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(54) Method of estimating the in cylinder temperature after combustion

(57) In a combustion temperature estimation method for an internal combustion engine, upon each arrival of fuel injection start timing, the pre-combustion temperature of cylinder interior gas at the time of ignition (ignition-time compressed cylinder interior gas temperature Tpump) is estimated on the basis of the fact that the state of the cylinder interior gas changes adiabatically. The quantity of heat generated as a result of combustion of fuel is divided by the product of the post-combustion mole amount and constant-pressure specific heat of the cylinder interior gas, which can be obtained from the concentration proportions of gas components contained in intake gas, to thereby estimate an increase in temperature of the cylinder interior gas stemming from combustion (combustion ascribable temperature increase ΔTburn). Further, an increase in temperature of the cylinder interior gas stemming from an increase in combustion speed (combustion-speed ascribable temperature increase ΔTb_velo) is estimated on the basis of fuel injection pressure and engine speed, which are factors which influence the combustion speed. Then, the highest combustion temperature Tflame is estimated by the equation Tflame = Tpump + ΔTburn + ΔTb_velo.

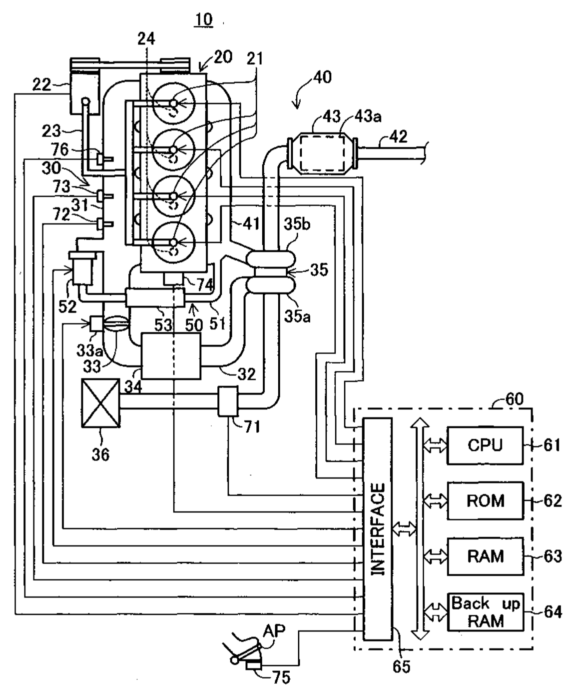

FIG.1

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Description

BACKGROUND OF THE INVENTION

Field of the Invention

[0001] The present invention relates to a combustion temperature estimation method for estimating combustion temperature in a cylinder (in a combustion chamber) of an internal combustion engine.

[0002] The quantity of emissions such as NO_x discharged from an internal combustion engine such as a spark-ignition engine or a diesel engine has a strong correlation with combustion temperature (highest combustion temperature, highest flame temperature) in each cylinder. Therefore, an effective way for reducing the quantity of emissions such as NO_x is controlling the combustion temperature to a predetermined temperature. Meanwhile, actual measurement of the combustion temperature is very difficult. Therefore, the combustion temperature must be accurately estimated in order to control the combustion temperature to a predetermined temperature.

[0003] In view of the foregoing, the combustion chamber temperature condition estimation apparatus for an internal combustion engine described in Japanese Patent Application Laid-Open (*kokai*) No. 2002-54491 includes an exhaust temperature sensor for detecting, at the outside of a combustion chamber of the engine, the temperature of exhaust gas discharged from the combustion chamber, and, on the basis of the detected exhaust gas temperature, estimates the temperature in the combustion chamber after completion of combustion (accordingly, the above-mentioned cylinder interior combustion temperature), which temperature has a strong correlation with the detected exhaust gas temperature.

[0004] The above-mentioned apparatus estimates the cylinder interior combustion temperature on the assumption that a strong correlation exists between the combustion temperature and the exhaust temperature as measured externally of the combustion chamber. However, in actuality, a strong correlation does not always exist therebetween; therefore, in some cases, the above-mentioned combustion temperature cannot be accurately estimated. In addition, the above-mentioned apparatus has a drawback of increased production cost and a complex structure, because the apparatus includes an exhaust temperature sensor as an essential component.

SUMMARY OF THE INVENTION

[0005] In view of the foregoing, an object of the present invention is to provide a combustion temperature estimation method for an internal combustion engine, which can accurately estimate the temperature of combustion within a cylinder interior (within a combustion chamber) of the internal combustion engine by use of a simple configuration.

[0006] In a combustion temperature estimation method for an internal combustion engine according to the present invention, an ignition-time compressed cylinder interior gas temperature, which is a pre-combustion cylinder interior gas temperature at the time of ignition, is first estimated, while utilizing the fact that at least cylinder interior gas present in a cylinder is compressed in the cylinder. In the case of a spark-ignition engine, the time of ignition refers to a time of ignition by a spark plug. In the case of a diesel engine, the time of ignition refers to a point in time when a predetermined ignition delay time elapses from a fuel injection timing (a main injection timing in the case where main injection operation is performed after at least one pilot injection operation).

[0007] The ignition-time compressed cylinder interior gas temperature can be obtained in an accurate and easy manner on the basis of a cylinder interior volume at the time of ignition and a general formula which represents an adiabatic change by use of a politropic index, on the assumption that the state (i.e., temperature and pressure) of cylinder interior gas adiabatically changes before combustion in a compression stroke (and an expansion stroke).

[0008] Further, in the estimation method, a combustion ascribable temperature increase, which is an increase in temperature of the cylinder interior gas as a result of combustion, is estimated on the basis of at least the composition of gas taken in the cylinder and the quantity of heat generated as a result of combustion of injected fuel. In general, an increase in temperature of the cylinder interior gas as a result of combustion (i.e., the combustion ascribable temperature increase) can be obtained by dividing the quantity of heat generated as a result of combustion of fuel by a post-combustion specific heat (constant-pressure specific heat) of the cylinder interior gas having participated in combustion and a post-combustion quantity (by mol) of the cylinder interior gas having participated in combustion.

[0009] The combustion specific heat and amount by mol (hereinafter may be referred to as "mole amount") of the cylinder interior gas having participated in combustion change depending on the composition of gas taken into a cylinder (hereinafter may be referred to as "intake gas"); i.e., proportions by concentrations (hereinafter may be referred to as "concentration proportions") of a plurality of components (e.g., oxygen and inert gases) of intake gas, and increase with, for example, the concentration proportions of inert gases (a detail description will be provided later). Accordingly, the combustion ascribable temperature increase can be accurately obtained on the basis of the constant-pressure specific heat and mole amount of cylinder interior gas having participated in combustion, and the quantity of heat generated as a result of combustion of fuel.

[0010] Moreover, in this estimation method, a combustion-speed ascribable temperature increase, which is an increase in temperature of cylinder interior gas stemming from an increase in combustion speed, is estimated on the basis of factors which influence combustion speed in a cylinder (hereinafter simply referred to "combustion speed"). In general, in the case of a diesel engine, in many cases, the time of ignition is after a point in time corresponding to the compression dead center (i.e., the time of ignition is a point in time in an expansion stroke). In the expansion stroke, the cylinder interior gas temperature decreases with time.

[0011] Accordingly, in such a case, the shorter the time between the time of ignition (i.e., combustion start time) and a point in time at which the cylinder interior gas temperature reaches the highest combustion temperature, the higher the highest combustion temperature. In other words, the (highest) combustion temperature of cylinder interior gas increases with combustion speed after initiation of combustion. Such an increase in temperature of cylinder interior gas is obtained as the combustion-speed ascribable temperature increase, on the basis of factors which influence the combustion speed.

[0012] Examples of factors which influence the combustion speed includes fuel injection pressure, engine speed, swirl ratio of gas taken into the cylinder, and boost pressure produced by a supercharger (for the case where the engine is equipped with such a supercharger), and oxygen concentration of gas taken into the cylinder.

[0013] In this estimation method, the (highest) combustion temperature in the cylinder (hereinafter simply referred to as "(highest) combustion temperature") is estimated from a value obtained through addition of the combustion ascribable temperature increase and the combustion-speed ascribable temperature increase to the ignition-time compressed cylinder interior gas temperature. Accordingly, the highest combustion temperature in the cylinder estimated in this manner can be a value which accurately represents various actual phenomena.

[0014] For example, when the time of ignition (in an expansion stroke) delays due to a delay in fuel injection timing, the above-mentioned estimated ignition-time compressed cylinder interior gas temperature drops because of an increase in cylinder volume at the time of ignition, whereby the estimated highest combustion temperature in the cylinder decreases. This estimation result matches an actual phenomenon that the quantity of generated NO_x decreases with a delay in fuel injection timing.

[0015] When the concentrations of inert gases (e.g., CO_2) contained in intake gas increases by means of, for example, increasing the EGR ratio (the ratio of the quantity of EGR gas to the quantity of intake gas), as described above, the constant-pressure specific heat and mole amount of cylinder interior gas having participated in combustion both increase, whereby the above-described estimated combustion ascribable temperature increase decreases, and thus, the above-described estimated highest combustion temperature decreases. This estimation result matches an actual phenomenon that the quantity of generated NO_x decreases with an increase in the EGR ratio.

[0016] When the combustion speed is increased by means of, for example, increasing the fuel injection pressure and the engine speed, the above-described estimated combustion-speed ascribable temperature increase increases, and thus, the above-described estimated highest combustion temperature increases. This estimation result matches an actual phenomenon that the quantity of generated NO_x increases when the fuel injection pressure and the engine speed are increased. As described above, the combustion temperature estimation method according to the present invention can accurately estimate the highest combustion temperature by use of a simple configuration to match various actual phenomena.

[0017] In the combustion temperature estimation method according to the present invention, the ignition-time compressed cylinder interior gas temperature is preferably estimated on the basis of the temperature of gas taken into the cylinder, the composition of the taken gas, and an increase in the ignition-time compressed cylinder interior gas temperature estimated on the basis of a factor which increases the pre-combustion cylinder interior gas temperature. In this case, examples of the factor which increases the pre-combustion cylinder interior gas temperature include heat generated as a result of combustion of fuel injected by means of pilot injection in the case where such pilot injection is performed before main fuel injection, and heat generated as a result of supply of electricity to a glow plug in the case where electricity is supplied to the glow plug.

[0018] The ignition-time compressed cylinder interior gas temperature naturally changes with intake temperature. Further, since the politropic index to be used with cylinder interior gas which causes adiabatic change changes depending on the composition of intake gas (accordingly, the composition of cylinder interior gas), the ignition-time compressed cylinder interior gas temperature (based on a general formula representing adiabatic change by use of the politropic index) also changes with the composition of intake gas.

[0019] Moreover, in the case where the above-mentioned heat generated as a result of supply of electricity to a glow plug is imparted to cylinder interior gas before combustion, the ignition-time compressed cylinder interior gas temperature increases by an amount corresponding to the quantity of the heat. Such an increase in temperature of the cylinder interior gas is obtained as an increase in the ignition-time compressed cylinder interior gas temperature.

[0020] As is understood from the above, the temperature of gas (intake gas) taken into the cylinder, the composition of the taken gas (intake gas), and an increase in the ignition-time compressed cylinder interior gas temperature can serve as values (parameters) which influence the ignition-time compressed cylinder interior gas temperature. Accordingly, as described above, in addition to the fact of cylinder interior gas being compressed in a cylinder, the above-described three parameters are taken into consideration when the ignition-time compressed cylinder interior gas temperature is estimated. Thus, the ignition-time compressed cylinder interior gas temperature can be estimated more accurately, and as a result, the (highest) combustion temperature can be estimated more accurately.

[0021] In the combustion temperature estimation method according to the present invention, when the time of ignition is prior to the point in time corresponding to the compression top dead center, the ignition-time compressed cylinder interior gas temperature is preferably estimated under the assumption that the time of ignition coincides with the point in time corresponding to the compression top dead center.

[0022] When the time of ignition is prior to the point in time corresponding to the compression top dead center (that is, the time of ignition is located in a compression stroke), the cylinder interior gas after ignition (i.e., during combustion or after combustion) is further compressed up to a point in time corresponding to the compression top dead center, and as a result, the highest combustion temperature is considered to increase up to a point in time corresponding to the compression top dead center. In other words, the highest combustion temperature in this case coincides with the highest combustion temperature in the case where the time of ignition coincides with the point in time corresponding to the compression top dead center.

[0023] Accordingly, by virtue of the above-described operation in which when the time of ignition is prior to the point in time corresponding to the compression top dead center, the ignition-time compressed cylinder interior gas temperature is estimated under the assumption that the time of ignition coincides with the point in time corresponding to the compression top dead center, the highest combustion temperature can be estimated more accurately for the case where the time of ignition is prior to the point in time corresponding to the compression top dead center.

BRIEF DESCRIPTION OF THE DRAWINGS

[0024]

FIG. 1 a schematic diagram showing the overall configuration of a system in which an engine control apparatus, which performs a combustion temperature estimation method for an internal combustion engine according to an embodiment of the present invention, is applied to a four-cylinder internal combustion engine (diesel engine);
FIG. 2 is a diagram schematically showing a state in which gas is taken from an intake manifold to a certain cylinder and is then discharged to an exhaust manifold;
FIG. 3 is a graph showing the relation between crank angle and cylinder interior gas temperature for the case where cylinder interior gas changes adiabatically in compression and expansion strokes;
FIG. 4 is an explanatory view showing an increase in the highest combustion temperature of cylinder interior gas with combustion speed;
FIG. 5 is a flowchart showing a routine which the CPU shown in FIG. 1 executes so as to control fuel injection quantity, etc;
FIG. 6 is a table for determining a fuel injection quantity, to which the CPU shown in FIG. 1 refers during execution of the routine shown in FIG. 5;
FIG. 7 is a table for determining a base fuel injection timing, to which the CPU shown in FIG. 1 refers during execution of the routine shown in FIG. 5;
FIG. 8 is a table for determining a base fuel injection pressure, to which the CPU shown in FIG. 1 refers during execution of the routine shown in FIG. 5;
FIG. 9 is a table for determining a target NO_x generation quantity, to which the CPU shown in FIG. 1 refers during execution of the routine shown in FIG. 5;
FIG. 10 is a table for determining an injection-timing correction value, to which the CPU shown in FIG. 1 refers during execution of the routine shown in FIG. 5;
FIG. 11 is a flowchart showing a routine which the CPU shown in FIG. 1 executes so as to compute combustion temperature;
FIG. 12 is a flowchart showing a routine which the CPU shown in FIG. 1 executes so as to compute ignition-time compressed cylinder interior gas temperature;
FIG. 13 is a flowchart showing a routine which the CPU shown in FIG. 1 executes so as to compute combustion ascribable temperature increase;
FIG. 14 is a flowchart showing a routine which the CPU shown in FIG. 1 executes so as to compute combustion-speed ascribable temperature increase; and
FIG. 15 is a flowchart showing a routine which the CPU shown in FIG. 1 executes so as to compute NO_x generation quantity (actual NO_x generation quantity).

DESCRIPTION OF THE PREFERRED EMBODIMENT

[0025] With reference to the drawings, there will now be described an control apparatus of an internal combustion engine (diesel engine), which apparatus performs a combustion temperature estimation method for an internal combustion engine according to an embodiment of the present invention, and estimates the quantity of NO_x generated within a cylinder as a result of combustion, on the basis of the combustion temperature estimated by the method.

[0026] FIG. 1 schematically shows the entire configuration of a system in which such an engine control apparatus is applied to a four-cylinder internal combustion engine (diesel engine) 10. This system comprises an engine main body 20 including a fuel supply system; an intake system 30 for introducing gas to combustion chambers (cylinder interiors) of individual cylinders of the engine main body 20; an exhaust system 40 for discharging exhaust gas from the engine main body 20; an EGR apparatus 50 for performing exhaust circulation; and an electronic control apparatus 60.

[0027] Fuel injection valves (injection valves, injectors) 21 are disposed above the individual cylinders of the engine main body 20. The fuel injection valves 21 are connected via a fuel line 23 to a fuel injection pump 22 connected to an unillustrated fuel tank. Moreover, a glow plug 24 is disposed above each cylinder to be located adjacent to the fuel injection valve 21. Each glow plug 24 is electrically connected to the electronic control apparatus 60. The glow plug 24 generates heat upon receipt of electricity in accordance with a signal from the electronic control apparatus 60 only when the engine is in a predetermined operating state such as a warming up state, so as to supply a predetermined quantity of heat to cylinder interior gas present in each cylinder.

[0028] The fuel injection pump 22 is electrically connected to the electronic control apparatus 60. In accordance with a drive signal from the electronic control apparatus 60 (an instruction signal corresponding to an (instruction) base fuel injection pressure Pcrbase to be described later), the fuel injection pump 22 pressurizes fuel in such a manner that the actual injection pressure (discharge pressure) of fuel becomes equal to the instruction base fuel injection pressure Pcrbase.

[0029] Thus, fuel pressurized to the base fuel injection pressure Pcrbase is supplied from the fuel injection pump 22 to the fuel injection valves 21. Moreover, the fuel injection valves 21 are electrically connected to the electronic control apparatus 60. In accordance with a drive signal (an instruction signal corresponding to an (instruction) fuel injection quantity (mass) qfin to be described later) from the electronic control apparatus 60, each of the fuel injection valves 21 opens for a predetermined period of time so as to inject, directly to the combustion chamber of the corresponding cylinder, the fuel pressurized to the instruction base fuel injection pressure Pcrbase, in the instruction fuel injection quantity qfin.

[0030] The intake system 30 includes an intake manifold 31, which is connected to the respective combustion chambers of the individual cylinders of the engine main body 20; an intake pipe 32, which is connected to an upstream-side branching portion of the intake manifold 31 and constitutes an intake passage in cooperation with the intake manifold 31; a throttle valve 33, which is rotatably held within the intake pipe 32; a throttle valve actuator 33a for rotating the throttle valve 33 in accordance with a drive signal from the electronic control apparatus 60; an intercooler 34, which is interposed in the intake pipe 32 to be located on the upstream side of the throttle valve 33; a compressor 35a of a turbocharger 35, which is interposed in the intake pipe 32 to be located on the upstream side of the intercooler 34; and an air cleaner 36, which is disposed at a distal end portion of the intake pipe 32.

[0031] The exhaust system 40 includes an exhaust manifold 41, which is connected to the individual cylinders of the engine main body 20; an exhaust pipe 42, which is connected to a downstream-side merging portion of the exhaust manifold 41; a turbine 35b of the turbocharger 35 interposed in the exhaust pipe 42; and a diesel particulate filter (hereinafter referred to as "DPNR") 43, which is interposed in the exhaust pipe 42. The exhaust manifold 41 and the exhaust pipe 42 constitute an exhaust passage.

[0032] The DPNR 43 is a filter unit which accommodates a filter 43a formed of a porous material such as cordierite and which collects, by means of a porous surface, the particulate matter contained in exhaust gas passing through the filter. In the DPNR 43, at least one metal element selected from alkaline metals such as potassium K, sodium Na, lithium Li, and cesium Cs; alkaline-earth metals such as barium Ba and calcium Ca; and rare-earth metals such as lanthanum La and yttrium Y is carried, together with platinum, on alumina serving as a carrier. Thus, the DPNR 43 also serves as a storage-reduction-type NO_x catalyst unit which, after absorption of NO_x, releases the absorbed NO_x and reduces it.

[0033] The EGR apparatus 50 includes an exhaust circulation pipe 51, which forms a passage (EGR passage) for circulation of exhaust gas; an EGR control valve 52, which is interposed in the exhaust circulation pipe 51; and an EGR cooler 53. The exhaust circulation pipe 51 establishes communication between an exhaust passage (the exhaust manifold 41) located on the upstream side of the turbine 35b, and an intake passage (the intake manifold 31) located on the downstream side of the throttle valve 33. The EGR control valve 52 responds to a drive signal from the electronic control apparatus 60 so as to change the quantity of exhaust gas to be circulated (exhaust-gas circulation quantity, EGR-gas flow rate).

[0034] The electronic control apparatus 60 is a microcomputer which includes a CPU 61, ROM 62, RAM 63, backup RAM 64, an interface 65, etc., which are connected to one another by means of a bus. The ROM 62 stores a program to be executed by the CPU 61, tables (lookup tables, maps), constants, etc. The RAM 63 allows the CPU 61 to temporarily store data when necessary. The backup RAM 64 stores data in a state in which the power supply is on, and holds the stored data even after the power supply is shut off. The interface 65 contains A/D converters.

[0035] The interface 65 is connected to a hot-wire-type airflow meter 71, which serves as air flow rate (new air flow rate) measurement means, and is disposed in the intake pipe 32; an intake gas temperature sensor 72, which is provided in the intake passage to be located downstream of the throttle valve 33 and downstream of a point where the exhaust circulation pipe 51 is connected to the intake passage; an intake pipe pressure sensor 73, which is provided in the intake passage to be located downstream of the throttle valve 33 and downstream of the point where the exhaust circulation pipe 51 is connected to the intake passage; a crank position sensor 74; an accelerator opening sensor 75; and an intake-gas oxygen concentration sensor 76 provided in the intake passage to be located downstream of the throttle valve 33 and downstream of the point where the exhaust circulation pipe 51 is connected to the intake passage. The interface 65 receives respective signals from these sensors, and supplies the received signals to the CPU 61. Further, the interface 65 is connected to the fuel injection valves 21, the fuel injection pump 22, the throttle valve actuator 33a, and the EGR control valve 52; and outputs corresponding drive signals to these components in accordance with instructions from the CPU 61.

[0036] The hot-wire-type airflow meter 71 measures the mass flow rate of intake gas (new air) passing through the intake passage (intake new air quantity per unit time), and generates a signal indicating the mass flow rate Ga (intake new air flow rate Ga). The intake gas temperature sensor 72 detects the temperature of the above-mentioned intake gas, and generates a signal representing the intake gas temperature Tb. The intake pipe pressure sensor 73 measures the pressure of intake gas (i.e., intake pipe pressure), and generates a signal representing the intake pipe pressure Pb.

[0037] The crank position sensor 74 detects the absolute crank angle of each cylinder, and generates a signal representing the crank angle CA and engine speed NE; i.e., rotational speed of the engine 10. The accelerator opening sensor 75 detects an amount by which an accelerator pedal AP is operated, and generates a signal representing the accelerator pedal operated amount Accp. The intake-gas oxygen concentration sensor 76 detects the oxygen concentration of intake gas (i.e., intake-gas oxygen concentration), and a signal representing intake-gas oxygen concentration RO2_in.

<u>Outline of Combustion Temperature Estimation Method</u>

[0038] Next, there will be described an outline of a combustion temperature method according to the embodiment of the present invention performed by the control apparatus of the internal combustion engine having the above-described configuration (hereinafter may be referred to as the "present apparatus"). FIG. 2 is a diagram schematically showing a state in which gas (intake gas) is taken from the intake manifold 31 into a certain cylinder (cylinder interior) of the engine 10 and is then discharged to the exhaust manifold 41 after combustion.

[0039] As shown in FIG. 2, intake gas (accordingly, cylinder interior gas) includes new air taken from the tip end of the intake pipe 32 via the throttle valve 33, and EGR gas taken from the exhaust circulation pipe 51 via the EGR control valve 52. The mass ratio (i.e., EGR ratio) of the mass of the taken EGR gas (EGR gas mass) to the sum of the mass of the taken new air (new air mass) and the mass of the taken EGR gas (EGR gas mass) changes depending on the opening of the throttle valve 33 and the opening of the EGR control valve 52, which are properly controlled by the electronic control apparatus 60 (CPU 61) in accordance with the operating condition.

[0040] During an intake stroke, the intake gas (i.e., gas composed of the new air and the EGR gas) is taken in the cylinder via an opened intake valve Vin as the piston moves downward, and the thus-produced gas mixture serves as cylinder interior gas. The cylinder interior gas is confined within the cylinder when the intake valve Vin closes upon the piston having reached bottom dead center (hereinafter referred to as "ATDC-180°"), and then compressed in a subsequent compression stroke as the piston moves upward. When the piston reaches the vicinity of compression top dead center (hereinafter referred to as "ATDC0°")(specifically, when a final fuel injection timing finjfin to be described later comes), the present apparatus opens the corresponding fuel injection valve 21 for a predetermined period of time corresponding to the instruction fuel injection quantity qfin, to thereby inject fuel directly into the cylinder. As a result, the injected fuel disperses in the cylinder with elapse of time, while mixing with the cylinder interior gas to produce a gas mixture. After elapse of a predetermined ignition delay time, the gas mixture starts combustion by means of self ignition.

[0041] In the present embodiment, such combustion is assumed to occur only in a combustion region (hereinafter may be referred to as "region B"; see FIG. 2), which is a portion of the combustion chamber and is estimated as described later, and not to occur in a non-combustion region (hereinafter may be referred to as "region A"; see FIG. 2), which is the remaining portion of the combustion chamber other than the region B. Cylinder interior gas remaining in the combustion chamber after combustion is discharged, as exhaust gas, to the exhaust manifold 41 via the exhaust valve Vout, which is held open during the exhaust stroke, as the piston moves upward. A portion of the exhaust gas is circulated to the intake side as EGR gas via the exhaust circulation pipe 51, and the remaining exhaust gas is discharged to the outside via the exhaust pipe 42.

[0042] Next, a specific combustion temperature estimation method performed by the present apparatus will be described. In this combustion temperature estimation method, immediately upon arrival of each final fuel injection timing finjfin for a cylinder to which fuel is injected (hereinafter referred to as "fuel injection cylinder"), the highest combustion temperature Tflame of the cylinder interior gas generated as a result of combustion in the region B is estimated immediately after the arrival (after elapse of the above-mentioned ignition delay time). The present apparatus obtains the highest combustion temperature Tflame in accordance with the following Eq. (1).

$$Tflame = Tpump + \Delta Tburn + \Delta Tb\_velo \qquad (1)$$

[0043] In Eq. (1), Tpump represents ignition-time compressed cylinder interior gas temperature; i.e., the pre-combustion temperature of cylinder interior gas at the time of ignition. ΔTburn represents a combustion ascribable temperature increase; i.e., an increase in temperature of cylinder interior gas ascribable to combustion. ΔTb_velo represents an increase in temperature of the cylinder interior gas ascribable to an increase in combustion speed. Next, methods for obtaining these values will be described individually.

<Obtainment of Ignition-Time Compressed Cylinder Interior Gas Temperature Tpump>

[0044] For obtainment of the ignition-time compressed cylinder interior gas temperature Tpump, it is first assumed that no heat exchange occurs between cylinder interior gas and the outside in compression and expression strokes. In this case, since the status of the cylinder interior gas changes adiabatically, as indicated by a solid line in FIG. 3, cylinder interior gas temperature Ta changes with crank angle CA (ATDC), and the cylinder interior gas temperature at the time of ignition (i.e., the ignition-time compressed cylinder interior gas temperature Tpump) can be obtained in accordance with the following Eq. (2), which is a general formula for representing adiabatic changes.

$$Tpump = Ta0 \cdot (Va0/Vig)^{\kappa-1} \qquad (2)$$

[0045] In Eq. (2), Ta0 represents cylinder interior gas temperature at ATDC-180°; i.e., bottom-dead-center cylinder interior gas temperature. At ATDC-180°, the cylinder interior gas temperature is considered to be substantially equal to the intake temperature Tb. Therefore, the bottom-dead-center cylinder interior gas temperature Ta0 can be obtained as the intake temperature Tb detected by means of the intake temperature sensor 72 at ATDC-180°. Va0 represents cylinder interior volume at ATDC-180°; i.e., bottom-dead-center cylinder interior volume. Since the cylinder interior volume Va can be represented in the form of a function Va(CA) of crank angle CA on the basis of the design specifications of the engine 10, the bottom-dead-center cylinder interior volume Va0 can be obtained on the basis of this function.

[0046] In Eq. (2), Vig represents cylinder interior volume at the time of ignition. Since the ignition time is a point in time after passage of a predetermined ignition delay time from a corresponding fuel injection timing, as shown in FIG. 3, a crank angle CAig at the time of ignition can be obtained through addition of a crank angle ΔCAdelay, which corresponds to the above-mentioned ignition delay time, to a fuel injection crank angle CAinj corresponding to the above-mentioned final fuel injection timing finjfin. Accordingly, the cylinder interior volume Vig at the time of ignition can be obtained as Va(CAig).

[0047] In Eq. (2), κ represents a politropic index. The politropic index κ, to be used with the cylinder interior gas which causes adiabatic changes, changes depending on the composition of intake gas (accordingly, the composition of cylinder interior gas). In the present example, the politropic index κ can be obtained as g(RO2c), where RO2c represents bottom-dead-center intake-gas oxygen concentration; i.e., the intake-gas oxygen concentration RO2_in detected by the intake-gas oxygen concentration sensor 76 (specifically, the intake-gas oxygen concentration RO2_in detected at bottom dead center (ATDC-180°); and g represents a function for obtaining the politropic index from the intake-gas oxygen concentration.

[0048] As described above, in principle, the present apparatus obtains the ignition-time compressed cylinder interior gas temperature Tpump in accordance with Eq. (2). As a result, as indicated by the solid line in FIG. 3, the ignition-time compressed cylinder interior gas temperature Tpump increases with a delay in ignition time when the ignition time (ignition-time crank angle CAig) is before a point in time corresponding to the compression top dead center (ATDC0°) (i.e., when the ignition time falls in the compression stroke). Meanwhile, the ignition-time compressed cylinder interior gas temperature Tpump decreases with a delay in ignition time when the ignition time is after the point in time corresponding to the compression top dead center (i.e., when the ignition time falls in the expansion stroke).

[0049] Incidentally, when the ignition time is before a point in time corresponding to the compression top dead center, the cylinder interior gas having been ignited (i.e., during or after combustion) is further compressed up to the point in time corresponding to the compression top dead center, and as a result, the highest combustion temperature Tflame, which a final value to be calculated in accordance with the above-described Eq. (1), is considered to increase up to the point in time corresponding to the compression top dead center. That is, in this case, the highest combustion temperature Tflame is preferably made equal to the highest combustion temperature for the case where the ignition time coincides with the point in time corresponding to the compression top dead center.

[0050] In view of the above, when the ignition time (i.e., the ignition-time crank angle CAig) is before the point in time corresponding to the compression top dead center, under the assumption that the ignition time coincides with the point in time corresponding to the compression top dead center, the present apparatus obtains the ignition-time compressed cylinder interior gas temperature Tpump in accordance with the above-mentioned Eq. (2) while using, as the value of the ignition-time cylinder interior volume Vig, a cylinder interior volume Vtop (const value) at ATDC0° in place of the value of Va(CAig). As a result, as indicated by a broken line in FIG. 3, the ignition-time compressed cylinder interior gas temperature Tpump in this case is estimated as a cylinder interior gas temperature Tal (constant value) at compression top dead center.

[0051] In the above, the description has been provided for the case where no heat exchange occurs between cylinder interior gas and the outside in the compression and expression strokes. However, in actuality, during a period in which electricity is supplied to the glow plug 24, the cylinder interior gas receives a predetermined quantity of heat from the glow plug 24 in the compression and expression strokes, and consequently, the cylinder interior gas temperature at the time of ignition (i.e., the ignition-time compressed cylinder interior gas temperature Tpump) increases by an amount corresponding to the quantity of heat.

[0052] In view of the above, during a period in which electricity is supplied to the glow plug 24, the present apparatus adds a cylinder interior gas temperature increase ΔTpump (= Tglow (constant value in the present example)) corresponding to the above-mentioned heat quantity to the ignition-time compressed cylinder interior gas temperature Tpump obtained in the above-described manner so as to estimate the final value of the ignition-time compressed cylinder interior gas temperature. As described above, upon arrival of each final fuel injection timing finjfin, the ignition-time compressed cylinder interior gas temperature Tpump is estimated on basis of at least the above-mentioned Eq. (2) regarding adiabatic compression, the intake-gas temperature Tb at bottom dead center (ATDC-180°), the composition of the intake gas (the above-mentioned bottom-dead-center intake-gas oxygen concentration R02c), and the above-mentioned cylinder interior gas temperature increase ΔTpump.

<Obtainment of Combustion Ascribable Temperature Increase ΔTburn>

[0053] In order to obtain a combustion ascribable temperature increase ΔTburn, which is an increase in cylinder interior gas temperature ascribable to combustion, combustion of 1 mol of injected fuel is considered. In this case, the combustion ascribable temperature increase ΔTburn can be represented by the following Eq. (3).

$$\Delta Tburn = Qfuel/(Cp \cdot ngas) \tag{3}$$

[0054] In Eq. (3), Qfuel represents a quantity of heat generated as a result of combustion of 1 mol of injected fuel, and is a known value (constant) which is univocally determined by the composition of the fuel. Cp represents a post-combustion constant-pressure specific heat of cylinder interior gas involved in the combustion of 1 mol of injected fuel. ngas represents the post-combustion mole amount of cylinder interior gas involved in the combustion of 1 mol of injected fuel. Accordingly, in order to obtain the combustion ascribable temperature increase ΔTburn in accordance with Eq. (3), the above-mentioned constant-pressure specific heat Cp and the above-mentioned mole amount ngas must be obtained. Here, a chemical reaction per mol of combustion of fuel (C_mH_n) containing typical gas components of the intake gas is represented by the following Eq. (4).

$$C_mH_n + (m + \tfrac{n}{4})O_2 + \alpha CO_2 + \beta H_2O + \gamma N_2 \rightarrow mCO_2 + \tfrac{n}{2}H_2O + \alpha CO_2 + \beta H_2O + \gamma N_2 \tag{4}$$

[0055] In Eq. (4), m and n are values univocally determined on the basis of the composition of fuel to be injected. For example, in the case of a typical fuel used for diesel engines, m = 15.75 and n = 30.55. α, β, and γ respectively represent the mole amounts of inert gases CO_2, H_2O, and N_2 involved in the combustion of fuel of 1 mol. That is, in the present embodiment, four components; i.e., O_2, CO_2, H_2O, and N_2, are considered as the gas components of the intake gas. As is easily understood from the right side of Eq. (4), the above-mentioned post-combustion mole amount ngas and constant-pressure specific heat Cp (equivalent constant-pressure specific heat) of cylinder interior gas involved in the combustion of 1 mol of injected fuel can be represented by the following Eqs. (5) and (6), respectively. In Eq. (6), Cp_CO_2, Cp_H_2O, and Cp_N_2 represent the constant-pressure specific heats of CO_2, H_2O, and N_2, respectively.

$$\mathrm{ngas} = m + \frac{n}{2} + \alpha + \beta + \gamma \tag{5}$$

$$C_p = \frac{1}{n_{gas}}\left\{(m+\alpha)C_{p\_CO_2} + \left(\frac{n}{2}+\beta\right)C_{p\_H_2O} + \gamma C_{p\_N_2}\right\} \tag{6}$$

[0056] As is apparent from Eqs. (5) and (6), in order to obtain the above-mentioned mole amount ngas and constant-pressure specific heat Cp, the above-mentioned mole amounts α, β, and γ must be obtained. Meanwhile, these values change in accordance with the composition of intake gas (specifically, the concentrations (proportions) of the above-mentioned four components that constitute intake gas). Therefore, the concentrations (proportions) of the above-mentioned four components in intake gas must be obtained in order to obtain the values of α, β, and γ. Methods for obtaining the respective concentrations of the above-mentioned four components in intake gas will now be described.

[0057] In the following description, $[X]_{in}$, $[X]_{egr}$, and $[X]_{air}$ (X: O_2, CO_2, H_2O, N_2) represent the mass concentration of component X in intake gas, the mass concentration of component X in EGR gas, and the mass concentration of component X in new air, respectively. The $[X]_{air}$ is assumed to have a known value (e.g., $[CO2]_{air}$ = $[H2O]_{air}$ = 0, [02] air = 0.233, and $[N_2]_{air}$ = 0.767).

[0058] In the following description, Gcyl represents the total mass of intake gas taken in a cylinder in a single intake stroke (hereinafter referred to as "cylinder interior total gas quantity Gcyl"); Gegr represents the mass of EGR gas taken from the EGR apparatus 50 into the cylinder in a single intake stroke as a part of intake gas (hereinafter referred to as "EGR gas quantity"); and Gm represents the mass of new air taken from the end portion of the intake pipe 32 into the cylinder in a single intake stroke as a part of intake gas (hereinafter referred to as "intake new air quantity").

[0059] The cylinder interior total gas quantity Gcyl can be obtained in accordance with the following Eq. (7), which is based on the gas state equation at ATDC-180°.

$$\mathrm{Gcyl} = (\mathrm{Pa0}\cdot\mathrm{Va0})/(\mathrm{R}\cdot\mathrm{Ta0}) \tag{7}$$

[0060] In Eq. (7), Pa0 represents bottom-dead-center cylinder interior gas pressure; i.e., cylinder interior gas pressure at ATDC-180°. At ATDC-180°, the cylinder interior gas pressure is considered to be substantially equal to the intake pipe pressure Pb. Therefore, the bottom-dead-center cylinder interior gas pressure Pa0 can be obtained from the intake pipe pressure Pb detected by means of the intake pipe pressure sensor 73 at ATDC-180°. R represents a gas constant of cylinder interior gas. Ta0 and Va0 represent bottom-dead-center cylinder interior gas temperature and bottom-dead-center combustion chamber volume, respectively, as in the case of Eq. (2).

[0061] The intake new air quantity Gm can be calculated on the basis of the intake new air quantity per unit time (intake new air flow rate Ga) measured by means of the airflow meter 71, the engine speed NE based on the output of the crank position sensor 74, and a function f(Ga, NE) which uses the intake new air flow rate Ga and the engine speed NE, as arguments, so as to obtain quantity of intake new air per intake stroke. A bottom-dead-center intake new air flow rate Ga0 and a bottom-dead-center engine speed NE0, which are detected by the corresponding sensors at ATDC-180°, are used as the intake new air flow rate Ga and the engine speed NE, respectively.

[0062] Since the above-mentioned cylinder interior total gas quantity Gcyl is the sum of the above-mentioned EGR gas quantity Gegr and intake new air quantity Gm, the EGR gas quantity Gegr can be obtained in accordance with the following Eq. (8) on the basis of the cylinder interior total gas quantity Gcyl and the intake new air quantity Gm, which are obtained in the above-described manner.

$$\mathrm{Gegr} = \mathrm{Gcyl} - \mathrm{Gm} \tag{8}$$

[0063] First, a method of obtaining the concentration $[CO_2]_{in}$ of CO_2 contained in intake gas (hereinafter referred to as "intake-gas CO_2 concentration") will be described. Since the intake-gas CO_2 concentration is equal to the CO_2 concentration of cylinder interior gas taken into a cylinder but not having undergone combustion, the intake-gas CO_2 concentration can be obtained in accordance with the following Eq. (9) as a mass ratio of the "sum of the mass $[CO_2]_{air}\cdot Gm$ of CO_2 contained in new air taken into the cylinder and the mass $[CO_2]_{egr}\cdot Gegr$ of CO_2 contained in EGR gas taken into the cylinder" to the cylinder interior total gas quantity Gcyl.

$$[CO_2]_{in} = \frac{[CO_2]_{air}G_m + [CO_2]_{egr}\,G_{egr}}{Gcyl} \tag{9}$$

[0064] The concentration $[CO_2]_{egr}$ of CO_2 contained in EGR gas is considered to be equal to the concentration of CO_2 contained in exhaust gas (passing through the exhaust valve Vout), and the concentration of CO_2 contained in the exhaust gas is equal to the CO_2 concentration of cylinder interior gas after combustion. The CO_2 concentration of cylinder interior gas is a mass ratio of the "sum of the mass $[CO_2]_{air}\cdot Gm$ of CO_2 contained in new air taken into the cylinder, the mass $[CO_2]_{egr}\cdot Gegr$ of CO_2 contained in EGR gas taken into the cylinder, and the mass of CO_2 generated as a result of combustion" to the "sum of the cylinder interior total gas quantity Gcyl and the instruction fuel injection quantity (mass) qfinc (= qfin) for the present operation cycle."

[0065] The mass of CO_2 generated as a result of combustion can be represented as $KCO_2\cdot qfinc$, where KCO_2 represents the mass of CO_2 generated as a result of combustion of fuel of unit quantity. Further, as can be understood from the above-described Eq. (4), CO_2 of m mol is generated as a result of combustion of fuel (C_mH_n) of 1 mol. Therefore, when the molecular weight of fuel (C_mH_n) is represented by Mfuel and the molecular weight of CO_2 by MCO_2 (= 44), combustion of fuel having a mass Mfuel results in generation of CO_2 having a mass ($m\cdot MCO_2$). Accordingly, the value of $KCO_2\cdot$can be obtained from $m\cdot(MCO_2/Mfuel)$. From the above, the concentration $[CO_2]_{egr}$ of CO_2 contained in EGR gas can be obtained in accordance with the following Eq. (10). When Eq. (10) is rearranged, the following Eq. (11) can be obtained.

$$[CO_2]_{egr} = \frac{KCO_2\cdot qfinc + [CO_2]_{air}\,G_m + [CO_2]_{egr}\,G_{egr}}{Gcyl + qfinc} \tag{10}$$

$$[CO_2]_{egr} = \frac{KCO_2\cdot qfinc + [CO_2]_{air}\,G_m}{G_m + qfinc} \tag{11}$$

[0066] When Eq. (11) is substituted in Eq. (9), the following Eq. (12) can be obtained. The intake-gas CO_2 concentration $[CO_2]_{in}$ can be obtained in accordance with Eq. (12). In Eq. (12), R represents an EGR ratio (= Gegr/Gcyl).

$$[CO_2]_{in} = [CO_2]_{air}\cdot(1-R) + \frac{KCO_2\cdot qfinc + [CO_2]_{air}G_m}{G_m + qfinc}\,R \tag{12}$$

[0067] Next, a method of obtaining the concentration $[H_2O]_{in}$ of H_2O contained in the intake gas (hereinafter referred to as "intake-gas H_2O concentration") will be described. The intake-gas H_2O concentration $[H_2O]_{in}$ can be obtained in the same manner as in the above-described method of obtaining the intake-gas CO_2 concentration $[CO_2]_{in}$ with the $[CO_2]$ in Eqs. (9) to (12) replaced with $[H_2O]$ and KCO_2 in Eqs. (9) to (12) with KH_2O. That is, the intake-gas H_2O concentration $[H_2O]_{in}$ can be obtained in accordance with the following Eq. (13).

$$[H_2O]_{in} = [H_2O]_{air}\cdot(1-R) + \frac{KH_2O\cdot qfinc + [H_2O]_{air}G_m}{G_m + qfinc}\,R \tag{13}$$

[0068] In Eq. (13), KH_2O represents the mass of H_2O generated as a result of combustion of a unit quantity of fuel. As can be understood from the above-described Eq. (4), H_2O of (n/2) mol is generated as a result of combustion of fuel (C_mH_n) of 1 mol. Therefore, when the molecular weight of fuel (C_mH_n) is represented by Mfuel and the molecular weight of H_2O by MH_2O (= 18), combustion of fuel having a mass Mfuel results in generation of H_2O having a mass $\{(n/2)\cdot MH_2O\}$. Accordingly, the value of KH_2O can be obtained from $(n/2)\cdot(MH_2O/Mfuel)$.

[0069] Next, a method of obtaining the concentration $[N_2]_{in}$ of N_2 contained in the intake gas (hereinafter referred to as "intake-gas N_2 concentration") will be described. N_2 is neither consumed nor generated in the cylinder as a result of combustion of fuel (C_mH_n). Accordingly, the intake-gas N_2 concentration $[N_2]_{in}$ can be obtained in accordance with Eqs. (9) to (12), with the $[CO_2]$ in Eqs. (9) to (12) replaced with $[N_2]$ and the terms of KCO_2 in Eqs. (9) to (12) removed. That is, the intake-gas N_2 concentration $[N_2]_{in}$ can be obtained in accordance with the following Eq. (14).

$$[N_2]_{in} = [N_2]_{air} \cdot (1-R) + \frac{[N_2]_{air} G_m}{G_m + qfinc} R \tag{14}$$

[0070] Next, a method of obtaining the concentration $[O_2]$in of O_2 contained in the intake gas (hereinafter referred to as "intake-gas O_2 concentration") will be described. O_2 contained in the cylinder interior gas is consumed in the cylinder as a result of combustion of fuel (C_mH_n). The mass of O_2 consumed as a result of combustion can be represented by $KO_2 \cdot qfinc$, where KO_2 represents the mass of O_2 consumed as a result of combustion of a unit quantity of fuel. Accordingly, the intake-gas O_2 concentration $[O_2]_{in}$ can be obtained in accordance with Eqs. (9) to (12), with the $[CO_2]$ in Eqs. (9) to (12) replaced with $[O_2]$ and KCO_2 in Eqs. (9) to (12) with $-KO_2$. That is, the intake-gas O_2 concentration $[O_2]_{in}$ can be obtained in accordance with the following Eq. (15).

$$[O_2]_{in} = [O_2]_{air} \cdot (1-R) + \frac{[O_2]_{air} G_m - KO_2 \cdot qfinc}{G_m + qfinc} R \tag{15}$$

[0071] As can be understood from the above-described Eq. (4), O_2 of (m+n/4) mol is consumed as a result of combustion of 1 mol of fuel (C_mH_n). Therefore, when the molecular weight of fuel (C_mH_n) is represented by Mfuel and the molecular weight of O_2 by MO_2 (= 32), combustion of fuel having a mass Mfuel results in consumption of O_2 having a mass $\{(m+n/4) \cdot MO_2\}$. Accordingly, the value of KO_2 in Eq. (15) can be obtained from $(m+n/4) \cdot (MO_2/Mfuel)$.

[0072] When KO_2 and Gm are removed from Eq. (15) by making use of the relation the stoichiometric air-fuel ratio stoich = $KO_2/[O_2]_{air}$ and the relation the excessive air ratio λ = Gm/(stoich·qfinc), the following Eq. (16) can be obtained. Further, when the relation $[(1/\text{stoich})] \cong 0$ is taken into consideration in Eq. (16), the intake-gas O_2 concentration$[O_2]_{in}$ can be obtained in accordance with Eq. (17).

$$[O_2]_{in} = [O_2]_{air} \left\{ 1 - R \frac{1 + \frac{1}{\text{stoich}}}{\lambda + \frac{1}{\text{stoich}}} \right\} \tag{16}$$

$$[O_2]_{in} = [O_2]_{air} \left\{ 1 - \frac{R}{\lambda} \right\} \tag{17}$$

[0073] In the above-described manner, the mass concentration $[X]_{in}$ (X: O_2, CO_2, H_2O, N_2) of the above-mentioned four components contained in the intake gas can be obtained. Next, a method of obtaining the above-mentioned mole amounts α, β, and γ by use of the thus-obtained mass concentrations will be described. Once the mass concentrations of the four components contained in the intake gas are obtained, proportions (on the basis of concentration by mass) of the four components in the intake gas, $[O_2]_{in} : [CO_2]_{in} : [H_2]_{in} : [N_2]_{in}$, can be obtained. The proportions of the four components in the intake gas are assumed to be maintained as proportions of the four components in cylinder interior gas; i.e., gas taken into a cylinder (more specifically, a gas present in the above-mentioned combustion region (region B)).

[0074] Meanwhile, the proportions of the four components in the cylinder interior gas are equal to those of the four components in the cylinder interior gas. Moreover, as can be understood from the above-mentioned Eq. (4), since the molar ratio of the four components in the cylinder interior gas is $(m+n/4) : \alpha : \beta : \gamma$, the mass ratio of the four components in the cylinder interior gas can be represented as $(m+n/4)MO_2 : \alpha MCO_2 : \beta MH_2O : \gamma MN_2$. From the above, the following Eq. (18) can be obtained.

$$(m + \frac{n}{4})MO_2 : \alpha MCO_2 : \beta MH_2O : \gamma MN_2 = [O_2]_{in} : [CO_2]_{in} : [H_2O]_{in} : [N_2]_{in} \quad (18)$$

[0075] Use of Eq. (18) enables the above-mentioned mole amounts α, β, and γ to be obtained in accordance with the following Eqs. (19) to (21), respectively.

$$\alpha = \frac{[CO_2]_{in}}{[O_2]_{in}} \frac{MO_2}{MCO_2}(m + \frac{n}{4}) \quad (19)$$

$$\beta = \frac{[H_2O]_{in}}{[O_2]_{in}} \frac{MO_2}{MH_2O}(m + \frac{n}{4}) \quad (20)$$

$$\lambda = \frac{[N_2]_{in}}{[O_2]_{in}} \frac{MO_2}{MN_2}(m + \frac{n}{4}) \quad (21)$$

[0076] Once the above-mentioned mole amounts α, β, and γ are obtained, the above-mentioned mole amount ngas and constant-pressure specific heat Cp can be obtained in accordance with the above-mentioned Eqs. (5) and (6). As a result, the combustion ascribable temperature increase ΔTburn can be obtained in accordance with the above-mentioned Eq. (3).

[0077] As described above, upon arrival of each final fuel injection timing finjfin, the present apparatus obtains the combustion ascribable temperature increase ΔTburn by use of the above-mentioned Eqs. (3) to (21) and on the basis of at least the composition of intake gas (the concentration proportions of gas components contained in intake gas) and the quantity (Qfuel) of heat generated as a result of combustion of injected fuel.

<Obtainment of Combustion-Speed Ascribable Temperature Increase ΔTb_velo>

[0078] A predetermined period of time (hereinafter referred to as "highest-temperature reaching time") is needed for the cylinder interior gas temperature Ta to increase by the above-mentioned combustion ascribable temperature increase ΔTburn from the temperature at the time of ignition (i.e., ignition-time compressed cylinder interior gas temperature Tpump) and to reach a temperature of (Tpump + ΔTburn) (i.e., reach the highest combustion temperature Tflame). Meanwhile, in the case where the time of ignition is after the point in time corresponding to the compression top dead center (i.e., the time of ignition falls in the expansion stroke), the cylinder interior gas temperature Ta decreases with time because of an increase in the cylinder interior volume Va (an increase in the cylinder interior gas temperature Ta stemming from combustion is suppressed).

[0079] Accordingly, the shorter the highest-temperature reaching time, the higher the highest combustion temperature Tflame. In other words, the highest combustion temperature Tflame increases with combustion speed in the cylinder after start of combustion. Such a temperature increase of cylinder interior gas corresponds to the combustion-speed ascribable temperature increase ΔTb_velo. Next, a method of obtaining the combustion-speed ascribable temperature increase ΔTb_velo will be described with reference to FIG. 4.

[0080] Although various factors may influence combustion speed in the cylinder, in the present embodiment, fuel injection pressure Pcrc (= the above-mentioned instruction base fuel injection pressure Pcrbase) (for the present operation cycle) and engine speed NE (more specifically, bottom-dead-center engine speed NE0) are selected as influencing factors. Combustion speed increases with the fuel injection pressure Pcrc for the present operation cycle or the bottom-dead-center engine speed NE0.

[0081] Here, a combustion speed when the fuel injection pressure Pcrc for the present operation cycle is a reference fuel injection pressure Pcrref and the bottom-dead-center engine speed NE0 is a reference engine speed NEref is defined as ordinary combustion speed. Further, the highest combustion temperature Tflame of cylinder interior gas is assumed to become equal to the temperature (Tpump + ΔTburn) (hereinafter referred to as "highest combustion temperature Tig for ordinary combustion speed") (i.e., the combustion-speed ascribable temperature increase ΔTb_velo = 0) when the combustion speed in the cylinder is the above-mentioned ordinary combustion speed.

[0082] In this case, a highest temperature reaching time shorting amount Δtadv with respect to the highest temperature reaching time corresponding to the ordinary combustion speed can be obtained by the following Eq. (22), which is a function of the fuel injection pressure Pcrc for the present operation cycle and the bottom-dead-center engine speed NE0.

$$\Delta tadv = t0 + Ka \cdot Pcrc + Kb \cdot NE0 \qquad (22)$$

[0083] In Eq. (22), t0 represents a constant having a negative value, and each of Ka and Kb is a constant having a positive value. The values of t0, Ka, and Kb are set in such a manner that Δtadv becomes zero when the fuel injection pressure Pcrc for the present operation cycle is the reference fuel injection pressure Pcrref and the bottom-dead-center engine speed NE0 is the reference engine speed NEref. Moreover, a Δtadv-corresponding advancing angle ΔCAadv, which is an advancing amount of crank angle CA corresponding to the highest temperature reaching time shorting amount Δtadv obtained in accordance with Eq. (22), can be obtained on the basis of the highest temperature reaching time shorting amount Δtadv, the bottom-dead-center engine speed NE0, and a function h whose arguments are Δtadv and NE0; i.e., as a value of h(Δtadv, NE0).

[0084] Then, as shown in FIG. 4, cylinder interior gas which has the highest combustion temperature for ordinary combustion speed Tig at the above-mentioned ignition-time crank angle CAig (at which the cylinder interior volume becomes the ignition-time cylinder interior volume Vig) is assumed to have been compressed and have a temperature Tadv because of an adiabatic change caused through advancement of the crank angle from the ignition-time crank angle CAig to a corrected ignition-time crank angle CAadv (a crank angle advanced from the ignition-time crank angle CAig by the Δtadv-corresponding advancing angle ΔCAadv), at which the cylinder interior volume becomes a corrected ignition-time cylinder interior volume Vadv (= Va(CAadv)).

[0085] In this case, a value obtained through substation of the above-mentioned highest combustion temperature for ordinary combustion speed Tig from the temperature Tadv can be considered to correspond to an increase in cylinder interior gas temperature (accordingly, the above-mentioned combustion-speed ascribable temperature increase ΔTb_velo) caused by advancement of the time at which the cylinder interior gas temperature reaches the highest combustion temperature Tflame by an amount corresponding to the highest temperature reaching time shorting amount Δtadv, as compared with the case where the combustion speed becomes the ordinary combustion speed. Accordingly, the combustion-speed ascribable temperature increase ΔTb_velo can be obtained in accordance with the following Eq. (23).

$$\Delta Tb\_velo = Tig \cdot (Vig/Vadv)^{k-1} - Tig \qquad (23)$$

[0086] As described above, upon arrival of each final fuel injection timing finjfin, the present apparatus obtains the combustion-speed ascribable temperature increase ΔTb_velo by use of the above-mentioned Eqs. (22) to (23) and on the basis of the factors which influence the cylinder interior combustion speed; i.e., the fuel injection pressure Pcrc for the present operation cycle and the bottom-dead-center engine speed NE0.

[0087] Then, upon arrival of each final fuel injection timing finjfin, the present apparatus estimates the highest combustion temperature Tflame in accordance with the above-mentioned Eq. (1); i.e., from the value obtained through addition of the combustion ascribable temperature increase ΔTburn and the combustion-speed ascribable temperature increase ΔTb_velo to the ignition-time compressed cylinder interior gas temperature Tpump. The above is the outline of the combustion temperature estimation method according to the present invention.

Outline of NO_x Geneation Quantityr Estimation Method

[0088] Next, there will be described a method by which the present apparatus estimates NO_x generation quantity on the basis of the highest combustion temperature Tflame estimated in the above-described manner. In the NO_x generation quantity estimation method, upon each arrival of the final fuel injection timing finjfin for a fuel injection cylinder, actual NO_x generation quantity NOxact (quantity of NO_x generated in the above-mentioned region B as a result of combustion in an explosion (expansion) stoke immediately after the arrival) is estimated.

[0089] The actual NO_x generation quantity NOxact can be obtained in accordance with the following Eq. (24); i.e., by multiplying the quantity of NO_x generated as a result of combustion of fuel of unit quantity (hereinafter referred to as "combustion-generated NOx ratio RNOx_burn") by the above-mentioned instruction fuel injection quantity qfinc for the present operation cycle.

$$NOxact = RNOx\_burn \cdot qfinc \qquad (24)$$

[0090] Here, the combustion generated NOx ratio RNOx_burn in Eq. (24) is estimated by the following Eq. (25). In Eq. (25), e is the base of a natural logarithm. As described above, RO2c represents bottom-dead-center intake-gas oxygen concentration; qfinc represents instruction fuel injection quantity for the present operation cycle; Pcrc represents instruction fuel injection pressure (= Pcrbase) for the present operation cycle, and Tflame represents the highest combustion temperature in the present explosion stroke obtained by the above-described Eq. (1). K0 to K4 are fitting constants which are determined in the manner described below on the basis of typical known multiple regression analysis.

$$RNOx\_burn = e^{K0} \cdot (RO2c)^{K1} \cdot (qfinc)^{K2} \cdot (Pcrc)^{K3} \cdot e^{(K4/Tflame)} \qquad (25)$$

[0091] That is, Eq. (25) is an empirical formula for obtaining the combustion-generated NO_x ratio RNOx_burn. The combustion-generated NO_x ratio RNOx_burn estimated by Eq. (25) is a function of the bottom-dead-center intake-gas oxygen concentration RO2c, the instruction fuel injection quantity qfinc in the present operation cycle, the instruction fuel injection pressure Pcrc in the present operation cycle, and the highest combustion temperature Tflame. More specifically, the combustion-generated NO_x ratio RNOx_burn is calculated on the basis of the product of the power of the bottom-dead-center intake-gas oxygen concentration RO2c, the power of the instruction fuel injection quantity qfinc in the present operation cycle, the power of the instruction fuel injection pressure Pcrc in the present operation cycle, and an exponential function whose exponent is determined in accordance with the highest combustion temperature Tflame.

[0092] The fitting constants K0 to K4 can be determined, for example, through performance of an experiment as follows. That is, first, the engine 10 is operated while the EGR control valve 52 is maintained closed, whereby all the exhaust gas (accordingly, NO_x contained in the exhaust gas) discharged via the exhaust valve Vout is discharged to the outside from the exhaust passage. With this operation, the quantity of NO_x contained in the exhaust gas discharged to the outside from the exhaust passage becomes equal to the actual NO_x generation quantity NOxact, whereby it becomes possible to measure the actual NO_x generation quantity NOxact (accordingly, the combustion-generated NO_x ratio RNOx_burn (= NOxact/qfinc)) through measurement of the NO_x discharge quantity on the basis of output of a predetermined NO_x concentration sensor.

[0093] Next, in this state, the values of the bottom-dead-center intake-gas oxygen concentration RO2c, the instruction fuel injection quantity qfinc in the present operation cycle, the instruction fuel injection pressure Pcrc in the present operation cycle, and the highest combustion temperature Tflame obtained by the above-described Eq. (1) are successively changed so that combinations of the respective values are attained in various predetermined patterns. Subsequently, the combustion-generated NO_x ratio RNOx_burn is successively measured for each pattern.

[0094] Subsequently, the predetermined known multiple regression analysis is performed on the basis of a large number of data sets regarding the relationship between measured values of the combustion-generated NO_x ratio RNOx_burn and the combinations of the above-mentioned respective values, which were obtained as a result of such a work (experiment), whereby the above-mentioned fitting constants K0 to K4 can be obtained. Here, at least the fitting constants K1 to K3 are determined to assume positive values, and the fitting constant K4 is determined to assume a negative value.

[0095] Accordingly, as is understood from Eq. (25), the combustion-generated NO_x ratio RNOx_burn (accordingly, actual NO_x generation quantity NOxact) calculated and estimated in accordance with Eq. (25) increases with an increase in any one of the bottom-dead-center intake-gas oxygen concentration RO2c, the instruction fuel injection quantity qfinc in the present operation cycle, the instruction fuel injection pressure Pcrc in the present operation cycle, and the highest combustion temperature Tflame. This matches the actual phenomena described below.

[0096] First, the actual NO_x generation quantity NOxact increases with the intake-gas oxygen concentration RO2_in. This phenomenon occurs because oxygen is a material for generation of NO_x, and an increase in the quantity of oxygen within the combustion chamber naturally facilitates generation of NO_x.

[0097] The actual NO_x generation quantity NOxact increases with the fuel injection quantity qfin. This phenomenon occurs as follows. When the fuel injection quantity qfin increases, the load of the engine 10 increases, so that the inner wall temperature of the combustion chamber increases. Therefore, the greater the fuel injection quantity qfin (i.e., the greater the load of the engine), the greater the quantity of NO_x that is generated.

[0098] The actual NO_x generation quantity NOxact increases with the fuel injection pressure Pcr. This phenomenon occurs as follows. When the fuel injection pressure Pcr is increased, the injection speed of fuel increases with a resultant increase in the degree of atomization of the fuel, whereby the above-mentioned excess air ratio increases. Therefore, the greater the fuel injection pressure Pcr (i.e., the greater the degree of atomization of injected fuel), the greater the quantity of NO_x that is generated.

[0099] Moreover, the actual NO_x generation quantity NOxact increases with the highest combustion temperature Tflame. This phenomenon occurs because increased gas temperature accelerates a chemical reaction of producing NO_x from nitrogen. As is understood from above, when the combustion-generated NO_x ratio RNOx_burn is calculated in accordance with Eq. (25), the combustion-generated NO_x ratio RNOx_burn can be accurately estimated (thus, the actual NO_x generation quantity NOxact can be accurately estimated in accordance with Eq. (24)) in such a manner that the estimated values follow at least the above-described four actual phenomena. The above is the outline of the NO_x generation quantity estimation method.

<Outline of Fuel Injection Control>

[0100] The present apparatus, which performs the above-mentioned NO_x generation quantity estimation method, calculates, at predetermined intervals, a target NO_x generation quantity per operation cycle NOxt on the basis of the above-mentioned fuel injection quantity qfin and engine speed NE. Subsequently, the present apparatus feedback-controls the final fuel injection timing finjfin and the opening of the EGR control valve 52 in such a manner that the actual NO_x generation quantity NOxact estimated in the previous operation cycle coincides with the target NO_x generation quantity NOxt.

[0101] Specifically, when the actual NO_x generation quantity NOxact estimated in the previous operation cycle is greater than the target NO_x generation quantity NOxt, the final fuel injection timing finjfin to be applied for the fuel injection cylinder in the present operation cycle is delayed from the base fuel injection timing finjbase by a predetermined amount, and the opening of the EGR control valve 52 is increased from the current degree by a predetermined amount. As a result, the highest combustion temperature Tflame of the fuel injection cylinder in the present operation cycle is controlled to decrease, whereby the actual NO_x generation quantity NOxact; i.e., the quantity of NO_x generated in the fuel injection cylinder in the present operation cycle, is rendered coincident with the target NO_x generation quantity NOxt.

[0102] Meanwhile, when the actual NO_x generation quantity NOxact estimated in the previous operation cycle is smaller than the target NO_x generation quantity NOxt, the final fuel injection timing finjfin to be applied for the fuel injection cylinder in the present operation cycle is advanced from the base fuel injection timing finjbase by a predetermined amount, and the opening of the EGR control valve 52 is decreased from the current degree by a predetermined amount. As a result, the highest combustion temperature Tflame of the fuel injection cylinder in the present operation cycle is controlled to increase, whereby the actual NO_x generation quantity NOxact; i.e., the quantity of NO_x discharged from the fuel injection cylinder to the outside in the present operation cycle, is rendered coincident with the target NO_x generation quantity NOxt. The above is the outline of fuel injection control.

<Actual Method of Calculating Combustion-Generated NO_x Ratio RNOx_burn>

[0103] Calculation of the combustion-generated NO_x ratio RNOx_burn performed in accordance with Eq. (25) requires calculation of "power" and "multiplication." However, in general, when calculation of "power" is performed by use of a microcomputer, the calculation load tends to increase; and when calculation of "multiplication" is performed by use of a microcomputer, the calculation accuracy tends to decrease. Therefore, in order to avoid calculation of "power" and "multiplication," the present apparatus (CPU 61) calculates the combustion-generated NO_x ratio RNOx_burn by means of only table search and "addition," while utilizing the following Eq. (26), which is obtained by taking natural logarithms of both sides of Eq. (25).

$$\log(RNOx\_burn) = K0 + K1 \cdot \log(RO2c) + K2 \cdot \log(qfinc) + K3 \cdot \log(Pcrc) + K4/Tflame \qquad (26)$$

[0104] That is, on the basis of tables Maplog1 (R02c), Maplog2(qfinc), Maplog3(Pcrc), and Mapinvpro(Tflame), which are previously stored in the ROM 62 for obtaining the respective values of the second through fifth terms of the right side of Eq. (26), the present apparatus determines respective table search values dataMap1 (= K1·log(RO2c)), dataMap2 (= K2·log(qfinc)), dataMap3 (= K3·log(Pcrc)), and dataMap4 (= K4/Tflame), and then obtains the value of "log(RNOx_burn)" in accordance with the following Eq. (27), which includes "addition calculation" only.

$$\log(RNOx\_burn) = K0 + dataMap1 + dataMap2 + dataMap3 + dataMap4 \qquad (27)$$

[0105] Subsequently, the present apparatus obtains the combustion-generated NO_x ratio RNOx_burn on the basis of a table Mapinvlog(log(RNOx_burn)), which is stored in the ROM 62 in order to obtain the combustion-generated NO_x ratio RNOx_burn from the "log(RNOx_burn)" obtained in accordance with Eq. (27). This calculation procedure reduces the calculation load of the CPU 61 and prevents deterioration of calculation accuracy.

Actual Operation

[0106] Next, actual operations of the control apparatus of the internal combustion engine having the above-described configuration will be described.

<Control of Fuel Injection Quantity, Etc.>

[0107] The CPU 61 repeatedly executes, at predetermined intervals, a routine shown by the flowchart of FIG. 5 and adapted to control fuel injection quantity, etc. Therefore, when a predetermined timing has been reached, the CPU 61 starts the processing from step 500, and then proceeds to step 505 so as to obtain an (instruction) fuel injection quantity qfin from an accelerator opening Accp, an engine speed NE, and a table (map) Mapqfin shown in FIG. 6. The table Mapqfin defines the relation between accelerator opening Accp and engine speed NE, and fuel injection quantity qfin; and is stored in the ROM 62.

[0108] Subsequently, the CPU 61 proceeds to step 510 so as to determine a base fuel injection timing finjbase from the fuel injection quantity qfin, the engine speed NE, and a table Mapfinjbase shown in FIG. 7. The table Mapfinjbase defines the relation between fuel injection quantity qfin and engine speed NE, and base fuel injection timing finjbase; and is stored in the ROM 62.

[0109] Subsequently, the CPU 61 proceeds to step 515 so as to determine a base fuel injection pressure Pcrbase from the fuel injection quantity qfin, the engine speed NE, and a table MapPcrbase shown in FIG. 8. The table MapPcrbase defines the relation between fuel injection quantity qfin and engine speed NE, and base fuel injection pressure Pcrbase; and is stored in the ROM 62.

[0110] Subsequently, the CPU 61 proceeds to step 520 so as to determine a target NO_x generation quantity NOxt from the fuel injection quantity qfin, the engine speed NE, and a table MapNOxt shown in FIG. 9. The table MapNOxt defines the relation between fuel injection quantity qfin and engine speed NE, and target NOx generation quantity NOxt; and is stored in the ROM 62.

[0111] Subsequently, the CPU 61 proceeds to step 525 so as to store, as an NO_x generation quantity deviation ΔNOx, a value obtained through subtraction, from the target NO_x generation quantity NOxt, of the latest actual NO_x generation quantity NOxact, which is computed at a fuel injection timing in a previous operation cycle by a routine to be described later.

[0112] Subsequently, the CPU 61 proceeds to step 530 so as to determine an injection-timing correction value $\Delta\theta$ from the NO_x generation quantity deviation ΔNOx and a table MapΔθ shown in FIG. 10. The table MapΔθ defines the relation between NOx generation quantity deviation ΔNOx and injection-timing correction value $\Delta\theta$, and is stored in the ROM 62.

[0113] Next, the CPU 61 proceeds to step 535 so as to correct the base fuel injection timing finjbase by the injection-timing correction value $\Delta\theta$ to thereby obtain a final fuel injection timing finjfin. Thus, the fuel injection timing is corrected in accordance with the NO_x generation quantity deviation ΔNOx. As is apparent from FIG. 10, when the NO_x generation quantity deviation ΔNOx is positive, the injection-timing correction value $\Delta\theta$ becomes positive, and its magnitude increases with the magnitude of the NO_x generation quantity deviation ΔNOx, whereby the final fuel injection timing finjfin is shifted toward the advance side. When the NO_x generation quantity deviation ΔNOx is negative, the injection-timing correction value $\Delta\theta$ becomes negative, and its magnitude increases with the magnitude of the NO_x generation quantity deviation ΔNOx, whereby the final fuel injection timing finjfin is shifted toward the retard side.

[0114] Subsequently, the CPU 61 proceeds to step 540 so as to determine whether the injection start timing (i.e., the final fuel injection timing finjfin) is reached for the fuel injection cylinder. When the CPU 61 makes a "No" determination in step 540, the CPU 61 proceeds directly to step 595 so as to end the current execution of the present routine.

[0115] In contrast, when the CPU 61 makes a "Yes" determination in step 540, the CPU 61 proceeds to step 545 so as to inject fuel in an amount of the (instruction) fuel injection quantity qfin into the fuel injection cylinder from the fuel injection valve 21 at the base fuel injection pressure Pcrbase. In the subsequent step 550, the CPU 61 determines whether the NO_x generation quantity deviation ΔNOx is positive. When the CPU 61 makes a "Yes" determination in step 550, the CPU 61 proceeds to step 555 so as to reduce the opening of the EGR control valve 52 from the current degree by a predetermined amount. Subsequently, the CPU 61 proceeds to step 570.

[0116] When the CPU 61 makes a "No" determination in step 550, the CPU 61 proceeds to step 560 so as to determine whether the NO_x generation quantity deviation ΔNOx is negative. When the CPU 61 makes a "Yes" determination in step 560, the CPU 61 proceeds to step 565 so as to increase the opening of the EGR control valve 52 from the current degree by a predetermined amount. Subsequently, the CPU 61 proceeds to step 570. When the CPU 61 makes a "No" determination in step 560 (i.e., when the NO_x generation quantity deviation ΔNOx is zero), the CPU 61 proceeds to step 570 without changing the opening of the EGR control valve 52.

[0117] In this manner, the opening of the EGR control valve 52 is changed according to the NO_x generation quantity deviation ΔNOx. In step 570, the CPU 61 stores, as the fuel injection quantity qfinc in the present operation cycle, the fuel injection quantity qfin actually injected. In the subsequent step 575, the CPU 61 stores, as the fuel injection pressure Pcrc in the present operation cycle, the base fuel injection pressure Pcrbase at which fuel was actually injected. Subsequently, the CPU 61 proceeds to step 595 so as to end the current execution of the present routine. Through the above-described processing, control of fuel injection quantity, fuel injection timing, fuel injection pressure, and opening of the EGR control valve 52 is achieved.

<Calculation of Highest Combustion Temperature>

[0118] Meanwhile, the CPU 61 repeatedly executes, at predetermined intervals, a routine shown by the flowchart of FIG. 11 and adapted to calculate highest combustion temperature Tflame. Therefore, when a predetermined timing has been reached, the CPU 61 starts the processing from step 1100, and then proceeds to step 1105 so as to determine whether the crank angle CA at the present point in time coincides with ATDC-180°.

[0119] Description will be continued under the assumption that the crank angle CA at the present point in time has not yet reached ATDC-180°. In this case, the CPU 61 makes a "No" determination in step 1105, and then proceeds directly to step 1145 so as to determine whether the fuel injection start timing (i.e., the final fuel injection timing finjfin) for the fuel injection cylinder has come. Since the crank angle CA at the present point in time has not yet reached ATDC-180°, the CPU 61 makes a "No" determination in step 1145, and then proceeds directly to step 1195 so as to end the current execution of the present routine.

[0120] After that, the CPU 61 repeatedly performs the processing of steps 1100, 1105, 1145, and 1195 until the crank angle CA reaches ATDC-180°. When the crank angle CA has reached ATDC-180°, the CPU 61 makes a "Yes" determination when it proceeds to step 1105, and then proceeds to step 1110. In step 1110, the CPU 61 stores, as bottom-dead-center cylinder interior gas temperature Ta0, bottom-dead-center cylinder interior gas pressure Pa0, bottom-dead-center intake new air flow rate Ga0, and bottom-dead-center engine speed NE0, respectively, the intake gas temperature Tb, the intake pipe pressure Pb, the intake new air flow rate Ga, and the engine speed NE, which are detected by means of the intake gas temperature sensor 72, the intake pipe pressure sensor 73, the airflow meter 71, and the crank position sensor 74, respectively, at the present point in time (ATDC-180°).

[0121] Subsequently, the CPU 61 proceeds to step 1115 so as to store, as bottom-dead-center intake-gas oxygen concentration R02c, the intake-gas oxygen concentration RO2_in detected by means of the intake-gas oxygen concentration sensor 76 at the present point in time (ATDC-180°). In the subsequent step 1120, the CPU 61 computes the cylinder interior total gas quantity Gcyl in accordance with the above-described Eq. (7). Here, the values stored at step 1110 are employed as the bottom-dead-center cylinder interior gas pressure Pa0 and the bottom-dead-center cylinder interior gas temperature Ta0.

[0122] Subsequently, the CPU 61 proceeds to step 1125 so as to compute an intake new air quantity Gm from the bottom-dead-center intake new air flow rate Ga0 and the bottom-dead-center engine speed NE0 in accordance with the above-defined function f. In the subsequent step 1130, the CPU 61 computes an EGR gas quantity Gegr on the basis of the cylinder interior total gas quantity Gcyl computed in step 1120 and the intake new air quantity Gm, and in accordance with the above-described Eq. (8). Subsequently, the CPU 61 proceeds to step 1135 so as to obtain an EGR ratio R on the basis of the above-mentioned intake new air quantity Gm, the above-mentioned EGR gas quantity Gegr, and the equation described in the box of step 1135, and then proceeds to step 1140 so as to obtain an excessive air ratio λ on the basis of the above-mentioned intake new air quantity Gm, the fuel injection quantity qfinc in the present operation cycle stored in the above-mentioned step 570, and the equation described in the box of step 1140. Subsequently, the CPU 61 proceeds to step 1145 so as to make a "No" determination, and then proceeds to step 1195 so as to end the current execution of the present routine.

[0123] After that, the CPU 61 repeatedly performs the processing of steps 1100, 1105, 1145, and 1195 until the fuel injection timing (i.e., the final fuel injection timing finjfin) comes. When the final fuel injection timing finjfin has come, the CPU 61 makes a "Yes" determination in step 1145 and then proceeds to step 1150 so as to calculate the ignition-time crank angle CAig from the above-mentioned final fuel injection timing finjfin and the above-described ignition delay time.

[0124] Subsequently, the CPU 61 proceeds via step 1155 to a routine shown in FIG. 12 and adapted to calculate the ignition-time compressed cylinder interior gas temperature Tpump. That is, the CPU 61 starts the processing from step 1200, and then proceeds to step 1205 so as to obtain the politropic index κ from the latest bottom-dead-center intake-gas oxygen concentration RO2c obtained in the above-described step 1115 and the above-mentioned function g.

[0125] Next, the CPU 61 proceeds to step 1210 so as to determine whether the latest ignition-time crank angle CAig obtained in the above-described step 1150 is delayed from ATDC0°. When the CPU 61 makes a "Yes" determination, it proceeds to step 1215 so as to store a cylinder interior volume corresponding to the ignition-time crank angle CAig as the ignition-time cylinder interior volume Vig. Meanwhile, when the CPU 61 makes a "No" determination in step 1210, it proceeds to step 1220 so as to store a cylinder interior volume Vtop corresponding to the top dead center (ATDC0°) as the ignition-time cylinder interior volume Vig.

[0126] Subsequently, the CPU 61 proceeds to step 1225 so as to determine whether electricity is supplied to the glow plug 24. When the CPU 61 makes a "Yes" determination, it proceeds to step 1230 so as to store the above-mentioned predetermined value Tglow as the cylinder interior gas temperature increase ΔTpump. When the CPU 61 makes a "No" determination, it proceeds to step 1235 so as to set the value of the cylinder interior gas temperature increase ΔTpump to zero.

[0127] The CPU 61 then proceeds to step 1240 so as to obtain the ignition-time compressed cylinder interior gas temperature Tpump on the basis of the latest bottom-dead-center cylinder interior gas temperature Ta0 obtained in the above-described step 1110, the above-mentioned ignition-time cylinder interior volume Vig, the above-mentioned cylinder interior gas temperature increase ΔTpump, and the equation described in the box of step 1240. Subsequently, the CPU 61 proceeds to step 1160 of FIG. 11 via step 1295.

[0128] When the CPU 61 proceeds to step 1160, it executes a routine shown in FIG. 13 and adapted to calculate the combustion ascribable temperature increase ΔTburn. That is, the CPU 61 starts the processing from step 1300, and then proceeds to step 1305 so as to obtain the intake-gas O_2 concentration $[O_2]_{in}$ on the basis of the latest EGR ratio R obtained in the above-described step 1135, the latest excess air ratio λ obtained in the above-described step 1140, and the equation described in the box of step 1305 and corresponding to the above-mentioned Eq. (17).

[0129] Next, the CPU 61 proceeds to step 1310 so as to obtain the intake-gas CO_2 concentration $[CO_2]_{in}$ on the basis of the EGR ratio R, the fuel injection quantity qfinc for the present operation cycle, the latest intake new air quantity Gm obtained in the above-described step 1125, and the equation described in the box of step 1310 and corresponding to the above-mentioned Eq. (12). Similarly, the CPU 61 proceeds to step 1315 so as to obtain the intake-gas H_2O concentration $[H_2O]_{in}$ in accordance with the above-mentioned Eq. (13), and then proceeds to step 1320 so as to obtain the intake-gas N_2 concentration $[N_2]_{in}$ in accordance with the above-mentioned Eq. (14).

[0130] Subsequently, the CPU 61 proceeds to step 1325 so as to obtain the above-described mole amount α on the basis of the intake-gas CO_2 concentration $[CO_2]_{in}$, the intake-gas O_2 concentration $[O_2]_{in}$, and the above-described Eq. (19). Similarly, the CPU 61 proceeds to step 1330 so as to obtain the above-described mole amount β in accordance with the above-mentioned Eq. (20), and then proceeds to step 1335 so as to obtain the above-described mole amount γ in accordance with the above-mentioned Eq. (21).

[0131] After that, the CPU 61 proceeds step 1340 so as to obtain the mole amount ngas of cylinder interior gas after combustion on the basis of the mole amounts α, β, γ obtained in the above-described manner, and the above-mentioned Eq. (5). In subsequent step 1345, the CPU 61 obtains the constant-pressure specific heat Cp of cylinder interior gas after combustion on the basis of the mole amounts α, β, r, and the above-mentioned Eq. (6). The CPU 61 then proceeds to step 1350 so as to obtain the combustion ascribable temperature increase ΔTburn on the basis of the mole amount ngas of cylinder interior gas after combustion, the constant-pressure specific heat Cp of cylinder interior gas after combustion, and the above-mentioned Eq. (3), and then proceeds to step 1165 of FIG. 11 via step 1395.

[0132] When the CPU 61 proceeds to step 1165, it executes a routine shown in FIG. 14 and adapted to calculate the combustion-speed ascribable temperature increase ΔTb_velo. That is, the CPU 61 starts the processing from step 1400, and then proceeds to step 1405 so as to store, as the highest combustion temperature for ordinary combustion speed Tig, a value obtained through addition of the latest combustion ascribable temperature increase ΔTburn obtained in the above-mentioned step 1350 to the latest ignition-time compressed cylinder interior gas temperature Tpump obtained in the above-mentioned step 1240.

[0133] Next, the CPU 61 proceeds to step 1410 so as to obtain the highest temperature reaching time shorting amount Δtadv on the basis of the fuel injection pressure Pcrc in the present operation cycle stored in the above-described step 575, the latest bottom-dead-center engine speed NE0 stored in the above-described step 1110, and the above-mentioned Eq. (22). In subsequent step 1415, the CPU 61 obtains the Δtadv-corresponding advancing angle ΔCAadv on the basis of the highest temperature reaching time shorting amount Δtadv, the bottom-dead-center engine speed NE0, and the above-described function h.

[0134] Subsequently, the CPU 61 proceeds to step 1420 so as to obtain a corrected ignition-time crank angle CAadv by advancing the latest ignition-time crank angle CAig obtained in the above-described step 1150 by the Δtadv-corresponding advancing angle ΔCAadv. In subsequent step 1425, the CPU 61 determines whether the corrected ignition-time crank angle CAadv is delayed from ATDC0°.

[0135] When the CPU 61 makes a "Yes" determination in step 1425, it proceeds to step 1430 so as to store a cylinder interior volume corresponding to the corrected ignition-time crank angle CAadv as a corrected ignition-time cylinder interior volume Vadv. Meanwhile, when the CPU 61 makes a "No" determination in step 1425, it proceeds to step 1435 so as to store a cylinder interior volume Vtop corresponding to the top-dead center (ATDC0°) as the corrected ignition-time cylinder interior volume Vadv.

[0136] Subsequently, the CPU 61 proceeds to step 1440 so as to obtain the combustion-speed ascribable temperature increase ΔTb_velo on the basis of the ignition-time cylinder interior volume Vig obtained in the above-described step 1215 or 1220, the above-mentioned corrected ignition-time cylinder interior volume Vadv, the above-mentioned highest combustion temperature for ordinary combustion speed Tig, and the above-mentioned Eq. (23). The CPU 61 then proceeds to step 1170 of FIG. 11 via step 1495. When the CPU 61 proceeds to step 1170, it obtains the highest combustion temperature Tflame of the cylinder interior gas on the basis of the latest ignition-time compressed cylinder interior gas temperature Tpump obtained in the above-described step 1240, the latest combustion ascribable temperature increase ΔTburn obtained in the above-described step 1350, the latest combustion-speed ascribable temperature increase ΔTb_velo obtained in the above-described step 1440, and the above-described Eq. (1). After that, the CPU 61 proceeds to step 1195 to end the current execution of the present routine.

[0137] After that point in time, the CPU 61 repeatedly executes the processing of steps 1100, 1105, 1145, and 1195 until ATDC-180° for the fuel injection cylinder comes again. In the above-described manner, the highest combustion temperature Tflame of cylinder interior gas is newly obtained every time the fuel injection start timing comes.

<Calculation of NO_x Generation Quantity>

[0138] Moreover, the CPU 61 repeatedly executes, at predetermined intervals, a routine shown by the flowchart of FIG. 15 and adapted to calculate actual NO_x generation quantity NOxact. Therefore, when a predetermined timing has been reached, the CPU 61 starts the processing from step 1500, and then proceeds to step 1505 so as to determine whether the fuel injection start timing (i.e., the final fuel injection timing finjfin) has come. When the CPU 61 makes a "No" determination in step 1505, it proceeds directly to step 1595 so as to end the current execution of the present routine.

[0139] Now, it is assumed that the fuel injection start timing has come. In this case, the CPU 61 proceeds to step 1510 so as to obtain the above-mentioned table search value dataMap1 (= K1·log(RO2c)) on the basis of the latest bottom-dead-center intake-gas oxygen concentration R02c obtained in the above-described step 1115 and the above-described table Maplogl.

[0140] Similarly, the CPU 61 proceeds to step 1515 so as to obtain the above-mentioned table search value dataMap2 (= K2·log(qfinc)) on the basis of the fuel injection quantity qfinc in the present operation cycle, which has been stored in the above-described step 570, and the above-described table Maplog2. In the subsequent step 1520, the CPU 61 obtains the above-mentioned table search value dataMap3 (= K3·log(Pcrc)) on the basis of the fuel injection pressure Pcrc in the present operation cycle, which has been stored in the above-described step 575, and the above-described table Maplog3. In the subsequent step 1525, the CPU 61 obtains the above-mentioned table search value dataMap4 (= K4/Tflame)) on the basis of the latest highest combustion temperature Tflame, which has been determined in the above-described step 1170, and the above-described table Mapinvpro.

[0141] Subsequently, the CPU 61 proceeds to step 1530 so as to obtain "log(RNOx_burn)" in accordance with the above-described Eq. (27). In the subsequent step 1535, the CPU 61 determines the combustion-generated NO_x ratio RNOx_burn on the basis of the log(RNOx_burn) and the above-described table Mapinvlog. The CPU 61 then proceeds to step 1540 so as to obtain the actual NO_x generation quantity NOxact in accordance with the above-described Eq. (24) and on the basis of the above-mentioned fuel injection quantity qfinc in the present operation cycle and the above-mentioned combustion-generated NO_x ratio RNOx_burn. After that, the CPU 61 proceeds to 1595 so as to end the current execution of the present routine. After that point in time, the CPU 61 repeatedly executes the processing of steps 1500, 1505, and 1595 until the fuel injection start timing for the fuel injection cylinder comes again.

[0142] As described above, a new actual NO_x generation quantity NOxact is obtained each time the fuel injection start timing comes. The obtained new actual NO_x generation quantity NOxact is used in step 525 of FIG. 5 as described above. As a result, the final fuel injection timing finjfin and the opening of the EGR control valve 52 to be applied to the fuel injection cylinder in the next operation cycle are feedback-controlled on the basis of the new actual NO_x generation quantity NOxact.

[0143] As described above, under the combustion temperature estimation method for an internal combustion engine according to the embodiment of the present invention, upon each arrival of fuel injection start timing, the cylinder interior gas temperature (before combustion) at the time of ignition (ignition-time compressed cylinder interior gas temperature Tpump) is estimated, while the fact that the state of cylinder interior gas changes adiabatically is used as a general rule. The quantity Qfuel of heat generated as a result of combustion of fuel is divided by the product of the post-combustion mole amount ngas and constant-pressure specific heat Cp of the cylinder interior gas, which can be obtained from the composition of intake gas (the concentration proportions of gas components), to thereby estimate an increase in temperature of the cylinder interior gas stemming from combustion (combustion ascribable temperature increase ΔTburn). Further, an increase in temperature of cylinder interior gas stemming from an increase in combustion speed (combustion-speed ascribable temperature increase ΔTb_velo) is estimated on the basis of fuel injection pressure Pcrc and engine speed NE0, which are factors which influence the combustion speed. Then, the highest combustion temperature Tflame is estimated from a value obtained through addition of the combustion ascribable temperature increase ΔTburn and the combustion-speed ascribable temperature increase ΔTb_velo to the ignition-time compressed cylinder interior gas temperature Tpump. Accordingly, the highest combustion temperature Tflame can be accurately estimated by use of a simple configuration to match various actual phenomena.

[0144] The present invention is not limited to the above-described embodiment, and may be modified in various manners within the scope of the present invention. For example, the following modifications may be employed. In the above-described embodiment, fuel injection pressure and engine speed are employed as factors which influence combustion speed. However, at least one of the swirl ratio of gas taken into cylinders, the boost pressure produced by a supercharger, and the oxygen concentration of gas taken into cylinders may be employed as a factor which influences combustion speed.

[0145] In the above-described embodiment, heat generated as a result of supply of electricity to a glow plug is employed as a factor which causes an increase in ignition-time compressed cylinder interior gas temperature (cylinder interior gas temperature increase ΔTpump). However, in the case where pilot injection is performed before final fuel injection timing finjfin (main injection), heat generated as a result of combustion of fuel injected by means of the pilot injection may be employed as such a factor. In this case, the apparatus according to the present invention is preferably configured to calculate the heat generated as a result of combustion of fuel injected by means of the pilot injection (accordingly, an increase in temperature of cylinder interior gas) on the basis of, for example, the quantity of fuel injected by means of the pilot injection and the timing of the pilot injection (the time span (interval) between the pilot injection and the main injection).

[0146] In a combustion temperature estimation method for an internal combustion engine, upon each arrival of fuel injection start timing, the pre-combustion temperature of cylinder interior gas at the time of ignition (ignition-time compressed cylinder interior gas temperature Tpump) is estimated on the basis of the fact that the state of the cylinder interior gas changes adiabatically. The quantity of heat generated as a result of combustion of fuel is divided by the product of the post-combustion mole amount and constant-pressure specific heat of the cylinder interior gas, which can be obtained from the concentration proportions of gas components contained in intake gas, to thereby estimate an increase in temperature of the cylinder interior gas stemming from combustion (combustion ascribable temperature increase ΔTburn). Further, an increase in temperature of the cylinder interior gas stemming from an increase in combustion speed (combustion-speed ascribable temperature increase ΔTb_velo) is estimated on the basis of fuel injection pressure and engine speed, which are factors which influence the combustion speed. Then, the highest combustion temperature Tflame is estimated by the equation Tflame = Tpump + ΔTburn + ΔTb_velo.

Claims

1. A combustion temperature estimation method for estimating combustion temperature in a cylinder of an internal combustion engine, **characterized by** comprising:

    estimating an ignition-time compressed cylinder interior gas temperature, which is a pre-combustion cylinder interior gas temperature at the time of ignition, while utilizing the fact that at least cylinder interior gas present in the cylinder is compressed in the cylinder;
    estimating a combustion ascribable temperature increase, which is an increase in temperature of the cylinder interior gas as a result of combustion, on the basis of at least the composition of gas taken in the cylinder and the quantity of heat generated as a result of combustion of injected fuel;
    estimating a combustion-speed ascribable temperature increase, which is an increase in temperature of the cylinder interior gas stemming from an increase in combustion speed, on the basis of a factor which influences combustion speed in the cylinder; and
    estimating the combustion temperature in the cylinder from a value obtained through addition of the combustion ascribable temperature increase and the combustion-speed ascribable temperature increase to the ignition-time compressed cylinder interior gas temperature.

2. A combustion temperature estimation method according to claim 1, wherein the ignition-time compressed cylinder interior gas temperature is estimated under the assumption that the state of the cylinder interior gas adiabatically changes before combustion in a compression stroke.

3. A combustion temperature estimation method according to claim 1 or 2, wherein the ignition-time compressed cylinder interior gas temperature is estimated on the basis of the temperature of gas taken into the cylinder, the composition of the taken gas, and an increase in the ignition-time compressed cylinder interior gas temperature estimated on the basis of a factor which increases the pre-combustion cylinder interior gas temperature.

4. A combustion temperature estimation method according to claim 3, wherein the increase in the ignition-time compressed cylinder interior gas temperature is estimated on the basis of at least one of heat generated as a result of combustion of fuel injected by means of pilot injection in the case where such pilot injection is performed before

the injection of fuel, and heat generated as a result of supply of electricity to a glow plug in the case where electricity is supplied to the glow plug, wherein the heat generated as a result of combustion of fuel injected by means of the pilot injection and the heat generated as a result of supply of electricity to the glow plug each serving as the factor which increases the pre-combustion cylinder interior gas temperature.

5. A combustion temperature estimation method according to any one of claims 1 to 4, wherein when the time of ignition is prior to a point in time corresponding to a compression top dead center, the ignition-time. compressed cylinder interior gas temperature is estimated under the assumption that the time of ignition coincides with the point in time corresponding to the compression top dead center.

6. A combustion temperature estimation method according to any one of claims 1 to 5, wherein the combustion ascribable temperature increase is estimated on the basis of a post-combustion specific heat and a post-combustion quantity of the cylinder interior gas having participated in combustion, which are obtained from the composition of gas taken in the cylinder, and the quantity of heat generated as a result of combustion of the injected fuel.

7. A combustion temperature estimation method according to any one of claims 1 to 6, wherein the combustion-speed ascribable temperature increase is estimated on basis of at least one of fuel injection pressure, engine speed, swirl ratio of gas taken into the cylinder, boost pressure produced by a supercharger in the case where the engine is equipped with the supercharger, and oxygen concentration of gas taken into the cylinder, wherein the fuel injection pressure, the engine speed, the swirl ratio, the boost pressure, and the oxygen concentration each serving as the factor which influences combustion speed in the cylinder.

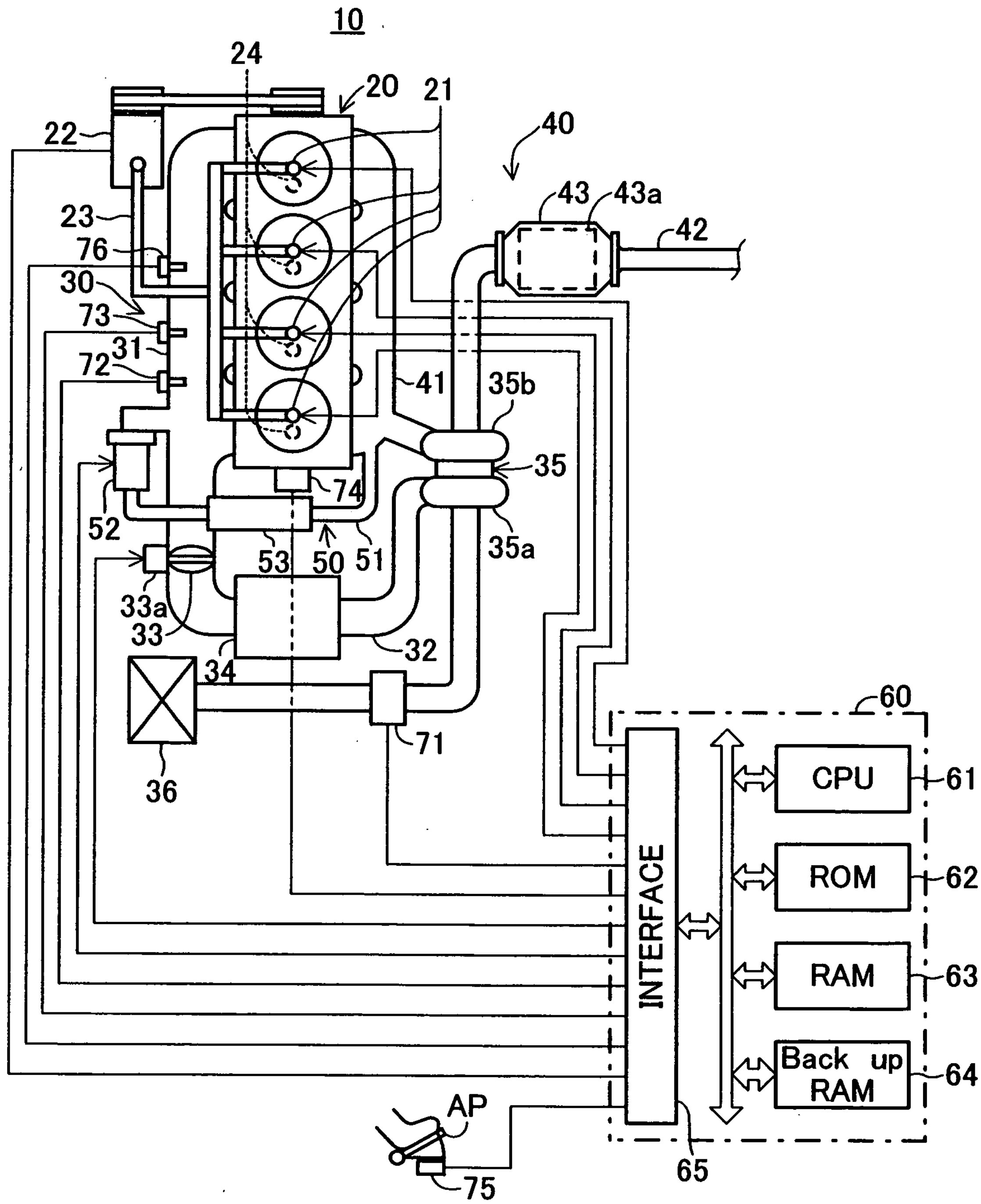
10
24
20
21
22
23
76
30
73
31
72
40
43
43a
42
41
35b
35
35a
74
52
53
50
51
33a
33
34
32
36
71
60
CPU
61
ROM
62
RAM
63
Back up RAM
64
INTERFACE
65
AP
75

FIG.1

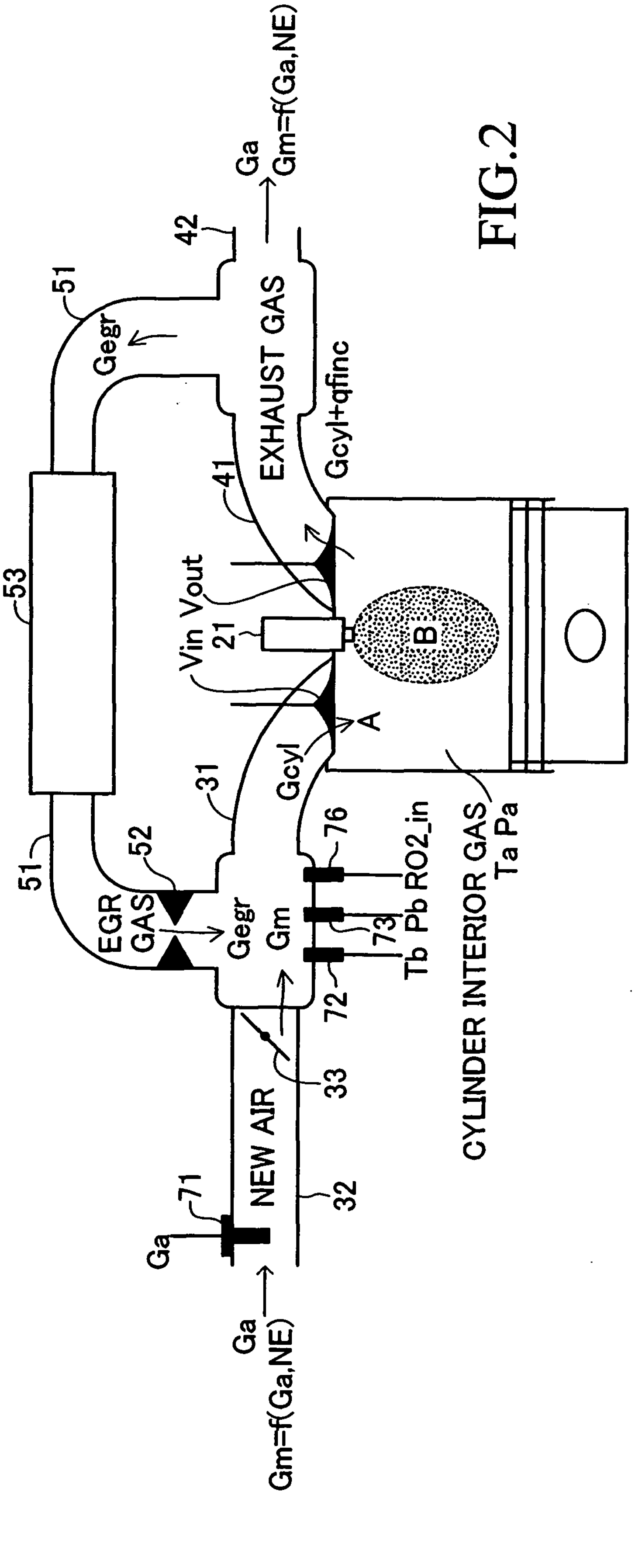
Gm=f(Ga,NE)
Ga
42
51
Gegr
EXHAUST GAS
Gcyl+qfinc
41
53
Vin Vout
21
B
A
Gcyl
31
52
EGR
GAS
Gegr
Gm
76
73
72
Tb Pb RO2_in
CYLINDER INTERIOR GAS
Ta Pa
33
NEW AIR
71
Ga
32
Ga
Gm=f(Ga,NE)

FIG.2

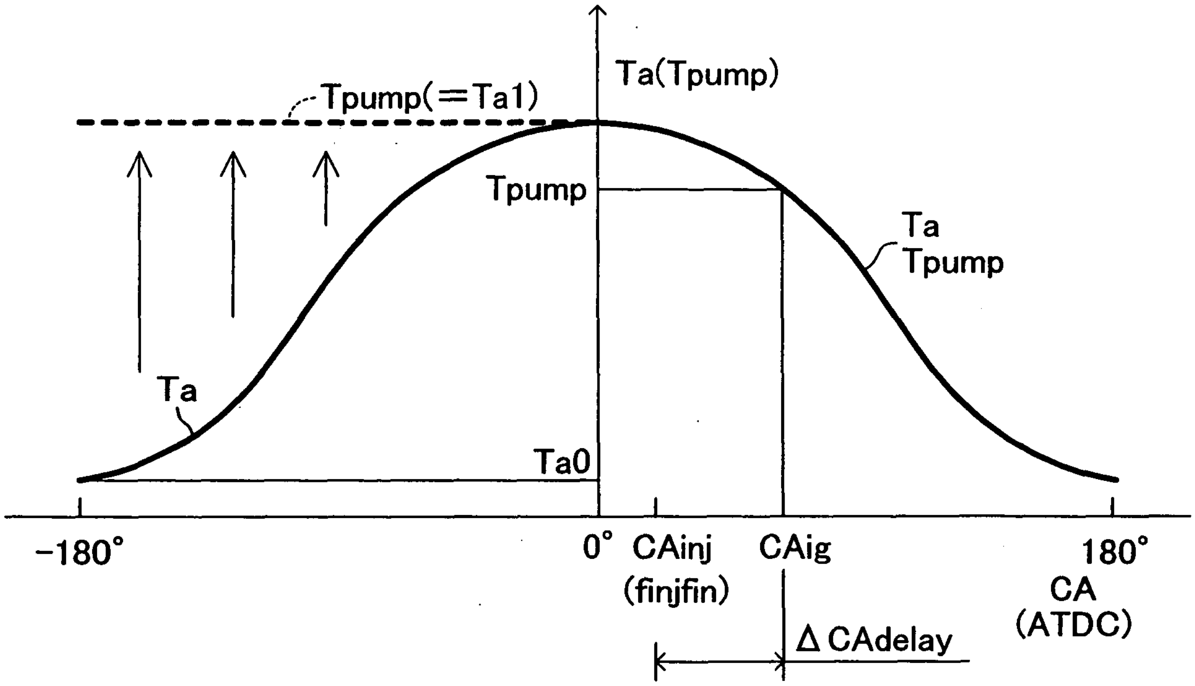
Ta(Tpump)
Tpump(=Ta1)
Tpump
Ta
Tpump
Ta
Ta0
-180°
0°
CAinj
(finjfin)
CAig
180°
CA
(ATDC)
Δ CAdelay

FIG.3

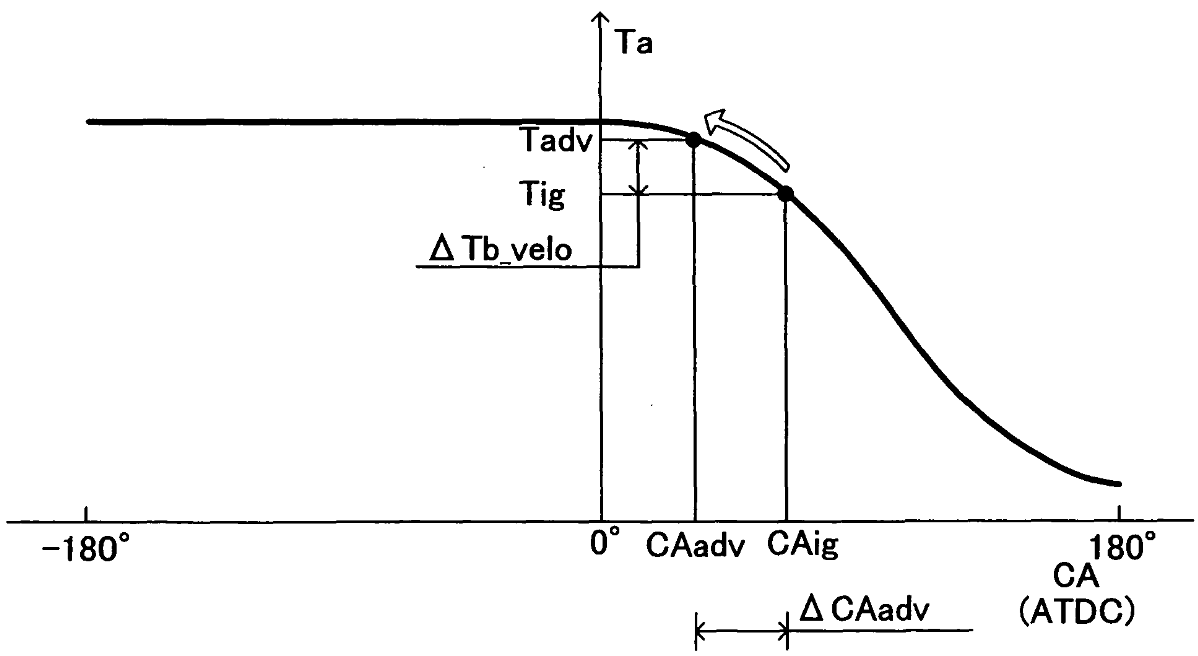
Ta
Tadv
Tig
Δ Tb_velo
-180°
0°
CAadv
CAig
180°
CA
(ATDC)
Δ CAadv

FIG.4

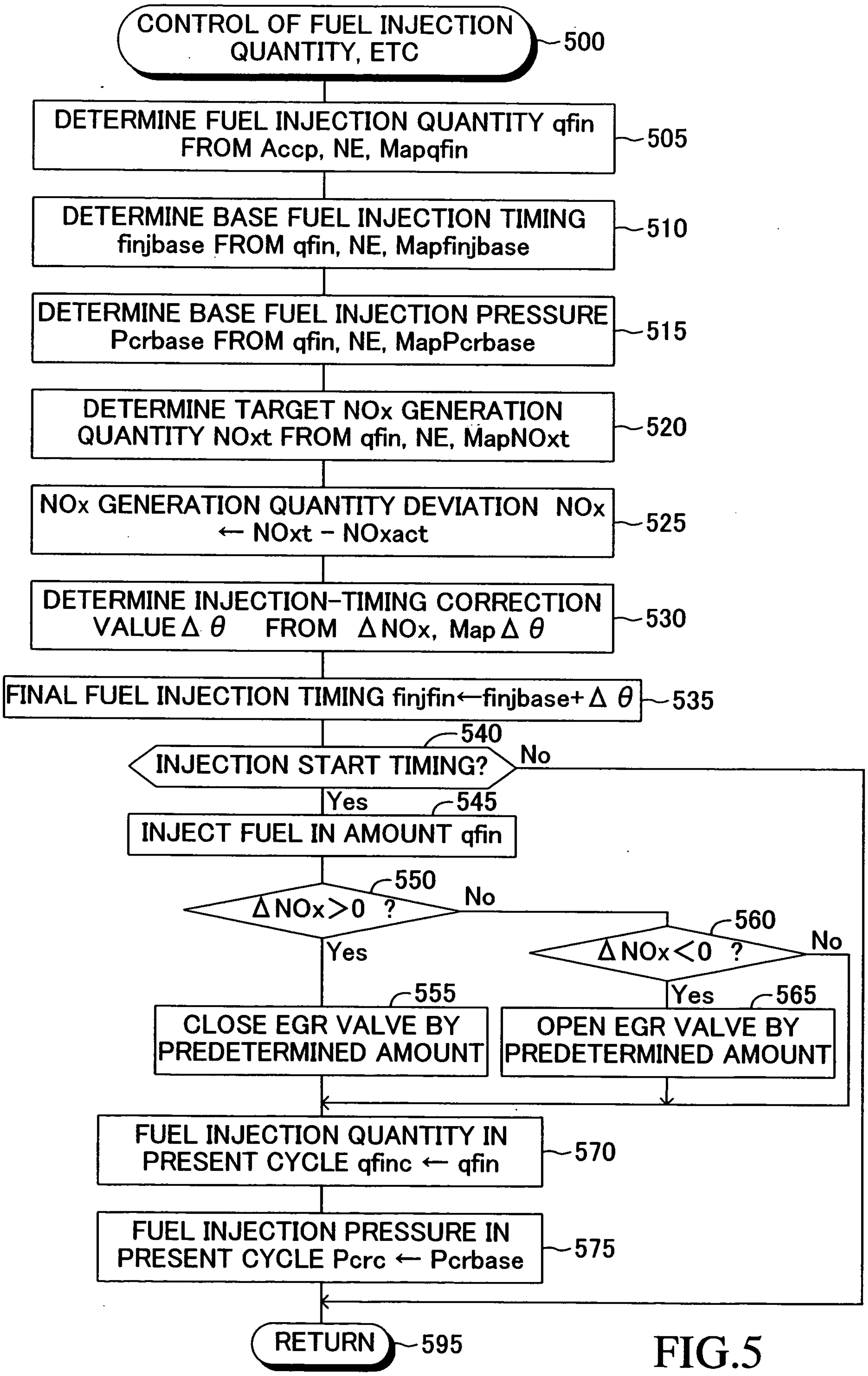
CONTROL OF FUEL INJECTION QUANTITY, ETC
500
DETERMINE FUEL INJECTION QUANTITY qfin FROM Accp, NE, Mapqfin
505
DETERMINE BASE FUEL INJECTION TIMING finjbase FROM qfin, NE, Mapfinjbase
510
DETERMINE BASE FUEL INJECTION PRESSURE Pcrbase FROM qfin, NE, MapPcrbase
515
DETERMINE TARGET NOx GENERATION QUANTITY NOxt FROM qfin, NE, MapNOxt
520
NOx GENERATION QUANTITY DEVIATION NOx ← NOxt − NOxact
525
DETERMINE INJECTION-TIMING CORRECTION VALUE Δθ FROM ΔNOx, MapΔθ
530
FINAL FUEL INJECTION TIMING finjfin←finjbase+Δθ
535
540
INJECTION START TIMING?
No
Yes
545
INJECT FUEL IN AMOUNT qfin
550
ΔNOx>0 ?
No
Yes
560
ΔNOx<0 ?
No
Yes
555
CLOSE EGR VALVE BY PREDETERMINED AMOUNT
565
OPEN EGR VALVE BY PREDETERMINED AMOUNT
FUEL INJECTION QUANTITY IN PRESENT CYCLE qfinc ← qfin
570
FUEL INJECTION PRESSURE IN PRESENT CYCLE Pcrc ← Pcrbase
575
RETURN
595

FIG.5

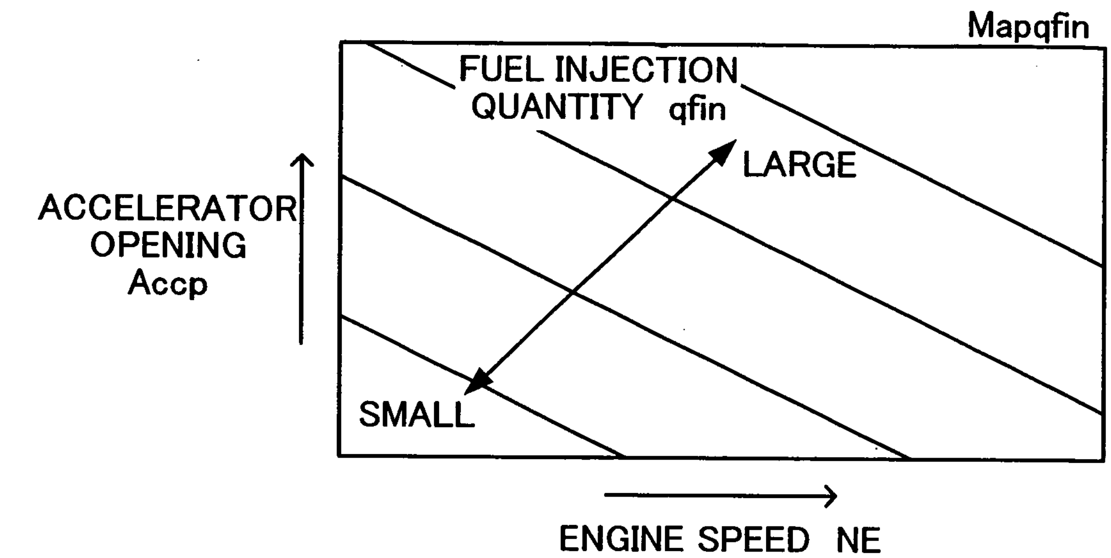
Mapqfin
FUEL INJECTION QUANTITY qfin
LARGE
SMALL
ACCELERATOR OPENING Accp
ENGINE SPEED NE

FIG.6

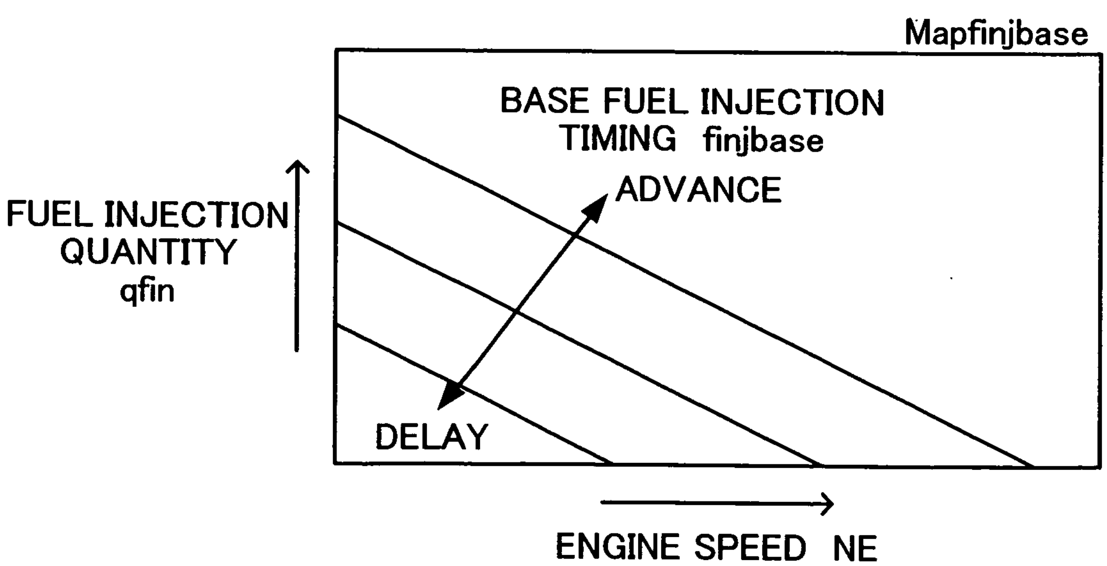
Mapfinjbase
BASE FUEL INJECTION TIMING finjbase
ADVANCE
DELAY
FUEL INJECTION QUANTITY qfin
ENGINE SPEED NE

FIG.7

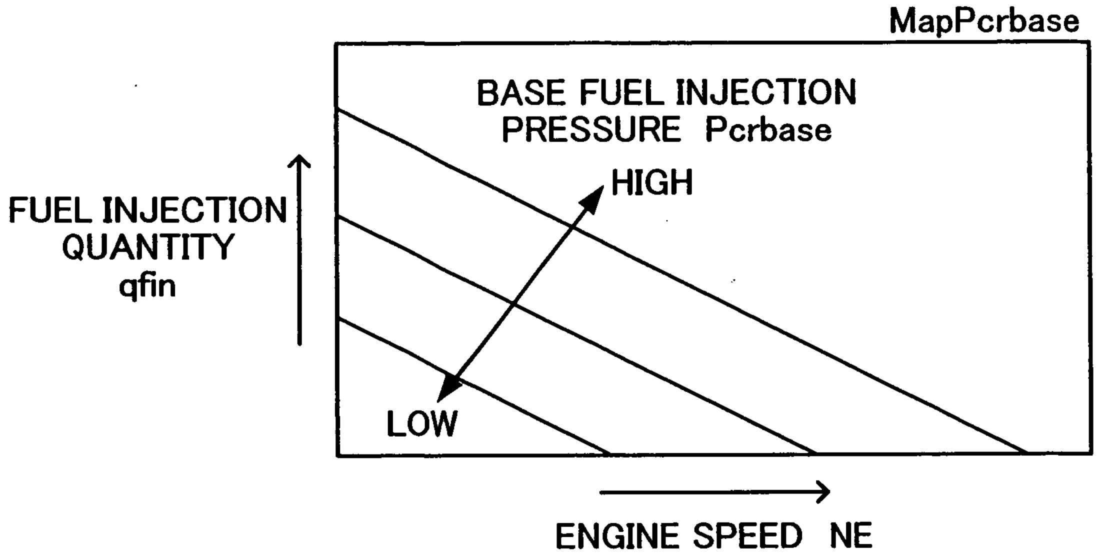
MapPcrbase
BASE FUEL INJECTION PRESSURE Pcrbase
HIGH
LOW
FUEL INJECTION QUANTITY qfin
ENGINE SPEED NE

FIG.8

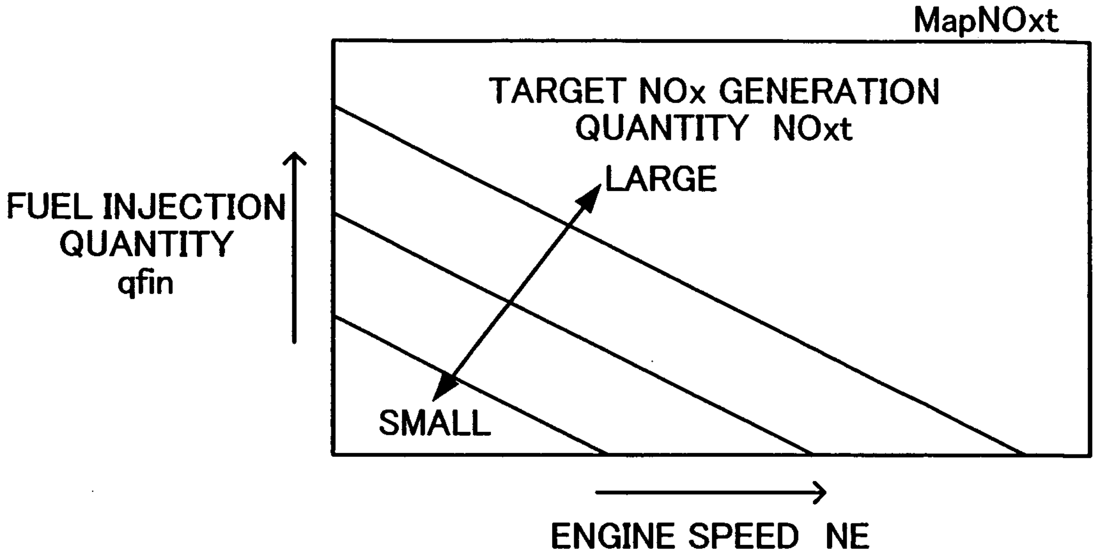
MapNOxt
TARGET NOx GENERATION QUANTITY NOxt
LARGE
SMALL
FUEL INJECTION QUANTITY qfin
ENGINE SPEED NE

FIG.9

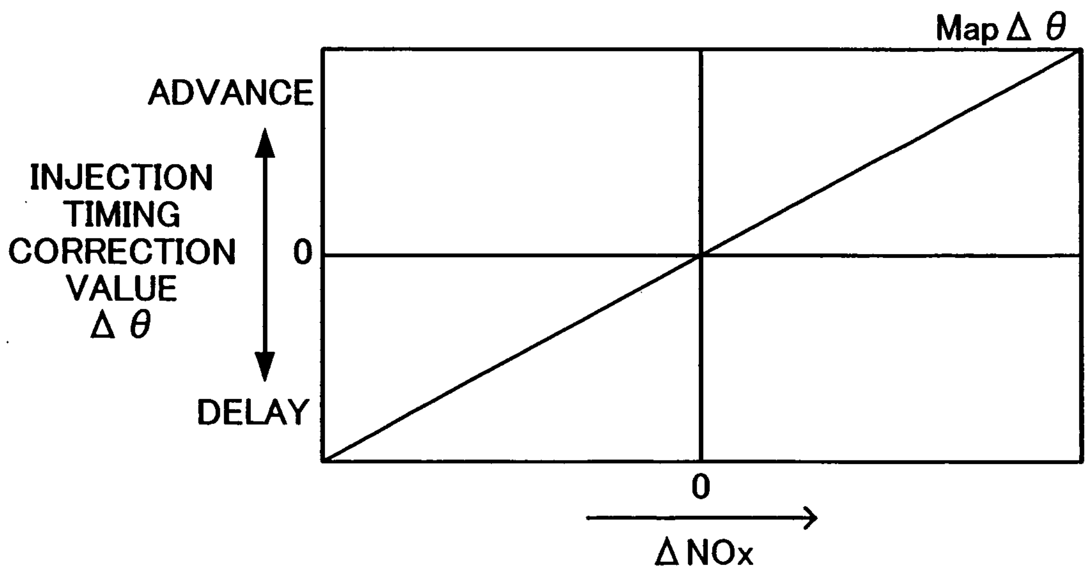
Map Δ θ
ADVANCE
INJECTION TIMING CORRECTION VALUE Δ θ
0
DELAY
0
Δ NOx

FIG.10

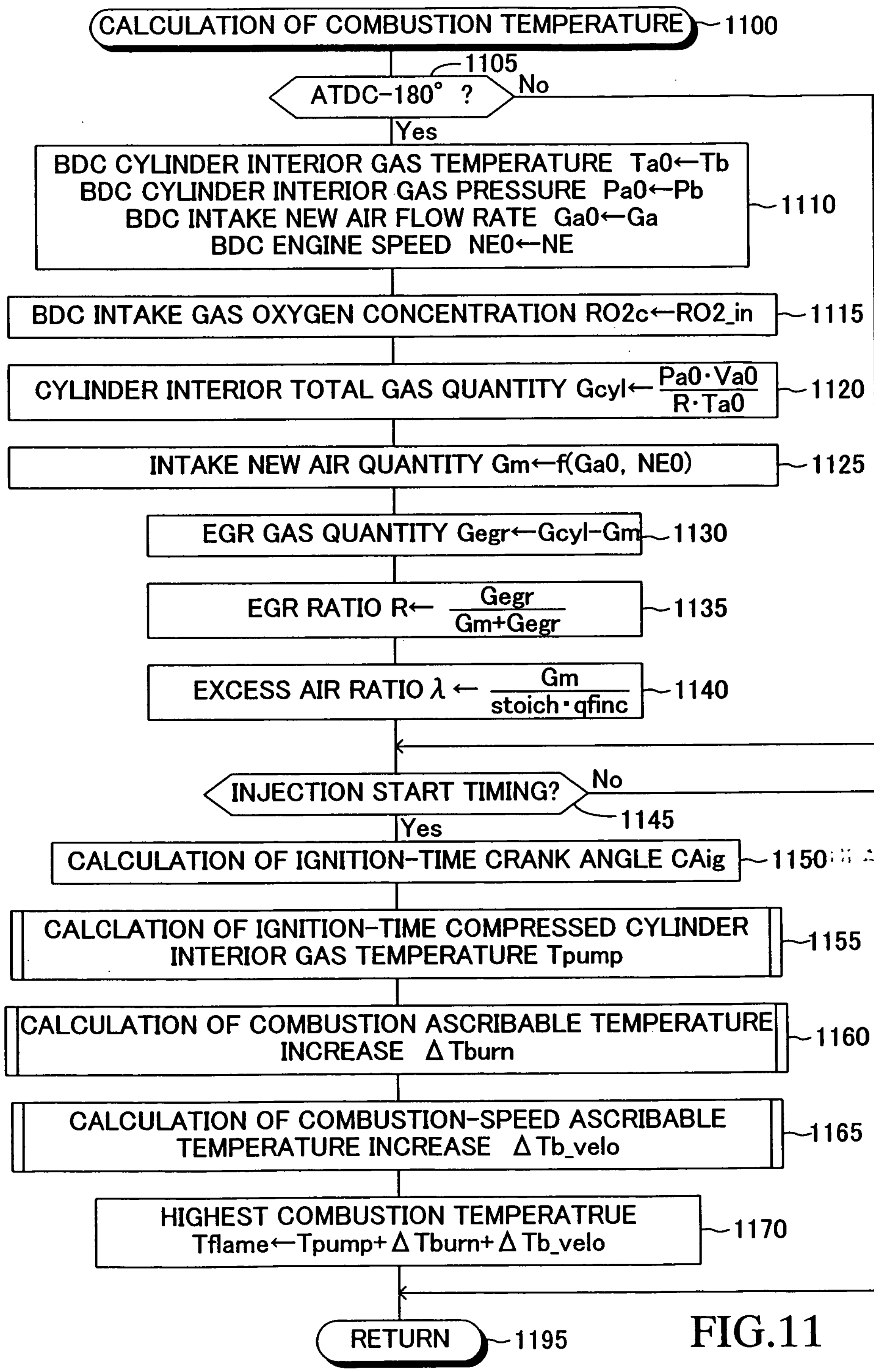
CALCULATION OF COMBUSTION TEMPERATURE
1100
1105
ATDC−180° ?
No
Yes
BDC CYLINDER INTERIOR GAS TEMPERATURE Ta0←Tb
BDC CYLINDER INTERIOR GAS PRESSURE Pa0←Pb
BDC INTAKE NEW AIR FLOW RATE Ga0←Ga
BDC ENGINE SPEED NE0←NE
1110
BDC INTAKE GAS OXYGEN CONCENTRATION RO2c←RO2_in
1115
CYLINDER INTERIOR TOTAL GAS QUANTITY Gcyl← Pa0·Va0 / R·Ta0
1120
INTAKE NEW AIR QUANTITY Gm←f(Ga0, NE0)
1125
EGR GAS QUANTITY Gegr←Gcyl−Gm
1130
EGR RATIO R← Gegr / Gm+Gegr
1135
EXCESS AIR RATIO λ ← Gm / stoich·qfinc
1140
INJECTION START TIMING?
No
1145
Yes
CALCULATION OF IGNITION−TIME CRANK ANGLE CAig
1150
CALCLATION OF IGNITION−TIME COMPRESSED CYLINDER INTERIOR GAS TEMPERATURE Tpump
1155
CALCULATION OF COMBUSTION ASCRIBABLE TEMPERATURE INCREASE ΔTburn
1160
CALCULATION OF COMBUSTION−SPEED ASCRIBABLE TEMPERATURE INCREASE ΔTb_velo
1165
HIGHEST COMBUSTION TEMPERATRUE
Tflame←Tpump+ΔTburn+ΔTb_velo
1170
RETURN
1195

FIG.11

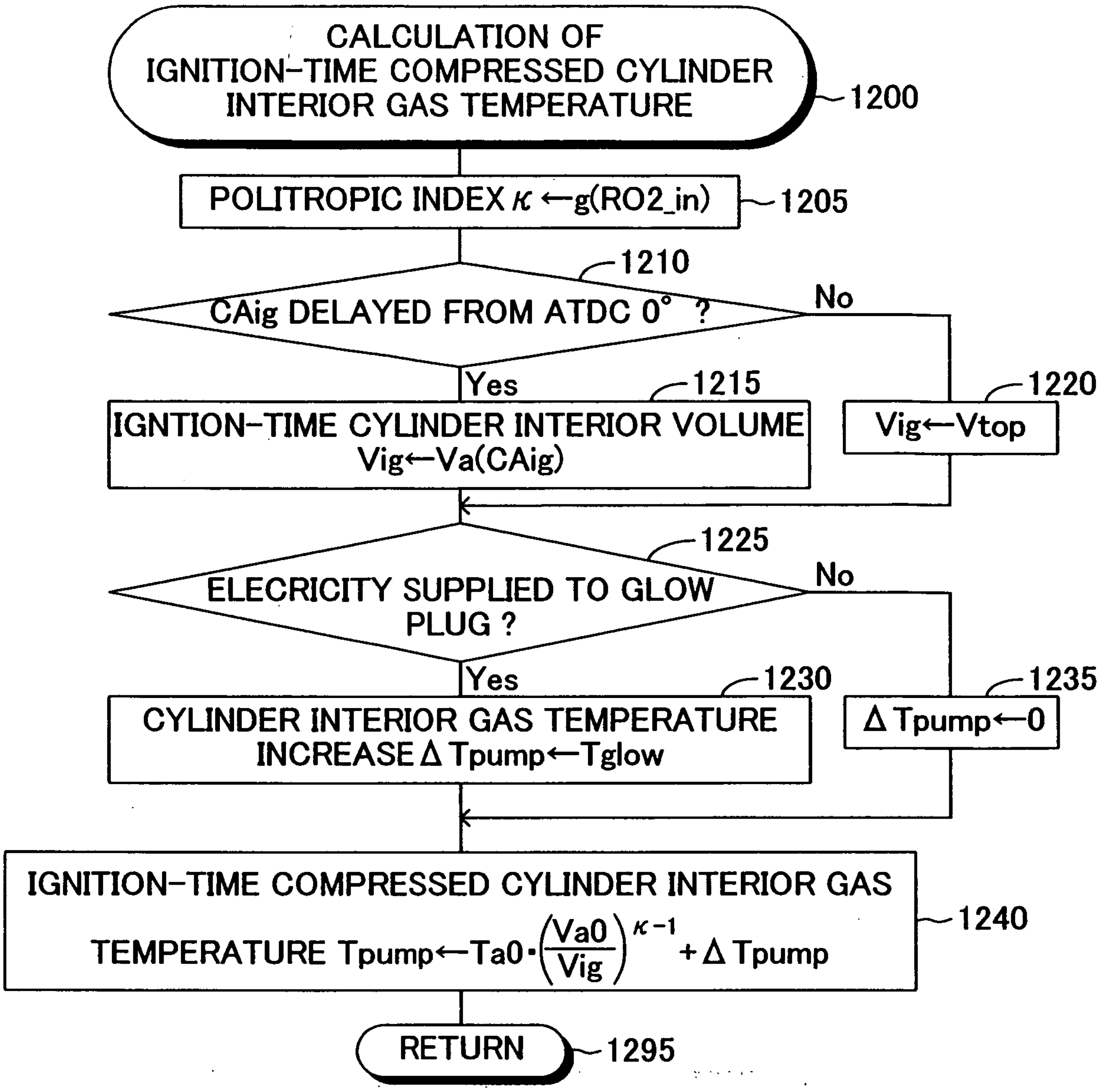
CALCULATION OF
IGNITION-TIME COMPRESSED CYLINDER
INTERIOR GAS TEMPERATURE
1200
POLITROPIC INDEX $\kappa \leftarrow$g(RO2_in)
1205
1210
CAig DELAYED FROM ATDC 0° ?
No
Yes
1215
IGNTION-TIME CYLINDER INTERIOR VOLUME
Vig←Va(CAig)
1220
Vig←Vtop
1225
ELECRICITY SUPPLIED TO GLOW PLUG ?
No
Yes
1230
CYLINDER INTERIOR GAS TEMPERATURE
INCREASE ΔTpump←Tglow
1235
ΔTpump←0
IGNITION-TIME COMPRESSED CYLINDER INTERIOR GAS
TEMPERATURE $Tpump \leftarrow Ta0 \cdot \left(\frac{Va0}{Vig}\right)^{\kappa-1} + \Delta Tpump$
1240
RETURN
1295

FIG.12

CALCULATION OF COMBUSTION ASCRIBABLE TEMPERATURE INCREASE 1300

INTAKE GAS O_2 CONCENTRATION $[O_2]in \leftarrow [O_2]air \left(1-\frac{R}{\lambda}\right)$ 1305

INTAKE GAS CO_2 CONCENTRATION $[CO_2]in$

$\leftarrow [CO_2]air \cdot (1-R) + \frac{KCO_2 \cdot qfinc + [CO_2]air \cdot Gm}{Gm + qfinc} \cdot R$ 1310

INTAKE GAS H2O CONCENTRATION [H2O]in

$\leftarrow [H2O]air \cdot (1-R) + \frac{KH_2O \cdot qfinc + [H_2O]air \cdot Gm}{Gm + qfinc} \cdot R$ 1315

INTAKE GAS N_2 CONCENTRATION $[N_2]in \leftarrow [N_2]air \cdot (1-R) + \frac{[N_2]air \cdot Gm}{Gm + qfinc} \cdot R$ 1320

$\alpha \leftarrow \frac{[CO_2]in}{[O_2]in} \cdot \frac{MO_2}{MCO_2} \cdot \left(m + \frac{n}{4}\right)$ 1325

$\beta \leftarrow \frac{[H_2O]in}{[O_2]in} \cdot \frac{MO_2}{MH_2O} \cdot \left(m + \frac{n}{4}\right)$ 1330

$\gamma \leftarrow \frac{[N_2]in}{[O_2]in} \cdot \frac{MO_2}{MN_2} \cdot \left(m + \frac{n}{4}\right)$ 1335

POST-COMBUSTION CYLINDER INTERIOR GAS MOLAR AMOUNT 1340

$ngas \leftarrow m + \frac{n}{2} + \alpha + \beta + \gamma$

POST-COMBUSTION CYLINDER INTERIOR GAS CONSTANT-PRESSRUE 1345

SPECIFIC HEAT $Cp \leftarrow \frac{1}{ngas} \cdot \left\{ (m+\alpha) \cdot Cp\_CO_2 + \left(\frac{n}{2} + \beta\right) \cdot Cp\_H_2O + \gamma \cdot Cp\_N_2 \right\}$

COMBUSTION ASCRIBABLE TEMPERATURE INCREASE 1350

$\Delta Tburn \leftarrow \frac{Qfuel}{Cp \cdot ngas}$

RETURN 1395

FIG.13

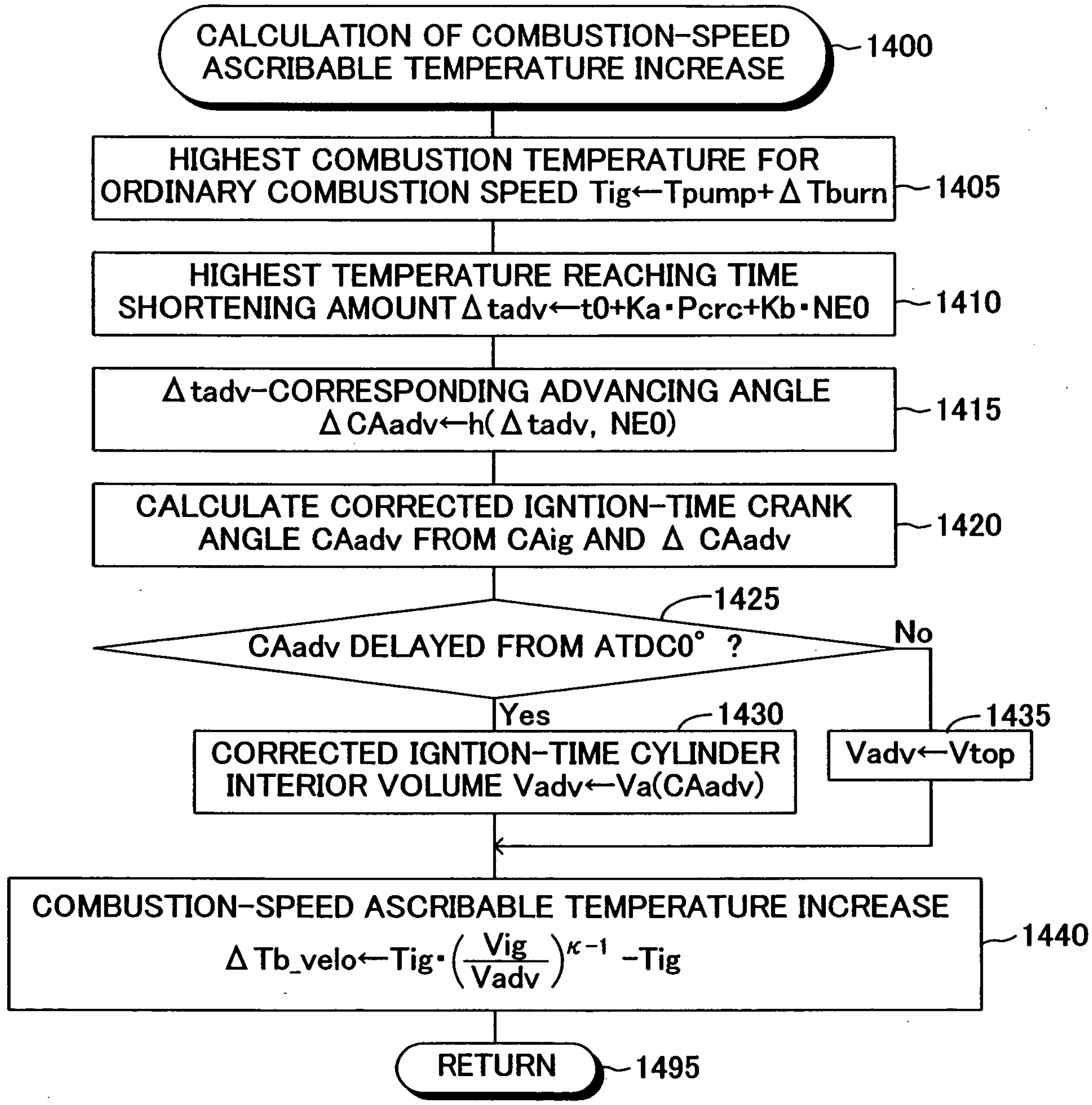
CALCULATION OF COMBUSTION-SPEED ASCRIBABLE TEMPERATURE INCREASE
1400
HIGHEST COMBUSTION TEMPERATURE FOR ORDINARY COMBUSTION SPEED Tig←Tpump+ΔTburn
1405
HIGHEST TEMPERATURE REACHING TIME SHORTENING AMOUNT Δtadv←t0+Ka•Pcrc+Kb•NE0
1410
Δtadv-CORRESPONDING ADVANCING ANGLE ΔCAadv←h(Δtadv, NE0)
1415
CALCULATE CORRECTED IGNTION-TIME CRANK ANGLE CAadv FROM CAig AND Δ CAadv
1420
1425
CAadv DELAYED FROM ATDC0° ?
No
Yes
1430
CORRECTED IGNTION-TIME CYLINDER INTERIOR VOLUME Vadv←Va(CAadv)
1435
Vadv←Vtop
COMBUSTION-SPEED ASCRIBABLE TEMPERATURE INCREASE
$\Delta Tb\_velo \leftarrow Tig \cdot \left(\frac{Vig}{Vadv}\right)^{\kappa-1} - Tig$
1440
RETURN
1495

FIG.14

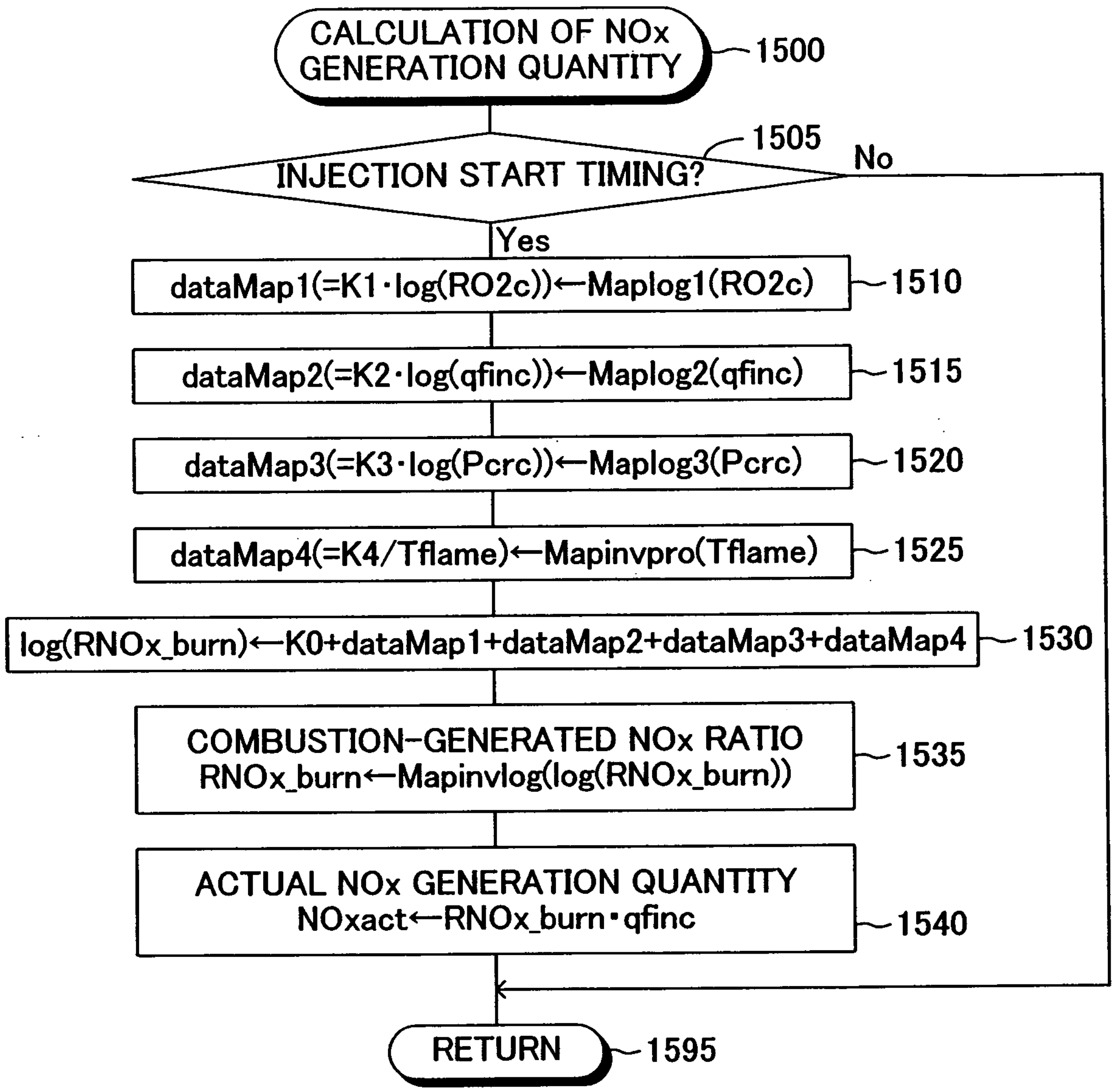
CALCULATION OF NOx GENERATION QUANTITY
1500
1505
INJECTION START TIMING?
No
Yes
dataMap1(=K1·log(RO2c))←Maplog1(RO2c)
1510
dataMap2(=K2·log(qfinc))←Maplog2(qfinc)
1515
dataMap3(=K3·log(Pcrc))←Maplog3(Pcrc)
1520
dataMap4(=K4/Tflame)←Mapinvpro(Tflame)
1525
log(RNOx_burn)←K0+dataMap1+dataMap2+dataMap3+dataMap4
1530
COMBUSTION-GENERATED NOx RATIO
RNOx_burn←Mapinvlog(log(RNOx_burn))
1535
ACTUAL NOx GENERATION QUANTITY
NOxact←RNOx_burn·qfinc
1540
RETURN
1595

FIG.15

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 04 02 9561

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.7)
D,A	PATENT ABSTRACTS OF JAPAN vol. 2002, no. 06, 4 June 2002 (2002-06-04) & JP 2002 054491 A (NISSAN MOTOR CO LTD), 20 February 2002 (2002-02-20) * abstract * -----	1-7	F02D41/04
A	PATENT ABSTRACTS OF JAPAN vol. 2000, no. 11, 3 January 2001 (2001-01-03) & JP 2000 213391 A (MITSUBISHI MOTORS CORP), 2 August 2000 (2000-08-02) * abstract * -----	1-7	
A	PATENT ABSTRACTS OF JAPAN vol. 1997, no. 11, 28 November 1997 (1997-11-28) & JP 09 189256 A (UNISIA JECS CORP), 22 July 1997 (1997-07-22) * abstract * -----	1-7	

TECHNICAL FIELDS SEARCHED (Int.Cl.7)

F02D

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
Munich	1 April 2005	Jackson, S

2

CATEGORY OF CITED DOCUMENTS

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01-04-2005

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
JP 2002054491 A	20-02-2002	NONE	
JP 2000213391 A	02-08-2000	NONE	
JP 09189256 A	22-07-1997	NONE	

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