



(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:  
**12.04.2000 Bulletin 2000/15**

(51) Int. Cl.<sup>7</sup>: **F02D 43/00**, F02D 21/08,  
F02D 41/14

(21) Application number: **99118823.6**

(22) Date of filing: **23.09.1999**

(84) Designated Contracting States:  
**AT BE CH CY DE DK ES FI FR GB GR IE IT LI LU  
MC NL PT SE**  
Designated Extension States:  
**AL LT LV MK RO SI**

(30) Priority: **02.10.1998 JP 28121398**  
**16.10.1998 JP 29538198**  
**11.11.1998 JP 32101598**

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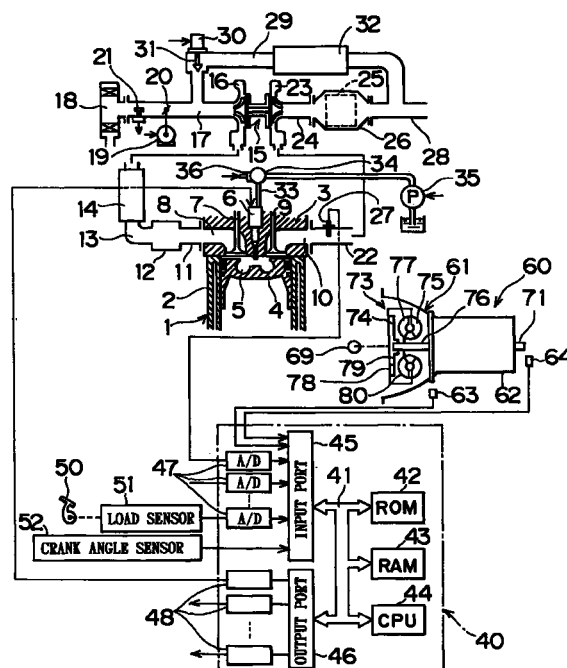
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(54) **Internal combustion engine**

(57) At a time of executing a low temperature combustion in which an amount of an EGR gas supplied within a combustion chamber (5) is more than an amount of the EGR gas when a generation amount of a soot becomes peak and the soot is hardly generated and changing a speed of an automatic transmission (60), in order to reduce a generated torque, an injection amount of a fuel injected from a fuel injection valve (6) is corrected to be increased so as to reduce an air fuel ratio, or a fuel injection timing is delayed, or an opening degree of an EGR control valve (31) is corrected to be increased so as to correct an amount of the EGR gas to increase, and further at a time of executing a combustion in which the amount of the EGR gas supplied within a combustion chamber (5) is less than the amount of the EGR gas when a generation amount of a soot becomes peak and the soot is hardly generated and changing a speed of an automatic transmission (60), in order to reduce a generated torque, the injection amount of the fuel injected from the fuel injection valve (6) is corrected to be reduced.

**FIG. 1**



## Description

### BACKGROUND OF THE INVENTION

#### 1. FIELD OF THE INVENTION

[0001] The present invention relates to an internal combustion engine introducing an inert gas into a combustion chamber so as to perform combustion in accordance with the preamble of claim 1.

#### 2. DESCRIPTION OF THE RELATED ART

[0002] Conventionally, in an internal combustion engine, for example, in a diesel engine, in order to restrict generation of NOx, it is structured an engine exhaust passage and an engine intake passage are connected by an exhaust gas recirculation (hereinafter, refer to an EGR: an Exhaust Gas Recirculation) so as to recirculate the exhaust gas, that is, EGR gas into the engine intake passage via the EGR passage. In this case, since the EGR gas has a relatively high specific heat and accordingly can absorb a large amount of heat, a combustion temperature within the combustion chamber is lowered as an amount of the EGR gas is increased, that is, a rate of the EGR (EGR gas amount/(EGR gas amount + intake air amount)) is increased. When the combustion temperature is lowered, the generation amount of NOx is lowered, so that the more the EGR rate is increased, the less the generation amount of NOx becomes.

[0003] As mentioned above, it has been conventionally known that the generation amount of NOx can be lowered when the EGR rate is increased. However, in the case where the EGR rate is increased, the generation amount of a soot, that is, the smoke suddenly starts increasing when the EGR rate exceeds a certain limit. With respect to this point, it has been conventionally considered that the smoke is unlimitedly increased when the EGR rate is increased further, so that it has been considered that the EGR rate at which the smoke suddenly starts increasing is the maximum allowable limit of the EGR rate.

[0004] Accordingly, the EGR rate has been conventionally defined within a range which does not exceed the maximum allowable limit. The maximum allowable limit of the EGR rate is significantly different in correspondence to a type of the engine and a fuel, however, is within a range from about 30 % to 50 %. Therefore, in the conventional diesel engine, the EGR rate is restricted to the range from about 30 % to 50 % at the maximum.

[0005] As mentioned above, since it has been conventionally considered that the maximum allowable limit exists with respect to the EGR rate, the EGR rate has been defined within the range which does not exceed the maximum allowable limit and so that the generation amount of NOx and the smoke becomes as least as

possible. However, even when the EGR rate is defined so that the generation amount of NOx and the smoke becomes as least as possible, the reduction of the generation amount of NOx and the smoke has a limit, so that actually a significant amount of NOx and smoke are still generated.

[0006] However, in the process of researching combustion in the diesel engine, it has been found that when making the EGR rate greater than the maximum allowable limit, the smoke is suddenly increased as mentioned above, however, the generation amount of the smoke has a peak, and when further increasing the EGR rate to exceed the peak, then the smoke suddenly starts reducing at this time. When setting the EGR rate to the value equal to or more than 70 % at a time of an idling operation or strongly cooling the EGR gas, the smoke becomes substantially 0, that is, the soot is hardly generated when setting the EGR rate to the value equal to or more than 55 %. Further, it has been found that the generation amount of NOx becomes significantly small amount at this time. Thereafter, on the basis of this information, a consideration has been performed with respect to the reason why the soot is not generated, as a result, a new combustion system which has not been obtained and can simultaneously reduce the soot and NOx has been constructed. The new combustion system will be in detail described below. In a word, it is based on a principle that the growth of a hydrocarbon is stopped in the middle of a step by which the hydrocarbon grows the soot.

[0007] That is, it is ascertained as a result of many experiments and researches that the growth of the hydrocarbon stops in the middle of the step before becoming the soot when the temperature of the fuel and the surrounding gas at a time of combustion within the combustion chamber is equal to or less than a certain temperature, and that the hydrocarbon grows the soot at a stroke when the temperature of the fuel and the surrounding gas reaches a certain temperature. In this case, the temperature of the fuel and the surrounding gas is greatly influenced by an endothermic effect of the gas surrounding the fuel at a time when the fuel is burned, so that it is possible to control the temperature of the fuel and the surrounding gas by adjusting the heat absorption amount of the gas surrounding the fuel in correspondence to the generation amount at a time of the fuel combustion.

[0008] Accordingly, when restricting the temperature of the fuel and the surrounding gas at a time of the combustion within the combustion chamber to a level equal to or less than the temperature at which the growth of the hydrocarbon stops halfway, the soot is not generated, so that it is possible to restrict the temperature of the fuel and the surrounding gas at a time of the combustion within the combustion chamber to the level equal to or less than the temperature at which the growth of the hydrocarbon stops on the way by adjusting the heat absorption amount of the gas surrounding the

fuel. On the contrary, the hydrocarbon that stops growing on the way before becoming the soot can be easily purified by the after treatment using an oxidation catalyst or the like. This is a basic principle of the new combustion system. The internal combustion engine employing the new combustion system was already filed as EP 879,946 A2 by the present applicant.

**[0009]** Here, in a vehicle in which an automatic transmission is provided in an internal combustion engine performing a conventional combustion, when the automatic transmission is shifted up, for example, due to the increase in a vehicle speed, an ignition timing of the internal combustion engine is delayed in order to reduce a shift change shock, an amount of the intake air is reduced or an output torque of the engine is reduced.

**[0010]** Further, in the new combustion system as mentioned above, it is desired to reduce the shock due to the torque change generated together with the shift change by the automatic transmission.

## SUMMARY OF THE INVENTION

**[0011]** An object of the present invention is to provide an internal combustion engine which can reduce a shock generated together with a change of a torque generated by the engine when the shift change is performed by the automatic transmission while simultaneously preventing a soot (a smoke) from being discharged from the internal combustion engine and preventing NOx from being discharged.

**[0012]** The above object is solved by combination of features of the main claim, the sub-claims disclose further advantageous embodiments of the invention.

**[0013]** In accordance with the present invention, there is provided an internal combustion engine structured such that a generation amount of a soot is gradually increased to the peak when increasing an amount of an inert gas supplied within a combustion chamber, a temperature of a fuel and a surrounding gas thereof at a time of burning within the combustion chamber becomes lower than a temperature for generating a soot when further increasing the amount of the inert gas supplied within the combustion chamber, so that a soot is hardly generated, and an automatic transmission is connected to the internal combustion engine, wherein it is possible to execute a combustion structured such that an amount of an inert gas supplied within the combustion chamber is more than an amount of the inert gas when the generation amount of the soot becomes peak and a soot is hardly generated, and an air fuel ratio is reduced, an injection timing of the fuel is delayed, or an amount of a recirculated exhaust gas is corrected to be increased when the combustion such that the soot is hardly generated is performed and the automatic transmission is under a control state.

**[0014]** Accordingly, in the case where the air fuel ratio is reduced, since the combustion in which the soot is hardly generated is performed under a condition that

an air tends to be insufficient, the combustion is deteriorated due to the reduction of the air fuel ratio when the combustion in which the soot is hardly generated is performed, so that the torque generated by the engine is reduced. Accordingly, it is possible to soften the shock due to the torque change when the torque generated by the engine is changed together with the shift change by the automatic transmission.

**[0015]** Further, in the case where the injection timing of the fuel is delayed, the combustion is deteriorated in accordance that the fuel supply is too late for the combustion when the injection timing of the fuel supplied within the combustion chamber is delayed, thus lowering the torque generated in the engine is lowered. Accordingly, it is possible to reduce the shock due to the torque change when the torque generated in the engine is changed in accordance that the shift change by the automatic transmission is performed.

**[0016]** Still further, in the case where the amount of the recirculated exhaust gas is corrected to be increased, since the combustion in which the soot is hardly generated is performed under a condition where the air tends to be insufficient, it is hard that the air is supplied to the combustion chamber when the amount of the recirculated exhaust gas is corrected to be increased when the combustion in which the soot is hardly generated is performed, so that the air tends to be further insufficient. Accordingly, the combustion is deteriorated and the torque generated in the engine is lowered. Therefore, it is possible to reduce the shock generated by the torque change when the torque generated in the engine is changed in accordance that the shift change by the automatic transmission is performed.

**[0017]** Although this summary does not describe all the features of the present invention, it should be understood that any combination of the features stated in the dependent claims is within the scope of the present invention.

## BRIEF DESCRIPTION OF THE DRAWINGS

### **[0018]**

Fig. 1 is a total view of a compression ignition type internal combustion engine in accordance with the present invention;

Fig. 2 is a graph which shows a change of a smoke, HC, CO and NOx in correspondence to a change of an output torque and an air fuel ratio;

Figs. 3A and 3B are graphs which show a combustion pressure;

Fig. 4 is a view which shows a molecule of a fuel;

Fig. 5 is a graph which shows a relation between a generation amount of a smoke and an EGR rate;

Fig. 6 is a graph which shows a relation between a total amount of an intake gas and a required load;

Fig. 7 is a graph which shows a first operation area

I and a second operation area II;

Fig. 8 is a graph which shows an output of an air fuel ratio sensor;

Fig. 9 is a graph which shows an opening degree of a throttle valve, an opening degree of an EGR control valve, an EGR rate, an air fuel ratio, an injection timing and an injection amount in the required load; Fig. 10 is a graph which shows an air fuel ratio in the first operation area I;

Figs. 11A and 11B are views which shows a map of a target opening degree of the throttle valve and the like;

Fig. 12 is a graph which shows an air fuel ratio in a second combustion;

Figs. 13A and 13B are views which show a map of a target opening degree of the throttle valve and the like;

Fig. 14 is a view which shows a map of a fuel injection amount;

Figs. 15 and 16 are flow charts for controlling an operation of an engine in accordance with a first embodiment;

Fig. 17 is a graph which shows a relation between an air fuel ratio and a torque generated in the engine;

Figs. 18 and 19 are flow charts for controlling an operation of an engine in accordance with a second embodiment; and

Figs. 20 and 21 are flow charts for controlling the operation of the engine in accordance with the second embodiment.

## DESCRIPTION OF THE PREFERRED EMBODIMENT

**[0019]** Fig. 1 shows an embodiment in which the present invention is applied to a 4-stroke compression ignition type internal combustion engine.

**[0020]** With reference to Fig. 1, a reference numeral 1 denotes a main body of an engine, reference numeral 2 denotes a cylinder block, reference numeral 3 denotes a cylinder head, reference numeral 4 denotes a piston, reference numeral 5 denotes a combustion chamber, reference numeral 6 denotes an electrically controlled type fuel injection valve, reference numeral 7 denotes an intake valve, reference numeral 8 denotes an intake port, reference numeral 9 denotes an exhaust valve, and reference numeral 10 denotes an exhaust port, respectively. The intake port 8 is connected to a surge tank 12 via a corresponding intake branch pipe 11, and the surge tank 12 is connected to a supercharger, for example, an outlet portion of a compressor 16 of an exhaust turbo charger 15 via an intake duct 13 and an inter cooler 14. An inlet portion of the compressor 16 is connected to an air cleaner 18 via an air intake pipe 17, and a throttle valve 20 driven by a step motor 19 is arranged within the air intake pipe 17. Further, a mass flow amount detecting device 21 for detecting a mass flow amount of a intake air is arranged within the air

intake pipe 17 disposed upward the throttle valve 20.

**[0021]** An automatic transmission 60 is connected to a crank shaft 69 serving as an output shaft of the engine main body 1. The automatic transmission 60 is provided with a torque converter 61 and a transmission 62, and an output shaft 71 of the transmission 62 is connected to a drive wheel of a vehicle via a differential gear (not shown).

**[0022]** The transmission 62 is of a known type which is provided with a planetary gear train and frictional elements (a brake, a clutch and the like), and is structured to switch an engaging state of the frictional elements by switching a control hydraulic pressure and perform a fixation and connection of each of the elements in the planetary gear train, thereby performing a shift change operation. The torque converter 61 is of a known type which is provided with a pump directly connected to the engine output shaft and a turbine driven by a fluid discharged by the pump, and an output shaft of the turbine (hereinafter, referred to as „a converter output shaft“) is directly connected to an input shaft of the transmission 62. The torque converter 61 has a known torque amplification function of amplifying the torque input from the engine output shaft so as to output to the converter output shaft. The automatic transmission 60 is provided with a converter output shaft rotating speed sensor 63 which outputs a pulse signal having a frequency corresponding to a rotating speed of the converter output shaft (that is, a rotating speed of the input shaft of the transmission 62), and a transmission output shaft rotating speed sensor 64 which outputs a pulse signal having a frequency corresponding to a rotating speed of the output shaft of the transmission 62, respectively.

**[0023]** A lockup mechanism 73 is provided within the torque converter 61. That is, the torque converter 61 is connected to the crank shaft 69 so as to be rotated together with the crank shaft 69, however, is provided with a pump cover 74, a pump impeller 75 supported by the pump cover 74, a turbine runner 77 mounted to an input shaft 76 of an automatic transmission 70 and a stator, and a rotational movement of the crank shaft 69 is transmitted to the input shaft 76 via the pump cover 74, the pump impeller 75 and the turbine runner 77.

**[0024]** The lockup mechanism 73 is mounted to the input shaft 76 in such a manner as to freely move in an axial direction thereof, and is provided with a lockup clutch plate 78 which rotates together with the input shaft 76. In general, that is, when the lockup is in on-state, a pressurized oil is supplied within a room 79 between the lockup clutch plate 78 and the pump cover 74 via an oil passage within the input shaft 76, and next the pressurized oil flown out from the room 79 is discharged via the oil passage within the input shaft 76 after being fed within a room 80 around the pump impeller 75 and the turbine runner 77. At this time, since a pressure difference between the rooms 79 and 80 is disposed in both sides of the lockup clutch plate 78 is

hardly generated, the lockup clutch plate 78 is apart from an inner wall surface of the pump cover 74, so that at this time, a rotational force of the crank shaft 69 is transmitted to the input shaft 76 via the pump cover 74, the pump impeller 75 and the turbine runner 77.

**[0025]** When the lockup should be turned on, the pressurized oil is supplied within the room 80 via the oil passage within the input shaft 76, and the oil within the room 79 is discharged via the oil passage within the input shaft 76. At this time, the pressure within the room 80 becomes higher than the pressure within the room 79 and the lockup clutch plate 78 is press contacted onto the inner peripheral surface of the pump cover 74, so that the crank shaft 69 and the input shaft 76 are in a directly connected state in which they are rotated at a constant speed. A control of supplying an oil within the rooms 79 and 80, that is, an on/off control of the lockup mechanism 73 is controlled by a control valve provided within the automatic transmission 70, and the control valve is controlled on the basis of an output signal of an electronic control unit 40. Further, a large number of clutches for performing a shift change operation are provided within the automatic transmission 70, and these clutches are controlled on the basis of the output signal of the electronic control unit 40.

**[0026]** On the contrary, the exhaust port 10 is connected to an inlet portion of an exhaust turbine 23 of the exhaust turbo charger 15 via an exhaust manifold 22, and an outlet portion of the exhaust turbine 23 is connected to a catalytic converter 26 containing a catalyst 25 having an oxidation function therein via an exhaust pipe 24. An air fuel ratio sensor 27 is arranged within the exhaust manifold 21.

**[0027]** An exhaust pipe 28 connected to an outlet portion of the catalyst converter 26 and the air intake pipe 17 disposed downstream the throttle valve 20 are connected to each other via an exhaust gas recirculation (hereinafter, referred to as an EGR) passage 29, and an EGR control valve 31 driven by a step motor 30 is arranged within the EGR passage 29. Further, an inter cooler 32 for cooling an EGR gas flowing within the EGR passage 29 is arranged within the EGR passage 29. In the embodiment shown in Fig. 1, an engine cooling water is introduced into the inter cooler 32, and the EGR gas is cooled by the engine cooling water.

**[0028]** On the contrary, the fuel injection valve 6 is connected to a fuel reservoir, so-called a common rail 34 via a fuel supply pipe 33. A fuel is supplied into the common rail 34 from an electrically controlled type fuel pump 35 in which a discharge amount is variable, and the fuel supplied into the common rail 34 is supplied to the fuel injection valve 6 via each of the fuel supply pipe 33. A fuel pressure sensor 36 for detecting a fuel pressure within the common rail 34 is mounted to the common rail 34, so that a discharge amount of the fuel pump 35 can be controlled. Therefore, the fuel pressure within the common rail 34 becomes a target fuel pressure on the basis of an output signal of the fuel pressure sensor

36.

**[0029]** An electronic control unit 40 is constituted by a digital computer, and is provided with a read only memory (ROM) 42, a random access memory (RAM) 43, a microprocessor (CPU) 44, an input port 45 and an output port 46 mutually connected by a two way bus 41. An output signal of the mass flow amount detecting device 21 is input to the input port 45 via a corresponding AD converter 47, and output signals of the air fuel ratio sensor 27 and the fuel pressure sensor 36 are also input to the input port 45 via the corresponding AD converter 47, respectively. Pulse signals from the converter output shaft rotating speed sensor 63 and the transmission output shaft rotating speed sensor 64 are respectively input to the input port 45. A load sensor 51 for generating an output voltage in proportion to a depression amount L of an accelerator pedal 50 is connected to the accelerator pedal 50, and an output voltage of the load sensor 51 is input to the input port 45 via the corresponding AD converter 47. Further, a crank angle sensor 52 for generating an output pulse every time when the crank shaft rotates, for example, at 30 degrees is connected to the input port 45. The engine speed is calculated on the basis of the output value of the crank angle sensor 52. On the contrary, the output port 46 is connected to the fuel injection valve 6, the throttle valve controlling step motor 19, the EGR control valve controlling step motor 30 and the fuel pump 35 via the corresponding drive circuit 48.

**[0030]** Fig. 2 expresses an experimental embodiment which shows a change of an output torque and a change of a discharge amount of a smoke, HC, CO and NOx when changing an air fuel ratio A/F (an axis of abscissas in Fig. 2) by changing an opening degree of the throttle valve 20 and the EGR rate at a time of operating the engine at a low load. As is understood from Fig. 2, in this experimental embodiment, the smaller the air fuel ratio A/F becomes, the greater the EGR rate is, and the EGR rate becomes equal to or more than 65 % when the air fuel ratio is equal to or less than a stoichiometric air fuel ratio (14.6).

**[0031]** As shown in Fig. 2, in the case of making the air fuel ratio A/F smaller by increasing the EGR rate, the generation amount of the smoke starts increasing when the EGR rate becomes near 40 % and the air fuel ratio A/F becomes about 30 %. Next, when further increasing the EGR rate and making the air fuel ratio A/F smaller, the generation amount of the smoke is suddenly increased to the peak. Next, when further increasing the EGR rate and making the air fuel ratio A/F smaller, the smoke is suddenly reduced at this time, and when the air fuel ratio A/F becomes near 15.0 with the EGR rate set to a value equal to or more than 65 %, the amount of the smoke is substantially 0. That is, the soot is hardly generated. At this time, the output torque of the engine is slightly reduced and the generation amount of NOx is significantly reduced. On the contrary, the generation amount of HC and CO starts increasing at this time.

**[0032]** Fig. 3A shows a change of a combustion pressure within the combustion chamber 5 when the air fuel ratio A/F is near 18 and the generation amount of the smoke is the largest, and Fig. 3B shows a change of a combustion pressure within the combustion chamber 5 when the air fuel ratio A/F is near 13 and the generation amount of the smoke is substantially 0. As is understood from comparison between Figs. 3A and 3B, the combustion pressure in the case as shown in Fig. 3B in which the generation amount of the smoke is substantially 0 is lower than that shown in Fig. 3A in which the generation amount of the smoke is large.

**[0033]** The following results can be introduced from the experimental results shown in Figs. 2 and 3. That is, at first, when the air fuel ratio A/F is equal to or less than 15.0 and the generation amount of the smoke is substantially 0, the generation amount of NOx is significantly reduced as shown in Fig. 2. The reduction of the generation amount of NOx means the reduction of the combustion temperature within the combustion chamber 5, so that it is said that the combustion temperature within the combustion chamber 5 becomes low when the soot is hardly generated. The same matter can be applied to the case as shown in Fig. 3. That is, in a state shown in Fig. 3B where the soot is hardly generated, the combustion pressure becomes low, so that the combustion temperature within the combustion chamber 5 becomes low.

**[0034]** Secondly, when the generation amount of the smoke, that is, the generation amount of the soot becomes substantially 0, the discharge amount of HC and CO is increased as shown in Fig. 2. This means that the hydrocarbon is discharged without growing to the soot. That is, in a straight chain hydrocarbon or an aromatic hydrocarbon as shown in Fig. 4 and contained in the fuel, when the temperature is increased in an oxygen poor state, a precursor of the soot is formed due to a thermal decomposition, and the soot containing a solid mainly formed by an aggregation of carbon atoms is produced. In this case, the process of actually producing the soot is complex and it is indefinite what aspect the precursor of the soot forms, however, in any event, the hydrocarbon as shown in Fig. 4 grows to the soot via the precursor of the soot. Accordingly, as mentioned above, when the generation amount of the soot becomes substantially 0, the discharge amount of HC and CO is increased as shown in Fig. 2, however, HC at this time corresponds to the precursor of the soot or the hydrocarbon in the preceding state.

**[0035]** Putting in order the considerations on the basis of the experimental results as shown in Figs. 2 and 3, when the combustion temperature within the combustion chamber 5 is low, the generation amount of the soot becomes substantially 0, so that the precursor of the soot or the hydrocarbon in the preceding state is discharged from the combustion chamber 5. As a result of further performing the experiments and researches with respect to the matter in detail, it becomes clear that

the growing process of the soot stops on the way, that is, no soot is generated, in the case where the temperature of the fuel and the surrounding gas within the combustion chamber 5 is equal to or less than a certain temperature, and that the soot is generated when the temperature of the fuel and the surrounding gas within the combustion chamber 5 is equal to or more than a certain temperature.

**[0036]** In this case, since the temperature of the fuel and the surrounding gas when the growing process of the hydrocarbon stops in a state of the precursor of the soot, that is, the certain temperature as mentioned above is changed due to various reasons, for example, a kind of the fuel, a compression ratio of the air fuel ratio and the like, it is not exactly said what degree the temperature is. However, the certain temperature has a great relation to the generation amount of NOx, so that the certain temperature can be defined from the generation amount of NOx at a certain level. That is, as the EGR rate is increased, the temperature of the fuel and the surrounding gas at a time of combustion is reduced, so that the generation amount of NOx is reduced. At this time, the soot is hardly generated when the generation amount of NOx becomes near 10 p.p.m. or less. Accordingly, the certain temperature mentioned above substantially coincides with the temperature when the generation amount of NOx becomes near 10 p.p.m. or less.

**[0037]** Once the soot is generated, the soot can not be purified in accordance with the after treatment using the catalyst having an oxidation function. On the contrary, the precursor of the soot or the hydrocarbon in the preceding state can be easily purified in accordance with the after treatment using the catalyst having an oxidation function. As mentioned above, considering the after treatment by the catalyst having an oxidation function, there is a significantly great difference between the case of discharging the hydrocarbon from the combustion chamber 5 as the precursor of the soot or the preceding state, and the case of discharging the hydrocarbon from the combustion chamber 5 as the soot. The new combustion system employed in the present invention is mainly structured so as to discharge the hydrocarbon from the combustion chamber 5 as the precursor of the soot or the preceding state without generating the soot within the combustion chamber 5 and oxidize the hydrocarbon by the catalyst having the oxidation function.

**[0038]** Further, in order to stop the growth of the hydrocarbon in the state prior to the generation of the soot, it is necessary to restrict the temperature of the fuel and the surrounding gas at a time of the combustion within the combustion chamber 5 to a temperature lower than the temperature at which the soot is generated. In this case, it is clearly understood that an endothermic effect of the gas around the fuel when the fuel is burned affects restriction of the temperature of the fuel and the surrounding gas at a significantly great amount.

**[0039]** That is, when only an air exists around the fuel, the evaporated fuel immediately reacts with an oxygen in the air so as to be burned. In this case, the temperature of the air apart from the fuel is not increased so much, only the temperature around the fuel becomes locally increased in a significant manner. That is, at this time, the air apart from the fuel hardly perform an endothermic effect of the combustion heat in the fuel. In this case, since the combustion temperature becomes locally high in a significant manner, an unburned hydrocarbon to which the combustion heat is applied generates the soot.

**[0040]** On the contrary, in the case where the fuel exists in the mixed gas including a large amount of inert gas and a small amount of air, the condition becomes slightly different. In this case, the evaporated fuel diffuses to the periphery and reacts with an oxygen contained in the inert gas in a mixed manner so as to burn. In this case, since the combustion heat is absorbed into the peripheral inert gas, the combustion temperature is not increased so much. That is, it is possible to restrict the combustion temperature to a low level. That is, the existing inert gas plays an important part to restrict the combustion temperature to a low level due to the endothermic effect of the inert gas.

**[0041]** In this case, in order to restrict the temperature of the fuel and the surrounding gas to the temperature lower than the temperature at which the soot is generated, the inert gas amount sufficient for absorbing sufficient heat is required. Accordingly, when the fuel amount increases, the required inert gas amount increases accordingly. Here, in this case, the greater the specific heat of the inert gas is, the stronger the endothermic effect is. Therefore, a gas having a great specific heat is preferable for the inert gas. In view of this, since  $\text{CO}_2$  and the EGR gas have the relatively higher specific heat, it is said that employing the EGR gas as the inert gas is preferable.

**[0042]** Fig. 5 shows a relation between the EGR rate and the smoke when using the EGR gas as the inert gas and changing the cooling degree of the EGR gas. That is, in Fig. 5, a curve A shows the case where the EGR gas is strongly cooled so as to maintain the EGR gas temperature to substantially 90°, a curve B shows the case where the EGR gas is cooled by a compact cooling apparatus and a curve C shows the case where the EGR gas is not forcibly cooled.

**[0043]** As shown by the curve A in Fig. 5, in the case where the EGR gas is strongly cooled, the generation amount of the soot becomes peak when the EGR rate is slightly lower than 50%, and in this case, the soot is hardly generated when setting the EGR rate to the level equal to or more than substantially 55%. On the contrary, as shown by the curve B in Fig. 5, in the case where the EGR gas is slightly cooled, the generation amount of the soot becomes peak when the EGR rate is slightly higher than 50%, and in this case, the soot is hardly generated when setting the EGR rate to the level

equal to or more than substantially 65%.

**[0044]** Further, as shown by the curve C in Fig. 5, in the case where the EGR gas is not forcibly cooled, the generation amount of the soot becomes peak when the EGR rate is near 55%, and in this case, the soot is hardly generated when setting the EGR rate to the level equal to or more than substantially 70%.

**[0045]** In this case, Fig. 5 shows a generation amount of the smoke when the engine load is comparatively high, when the engine load becomes small, the EGR rate at which the generation amount of the soot becomes peak is slightly reduced, and a lower limit of the EGR rate at which the soot is hardly generated is slightly reduced. As mentioned above, the lower limit of the EGR rate at which the soot is hardly generated changes in correspondence to a cooling degree of the EGR gas and the engine load.

**[0046]** Fig. 6 shows a mixed gas amount of the EGR gas and an air necessary for making the temperature of the fuel and the surrounding gas at a time of combustion in the case of employing the EGR gas for the inert gas lower than the temperature at which the soot is generated, a rate of the air in the mixed gas, and a rate of the EGR gas in the mixed gas. Here, in Fig. 6, an axis of ordinates shows a total intake gas amount admitted within the combustion chamber 5, and a chain line Y shows a total intake gas amount capable of being admitted within the combustion chamber 5 when supercharging is not performed. Further, an axis of abscissas shows a required load.

**[0047]** Fig. 6 shows a mixed gas amount of the EGR gas and an air necessary for making the temperature of the fuel and the surrounding gas at a time of combustion in the case of employing the EGR gas for the inert gas lower than the temperature at which the soot is generated, a rate of the air in the mixed gas, and a rate of the EGR gas in the mixed gas. Referring to Fig. 6, an axis of ordinates shows a total intake gas amount admitted within the combustion chamber 5, and a single dot chain line Y shows a total intake gas amount capable of being admitted within the combustion chamber 5 when supercharging is not performed. Further, an axis of abscissas shows a required load.

**[0048]** With reference to Fig. 6, the rate of the air, that is, the air amount in the mixed gas shows an amount of the air necessary for completely burning the injected fuel. That is, in the case shown in Fig. 6, a ratio between the air amount and the injection fuel amount corresponds to a stoichiometric air fuel ratio. On the contrary, in Fig. 6, the rate of the EGR gas, that is, the EGR gas amount in the mixed gas shows the EGR gas amount necessary at the lowest for setting the temperature of the fuel and the surrounding gas to the temperature lower than the temperature at which the soot is formed when the injected fuel is burned. The EGR gas amount is equal to or more than 55% of the EGR rate, and that of the embodiment shown in Fig. 6 is equal to or more than 70%. That is, when setting the total intake

gas amount admitted into the combustion chamber 5 to a solid line X in Fig. 6 and setting the rate between the air amount and the EGR gas amount among the total intake gas amount X to a rate shown in Fig. 6, the temperature of the fuel and the surrounding gas becomes lower than the temperature at which the soot is generated, and accordingly the soot is not completely generated. Further, the generation amount of NOx at this time is about 10 p.p.m. or less, so that the generation amount of NOx is significantly small.

**[0049]** Since the heat generated when the fuel is burned is increased as the fuel injection amount is increased, in order to maintain the temperature of the fuel and the surrounding gas to the temperature lower than the temperature at which the soot is generated, it is necessary to increase the absorption amount of the heat due to the EGR gas. Accordingly, as shown in Fig. 6, the EGR gas amount should be increased in accordance that the injection fuel amount is increased. That is, the EGR gas amount should be increased as a required load becomes high.

**[0050]** Here, in the case where the supercharging is not performed, an upper limit of the amount X of the total intake gas admitted into the combustion chamber 5 is Y, so that in Fig. 6, in the area having a required load larger than  $L_0$ , the air fuel ratio can not be maintained to the stoichiometric air fuel ratio unless the EGR gas rate is reduced in accordance that the required load becomes greater. In other words, in the case where it is intended to maintain the air fuel ratio to the stoichiometric air fuel ratio in the area having the desired load larger than  $L_0$  when the supercharging is not performed, the EGR rate is reduced in accordance that the required load becomes high, and accordingly, in the area having the desired load larger than  $L_0$ , it is impossible to maintain the temperature of the fuel and the surrounding gas to the temperature lower than the temperature at which the soot is produced.

**[0051]** However, as shown in Fig. 1, when recirculating the EGR gas into the inlet side of the supercharger, that is, the air intake pipe 17 of the exhaust turbo charger 15 via the EGR passage 29, in the area having the required load larger than  $L_0$ , it is possible to maintain the EGR rate to the level equal to or more than 55 %, for example, 70 %, so that it is possible to maintain the temperature of the fuel and the surrounding gas to the temperature lower than the temperature at which the soot is produced. That is, when recirculating the EGR gas so that the EGR rate within the air intake pipe 17 becomes, for example, 70 %, the EGR rate of the intake gas at the pressure increased by the compressor 16 of the exhaust turbo charger 15 also becomes 70 %, so that it is possible to maintain the temperature of the fuel and the surrounding gas to the temperature at which the soot is produced as long as the compressor 16 can increase the pressure. Accordingly, it is possible to expand an operation range of the engine which can produce the low temperature combustion.

**[0052]** Here, in this case, when setting the EGR rate to the level equal to or more than 55 % in the area having the required load larger than  $L_0$ , the EGR control valve 31 is fully opened and the throttle valve 20 is slightly closed.

**[0053]** As mentioned above, Fig. 6 shows the case where the fuel is burned under the stoichiometric air fuel ratio, however, even when setting the air amount to the level less than the air amount shown in Fig. 6, that is, setting the air fuel ratio to rich, it is possible to restrict the generation amount of NOx near to 10 p.p.m. or less while restricting the generation of the soot, and further, even when setting the air amount to the level more than the air amount shown in Fig. 6, that is, setting an average value of the air fuel ratio to the lean value such as 17 to 18, it is possible to restrict the generation amount of NOx near to 10 p.p.m. or less while restricting the generation of the soot.

**[0054]** That is, when the air fuel ratio is made rich, the fuel becomes excessive, however, since the combustion temperature is restricted to the low temperature, the excessive fuel does not grow to the soot, so that the soot is not produced. Further, at this time, a significantly small amount of NOx is only produced. On the contrary, when the average air fuel ratio is lean, or even when the air fuel ratio is the stoichiometric air fuel ratio, a small amount of soot is produced if the combustion temperature becomes high. However, in accordance with the present invention, since the combustion temperature is restricted to the low temperature, the soot is not produced at all. Further, only a significantly small amount of NOx is generated.

**[0055]** As mentioned above, when the low temperature combustion is performed, the soot is not produced irrespective of the air fuel ratio, that is, whether or not the air fuel ratio is rich, stoichiometric, or lean. As a result, the generation amount of NOx is significantly small. Accordingly, in view of the improvement of a specific fuel consumption, it is said that it is preferable to make the average air fuel ratio lean.

**[0056]** In this case, it is limited to an engine at the middle or low load operation at which the amount of heat generated by the combustion is relatively small to restrict the temperature of the fuel and the surrounding gas at a time of combustion within the combustion chamber to the level equal to or less than the temperature at which the growth of the hydrocarbon stops on the way. Accordingly, in the embodiment in accordance with the present invention, at a time of operating the engine at the middle or low load, the temperature of the fuel and the surrounding gas at a time of combustion is restricted to the temperature equal to or less than the temperature at which the growth of the hydrocarbon stops on the way so as to perform the first combustion, that is, the low temperature combustion, and at a time of operating the engine at high load, the second combustion, that is, the conventionally performed combustion is performed. In this case, the first combustion, that is, the

low temperature combustion means the combustion in which the amount of the inert gas within the combustion chamber is larger than the amount of the inert gas at which the generation amount of the soot becomes peak and the soot is hardly generated, as is apparent from the explanation as mentioned above. The second combustion, that is, the conventionally performed combustion means the combustion in which the amount of the inert gas within the combustion chamber is smaller than the amount of the inert gas at which the generation amount of the soot becomes peak.

**[0057]** Fig. 7 shows a first operation area I in which the first combustion, that is, the low temperature combustion is performed and a second operation area II in which the second combustion, that is, the combustion in accordance with the conventional combustion method is performed. In this case, in Fig. 7, an axis of ordinates L indicates a depression amount of the acceleration pedal 50, that is, a required load, and an axis of abscissas N indicates an engine speed. Further, in Fig. 7, X(N) shows a first boundary between the first operation area I and the second operation area II, and Y(N) shows a second boundary between the first operation area I and the second operation area II. A change of the operation area from the first operation area I to the second operation area II is judged on the basis of the first boundary X(N), and a change of the operation area from the second operation area II to the first operation area I is judged on the basis of the second boundary Y(N).

**[0058]** That is, when the required load L exceeds the first boundary X(N) corresponding to a function of the engine speed N when the operation state of the engine is in the first operation area I and the low temperature combustion is performed, it is judged that the operation area is moved to the second operation area II, so that the combustion in accordance with the conventional combustion method is performed. Next, when the required load L becomes lower than the second boundary Y(N) corresponding to a function of the engine speed N, it is judged that the operation area goes to the first operation area I, so that the low temperature combustion is performed again.

**[0059]** As mentioned above, two boundaries comprising the first boundary X(N) and the second boundary Y(N) closer to the lower load than the first boundary X(N) are provided for the following two reasons. The first reason is that since the combustion temperature is relatively high in a side of the high load in the second operation area II, the low temperature combustion can not be immediately performed even when the required load L becomes lower than the first boundary X(N) at this time. That is, because the low temperature combustion is started only when the required load L becomes significantly low, that is, lower than the second boundary Y(N). The second reason is that the hysteresis is provided with respect to the change in the operation area between the first operation area I and the second operation area II.

**[0060]** In this case, when the engine operation state exists in the first operation area I and the low temperature combustion is performed, the soot is hardly generated, and in place thereof, unburned hydrocarbon is discharged from the combustion chamber 5 as the precursor of the soot or the state prior thereto. At this time, the unburned hydrocarbon discharged from the combustion chamber 5 is well oxidized by the catalyst 25 having an oxidization function. As the catalyst 25, an oxidation catalyst, a three way catalyst or an NOx absorbent can be employed. The NOx absorbent has the function of absorbing NOx when the average air fuel ratio within the combustion chamber 5 is lean and discharging NOx when the average air fuel ratio within the combustion chamber 5 becomes rich.

**[0061]** The NOx absorbent is structured such that, for example, an alumina is set as a carrier and at least one selected from an alkaline metal such as a potassium K, a sodium Na, a lithium Li and a cesium Cs, an alkaline earth metal such as a barium Ba and a calcium Ca and a rare earth metal such as a lanthanum La and an yttrium Y, and a noble metal such as a platinum Pt are carried on the carrier.

**[0062]** As well as the oxidation catalyst, the three way catalyst and the NOx absorbent have the oxidation function. As mentioned above, the three way catalyst and the NOx absorbent can be used as the catalyst 25.

**[0063]** Fig. 8 shows the output of the air fuel ratio sensor 27. As shown in Fig. 8, an output current I of the air fuel ratio sensor 27 is changed in accordance with the air fuel ratio A/F. Accordingly, the air fuel ratio can be derived from the output current I of the air fuel ratio sensor 27.

**[0064]** Next, the outline of the operation control in the first operation area I and the second operation area II is described with reference to Fig. 9.

**[0065]** Fig. 9 shows an opening degree of the throttle valve 20 with respect to the required load L, an opening degree of the EGR control valve 31, an EGR rate, an air fuel ratio, an injection timing and an injection amount. As shown in Fig. 9, in the first operation area I having a low required load L, the opening degree of the throttle valve 20 is gradually increased to about two-third of the opening degree from a nearly full close state as the required load L is increased, and the opening degree of the EGR control valve 31 is gradually increased to the full open state from a nearly full closed state as the required load L is increased. Further, in the embodiment shown in Fig. 9, the EGR rate is set to substantially 70 % in the first operation area I, and the air fuel ratio is set to the lean air fuel ratio which is slightly leaner.

**[0066]** In other words, in the first operation area I, the opening degree of the throttle valve 20 and the opening degree of the EGR control valve 31 are controlled so that the EGR rate becomes substantially 70 % and the air fuel ratio becomes lean which is slightly leaner. Further, in the first operation area I, a fuel injection is performed prior to a compression top dead center

TDC. In this case, an injection start timing S is delayed in accordance that the required load L becomes high, and an injection end timing E is also delayed in accordance that the injection start timing S is delayed.

**[0067]** Further, at a time of an idling operation, the throttle valve 20 is closed near to the full closed state, and at this time, the EGR control valve 31 is also closed near to the full closed state. When closing the throttle valve 20 near to the full closed state, a pressure within the combustion chamber 5 at the beginning of the compression becomes low, so that the compression pressure becomes small. When the compression pressure becomes small, a compression work by the piston 4 is reduced. Accordingly, a vibration of the engine main body 1 is restricted. That is, at a time of the idling operation, in order to restrict the vibration of the engine main body 1, the throttle valve 20 is closed near to the fully closed state.

**[0068]** On the contrary, the operation area of the engine changes from the first operation area I to the second operation area II, the opening degree of the throttle valve 20 is increased stepwise from about two-third of the opening degree to the fully open direction. At this time, in the embodiment shown in Fig. 9, the EGR rate is reduced stepwise from substantially 70 % to 40 % or less and the air fuel ratio is increased stepwise. That is, since the EGR rate flies over the EGR rate range (Fig. 5) in which a lot of smoke is generated, a lot of smoke is not generated when the operation area of the engine changes from the first operation area I to the second operation area II.

**[0069]** In the second operation area II, the conventionally performed combustion is performed. In this second operation area II, the throttle valve 20 is kept in the fully open state except a portion thereof, and the opening degree of the EGR control valve 31 is gradually reduced as the required load L becomes high. Further, in this operation area II, the EGR rate becomes low as the required load L becomes high, and the air fuel ratio becomes small as the required load L becomes high. However, the air fuel ratio is set to lean even when the required load L becomes high. Further, in the second operation area II, the injection start timing S is set near the compression top dead center TDC.

**[0070]** Fig. 10A shows a target air fuel ratio A/F in the first operation area I. In Fig. 10A, curves indicated by  $A/F = 15.5$ ,  $A/F = 16$ ,  $A/F = 17$  and  $A/F = 18$  respectively show states having air fuel ratios 15.5, 16, 17 and 18, each of the air fuel ratios between the curves is defined in accordance with a proportional allotment. As shown in Fig. 10A, the air fuel ratio becomes lean in the first operation area I, and further, in the first operation area I, the air fuel ratio A/F is made lean in accordance that the required load L becomes low.

**[0071]** That is, the heat generated by the combustion is reduced as the required load L becomes low. Accordingly, the low temperature combustion can be performed even when lowering the EGR rate as the

required load L becomes low. When lowering the EGR rate, the air fuel ratio becomes large, so that as shown in Fig. 10A, the target air fuel ratio A/F is made large as the required load L becomes low. As the target air fuel ratio A/F is increased, the specific fuel consumption is improved, so that in order to make the air fuel ratio as lean as possible, in accordance with the embodiment of the present invention, the target air fuel ratio A/F is made large as the required load L becomes low.

**[0072]** In this case, the target air fuel ratio A/F shown in Fig. 10A is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 10B. Further, a target opening degree ST of the throttle valve 20 necessary for setting the air fuel ratio to the target air fuel ratio A/F as shown in Fig. 10A is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 11A, and a target opening degree SE of the EGR control valve 31 necessary for setting the air fuel ratio to the target air fuel ratio A/F shown in Fig. 10A is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 11B.

**[0073]** Fig. 12A shows a target air fuel ratio A/F when the second combustion, that is, the combustion in accordance with the conventional combustion method is performed. In this case, in Fig. 12A, curves indicated by  $A/F = 24$ ,  $A/F = 35$ ,  $A/F = 45$  and  $A/F = 60$  respectively show states having target air fuel ratios 24, 35, 45 and 60. The target air fuel ratio A/F shown in Fig. 12A is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 12B. Further, a target opening degree ST of the throttle valve 20 necessary for setting the air fuel ratio to the target air fuel ratio A/F as shown in Fig. 12A is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 13A, and a target opening degree SE of the EGR control valve 31 necessary for setting the air fuel ratio to the target air fuel ratio A/F is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 13B.

**[0074]** Further, the fuel injection amount Q when the second combustion is performed is calculated on the basis of the required load L and the engine speed N. The fuel injection amount Q is preliminarily stored within the ROM 42 as a function of the required load L and the engine speed N in the form of a map as shown in Fig. 14.

**[0075]** Next, the operation control in accordance with the present embodiment will be described below with reference to Figs. 15 and 16. With reference to Figs. 15 and 16, at first, in step 100, it is judged whether or not a flag I indicating that the operation area of the engine is in the first operation area I. When the flag I is set, that is, the operation area of the engine is in the first

operation area I, the operation control goes to step 101 where it is judged whether or not the required load  $L$  becomes greater than the first boundary  $X1(N)$ . When a relation  $L > X1(N)$  is established, the operation control goes to step 105 where the low temperature combustion is performed.

**[0076]** In step 101, when it is judged that the relation  $L > X(N)$  is established, the operation control goes to step 102 where a flag  $I$  is reset, and next, the operation control goes to step 112 where the second combustion is performed.

**[0077]** On the contrary, in step 100, when it is judged that the flag  $I$  indicating that the operating state of the engine is the first operation area I is not set, that is, when the operation state of the engine is in the second operation area II, the operation control goes to step 103 where it is judged whether or not the required load  $L$  becomes lower than the second boundary  $Y(N)$ . When the relation  $L < Y(N)$  is established, the recirculation goes to step 112 where the second combustion is performed at a lean air fuel ratio.

**[0078]** On the contrary, in step 103, when it is judged that the relation  $L < Y(N)$  is established, the process goes to step 104 where the flag  $I$  is set, and next the process goes to step 105 where the low temperature combustion is performed.

**[0079]** In step 105, the target opening degree  $ST$  of the throttle valve 20 is calculated from a map shown in Fig. 11 A, and the opening degree of the throttle valve 20 is set to the target opening degree  $ST$ . Next, in step 106, the target opening degree  $SE$  of the EGR control valve 31 is calculated from a map shown in Fig. 11 B, and the opening degree of the EGR control valve 31 is set to the target opening degree  $SE$ . Next, in step 107, a mass flow amount of the intake air detected by the mass flow amount detecting device 21 (hereinafter, simply refer to as the intake air amount)  $G_a$  is taken in, and next in step 108, the target air fuel ratio  $A/F$  is calculated on the basis of the map shown in Fig. 10B. Next, in step 109, the fuel injection amount  $Q$  required for setting the air fuel ratio to the target air fuel ratio  $A/F$  is calculated on the basis of the intake air amount  $G_a$  and the target air fuel ratio  $A/F$ .

**[0080]** Next, in step 110, it is judged whether or not the automatic transmission 60 is under operation. Since the torque generated in the engine is changed during the operation of the automatic transmission 60, it is desired to soften the shock. Accordingly, when it is judged YES in step 110, the process goes to step 111 where it is intended to soften the shock on the basis of the change of the torque generated in the engine. Hereinafter, the description will be given with respect to the idea for softening the shock on the basis of the change of the torque generated in the engine.

**[0081]** Fig. 17 is a graph which shows the relation between the air fuel ratio and the torque generated in the engine. In Fig. 17, an axis of abscissas indicates an air fuel ratio  $A/F$  and an axis of ordinates indicates a

torque  $T$  generated in the engine. As shown in Fig. 17, the second combustion (the combustion in accordance with the conventional combustion method) is performed in an area where the air amount is sufficiently excessive and the air fuel ratio is relatively lean. Accordingly, in the case where the second combustion is performed, when the fuel injection amount is corrected to be reduced, that is, the air fuel ratio is increased from  $A/F3$  to  $A/F4$  (becomes leaner) together with the correction for reducing the fuel injection amount, the torque generated in the engine is reduced by  $T2$  as the fuel amount serving for the combustion is reduced. On the contrary, the low temperature combustion (the first combustion) is performed in an area where the air amount is likely to be insufficient and the air fuel ratio is richer than that of the case in the second combustion. Accordingly, in the case where the low temperature combustion is performed, when the fuel injection amount is corrected to be increased, that is, the air fuel ratio is reduced from  $A/F1$  to  $A/F2$  (becomes richer) together with the correction for increasing the fuel injection amount, the combustion is deteriorated and the torque generated in the engine is reduced by  $T1$ .

**[0082]** Returning to the explanation of Figs. 15 and 16, the fuel injection amount is corrected to be increased in step 111 on the basis of the idea as mentioned above ( $Q \leftarrow Q + Q1$ ). In accordance with step 111, when the low temperature combustion is performed and the shift change is performed by the automatic transmission 60, the fuel injection amount is corrected to be increased and the air fuel ratio is reduced, such that the shock of the torque change due to the automatic transmission 60 is softened. On the contrary, when it is judged NO in step 110, it is not required to soften the shock of the torque change due to the automatic transmission. Accordingly, the present routine is finished without correcting the fuel injection amount to increase. In this case, in accordance with another embodiment, in place of correcting the fuel injection amount to increase in step 111, it is possible to reduce the intake air amount by reducing the opening degree of the throttle valve 20.

**[0083]** As mentioned above, in the case where the low temperature combustion is performed, when the required load  $L$  or the engine speed  $N$  is changed, the opening degree of the throttle valve 20 and the opening degree of the EGR control valve 31 is immediately made to be coincident with the target opening degree  $ST$  and  $SE$  corresponding to the required load  $L$  and the engine speed  $N$ . Accordingly, for example, when the required load  $L$  is increased, the amount of the air within the combustion chamber 5 is immediately increased, thus immediately increasing the torque generated in the engine.

**[0084]** On the contrary, when the opening degree of the throttle valve 20 or the opening degree of the EGR control valve 31 is changed and the intake air amount is changed, the change of the intake air amount  $G_a$  is

detected by the mass flow amount detecting device 21, and the fuel injection amount  $Q$  is controlled on the basis of the detected intake air amount  $G_a$ . That is, the fuel injection amount  $Q$  is changed after the intake air amount  $G_a$  is actually changed.

**[0085]** In step 112 where the second combustion is performed, the target fuel injection amount  $Q$  is calculated on the basis of the map shown in Fig. 14, and the fuel injection amount is set to the target fuel injection amount  $Q$ . Next, in step 113, in the same manner as step 110, it is judged whether or not the shift change by the automatic transmission 60 is going to be performed. Since the torque generated in the engine is changed during the shift change of the automatic transmission 60, it is desired to soften the shock. Accordingly, when it is judged YES in step 113, the process goes to step 114 where the shock on the basis of the change of the torque generated in the engine is aimed to be softened. In step 114, on the basis of the idea as mentioned above, the fuel injection amount is corrected to be reduced ( $Q \leftarrow Q - Q_2$ ). In accordance with step 114, when the second combustion is performed and the shift change is performed by the automatic transmission 60, the fuel injection amount is corrected to be reduced and the air fuel ratio is increased, such that the shock of the torque change due to the automatic transmission 60 is softened. On the contrary, when it is judged NO in step 113, it is not required to soften the shock of the torque change due to the automatic transmission. Therefore, the process goes to step 115 without correcting to reduce the fuel injection amount.

**[0086]** Next, in step 115, the target opening degree ST of the throttle valve 20 is calculated on the basis of the map shown in Fig. 13A. Next, in step 116, the target opening degree SE of the EGR control valve 31 is calculated on the basis of the map shown in Fig. 13B, and the opening degree of the EGR control valve 31 is set to the target opening degree SE.

**[0087]** Next, in step 117, the intake air amount  $G_a$  detected by the mass flow amount detecting device 21 is taken in. Next, in step 118, an actual air fuel ratio  $(A/F)_R$  is calculated on the basis of the fuel injection amount  $Q$  and the intake air amount  $G_a$ . Next, in step 119, the target air fuel ratio  $A/F$  is calculated on the basis of the map shown in Fig. 12B. Next, in step 120, it is judged whether or not the actual air fuel ratio  $(A/F)_R$  is larger than the target air fuel ratio  $A/F$ . When the relation  $(A/F)_R > A/F$  is established, the process goes to step 121, a correcting value ST of the throttle opening degree is reduced at a fixed value, and next the process goes to step 123. On the contrary, when the relation  $(A/F)_R < A/F$  is established, the process goes to step 122, the correcting value ST is increased at the fixed value, and next the process goes to step 123. In step 123, a final target opening degree ST is calculated by adding the correcting value ST to the target opening degree ST of the throttle valve 20, and the opening degree of the throttle valve 20 is set to the final target opening degree

ST. That is, the opening degree of the throttle valve 20 is controlled such that the actual air fuel ratio  $(A/F)_R$  becomes the target air fuel ratio  $A/F$ .

**[0088]** As mentioned above, in the case where the second combustion is performed, when the required load  $L$  or the engine speed  $N$  is changed, the fuel injection amount is immediately made to be coincident with the target fuel injection amount  $Q$  corresponding to the required load  $L$  and the engine speed  $N$ . For example, when the required load  $L$  is increased, the fuel injection amount is immediately increased, thus immediately increasing the torque generated in the engine.

**[0089]** On the contrary, when the fuel injection amount  $Q$  is increased and the air fuel ratio is shifted from the target air fuel ratio  $A/F$ , the opening degree of the throttle valve 20 is controlled so that the air fuel ratio becomes the target air fuel ratio  $A/F$ . That is, the air fuel ratio is changed after the fuel injection amount  $Q$  is changed.

**[0090]** In the embodiments as mentioned above, the fuel injection amount  $Q$  is controlled in accordance with an open loop when the low temperature combustion is performed, and the air fuel ratio is controlled by changing the opening degree of the throttle valve 20 when the second combustion is performed. However, it is possible to feedback control the fuel injection amount  $Q$  on the basis of the output signal of the air fuel ratio sensor 27 when the low temperature combustion is performed, and it is possible to control the air fuel ratio by changing the opening degree of the EGR control valve 31 when the second combustion is performed.

**[0091]** Hereinafter, a second embodiment of an internal combustion engine in accordance with the present invention will be described below. A structure of the present embodiment is substantially the same as the structure of the first embodiment as shown in Fig. 1.

**[0092]** Next, the description will be given with respect to the operation control in accordance with the present embodiment below with reference to Figs. 18 and 19. Since steps 100 to 109 are the same as those of the first embodiment, the explanation thereof will be omitted.

**[0093]** In step 110, it is judged whether or not the shift change is going to be performed by the automatic transmission 60. Since the torque generated in the engine is changed during the shift change of the automatic transmission 60, it is desired to soften the shock. Accordingly, when it is judged YES in step 110, the process goes to step 1800 where the shock on the basis of the change of the torque generated in the engine is aimed to be softened.

**[0094]** That is, in step 1800, the fuel injection timing is delayed in comparison with the case where the shift change is not performed. In this case, when the injection timing of the fuel supplied within the combustion chamber 5 is delayed, the combustion is deteriorated as the fuel supply is too late for the combustion, thus lowering the torque generated in the engine. Accordingly,

when the low temperature combustion is performed in step 1800 where the shift change is performed by the automatic transmission 60, the fuel injection timing is delayed, softening the shock of the torque change by the automatic transmission 60. On the contrary, when it is judged NO in step 110, it is not required to soften the shock of the torque change by the automatic transmission, the present routine is finished without delaying the fuel injection timing.

**[0095]** As mentioned above, in the case where the low temperature combustion is performed, when the required load L or the engine speed N is changed, the opening degree of the throttle valve 20 and the opening degree of the EGR control valve 31 is immediately made to be coincident with the target opening degree ST and SE corresponding to the required load L and the engine speed N. Accordingly, for example, when the required load L is increased, the amount of the air within the combustion chamber 5 is immediately increased, thus immediately increasing the torque generated in the engine.

**[0096]** On the contrary, when the opening degree of the throttle valve 20 or the opening degree of the EGR control valve 31 is changed and the intake air amount is changed, the change of the intake air amount Ga is detected by the mass flow amount detecting device 21, and the fuel injection amount Q is controlled on the basis of the detected intake air amount Ga. That is, the fuel injection amount Q is changed after the intake air amount Ga is actually changed.

**[0097]** Since the same operations are performed in steps 112 to 123 where the second combustion is performed as those of the first embodiment, the explanation thereof will be omitted.

**[0098]** Hereinafter, a third embodiment of an internal combustion engine in accordance with the present invention will be described below. The present embodiment has substantially the same structure as that of the first embodiment shown in Fig. 1.

**[0099]** Next, the description will be given with respect to the operation control in accordance with the present embodiment below with reference to Figs. 20 and 21. Since steps 100 to 106 are the same as those of the first embodiment, the explanation thereof will be omitted.

**[0100]** In step 2000, it is judged whether or not the shift change is going to be performed by the automatic transmission 60. Since the torque generated in the engine is changed during the shift change of the automatic transmission 60, it is desired to soften the shock. Accordingly, when it is judged YES in step 2000, the process goes to step 2001 where the shock on the basis of the change of the torque generated in the engine is intended to be softened.

**[0101]** That is, in step 2001, the target opening degree SE of the EGR control valve 31 is corrected to be increased, and the opening degree of the EGR control valve 31 is set to the target opening degree SE (SE

← SE + SE). In this case, as explained with reference to Fig. 17, since the low temperature combustion is performed under the state where the air amount tends to be insufficient, the air admitted via the throttle valve 20 is hard to be supplied to the combustion chamber 5 when the target opening degree SE of the EGR control valve 31 is corrected to be increased and the amount of the EGR gas is corrected to be increased in the case where the low temperature combustion is performed. Then the air within the combustion chamber 5 tends to be further insufficient. As a result, the combustion is deteriorated and the torque generated in the engine is lowered. Accordingly, when the low temperature combustion is performed in step 2001 and the shift change by the automatic transmission 60 is performed, the target opening degree SE of the EGR control valve 31 is corrected to be increased and the amount of the EGR gas is corrected to be increased, whereby the shock of the torque change due to the automatic transmission 60 is softened. On the contrary, when it is judged NO in step 2000, it is not required to soften the shock of the torque change by the automatic transmission, the process goes to step 107 without delaying the fuel injection timing.

**[0102]** Next, in step 107, a mass flow amount of the intake air detected by the mass flow amount detecting device 21 (hereinafter, simply refer to as the intake air amount) Ga is taken in, and next in step 108, the target air fuel ratio A/F is calculated on the basis of the map shown in Fig. 10B. Next, in step 109, the fuel injection amount Q required for setting the air fuel ratio to the target air fuel ratio A/F is calculated on the basis of the intake air amount Ga and the target air fuel ratio A/F.

**[0103]** As mentioned above, in the case where the low temperature combustion is performed, when the required load L or the engine speed N is changed, the opening degree of the throttle valve 20 and the opening degree of the EGR control valve 31 is immediately made to be coincident with the target opening degree ST and SE corresponding to the required load L and the engine speed N. Accordingly, for example, when the required load L is increased, the amount of the air within the combustion chamber 5 is immediately increased, so that the torque generated in the engine is immediately increased.

**[0104]** On the contrary, when the opening degree of the throttle valve 20 or the opening degree of the EGR control valve 31 is changed and the intake air amount is changed, the change of the intake air amount Ga is detected by the mass flow amount detecting device 21, and the fuel injection amount Q is controlled on the basis of the detected intake air amount Ga. That is, the fuel injection amount Q is changed after the intake air amount Ga is actually changed.

**[0105]** Since the same operations are performed in steps 112 to 123 in which the second combustion is performed as those of the first embodiment, the explanation thereof will be omitted.

**[0106]** Next, at a time of switching between the low temperature combustion corresponding to the first combustion and the second combustion, the intake air amount supplied within the combustion chamber 5 and the fuel injection amount are changed. In this case, since the intake air amount supplied within the combustion chamber 5 is actually changed with respect to the timing at which the fuel injection amount is changed, the generated torque is temporarily changed. Accordingly, in the internal combustion engine provided with the automatic transmission 60, there is a risk that the shock of the generated torque is increased in accordance with the control state of the automatic transmission 60. Then, the description will be given with respect to the control content of performing a switching control of the combustion in accordance with the control state of the automatic transmission 60 in order to reduce the torque shock.

**[0107]** At first, the generated torque is changed at a time of switching the shift change ratio of the automatic transmission 60. In this case, it is judged whether the shift change ratio is at the switching state after judging the switching of the combustion state against the required load. At a time of switching the shift change ratio, it is structured so as to inhibit the switching between the first combustion and the second combustion. Accordingly, it is possible to reduce the increase in the torque change due to the change of the torque generated together with the switching between the first combustion and the second combustion. Further, at a time of switching the shift change ratio, it is possible to change the first combustion and the second combustion in synchronous with the switching of the shift change. By synchronously switching the combustion, it is possible to reduce the change of the torque in comparison with the case in which the switching of the shift change ratio and the switching of the combustion are performed at the different timing.

**[0108]** The generated torque is also changed at a time when the lockup mechanism 73 of the automatic transmission 60 is turned between on-state and off-state. Also, in this case, after judging the switching of the combustion state with respect to the required load, it is judged whether the shift change ratio is at the switching state. When the lockup mechanism 73 is turned between on-state and off-state, it is structured so as to inhibit the switching between the first combustion and the second combustion. Further, at a time of switching the lockup mechanism 73, it is possible to switch the first combustion and the second combustion in synchronous with the switching between on-state and off-state of the lockup mechanism 73. By synchronously switching the combustion, it is possible to reduce the change of the torque in comparison with the case in which the switching between on-state and off-state of the lockup mechanism 73 and the switching of the combustion are performed at the different timing.

**[0109]** Further, in addition to the torque change

generated at a time of switching between the low temperature combustion corresponding to the first combustion mentioned above and the second combustion, the generated torque is also changed when the air fuel ratio of a mixed gas to be burned in the engine for discharging NOx from the NOx absorbent 25 is switched to a rich state. That is, in the case of being switched to the rich air fuel ratio operation in the engine. In the switching of the operation state of the engine as mentioned above, it is also possible to synchronously perform a switching control of the rich air fuel ratio operation in accordance with the control state of the automatic transmission.

**[0110]** The control state of the automatic transmission includes a shift change state called as a power down shift. It is different from the down shift which is performed in the case where a speed reduction is performed in a vehicle, and corresponds to the shift down operation which is performed in a state where a positive drive torque is transmitted from the engine side to the transmission side. When the rotational synchronism at a time of operating the power down shift is detected, the generated torque is changed. In the control state of the automatic transmission as mentioned above, it is possible to inhibit the switching control between the first combustion and the second combustion. Further, in the case where it is required to increase the engine speed at a time of operating the power down shift, it is possible to switch the air fuel ratio of the mixed gas to be burned in the engine for discharging NOx from the NOx absorbent 25 to the rich state.

**[0111]** In the internal combustion engine described in the present embodiment, since an amount of an inert gas supplied within the combustion chamber is large in the low temperature combustion corresponding to the first combustion state, the combustion is performed under the condition where oxygen is relatively insufficient, that is, the combustion is performed under a relatively severe combustion condition. Accordingly, when the engine stall is generated if the low temperature combustion corresponding to the first combustion is performed, it is also possible to inhibit the low temperature combustion, whereby the engine stall is not generated. In this case, it is judged whether or not the condition is at a state where the engine stall is likely to occur, for example, whether or not the braking operation is performed, whether or not the engine speed becomes lower than a predetermined engine speed (for example, 2000 rpm), whether or not an external load (an air conditioner and the like) connected to the engine is increased, a reduction amount of the engine speed at a unit time is greater than a predetermined amount, and the like, such that it is possible to inhibit the low temperature combustion.

**[0112]** At a time of executing a low temperature combustion in which an amount of an EGR gas supplied within a combustion chamber (5) is more than an amount of the EGR gas when a generation amount of a soot becomes peak and the soot is hardly generated

and changing a speed of an automatic transmission (60), in order to reduce a generated torque, an injection amount of a fuel injected from a fuel injection valve (6) is corrected to be increased so as to reduce an air fuel ratio, or a fuel injection timing is delayed, or an opening degree of an EGR control valve (31) is corrected to be increased so as to correct an amount of the EGR gas to increase, and further at a time of executing a combustion in which the amount of the EGR gas supplied within a combustion chamber (5) is less than the amount of the EGR gas when a generation amount of a soot becomes peak and the soot is hardly generated and changing a speed of an automatic transmission (60), in order to reduce a generated torque, the injection amount of the fuel injected from the fuel injection valve (6) is corrected to be reduced.

### Claims

1. An internal combustion engine adapted for executing at least a first kind of combustion in which a combustion chamber (5) contains inert gas of an amount larger than that corresponding to a peak generation amount of soot, said internal combustion engine is connected to an automatic transmission (60), characterized in that
 

when in the first kind of combustion a shift change is going to be performed by the automatic transmission (60) being under a control state an air fuel ratio prevailing during the first kind of combustion is reduced.
2. An internal combustion engine according to claim 1, characterized in that the air fuel ratio is reduced by correcting to increase an amount of injected fuel supplied within said combustion chamber (5).
3. An internal combustion engine according to claim 1, characterized in that the air fuel ratio is reduced by correcting to reduce an amount of air supplied within said combustion chamber (5).
4. An internal combustion engine according to the preamble of claim 1, characterized in that
 

when in the first kind of combustion a shift change is going to be performed by the automatic transmission (60) being under a control state an injection timing of the fuel supplied within said combustion chamber (5) during the first kind of combustion is delayed.
5. An internal combustion engine according to claim 1 or 4, characterized in that there is provided switching means (40) for selectively switching the first kind of combustion corresponding to a combustion in which said soot is hardly generated and a second kind of combustion in which an amount of an inert gas supplied within said combustion chamber is less than an amount of an inert gas when an amount of generating soot becomes peak, whereby the fuel injection amount is corrected to be reduced when said second kind of combustion is performed and a shift change by said automatic transmission (60) is performed.
6. An internal combustion engine according to one of claims 1 to 5, characterized in that there is arranged a catalyst (25, 26) having an oxidation function within the engine exhaust passage for oxidizing an unburned hydrocarbon discharged from said combustion chamber (5).
7. An internal combustion engine according to claim 5, characterized in that there is provided an exhaust gas recirculating apparatus (29, 30, 31) for recirculating an exhaust gas discharged from said combustion chamber (5) into an engine intake passage (17), and said inert gas is constituted by a recirculated exhaust gas recirculated into said engine intake passage (17), and
 

an exhaust gas recirculation rate when said first kind of combustion is performed is substantially equal to or more than 55 %, and an exhaust gas recirculation rate when said second kind of combustion is performed is substantially equal to or less than 50 %.
8. An internal combustion engine according to one of claims 1 to 4, characterized in that the control state of said automatic transmission (60) corresponds to a time when the shift change is performed, at which it is inhibited to switch between the first kind of combustion and the second kind of combustion.
9. An internal combustion engine according to one of claims 1 to 4, characterized in that the control state of said automatic transmission (60) corresponds to a time when the shift change is performed, at which it is synchronously performed to switch between the first kind of combustion and the second kind of combustion.
10. An internal combustion engine according to one of claims 1 to 4, characterized in that the control state of said automatic transmission (60) corresponds to a switching between on-state and off-state in a lockup mechanism (73), at which it is inhibited to switch between the first kind of combustion and the second kind of combustion.
11. An internal combustion engine according to one of

claims 1 to 4, characterized in that the control state of said automatic transmission (60) corresponds to a switching between on-state and off-state in a lockup mechanism (73), at which it is synchronously performed to switch between the first kind of combustion and the second kind of combustion. 5

12. An internal combustion engine according to claim 1, characterized in that an NOx absorbent (25) which absorbs an NOx when an air fuel ratio of a flowing exhaust gas is lean and discharges the absorbed NOx when the air fuel ratio of the flowing exhaust gas is a stoichiometric air fuel ratio or rich is arranged within an engine exhaust passage (24), so as to switch an air fuel ratio of a mixed gas to be burned in the engine for discharging NOx from said NOx absorbent (25) to a rich state in synchronous with a timing at which the shift change in the control state of the automatic transmission (60) is performed. 10 15 20

13. An internal combustion engine according to claim 1, characterized in that an NOx absorbent (25) which absorbs an NOx when an air fuel ratio of a flowing exhaust gas is lean and discharges the absorbed NOx when the air fuel ratio of the flowing exhaust gas is a stoichiometric air fuel ratio or rich is arranged within an engine exhaust passage (24), so as to switch an air fuel ratio of a mixed gas to be burned in the engine for discharging NOx from said NOx absorbent to a rich state in synchronous with a timing at which a switching between on-state and off-state of a lockup mechanism (73) in the control state of the automatic transmission (60) is performed. 25 30 35

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FIG. 1

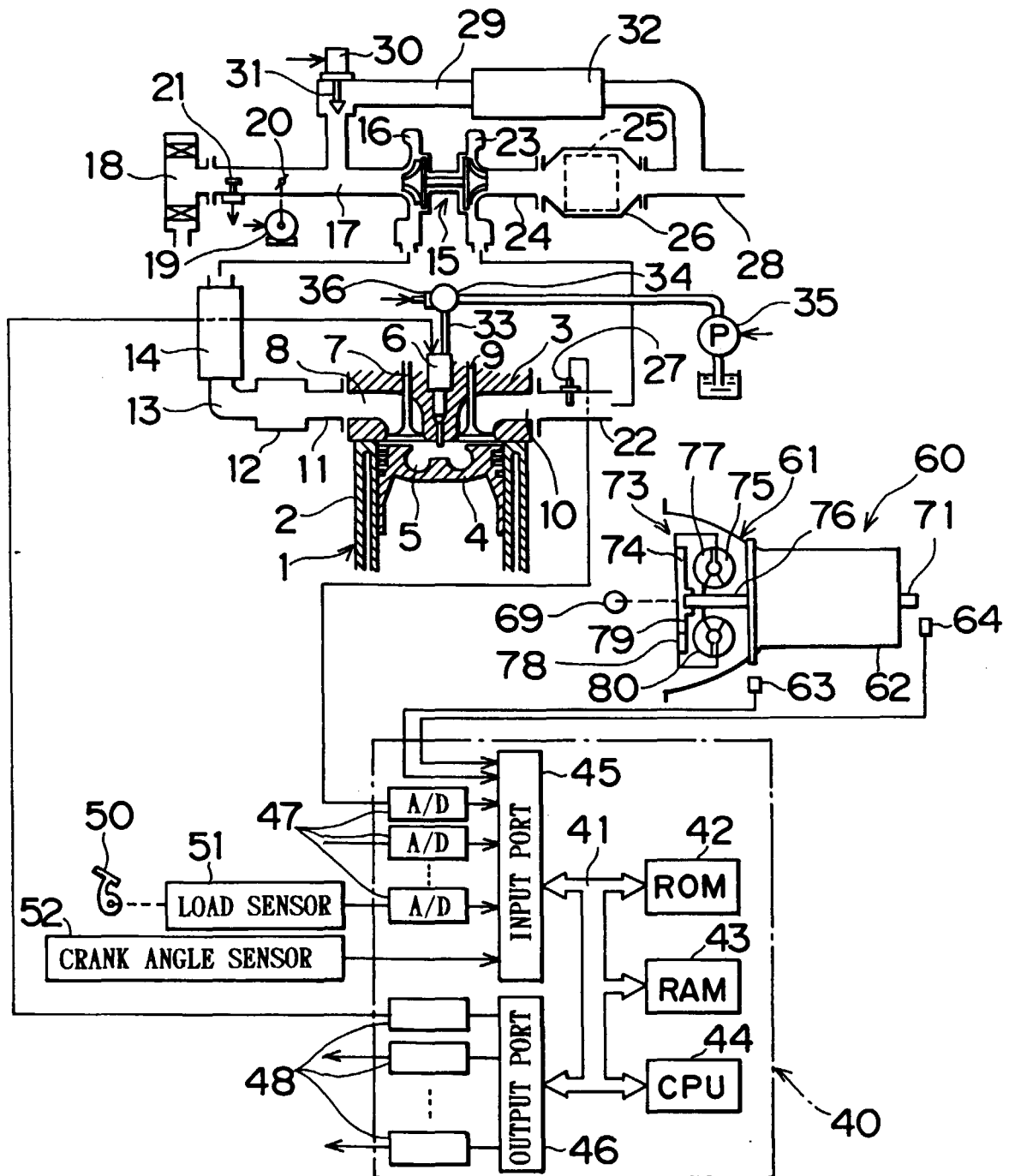


FIG. 2

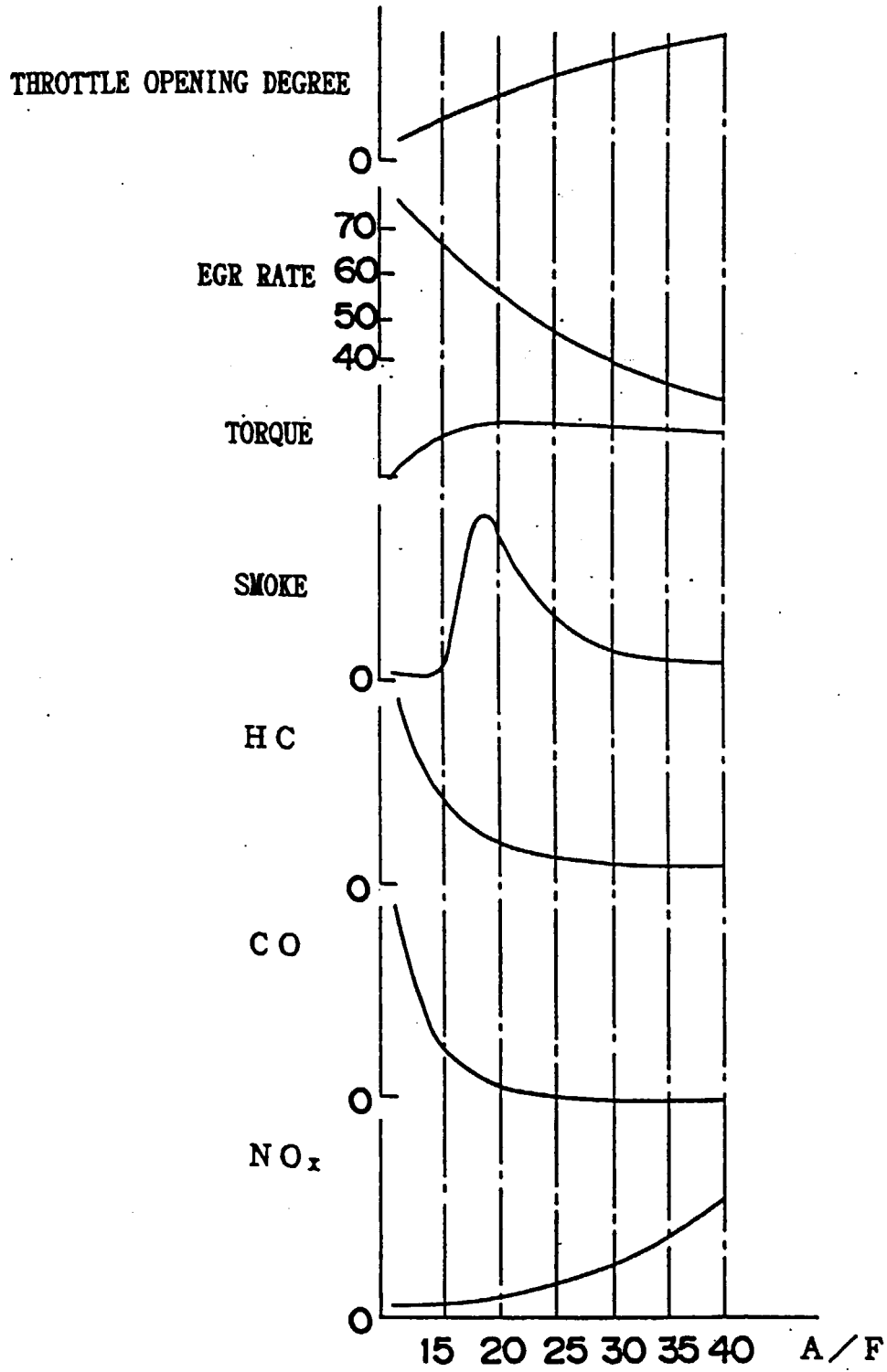


FIG. 3A

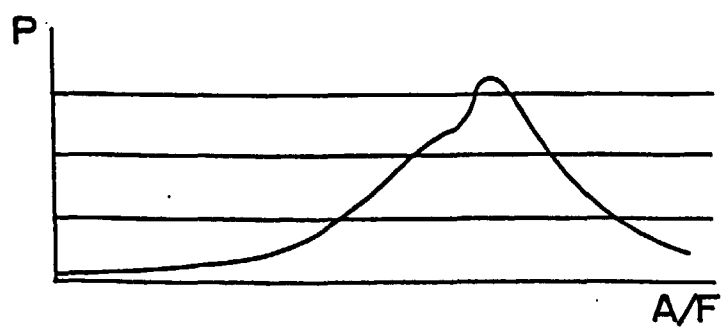


FIG. 3B

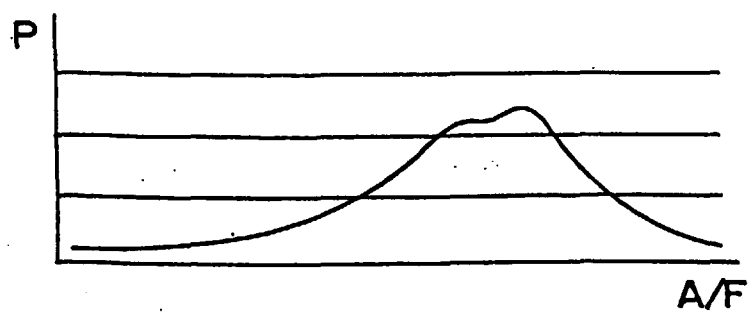


FIG. 4

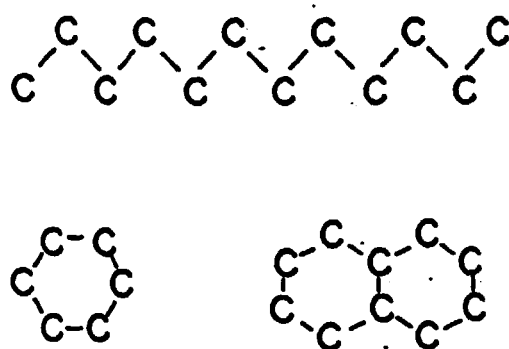


FIG. 5

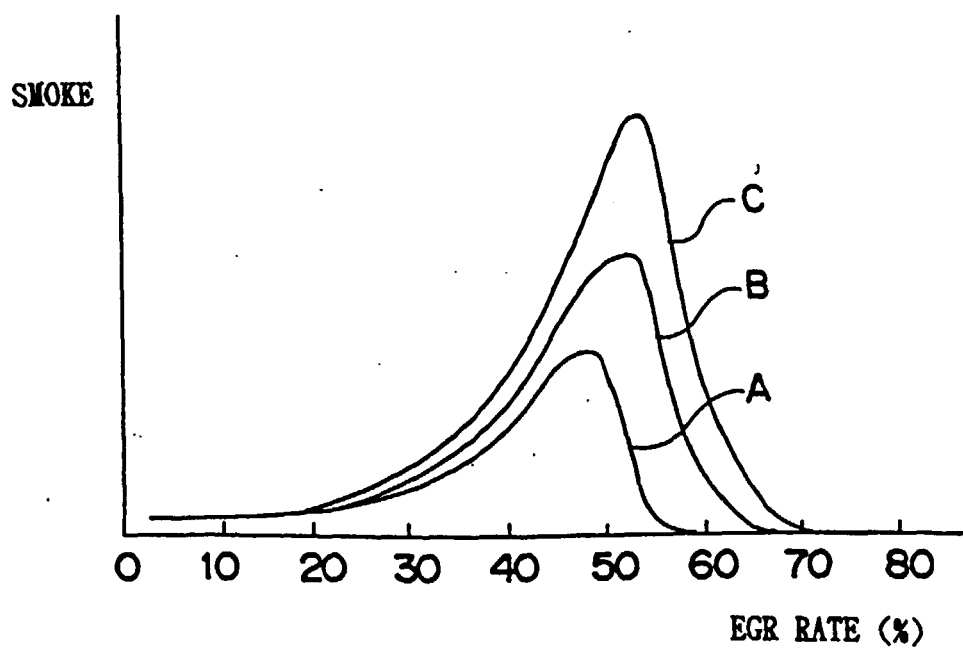


FIG. 6

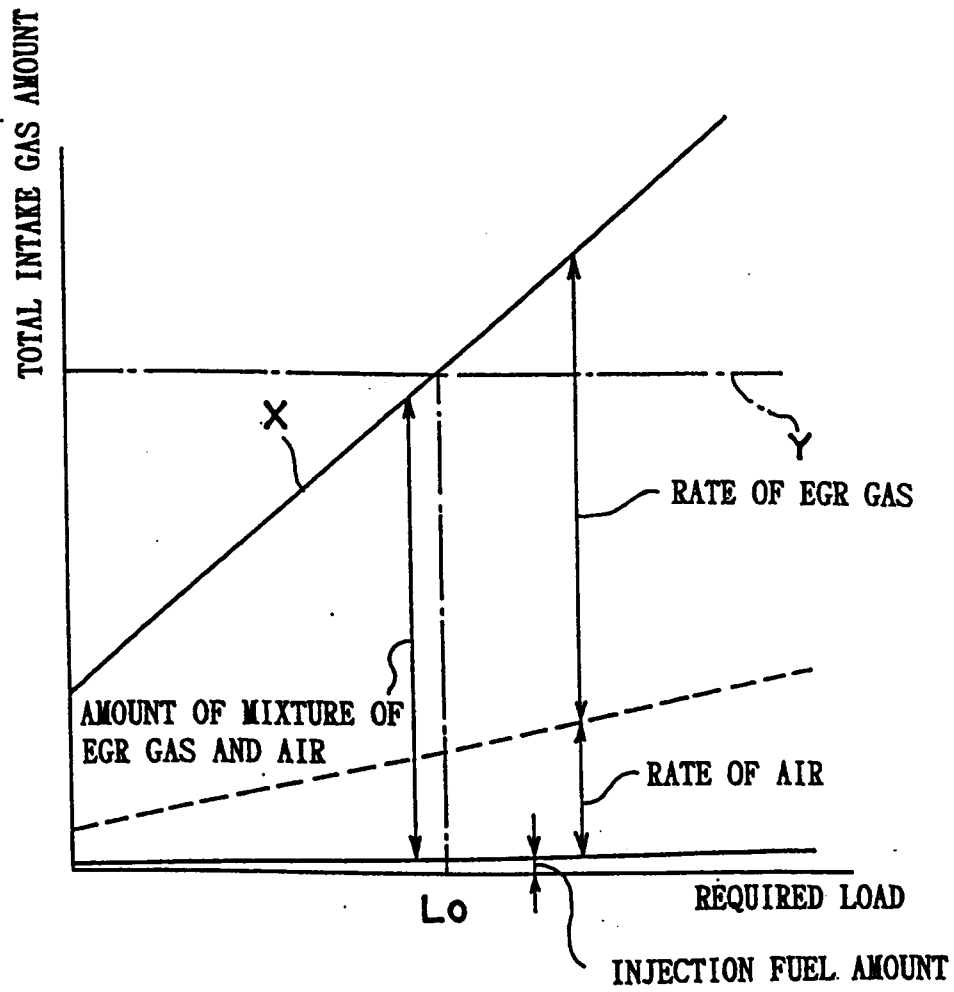


FIG. 7

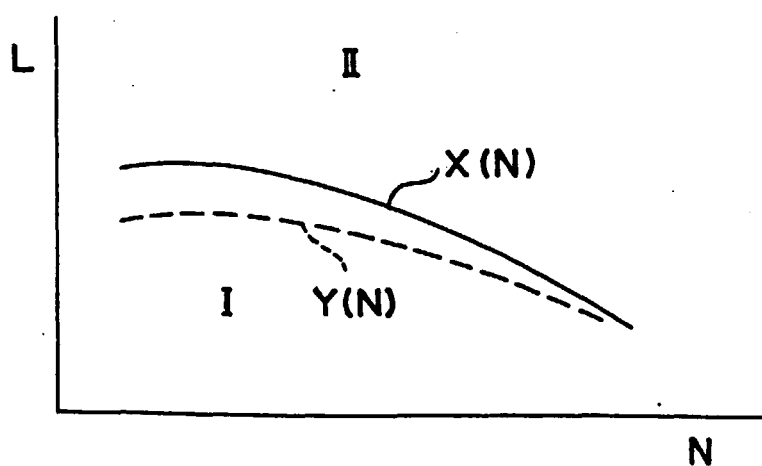


FIG. 8

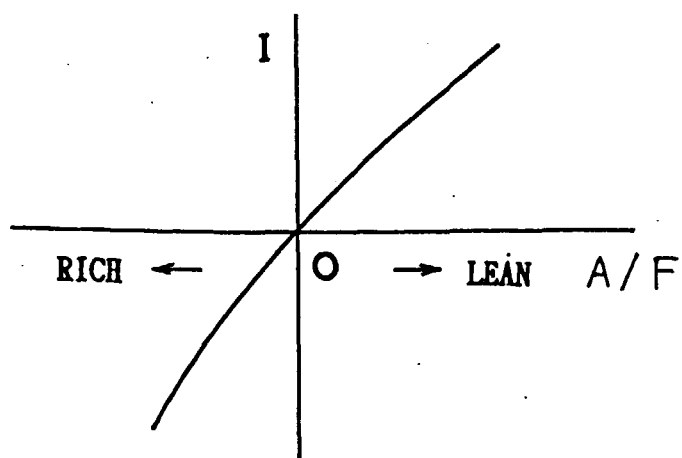


FIG. 9

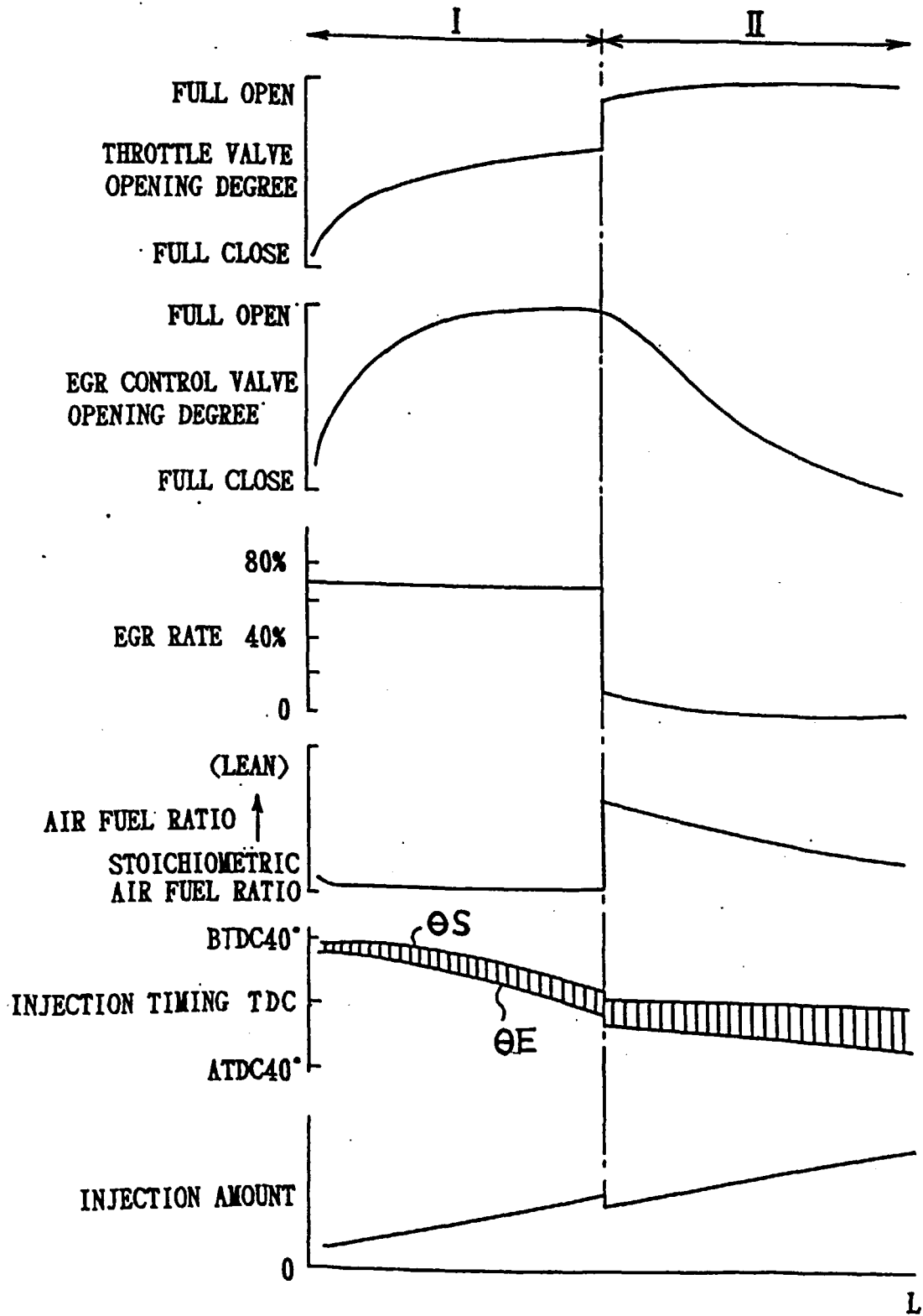


FIG. 10A

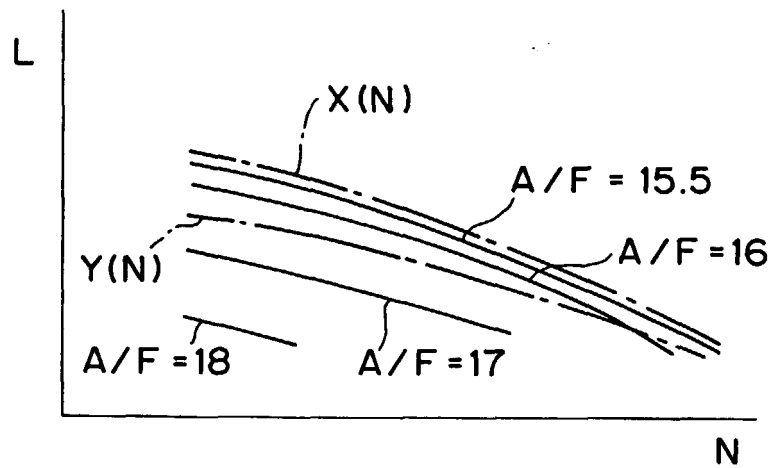


FIG. 10B

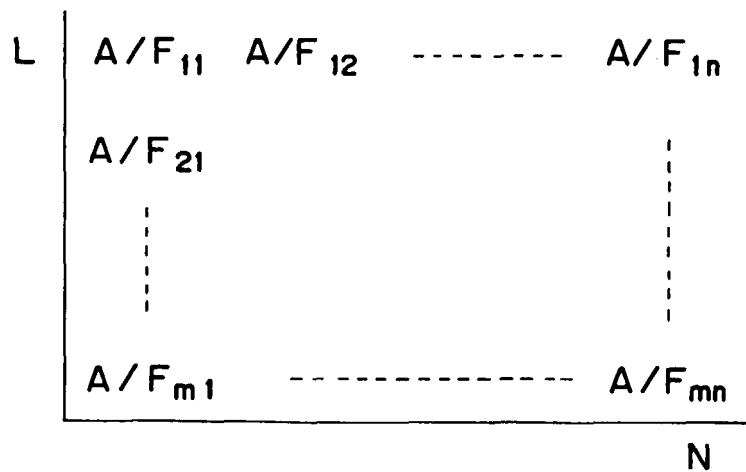


FIG. 11A

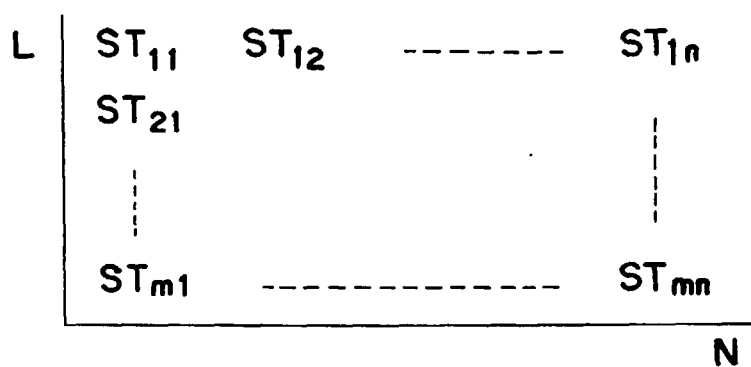


FIG. 11B

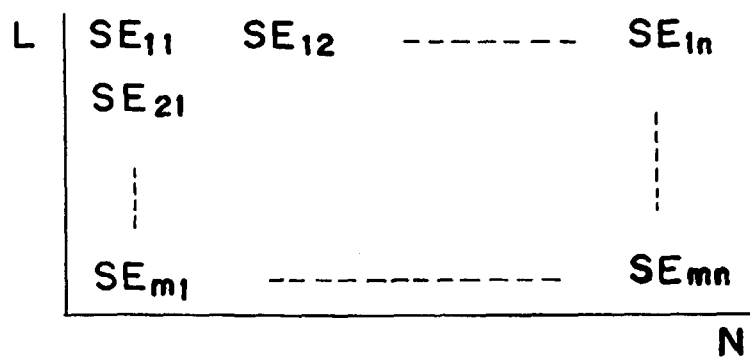


FIG. 12A

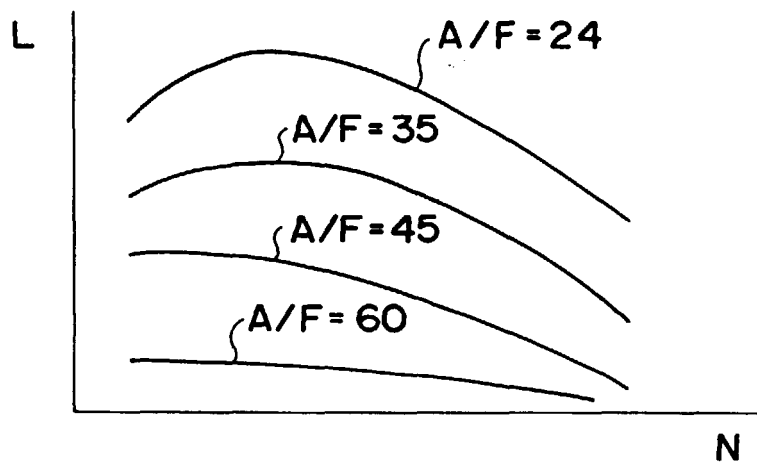


FIG. 12B

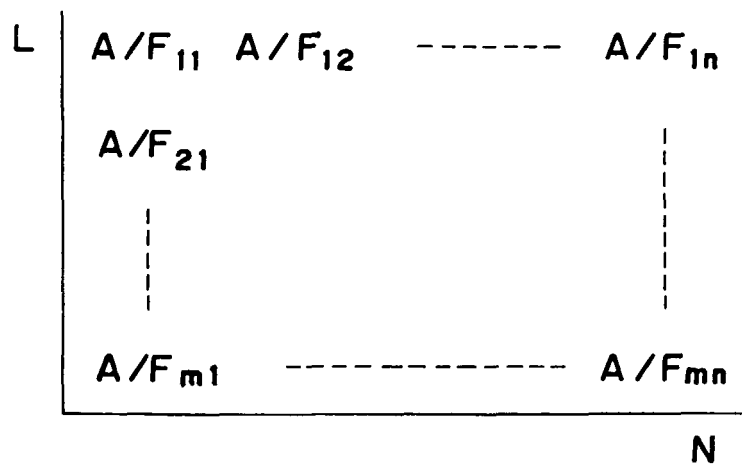


FIG. 13A

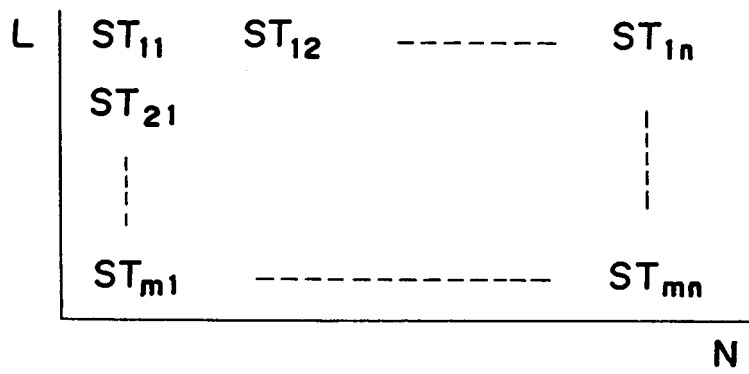


FIG. 13B

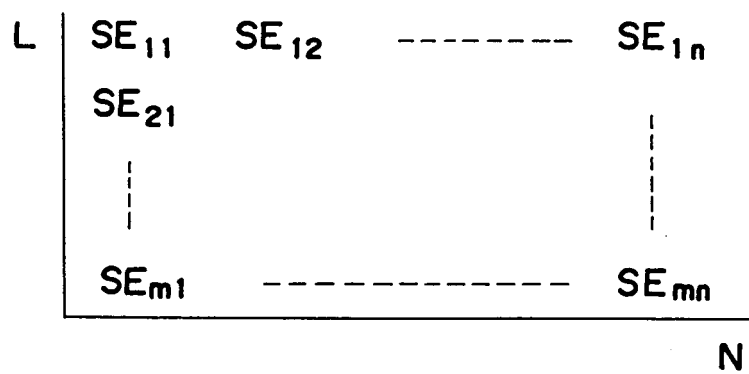


FIG. 14

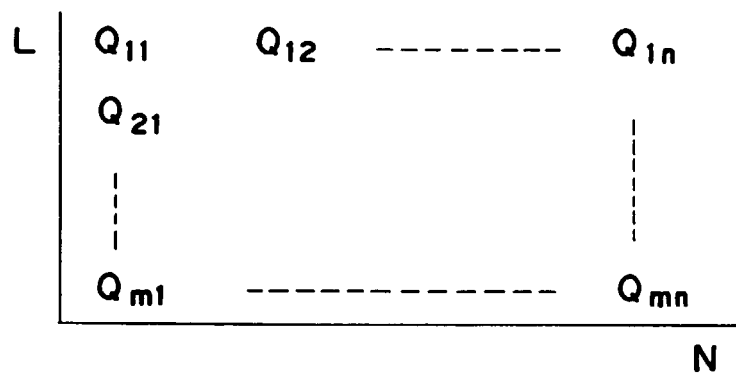
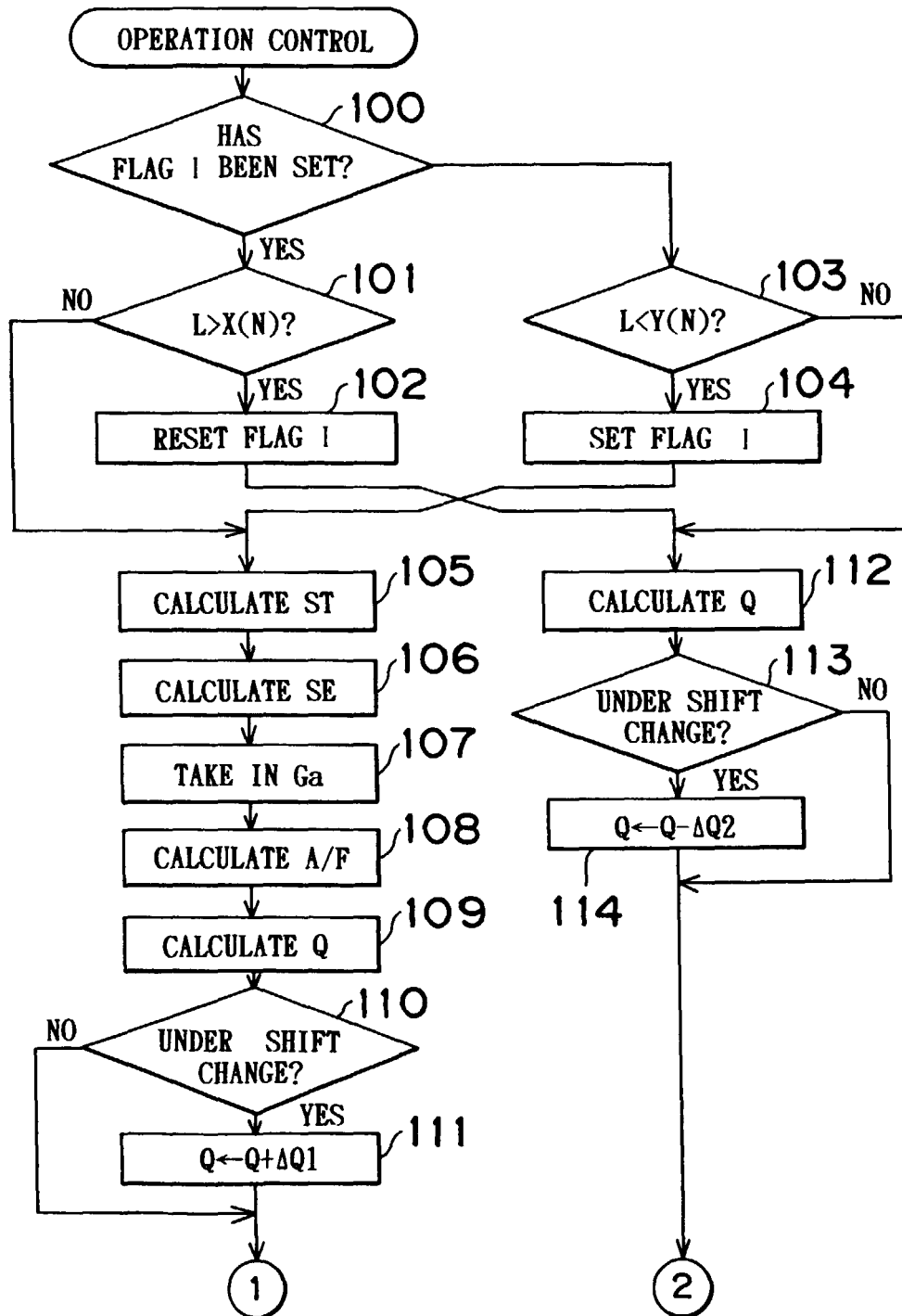


FIG. 15



## FIG. 16

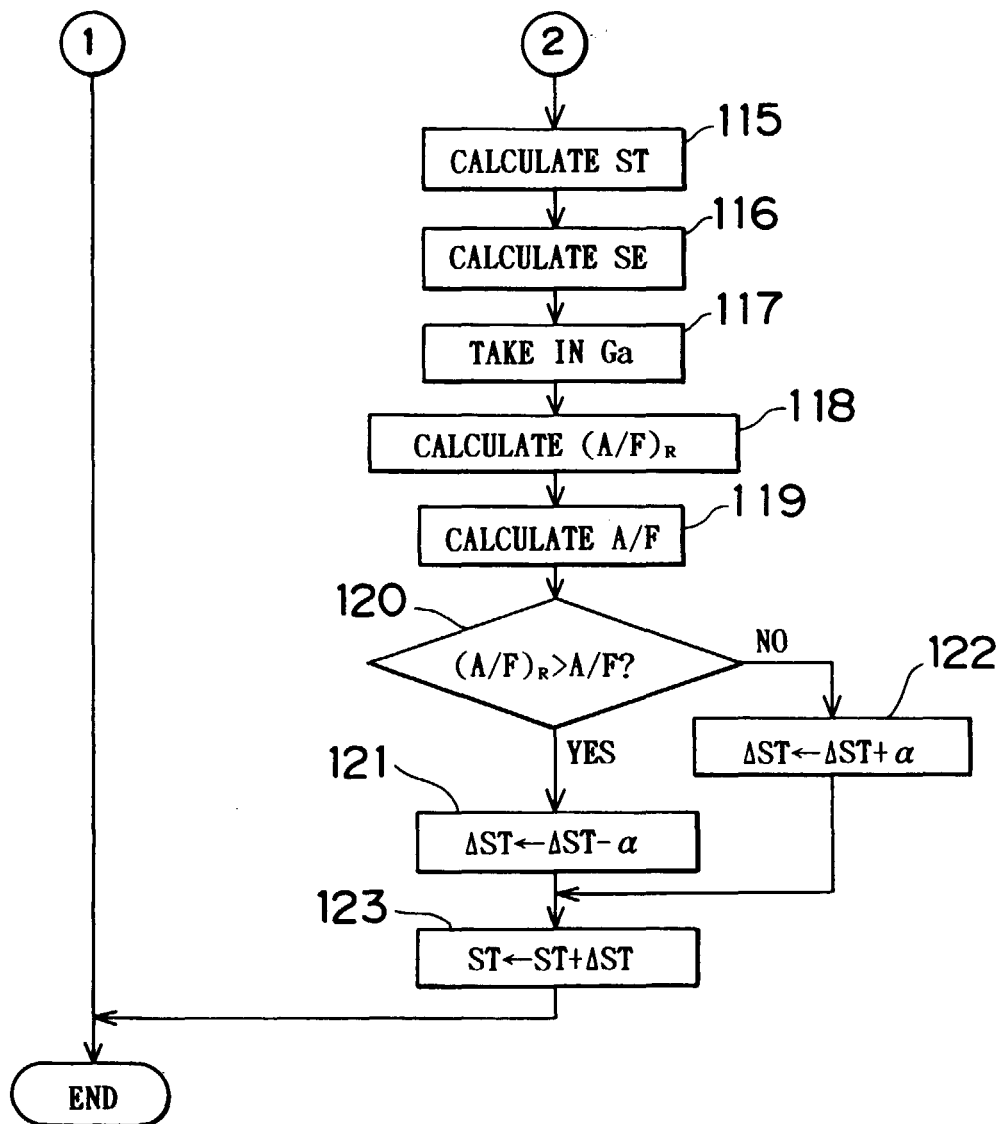


FIG. 17

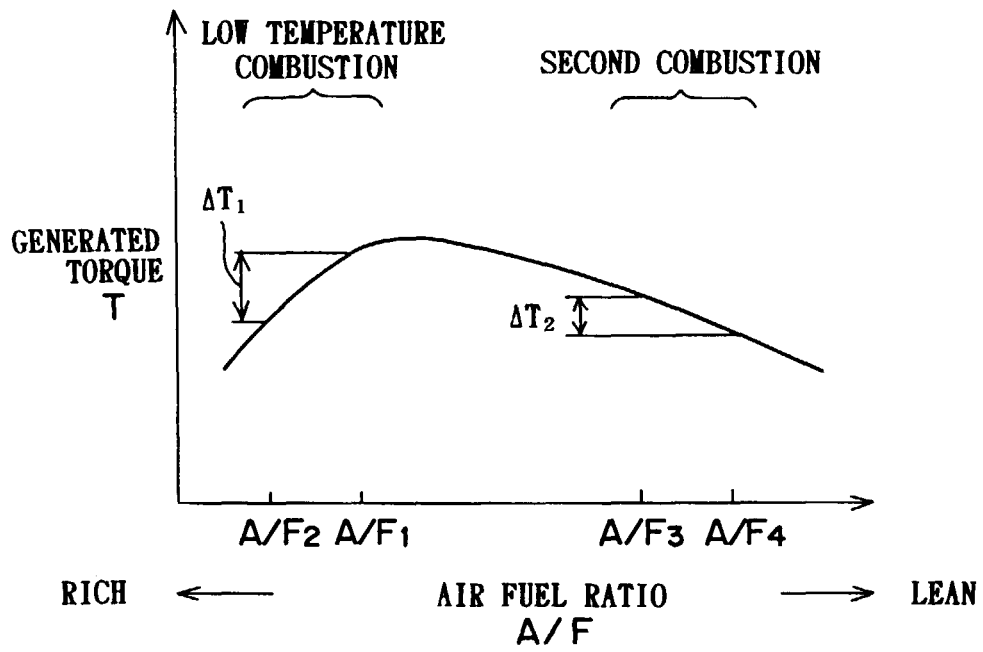


FIG. 18

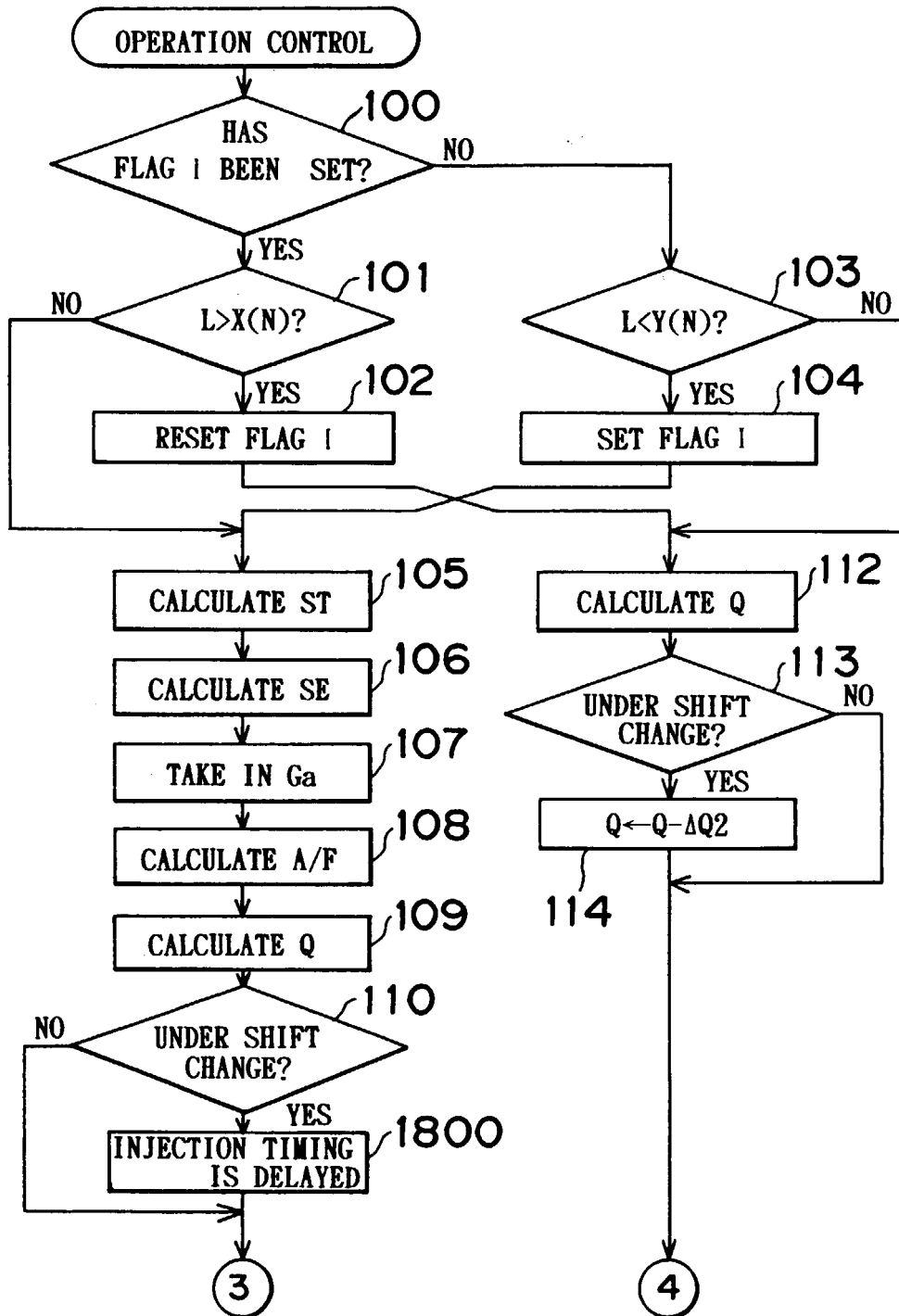


FIG. 19

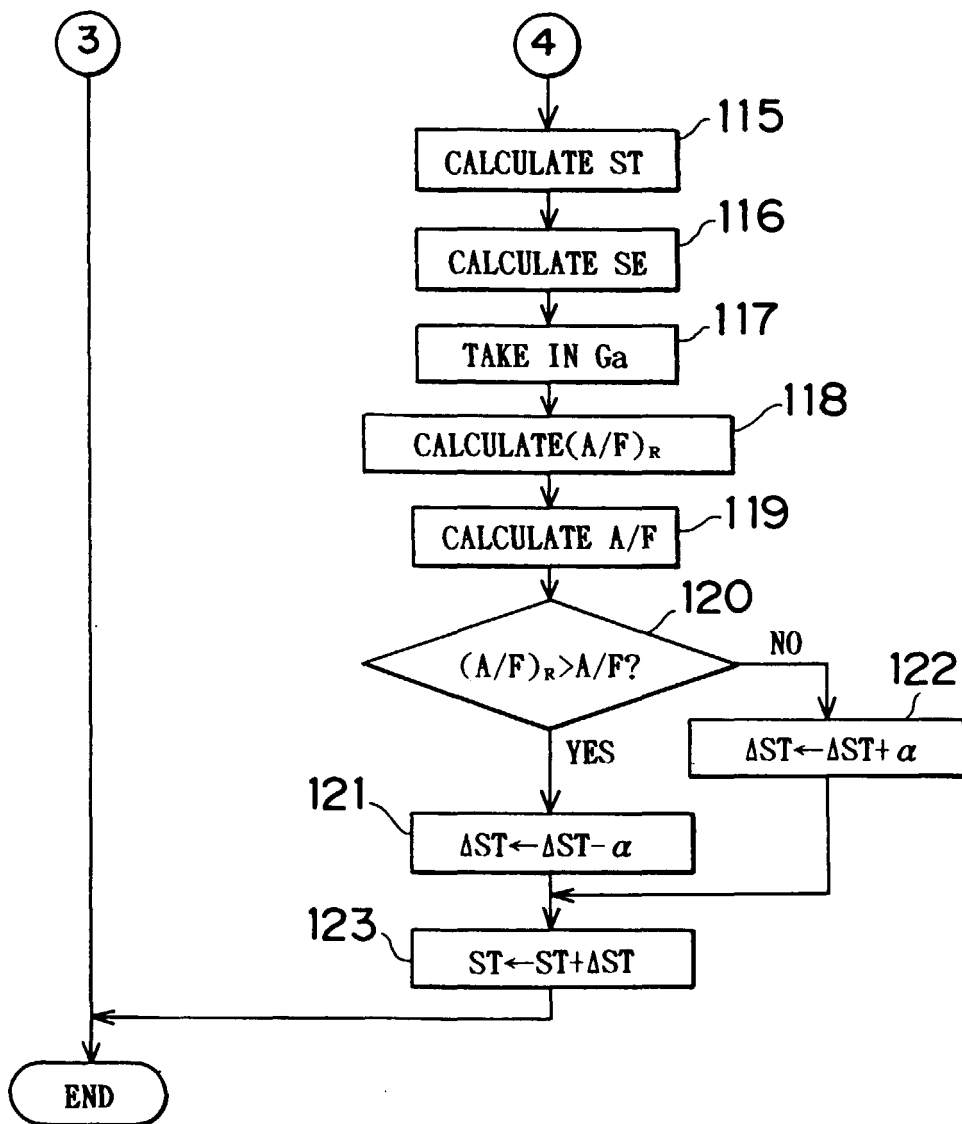


FIG. 20

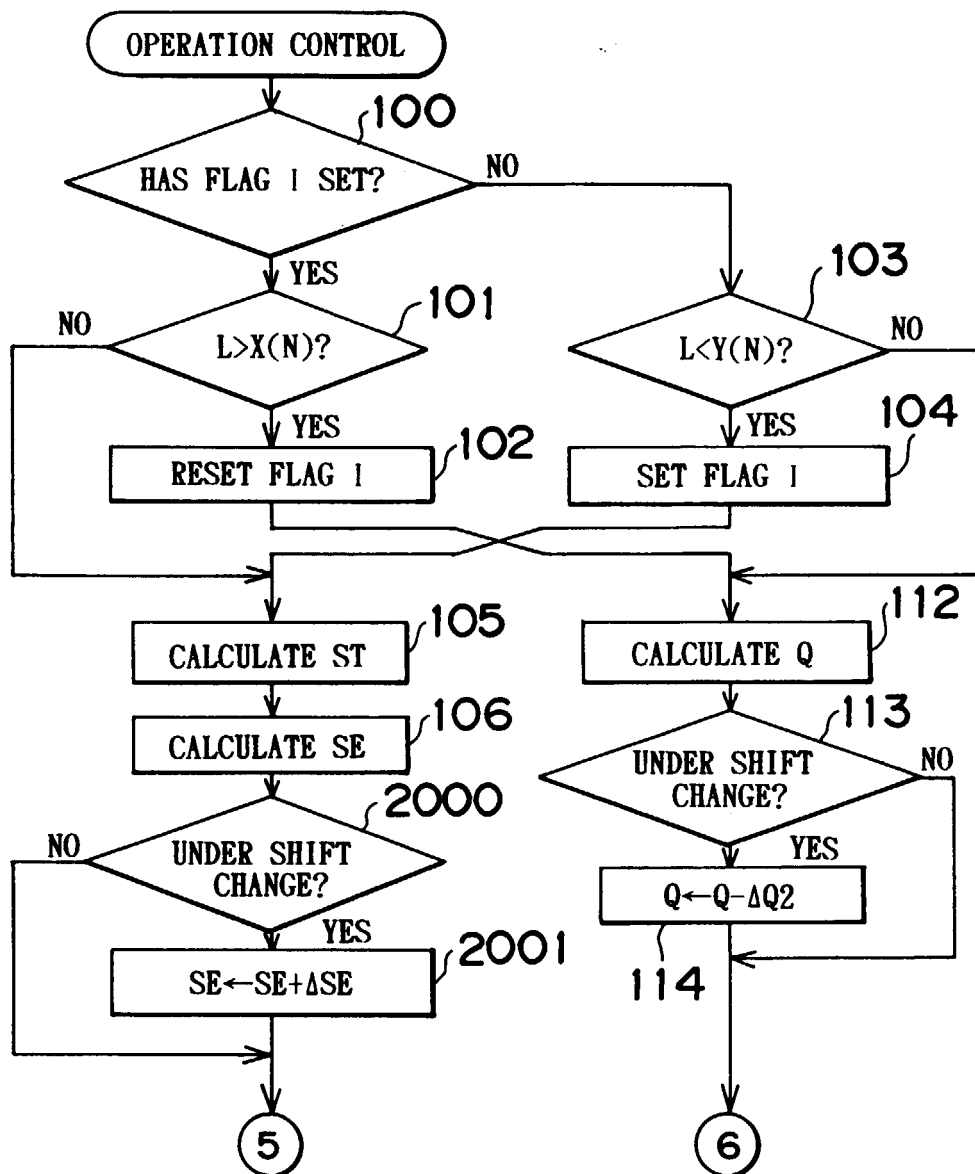


FIG. 21

