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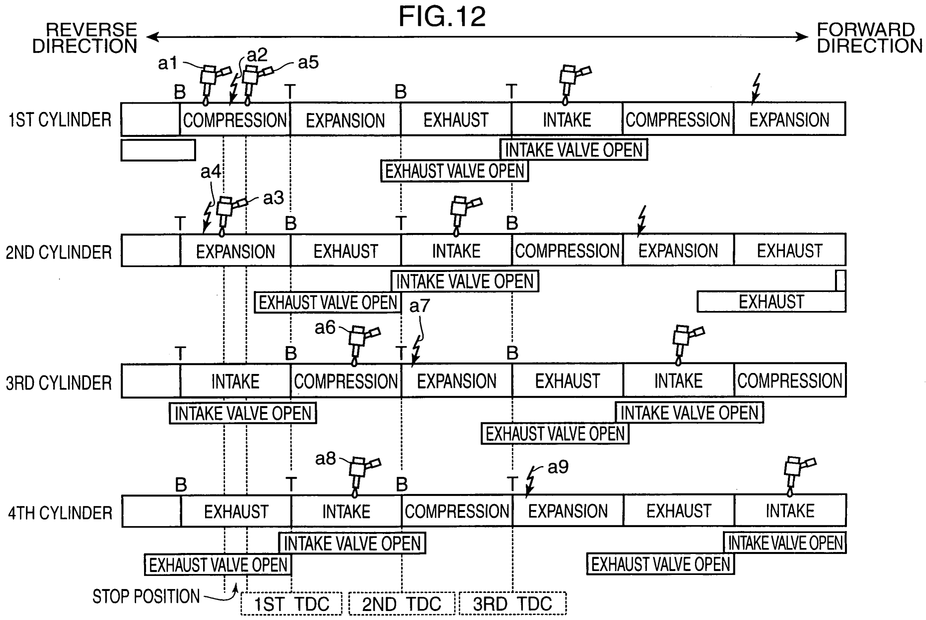
(54) **Engine starting system**

(57) A rich mixture having a low air-fuel ratio is combusted in a cylinder which is on a compression stroke at engine stop to produce a torque acting on an engine in a reverse running direction thereof and a mixture produced in a cylinder which is on an expansion stroke at engine stop is combusted after the mixture has been sufficiently compressed as a result of reverse running motion of the engine. Subsequently, an additional quantity of fuel is injected into the cylinder which was initially on the compression stroke when gas in this cylinder has been compressed due to the reverse running motion of the engine, so that a compressive reaction force exerted by the same cylinder decreases due to a cooling effect produced by absorption of latent heat by evaporation of the injected fuel.

FIG.12



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Description

BACKGROUND OF THE INVENTION

1. Field of the Invention

[0001] The present invention relates generally to an engine starting system for restarting an engine which has once stopped automatically under engine idle conditions in response to an engine restart request, and in particular to a fuel supply control technique which makes it possible to reverse an engine a little at first for increasing starting torque at engine restart.

2. Description of the Related Art

[0002] There exist conventionally known engine control systems devised for reducing fuel consumption and carbon dioxide (CO_2) emissions of an engine, for instance. One example of such engine control systems is an idle stop system which causes an engine to automatically stop at engine idle. In this kind of system, it is desirable to instantly start the engine in response to an engine restart request, such as a driver's action for restarting. A generally known method of restarting a once stopped engine by cranking the engine with a starter motor tends to require a relatively long period of time for engine restart. In addition, this conventional starting system has a drawback in that cranking operation could cause uncomfortable noise or a rapid increase in engine speed.

[0003] If the engine is caused to stop and restart each time the engine is brought to idle, the starter motor must restart the engine a significantly larger number of times compared to the conventional starting system which restarts the engine only when an ignition switch is operated. Therefore, the starter motor, if employed, must have considerably high durability, which results in an unwanted cost increase.

[0004] Under these circumstances, techniques for restarting the engine without using the starter motor have been developed in recent years. According to direct injection engines disclosed in Japanese Laid-open Utility Model Publication No. 1985-128975 and Japanese Laid-open Patent Publication No. 2003-517134, for example, fuel is injected into a cylinder which is currently on an expansion stroke under engine stop conditions and a mixture produced in the cylinder is ignited and burnt such that the engine would restart by its own motive power.

[0005] In a case where the mixture is ignited and burnt in the cylinder in which a piston is stopped halfway on the expansion stroke, the cylinder does not hold a sufficient quantity of air and the distance traveled by the piston in the expansion stroke cylinder is too short, so that a sufficient starting torque will not be obtained by combustion in the expansion stroke cylinder. Typically, the piston in the expansion stroke cylinder is situated approximately at the middle of the stroke at idle stop and, thus, the quantity of air usable for combustion in the cylinder is approximately one half of the cylinder capacity at atmospheric pressure. Consequently, an increase in in-cylinder pressure caused by combustion in the expansion stroke cylinder would be insufficient an exhaust valve of the expansion stroke cylinder would shortly open when the piston descends as a result of combustion.

[0006] To solve this problem Japanese Laid-open Patent Publication No. 2003-515052 proposes an engine starting method in which combustion is first produced in a cylinder which is on a compression stroke at idle stop to reversely turn a crankshaft and thereby compress gas in the aforementioned expansion stroke cylinder and the mixture produced therein is burnt to restart the engine. In this engine starting method (hereinafter referred to as the reverse action starting method), the piston in the cylinder which was on the expansion stroke at idle stop is caused to move toward the top dead center due to initial reversing action of the engine so that the distance traveled by the piston in the expansion stroke cylinder increases and combustion in the cylinder is produced under conditions of high in-cylinder pressure and high density of air-fuel mixture. This makes it possible to obtain a high combustion pressure and a significantly increased starting torque.

[0007] In the aforementioned approach of Japanese Laid-open Patent Publication No. 2003-515052 in which the engine is restarted after once reversing the engine, combustion is first produced in the compression stroke cylinder for reversing the engine as stated above. As a result, the cylinder which was initially on the compression stroke is fully filled with burned gas and, therefore, it is impossible to recombust in the same cylinder unless the burned gas produced therein is discharged.

[0008] It will be recognized from above that even though the starting torque can be increased by compressing a mixture in the compression stroke cylinder and producing combustion therein, it is impossible to combust the mixture in the same cylinder when combustion should normally be produced in a succeeding expansion stroke. Moreover, since the temperature and pressure in the cylinder which was initially on the compression stroke are increased by the burned gas produced by the first combustion, a fairly high compressive reaction force is produced in the same cylinder due to a volumetric gaseous expansion, and this could lead to a failure to restart the engine.

[0009] One approach to the solution of this problem might be to produce the first combustion in the cylinder which is initially on the compression stroke by burning a lean mixture, or at a high air-fuel ratio, therein and produce recombustion in the same cylinder by using excess air after the turning direction of the engine has changed from a reverse direction to a forward direction. It is, however, difficult to obtain a sufficient torque for producing the initial reversing action of the engine by combustion under lean burn conditions and, therefore, gas in the cylinder which is initially on the expansion stroke would not be sufficiently compressed by the reversing action and subsequent combustion in this expansion stroke cylinder might not provide a sufficiently increased starting torque.

[0010] After combustion in the cylinder which was on the compression stroke at engine stop, a mixture produced in a cylinder which was on an intake stroke at engine stop is combusted. Since the temperature of air in the intake stroke cylinder is well increased due to heat dissipation from a cylinder wall and the cylinder is fully charged with relatively warm air introduced through an intake passage as a result of forward running motion of the engine, the temperature and pressure in the cylinder which was initially on the intake stroke considerably increase and autoignition (spontaneous combustion) would easily occur when the same cylinder goes into the compression stroke.

[0011] The cylinder which was on the compression stroke at engine stop does not provide any starting torque as mentioned above. Therefore, if autoignition occurs in the cylinder which was initially on the intake stroke in which the mixture should normally be ignited next, a torque produced in the engine-reversing direction by the autoignition would lead to a failure in engine restart.

SUMMARY OF THE INVENTION

[0012] The present invention is intended to provide a solution to the aforementioned problem of the prior art. Accordingly, it is a general object of the invention to provide an engine starting system for restarting an engine from idle stop. It is a more specific object of the invention to provide an engine starting system capable of restarting an engine in a reliable fashion, in which initial combustion is produced in a cylinder which is on a compression stroke at engine stop to reverse the engine at first and subsequent combustion is produced in a cylinder which is on an expansion stroke at engine stop to start the engine in a forward running direction by using a novel technique for controlling fuel injection into the compression stroke cylinder to obtain as high a starting torque as possible from combustion in the expansion stroke cylinder and to decrease a reaction force exerted by the compression stroke cylinder when the engine begins to run in the forward direction, and in which a novel technique is used in controlling fuel injection into a cylinder which is on an intake stroke at engine stop and in controlling ignition in this cylinder to prevent autoignition during a succeeding compression stroke of the same cylinder and to decrease a compressive reaction force exerted thereby.

[0013] To achieve the aforementioned object of the invention, initial combustion in a cylinder which is on a compression stroke at engine stop is made with a rich mixture having a low air-fuel ratio to produce a torque acting in an engine-reversing direction, whereby gas in a cylinder which is on an expansion stroke at engine stop is sufficiently compressed and a starting torque produced by combustion in the expansion stroke cylinder is significantly increased. Then, an additional quantity of fuel is injected into the cylinder which was initially on the compression stroke when gas in this cylinder has been compressed as a result of forward running motion of the engine, so that a compressive reaction force exerted by the same cylinder decreases due to a cooling effect produced by absorption of latent heat by evaporation of the injected fuel.

[0014] Specifically, an engine starting system for starting a multicylinder engine includes a fuel injection controlling section for controlling actuation of fuel injectors of which nozzle ends are located on the interior of individual cylinders, an ignition controlling section for controlling ignition timing for the individual cylinders, and an air quantity detecting section for detecting the quantity of air in the cylinder which is on a compression stroke at engine stop. In this engine starting system, the fuel injection controlling section causes the fuel injector of the cylinder which is on the compression stroke at engine stop to inject fuel and a mixture produced in the compression stroke cylinder is ignited and burnt so that the engine once turns in a reverse direction from idle stop, and the fuel injection controlling section causes the fuel injector of the cylinder which is on an expansion stroke at engine stop to inject the fuel when gas in the expansion stroke cylinder has been compressed as a result of reverse running motion of the engine and a mixture produced in the expansion stroke cylinder is ignited and burnt so that a torque acting in a forward direction is generated to restart the engine. In addition, the fuel injection controlling section controls the quantity of fuel injected into the compression stroke cylinder such that an average air-fuel ratio produced therein becomes lower than the stoichiometric air-fuel ratio based on a value of the quantity of air detected by the air quantity detecting section, and the fuel injection controlling section causes the fuel injector of the cylinder which was on the compression stroke at engine stop to inject an additional quantity of fuel when gas in the same cylinder has been compressed as a result of forward running motion of the engine.

[0015] In the engine starting system thus configured, the fuel injection controlling section controls the fuel injector of the cylinder which is on the compression stroke at engine stop to inject the fuel into the same cylinder and the mixture produced therein is combusted so that the engine turns in the reverse direction. Since the average air-fuel ratio in the compression stroke cylinder is low at this point, initial combustion in this cylinder produces a sufficiently large a torque acting in the engine-reversing direction and, therefore, gas in the expansion stroke cylinder can be sufficiently compressed.

[0016] On the other hand, the fuel injection controlling section causes the fuel injector of the cylinder which is on the expansion stroke at engine stop to inject the fuel and the mixture produced in this cylinder is combusted under conditions where the mixture has been compressed and in-cylinder temperature and pressure have increased as a result of the reverse running motion of the engine. Therefore, the torque generated by combustion in the expansion stroke cylinder and the distance traveled by a piston in the same cylinder increases, so that the combustion in the expansion stroke cylinder generates a significantly increased starting torque acting in the forward running direction of the engine.

[0017] In addition, as the fuel injection controlling section controls the fuel injector of the cylinder which was on the compression stroke at engine stop to inject an additional quantity of fuel when gas in the same cylinder has been compressed as a result of the forward running motion of the engine, the interior of this cylinder is cooled down by absorption of latent heat by evaporation of the additionally injected fuel. As a result, an increase in in-cylinder temperature and pressure is greatly lessened even if the cylinder which was on the compression stroke at engine stop is filled with burned gas produced by the initial combustion therein for reversing the engine. This enables a piston in the same cylinder to go beyond the first compression stroke top dead center (TDC) after engine restart, causing the engine to continue running in the forward direction. Thus, the engine starting system of the invention can restart the engine in a reliable fashion.

[0018] These and other objects, features and advantages of the invention will become more apparent upon reading the following detailed description along with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0019]

FIG. 1 is a partially sectional diagram generally showing the structure of an engine control system according to an embodiment of the invention;

FIG. 2 is a schematic diagram showing the configuration of intake and exhaust systems of the engine of FIG. 1;

FIGS. 3A-3D are diagrams schematically showing a reverse action starting procedure for restarting the engine according to the embodiment;

FIG. 4 is a flowchart showing a first half of an engine stop control routine performed for automatically stopping the engine at idle;

FIG. 5 is a flowchart showing a second half of the engine stop control routine performed for automatically stopping the engine at idle;

FIGS. 6A-6E are diagrams schematically showing how engine speed, crank angle, throttle opening and intake air pipe negative pressure vary in successive strokes of individual cylinders during an engine stopping period;

FIG. 7 is a distribution chart showing how TDC engine speed during the engine stopping period is correlated with piston stop positions at engine stop;

FIG. 8 is a flowchart of a stop position detecting subroutine performed for detecting the piston stop position at engine stop;

FIGS. 9A and 9B are diagrams showing how crank angle signals output from a pair of crank angle sensors are related in phase, FIG. 9A showing a phase relation observed when the engine runs in a forward direction, and FIG. 9B showing a phase relation observed when the engine runs in a reverse direction;

FIG. 10 is a flowchart showing a first half of an engine restart control routine performed for automatically restarting the engine from idle stop;

FIG. 11 is a flowchart showing a second half of the engine restart control routine performed for automatically restarting the engine from idle stop;

FIG. 12 is a diagram showing fuel injection and ignition timings at engine restart for the individual cylinders in relation to the timing of the successive strokes and open/closed states of intake and exhaust valves;

FIGS. 13A-13F are time charts showing how the engine speed, the pressure in each cylinder and the amount of torque generated by the engine vary during engine restart;

FIGS. 14A and 14B are diagrams showing how the pressure in the cylinder which was on the intake stroke at engine stop and the amount of torque generated by the same cylinder vary when autoignition occurs, and does not occur, during engine restart in the cylinder which was initially on the intake stroke;

FIG. 15 is a graph showing a relationship between fuel injection timing and an effect of reducing in-cylinder pressure at engine restart; and

FIG. 16 is a graph showing a relationship between the air-fuel ratio in a cylinder and an effect of reducing the in-cylinder pressure at engine restart.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT OF THE INVENTION

[0020] A preferred embodiment of the invention is now described in detail with reference to the accompanying drawings. It is to be noted that the following discussion of the embodiment simply illustrates a typical example of implementation of the invention and is not intended to limit specific applications or use of the invention.

GENERAL STRUCTURE OF ENGINE CONTROL SYSTEM

[0021] FIGS. 1 and 2 are diagrams schematically showing the structure of an engine control system provided with an engine starting system according to the embodiment of the invention. An engine system E includes an engine 1 having a cylinder head 10 and a cylinder block 11 associated with an electronic control unit (ECU) 2. The engine 1 has four cylinders 12A-12D (which may be referred to simply as the cylinders 12 collectively) as shown in FIG. 2. Pistons 13 connected to a crankshaft 3 are fitted in the individual cylinders 12A-12D whereby a combustion chamber 14 is formed above the piston 13 in each of the cylinders 12A-12D as shown in FIG. 1.

[0022] Generally, a four-cycle multicylinder engine is constructed in such a manner that individual cylinders undergo successive combustion cycles of intake, compression, expansion and exhaust strokes with a specific phase delay from one cylinder to another. The cylinders 12A-12D of the four-cylinder engine of this embodiment, which are now referred to as the first cylinder 12A, the second cylinder 12B, the third cylinder 12C and the fourth cylinder 12D in this order as viewed from one end of a cylinder bank, undergo the aforementioned combustion cycles with a successive phase delay of 180° in terms of crank angle in the order of the first cylinder 12A, the third cylinder 12C, the fourth cylinder 12D and the second cylinder 12B as shown in FIG. 6E.

[0023] Disposed at the top of the combustion chamber 14 in each of the cylinders 12A-12D is a spark plug 15 for igniting and combusting a mixture in the combustion chamber 14 with an electrode of the spark plug 15 at a far end thereof located in the combustion chamber 14. There is installed a fuel injector 16 on one side (right side as illustrated in FIG. 1) of the combustion chamber 14 for injecting fuel directly into the combustion chamber 14.

The fuel injector 16 is oriented such that it sprays the fuel toward the vicinity of the electrode of the spark plug 15. Incorporating a needle valve and a solenoid which are not illustrated, the fuel injector 16 is actuated by a pulse signal fed from the ECU 2. When this pulse signal is input, the fuel injector 16 opens for a period of time corresponding to the pulselength of the pulse signal to inject the fuel in a quantity corresponding to valve opening time into the cylinder 12. The fuel is supplied to each fuel injector 16 from a fuel pump (not shown) via a fuel supply channel. A fuel supply system of the engine system E is constructed in such a way that the fuel supply system produces a fuel supply pressure higher than pressure in the combustion chamber 14 during each successive compression stroke.

[0024] There are formed intake ports 17 and exhaust ports 18 opening into the combustion chambers 14 of the individual cylinders 12A-12D at upper portions thereof with intake valves 19 and exhaust valves 20 provided in the intake ports 17 and the exhaust ports 18, respectively. The intake valves 19 and the exhaust valves 20 are actuated by valve actuating mechanisms (not shown) including camshafts. As already mentioned, opening and closing timing of the intake and exhaust valves 19, 20 of the individual cylinders 12A-12D is preset such that the cylinders 12A-12D undergo the combustion cycles with the aforementioned successive phase delay.

[0025] Instead of structuring the valve actuating mechanisms as stated above, the camshafts of the intake valves 19 may be provided with variable valve actuating mechanisms of the prior art such that at least the closing timing of the intake valves 19 can be advanced or retarded by control commands from the ECU 2 according to operating conditions of the engine 1, such as engine load and engine speed.

[0026] The intake ports 17 are connected to an intake passage 21 while the exhaust ports 18 are connected to an exhaust passage 22. A downstream part of the intake passage 21 close to the intake ports 17 is divided into four independent branched intake channels 21a which are connected to the individual cylinders 12A-12D as shown in FIG. 2. Upstream ends of these branched intake channels 21a are connected to a surge tank 21b. A portion of the intake passage 21 further upstream of the surge tank 21b constitutes a common intake passage portion 21c which supplies air to all of the cylinders 12A-12D. Provided in the common intake passage portion 21c is a throttle valve (intake air quantity regulator) 23 associated with an actuator 24 for driving the throttle valve 23 which is a butterfly valve, for example, for regulating intake airflow by varying the cross-sectional internal area of the common intake passage portion 21c. Further provided in the common intake passage portion 21c are an airflow sensor 25 for detecting the quantity of intake air and an intake air pressure sensor 26 for detecting intake air pressure (negative pressure). The airflow sensor 25 and the intake air pressure sensor 26 which are illustrated only in FIG. 2 are disposed upstream and downstream of the throttle valve 23, respectively.

[0027] Downstream of a joint part of branched exhaust channels where exhaust gas discharged from the individual cylinders 12A-12D are collected into the exhaust passage 22, there is provided a catalyst 29 for converting the exhaust gas. The catalyst 29 is, for example, a three-way catalyst which exhibits an extremely high converting efficiency with respect to hydrocarbons (HC), carbon monoxide (CO) and nitrogen oxides (NOx) when the exhaust gas has an air-fuel ratio approximately equal to the stoichiometric air-fuel ratio. The three-way catalyst 29 has a capability to adsorb oxygen under oxygen-rich conditions in which oxygen concentration is relatively high. When the oxygen concentration in the exhaust gas becomes relatively low, on the other hand, the three-way catalyst 29 releases adsorbed oxygen which reacts with HC and CO, for instance. It is to be noted that the catalyst 29 need not necessarily be the three-way catalyst but may be any kind of catalyst having a similar oxygen-adsorbing capability. One example of such catalysts is lean NOx catalyst which can convert NOx even under oxygen-rich environments.

[0028] The engine 1 is further provided with an alternator 28 which is connected to the crankshaft 3 by a belt, for example. Although not illustrated in detail, the alternator 28 has a built-in regulator circuit 28a which regulates the amount of electric power generated by the alternator 28 by controlling a current flowing through a field coil to vary output voltage according to a control command (e.g., voltage command) fed from the ECU 2 to the regulator circuit 28a. Essentially, the amount of electric power generated is controlled according to the amount of load of on-board electric devices and battery. When the amount of electric power generated by the alternator 28 is altered in this way, the amount of output power of the alternator 28, or the amount of external load applied to the engine 1, varies correspondingly.

[0029] The engine system E includes a pair of first and second crank angle sensors 30, 31 for detecting the angle of rotation of the crankshaft 3. The ECU 2 determines the engine speed based mainly on a signal output from one of these crank angle sensors, or the first crank angle sensor 30 and, more specifically, detects the direction and angle of rotation of the crankshaft 3 from mutually phase-offset crank angle signals (first crank angle signal CA1 and second crank angle signal CA2) output from the two crank angle sensors 30, 31 as will be later discussed in detail. The engine system E further includes cam angle sensors 32 for detecting specific rotational positions of the aforementioned camshafts and outputting detected signals as cylinder identification signals. Other constituent components of the engine system E for controlling the engine 1 include a water temperature sensor (not shown) for detecting the temperature of engine cooling water and an accelerator depression sensor 34 for detecting the amount of depression of an accelerator pedal.

[0030] Upon receiving signals from the individual sensors 25, 26, 30-32 and 34 mentioned above, the ECU 2 outputs a signal for controlling the quantity of fuel to be injected and fuel injection timing to each of the fuel injectors 16, a signal for controlling ignition timing to an ignition device 27 for actuating the individual spark plugs 15, and a signal for controlling the opening of each of the throttle valve 23 to the actuator 24.

[0031] While a detailed description is given below, the ECU 2 automatically stops the engine 1 by cutting fuel supply to the individual cylinders 12A-12D when specific conditions for engine stop are met during engine idle operation, and the ECU 2 automatically restarts the engine 1 when specific conditions for engine restart, such as depression of the accelerator pedal by a driver, are met subsequently.

[0032] The ECU 2 causes the engine 1 to restart by its own motive power without the aid of a starter motor. In this embodiment, the engine 1 is restarted as shown schematically in FIGS. 3A to 3D. Specifically, when the engine 1 is at rest due to idle stop, the ECU 2 produces initial combustion in the cylinder 12 (the first cylinder 12A in the illustrated example) of which piston 13 is stopped halfway on the compression stroke to lower the piston 13 and thereby turn the crankshaft 3 slightly in a reverse running direction as shown in FIG. 3A. As a consequence, the piston 13 in the cylinder 12 (the second cylinder 12B in the illustrated example) which was on the expansion stroke at engine stop ascends, thereby compressing a mixture in the same cylinder 12 as shown in FIG. 3B. Then, the ECU 2 causes the spark plug 15 of the cylinder 12 which was initially on the expansion stroke to ignite and burn the compressed mixture of which temperature and pressure have been increased to produce a torque acting on the crankshaft 3 in a forward running direction thereof to restart the engine 1.

[0033] To cause the engine 1 to restart by its own motive power alone, it is necessary to generate as great a torque as possible acting on the crankshaft 3 in the forward running direction by producing combustion in the aforementioned cylinder 12B which was on the expansion stroke at engine stop so that the piston 13 in the cylinder 12A which will reach the compression stroke top dead center (TDC) as shown in FIG. 3C can next go beyond TDC overwhelming a compressive reaction force (compressive pressure) exerted by the cylinder 12A itself. It is therefore essential for the cylinder 12B which was on the expansion stroke at engine stop to hold a sufficient quantity of air necessary for combustion to ensure reliable engine restart.

[0034] When automatically stopping the engine 1 at idle, the ECU 2 controls the engine 1 in such a manner that the fuel supply is cut at a specific engine speed which is slightly higher than idle speed so that gases are sufficiently driven out of the individual cylinders 12A-12D and, then, the throttle valve 23 is opened and kept at a preset valve opening for a specific period in the present embodiment. The throttle valve 23 is subsequently closed with predetermined appropriate timing so that air is introduced in sufficient quantities into the cylinder 12A which is on the compression stroke at engine stop and into the cylinder 12B which is on the expansion stroke at engine stop. In particular, the ECU 2 controls the engine 1 in such a manner that a larger quantity of air is introduced into the cylinder 12B which is on the expansion stroke at engine stop than into the cylinder 12A which is on the compression stroke at engine stop.

[0035] Controlling the engine 1 in this way ensures that the piston 13 in the expansion stroke cylinder 12B stops at a position within a later-described specified range R suited for engine restart more or less closer to the bottom dead center (BDC) than the middle of the expansion stroke due to a proper balance between compressive air pressures exerted by the two cylinders 12A and 12B.

ENGINE STOP CONTROL OPERATION

[0036] Engine stop control operation performed by the ECU 2 is described in detail referring to FIGS. 4 to 7. FIGS. 4 and 5 are flowcharts showing first and second halves of an engine stop control routine. FIGS. 6A-6E are diagrams schematically showing how the engine speed and crank angle vary in successive strokes of the individual cylinders 12A-12D during an engine stopping period which begins a point of fuel supply interruption and ends at a point of engine stop during which the engine 1 continues to run by inertia as well as how the throttle opening is controlled and the intake air pressure (intake air pipe negative pressure) varies during the same period. FIG. 7 is a distribution chart showing how decreases in later discussed TDC engine speed during the engine stopping period are correlated with piston stop positions in the cylinder 12 which will be on the expansion stroke at engine stop.

[0037] Referring to FIG. 6A, if the fuel supply is cut at time t0 when the engine 1 is at a preset engine speed (approximately 800 rpm in the illustrated example), kinetic energy possessed by various moving parts, such as the crankshaft 3 and a flywheel, is consumed by mechanical friction and pumping work, for instance, so that the engine speed gradually decreases after fuel supply interruption. The engine 1 eventually stops after turning several times by inertia. During the engine stopping period when the engine 1 continues to run by inertia, the engine speed briefly decreases considerably and increases again in repetitive cycles as the pistons 13 in the individual cylinders 12A-12D reach and go beyond the compression stroke TDC. The engine speed gradually approaches zero while going up and down repetitively in this way. If the fuel supply is cut at an engine speed of approximately 800 rpm as depicted in FIG. 6A, for example, the engine 1 typically goes through 8 or 9 successive TDCs. When the piston 13 in a particular cylinder 12 can not go beyond succeeding TDC after the piston 13 has lastly gone beyond TDC (time t3), the engine 1 can not turn any longer and eventually stops (times t4, t5, t6).

[0038] More specifically, if the piston 13 in a particular cylinder 12 fails to go beyond the compression stroke TDC, that cylinder 12 (the first cylinder 12A in the illustrated example) is left on the compression stroke. As the piston 13 in the cylinder 12 ascends by inertial force, the air pressure in the cylinder 12 increases and a resultant compressive reaction force causes the piston 13 to be momentarily stopped (time t4) and forced back toward BDC. This backward movement of the piston 13 causes the crankshaft 3 to turn in the reverse direction so that the engine speed becomes negative as shown in FIG. 6A. Should this situation occur, the air pressure in the cylinder 12 (the second cylinder 12B in the illustrated example) which is currently on the expansion stroke after lastly going beyond the compression stroke TDC increases, producing a compressive reaction force acting on the piston 13 in this cylinder 12 toward BDC. This compressive reaction force causes the piston 13 in the expansion stroke cylinder 12 to be momentarily stopped (time t5) and forced toward BDC. This movement of the piston 13 in the expansion stroke cylinder 12 causes the crankshaft 3 to turn again in the forward direction so that the engine speed reverts to a positive value.

[0039] As the compressive reaction forces act on the pistons 13 in the compression stroke cylinder 12 and the expansion stroke cylinder 12 in opposite directions as discussed above, the pistons 13 in the individual cylinders 12A-12D stop (time t6) after moving up and down a few times. While positions where these pistons 13 stop are determined generally by the balance between the compressive reaction forces exerted by the compression stroke cylinder 12 and the expansion stroke cylinder 12, the piston stop positions are more or less affected by mechanical friction occurring in the engine 1, so that the piston stop positions vary with the rotational inertia of the engine 1, or with the engine speed, at a point in time when one of the pistons 13 lastly goes beyond TDC before eventual engine stop.

[0040] To ensure that the piston 13 in the cylinder 12 which will be on the expansion stroke at idling engine stop will stop within the aforementioned specified range R suited for engine restart, it is necessary to regulate the quantities of air introduced into the two cylinders 12 which will be on the expansion stroke and on the compression stroke at engine stop in such a manner that sufficiently large compressive reaction forces will be exerted by the two cylinders 12 with the compressive reaction force exerted by the expansion stroke cylinder 12 exceeding the compressive reaction force exerted by the compression stroke cylinder 12 by at least a specific amount. To achieve this, the ECU 2 of the present embodiment opens the throttle valve 23 immediately after interrupting the fuel supply (time t1) and closes the throttle valve 23 after a lapse of a specific time period (time t2) as shown in FIG. 6C so that the intake air pipe negative pressure decreases (and the intake air quantity increases) for a while as shown in FIG. 6D and, as a result, specific quantities of air are introduced into the cylinders 12 which will be on the expansion stroke and on the compression stroke at engine stop.

[0041] In the actual engine 1, however, individual components, such as the throttle valve 23, the intake ports 17 and the branched intake channels 21a, vary in shape and airflows drawn through these components exhibit different behaviors, causing a certain degree of variations in the quantities of air introduced into the individual cylinders 12A-12D during the engine stopping period. Therefore, even when the throttle valve 23 is controlled to open and close as discussed above, it is difficult to ensure that the pistons 13 in the cylinders 12 which will be on the expansion stroke and on the compression stroke at engine stop will stop at positions exactly within the aforementioned targeted range R.

[0042] In this embodiment, particular attention is given to the fact that there is a distinct correlation between the aforementioned TDC engine speed (which is the engine speed measured when any of the pistons 13 is at the compression stroke TDC) during a process of gradual engine speed decrease in the engine stopping period and the position where the piston 13 in the cylinder 12 which will be on the expansion stroke at engine stop will stop as shown in FIG. 7. Taking this correlation into account, the ECU 2 detects the TDC engine speed at successive TDCs occurring at intervals of 180° crank angle during the aforementioned process of gradual engine speed decrease shown in FIG. 6A and adjusts the rate of engine speed decrease by regulating the amount of electric power generated by the alternator 28 and the opening of the throttle valve 23 according to detected engine speed values during the engine stopping period.

[0043] After the fuel supply is cut at the engine speed of approximately 800 rpm as stated above, the ECU 2 holds the throttle valve 23 open for a specific time period during which the ECU 2 measures the engine speed each time the pistons 13 in the individual cylinders 12A-12D successively go beyond TDC while the engine 1 continues to run by inertia. The distribution chart of FIG. 7 shows the correlation between the TDC engine speed so measured during the engine stopping period and the position where the piston 13 in the cylinder 12 which is on the expansion stroke at engine stop stops, the horizontal axis of the chart representing the TDC engine speed and the vertical axis of the chart representing the piston stop position. The distribution chart of FIG. 7 showing the correlation between the TDC engine speed and the piston stop position is obtained by repeating the aforementioned measurement and plotting cycles for a specific number of times during the engine stopping period.

[0044] The distribution chart of FIG. 7 does not show the engine speed observed when one of the pistons 13 lastly goes beyond TDC but shows plots of TDC engine speeds observed from a point immediately after fuel supply interruption (or at the ninth TDC from the last TDC) up to a point of TDC immediately before the last TDC (or at the second TDC from the last TDC). As can be seen from FIG. 7, the TDC engine speeds observed at the ninth to second TDCs from the last TDC are distributed in distinct groups. It is apparent from the plots of TDC engine speeds measured particularly at the sixth to second TDCs from the last TDC that the piston 13 in the cylinder 12 which will be on the expansion stroke at engine stop will stop within the aforementioned specified range R (100° to 120° after TDC, or ATDC, in terms of crank angle in the illustrated example) suited for engine restart if the measured TDC engine speeds fall within particular ranges (which are hatched in FIG. 7).

[0045] The aforementioned particular ranges of the TDC engine speeds at which the piston 13 in the cylinder 12 which will be on the expansion stroke at engine stop will stop within the specified range R suited for engine restart are hereinafter referred to as appropriate TDC engine speed ranges. In this embodiment, the ECU 2 detects the TDC engine speed for each of the cylinders 12A-12D as the engine speed gradually decreases while going up and down repetitively as shown in FIG. 6A, compares TDC engine speed values thus detected with the appropriate TDC engine speed ranges, and regulates the amount of electric power generated by the alternator 28 and the opening of the throttle valve 23 according to deviations of the TDC engine speeds from the appropriate TDC engine speed ranges as will be later discussed in detail.

[0046] During a specific period immediately following the fuel supply interruption, the throttle valve 23 is kept relatively wide-open for driving gases out of the individual cylinders 12A-12D as mentioned earlier. Even if the opening of the throttle valve 23 is further adjusted, the amount of pumping work produced by the individual cylinders 12A-12D does not vary so much that it is difficult to adjust the engine speed by regulating the opening of the throttle valve 23 during this period. Thus, the ECU 2 intentionally activates the alternator 28 and controllably varies the amount of electric power generated to regulate a driving force produced thereby, so that the rate of engine speed decrease can be adjusted. During this period, the ECU 2 adjusts the amount of electric power generated by the alternator 28 at a larger value so that the TDC engine speed is kept closer to a lower limit of each appropriate TDC engine speed range, that is, the engine speed is kept relatively low.

[0047] After the aforementioned specific period immediately following the fuel supply interruption has elapsed, the ECU 2. controls the opening of the throttle valve 23 to regulate the amount of pumping work produced by the individual cylinders 12A-12D of the engine 1 and thereby adjust the rate of engine speed decrease. In a case where the throttle valve 23 is disposed upstream of the surge tank 21b, however, the quantities of air introduced into the individual cylinders 12A-12D do not change so sharply even if the opening of the throttle valve 23 is decreased. Therefore, the alternator 28 is controlled to keep the engine speed relatively low as mentioned above at first in the aforementioned period immediately following the fuel supply interruption, and the opening of the throttle valve 23 is increased to moderate the rate of engine speed decrease when the TDC engine speed becomes lower than the appropriate TDC engine speed range.

[0048] If the rate of engine speed decrease is adjusted by controlling the amount of electric power generated by the alternator 28 and regulating the opening of the throttle valve 23 in this way so that the TDC engine speed falls within the appropriate TDC engine speed range before one of the pistons 13 lastly goes beyond TDC at the latest, kinetic energy possessed by various moving parts, such as the crankshaft 3, the flywheel, the pistons 13 and connecting rods, combined with potential energy possessed by high-pressure air in the compression stroke cylinder 12 becomes balanced with mechanical friction and other reaction forces which will subsequently be exerted. Consequently, it becomes possible to stop the piston 13 in the cylinder 12 which will be on the expansion stroke at engine stop.

[0049] An example of a procedure of the aforementioned engine stop control operation is now described specifically referring to the flowcharts of FIGS. 4 and 5.

Operation flow shown in FIGS. 4 and 5 starts with specific timing while the engine 1 is running. First, the ECU 2 judges whether specific idle stop conditions have been met in step SA1. This judgment is made based on vehicle speed, braking conditions, engine cooling water temperature, for instance. Specifically, the ECU 2 judges that the idle stop conditions have been met if the vehicle speed is lower than a specific value, brakes are applied, the engine cooling water temperature is within a specific range and it is not particularly inconvenient to stop the engine 1, for example.

[0050] When the idle stop conditions have been satisfied (Yes in step SA1 in the flowchart of FIG. 4), the ECU 2 judges whether specific conditions necessary for specified one of the cylinders 12 (the first cylinder 12A, for example) for stopping the engine 1 have been met in step SA2. More specifically, the ECU 2 judges whether the engine 1 has slowed down to the aforementioned preset engine speed (approximately 800 rpm in this embodiment) at which the fuel supply should be cut and the specified cylinder 12 is currently on a predetermined stroke (e.g., the intake stroke).

[0051] When all of these conditions have been satisfied (Yes in step SA2), the ECU 2 proceeds to step SA3 and interrupts the fuel supply to the individual cylinders 12A-12D. Then, the ECU 2 opens the throttle valve 23 to a predefined opening in step SA4 in step SA5. Thus, the quantities of air introduced into the individual cylinders 12A-12D increase as shown in FIGS. 6C and 6D and gases are sufficiently driven out of the cylinders 12A-12D. As a consequence, an ample amount of fresh air is supplied to the catalyst 29 disposed in the exhaust passage 22 and the catalyst 29 adsorbs a sufficiently large quantity of oxygen.

[0052] Proceeding next to step SA5, the ECU 2 judges whether the TDC engine speed obtained from the first crank angle signal CA1 output from the first crank angle sensor 30 falls within one of the aforementioned appropriate TDC engine speed ranges. If the measured TDC engine speed falls within one of the appropriate TDC engine speed ranges (Yes in step SA5), the ECU 2 proceeds to step SA6 and judges whether the engine speed is equal to or lower than a predefined value. This predefined value is for closing the throttle valve 23 at such a timing that the quantity of air introduced into the cylinder 12 (the second cylinder 12B in the illustrated example) which will be on the expansion stroke at engine stop becomes larger than the quantity of air introduced into the cylinder 12 (the first cylinder 12A in the illustrated example) which will be on the compression stroke at engine stop taking into consideration a delay in transporting intake air as shown in FIGS. 6C and 6D. Specifically, this timing corresponds to time t2 and the aforementioned predefined value of the engine speed is set within a range of approximately 500 to 600 rpm, for example, in this embodiment.

[0053] When the engine speed becomes equal to or lower than the predefined value (Yes in step SA6), the ECU 2 proceeds to step SA9. If the engine speed is still higher than the predefined value (No in step SA6), on the other hand, the ECU 2 returns to step SA5.

[0054] If the TDC engine speed is judged to be out of the appropriate TDC engine speed range (No in step SA5), the ECU 2 proceeds to step SA7, in which the ECU 2 calculates the amount of electric power to be generated by the alternator 28 based on the deviation of the TDC engine speed from the appropriate TDC engine speed range. The amount of electric power to be generated by the alternator 28 is read out of a preprogrammed map which defines the amounts of electric power to be generated according to deviations of the engine speed from the appropriate TDC engine speed ranges, for example. Specifically, if the TDC engine speed is higher than an upper limit of the appropriate TDC engine speed range, for instance, the amount of electric power generated by the alternator 28 is increased so that the engine load would increase. If the TDC engine speed is lower than the lower limit of the appropriate TDC engine speed range, on the other hand, the amount of electric power generated by the alternator 28 is decreased so that the engine load would decrease. In the aforementioned map, target values of the amount of electric power to be generated are preset at more or less larger values such that the TDC engine speed is kept close to each appropriate TDC engine speed range.

[0055] In step SD8 succeeding to step SD7, the ECU 2 outputs a control command to the regulator circuit 28a in accordance with the result of the aforementioned calculation to cause the alternator 28 to generate electric power accordingly. As the alternator 28 generates electric power according to the control command, the engine load is properly regulated so that a plotted curve of the turning speed of the engine 1 which continues to run by inertia is caused to shift toward a higher-speed or lower-speed side and the engine speed gradually approaches a targeted curve. When the engine speed eventually becomes equal to or lower than the predefined value as a consequence (Yes in step SA6), the ECU 2 proceeds to step SA9, in which the ECU 2 closes the throttle valve 23. Then, the ECU 2 proceeds to step SA10 of FIG. 5.

[0056] In step SA10 of the operation flow shown in FIG. 5, the ECU 2 judges whether the current TDC engine speed falls within one of the appropriate TDC engine speed ranges in the same manner as in step SA5. If the measured TDC engine speed falls within one of the appropriate TDC engine speed ranges (Yes in step SA10), the ECU 2 proceeds to step SA11. If the TDC engine speed is judged to be out of the appropriate TDC engine speed range (No in step SA10), on the other hand, the ECU 2 proceeds to step SA12, in which the ECU 2 calculates a target opening of the throttle valve 23 based on the deviation of the TDC engine speed from the appropriate TDC engine speed range. The target opening of the throttle valve 23 is read out of a preprogrammed map which defines valve openings to be achieved according to the current engine speed, deviations of the engine speed from the appropriate TDC engine speed ranges and the current valve opening, for example. Data preprogrammed in the map is such that the ECU 2 would increase the throttle valve opening as shown by "TVO" in FIG. 6C to decrease the amount of pumping work done by the individual cylinders 12A-12D of the engine 1 when the measured TDC engine speed is lower than the lower limit of the appropriate TDC engine speed range, and the ECU 2 would not vary the throttle valve opening when the measured TDC engine speed is higher than the upper limit of the appropriate TDC engine speed range.

[0057] When the throttle valve 23 is located upstream of the surge tank 21b as in the aforementioned structure of the present embodiment, engine response to throttling action for restricting the intake airflow has a larger delay so that adequate controllability can not be achieved. For this reason, the amount of electric power generated by the alternator 28 is increased as mentioned above to increase the engine load, as necessary, so that the rate of engine speed decrease would increase (or the engine speed would decrease), whereas the rate of engine speed decrease is lessened by increasing the opening of the throttle valve 23 only when the TDC engine speed is lower than the lower limit of the appropriate TDC engine speed range. In step SA13 succeeding to step SD12, the ECU 2 activates the actuator 24 to increase the opening of the throttle valve 23, as necessary, and then proceeds to step SA11.

[0058] The ECU 2 regulates the rate of engine speed decrease after fuel supply interruption by controlling the alternator 28 and the throttle valve 23 in the aforementioned manner, whereby the plotted curve of the engine speed which gradually decreases while going up and down repetitively can be adjusted step by step so that the TDC engine speed would fall within the appropriate TDC engine speed range before one of the pistons 13 lastly goes beyond TDC at the latest.

[0059] In step SA11, the ECU 2 judges whether the TDC engine speed detected in step SA10 is equal to or lower than a predefined value "A". This predefined value "A" is an experimentally determined preset value corresponding to the TDC engine speed lastly measured before engine stop. If the TDC engine speed obtained in step SA10 is higher than the predefined value "A" (No in step SA11), the engine 1 has not gone beyond the last TDC yet. In this case, the ECU 2 returns to step SA10 and continues to regulate the throttle valve 23.

[0060] If the TDC engine speed obtained in step SA10 is equal to or lower than the predefined value "A" (Yes in step SA11), the engine 1 has already gone beyond the last TDC. In this case, the engine 1 will stop after turning alternately in the forward and reverse directions a few times due to the compressive reaction forces exerted by the two cylinders 12 which are on the compression stroke and on the expansion stroke.

[0061] Subsequently, the ECU 2 proceeds to step SA14, in which the ECU 2 judges whether the engine 1 has completely stopped based on the first and second crank angle signals CA1, CA2 output from the two crank angle sensors 30, 31. If the judgment result in step SA14 is in the affirmative with the engine 1 judged to have completely stopped, the ECU 2 proceeds to step SA22 to perform a later discussed stop position detecting subroutine (FIGS. 8, 9A and 9B) in which the ECU 2 detects the piston stop position in the cylinder 12 which is on the expansion stroke at engine stop based on the mutually phase-offset crank angle signals CA1, CA2 output from the two crank angle sensors 30, 31. The ECU 2 stores the detected piston stop position in an internal memory and quits the engine stop control routine of FIGS. 4 and 5.

[0062] Since the crankshaft 3 repeatedly turns in the forward and reverse directions a few times as mentioned above immediately before the engine 1 stops, it is impossible to detect the piston stop position by just counting pulses of the first crank angle signal CA1 output from the first crank angle sensor 30 alone. For this reason, the ECU 2 determines the piston stop positions in the individual cylinders 12A-12D in terms of crank angle with respect to TDC or BDC by detecting the direction and angle of rotation of the crankshaft 3 based on the mutually phase-offset crank angle signals CA1, CA2 output from the two crank angle sensors 30, 31 as will be discussed in detail below.

[0063] FIG. 8 is a flowchart showing the aforementioned stop position detecting subroutine for detecting the piston stop position in the cylinder 12 which is on the expansion stroke at engine stop. After operation flow shown in FIG. 8 has begun, the ECU 2 first judges in step SC1, based on the first crank angle signal CA1 output from the first crank angle sensor 30 and the second crank angle signal CA2 output from the second crank angle sensor 31, whether the second crank angle signal CA2 is Low at each rising edge of the first crank angle signal CA1 and High at each falling edge of the first crank angle signal CA1 or the second crank angle signal CA2 is Low at each falling edge of the first crank angle signal CA1 and High at each rising edge of the first crank angle signal CA1. In other words, the ECU 2 judges whether the first and second crank angle signals CA1, CA2 are phase-offset as shown in FIG. 9A indicating that the crankshaft 3 is turning in the forward direction or the first and second crank angle signals CA1, CA2 are phase-offset as shown in FIG. 9B indicating that the crankshaft 3 is turning in the reverse direction in step SC1.

[0064] More specifically, when the engine 1 runs in the forward direction, the second crank angle signal CA2 lags the first crank angle signal CA1 in phase by about half the pulselength and, therefore, the second crank angle signal CA2 becomes Low at the rising edge of each successive pulse of the first crank angle signal CA1 and High at the falling edge of each successive pulse of the first crank angle signal CA1 as shown in

FIG. 9A. When the engine 1 runs in the reverse direction, on the contrary, the second crank angle signal CA2 leads the first crank angle signal CA1 in phase by about half the pulselength and, therefore, the second crank angle signal CA2 becomes High at the rising edge of each successive pulse of the first crank angle signal CA1 and Low at the falling edge of each successive pulse of the first crank angle signal CA1 as shown in FIG. 9B.

[0065] If the engine 1 is judged to be running in the forward direction (Yes in step SC1), the ECU 2 increments a count value of a crank angle counter for measuring changes in the crank angle. If the engine 1 is judged to be running in the reverse direction (No in step SC1), on the contrary, the ECU 2 decrements the count value of the crank angle counter. The rising edges and the falling edges of the first crank angle signal CA1 and those of the second crank angle signal CA2 occur at intervals of a specific angle of rotation of the crankshaft 3. In this embodiment, the interval from one rising edge to the next, and from one falling edge to the next, is approximately 10° for both the first and second crank angle signals CA1, CA2. Since the intervals between the successive rising edges and between the falling edges are preset as stated above, it is possible to judge whether the engine 1 is running in the forward or reverse direction based on the state of the second crank angle signal CA2 at each rising edge and falling edge of the first crank angle signal CA1 and to determine the angle of rotation of the crankshaft 3 from the numbers of rising edges or falling edges of the first and second crank angle signals CA1, CA2. Therefore, the ECU 2 can determine the exact piston stop position in the cylinder 12 which is on the expansion stroke at engine stop by detecting the angle of rotation of the crankshaft 3 even though the crankshaft 3 turns in both the forward and reverse directions before the engine 1 completely stops.

[0066] In the aforementioned operation flow shown in FIGS. 4 and 5, step SA3 constitutes an engine stopping section 2a for stopping the engine 1 by interrupting the fuel supply to the individual cylinders 12A-12D while the engine 1 is running, and steps SA4, SA6 and SA9 together constitute an intake air quantity regulator controlling section 2b for controlling the throttle valve 23 such that the quantities of air introduced into the individual cylinders 12 increase for a specific period of time in the engine stopping period which begins at the point of fuel supply interruption by the engine stopping section 2a.

[0067] According to the engine stop control operation of the embodiment so far described, the ECU 2 opens the throttle valve 23 for the aforementioned specific period of time following the point of fuel supply interruption at first so that necessary quantities of air are introduced into the cylinders 12 which will be on the expansion stroke and on the compression stroke at engine stop and then controls the alternator 28 and the throttle valve 23 to regulate the rate of engine speed decrease as appropriate when automatically stopping the engine 1 at idle by cutting the fuel supply. Since the ECU 2 controls the engine 1 in this fashion during the engine stopping period, it is possible to cause the piston 13 in the cylinder 12 which will be on the expansion stroke at engine stop to stop at a position within the aforementioned specified range R suited for engine restart.

[0068] Also, as the throttle valve 23 is opened for the aforementioned specific period of time in the engine stopping period, almost all of burned gas is driven out of the individual cylinders 12A-12D and the cylinders 12A-12D are filled with fresh air with the catalyst 29 disposed in the exhaust passage 22 adsorbing a large quantity of oxygen. Although the intake and exhaust valves 19, 20 of the cylinders 12 which are on the expansion stroke and on the compression stroke are closed at engine stop, in-cylinder air even in these cylinders 12 quickly leaks out after engine stop. Therefore, the individual cylinders 12A-12D are filled with fresh air approximately at atmospheric pressure shortly after the engine 1 stops, the quantities of air held in current volumes of the combustion chambers 14 of the individual cylinders 12A-12D corresponding to the piston stop positions therein.

ENGINE RESTART CONTROL OPERATION

[0069] Engine restart control operation performed by the ECU 2 for automatically restarting the engine 1 which has been stopped at idle is now described referring mainly to FIGS. 3A-3D, 10, 11, 12 and 13A-13F, of which FIGS. 10 and 11 are flowcharts showing first and second halves of an engine restart control routine, FIG. 12 is a diagram showing fuel injection and ignition timings for the individual cylinders 12A-12D at engine restart in relation to the timing of the successive strokes and open/closed states of the intake and exhaust valves 19, 20, and FIGS. 13A-13F are time charts showing how the pressure in the individual cylinders 12A-12D, the amount of torque generated by the engine 1 and the engine speed vary as a result of fuel injection and ignition in the individual cylinders 12A-12D during engine restart.

[0070] The engine restart control operation of this embodiment is intended to restart the engine 1 by its own motive power only as mentioned earlier. Specifically, the initial combustion is produced in the cylinder 12 (the first cylinder 12A in the illustrated example) of which piston 13 is stopped halfway on the compression stroke at engine stop as shown in FIG. 3A and by the symbols a1 and a2 in FIG. 12 to cause the engine 1 to once turn in the reverse direction. As a result, a mixture produced in the cylinder 12 (the second cylinder 12B in the illustrated example) of which piston 13 is stopped halfway on the expansion stroke at engine stop is compressed, and the mixture in the expansion stroke cylinder 12B of which temperature and pressure have increased is ignited and burnt as shown in FIG. 3B and by the symbols a3 and a4 in FIG. 12. Consequently, a sufficiently high combustion pressure is produced in the expansion stroke cylinder 12B and the distance traveled by the piston 13 in the same cylinder 12B increases, making it possible to obtain a large starting torque.

[0071] If the initial combustion is produced in the cylinder 12A which is on the compression stroke at engine stop as stated above, however, the first cylinder 12A is filled with burned gas and, as a consequence, a fairly large compressive reaction force is exerted on the piston 13 in the compression stroke cylinder 12A when the piston 13 reaches the first TDC after engine restart as shown in FIG. 3C, resulting in a potential failure to restart the engine 1 due to an inability of the piston 13 to go beyond the first TDC. Even if the piston 13 in the compression stroke cylinder 12A can go beyond the first TDC overwhelming the compressive reaction force, the engine speed will greatly decrease at TDC and the starting torque can not be obtained by subsequent combustion in the cylinder 12A which is filled with the burned gas, so that the engine speed can not be smoothly increased.

[0072] In the example shown in FIGS. 3A-3D, the third cylinder 12C is a cylinder in which a mixture should normally be ignited after combustion in the first cylinder 12A. In the illustrated example, the piston 13 in the third cylinder 12C stops on the intake stroke at engine stop. While the engine 1 is at idle stop, air in the third cylinder 12C is heated due to heat dissipation from the cylinder wall. Since the temperature of air existing in the third cylinder 12C is well increased and the third cylinder 12C is fully charged with relatively warm air introduced through the intake passage 21 at approximately atmospheric pressure, a fairly large compressive reaction force is exerted on the piston 13 in the third cylinder 12C when the same cylinder 12C is on the compression stroke. In addition, as the temperature and pressure in the third cylinder 12C increase when the in-cylinder air is compressed, autoignition is likely to occur very easily in the third cylinder 12C on the compression stroke.

[0073] FIGS. 14A and 14B illustrate how the pressure in the cylinder 12C which is on the intake stroke at engine stop increases. Specifically, as the third cylinder 12C is filled with high-temperature air, the pressure in the same cylinder 12C begins to rise from an early part of the compression stroke as shown in FIG. 14A. The pressure in the third cylinder 12C increases with the lapse of time, producing a torque acting in the engine-reversing direction (negative torque) as shown in FIG. 14B. If autoignition occurs in the third cylinder 12C in a latter part of the compression stroke thereof as shown by imaginary lines in FIGS. 14A and 14B, the in-cylinder pressure sharply rises and, as a consequence, a large reversing torque acts on the engine 1, resulting in an eventual failure in engine restart.

[0074] In light of the aforementioned problem, the invention offers the following feature in the engine restart control operation of the present embodiment. Specifically, a torque acting in the engine-reversing direction is first produced by the initial combustion in the cylinder 12A which is on the compression stroke at engine stop to sufficiently compress the mixture produced in the cylinder 12B which is on the expansion stroke at engine stop, so that subsequent combustion in the cylinder 12B would generate a significantly increased starting torque acting in the forward running direction. When gas in the cylinder 12A which was on the compression stroke at engine stop is compressed as a result of the forward running motion of the engine 1, an additional quantity of fuel is injected into the compression stroke cylinder 12A at a point shown by the symbol a5 in FIG. 12. Consequently, the compressive pressure occurring in the compression stroke cylinder 12A is lessened due to a cooling effect produced by absorption of latent heat by evaporation of the injected fuel, whereby the piston 13 in the compression stroke cylinder 12A can reliably go beyond the first TDC after engine restart as shown in FIG. 3C and the engine 1 can continue to run in the forward direction.

[0075] Also, fuel is injected into the third cylinder 12C which was on the intake stroke at engine stop in which the mixture should be ignited after combustion in the first cylinder 12A which was initially on the compression stroke when the cylinder 12C has transferred to the compression stroke and the in-cylinder temperature and pressure have increased. Consequently, the compressive pressure occurring in the cylinder 12C which was initially on the intake stroke is lessened due to the cooling effect produced by absorption of latent heat by evaporation of the injected fuel, whereby autoignition in the same cylinder 12C is prevented. A mixture produced in the cylinder 12C is ignited and burnt when the piston 13 in the cylinder 12C has gone beyond TDC to give a forward-acting torque to the engine 1.

[0076] A specific example of a procedure of the engine restart control operation is described referring to the flowcharts of FIGS. 10 and 11. Operation flow shown in FIGS. 10 and 11 begins after the engine 1 has been stopped by the aforementioned engine stop control routine (FIGS. 4 and 5). First in step SB1, the ECU 2 judges whether the earlier-mentioned specific conditions for engine restart have been met. The engine restart conditions include such conditions as the brakes are released or the accelerator pedal is depressed for restarting the engine 1 from idle stop, or an air conditioner is switched on requiring the engine 1 to be restarted. If none of such engine restart conditions have been satisfied yet (No in step SB1), the ECU 2 waits until the conditions are met. When any of such conditions have been satisfied (Yes in step SB1), on the other hand, the ECU 2 proceeds to step SB2.

[0077] In step SB2, the ECU 2 calculates the quantities of air in the cylinder 12 which is on the compression stroke at engine stop (the first cylinder 12A as illustrated in FIGS. 3A-3D) and in the cylinder 12 which is on the expansion stroke at engine stop (the second cylinder 12B as illustrated in FIGS. 3A-3D) based on the stop positions of the pistons 13 determined in the aforementioned stop position detecting subroutine (FIGS. 8, 9A and 9B). More specifically, the ECU 2 calculates current volumes of the combustion chambers 14 in the compression stroke cylinder 12 and the expansion stroke cylinder 12 based on the stop positions of the pistons 13 and determines the quantities of air in the compression stroke cylinder 12 and the expansion stroke cylinder 12 in step SB2 on the assumption that the individual cylinders 12A-12D of the engine 1 are filled almost entirely with fresh air which is approximately at atmospheric pressure at engine stop as stated earlier.

[0078] In succeeding step SB3, the ECU 2 calculates the quantity of fuel to be injected for producing a specific air-fuel ratio (first-time air-fuel ratio for the compression stroke cylinder 12) with the quantity of air in the compression stroke cylinder 12 calculated in step SB2 above and causes the fuel injector 16 of the cylinder 12 which is on the compression stroke at engine stop to inject the fuel. This air-fuel ratio is determined from a preprogrammed map which defines desired air-fuel ratios in relation to the piston stop position in the compression stroke cylinder 12 at engine stop, for instance. Specifically, the air-fuel ratio in the compression stroke cylinder 12 is set to a value smaller than the stoichiometric air-fuel ratio (preferably within a range of approximately 11 to 14, and more preferably, at approximately 13). Here, it is necessary to control the quantity of fuel to be injected in such a way that the air-fuel ratio produced in the compression stroke cylinder 12 is smaller than a maximum combustible level on a rich mixture side (e.g., at approximately 7) for preventing misfire.

[0079] In succeeding step SB4, the ECU 2 causes the spark plug 15 of the compression stroke cylinder 12 to ignite a mixture produced therein after a lapse of a particular time period which is preset in consideration of fuel evaporation time required after fuel injection into the cylinder 12. Then, in step SB5, the ECU 2 judges whether the piston 13 in the compression stroke cylinder 12 has moved based on whether the rising and falling edges of the crank angle signals CA1, CA2 output from the crank angle sensors 30, 31 have been detected within a specific time from ignition in step SB4. (Refer to the aforementioned stop position detecting subroutine of FIGS. 8, 9A and 9B for details of how the ECU 2 detects the rising and falling edges of the crank angle signals CA1, CA2.) If the piston 13 in the compression stroke cylinder 12 has not moved (No in step SB5) due to misfire, for instance, the ECU 2 proceeds to step SB6, in which the ECU 2 causes the spark plug 15 of the compression stroke cylinder 12 to reignite the mixture.

[0080] If the edges of the crank angle signals CA1, CA2 are detected (Yes in step SB5) indicating that the piston 13 in the compression stroke cylinder 12 has moved, or the engine 1 has begun to run in the reverse direction, the ECU 2 proceeds to step SB7, in which the ECU 2 causes the fuel injector 16 of the cylinder 12 which is on the expansion stroke at engine stop to inject the fuel such that a specific air-fuel ratio for the expansion stroke cylinder 12 is produced with the quantity of air in the expansion stroke cylinder 12 calculated in step SB2 above. Here again, the air-fuel ratio for the expansion stroke cylinder 12 is determined from a preprogrammed map which defines desired air-fuel ratios in relation to the piston stop position in the expansion stroke cylinder 12 at engine stop, for instance. Specifically, the air-fuel ratio for the expansion stroke cylinder 12 is set to a value equal to or slightly smaller than the stoichiometric air-fuel ratio (preferably at approximately 13).

[0081] In succeeding step SB8, the ECU 2 causes the spark plug 15 of the expansion stroke cylinder 12 to ignite and combust a mixture produced therein after a lapse of a particular time period (ignition delay time) from a point of detecting reversing action of the engine 1. This ignition delay time corresponds to a period of time during which the mixture in the expansion stroke cylinder 12 is sufficiently compressed as a result of an upward movement of the piston 13 in the expansion stroke cylinder 12 and the piston 13 almost stops due to a resulting compressive reaction force. Specifically, the ignition delay time is determined from a preprogrammed map which defines appropriate ignition delay times in relation to the piston stop position at engine stop. As the mixture sufficiently compressed in the expansion stroke cylinder 12 ignited and burnt in this way, the engine 1 begins to run in the forward direction with a sufficiently large starting torque.

[0082] In succeeding step SB9, the ECU 2 causes the fuel injector 16 of the cylinder 12 which was on the compression stroke at engine stop to inject the fuel a second time (additional fuel injection) when the piston 13 in the same cylinder 12 reaches the first TDC as a result of the forward running motion of the engine 1 with timing determined in consideration of the fuel evaporation time. As the fuel injected into the compression stroke cylinder 12 evaporates, the temperature and pressure in the compression stroke cylinder 12 decrease since the evaporated fuel deprives surrounding gas of heat due to absorption of latent heat by evaporation of the injected fuel. Since the compressive reaction force exerted by the compression stroke cylinder 12 can be decreased even when the same cylinder 12 is fully filled with burned gas, it is possible to allow the piston 13 in the compression stroke cylinder 12 to go beyond TDC in a reliable fashion. Hence, the engine 1 which began to run in the forward direction as a result of combustion in the expansion stroke cylinder 12 in step SB8 is caused to continue running in the forward direction so that the piston 13 in the cylinder 12 which was on the compression stroke at engine stop goes beyond TDC and the individual cylinders 12A-12D proceed to the succeeding strokes.

[0083] Assuming that the compression stroke of the cylinder 12 which was on the compression stroke at engine stop while the engine 1 runs in the forward direction is divided into three approximately equal parts which are referred to as an early part, a middle part and a latter part, the timing of the additional fuel injection into the cylinder 12 which was on the compression stroke at engine stop should preferably be the middle part or latter of the compression stroke. This is because there is a relationship as shown in an example of FIG. 15 between the fuel injection timing and the aforementioned effect of reducing the in-cylinder pressure. More specifically, if the fuel is injected into the cylinder 12 in the early part of the compression stroke, the temperature of gas in the compression stroke cylinder 12 decreases too early. It follows that the amount of heat the fuel injected into the compression stroke cylinder 12 receives from the cylinder wall increases and in-cylinder gas density increases due to evaporation of the injected fuel, whereby the aforementioned effect of reducing the in-cylinder temperature and pressure is diminished, resulting in an increase in the amount of work required for engine restart.

[0084] If the timing of the additional fuel injection into the cylinder 12 which was on the compression stroke at engine stop is retarded too much, however, evaporation of the injected fuel would be delayed and it becomes impossible to obtain a sufficient cooling effect. Therefore, the additional fuel injection into the compression stroke cylinder 12 should preferably be made during the middle part to an early half of the latter part of the compression stroke.

[0085] In step SB10 of FIG. 11 succeeding to step SB9, the ECU 2 calculates the quantity of air filled in the cylinder 12 (the third cylinder 12C as illustrated in FIGS. 3A-3D) which is on the intake stroke at engine stop after the engine 1 has begun to run in the forward direction. The piston 13 in the cylinder which was on the intake stroke at engine stop goes beyond TDC after the piston 13 in the cylinder which was on the compression stroke at engine stop has first gone beyond TDC upon engine restart. Since the cylinder 12 which was on the compression stroke at engine stop is fully charged with relatively high-temperature air approximately at atmospheric pressure, autoignition is likely to occur very easily in this cylinder 12 on the subsequent compression stroke as previously mentioned.

[0086] More specifically, the ECU 2 estimates the density of air filled in the cylinder 12 which was on the intake stroke at engine stop based on the atmospheric pressure and in-cylinder temperature estimated from the engine cooling water temperature, duration of engine stop and intake air temperature, for instance, and calculates the quantity of air filled in the intake stroke cylinder 12 based on the estimated in-cylinder air density in step SB10. Then in step SB11, the ECU 2 calculates an air-fuel ratio correction value used for producing a richer mixture to prevent autoignition (spontaneous combustion) based mainly on the estimated temperature in the intake stroke cylinder 12.

[0087] In succeeding step SB12, the ECU 2 calculates an appropriate quantity of fuel to be injected into the intake stroke cylinder 12 based on an air-fuel ratio corrected by the aforementioned correction value and the quantity of air filled in the intake stroke cylinder 12 calculated in step SB10.

[0088] After the cylinder 12 which was on the intake stroke at engine stop has transferred to the compression stroke, the ECU 2 causes the fuel injector 16 of the same cylinder 12 to inject the fuel in the middle part or latter of the compression stroke of the cylinder 12 in step SB13. As the fuel injected into the cylinder 12 which was on the intake stroke at engine stop evaporates, the temperature and pressure in this cylinder 12 decrease due to absorption of latent heat by evaporation of the injected fuel in the same way as the cylinder 12 which was on the compression stroke at engine stop. Therefore, it is possible to prevent autoignition by suppressing an increase in temperature and pressure due to compression in the cylinder 12 which was initially on the intake stroke even if the same cylinder 12 is fully filled with relatively high-temperature air as mentioned above. In addition, a decrease in engine speed potentially occurring when the piston 13 in the cylinder 12 which was initially on the intake stroke goes beyond TDC is lessened as the compressive reaction force exerted by the same cylinder 12 is decreased.

[0089] The ECU 2 controls the quantity of fuel to be injected into the cylinder 12 which was on the intake stroke at engine stop when the same cylinder 12 has transferred to the compression stroke in such a way that an average air-fuel ratio produced in the cylinder 12 would fall within a specific range including and close to the stoichiometric air-fuel ratio (e.g., approximately 12 to 16), and more preferably, the average air-fuel ratio would become slightly lower than the stoichiometric air-fuel ratio (e.g., at approximately 13). The ECU 2 controls the engine 1 in this way taking into consideration a relationship between the air-fuel ratio and the aforementioned effect of reducing the in-cylinder pressure by evaporation of the injected fuel as shown in an example of FIG. 16. Specifically, if the air-fuel ratio is higher than approximately 16, the quantity of injected fuel is too small so that a decrease in in-cylinder temperature and pressure due to absorption of latent heat by evaporation of the injected fuel is insufficient and the amount of work required for engine restart increases. If the air-fuel ratio is lower than approximately 12, on the other hand, the quantity of injected fuel is too large so that the amount of work required for engine restart increases due to an increase in the density of mixture. The timing of fuel injection into the cylinder 12 which was on the intake stroke at engine stop and is currently on the compression stroke should preferably be the middle part or latter of the compression stroke for the same reason as explained earlier with respect to the cylinder 12 which was on the compression stroke at engine stop.

[0090] In step SB14, the ECU 2 causes the spark plug 15 of the cylinder 12 which was on the intake stroke at engine stop (now on the compression stroke) to ignite a mixture produced therein after the piston 13 in the same cylinder 12 has gone beyond the compression stroke TDC. While the mixture is typically ignited before the compression stroke TDC (e.g., 10° before TDC, or BTDC) in an ordinary engine started by a starter motor, ignition timing is retarded to a point beyond the compression stroke TDC, or on the expansion stroke, in the engine 1 of this embodiment. This is because a torque acting on the crankshaft 3 in the reverse direction via the piston 13 could impede successful engine restart if the mixture is ignited before the compression stroke TDC in the engine 1 which is started without using any starter motor.

[0091] In succeeding step SB15, the ECU 2 judges whether the intake air pressure (intake air pipe negative pressure) in a portion of the intake passage 21 downstream of the throttle valve 23 is higher than under normal engine idle conditions. If the intake air pressure in the downstream portion of the intake passage 21 is judged to be higher than under normal engine idle conditions (Yes in step SB15), the ECU 2 proceeds to step SB16, in which the ECU 2 also retards the ignition timing for the cylinder 12 (the fourth cylinder 12D as illustrated in FIGS. 3A-3D) of which piston 13 reaches TDC next to the piston 13 in the cylinder 12 which was initially on the intake stroke to a point beyond TDC, and returns to step SB15. The ECU 2 successively retards the ignition timing of the individual cylinders 12A-12D to points beyond TDC until the intake air pressure becomes equal to that under normal engine idle conditions. When the intake air pressure has become equal to or lower than that at idle (No in step SB15), the ECU 2 proceeds to step SB17 and returns to normal engine control operation.

[0092] Air which is approximately at atmospheric pressure is introduced through the intake passage 21 into the cylinder 12D which was on the exhaust stroke at engine stop when the same cylinder 12D has transferred to the intake stroke and into the cylinders 12B, 12A and 12C when these cylinders 12B, 12A, 12C have successively transferred to the intake stroke. Taking into consideration the fact that the individual cylinders 12A-12D are charged with relatively high-temperature air on the intake stroke, the ECU 2 retards the ignition timing of the individual cylinders 12A-12D during a period when the intake air pipe negative pressure is relatively small (or when the intake air pressure is relatively high) to decrease an increase in combustion torque and thereby prevent a rapid increase in engine speed.

[0093] In the aforementioned operation flow shown in FIG. 10, step SB2 constitutes an air quantity detecting section 2c for calculating the quantity of air in the cylinder 12 which is on the compression stroke at engine stop, whereas steps SB3, SB7 and SB9 together constitute a fuel injection controlling section 2d for controlling activation of the fuel injectors 16 of which nozzle ends are located on the interior of the individual cylinders 12A-12D.

[0094] The aforementioned fuel injection controlling section 2d controls the quantity of fuel to be first injected into the cylinder 12 which is on the compression stroke at engine stop in such a manner that an average air-fuel ratio produced therein becomes low to produce a rich mixture. Further, the fuel injection controlling section 2d controls the engine 1 to inject an additional quantity of fuel into the cylinder 12 which was initially on the compression stroke with specific timing in the middle part or latter of the compression stroke when the gas in the same cylinder 12 is compressed as a result of the forward running motion of the engine 1 and to inject the fuel into the cylinder 12 which was on the intake stroke at engine stop when this cylinder 12 has transferred to the compression stroke.

[0095] Step SB8 of FIG. 10 and steps SB14 and SB16 of FIG. 11 together constitute an ignition controlling section 2e for controlling ignition timing for the individual cylinders 12A-12D during engine restart. The ignition controlling section 2e retards a first ignition point of at least the cylinder 12 which was on the intake stroke or on the exhaust stroke at engine stop.

THE WORKING AND EFFECTS OF THE EMBODIMENT

[0096] When automatically stopping the engine 1 at idle, the engine system E (engine starting system) of the present embodiment drives the burned gas out of the cylinders 12A-12D and causes the piston 13 in the cylinder 12 which will be on the expansion stroke at engine stop to stop at a position within the specified range R suited for engine restart more or less closer to BDC than the middle of the expansion stroke by performing the aforementioned engine stop control operation (illustrated in FIGS. 4, 5 and 6A-6E). Also, the engine system E of the embodiment can supply a sufficient quantity of fresh air to the exhaust-converting catalyst 29 during the engine stopping period so that the catalyst 29 adsorbs a sufficiently large quantity of oxygen.

[0097] When restarting the engine 1, on the other hand, the engine system E starts up the engine 1 in response to an engine restart request by performing the aforementioned engine restart control operation (illustrated in FIGS. 10, 11 and 12) without using any starter motor. The engine restart control operation is now described time-sequentially referring to FIGS. 12 and 13A-13F. Specifically, when an engine restart request is issued under idle engine stop conditions (time 0 in FIGS. 13A-13F), the ECU 2 actuates the fuel injector 16 of the cylinder 12 (the first cylinder 12A in the illustrated example) which is on the compression stroke at engine stop to inject the fuel into the first cylinder 12A at a point shown by the symbol a1 in FIG. 13B, whereby a rich mixture having a low air-fuel ratio is produced in the first cylinder 12A. The ECU 2 causes the spark plug 15 of the same cylinder 12A to ignite this rich mixture at a point shown by the symbol a2 in FIG. 13B, whereby a negative torque (reversing torque) is generated as shown by the symbol T1 in FIG. 13F and, as a consequence, the value of the engine speed momentarily becomes negative as shown in FIG. 13A.

[0098] Upon detecting reverse running motion of the engine 1 based on the crank angle signals CA1, CA2 output from the two crank angle sensors 30, 31, the ECU 2 actuates the fuel injector 16 of the cylinder 12 (the second cylinder 12B in the illustrated example) which is on the expansion stroke at engine stop to inject the fuel at a point shown by the symbol a3 in FIG. 13C and a mixture produced in the same cylinder 12B is compressed as the piston 13 ascends as a result of the reverse running motion of the engine 1. Since the reversing torque T1 is large enough, the piston 13 in the cylinder 12B which was on the expansion stroke at engine stop ascends up to the vicinity of TDC and the mixture produced in the cylinder 12B is sufficiently compressed, resulting in an increase in the temperature and pressure of the mixture. When the turning direction of the engine 1 changes from the reverse direction to the forward direction due to the compressive reaction force, that is, immediately after the engine speed has changed from a negative value to zero, the mixture produced in the cylinder 12B is ignited at a point shown by the symbol a4 in FIG. 13C. As a result, the starting torque sharply rises as shown by the symbol T2 in FIG. 13F and the engine 1 begins to run in the forward direction with the engine speed increasing as shown in FIG. 13A.

[0099] As burned gas in the cylinder 12A which was on the compression stroke at engine stop is compressed as a result of the forward running motion of the engine 1, the ECU 2 actuates the fuel injector 16 of the same cylinder 12A to reinject the fuel (additional fuel injection) into the cylinder 12A in the middle part or latter of the compression stroke thereof at a point shown by the symbol a5 in FIG. 13B. Thus, the interior of the first cylinder 12A is cooled down by absorption of latent heat by evaporation of the injected fuel, so that the increase in the temperature and pressure in the first cylinder 12A is greatly lessened compared to a case in which the aforementioned additional fuel injection is not done as shown by a broken line in FIG. 13B. As a consequence, the piston 13 in the first cylinder 12A can go beyond the first TDC after engine restart in a reliable fashion and with a minimum decrease in the engine speed. Moreover, since the fuel for cooling the interior of the cylinder 12A which was on the compression stroke at engine stop is injected into the burned gas at a low air-fuel ratio, the additionally injected fuel does not burn but reacts with oxygen adsorbed by the catalyst 29 disposed in the exhaust passage 22. Therefore, the fuel injected at the point a5 is made nontoxic and produces no problem.

[0100] After the piston 13 in the cylinder 12A which was on the compression stroke at engine stop has gone beyond the first TDC upon engine restart, the ECU 2 causes the fuel injector 16 of the cylinder 12 (the third cylinder 12C in the illustrated example) which was on the intake stroke at engine stop and has now transferred to the compression stroke to inject the fuel into the third cylinder 12C in the middle part or latter of the compression stroke at a point shown by the symbol a6 in FIG. 13D and, as a consequence, the interior of the third cylinder 12C is cooled down by absorption of latent heat by evaporation of the injected fuel. For this reason, the increase in the temperature and pressure due to compression in the third cylinder 12C is lessened, the occurrence of autoignition is avoided and the compressive reaction force exerted by the third cylinder 12C is reduced. In addition, the ignition timing for the third cylinder 12C is retarded to a point beyond TDC. Since this avoids an increase in the in-cylinder pressure potentially caused by ignition and combustion made before TDC, the engine 1 can also go beyond the second TDC (of the piston 13 in the third cylinder 12C) after engine restart in a reliable fashion.

[0101] When the piston 13 in the cylinder 12C which was on the intake stroke at engine stop has gone beyond the second TDC after engine restart, the ECU 2 causes the spark plug 15 of the cylinder 12C which has now transferred to the expansion stroke to ignite and combust a mixture produced in the same cylinder 12C at a point shown by the symbol a7 in FIG. 13D. As a consequence, an additional forward-acting torque is given to the engine 1 and the starting torque rises as shown by the symbol T3 in FIG. 13F, whereby the engine speed increases up to about the idle speed (650 rpm in the illustrated example) as shown in FIG. 13A. At this point, the aforementioned engine restart operation may be regarded as having been almost finished successfully. Since initial combustion in the cylinder 12C which was initially on the intake stroke is made on the expansion stroke at engine restart, the starting torque does not increase at an extraordinary rate so that the engine speed would not increase so rapidly beyond the idle speed.

[0102] Fuel injection into the fourth cylinder 12D which was on the exhaust stroke at engine stop is made at a point shown by the symbol a8 in FIG. 13E when the same cylinder 12D has transferred to the intake stroke as a result of the forward running motion of the engine 1. When the fuel injected into the fourth cylinder 12D has been sufficiently mixed with air through a process of evaporation and atomization and the piston 13 in the fourth cylinder 12D has gone beyond subsequent TDC (third TDC), a mixture produced in the same cylinder 12D is ignited and burnt at a point shown by the symbol a9 in FIG. 13E. As the ignition timing is retarded to the point beyond TDC and initial combustion in the cylinder 12D which was initially on the exhaust stroke is made on the expansion stroke at engine restart, the starting torque increases at a moderate rate as shown in FIG. 13F as is the case with the combustion in the cylinder 12C which was on the intake stroke at engine stop. As a result, the engine speed gradually increases as shown in FIG. 13A.

[0103] If the engine 1 of the aforementioned embodiment is provided with variable valve actuating mechanisms, the closing timing of the intake valves 19 may be retarded such that the intake valves 19 are closed within a range of 70° to 90° after the intake stroke BDC in terms of crank angle at least when the engine 1 is stopped at idle. If the engine 1 is so structured, the quantity of air introduced into the cylinder 12 which was on the intake stroke at engine stop for the first time upon engine restart becomes relatively small, and an increase in the temperature and pressure when the same cylinder 12 transfers to the compression stroke can be lessened. This is advantageous for preventing autoignition and for reducing the compressive reaction force.

[0104] Although the fuel is injected into the cylinder 12 which was on the exhaust stroke at engine stop when the same cylinder 12 has transferred to the intake stroke according to the engine restart control operation of the foregoing embodiment, the invention is not limited to this arrangement. For example, the engine 1 may be provided with an engine temperature sensor for judging whether or not the engine temperature is equal to or higher than a specific temperature or an engine stop time counter for judging whether or not the time elapsed after engine stop is equal to or shorter than a specific time period. Provided with one of such means, the ECU 2 can determine whether or not the engine temperature is equal to or higher than the specific temperature, or the time elapsed after engine stop is equal to or shorter than the specific time period based on the engine cooling water temperature, intake air temperature or the measured time elapsed after engine stop. If the temperature of air introduced into the cylinder 12 which was on the exhaust stroke at engine stop is assumed to be relatively high based on the judgment result of the engine temperature sensor or the engine stop time counter, the fuel may be injected into this cylinder 12 on the compression stroke (not on the intake stroke) in the same way as for the cylinder 12 which was on the intake stroke at engine stop.

[0105] If the temperature of air introduced into the cylinder 12 which was on the exhaust stroke at engine stop is high, autoignition could potentially occur in this cylinder 12 as in the cylinder 12 which was on the intake stroke at engine stop. Therefore, it is preferable to inject the fuel on the compression stroke so that the interior of the cylinder 12 is cooled by absorption of latent heat by evaporation of the injected fuel for preventing autoignition. If the temperature of air introduced into the cylinder 12 which was on the exhaust stroke at engine stop is not so high, on the other hand, it is preferable to inject the fuel earlier (i.e., on the intake stroke) to ensure a longer period of time to allow evaporation and atomization of the fuel and sufficient mixing with air.

[0106] Although the engine 1 of the foregoing embodiment employs the single throttle valve 23 located upstream of the surge tank 21b as an intake air quantity regulator for regulating the quantity of air introduced into the individual cylinders 12A-12D, the invention is not limited to this arrangement. For example, the engine 1 may be provided with the aforementioned variable valve actuating mechanisms of the prior art as an intake air quantity regulator for varying the lift of the intake valve 19 of each of the four cylinders 12A-12D. Alternatively, the engine 1 may employ multiple throttle valves of which valve bodies are disposed in the branched intake channels 21a for the individual cylinders 12A-12D instead of the throttle valve 23.

[0107] If the multiple throttle valves disposed in downstream portions of the intake passage 21, or in the branched intake channels 21a, are employed, it is possible to restrict the quantity of air introduced into the individual cylinders 12A-12D by reducing valve openings with good response characteristics. Therefore, unlike the single throttle valve 23 of the foregoing embodiment located upstream of the surge tank 21b, the openings of the multiple throttle valves can not only be increased to reduce the rate of engine speed decrease but also be decreased to increase the rate of engine speed decrease.

[0108] In summary, the present invention, as described in the above, is intended to provide a solution to the aforementioned problem of the prior art. Accordingly, it is a general object of the invention to provide an engine starting system for restarting an engine from idle stop.

It is a more specific object of the invention to provide an engine starting system capable of restarting an engine in a reliable fashion, in which initial combustion is produced in a cylinder which is on a compression stroke at engine stop to reverse the engine at first and subsequent combustion is produced in a cylinder which is on an expansion stroke at engine stop to start the engine in a forward running direction by using a novel technique for controlling fuel injection into the compression stroke cylinder to obtain as high a starting torque as possible from combustion in the expansion stroke cylinder and to decrease a reaction force exerted by the compression stroke cylinder when the engine begins to run in the forward direction, and in which a novel technique is used in controlling fuel injection into a cylinder which is on an intake stroke at engine stop and in controlling ignition in this cylinder to prevent autoignition during a succeeding compression stroke of the same cylinder and to decrease a compressive reaction force exerted thereby.

[0109] To achieve the aforementioned object of the invention, initial combustion in a cylinder which is on a compression stroke at engine stop is made with a rich mixture having a low air-fuel ratio to produce a torque acting in an engine-reversing direction, whereby gas in a cylinder which is on an expansion stroke at engine stop is sufficiently compressed and a starting torque produced by combustion in the expansion stroke cylinder is significantly increased. Then, an additional quantity of fuel is injected into the cylinder which was initially on the compression stroke when gas in this cylinder has been compressed as a result of forward running motion of the engine, so that a compressive reaction force exerted by the same cylinder decreases due to a cooling effect produced by absorption of latent heat by evaporation of the injected fuel.

[0110] Specifically, an engine starting system for starting a multicylinder engine includes a fuel injection controlling section for controlling actuation of fuel injectors of which nozzle ends are located on the interior of individual cylinders, an ignition controlling section for controlling ignition timing for the individual cylinders, and an air quantity detecting section for detecting the quantity of air in the cylinder which is on a compression stroke at engine stop. In this engine starting system, the fuel injection controlling section causes the fuel injector of the cylinder which is on the compression stroke at engine stop to inject fuel and a mixture produced in the compression stroke cylinder is ignited and burnt so that the engine once turns in a reverse direction from idle stop, and the fuel injection controlling section causes the fuel injector of the cylinder which is on an expansion stroke at engine stop to inject the fuel when gas in the expansion stroke cylinder has been compressed as a result of reverse running motion of the engine and a mixture produced in the expansion stroke cylinder is ignited and burnt so that a torque acting in a forward direction is generated to restart the engine. In addition, the fuel injection controlling section controls the quantity of fuel injected into the compression stroke cylinder such that an average air-fuel ratio produced therein becomes lower than the stoichiometric air-fuel ratio based on a value of the quantity of air detected by the air quantity detecting section, and the fuel injection controlling section causes the fuel injector of the cylinder which was on the compression stroke at engine stop to inject an additional quantity of fuel when gas in the same cylinder has been compressed as a result of forward running motion of the engine.

[0111] In the engine starting system thus configured, the fuel injection controlling section controls the fuel injector of the cylinder which is on the compression stroke at engine stop to inject the fuel into the same cylinder and the mixture produced therein is combusted so that the engine turns in the reverse direction. Since the average air-fuel ratio in the compression stroke cylinder is low at this point, initial combustion in this cylinder produces a sufficiently large a torque acting in the engine-reversing direction and, therefore, gas in the expansion stroke cylinder can be sufficiently compressed.

[0112] On the other hand, the fuel injection controlling section causes the fuel injector of the cylinder which is on the expansion stroke at engine stop to inject the fuel and the mixture produced in this cylinder is combusted under conditions where the mixture has been compressed and in-cylinder temperature and pressure have increased as a result of the reverse running motion of the engine. Therefore, the torque generated by combustion in the expansion stroke cylinder and the distance traveled by a piston in the same cylinder increases, so that the combustion in the expansion stroke cylinder generates a significantly increased starting torque acting in the forward running direction of the engine.

[0113] In addition, as the fuel injection controlling section controls the fuel injector of the cylinder which was on the compression stroke at engine stop to inject an additional quantity of fuel when gas in the same cylinder has been compressed as a result of the forward running motion of the engine, the interior of this cylinder is cooled down by absorption of latent heat by evaporation of the additionally injected fuel. As a result, an increase in in-cylinder temperature and pressure is greatly lessened even if the cylinder which was on the compression stroke at engine stop is filled with burned gas produced by the initial combustion therein for reversing the engine. This enables a piston in the same cylinder to go beyond the first compression stroke top dead center (TDC) after engine restart, causing the engine to continue running in the forward direction. Thus, the engine starting system of the invention can restart the engine in a reliable fashion.

[0114] In one feature of the invention, the fuel injection controlling section causes the fuel injector of the cylinder which was on the compression stroke at engine stop to inject the additional quantity of fuel at a point in a period including a middle part and a latter part of the compression stroke.

[0115] Here, it is assumed that the compression stroke of the cylinder which was on the compression stroke at engine stop while the engine runs in the forward direction is divided into three approximately equal parts of crank angle ranges, that is, the early part, the middle part and the latter part. A reason why the fuel is additionally injected into the compression stroke cylinder at a point in the period including the middle part and the latter part of the compression stroke is as follows. If the fuel is injected into the compression stroke cylinder in the early part of the compression stroke, the temperature of gas in this cylinder decreases too early. It follows that the amount of heat the fuel injected into the compression stroke cylinder receives from the cylinder wall increases and in-cylinder gas density increases due to evaporation of the injected fuel, whereby the aforementioned effect of reducing the in-cylinder temperature and pressure is diminished.

[0116] If the timing of the additional fuel injection into the cylinder which was on the compression stroke at engine stop is retarded too much, however, evaporation of the injected fuel would be delayed and it becomes impossible to obtain a sufficient cooling effect. Therefore, the additional fuel injection into the compression stroke cylinder should preferably be made during the middle part to an early half of the latter part of the compression stroke.

[0117] In another feature of the invention, an exhaust-converting catalyst having an oxygen-adsorbing capability is provided in an exhaust passage of the engine, and the engine starting system further includes an engine stopping section for stopping the engine by interrupting fuel supply to the individual cylinders while the engine is running, an intake air quantity regulator for regulating the quantities of air introduced into the individual cylinders, and an intake air quantity regulator controlling section for controlling the intake air quantity regulator in such a manner that the quantities of air introduced into the individual cylinders increase for a specific period of time in an engine stopping period which begins at a point of fuel supply interruption and ends at a point of engine stop.

[0118] In the engine starting system thus configured, the quantities of air introduced into the individual cylinders increase in the aforementioned specific period of time in the engine stopping period after fuel supply interruption during which the engine turns several times by inertia before stopping. Since a large quantity of fresh air is supplied also to the exhaust-converting catalyst as a result, the quantity of oxygen adsorbed by the catalyst sufficiently increases and unburned fuel discharged from the cylinder which was on the compression stroke at engine stop during engine restart reacts with oxygen adsorbed in the catalyst, so that exhaust gas emissions are made nontoxic in a reliable fashion.

[0119] In another feature of the invention, the fuel injection controlling section causes the fuel injector of the cylinder which was on an intake stroke at engine stop to inject the fuel when the same cylinder has transferred to the compression stroke as a result of the forward running motion of the engine.

[0120] In the engine starting system thus configured, air is once driven out of the cylinder which was on the intake stroke at engine stop into an intake port via an intake valve as a result of the reverse running motion of the engine and reintroduced into the same cylinder when the engine runs in the forward direction subsequently. When the fuel injection controlling section causes the fuel injector of the cylinder which was on an intake stroke at engine stop to inject the fuel after the same cylinder has transferred to the compression stroke, the interior of this cylinder is cooled down by absorption of latent heat by evaporation of the injected fuel. As a result, an increase in in-cylinder temperature and pressure is lessened and this serves to prevent the occurrence of autoignition in the cylinder which was on the intake stroke at engine stop and make it easier for a piston in the same cylinder to go beyond the compression stroke TDC due to a reduction in compressive reaction force.

[0121] In another feature of the invention, the fuel injection controlling section causes the fuel injector of the cylinder which was on the intake stroke at engine stop to inject the fuel at a point in a period including a middle part and a latter part of the compression stroke when the same cylinder has transferred to the compression stroke.

[0122] Here again, it is assumed that a period from the intake stroke bottom dead center (BDC) to the compression stroke TDC of the cylinder which was on the intake stroke at engine stop is divided into three approximately equal parts of crank angle ranges, that is, the early part, the middle part and the latter part. A reason why the fuel is injected into the cylinder which was on the intake stroke at engine stop at a point in the period including the middle part and the latter part of the compression stroke is as follows. If the fuel is injected into the cylinder which was on the intake stroke at engine stop in the early part of the compression stroke, the temperature of gas in this cylinder decreases too early. It follows that the amount of heat the fuel injected into the cylinder receives from the cylinder wall increases and in-cylinder gas density increases due to evaporation of the injected fuel, whereby the aforementioned effect of reducing the in-cylinder temperature and pressure is diminished.

[0123] In another feature of the invention, the ignition controlling section retards the ignition timing for the cylinder which was on the intake stroke at engine stop to a point beyond the compression stroke top dead center.

[0124] In the engine starting system thus configured, the ignition controlling section causes a spark plug of the cylinder which was on the intake stroke at engine stop to ignite and combust a mixture produced in the cylinder which was on the intake stroke at engine stop after the piston in the same cylinder has gone beyond the compression stroke TDC and the cylinder has transferred to the exhaust stroke, so that an additional forward-acting torque is given to the engine. Since the ignition timing is retarded to the point beyond the compression stroke TDC, it is possible to avoid an increase in the pressure in the cylinder which was on the intake stroke at engine stop potentially caused by ignition and combustion made before TDC. This also enables the piston in the cylinder which was on the intake stroke at engine stop to go beyond the compression stroke TDC.

[0125] In other words, it is possible to securely prevent autoignition in the cylinder which was on the intake stroke at engine stop when the same cylinder has transferred to the compression stroke and to restart the engine in a more reliable fashion by retarding the ignition timing to the point beyond the compression stroke TDC.

[0126] In another feature of the invention, the engine starting system further includes an engine stopping section for stopping the engine by interrupting fuel supply to the individual cylinders while the engine is running, wherein closing timing of an intake valve of each of the cylinders is set at a point retarded by a specific amount at least when the engine is stopped by the engine stopping section.

[0127] In the engine starting system thus configured, the closing timing of the intake valves of the individual cylinders are retarded such that the intake valves are closed within a range of 70° to 90° after the intake stroke BDC in terms of crank angle, for instance. To make this possible, there may be provided valve actuating mechanisms or variable valve actuating mechanisms of the prior art to control the closing timing of the intake valves.

[0128] If the closing timing of the intake valves are retarded as stated above, the quantity of air introduced into the cylinder which was on the intake stroke at engine stop for the first time upon engine restart becomes relatively small, and an increase in the in-cylinder temperature and pressure when the same cylinder has transferred to the compression stroke can be lessened. This is advantageous for preventing autoignition and for reducing the compressive reaction force.

[0129] In another feature of the invention, the fuel injection controlling section causes the fuel injector of the cylinder which was on the intake stroke at engine stop to inject the fuel for initial combustion in the same cylinder on the compression stroke and the fuel for second and subsequent combustions in the same cylinder on the intake stroke for preventing the occurrence of autoignition in the cylinder which was on the intake stroke at engine stop when the same cylinder is on the initial compression stroke.

[0130] In the engine starting system thus configured, engine speed increases by the time when the fuel for the second combustion in the cylinder which was on the intake stroke at engine stop is injected so that an interval between fuel injection and ignition in the same cylinder tends to become short. However, since fuel injection timing for the second and subsequent combustions in the cylinder which was on the intake stroke at engine stop is retarded from a point in the compression stroke to a point in the intake stroke as stated above to lengthen the interval between fuel injection and ignition, a longer period of time is ensured to allow evaporation and atomization of the fuel and sufficient mixing with air, making it possible to prevent deterioration of combustibility and to achieve engine restart in a more reliable fashion.

[0131] Even when the engine is restarted by use of a starter motor, it is preferable that the fuel injection controlling section cause the fuel injector of each of the cylinders to inject the fuel on the intake stroke to ensure sufficient time to allow evaporation and atomization of the fuel which is injected directly into the individual cylinders as well as sufficient mixing of the fuel with air.

[0132] In still another feature of the invention, the engine starting system further includes an engine temperature sensor for judging engine temperature conditions, wherein the fuel injection controlling section causes the fuel injector of the cylinder which was on an exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the compression stroke via an intake stroke as a result of the forward running motion of the engine if the engine temperature is judged to be equal to or higher than a specific temperature by the engine temperature sensor, whereas the fuel injection controlling section causes the fuel injector of the cylinder which was on the exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the intake stroke if the engine temperature is judged to be lower than the specific temperature by the engine temperature sensor.

[0133] In the engine starting system thus configured, the fuel is injected on the compression stroke into the cylinder which was on the exhaust stroke at engine stop as well if the engine temperature is judged to be equal to or higher than the specific temperature by the engine temperature sensor, because autoignition could also occur in the cylinder which was on the exhaust stroke at engine stop as in the cylinder which was on the intake stroke at engine stop when the engine temperature is relatively high. When the engine temperature is not so high, on the other hand, the fuel is injected on the intake stroke into the cylinder which was on the exhaust stroke at engine stop to ensure a longer period of time to allow evaporation and atomization of the injected fuel and sufficient mixing of the fuel with air and thereby improve combustibility.

[0134] While the engine temperature conditions can be judged based on a signal fed from an engine water temperature sensor or an intake air temperature sensor, for instance, the engine temperature conditions are greatly affected by the time elapsed from a point of idle engine stop to a point of engine restart. Taking this into consideration, the engine starting system may include an engine stop time counter for measuring time elapsed from a point of engine stop to a point of engine restart instead of the aforementioned engine temperature sensor, so that the fuel injection controlling section can vary the timing of injecting the fuel into the cylinder which was on an exhaust stroke at engine stop.

[0135] More specifically, the fuel injection controlling section causes the fuel injector of the cylinder which was on an exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the compression stroke if the elapsed time measured by the engine stop time counter is equal to or shorter than a specific time period, whereas the fuel injection controlling section causes the fuel injector of the cylinder which was on the exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the intake stroke if the elapsed time measured by the engine stop time counter is longer than the specific time period.

[0136] In yet another feature of the invention, the ignition controlling section should preferably retard the ignition timing for the cylinder which was on the exhaust stroke at engine stop to a point beyond the compression stroke top dead center regardless of the result of judgment by the aforementioned engine temperature sensor or the elapsed time measured by the engine stop time counter.

[0137] If the engine is so controlled, it is possible to prevent an increase in the pressure in the cylinder which was on the exhaust stroke at engine stop potentially caused by ignition and combustion made before the compression stroke TDC and lessen a reduction in engine speed at the compression stroke TDC. Since the combustion in the cylinder which was on the exhaust stroke at engine stop begins on the expansion stroke, a torque generated by this combustion is relatively small and, therefore, it is possible to smoothly increase the engine speed while preventing a rapid increase in engine speed.

[0138] Although the present invention has been fully described by way of example with reference to the accompanying drawings, it is to be understood that various changes and modifications will be apparent to those skilled in the art. Therefore, unless otherwise such changes and modifications depart from the scope of the present invention hereinafter defined, they should be construed as being included therein.

Claims

1. An engine starting system (E) for starting a multi-cylinder engine, said engine starting system comprising:

a fuel injection controlling section (2d) for controlling actuation of fuel injectors (16) of which nozzle ends are located on the interior of individual cylinders;
an ignition controlling section (2e) for controlling ignition timing for the individual cylinders; and
an air quantity detecting section (2c) for detecting the quantity of air in the cylinder which is on a compression stroke at engine stop;

wherein said fuel injection controlling section (2d) causes the fuel injector (16) of the cylinder which is on the compression stroke at engine stop to inject fuel and a mixture produced in said compression stroke cylinder is ignited and burnt so that the engine once turns in a reverse direction from idle stop, and said fuel injection controlling section (2d) causes the fuel injector (16) of the cylinder which is on an expansion stroke at engine stop to inject the fuel when gas in said expansion stroke cylinder has been compressed as a result of reverse running motion of the engine and a mixture produced in said expansion stroke cylinder is ignited and burnt so that a torque acting in a forward direction is generated to restart the engine;

characterized in that said fuel injection controlling section (2d) controls the quantity of fuel injected into said compression stroke cylinder such that an average air-fuel ratio produced therein becomes lower than the stoichiometric air-fuel ratio based on a value of the quantity of air detected by said air quantity detecting section (2c), and said fuel injection controlling section (2d) causes the fuel injector (16) of the cylinder which was on the compression stroke at engine stop to inject an additional quantity of fuel when gas in the same cylinder has been compressed as a result of forward running motion of the engine.

2. The engine starting system according to claim 1, wherein said fuel injection controlling section causes the fuel injector of the cylinder which was on the compression stroke at engine stop to inject the additional quantity of fuel at a point in a period including a middle part and a latter part of the compression stroke.

3. The engine starting system according to claim 1 or 2, wherein an exhaust-converting catalyst having an oxygen-adsorbing capability is provided in an exhaust passage of the engine, said engine starting system further comprising:

an engine stopping section for stopping the engine by interrupting fuel supply to the individual cylinders while the engine is running;
an intake air quantity regulator for regulating the quantities of air introduced into the individual cylinders; and
an intake air quantity regulator controlling section for controlling said intake air quantity regulator in such a manner that the quantities of air introduced into the individual cylinders increase for a specific period of time in an engine stopping period which begins at a point of fuel supply interruption and ends at a point of engine stop.

4. The engine starting system according to claim 1, wherein said fuel injection controlling section causes the fuel injector of the cylinder which was on an intake stroke at engine stop to inject the fuel when the same cylinder has transferred to the compression stroke as a result of the forward running motion of the engine.

5. The engine starting system according to claim 4, wherein said fuel injection controlling section causes the fuel injector of the cylinder which was on the intake stroke at engine stop to inject the fuel at a point in a period including a middle part and a latter part of the compression stroke when the same cylinder has transferred to the compression stroke.

6. The engine starting system according to claim 4 or 5, wherein said ignition controlling section retards the ignition timing for the cylinder which was on the intake stroke at engine stop to a point beyond the compression stroke top dead center.

7. The engine starting system according to one of claims 4 to 6 further comprising:

an engine stopping section for stopping the engine by interrupting fuel supply to the individual cylinders while the engine is running;

wherein closing timing of an intake valve of each of the cylinders is set at a point retarded by a specific amount at least when the engine is stopped by said engine stopping section.

8. The engine starting system according to one of claims 4 to 7, wherein said fuel injection controlling section causes the fuel injector of the cylinder which was on the intake stroke at engine stop to inject the fuel for initial combustion in the same cylinder on the compression stroke and the fuel for second and subsequent combustions in the same cylinder on the intake stroke.

9. The engine starting system according to one of claims 4 to 7, wherein said fuel injection controlling section causes the fuel injector of each of the cylinders to inject the fuel on the intake stroke if the engine is restarted by use of a starter motor.

10. The engine starting system according to claim 1 further comprising:

an engine temperature sensor for judging engine temperature conditions;

wherein said fuel injection controlling section causes the fuel injector of the cylinder which was on an exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the compression stroke via an intake stroke as a result of the forward running motion of the engine if the engine temperature is judged to be equal to or higher than a specific temperature by said engine temperature sensor, whereas said fuel injection controlling section causes the fuel injector of the cylinder which was on the exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the intake stroke if the engine temperature is judged to be lower than the specific temperature by said engine temperature sensor.

11. The engine starting system according to claim 1 further comprising:

an engine stop time counter for measuring time elapsed from a point of engine stop to a point of engine restart;

wherein said fuel injection controlling section causes the fuel injector of the cylinder which was on an exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the compression stroke via an intake stroke as a result of the forward running motion of the engine if the elapsed time measured by said engine stop time counter is equal to or shorter than a specific time period, whereas said fuel injection controlling section causes the fuel injector of the cylinder which was on the exhaust stroke at engine stop to inject the fuel when the same cylinder has transferred to the intake stroke if the elapsed time measured by said engine stop time counter is longer than the specific time period.

12. The engine starting system according to claim 10 or 11, wherein said ignition controlling section retards the ignition timing for the cylinder which was on the exhaust stroke at engine stop to a point beyond the compression stroke top dead center.

FIG.1

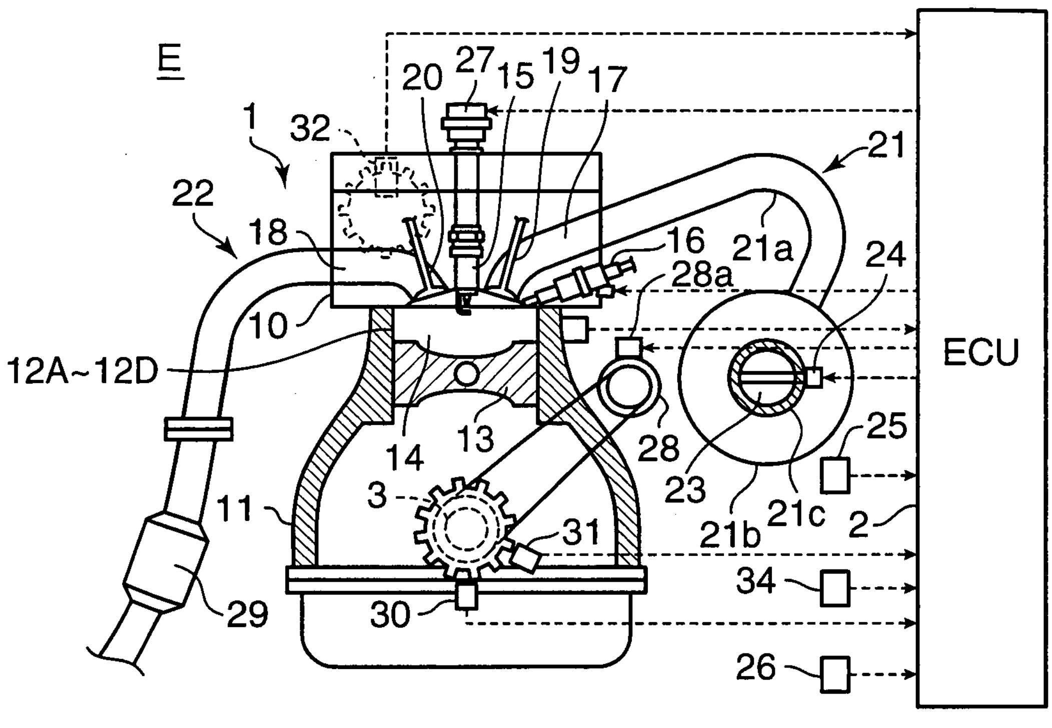

E
1
32
22
18
10
12A~12D
20
27
15
19
17
21
21a
16
28a
24
ECU
13
14
28
23
21b
21c
25
2
3
11
31
34
30
29
26

FIG.2

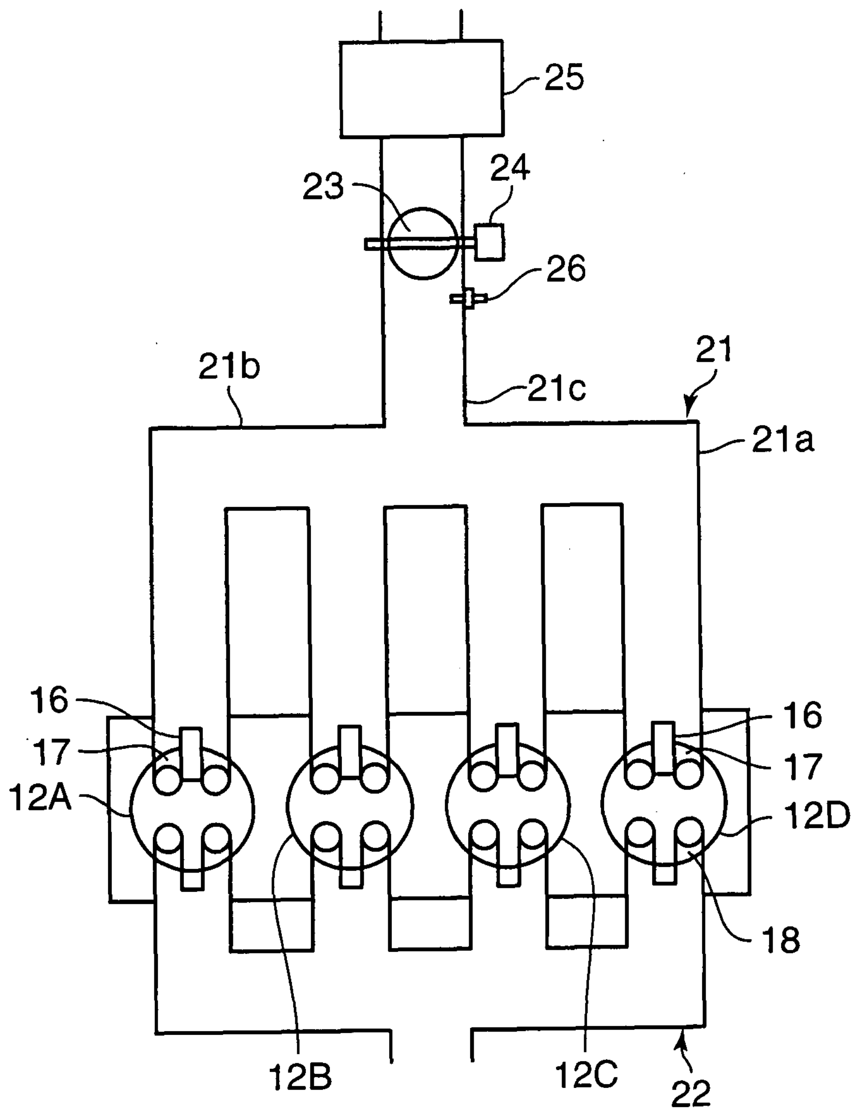


FIG.3A

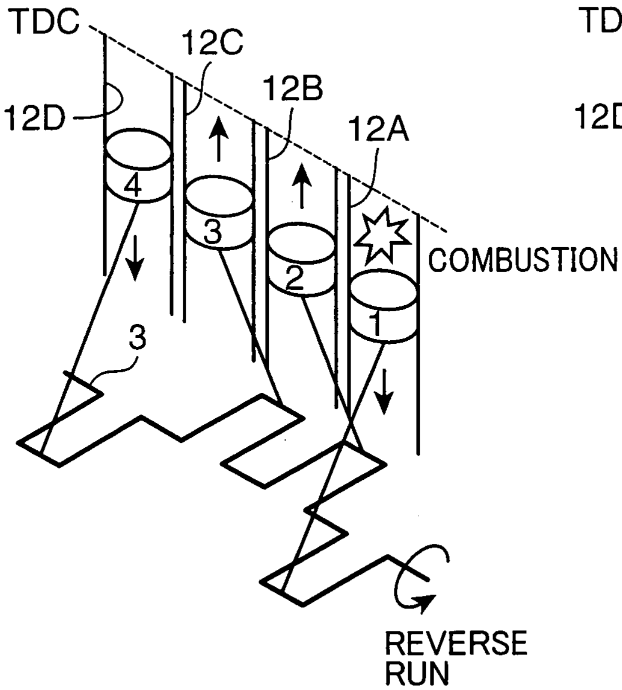


FIG.3B

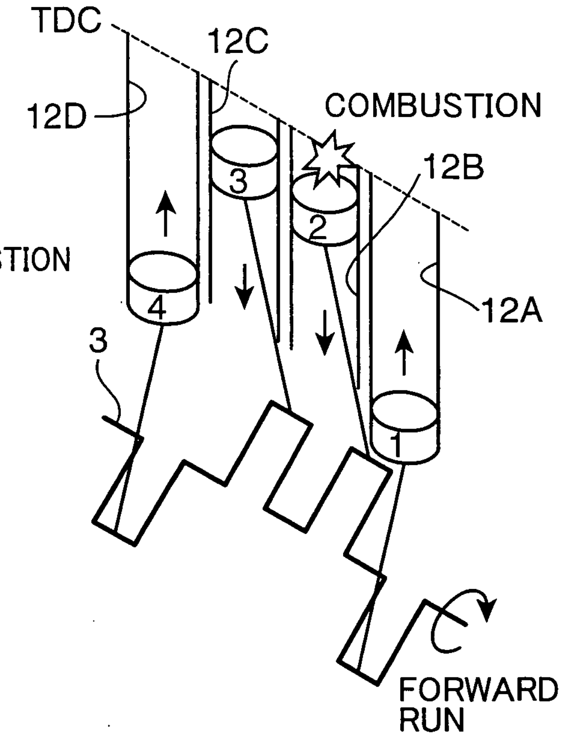


FIG.3C

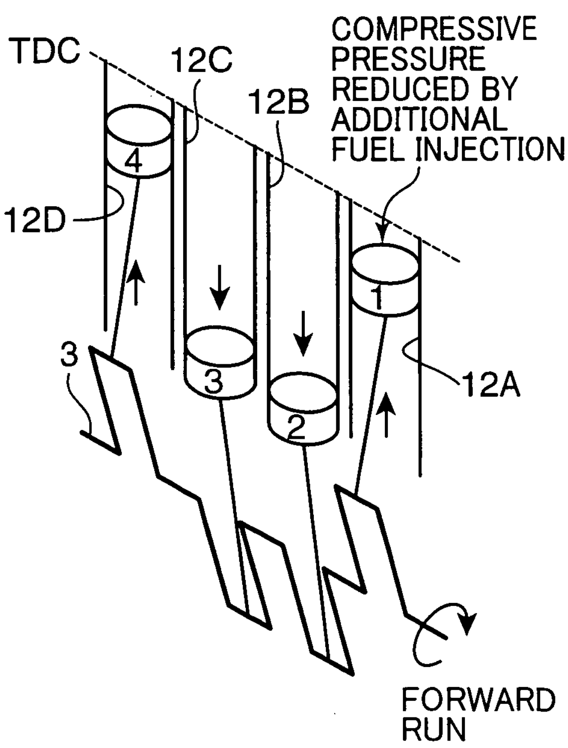


FIG.3D

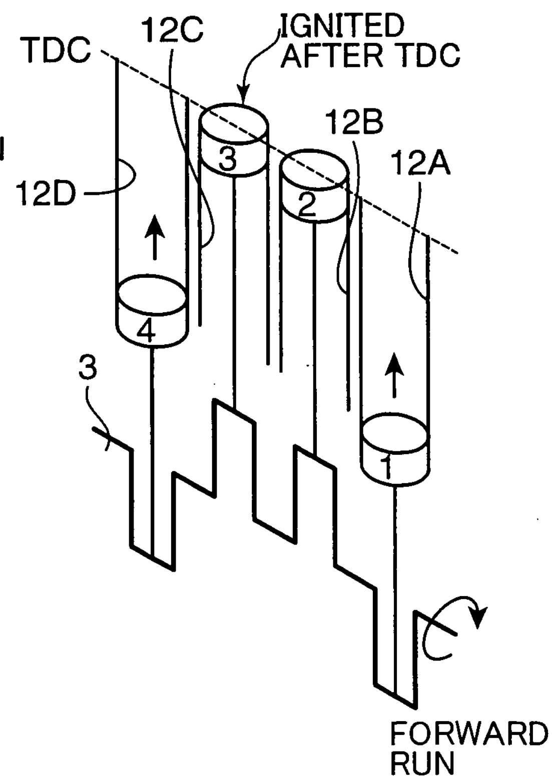


FIG.4

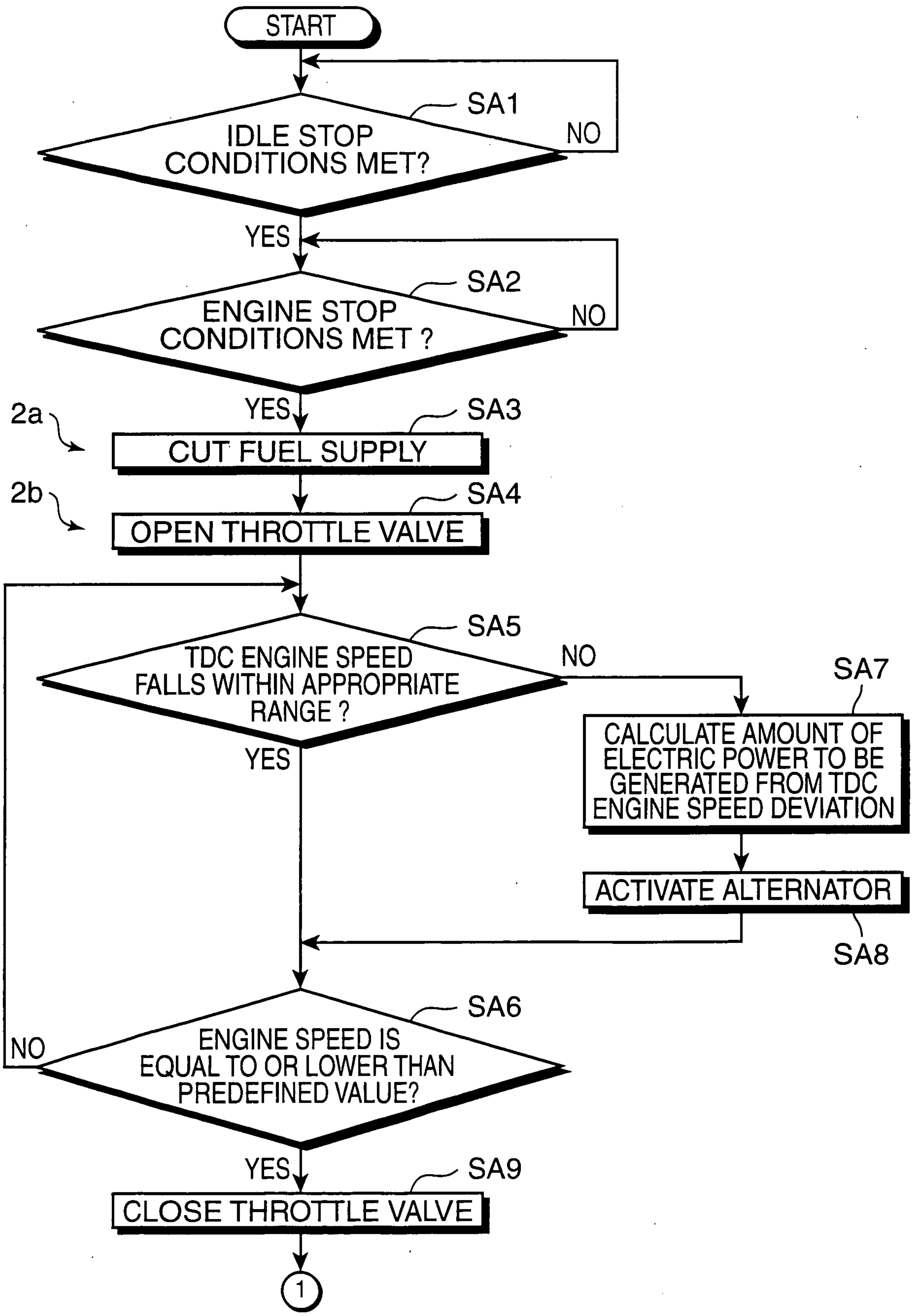
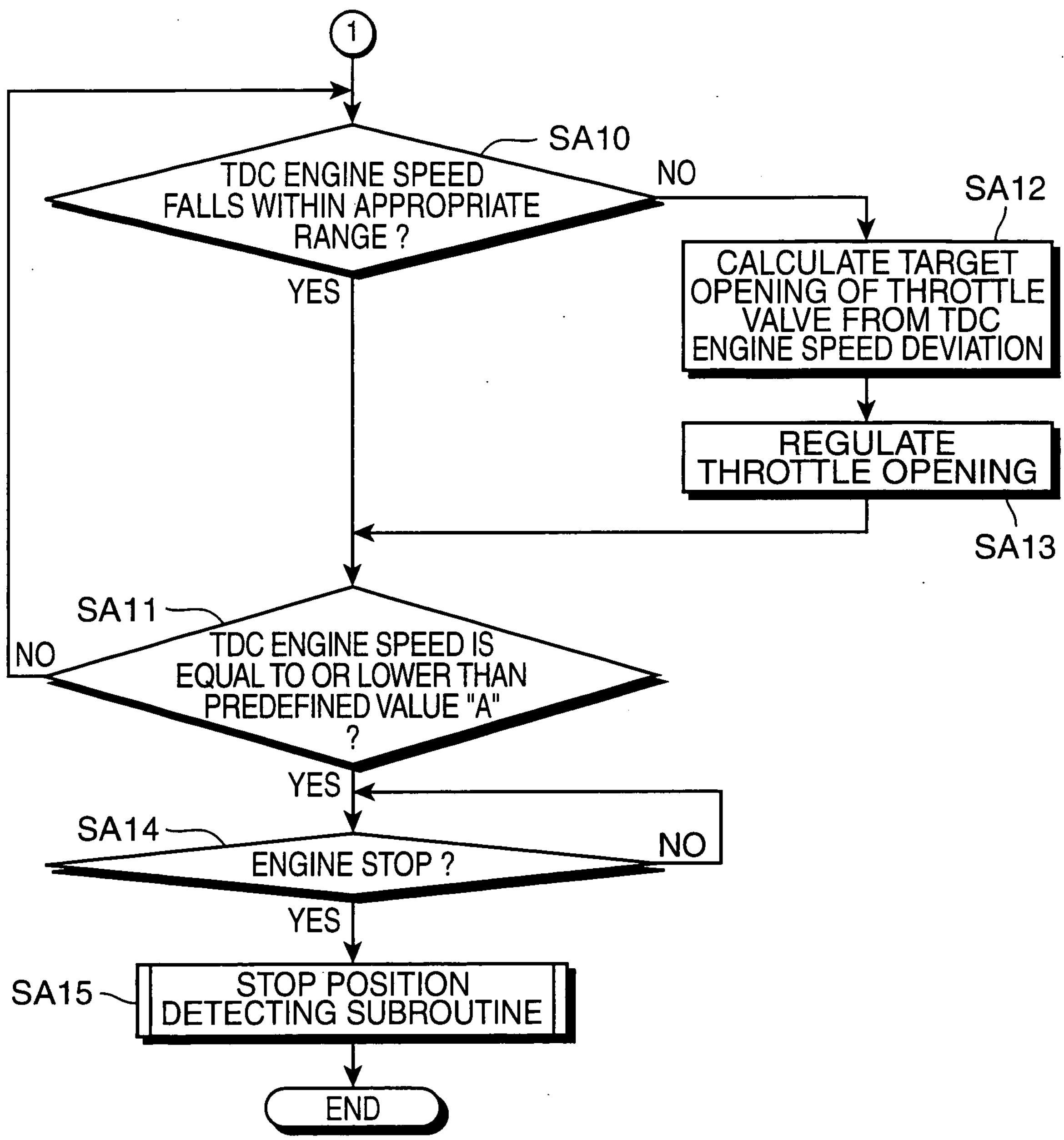

START
SA1
IDLE STOP CONDITIONS MET?
NO
YES
SA2
ENGINE STOP CONDITIONS MET ?
NO
YES
2a
SA3
CUT FUEL SUPPLY
2b
SA4
OPEN THROTTLE VALVE
SA5
TDC ENGINE SPEED FALLS WITHIN APPROPRIATE RANGE ?
NO
YES
SA7
CALCULATE AMOUNT OF ELECTRIC POWER TO BE GENERATED FROM TDC ENGINE SPEED DEVIATION
ACTIVATE ALTERNATOR
SA8
SA6
ENGINE SPEED IS EQUAL TO OR LOWER THAN PREDEFINED VALUE?
NO
YES
SA9
CLOSE THROTTLE VALVE
1

FIG.5


1
SA10
TDC ENGINE SPEED FALLS WITHIN APPROPRIATE RANGE ?
NO
YES
SA12
CALCULATE TARGET OPENING OF THROTTLE VALVE FROM TDC ENGINE SPEED DEVIATION
REGULATE THROTTLE OPENING
SA13
SA11
NO
TDC ENGINE SPEED IS EQUAL TO OR LOWER THAN PREDEFINED VALUE "A" ?
YES
SA14
ENGINE STOP ?
NO
YES
SA15
STOP POSITION DETECTING SUBROUTINE
END

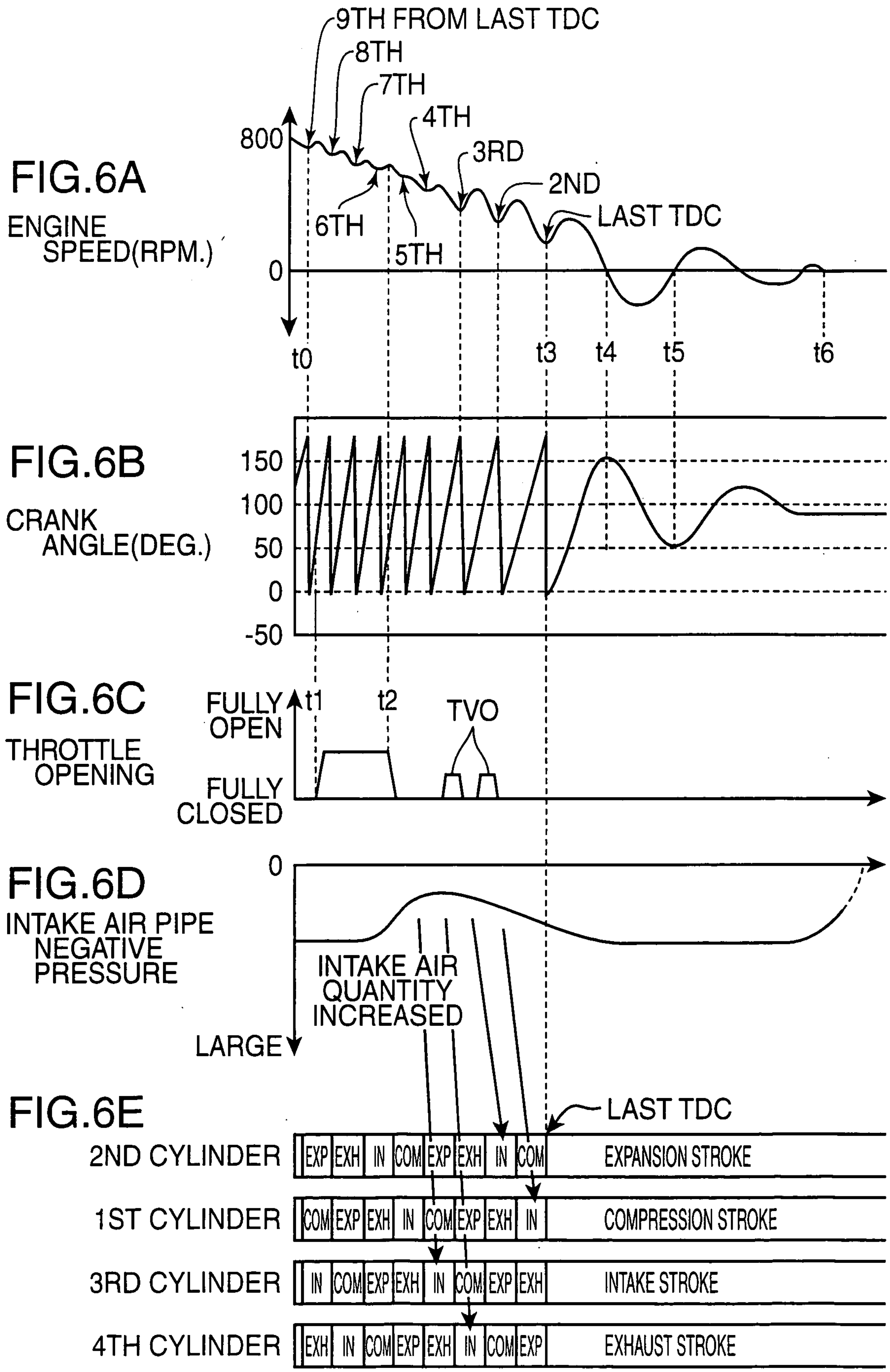
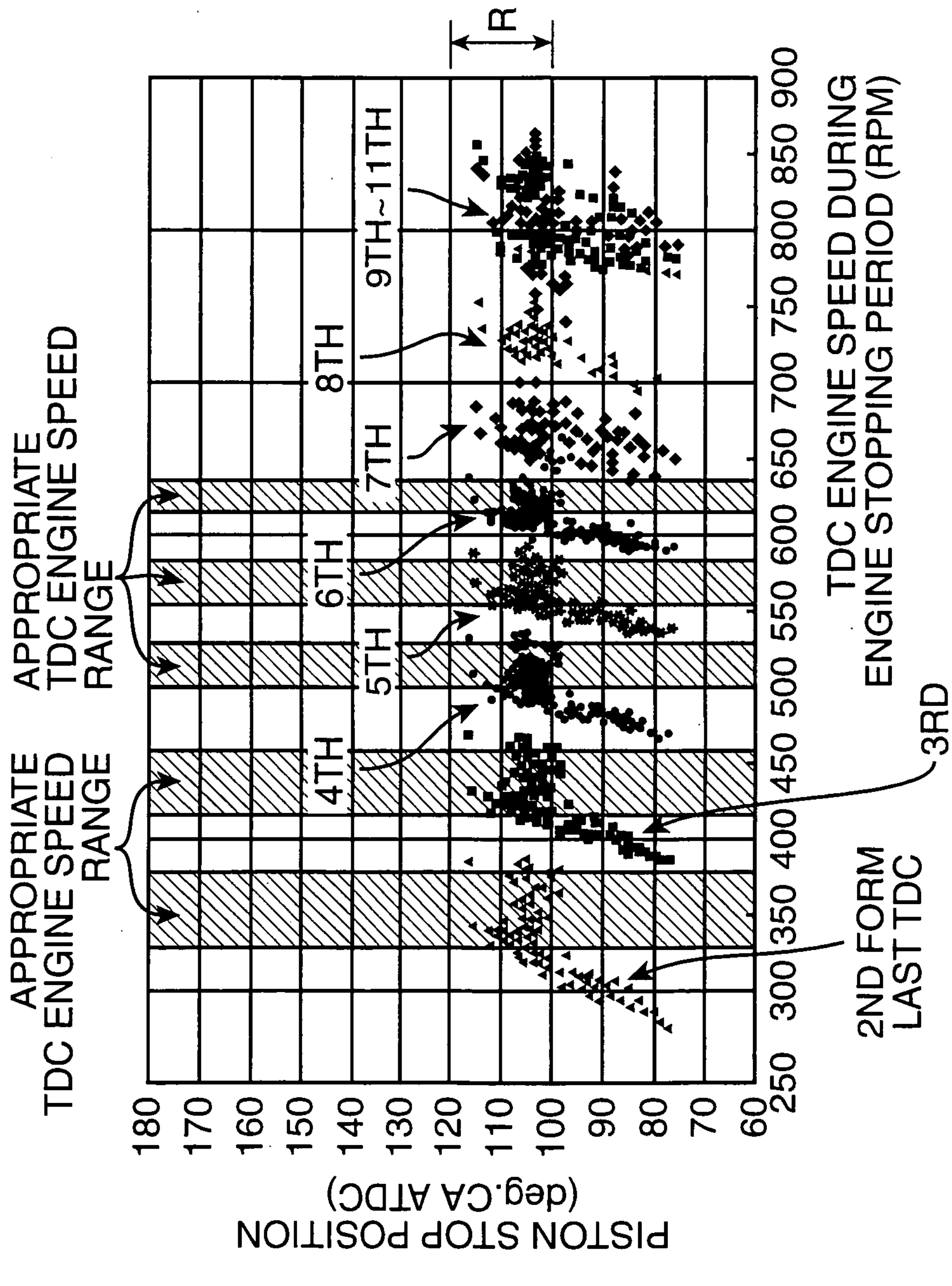


FIG.6A
ENGINE SPEED(RPM.)
9TH FROM LAST TDC
8TH
7TH
6TH
5TH
4TH
3RD
2ND
LAST TDC
800
0
t0
t3
t4
t5
t6
FIG.6B
CRANK ANGLE(DEG.)
150
100
50
0
-50
FIG.6C
THROTTLE OPENING
FULLY OPEN
FULLY CLOSED
t1
t2
TVO
FIG.6D
INTAKE AIR PIPE NEGATIVE PRESSURE
0
LARGE
INTAKE AIR QUANTITY INCREASED
FIG.6E
LAST TDC
2ND CYLINDER
EXP EXH IN COM EXP EXH IN COM
EXPANSION STROKE
1ST CYLINDER
COM EXP EXH IN COM EXP EXH IN
COMPRESSION STROKE
3RD CYLINDER
IN COM EXP EXH IN COM EXP EXH
INTAKE STROKE
4TH CYLINDER
EXH IN COM EXP EXH IN COM EXP
EXHAUST STROKE

FIG.7



APPROPRIATE TDC ENGINE SPEED RANGE
APPROPRIATE TDC ENGINE SPEED RANGE
PISTON STOP POSITION (deg.CA ATDC)
180 170 160 150 140 130 120 110 100 90 80 70 60
250 300 350 400 450 500 550 600 650 700 750 800 850 900
TDC ENGINE SPEED DURING ENGINE STOPPING PERIOD (RPM)
R
4TH
5TH
6TH
7TH
8TH
9TH~11TH
3RD
2ND FORM LAST TDC

FIG.8

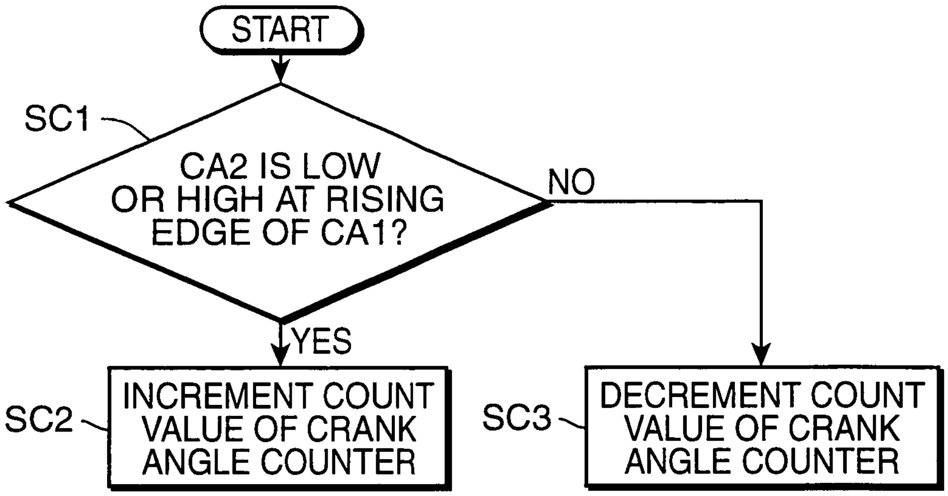


FIG.9A

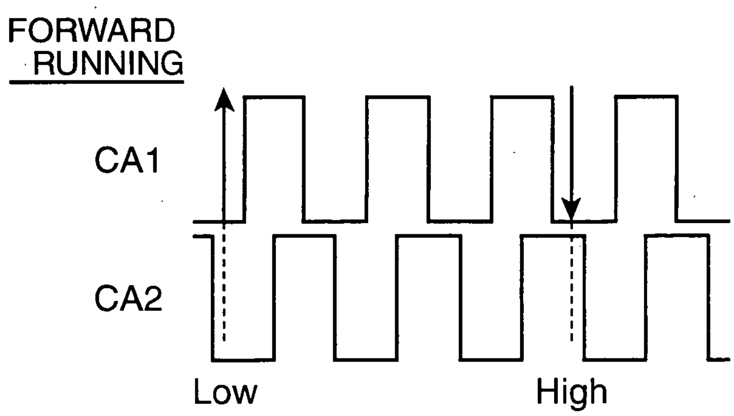


FIG.9B

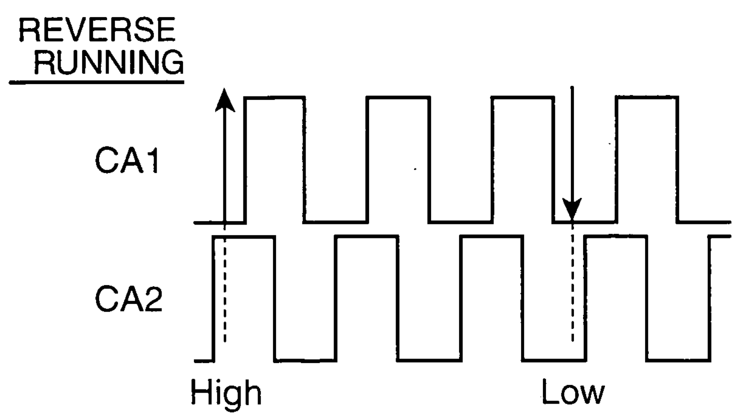


FIG.10

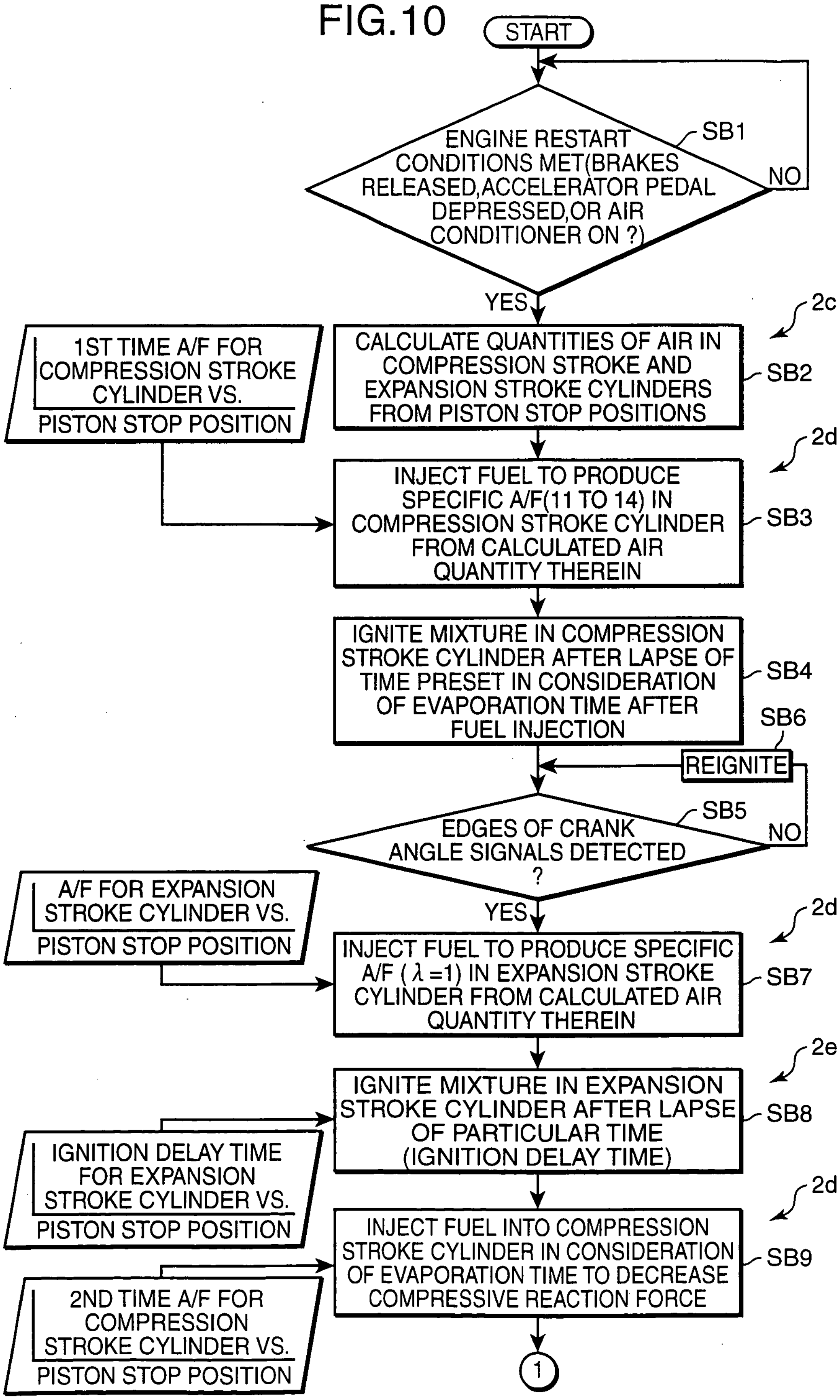

START
ENGINE RESTART CONDITIONS MET(BRAKES RELEASED,ACCELERATOR PEDAL DEPRESSED,OR AIR CONDITIONER ON ?)
SB1
NO
YES
2c
1ST TIME A/F FOR COMPRESSION STROKE CYLINDER VS. PISTON STOP POSITION
CALCULATE QUANTITIES OF AIR IN COMPRESSION STROKE AND EXPANSION STROKE CYLINDERS FROM PISTON STOP POSITIONS
SB2
2d
INJECT FUEL TO PRODUCE SPECIFIC A/F(11 TO 14) IN COMPRESSION STROKE CYLINDER FROM CALCULATED AIR QUANTITY THEREIN
SB3
IGNITE MIXTURE IN COMPRESSION STROKE CYLINDER AFTER LAPSE OF TIME PRESET IN CONSIDERATION OF EVAPORATION TIME AFTER FUEL INJECTION
SB4
SB6
REIGNITE
EDGES OF CRANK ANGLE SIGNALS DETECTED ?
SB5
NO
YES
2d
A/F FOR EXPANSION STROKE CYLINDER VS. PISTON STOP POSITION
INJECT FUEL TO PRODUCE SPECIFIC A/F (λ=1) IN EXPANSION STROKE CYLINDER FROM CALCULATED AIR QUANTITY THEREIN
SB7
2e
IGNITE MIXTURE IN EXPANSION STROKE CYLINDER AFTER LAPSE OF PARTICULAR TIME (IGNITION DELAY TIME)
SB8
IGNITION DELAY TIME FOR EXPANSION STROKE CYLINDER VS. PISTON STOP POSITION
2d
INJECT FUEL INTO COMPRESSION STROKE CYLINDER IN CONSIDERATION OF EVAPORATION TIME TO DECREASE COMPRESSIVE REACTION FORCE
SB9
2ND TIME A/F FOR COMPRESSION STROKE CYLINDER VS. PISTON STOP POSITION
1

FIG.11

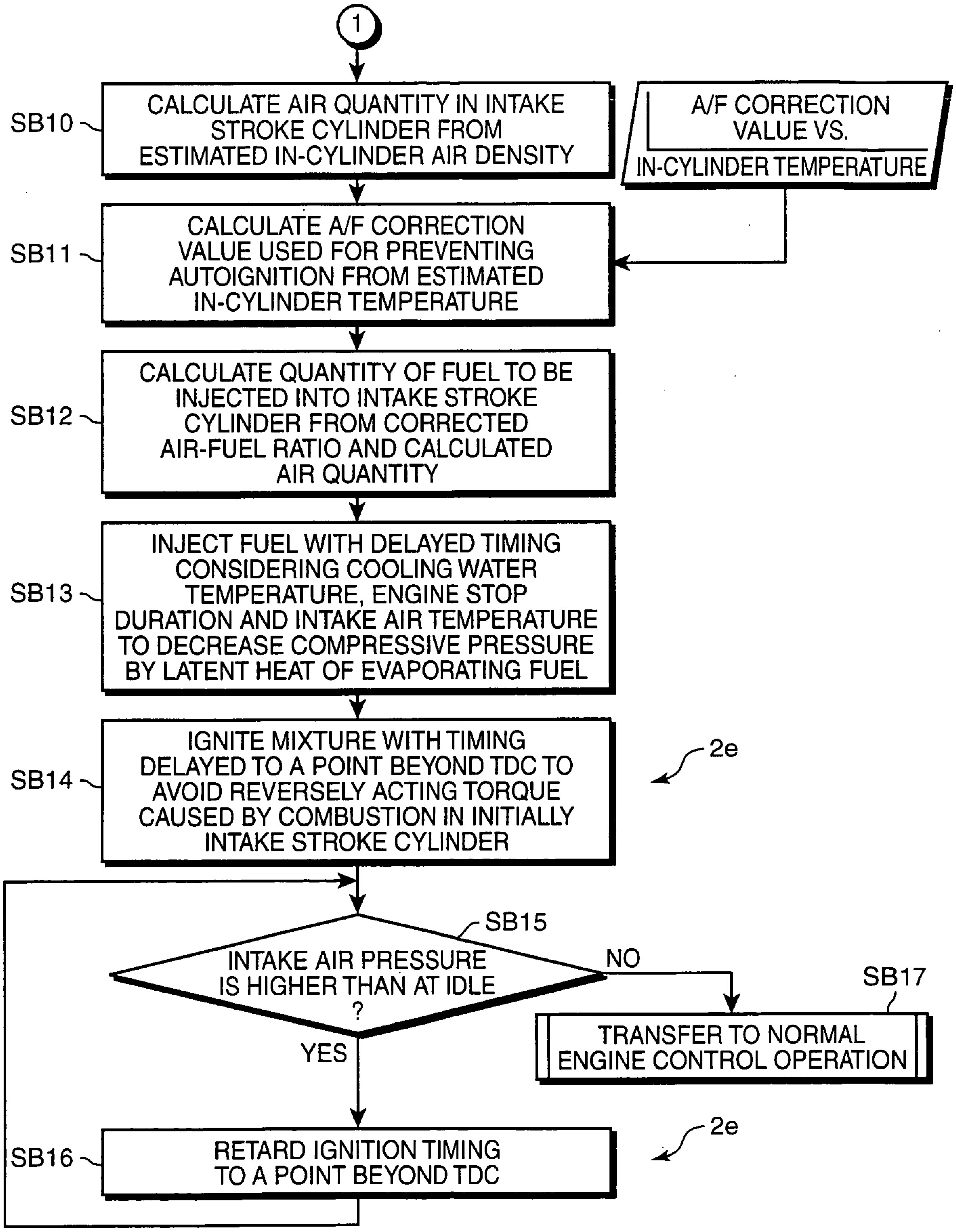
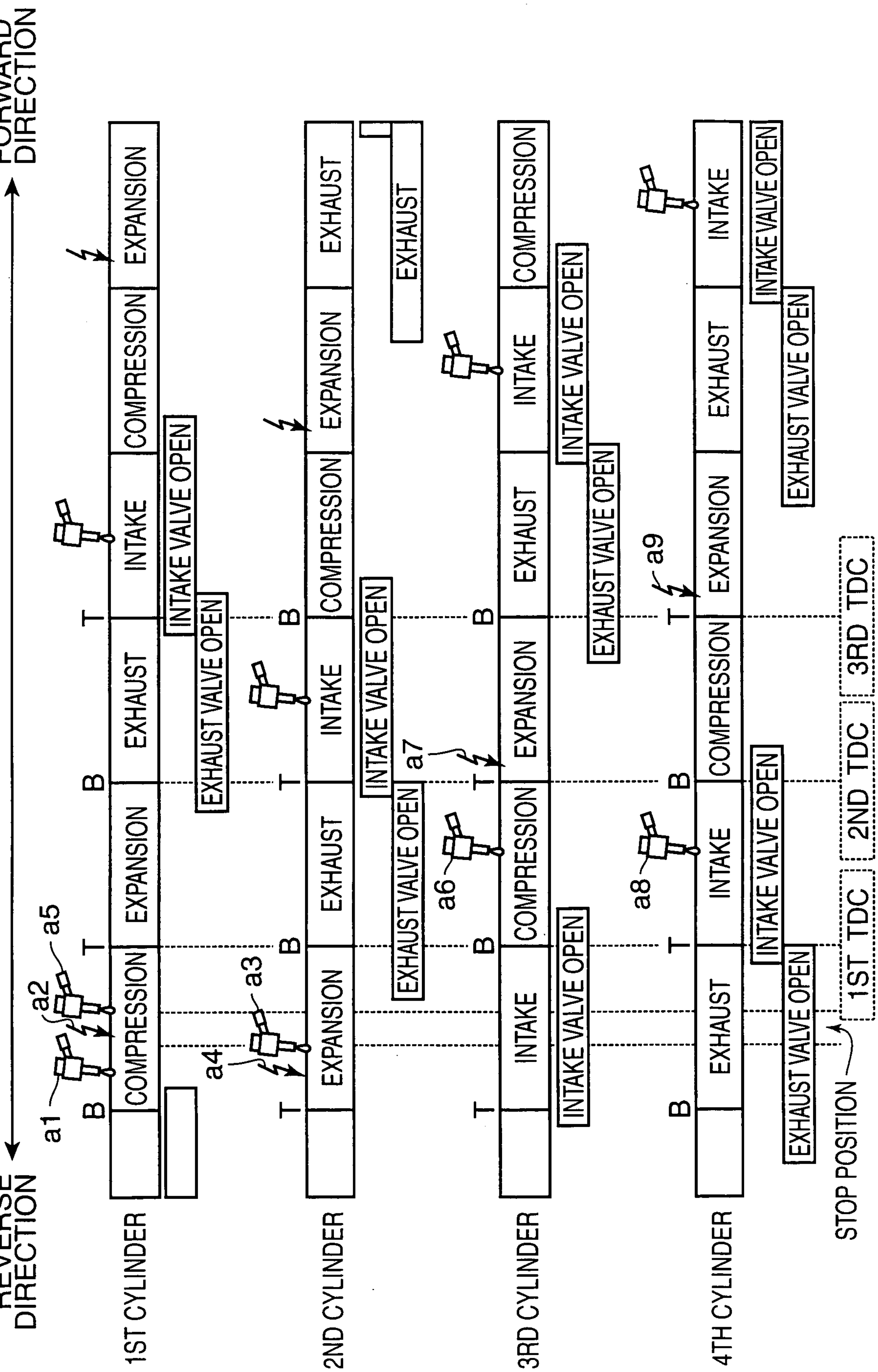

1
SB10
CALCULATE AIR QUANTITY IN INTAKE STROKE CYLINDER FROM ESTIMATED IN-CYLINDER AIR DENSITY
A/F CORRECTION VALUE VS.
IN-CYLINDER TEMPERATURE
SB11
CALCULATE A/F CORRECTION VALUE USED FOR PREVENTING AUTOIGNITION FROM ESTIMATED IN-CYLINDER TEMPERATURE
SB12
CALCULATE QUANTITY OF FUEL TO BE INJECTED INTO INTAKE STROKE CYLINDER FROM CORRECTED AIR-FUEL RATIO AND CALCULATED AIR QUANTITY
SB13
INJECT FUEL WITH DELAYED TIMING CONSIDERING COOLING WATER TEMPERATURE, ENGINE STOP DURATION AND INTAKE AIR TEMPERATURE TO DECREASE COMPRESSIVE PRESSURE BY LATENT HEAT OF EVAPORATING FUEL
SB14
IGNITE MIXTURE WITH TIMING DELAYED TO A POINT BEYOND TDC TO AVOID REVERSELY ACTING TORQUE CAUSED BY COMBUSTION IN INITIALLY INTAKE STROKE CYLINDER
2e
SB15
INTAKE AIR PRESSURE IS HIGHER THAN AT IDLE ?
NO
YES
SB17
TRANSFER TO NORMAL ENGINE CONTROL OPERATION
SB16
RETARD IGNITION TIMING TO A POINT BEYOND TDC
2e

FIG.12


REVERSE DIRECTION
FORWARD DIRECTION
1ST CYLINDER
2ND CYLINDER
3RD CYLINDER
4TH CYLINDER
COMPRESSION
EXPANSION
EXHAUST
INTAKE
INTAKE VALVE OPEN
EXHAUST VALVE OPEN
B
T
a1
a2
a3
a4
a5
a6
a7
a8
a9
STOP POSITION
1ST TDC
2ND TDC
3RD TDC

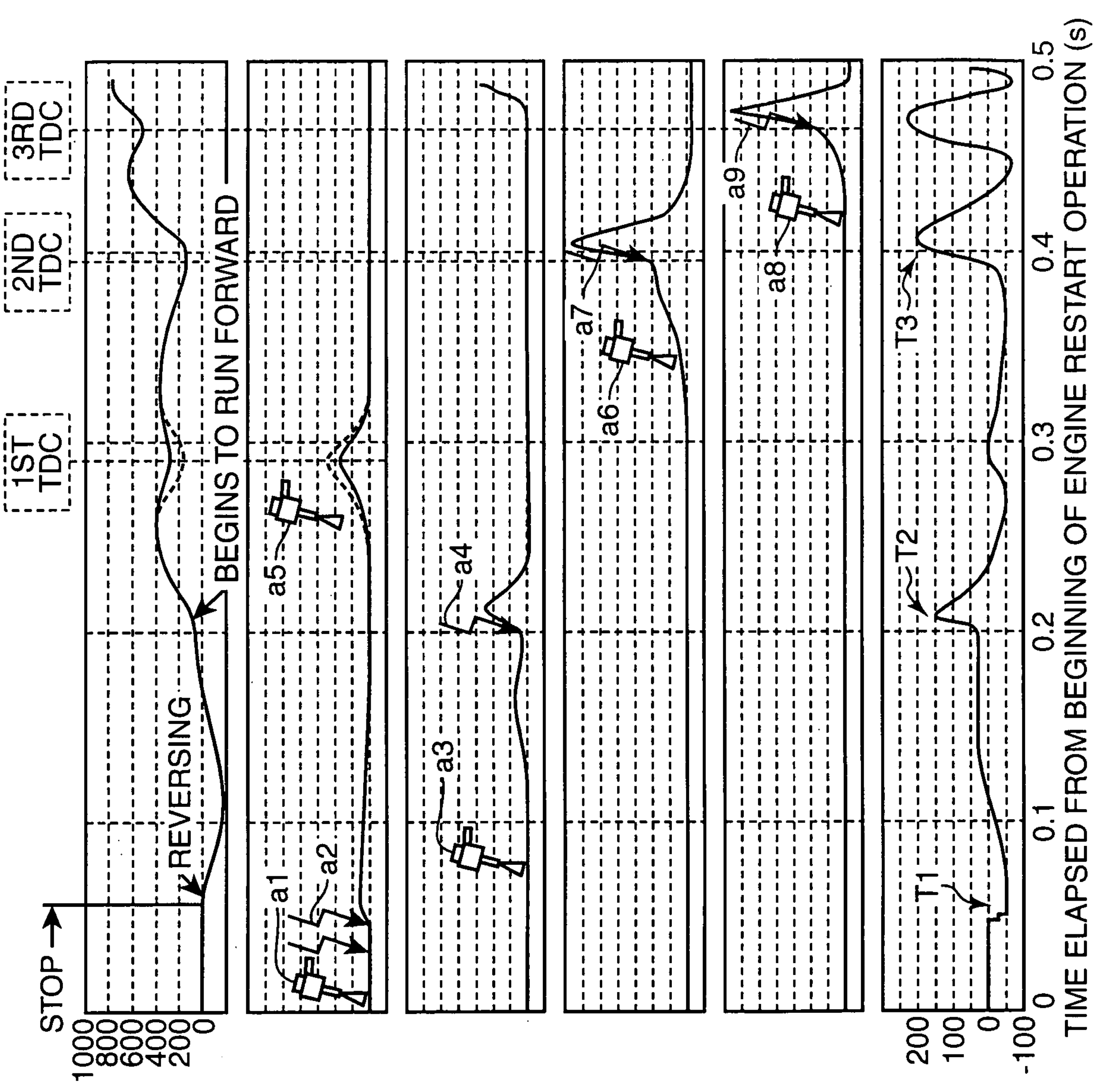


FIG.13A ENGINE SPEED(RPM)

FIG.13B PRESSURE IN 1ST CYLINDER(Mpa)

FIG.13C PRESSURE IN 2ND CYLINDER(Mpa)

FIG.13D PRESSURE IN 3RD CYLINDER(Mpa)

FIG.13E PRESSURE IN 4TH CYLINDER(Mpa)

FIG.13F TORQUE(Nm)

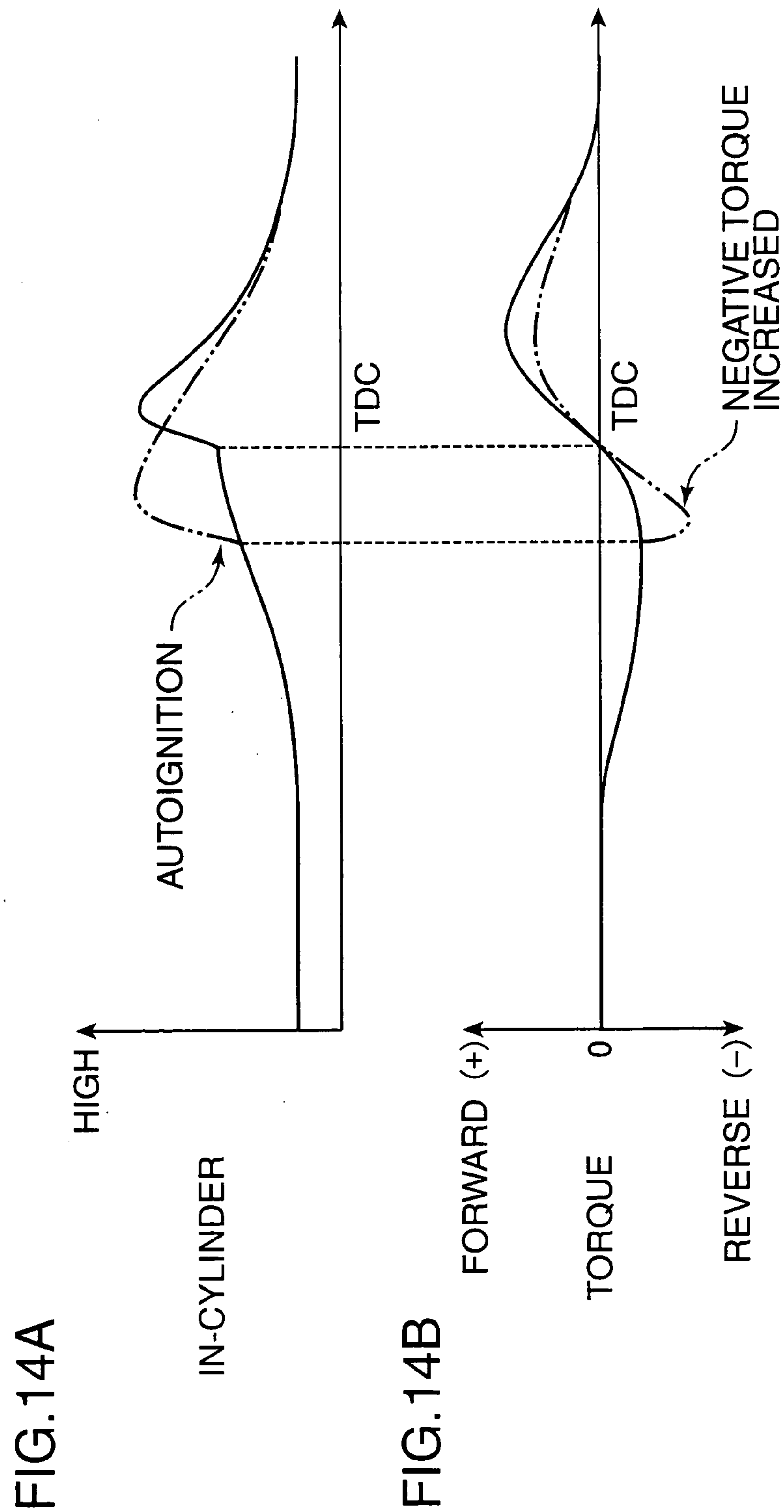


FIG.14A

FIG.14B

FIG.15

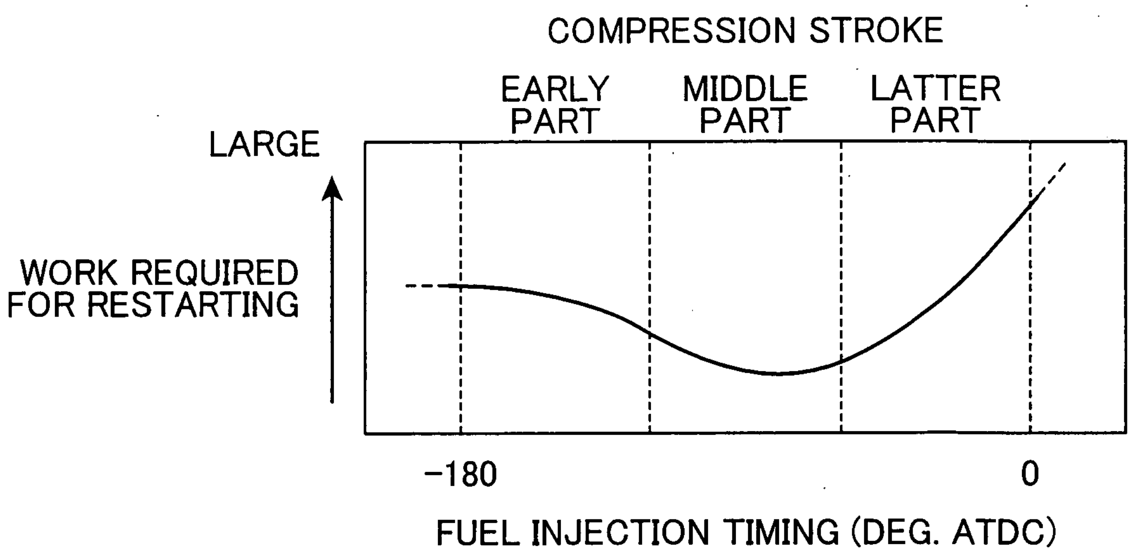
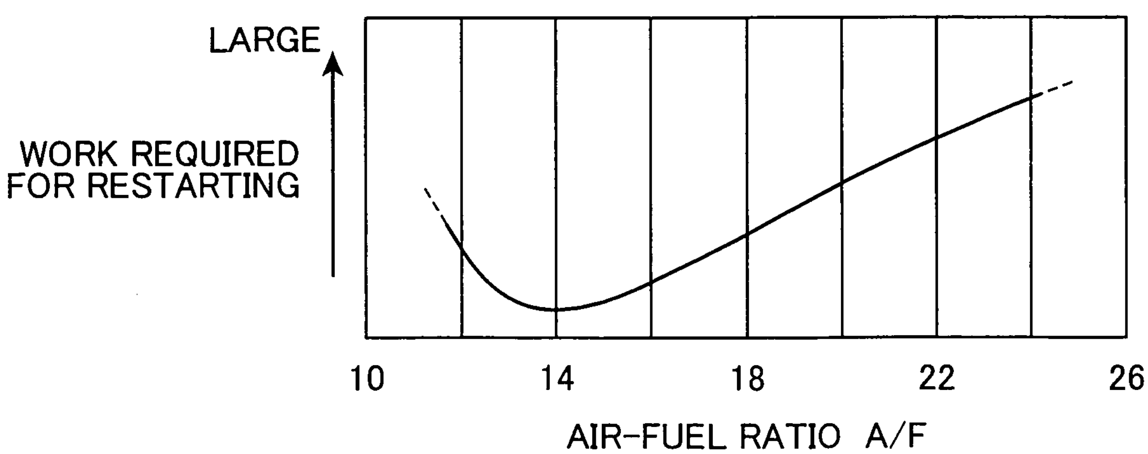

COMPRESSION STROKE
EARLY PART
MIDDLE PART
LATTER PART
LARGE
WORK REQUIRED FOR RESTARTING
−180
0
FUEL INJECTION TIMING (DEG. ATDC)

FIG.16


LARGE
WORK REQUIRED FOR RESTARTING
10
14
18
22
26
AIR-FUEL RATIO A/F