evaporation rate map. The steady-state fuel evaporation rate map prepared here is a steady-state map prepared using the pressure and the temperature in a cylinder that are specified

in advance as a reference pressure and a reference temperature, respectively (a map that shows a fuel evaporation rate at each crank angle). The values of this reference pressure and reference temperature can be suitably set. Multiplication by correction coefficients according to the environmental conditions, operating conditions, or the like of the engine 1 (a correction coefficient according to the pressure in a cylinder actually measured or estimated, and a correction coefficient according to the temperature in a cylinder actually measured or estimated) is performed on this steady-state fuel evaporation rate map to calculate the evaporation rate at each crank angle (formula (3) below). As these correction coefficients, in addition to the aforementioned correction coefficients according to the pressure in a cylinder and the aforementioned correction coefficient according to the temperature in a cylinder, a correction coefficient according to the actually measured engine speed may be used.

[0105] [Formula 1]

$$\frac{dm_v}{dt} = -2\pi \frac{k}{c_p} D_d (1 + 0.23 \text{Re}^{0.5}) \ln \left[1 + \frac{c_p (T - T_d)}{h_{fg}} \right] \quad (1)$$

k : thermal conductivity
 c_p : gas specific heat
 D_d : particle size of droplet
 Re : Reynolds number
 T : gas specific heat
 T_d : particle size temperature
 h_{fg} : latent heat of vaporization

[0106] [Formula 2]

$$\text{Re} = \left(V_{sw}^2 + V_{sq}^2 \right)^{\frac{1}{2}} \times A / \text{kinematic viscosity coefficient} \quad (2)$$

V_{sw} : swirl rate
 V_{sq} : squish rate
 A : constant

[0107]

Fuel evaporation rate = Map value on steady-state fuel evaporation rate map
 × In-cylinder pressure correction coefficient
 × In-cylinder temperature correction coefficient (3)

The swirl rate V_{sw} and the squish rate V_{sq} in the formula (2) above are values determined according to the engine shape (in particular, the shape of the combustion chamber 3) and the engine speed. The swirl rate V_{sw} in this case is, for example, a swirl rate at the circumferential portion in the combustion chamber 3. Constant A is a value determined in advance by an experiment or the like according to the type of the engine 1. Also, the kinetic viscosity coefficient of an air-fuel mixture is a value dependent on the temperature.

[0108] More specifically, a steady-state fuel evaporation rate map prepared as described above is stored in the ROM102 in advance, and an evaporation rate value obtained from the aforementioned steady-state evaporation rate map is multiplied by correction coefficients according to the environmental conditions, operating conditions, or the like (a correction coefficient set according to the pressure and the temperature in a cylinder) in an actual operation of an automobile to calculate the fuel evaporation rate at the current crank angle position (the operation in the aforementioned step ST2).

[0109] The aforementioned in-cylinder pressure correction coefficient and in-cylinder temperature correction coefficient are obtained from a two-dimensional map prepared in advance by conducting an experiment or a simulation or the like.

(Calculation of fuel oxidation rate)

[0110] Next, fuel oxidation rate calculation (the operation in the step ST3 of the flowchart shown in FIG. 8) is described. This fuel oxidation rate calculating operation is performed at the same timing as the above-described fuel evaporation rate calculating operation. This fuel oxidation rate calculating operation is performed by reading the map value of the current crank angle position on a steady-state fuel oxidation map prepared in advance (a reference oxidation rate as referred to herein), and multiplying the map value by correction coefficients according to the environmental conditions, operating conditions, or the like of the engine 1 (formula (4) below: a fuel oxidation rate calculating operation by the fuel oxidation rate calculating means).

[0111]

$$\begin{aligned} \text{Fuel oxidation rate} = & \text{Map value on steady-state fuel oxidation rate map} \\ & \times \text{In-cylinder temperature correction coefficient} \\ & \times \text{In-cylinder pressure correction coefficient} \\ & \times \text{In-cylinder oxygen concentration correction coefficient} \quad (4) \end{aligned}$$

A technique for preparing the aforementioned steady-state oxidation rate map for obtaining the aforementioned steady-state oxidation rate map value at each crank angle is described below. In an experiment with an engine performance testing device, the pressure in a cylinder in a combustion process in a combustion chamber is measured with a pressure sensor (pressure indicator), and the indicated change in pressure in a cylinder is regarded as a change in heat generation rate in the combustion chamber 3. It is thus possible to obtain a heat generation rate waveform as shown with a solid line in FIG. 7. The combustion waveform (diffusion combustion waveform) of fuel injected in a main injection is obtained as indicated by the dashed-dotted line in the figure, and the starting point (the intersection with the crank angle axis) of this waveform (diffusion combustion waveform) is defined as a diffusion combustion start time.

[0112] Then, as shown in FIG. 10(a), a diffusion combustion start time (a fuel evaporation rate at a diffusion combustion start time) on the steady-state evaporation rate map obtained by the aforementioned fuel evaporation rate calculating operation is obtained (point X in FIG. 10(a)), and as shown in FIG. 10(b), a straight line that connects the fuel evaporation rate at this diffusion combustion start time (point X) and the point where the fuel oxidation rate is "0" (point Y in FIG. 10(a)) is regarded as a steady-state oxidation rate map (see FIG. 10(b)). Actually, although the fuel oxidation rate is a cubic curve as indicated by a dashed line in FIG. 10(b), a fuel oxidation rate after straight-line approximation is obtained as a steady-state oxidation rate map.

[0113] The relationship between the steady-state oxidation rate map obtained in this manner and the heat generation rate waveform shown in FIG. 7 is described as follows. The main injection start timing (on the angle of advance side relative to the compression top dead center (TDC) of the piston 13 in FIG. 7) corresponds to point Y in FIG. 10, and the diffusion combustion start time (the compression top dead center (TDC)) of the piston 13 in FIG. 7) corresponds to point X in FIG. 10.

[0114] A value on the steady-state oxidation rate map thus specified is multiplied by each of the in-cylinder temperature correction coefficient, in-cylinder pressure correction coefficient, and in-cylinder oxygen concentration correction coefficient, thus giving an oxidation rate at each crank angle. This in-cylinder temperature correction coefficient, in-cylinder pressure correction coefficient, and in-cylinder oxygen concentration correction coefficient are obtained according to the correction maps shown in FIG. 11. FIG. 11(a) is an in-cylinder temperature correction map, and in the case where the map value at the aforementioned reference temperature is A1 and the map value at the actual temperature in a cylinder is A2, the in-cylinder temperature correction coefficient is "A2/A1." FIG. 11(b) is an in-cylinder pressure correction map, and in the case where the map value at the aforementioned reference pressure is B1 and the map value at the actual pressure in a cylinder is B2, the in-cylinder pressure correction coefficient is "B2/B1." FIG. 11(c) is an in-cylinder oxygen concentration correction map, and in the case where the map value at the reference oxygen concentration specified in advance is C1 and the map value at the actual oxygen concentration in a cylinder is C2, the in-cylinder oxygen concentration correction coefficient is "C2/C1."

[0115] That is, an steady-state oxidation rate map showing an oxidation rate change in a reference state is prepared based on a diffusion combustion start time obtained through an experiment and a steady-state evaporation rate map obtained by the aforementioned fuel evaporation rate calculating operation, and this is corrected by being multiplied by the respective correction coefficients (in-cylinder temperature correction coefficient, in-cylinder pressure correction coefficient, and in-cylinder oxygen concentration correction coefficient) to calculate the fuel oxidation rate at the current crank angle position (the operation in the aforementioned step ST3).

(Estimation of diffusion combustion start time)

[0116] As stated above, the diffusion combustion start time is defined as a time when the aforementioned fuel oxidation rate matches the fuel evaporation rate (when the fuel oxidation rate reaches the same level as the fuel evaporation rate). Therefore, the above-described fuel evaporation rate at each crank angle is compared with the corresponding fuel oxidation rate at each crank angle, and in the case where the fuel oxidation rate is still lower than the fuel evaporation rate, it is judged that diffusion combustion is not started, and when the fuel oxidation rate matches the fuel evaporation rate (at the crank angle position when they match), it is judged that diffusion combustion is started (a diffusion combustion start time estimating operation by the diffusion combustion start time estimating means). In this case, a main injection start timing (a crank angle position at which a main injection is started) is stored in advance, a crank rotation angle from that crank angle position to the estimated position of a diffusion combustion start time is obtained, and thereby the crank angle position at the diffusion combustion start time is calculated.

[0117] More specifically, relative to the fuel evaporation rate on the steady-state evaporation rate map obtained by the above-described fuel evaporation rate calculating operation, the actual fuel evaporation rate is changed to the extent of correction made based on the pressure and the temperature in a cylinder. Likewise, relative to the fuel oxidation rate on the steady-state oxidation rate map obtained by the above-described fuel oxidation rate calculating operation, the actual fuel oxidation rate is changed to the extent of correction made based on the pressure, temperature, and oxygen concentration in a cylinder. It is estimated that the crank angle at which the fuel evaporation rate and the fuel oxidation rate that are changed in such manners match is the diffusion combustion start time in that combustion stroke. For example, in the map shown in FIG. 12(a), the actual fuel evaporation rate is lower than the fuel evaporation rate on the aforementioned steady-state fuel evaporation rate map, and a change in fuel evaporation rate and fuel oxidation rate of a case where a diffusion combustion start time is shifted toward the angle of advance side is shown. In contrast, in the map shown in FIG. 12(b), the actual fuel evaporation rate is higher than the fuel evaporation rate in the aforementioned steady-state fuel evaporation rate map, and a change in fuel evaporation rate and fuel oxidation rate of a case where a diffusion combustion start time is shifted toward the angle of delay side is shown.

[0118] In the above-described diffusion combustion start time estimating operation, when this estimation is completed, the fuel evaporation rate calculating operation and the fuel oxidation rate calculating operation are terminated. In addition, it is possible that a fuel evaporation rate and a fuel oxidation rate over the entire period of a preset period (crank opening range) are calculated, and the fuel evaporation rate and the fuel oxidation rate are compared at each crank angle to estimate a diffusion combustion start time. For example, a diffusion combustion start time may be estimated by calculating a fuel evaporation rate and a fuel oxidation rate in each specific period (for example, every 1° CA crank angle) from a main injection start timing to the crank angle position 10° CA after the compression top dead center that is generally considered as reaching a diffusion combustion start time.

- Control of diffusion combustion start time -

[0119] Next, another feature of this embodiment, i.e., control of a diffusion combustion start time (correction on a diffusion combustion start time by the diffusion combustion start time correcting means), will now be described. This diffusion combustion start time control is performed in the case where there is a disparity between a diffusion combustion start time estimated as described above and the target diffusion combustion start time (for example, the crank angle position corresponding to the compression top dead center (TDC) of the piston 13: hereinafter referred to as a target diffusion combustion start time). Specifically, in the case where there is a disparity between the estimated diffusion combustion start time and the target diffusion combustion start time, the diffusion combustion start time of the next cylinder that reaches a combustion stroke after the cylinder on which a diffusion combustion start time estimation was performed is adjusted according to the extent of deviation so as to bring the diffusion combustion start time close to the target diffusion combustion start time. That is, the diffusion combustion start time control is performed as feedback control on the next cylinder that reaches a combustion stroke. In the case where this diffusion combustion start time control is performed, the above-described diffusion combustion start time estimating operation is performed until the aforementioned target diffusion combustion start time or a crank angle position that is located a specific amount on the angle of delay side relative to the target diffusion combustion start time (for example, as stated above, to the crank angle position 10° CA after compression top dead center that is generally considered as reaching a diffusion combustion start time) is reached. That is, the diffusion combustion start time control is configured such that the actual fuel evaporation rate at least at the target diffusion combustion start time is obtained.

[0120] An outline of this diffusion combustion start time control will now be described using the flowchart of FIG. 13. First, a deviation is calculated between the fuel evaporation rate at the target diffusion combustion start time (crank angle position) calculated by the above-described estimation operation (hereinafter referred to as the actual fuel evaporation rate) and the map value of the aforementioned steady-state fuel evaporation rate map at the target diffusion combustion start time (hereinafter referred to as the target fuel evaporation rate) (step ST11).

[0121] Specifically, using FIG. 14, this deviation calculating operation is described as follows. The steady-state fuel evaporation rate map is specified as the target evaporation rate, and in the case where the actual evaporation rate at the target diffusion combustion start time is lower than the evaporation rate (target evaporation rate) on the steady-state evaporation rate map, the deviation at the target diffusion combustion start time (for example, the compression top dead center (TDC) of the piston 13) is obtained as " $\Delta dm_v/dt$ " as presented in the figure. That is, increasing the fuel evaporation rate to the extent corresponding to this deviation " $\Delta dm_v/dt$ " increases the fuel evaporation rate in the combustion chamber 3 to the target evaporation rate, and it is thus possible to match the diffusion combustion start time with the target diffusion combustion start time.

[0122] Then, a fuel injection pressure is set according to the deviation between the aforementioned actual fuel evaporation rate and the target fuel evaporation rate (step ST12). As for the fuel injection pressure set here, in the case where the actual fuel evaporation rate is lower than the target evaporation rate, the greater the deviation therebetween, the higher the fuel injection pressure is set. That is, in the case where the aforementioned estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the fuel injection pressure is set. This is because, in the case where the fuel injection pressure is set to be high, the particle size of fuel in the aforementioned droplet region is small, and the fuel evaporation rate is high. The following formula (5) shows the relationship between fuel injection pressure and fuel particle size (particle size of droplet: Dd).

[0123] [Formula 3]

$$Dd = 72.36 \cdot P_{cr}^{-0.4} \cdot d_0 \quad (5)$$

P_{cr} : injection pressure (or rail pressure)

d_0 : hole diameter

[0124] Therefore, substituting this formula (5) into the aforementioned formula (1) enables the fuel evaporation rate after the fuel injection pressure is changed to be obtained. On the contrary, in the case where the actual fuel evaporation rate is higher than the target evaporation rate, the greater the deviation therebetween, the lower the fuel injection pressure is set. Adjustment of the fuel injection pressure is executed by the ECU100 through metering the fuel discharge amount of the supply pump 21.

[0125] After setting the fuel injection pressure in this manner and performing a fuel injection, whether the actual fuel evaporation rate matches the target fuel evaporation rate or not is determined in step ST13.

[0126] In the case where the actual fuel evaporation rate matches the target evaporation rate, a YES is given in the step ST13, and the current fuel injection pressure is maintained. On the other hand, in the case where the actual fuel evaporation rate does not match the target evaporation rate and a NO is given in the step ST13, the procedure advances to step ST14 and the intake operation amount is adjusted. That is, in the case where it is not possible to match the actual fuel evaporation rate with the target evaporation rate solely by the aforementioned fuel injection pressure adjustment, and this fuel injection pressure adjustment reaches its limit (the aforementioned upper limit or lower limit), the fuel evaporation rate is adjusted by adjusting the intake operation amount. Specifically, the opening of the aforementioned swirl control valve 66 is adjusted. That is, in the case where the actual fuel evaporation rate is lower than the target evaporation rate, the greater the deviation therebetween, the smaller the opening of the swirl control valve 66 so as to set the swirl rate to be high. That is, in the case where the aforementioned estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the swirl rate is set. On the other hand, in the case where the actual fuel evaporation rate is higher than the target evaporation rate, the greater the deviation therebetween, the greater the opening of the swirl control valve 66 so as to set the swirl rate to be low. Thereby, the fuel evaporation rate is adjusted, and the actual fuel evaporation rate is brought close to the target evaporation rate.

[0127] This swirl rate adjustment is performed according to the swirl rate adjustment map shown in FIG. 15. This swirl rate adjustment map specifies the relationship between the necessary swirl rate and the amount of operation of the swirl control valve 66 for attaining that swirl rate. A swirl rate necessary to match the actual fuel evaporation rate with the target evaporation rate is obtained, and the resulting rate is applied to the swirl rate adjustment map, thereby enabling the amount of operation of the swirl control valve 66 to be obtained.

- Other embodiments -

[0128] In the embodiments described above, a case was described in which the present invention is applied to an in-line four-cylinder diesel engine mounted in an automobile. The present invention is not limited to use in an automobile,

and is applicable also to engines used in other applications. Also, there is no particular limitation with respect to the number of cylinders or the engine type (classified as an in-line engine, V engine, horizontally opposed engine, and so forth).

[0129] In place of, or in addition to, the above-described swirl rate adjustment as a fuel evaporation rate adjustment operation, a valve opening time adjustment on the intake valve 16 by a variable valve timing (VVT) mechanism may be performed. That is, in the case where the actual fuel evaporation rate is lower than the target evaporation rate, the greater the deviation therebetween, the greater the valve opening time of the intake valve 16 is retarded so as to set the intake flow rate to be high. That is, in the case where the aforementioned estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the greater the valve opening time of the intake valve 16 is retarded so as to set the intake flow rate to be high.

[0130] Also, in the embodiments described above, the maniverter 77 is provided with the NSR catalyst 75 and the DPNR catalyst 76, but a maniverter provided with the NSR catalyst 75 and a diesel particulate filter (DPF) may be used as well.

[0131] The present invention is applicable also to diesel engines that performs split main injections.

[0132] In the embodiments described above, although a description was provided of an engine to which a piezo injector 23, which attains a full valve opening state only when a current is applied thereto and thus changes a fuel injection rate, is applied, the present invention is applicable also to engines to which a variable injection rate injector is applied.

[Industrial Applicability]

[0133] The present invention can be employed in a common rail in-cylinder direct injection multi-cylinder diesel engine mounted in an automobile to appropriately control a diffusion combustion start time.

[Reference Signs List]

[0134]

1	Engine (internal combustion engine)
3	Combustion chamber
21	Supply pump
23	Injector (fuel injection valve)
66	Swirl control valve

Claims

1. A diffusion combustion start time estimating apparatus for a compression self-igniting internal combustion engine that estimates a diffusion combustion start time when fuel injected from a fuel injection valve into a combustion chamber after premixed combustion of an air-fuel mixture has started starts diffusion combustion, wherein the diffusion combustion start time estimating apparatus comprising:

a fuel evaporation rate calculating means that calculates an evaporation rate of fuel injected from a fuel injection valve into a combustion chamber by performing a correction on a reference evaporation rate, which is specified in advance, according to at least one of environmental conditions and operating conditions;

a fuel oxidation rate calculating means that calculates an oxidation rate of fuel forming an air-fuel mixture due to evaporation in the combustion chamber by performing a correction on a reference oxidation rate, which is specified in advance, according to at least one of environmental conditions and operating conditions; and

a diffusion combustion start time estimating means that estimates that a diffusion combustion start time of the air-fuel mixture is a time when the fuel evaporation rate calculated by the fuel evaporation rate calculating means and the fuel oxidation rate calculated by the fuel oxidation rate calculating means arrive in a state where the fuel evaporation rate and the fuel oxidation rate match from a state where there is a disparity between the fuel evaporation rate and the fuel oxidation rate.

2. The diffusion combustion start time estimating apparatus for an internal combustion engine according to claim 1, wherein

the fuel evaporation rate calculating means is configured to calculate a fuel evaporation rate by multiplying by correction coefficients according to an actual pressure and temperature in a cylinder a map value on a steady-state fuel evaporation rate map that shows a fuel evaporation rate at each crank angle and that is prepared on the

assumption that inside of a cylinder is under a reference pressure and at a reference temperature.

3. The diffusion combustion start time estimating apparatus for an internal combustion engine according to claim 2, wherein

the fuel oxidation rate calculating means is configured to calculate a fuel oxidation rate by multiplying by correction coefficients according to an actual pressure, temperature, and oxygen concentration in a cylinder a map value on a steady-state fuel oxidation rate map that shows a fuel oxidation rate at each crank angle and that is prepared on the assumption that inside of a cylinder is under a reference pressure, at a reference temperature, and at a reference oxygen concentration.

4. The diffusion combustion start time estimating apparatus for an internal combustion engine according to claim 3, wherein

the steady-state fuel oxidation rate map, with the fuel evaporation rate at a time when fuel is injected into a combustion chamber from a fuel injection valve being set to be "0", is prepared by straight line approximation on the assumption that the fuel oxidation rate increases at a constant acceleration from the time when fuel is injected to a fuel oxidation rate that equals to a fuel evaporation rate on the steady-state evaporation rate map at a diffusion combustion start time that is specified based on a change in pressure in a cylinder in a combustion stroke.

5. A diffusion combustion start time control apparatus that controls a diffusion combustion start time estimated by an diffusion combustion start time estimating apparatus for an internal combustion engine of any one of claims 1 to 4, comprising:

a diffusion combustion start time correcting means that calculates a deviation of the estimated diffusion combustion start time relative to a target diffusion combustion start time, and performs based on this deviation a diffusion combustion start time correcting operation such that the estimated diffusion combustion start time matches the target diffusion combustion start time.

6. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 5, wherein the diffusion combustion start time correcting means is configured to perform a diffusion combustion start time correcting operation by changing a control parameter that varies an evaporation rate of fuel injected into a combustion chamber.

7. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 5, wherein the diffusion combustion start time correcting means is configured to perform a diffusion combustion start time correcting operation by changing, according to a deviation between a target fuel evaporation rate with which the target diffusion combustion start time is obtained and an actual fuel evaporation rate that corresponds to the estimated diffusion combustion start time, a control parameter that varies a fuel evaporation rate.

8. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 6, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber is a fuel injection pressure, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the fuel injection pressure is set.

9. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 7, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber is a fuel injection pressure, and in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the higher the fuel injection pressure is set.

10. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 6, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber is a swirl rate in a combustion chamber, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the swirl rate is set.

11. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 7, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber is a swirl rate in a combustion chamber, and in the case where the actual fuel evaporation rate is lower than the target fuel evaporation

rate, the greater the deviation therebetween, the higher the swirl rate is set.

5 12. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 6, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber includes a fuel injection pressure and a swirl rate in a combustion chamber, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the higher the fuel injection pressure is set, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time even when the fuel injection pressure is corrected to a correction limit, the greater the deviation therebetween, the higher the swirl rate is set.

10 13. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 7, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber includes a fuel injection pressure and a swirl rate in a combustion chamber, and in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the higher the fuel injection pressure is set, and in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate even when the fuel injection pressure is corrected to a correction limit, the greater the deviation therebetween, the higher the swirl rate is set.

15 14. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 6, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber is a valve timing of an intake valve, and in the case where the estimated diffusion combustion start time is on the angle of advance side relative to the target diffusion combustion start time, the greater the deviation therebetween, the farther the valve timing at which the intake valve opens is set toward the angle of delay side.

20 15. The diffusion combustion start time control apparatus for an internal combustion engine according to claim 7, wherein the control parameter that varies an evaporation rate of fuel injected into a combustion chamber is a valve timing of an intake valve, and in the case where the actual fuel evaporation rate is lower than the target fuel evaporation rate, the greater the deviation therebetween, the farther the valve timing at which the intake valve opens is set toward the angle of delay side.

FIG. 1

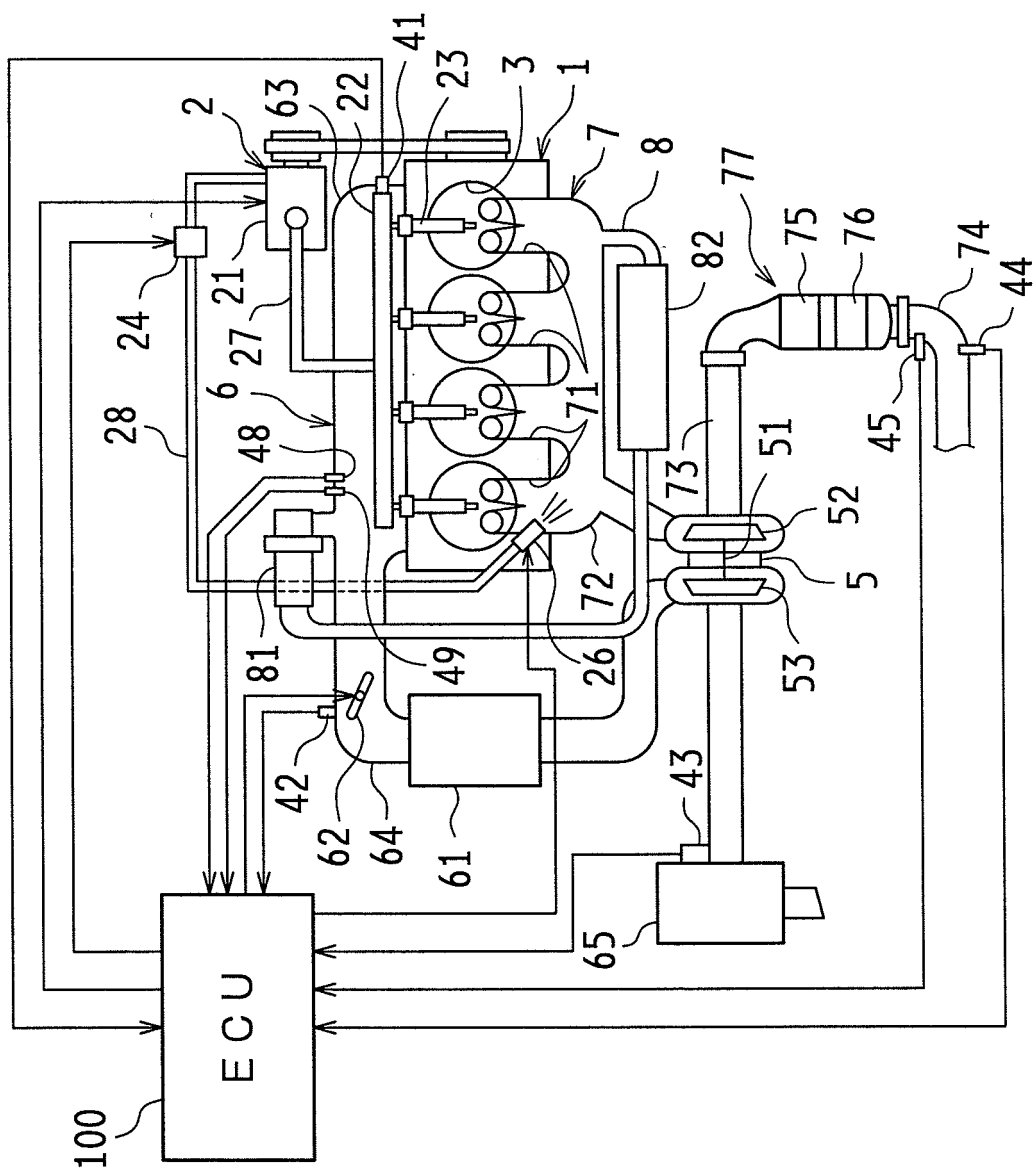


FIG.2

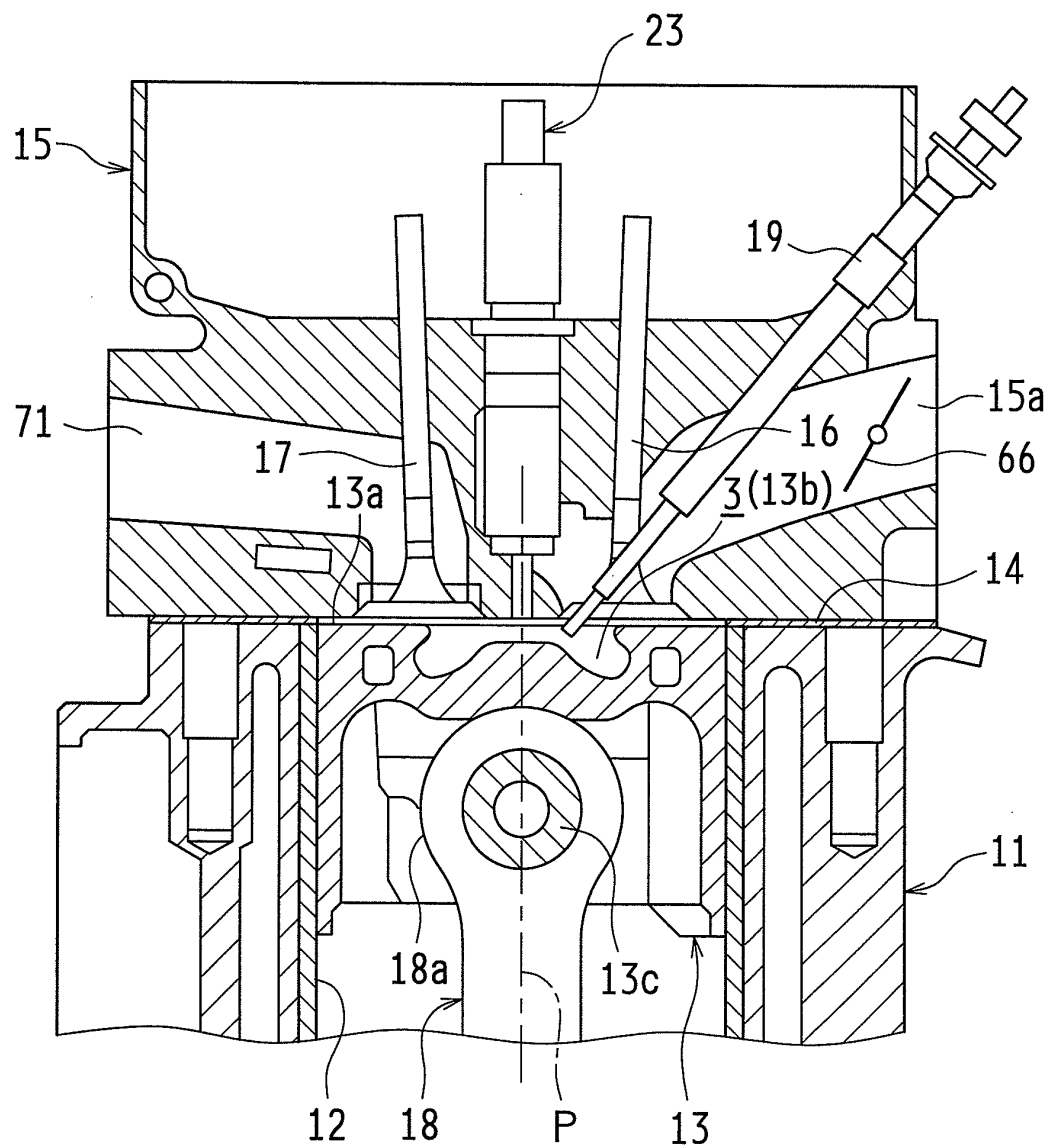


FIG.3

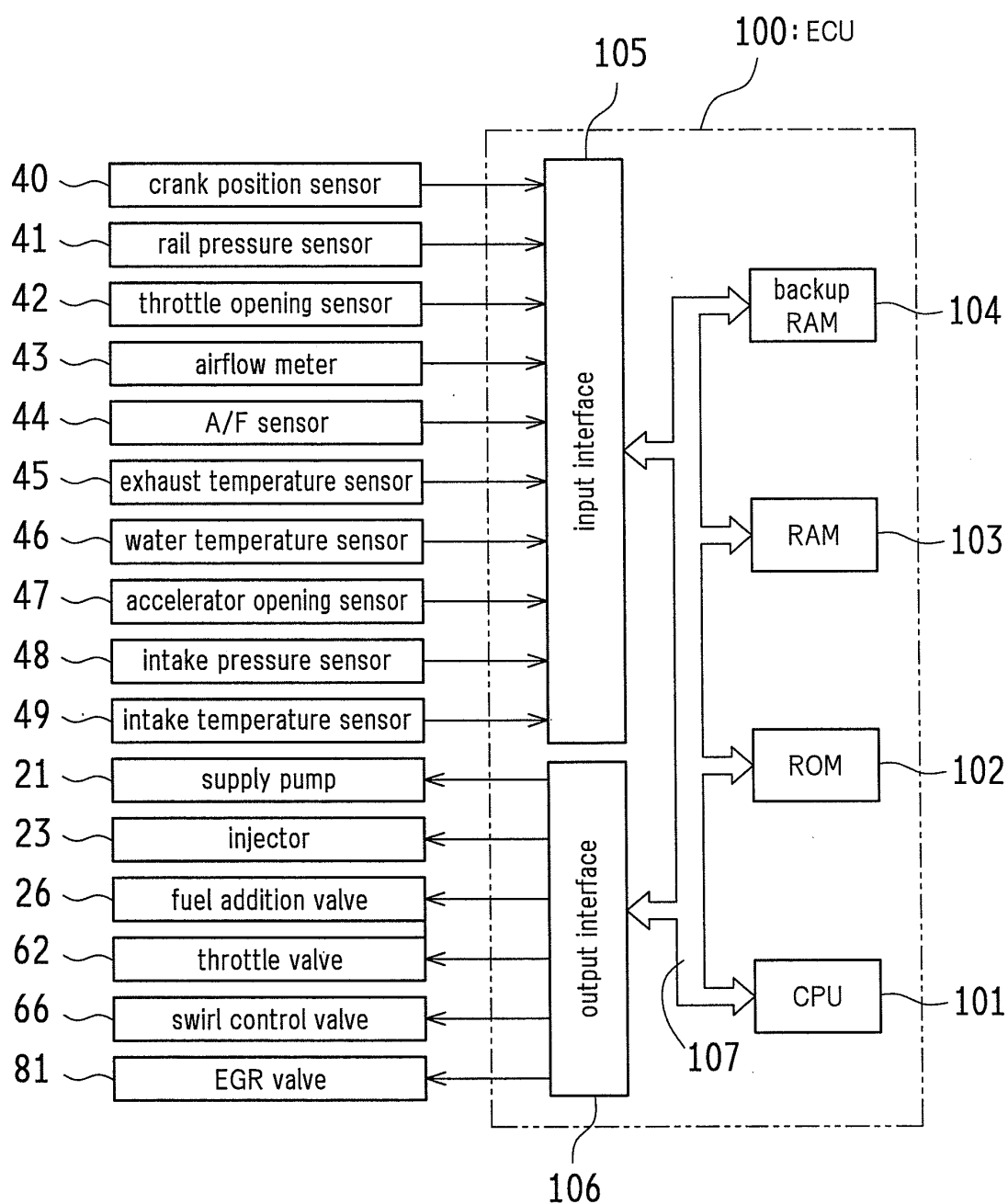


FIG.4

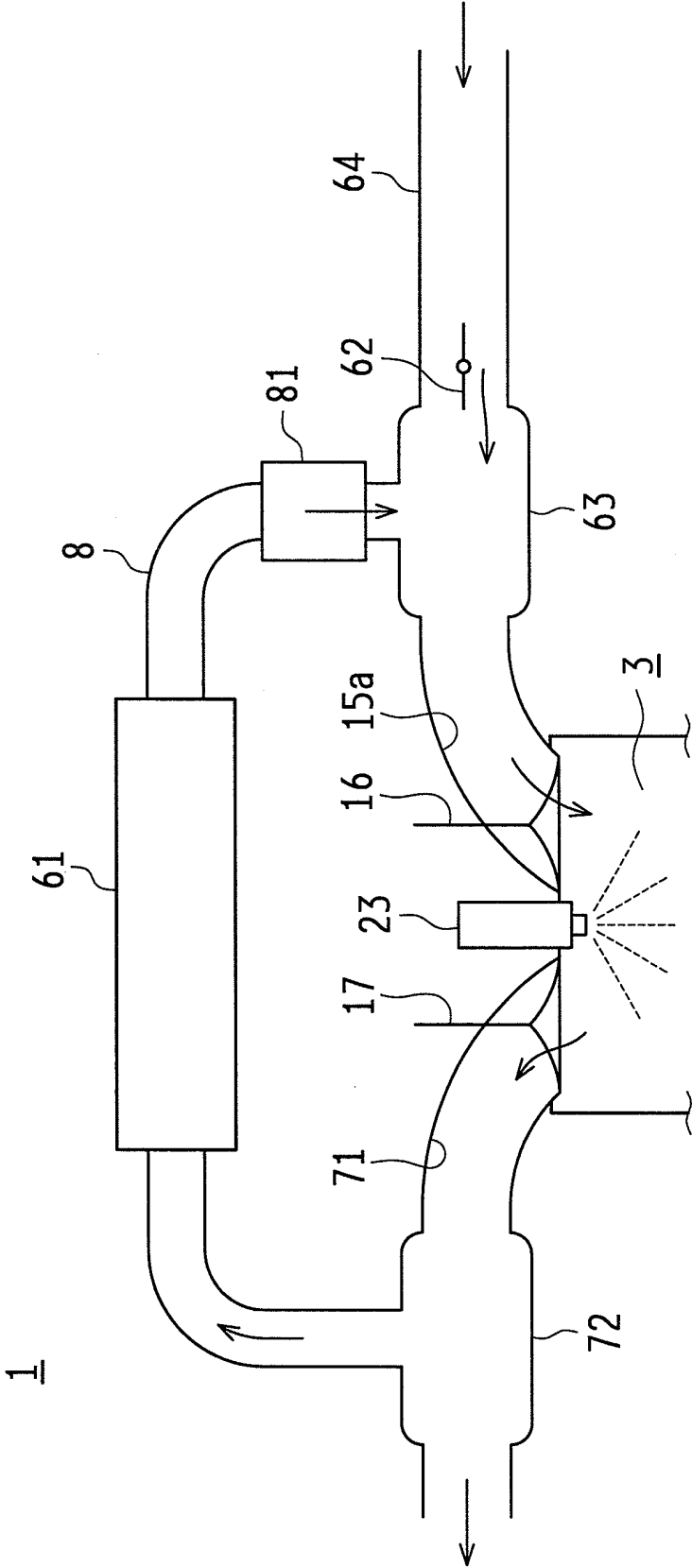


FIG.5

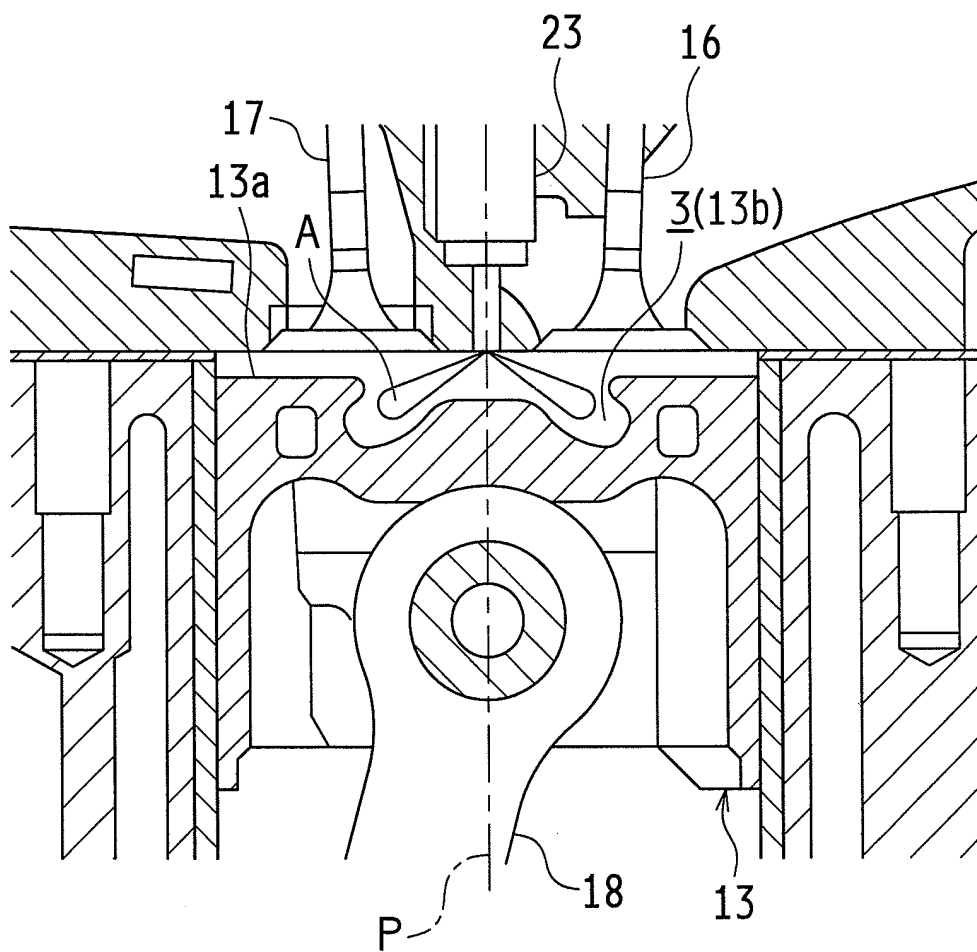


FIG.6

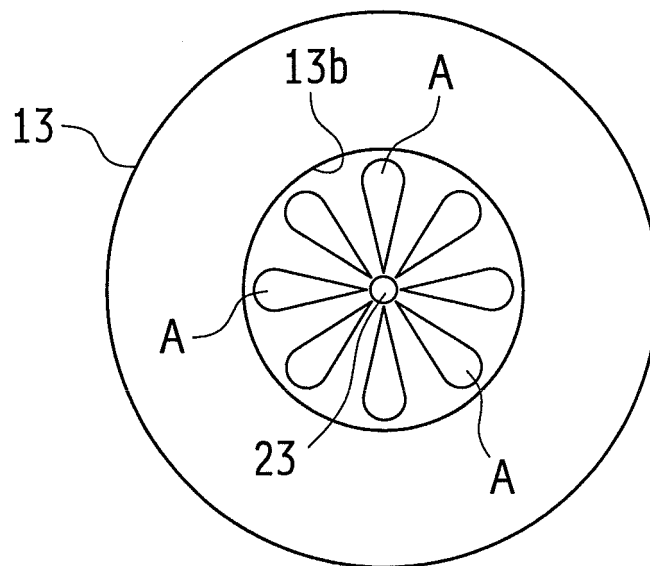


FIG.7

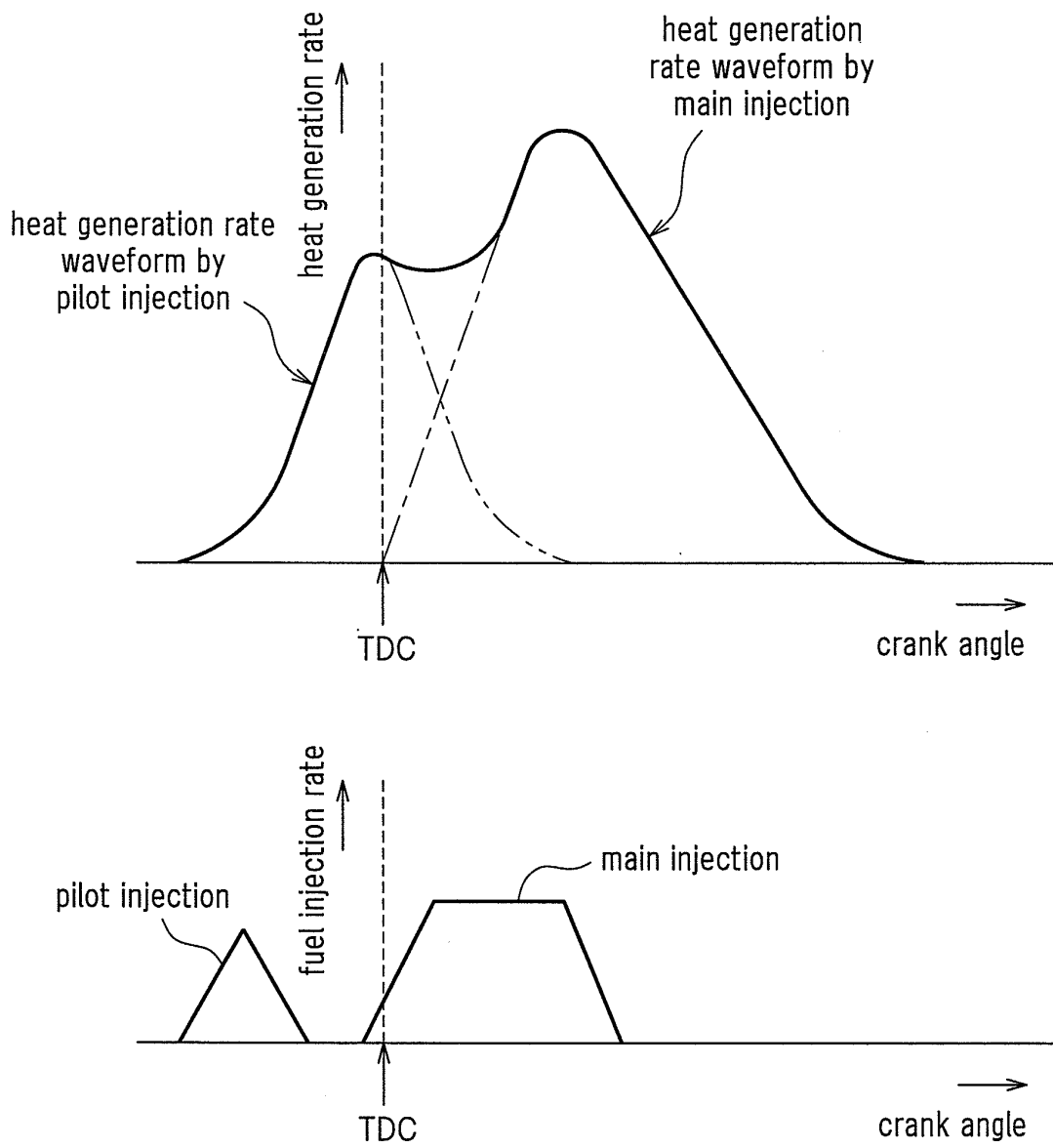


FIG.8

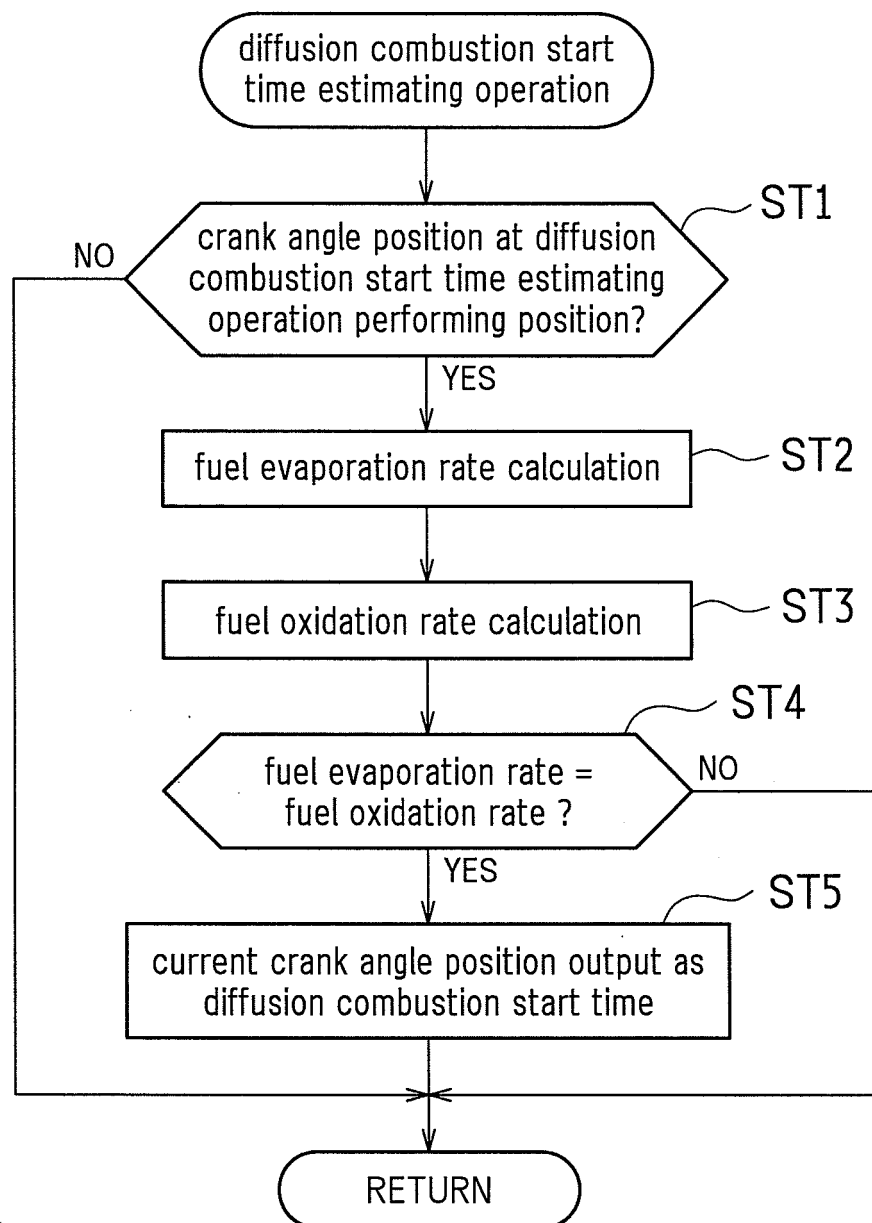
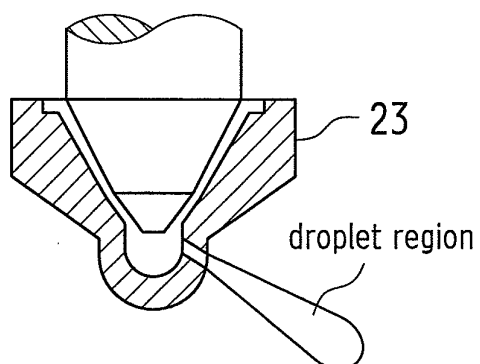
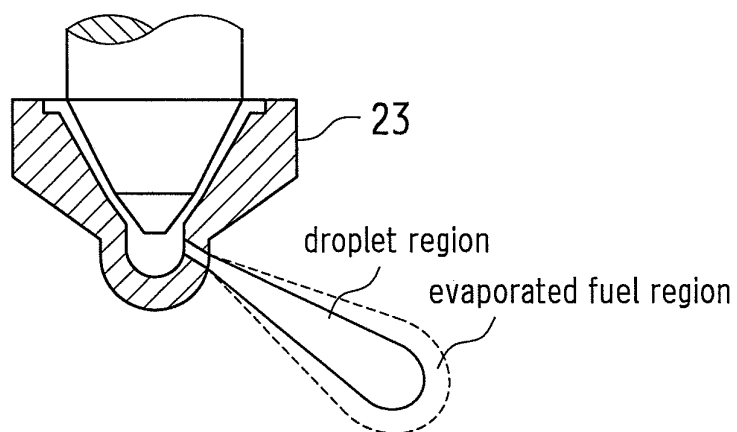


FIG.9

(a)



(b)



(c)

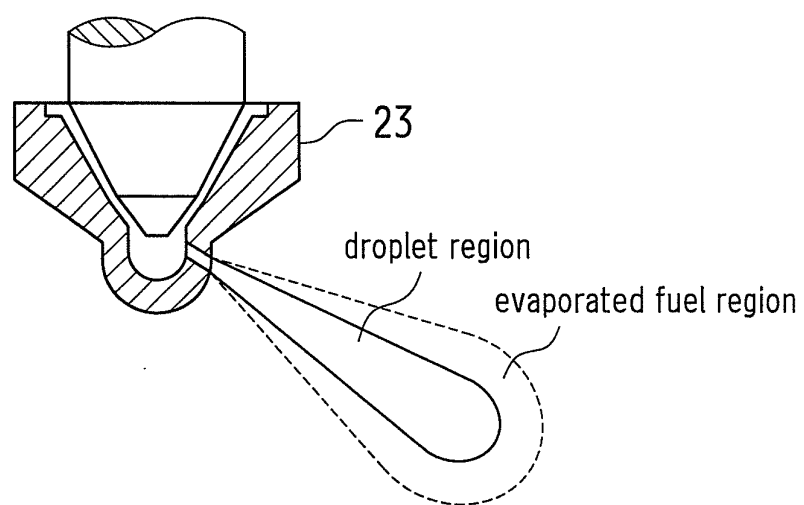


FIG.10

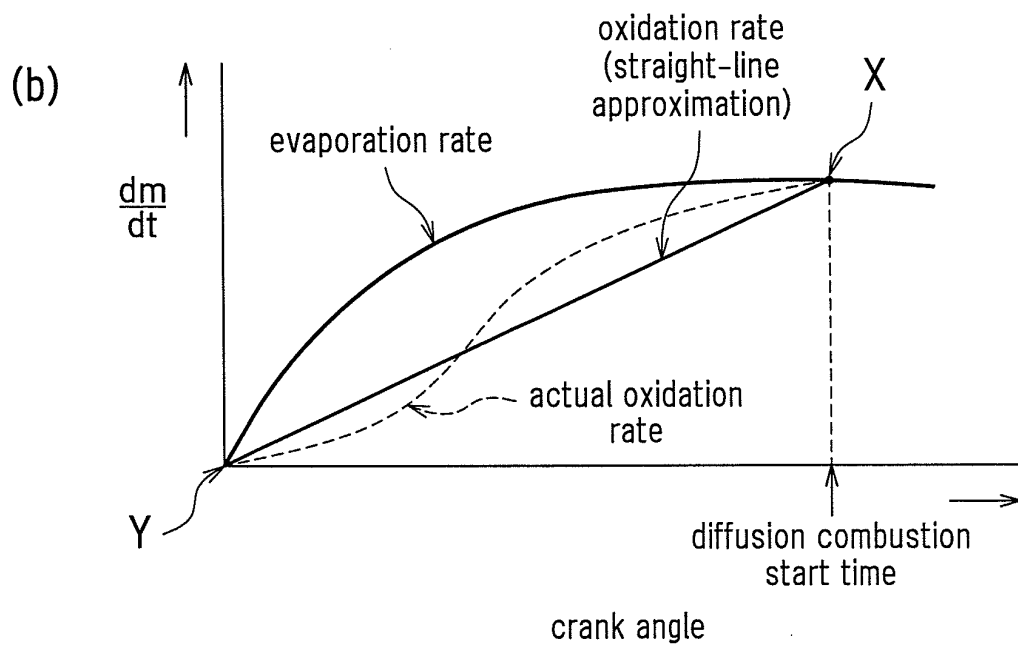
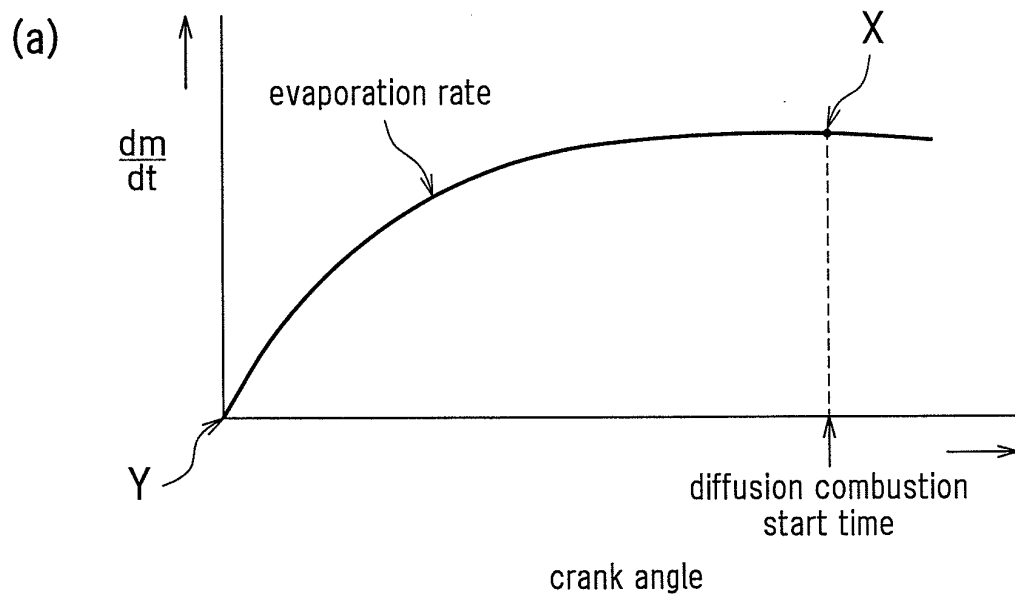
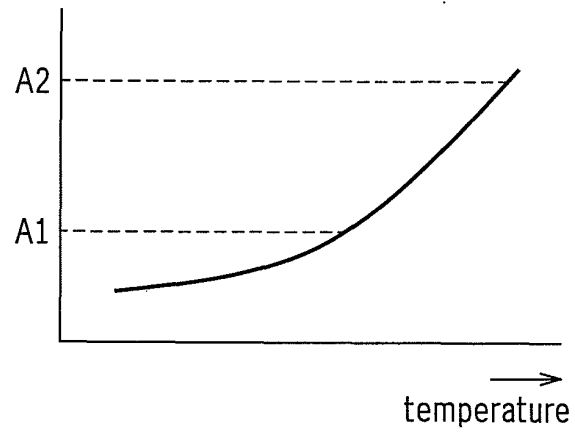
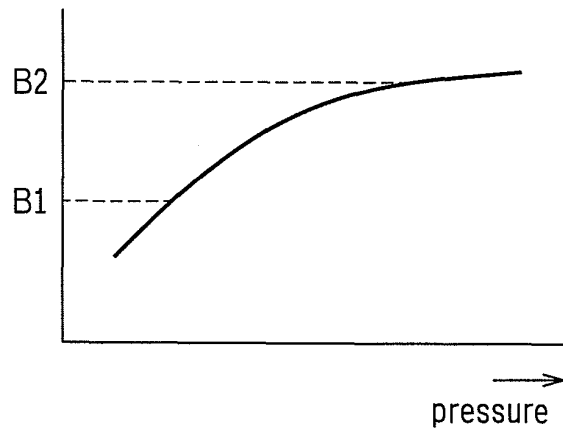


FIG.11

(a)



(b)



(c)

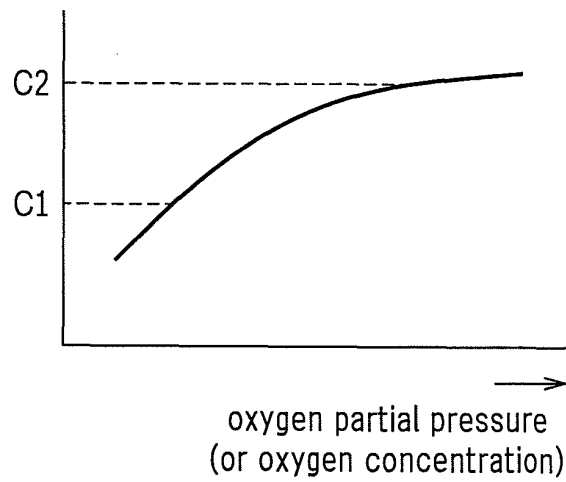


FIG.12

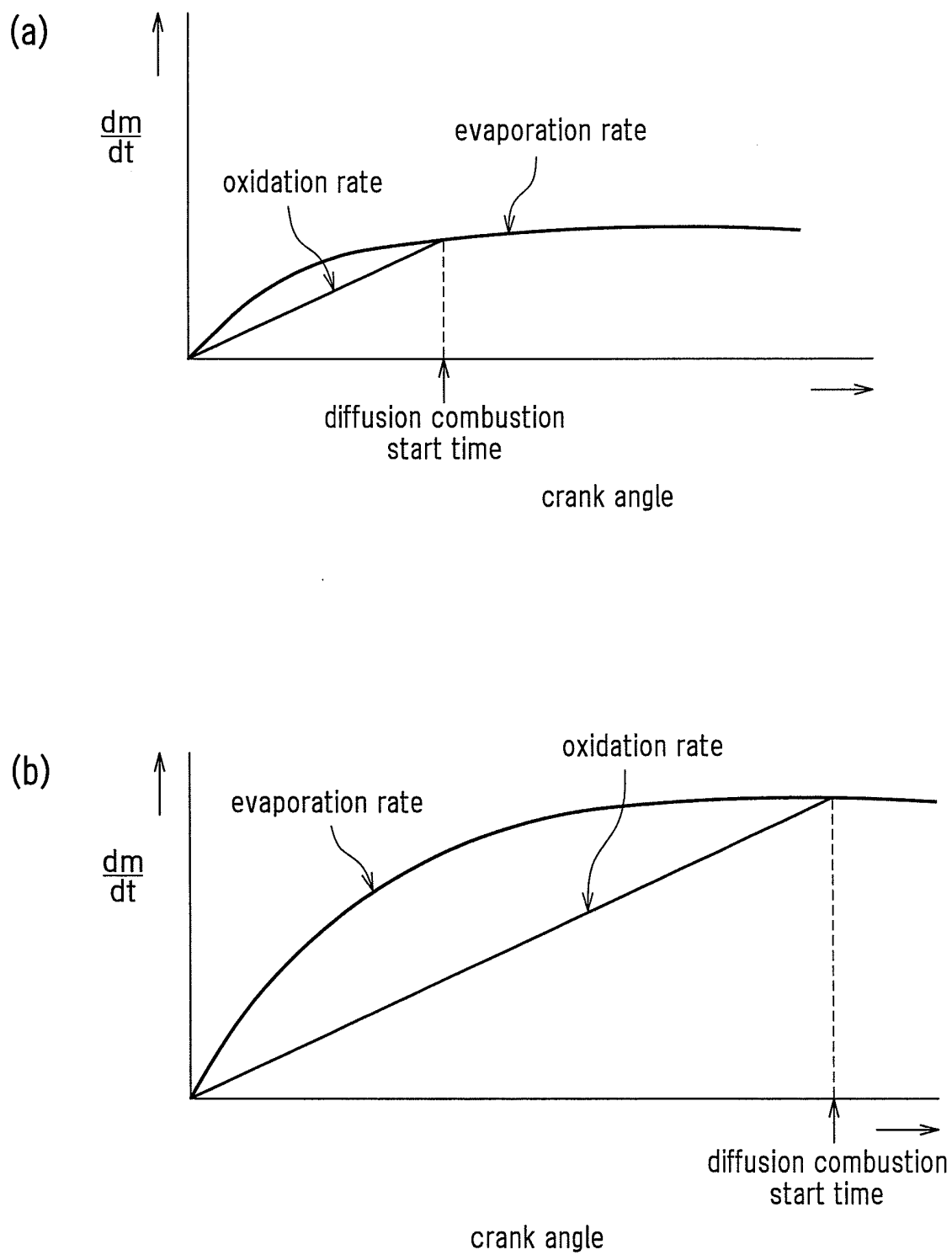


FIG.13

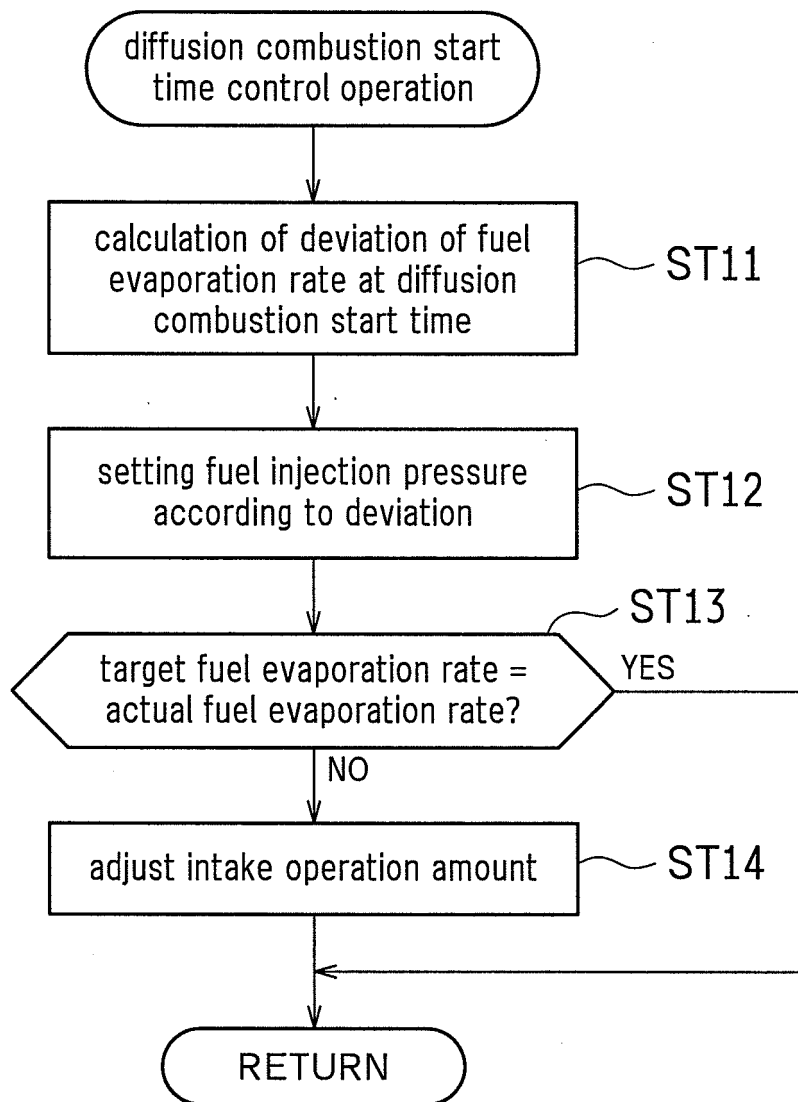


FIG.14

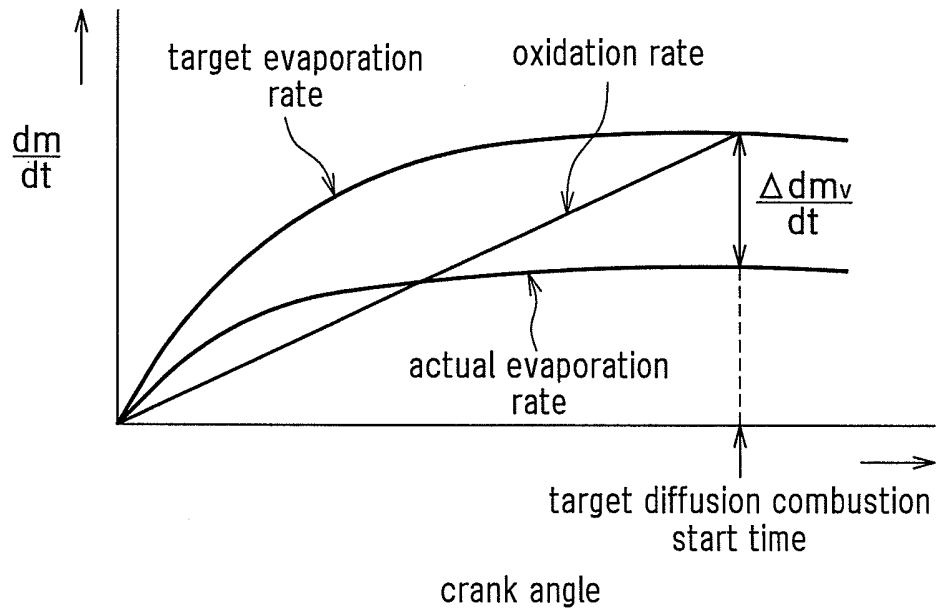
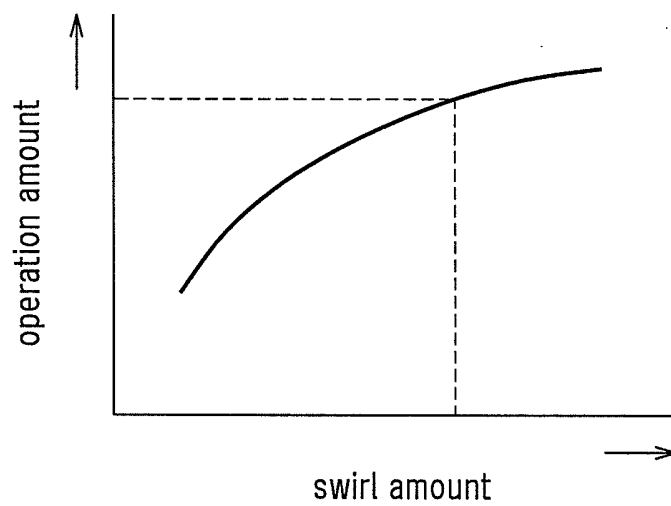


FIG.15



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2010/064397

A. CLASSIFICATION OF SUBJECT MATTER

F02D41/40 (2006.01) i, F02D41/38 (2006.01) i, F02D45/00 (2006.01) i

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)

F02D41/40, F02D41/38, F02D45/00

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2010
Kokai Jitsuyo Shinan Koho	1971-2010	Toroku Jitsuyo Shinan Koho	1994-2010

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2007-92583 A (Toyota Motor Corp.), 12 April 2007 (12.04.2007), claim 1 (Family: none)	1-15
A	WO 2010/035341 A1 (Toyota Motor Corp.), 01 April 2010 (01.04.2010), paragraphs [0104] to [0106] (Family: none)	1-15

☐ Further documents are listed in the continuation of Box C.☐ See patent family annex.

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Date of the actual completion of the international search
29 September, 2010 (29.09.10)Date of mailing of the international search report
12 October, 2010 (12.10.10)Name and mailing address of the ISA/
Japanese Patent Office

Authorized officer

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Form PCT/ISA/210 (second sheet) (July 2009)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2007092583 A [0007]
- JP 2008261312 A [0007]
- JP 2001254645 A [0007]