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(71) Applicant: **Nissan Motor Co., Ltd.**
Kanagawa 221-0023 (JP)

(72) Inventors:
- **Shiraishi, Taisuke**
  Atsugi-shi Kanagawa 243-0123 (JP)
- **Takahashi, Eiji**
  Atsugi-shi Kanagawa 243-0123 (JP)
- **Urushihara, Tomonori**
  Atsugi-shi Kanagawa 243-0123 (JP)

(74) Representative: **Holmes, Matthew William**
Nissan Motor Manufacturing (UK) Ltd
Nissan Technical Centre Europe
Cranfield Technology Park
Moulsoe Road
Bedfordshire MK43 0DB (GB)

(54) Internal combustion engine electric discharge structure

(57) An internal combustion engine electric discharge structure is provided which comprises a first electrode and a dielectric material. The first electrode includes a first voltage receiving end and a second engine attachment end with a long thin conductive material that discharges non-equilibrium plasma. The dielectric material covers the first electrode.

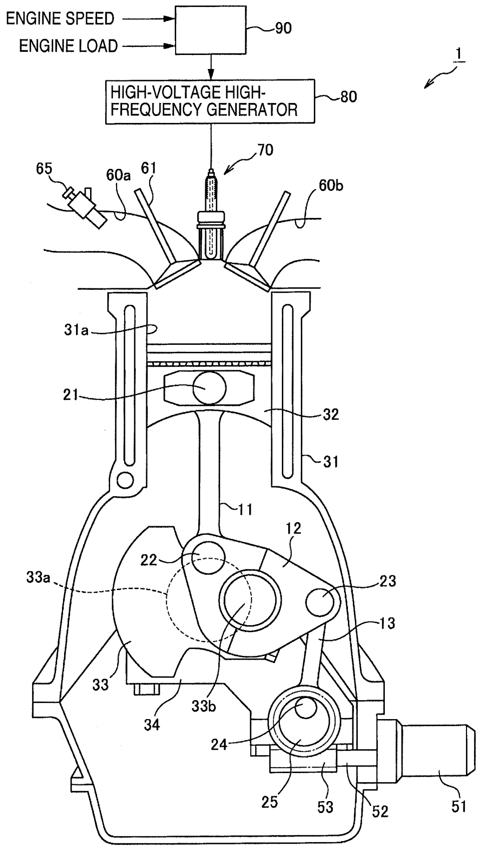

FIG. 1

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Description

[0001] The present invention generally relates to an internal combustion engine electric discharge apparatus and particularly, but not exclusively, to an electric discharge structure which discharges non-equilibrium plasma in order to increase a number of radicals and thereby improve auto-ignition properties of the internal combustion engine. Aspects of the invention relate to an apparatus, to a structure, to an engine, to a method and to a vehicle.

[0002] An electric discharge device has been proposed for an internal combustion engine in which the air-fuel mixture is ignited in an assisted manner by a sparkplug. In this electric discharge device radicals are generated in a cylinder and the auto-ignition properties of the air-fuel mixture are improved (see, Japanese Laid-Open Patent Application No. 2001-20842). The radicals tend to induce oxidation reactions (i.e. combustion), and the oxidation reactions (combustion) tend to become chain reactions. Therefore, when radicals are generated in the cylinder, the auto-ignition properties of the air-fuel mixture are improved.

[0003] As mentioned above, it has been discovered that, in order to improve the auto-ignition properties of the air-fuel mixture, a sparkplug can be used to generate radicals in the cylinder. However, since spark ignition is a thermal plasma discharge, the efficiency of radical generation is low even if spark ignition is induced by a sparkplug as in the conventional apparatus previously described. Moreover, in this conventional apparatus the amount of radicals generated is limited. It is therefore believed that the effects of improving the auto-ignition properties are small.

[0004] It is an aim of the present invention to address this issue and to improve upon such known technology. Embodiments of the invention may provide an electric discharge structure which is used in an internal combustion engine and which can improve the auto-ignition properties of an air-fuel mixture beyond that of conventional practice, and to provide a method for controlling the operation of the internal combustion engine. Other aims and advantages of the invention will become apparent from the following description, claims and drawings.

[0005] Aspects of the invention therefore provide an apparatus, a structure, a method, an engine and a vehicle as claimed in the appended claims.

[0006] According to another aspect of the invention for which protection is sought, there is provided an internal combustion engine electric discharge structure comprising a first electrode including a first voltage receiving end and a second engine attachment end with a conductive material that discharges non-equilibrium plasma by a barrier discharge and a dielectric material covering the first electrode.

[0007] The conductive material may be long and/or thin.

[0008] The structure may comprise a second electrode facing the first electrode on a periphery of the dielectric material.

[0009] In an embodiment, the second electrode includes a tubular electrode surrounding at least a portion of the first electrode.

[0010] The structure may comprise a cylinder head having the second electrode attached thereto, with the first electrode including a linear central electrode.

[0011] In an embodiment, the first electrode includes a linear central electrode and the second electrode is disposed as at least part of one of a wall surface of a combustion chamber and a top surface of a piston.

[0012] The structure may comprise a fuel injection valve for supplying fuel into a combustion chamber of an internal combustion engine and a voltage application device operatively coupled to the first voltage receiving end of the first electrode for applying a voltage between the first electrode and the second electrode, such that the non-equilibrium plasma generates radicals within the combustion chamber before an air-fuel mixture in the combustion chamber undergoes auto-ignition.

[0013] The structure may comprise a control unit operatively coupled to the voltage application device to vary a discharge start timing of the non-equilibrium plasma discharge in accordance with a mechanical load of the internal combustion engine.

[0014] In an embodiment, the control unit sets the discharge start timing of the non-equilibrium plasma discharge to occur during an intake stroke when the mechanical load of the internal combustion engine is comparatively low.

[0015] In an embodiment, the control unit sets the discharge start timing of the non-equilibrium plasma discharge to be increasingly advanced as the mechanical load of the internal combustion engine becomes lower.

[0016] In an embodiment, the control unit sets the discharge start timing of the non-equilibrium plasma discharge to occur during a compression stroke when the mechanical load of the internal combustion engine is comparatively high.

[0017] In an embodiment, the control unit sets the discharge start timing of the non-equilibrium plasma discharge to be increasingly delayed as the mechanical load of the internal combustion engine increases.

[0018] The structure may comprise a control unit operatively coupled to the voltage application device to set a discharge start timing of the non-equilibrium plasma discharge to occur after an intake valve has opened.

[0019] The structure may comprise a control unit operatively coupled to the voltage application device to set a discharge ending timing of the non-equilibrium plasma discharge to occur before an intake valve has closed.

[0020] The structure may comprise a control unit operatively coupled to the voltage application device to set a discharge energy of the non-equilibrium plasma discharge such that the discharge energy increases as the mechanical load of the internal combustion engine becomes lower when the mechanical load of the internal combustion engine is in a low load range.

[0021] In an embodiment, the control unit increases the discharge energy of non-equilibrium plasma discharge by at least one method selected from increasing a voltage value of an AC voltage applied between the first and second electrodes, increasing a frequency of the AC voltage applied between the first and second electrodes, and increasing an application duration of the AC voltage applied between the first and second electrodes.

[0022] The structure may comprise a variable compression ratio mechanism arranged to change a mechanical compression ratio of the internal combustion engine and a control unit operatively coupled to the variable compression ratio mechanism to reduce the mechanical compression ratio so that an air-fuel mixture does not undergo compression ignition when a mechanical load of the internal combustion engine is in a high load range, and volumetric ignition is performed.

[0023] The structure may comprise a fuel injection control unit operatively coupled to the fuel injection valve to control injection of fuel directly into a cylinder of the internal combustion engine such that a stratified air-fuel mixture is formed in the cylinder when a mechanical load of the internal combustion engine is in a low load range.

[0024] According to a further aspect of the invention for which protection is sought, there is provided an internal combustion engine control method for controlling an operating state of an internal combustion engine, comprising determining a mechanical load of the internal combustion engine, injecting fuel into a combustion chamber of the internal combustion engine, applying a voltage to an electric discharge device having a first electrode and a second electrode to produce a non-equilibrium plasma discharge generating radicals within the combustion chamber before an air-fuel mixture of the fuel undergoes auto-ignition and setting a discharge start timing of the non-equilibrium plasma discharge such that the discharge start timing varies in accordance with the mechanical load.

[0025] For example, in an embodiment an internal combustion engine electric discharge structure is provided which comprises a first electrode and a dielectric material. The first electrode includes a first voltage receiving end and a second engine attachment end with a long thin conductive material that discharges non-equilibrium plasma. The dielectric material covers the first electrode.

[0026] Within the scope of this application it is envisaged that the various aspects, embodiments, examples, features and alternatives set out in the preceding paragraphs, in the claims and/or in the following description and drawings may be taken individually or in any combination thereof.

[0027] The present invention will now be described, by way of example only, with reference to the accompanying drawings, in which:

Figure 1 is a simplified schematic cross-sectional view of a portion of a multi-link engine that is part of an electric discharge structure in accordance with a first embodiment;

Figure 2A is a partial cross-sectional view of the electric discharge structure of the engine shown in Figure 1;

Figure 2B is a cross-sectional view of the electric discharge structure illustrated in Figure 2A, taken along section line 2B-2B of Figure 2A;

Figure 3A is a diagram showing the electric discharges obtained when an AC voltage (electric potential) is applied to a spark ignition discharge mechanism in accordance with a comparative example of a conventional discharge mechanism;

Figure 3B is a diagram showing the electric discharges obtained when an AC voltage (electric potential) is applied to the electric discharge structure in accordance with the first illustrated embodiment;

Figure 4 is a diagram showing various methods for increasing the discharge energy of the electric discharge structure;

Figure 5 is a graph showing the problems with forming non-equilibrium plasma by the application of short pulses in accordance with a comparative example of a conventional discharge mechanism;

Figure 6A is a simple link diagram showing the arrangement of a multi-link variable compression ratio mechanism at a high compression ratio;

Figure 6B is a simple link diagram showing the arrangement of the multi-link variable compression ratio mechanism at a low compression ratio;

Figure 6C is a simple link diagram showing the method for varying the compression ratio using the multi-link variable compression ratio mechanism;

Figure 7 is a perspective view of a variable valve timing mechanism for adjusting the opening and closing timing of a valve;

Figure 8A is a simplified elevational view of the variable valve timing mechanism when valves are in a closed state;

Figure 8B is a simplified elevational view of the variable valve timing mechanism when the valves are in a state of maximum lift;

Figure 8C is a simplified elevational view showing the variable valve timing mechanism when the stroke amount of cam followers is minimized, cam noses are at the highest position, and the valves are in a closed state;

Figure 8D is a simplified elevational view of the variable valve timing mechanism when the stroke amount of cam followers is minimized, the cam noses are at the lowest position, and the valves are in a closed state;

Figure 9 is a graph showing the valve lift amount and the opening and closing timings in the variable valve timing mechanism;

Figure 10A is a graph showing the relationship of an air-fuel ratio to various operational states of the engine having the electric discharge structure in accordance with the first embodiment;

Figure 10B is a graph showing the relationship of a barrier discharge start timing to various operational states of the engine having the electric discharge structure in accordance with the first embodiment;

Figure 10C is a graph showing the relationship of discharge energy to various operational states of the engine having the electric discharge structure in accordance with the first embodiment;

Figure 10D is a graph showing the relationship of an intake valve close timing to various operational states of the engine having the electric discharge structure in accordance with the first embodiment;

Figure 10E is a graph showing the relationship of a mechanical compression ratio to various operational states of the engine having the electric discharge structure in accordance with the first embodiment;

Figure 11 is a graph showing the variation in the heat generation rate depending on if and when the barrier discharge start timing begins;

Figure 12A is a drawing schematically depicting the state in which radicals are distributed within the cylinder when barrier discharge does not occur;

Figure 12B is a drawing schematically depicting the state in which radicals are distributed within the cylinder when barrier discharge is initiated during compression stroke;

Figure 12C is a drawing schematically depicting the state in which radicals are distributed within the cylinder when barrier discharge is initiated during intake stroke;

Figure 13 is a graph showing the relationship between the barrier discharge start timing and the crank angle at which the mass combustion ratio is 50%;

Figure 14 is a graph showing the piston behavior in a multi-link variable compression ratio mechanism;

Figure 15 is a graph showing the relationship between the air-fuel ratio and combustion stability;

Figure 16 is a graph showing the problems due to the heat generation rate suddenly increasing to an excessive degree, and the effects of the illustrated embodiment;

Figure 17A is a graph showing the correlation between an air-fuel ratio and a fluctuation rate of the depicted average effective pressure;

Figure 17B is a graph showing that a fuel consumption rate can be reduced if a lean combustion limit is expanded;

Figure 18 is a simplified schematic cross-sectional view of a portion of an engine that is part of an electric discharge structure in accordance with a second embodiment;

Figure 19 is a simplified schematic cross-sectional view of a portion of the engine showing the manner in which fuel is injected into the engine in accordance with the second embodiment;

Figure 20A is a graph showing the relationship of an air-fuel ratio to various operational states of the engine having the electric discharge structure in accordance with the second embodiment;

Figure 20B is a graph showing the relationship of a barrier discharge start timing to various operational states of the engine having the electric discharge structure in accordance with the second embodiment;

Figure 20C is a graph showing the relationship of discharge energy to various operational states of the engine having an electric discharge structure in accordance with the second embodiment;

Figure 20D is a graph showing the relationship of an intake valve close timing to various operational states of the engine having an electric discharge structure in accordance with the second embodiment;

Figure 20E is a graph showing the relationship of a mechanical compression ratio to various operational states of the engine having an electric discharge structure in accordance with the second embodiment;

Figure 21 is a partial cross-sectional view showing the operational configuration of the engine having an electric discharge structure in accordance with a third embodiment;

Figure 22A is a partial cross-sectional view showing the operational configuration of the engine having an electric discharge structure in accordance with a fourth embodiment where a barrier discharge is formed within a combustion chamber;

Figure 22B is a partial cross-sectional view showing the operational configuration of the engine having an electric discharge structure in accordance with a fourth embodiment where a barrier discharge is formed within a concave part of a top surface of a piston;

Figure 23A is a partial cross-sectional view showing the operational configuration of the engine having an electric discharge structure in accordance with a fifth embodiment where a barrier discharge is formed within a combustion chamber;

Figure 23B is a partial cross-sectional view showing the operational configuration of the engine having an electric discharge structure in accordance with a fifth embodiment where a barrier discharge is formed within a concave part of a top surface of a piston;

Figure 24A is a diagram showing a waveform of an alternating current as a sine curve applied to the electric discharge structure; and

Figure 24B is a diagram showing a waveform of an alternating current as a bipolar multiple pulse applied to the electric discharge structure.

[0028] Selected embodiments of the present invention will now be explained with reference to the drawings. It will be apparent to those skilled in the art from this disclosure that the following descriptions of the embodiments of the present invention are provided for illustration only and not for the purpose of limiting the invention as defined by the appended claims and their equivalents.

[0029] First, the internal combustion engine electric discharge structure will be described.

[0030] As described above, an engine has been proposed in which spark ignition generates radicals (chemically active species which are in a state wherein molecular dissociation is induced by the collision of high-energy electrons with fuel or air molecules, and which promote ignition of an air-fuel mixture) in a cylinder and in which the auto-ignition properties (compression ignition properties) of the air-fuel mixture are improved.

[0031] However, the effects of improving ignition properties in such an engine have been small. Specifically, spark ignition involves a thermal plasma discharge. In a thermal plasma discharge, kinetic energy is adequately exchanged among electrons, ions, and molecules. The result is an establishment of a state of thermal equilibrium in which the electron energy, the ion energy, and the neutral particle energy are in equilibrium with each other. Radicals are chemically active species which are in a state wherein molecular dissociation is induced by collisions of high-energy electrons with fuel or air molecules, and which promote ignition of the air-fuel mixture. In spark ignition, energy is also imparted to ions and molecules which do not contribute to the generation of radicals, and the efficiency of conversion of input energy to electron energy is low. When the input energy is increased in order to increase the amount of radicals, there is a possibility that the electrodes will melt. Therefore, it is difficult to increase the amount of radicals.

[0032] In view of this, a non-equilibrium plasma discharge is beneficial. In a non-equilibrium plasma discharge, a thermally non-equilibrium state is achieved in which the electron temperature (electron energy) alone is extremely high (specifically, electron energy is much higher than ion energy and ion energy is equal to neutral particle energy), and the efficiency of converting input energy to electron energy is high. Heat loss is small in a non-equilibrium plasma discharge because the gas temperature is not increased. The danger that the electrodes will melt is also small.

[0033] Because of such reasons, radicals can be generated comparatively easily if a non-equilibrium plasma discharge is used. In view of this, a non-equilibrium plasma discharge mechanism for an engine is proposed herein. To conduct a non-equilibrium plasma discharge, possibilities include methods using a barrier discharge and methods using short pulse application. It has been discovered that of these methods, a barrier discharge is particularly advantageous.

[0034] Now referring to Figure 1, a simplified schematic cross-sectional view of a portion of a multi-link engine 1 is illustrated that forms a part of an electric discharge structure in accordance with a first embodiment. As explained hereinafter, the multi-link engine 1 utilizes a non-equilibrium plasma discharge function, advantageously barrier discharge, to improve the auto-ignition properties of the multi-link engine 1.

[0035] The engine 1 is provided with a barrier discharge device 70. The barrier discharge device 70 is provided between an intake port 60a and an exhaust port 60b, substantially in the center of a combustion chamber of a cylinder head. The barrier discharge device 70 generates radicals through barrier discharge, which is a non-equilibrium plasma discharge. The barrier discharge device 70 is also capable of igniting an air-fuel mixture through barrier discharge at a comparatively high load (when the air-to-fuel ratio of the air-fuel mixture is comparatively rich). The detailed structure of the barrier discharge device 70 will be described hereinafter with reference to an enlarged view (Figure 2).

[0036] The engine 1 having a barrier discharge function according to the present embodiment has a variable compression ratio mechanism (hereinafter referred to as a "multi-link variable compression ratio mechanism"), which uses a multi-link mechanism for connecting a piston 32 to a crankshaft 33 by two links. The multi-link variable compression ratio mechanism connects the piston 32 to the crankshaft 33 by an upper (first) link 11 and a lower (second) link 12. The multi-link variable compression ratio mechanism also controls the lower link 12 by using a control (third) link 13 to vary the mechanical compression ratio.

[0037] The upper link 11 is connected at the top end to the piston 32 via a piston pin 21. The upper link 11 is connected at the bottom end to one end of the lower link 12 via a connecting pin 22. The piston 32 receives combustion pressure that moves the piston 32 within a cylinder 31 a of a cylinder block 31 back and forth.

[0038] The lower link 12 is connected at one end to the upper link 11 via the connecting pin 22. The lower link 12 is connected at the other end to the control link 13 via a connecting pin 23. The lower link 12 also has a substantially central connecting hole in which crank pins 33b of the crankshaft 33 are disposed. Thus, the lower link 12 oscillates around the crank pins 33b as a center axis. The lower link 12 is divided into two left and right members. The crankshaft 33 comprises a plurality of crank journals 33a and a plurality of crank pins 33b for each cylinder. The journals 33a are rotatably supported by the cylinder block 31 and a ladder frame 34. The crank pins 33b are eccentric relative to the crank journals 33a by a predetermined amount, and the lower link 12 is oscillatably connected thereto.

[0039] The control link 13 is connected to the lower link 12 via the connecting pin 23. The control link 13 is also connected at the other end to a control shaft 25 via a connecting pin 24. The control link 13 oscillates or rocks around the connecting pin 24. A gear is formed on the control shaft 25, and this gear meshes with a pinion 53 provided to a rotating axle 52 of an actuator 51. The control shaft 25 is rotated by the actuator 51 to move the connecting pin 24.

[0040] Various sensors are provided for sensing the operating state of the engine, including the engine rotation speed and the engine load. The signals of various sensors are inputted to a controller 90. The controller 90 controls the actuator 51 to rotate the control shaft 25 and vary the compression ratio. The controller 90 also controls a high-voltage high-frequency generator 80 so that the AC voltage value, the application duration, the AC frequency, the application timing, and other parameters corresponding to the operating state of the engine are applied. Thus, the controller 90 may be considered to constitute a non-equilibrium plasma discharge control unit. In addition, the high-voltage high-frequency generator 80 constitutes a voltage application device. Furthermore, the controller 90 controls the fuel injection of a fuel injection valve 65 provided to the intake port 60a. An intake valve 61 is capable of varying the opening and closing timings thereof, as is described hereinafter. The controller 90 determines the engine load and performs control according to the load. The controller 90 is configured from a microcomputer comprising a central processing unit (CPU), a read-only memory (ROM), a random access memory (RAM), and an input/output interface (I/O interface). The controller 90 can also be configured from a plurality of microcomputers.

[0041] Figures 2A and 2B contain enlarged cross-sectional views of the barrier discharge device 70. The barrier discharge device 70 of the illustrated embodiment discharges non-equilibrium plasma by using a barrier discharge. Non-equilibrium plasma can also be formed by applying a short pulse instead of forming a barrier discharge, but barrier discharge is beneficial in the illustrated embodiment. The reasons for this are described hereinafter.

[0042] The barrier discharge device 70 comprises a central electrode 71 and a tubular electrode 72. The central electrode 71 is a rod-shaped electrical conductor. The entire periphery of the central electrode 71 is covered by a dielectric material (insulating material) 73. The central electrode 71 is connected to the high-voltage high-frequency generator 80 via a terminal 71 a. An AC voltage is applied to the central electrode 71 upon being generated by the high-voltage high-frequency generator 80. The value, application duration, AC frequency, application timing, and other characteristics of the AC voltage are controlled (set) according to the operating state of the engine 1,

[0043] The tubular electrode 72 is a tubular electrical conductor. The tubular electrode 72 is attached to the cylinder head. The inner periphery side of the tubular electrode 72 is a discharge chamber 72a. The central electrode 71 protrudes into the discharge chamber 72a. The central electrode 71 is provided on the top side of the substantial center of the fuel chamber. The center of the central electrode is substantially parallel to a line extending through the center of the fuel chamber. The distance from the central electrode 71 to the dielectric material and the distance from the dielectric material to the tubular electrode 72 are set to be substantially the same.

[0044] When an AC voltage is applied to the central electrode 71 from the high-voltage high-frequency generator 80, streamers S are generated between the tubular electrode 72 and the dielectric material 73 as shown in Figure 2A. A plurality of streamers S is generated in the vertical direction as shown in Figure 2A. The streamers are branched into thin streaks, and Figure 2A shows a state in which six streamers are generated on both the right and left sides of the dielectric material 73. The streamers are also formed in a radial pattern about the dielectric material 73, as shown in Figure 2B. Figure 2B shows a state in which twelve streamers are formed in a radial pattern about the dielectric material 73. The barrier discharge device 70 can generate a large amount of radicals in the discharge chamber 72a by forming a plurality of streamers S. It is also possible for multipoint simultaneous ignition, i.e., a volumetric ignition (hereinafter referred to as "volume ignition"), to occur within the discharge chamber.

[0045] The barrier discharge device 70 can perform multiple electric discharges within a predetermined time, whereby a large amount of radicals can be generated in the discharge chamber 72a. This will be described with reference to Figures 3A and 3B.
Figures 3A and 3B contain views showing the electric discharge obtained when an AC voltage (electric potential) is applied. Figure 3A is a diagram showing the electric discharges obtained when an AC voltage (electric potential) is applied by a spark ignition discharge mechanism in accordance with a comparative example of a conventional discharge mechanism. Figure 3B is a view showing the electric discharges obtained when an AC voltage (electric potential) is applied by the electric discharge structure in accordance with the illustrated embodiment.

[0046] First, as a comparison, a case will be described in which an AC voltage is applied to the spark ignition discharge mechanism of a conventional sparkplug. In cases in which an AC voltage is applied to the sparkplug, an arc discharge occurs between the electrodes when the absolute value of an electric potential V_0 formed between the electrodes by the applied voltage reaches a discharge voltage (insulation breakdown electric potential) Va, as shown in Figure 3A. Arc discharge similarly occurs when the polarity is inverted. With this sparkplug, four arc discharges occur within the discharge time t as shown in Figure 3A. A discharge takes place in one location, and the form of the discharge is either point or linear.

[0047] In the barrier discharge device 70, the dielectric material (insulating material) 73 covers the central electrode 71. The dielectric material 73 acts as a capacitor. After a barrier discharge (non-equilibrium plasma discharge) has occurred, an electric charge is accumulated on the surface of the dielectric material 73. The barrier discharge (non-equilibrium plasma discharge) occurs between the dielectric material 73 and the tubular electrode 72 when the absolute value of the difference between the electric potential V_0 created by the applied voltage and the electric potential Vw created by the surface electric charge of the dielectric material 73 reaches a discharge voltage Vd, as shown in Figure 3B. Therefore, streamers S are formed at a plurality of locations in the discharge chamber 72a in the barrier discharge device 70, and eight barrier discharges (non-equilibrium plasma discharges) occur within the discharge time t, as shown in Figure 3B.

[0048] Thus, the barrier discharge device 70 can increase the number of discharges in the same time (discharge time t) to a greater level than that obtained with a sparkplug in a conventional method.

[0049] Though not shown in the drawings, increasing the voltage value of the AC voltage in the barrier discharge device 70 also makes more likely that the absolute value of the difference between the electric potential V_0 created by the applied voltage and the electric potential Vw created by the surface electric charge of the dielectric material 73 will reach the discharge voltage Vd, and makes it possible to increase the number of discharges.

[0050] Figure 4 is a diagram showing various methods for increasing the discharge energy of the electric discharge structure.

[0051] The discharge energy of the barrier discharge device 70 is controlled by the voltage value, application duration, and AC frequency of the AC voltage from the high-voltage high-frequency generator 80. One possibility for increasing the discharge energy of the barrier discharge device 70 is a method for increasing the voltage value of the AC voltage in the manner shown in plot (B-1) of Figure 4 relative to the waveform of a reference AC applied voltage (plot (A) of Figure 4). The discharge energy of the barrier discharge part can also be increased by increasing the frequency of the AC voltage, the applied duration as in plot (B-2) of Figure 4, or the AC frequency as in plot (B-3) of Figure 4.

[0052] As described above, another method for forming non-equilibrium plasma aside from initiating a barrier discharge is a method for forming non-equilibrium plasma by applying a short pulse between the electrodes and blocking the electric potential before the transition to an arc discharge. However, a barrier discharge is beneficial in the illustrated embodiment. The reasons for this are described with reference to Figure 5 which is a graph showing the problems with forming non-equilibrium plasma by the application of short pulses in accordance with a comparative example of a conventional discharge mechanism.

[0053] To form non-equilibrium plasma by the application of short pulses, the required voltage (electric potential) corresponding to the discharge location (density, air-fuel mixture composition, and the like) must be applied. Non-equilibrium plasma is generated if the voltage V1 is applied at a pressure P0, but when the voltage V2 is applied, thermal plasma is generated, as shown in Figure 5. Thus, non-equilibrium plasma or thermal plasma is generated merely by slight variations in the applied voltage, and the discharge lacks robustness with short pulse application.

[0054] By contrast, with a barrier discharge, the electrodes are originally covered on one side with a dielectric material, and the voltage is kept substantially within a range that extends from the discharge start voltage (lower limit of voltage) to a voltage at which the withstand-voltage properties of the dielectric material can be ensured (upper limit of voltage), whereby non-equilibrium plasma can always be maintained regardless of the voltage. An arc transition does not take place because the electrodes are covered by a dielectric material. Thus, discharge robustness is high. In an internal combustion engine, the potential required for a discharge varies extensively, and it is difficult to form non-equilibrium plasma by the application of short pulses. Therefore, non-equilibrium plasma based on a barrier discharge is advantageous for application in an internal combustion engine.

[0055] Figures 6A-6C are simple link diagrams showing the arrangement of a multi-link variable compression ratio mechanism. With a multi-link variable compression ratio mechanism, the mechanical compression ratio can be varied by rotating the control shaft 25 and varying the position of the connecting pin 24. For example, if the connecting pin 24 is at position A as shown in Figure 6C, the top dead center (TDC) is at a high level, resulting in a high compression ratio. If the connecting pin 24 is at position B as shown in Figures 6B and 6C, the control link 13 is pushed upward, and the position of the connecting pin 23 rises. The lower link 12 is thereby rotated counterclockwise around the crank pins 33b, the connecting pin 22 moves down, and the piston 32 in the piston top dead center (TDC) moves to a lower position. Therefore, the compression ratio is low.

[0056] Figure 7 is a perspective view showing a variable valve timing mechanism for adjusting the opening and closing timing of a valve. The engine 1 having a barrier discharge function comprises a variable valve timing mechanism 200. The mechanism disclosed, for example, in Japanese Laid-Open Patent Application No. 11-107725 can be used as the variable valve timing mechanism 200. This is described with reference to the drawings.

[0057] The variable valve timing mechanism 200 comprises a camshaft 210, a link arm 220, a valve lift control shaft 230, a rocker arm 240, a link member 250, and oscillating cams 260. Cam followers 63 are pushed by the oscillation of the oscillating cams 260, thus opening and closing valves (intake valves) 61.

[0058] The camshaft 210 is rotatably supported at the top part of the cylinder head along the longitudinal direction of the engine. One end of the camshaft 210 is inserted through a cam sprocket 270. The cam sprocket 270 is rotated by the transmission of torque from a crank axle of the engine. The camshaft 210 rotates together with the cam sprocket 270. The camshaft 210 can rotate relative to the cam sprocket 270 by hydraulic pressure, and the phase of the camshaft 210 relative to the cam sprocket 270 can be varied. This type of structure makes it possible to vary the rotational phase of the camshaft 210 relative to the crank axle. A cam 211 is fixed to the camshaft 210. The cam 211 rotates integrally with the camshaft 210. The pair of oscillating cams 260 connected by pipes is inserted through the camshaft 210. The oscillating cams 260 oscillate around the camshaft 210 as a rotational center, and the cam followers 63 perform a stroke.

[0059] The link arm 220 is supported by the insertion of the cam 211. The valve lift control shaft 230 is disposed parallel to the camshaft 210. A cam 231 is formed integrally on the valve lift control shaft 230. The valve lift control shaft 230 is controlled by an actuator 280 so as to rotate within a predetermined range of rotational angles.

[0060] The rocker arm 240 is supported by the insertion of the cam 231 and is connected to the link arm 220. The link member 250 is connected to the rocker arm 240.

[0061] The camshaft 210 is inserted through the oscillating cams 260, which can oscillate around the camshaft 210. The oscillating cams 260 are connected to the link member 250. The oscillating cams 260 move up and down, pushing down on the cam followers 63 and opening and closing the valves 61.

[0062] Next, the action of the variable valve timing mechanism 200 will be described with reference to Figures 8A-8D.

[0063] Figures 8A and 8B are views showing the manner in which the stroke amount of the cam followers 63 is maximized to maximize the lift amount of the valves 61. Figure 8A shows the manner in which cam noses 262 are at their highest positions, and the oscillation direction of the oscillating cams 260 is inverted. At this time, the cam followers 63 are at top end positions, and the valves 61 are in a closed state. Figure 8B shows the manner in which the cam noses 262 are at their lowest positions, and the oscillation direction of the oscillating cams 260 is inverted. At this time, the cam followers 63 are at bottom end positions, and the valves 61 are in a state of maximum lift.

[0064] Figures 8C and 8D are views showing the manner in which the stroke amount of the cam followers 63 is minimized. Figure 8C shows the manner in which the cam noses 262 are at their highest positions and the oscillating direction of the oscillating cams 260 is inverted. Figure 8D shows the manner in which the cam noses 262 are at their lowest positions and the oscillation direction of the oscillating cams 260 is inverted. In the present embodiment, the stroke amount of the cam followers 63 is zero, and the lift amount of the valves 61 is also zero. Therefore, in Figures 8C and 8D, the valves 61 are always in a closed state regardless of the action of the oscillating cams 260.

[0065] To increase the stroke amount of the cam followers 63 and the lift amount of the valves 61, the valve lift control shaft 230 is rotated to lower the position of the cam 231 and to set the axial center P1 below the axial center P2, as shown in Figures 8A and 8B. The entire rocker arm 240 is thereby moved downward.

[0066] When the camshaft 210 is rotatably driven in this state, the drive force is transmitted first to the link arm 220 and then to the rocker arm 240, the link member 250, and the oscillating cams 260.

[0067] When the cam 211 is to the left of the camshaft 210, as shown in Figure 8A, the base-circle parts 261 of the oscillating cams 260 are in contact with the cam followers 63, at which time the cam followers 63 are at the top end position and the valves 61 are in a state of maximum lift.

[0068] When the cam 211 is to the right of the camshaft 210, as shown in Figure 8B, the cam noses 262 of the oscillating cams 260 are in contact with the cam followers 63, at which time the cam followers 63 are at the bottom end positions and the valves 61 are in an opened state.

[0069] To reduce the stroke amount of the cam followers 63 and the lift amount of the valves 61, the valve lift control shaft 230 is rotated to raise the position of the cam 231, and the axial center P1 is set above and to the right of the axial center P2, as shown in Figures 8C and 8D. The entire rocker arm 240 is thereby moved upward. When the camshaft 210 is rotatably driven in this state, the drive force is transmitted first to the link arm 220 and then to the rocker arm 240, the link member 250, and the oscillating cams 260. When the cam 211 is to the left of the camshaft 210, as shown in Figure 8C, the base-circle parts 261 of the oscillating cams 260 are in contact with the cam followers 63. When the cam 211 is to the right of the camshaft 210, as shown in Figure 8D, the base-circle parts 261 of the oscillating cams 260 are still in contact with the cam followers 63.

[0070] Thus, in cases in which the valve lift control shaft 230 is rotated, the position of the cam 231 is raised, and the axial center P1 is set above and to the right of the axial center P2, the cam followers 63 do not perform a stroke and the valves 61 remain closed, even though the camshaft 210 rotates and the oscillating cams oscillate.

[0071] Figure 9 is a graph showing the valve lift amount and the opening and closing timings in the variable valve timing mechanism 200. The solid lines indicate the lift amount and the opening and closing timings of the valves 61 when the valve lift control shaft 230 is rotated. The dashed lines indicate the opening and closing timings of the valves 61 when the phase of the camshaft 210 is varied relative to the cam sprocket 270.

[0072] According to the structure of the variable valve timing mechanism 200 described above, the lift amount and operating angle of the valves 61 can be continually varied. Thus, the lift amount and operating angle of the valves 61 can be continually and freely varied by varying the angle of the valve lift control shaft 230 and the phase of the camshaft 210 relative to the cam sprocket 270.

[0073] Figures10A-10E are graphs showing an example of an operation map of the engine having a barrier discharge function. The range of extremely low load (for example, engine is in an idle state) will now be discussed. When the load is in a range of extremely low load, the air-fuel ratio A/F is set to a constant value (Figure 10A). Also, the barrier discharge start timing is set to a constant timing of the intake stroke (Figure 10B). The constant timing is a timing in which the setting is made near the most advanced angle within the low load range described hereinafter. The discharge energy is set to a level that increases the lower the load is (Figure 10C). The intake valve close timing (IVC) is set to be nearer to the advance angle than the bottom dead center (BDC), and the operation proceeds according to the Miller cycle. This timing is set to an angle that is more advanced the lower the load is (Figure 10D). The mechanical compression ratio is set to a high level (Figure 10E).

[0074] The range of low load will now be discussed. In a low load range in which the load is greater than in the extremely low load range, the air-fuel ratio A/F is set to decrease (i.e., become richer) as the load increases (Figure 10A). The barrier discharge start timing is set to the intake stroke when the load is low, is set to approach the retard angle as the load increases, and is set to the compression stroke when the load is high (Figure 10B). The reasons for these settings are described hereinafter. The discharge energy is set to a constant value (Figure 10C). The intake valve close timing (IVC) is set to a constant value nearer the retard angle than the bottom dead center (BDC) (Figure 10D). The mechanical compression ratio is set to a high level (Figure 10E).

[0075] The range of low to moderate load will now be discussed. In a low-to-moderate load range in which the load is greater than in the low load range, the air-fuel ratio A/F is set to decrease (i.e., become richer) as the load increases (Figure 10A). The barrier discharge start timing is set to lag much more than in the low load range, and is also set to approach the retard angle as the load increases (Figure 10B). The discharge energy is set to a constant value (Figure 10C). The intake valve close timing (IVC) is set to a constant value nearer to the lag angle than the bottom dead center (BDC) (Figure 10D). The mechanical compression ratio is set to be much less than in the extremely low load range or the low load range, and is also set to decrease as the load increases (Figure 10E).

[0076] The range of moderate to high load will now be discussed. In a moderate-to-high load range in which the load is greater than in the low-to-moderate load range, the air-fuel ratio A/F is set to decrease (i.e., become richer) as the load increases (Figure 10A). The barrier discharge start timing is set to approach the retard angle as the load increases (Figure 10B). The discharge energy is set to a constant value (Figure 10C). The intake valve close timing (IVC) is set to a constant value nearer to the retard angle than the bottom dead center (BDC) (Figure 10D). The mechanical compression ratio is set to be even less than in the low-to-moderate load range, and is also set to decrease as the load increases (Figure 10E).

[0077] The reasons for setting the control map in the above manner will be described herein. In the low load range, the barrier discharge start timing is set to the intake stroke when the load is low, is set to approach the retard angle as the load increases, and is set to the compression stroke when the load is high (Figure 10B). The reasons for these settings will be explained with reference to Figure 11.

[0078] Figure 11 is a graph showing the variation in the heat generation rate outside of the barrier discharge start timing. Line A in the diagram is shown as a comparative example, and is a line indicating variation in the heat generation rate when a barrier discharge is not performed (i.e., radicals are not generated). It can be seen from line A that the peak of the heat generation rate is suppressed at the crank angle θa. The heat generation rate is substantially symmetrical before and after this peak, and the crank angle MBθ50% (discussed below) at which the mass combustion ratio is 50% substantially coincides with θa.

[0079] Line B in the diagram is a line indicating variation in the heat generation rate when a barrier discharge is initiated during the compression stroke (for example, 135 deg BTDC). It can be seen from line B that the peak of the heat generation rate is suppressed at the crank angle θb nearer to the advance angle than when the barrier discharge was not performed (line A), and the heat generation rate rises more rapidly than when the barrier discharge was not performed (line A). The heat generation rate is substantially symmetrical before and after this peak, and the crank angle MBθ 50%, at which the mass combustion ratio is 50%, substantially coincides with θb.

[0080] Line C in the diagram is a line indicating variation in the heat generation rate when a barrier discharge is initiated during the intake stroke (for example, 270 deg BTCD). It can be seen from line C that the peak of the heat generation rate is suppressed at the crank angle θc even nearer to the advance angle than when the barrier discharge was initiated during the compression stroke (line B), and the variation is steep. The heat generation rate is substantially symmetrical before and after this peak, and the crank angle MBθ 50%, at which the mass combustion ratio is 50%, substantially coincides with θc.

[0081] Figures 12A-C contain drawings schematically depicting the state in which radicals are distributed within the cylinder, which is the result of analyzing the reasons that bring about a state such as in Figure 11. The radicals are schematically depicted by the dots in the drawings. Research has shown that differences in the variation in the heat generation rate brought about by the barrier discharge start timing are caused by the state in which radicals are distributed within the cylinder, as shown in Figure 11.

[0082] When a barrier discharge is not performed (i.e., when radicals are not generated), there is naturally no distribution of radicals in the cylinder 31a (Figure 12A). When the air-fuel mixture undergoes compression ignition while no radicals are distributed, the heat generation rate varies comparatively slowly, as shown by line A in Figure 11.

[0083] In cases in which a barrier discharge is initiated during the intake stroke, it can be seen that radicals are distributed throughout substantially the entire cylinder 31a immediately before ignition, as shown in Figure 12C. This is because there is a long timing from the time when the barrier discharge device 70 performs a barrier discharge to generate radicals until the time of ignition, and the radicals are therefore carried by the intake flow to be widely dispersed throughout the cylinder 31 a. When compression ignition takes place in the state in which the radicals are widely distributed, the air-fuel mixture combusts substantially all at once throughout the entire cylinder 31 a. The radicals are in a state in which molecular dissociation is induced by collisions of high-energy electrons with fuel or air molecules. Such radicals have the characteristic of readily inducing oxidation reactions (i.e., combustion) and creating chain oxidation reactions. The radicals undergo combustion substantially all at once throughout the entire cylinder 31 a when the pressure in the cylinder increases while radicals having such characteristics are dispersed throughout the entire cylinder 31a. Research has shown that the heat generation rate also rises suddenly because a combustion reaction takes place in this manner throughout the entire cylinder 31 a.

[0084] Initiating a barrier discharge during the compression stroke brings about an intermediate state in the cylinder 31 a immediately before ignition, that is, a state between the case of no barrier discharge (Figure 12A) and the case in which a barrier discharge is initiated during the intake stroke (Figure 12C). In the intermediate state, fewer radicals are distributed in the vicinity of the barrier discharge device 70 (Figure 12B). This is because there is a short timing from the time when the barrier discharge device 70 performs a barrier discharge to generate radicals until the time of ignition, and the radicals are therefore unable to widely disperse. When compression ignition takes place in the state in which the radicals are dispersed in the vicinity of the barrier discharge device 70, the combustion process first involves the radicals and then spreads to the surrounding radical-free air-fuel mixture. It is because of this type of mechanism that line B is an intermediate line between line A and line C.

[0085] Figure 13 is a graph showing the relationship between the barrier discharge start timing and the crank angle at which the mass combustion ratio is 50%.

[0086] As described above, varying the barrier discharge start timing causes a change in the crank angle MBθ 50% at which the mass combustion ratio is 50%. In other words, the auto-ignition properties change. This relationship is plotted in Figure 13. Up until the barrier discharge start timing reaches approximately 270 deg BTDC, the crank angle MBθ 50% at which the mass combustion ratio is 50% advances as the barrier discharge start timing is advanced. In other words, auto-ignition properties are improved. When the barrier discharge start timing is advanced to 270 deg BTDC or greater, the crank angle MBθ 50% at which the mass combustion ratio is 50% lags behind as the barrier discharge start timing is advanced.

[0087] The following are thought to be the reasons that the crank angle MBθ 50% at which the mass combustion ratio is 50% advances the farthest (i.e., auto-ignition properties are best) when the barrier discharge start timing is approximately 270 deg BTDC. Specifically, there is an overlap between timings in which the intake valve and exhaust valve of the engine are normally opened and closed. It is believed that initiating a barrier discharge after the exhaust valve has closed causes the air-fuel mixture drawn in through the intake valve to scatter more readily and auto-ignition properties to improve in comparison with a case in which a barrier discharge is initiated during the timing in which the exhaust valve has not yet closed. It is also believed that the air-fuel mixture readily scatters and auto-ignition properties improve because the rate of air intake is higher during the latter half of the downward movement of the piston than the first half. The barrier discharge part continuously performs a barrier discharge for a predetermined time (predetermined crank angle timing) following discharge initiation. The air flow rate decreases after the intake valve is closed. When a non-equilibrium plasma discharge is performed while the air flow rate has decreased, the radicals do not disperse as readily as when the air flow rate is high. Therefore, to efficiently disperse radicals within the cylinder, the end timing of the non-equilibrium plasma discharge is before the closing of the intake valve.

[0088] As can be seen from Figure 13, the heat generation timing (the crank angle MBθ 50% at which the mass combustion ratio is 50%) can be controlled by adjusting the barrier discharge start timing. In other words, the auto-ignition properties of the air-fuel mixture can be controlled by adjusting the barrier discharge start timing. As the auto-ignition properties improve, the operability at a lean air-fuel ratio improves as well. However, if the auto-ignition properties improve excessively when the air-fuel ratio is not particularly lean, there is a danger that knocking will occur. In view of this, the barrier discharge start timing is adjusted according to the air-fuel ratio (load).

[0089] As a comparative example, Figure 13 also shows a case in which radicals are generated by a sparkplug. It is clear from the diagram that even if radicals are generated by a sparkplug, there is little difference from cases in which radicals are not generated.

[0090] Based on the above knowledge, an electric discharge structure is provided which causes a barrier discharge to be initiated during the intake stroke so that radicals are widely distributed within the cylinder when the air-fuel ratio corresponds to an extremely diluted (lean) condition.

[0091] Depending on the operating state, there is a danger that the auto-ignition properties will be improved to an excess and that knocking will occur if the amount of radicals generated within the cylinder is too great or the radicals are too widely distributed. In view of this, the auto-ignition properties are adjusted by delaying the barrier discharge start timing as the load increases (as the amount of fuel increases and the air-fuel ratio corresponds to a richer mixture). The above factors are the reasons that the barrier discharge start timing is set to occur during the intake stroke when the load is low, is set to approach a retard angle as the load increases, and is set to occur during the compression stroke when the load is high (Figure 10B).

[0092] The mechanical compression ratio is set to a high level in a load range at or below a low load (Figure 10E) The reasons for these settings will now be described.

[0093] An engine having a multi-link variable compression ratio mechanism has the characteristic of having a longer timing in which the piston stays in proximity to the top dead center in comparison with a common engine in which the compression ratio is constant (hereinafter referred to as a "normal engine"). Due to this characteristic, an engine having a multi-link variable compression ratio mechanism, even at a high compression ratio, is less susceptible to knocking than a common engine is, comparatively high combustion energy can be obtained even with ultra-lean combustion, and stable combustion can be maintained.

[0094] This aspect is described with reference to Figure 14. Figure 14 contains a graph showing the piston behavior in a multi-link variable compression ratio mechanism, wherein the upper portion of Figure 14 is an enlarged view of the dotted line portion of the lower portion of the figure. In Figure 14, the thin solid lines indicate the piston behavior in the multi-link variable compression ratio mechanism engine having the same compression ratio as a normal engine.

[0095] If the time in which the piston is within a predetermined distance from the top dead center is defined as the timing in which the piston is in proximity to the top dead center, it is clear from Figure 14 that the multi-link variable compression ratio mechanism engine has a longer timing in which the piston is in proximity to the top dead center than does a normal engine having the same compression ratio. Specifically, in the multi-link variable compression ratio mechanism engine, the timing L1 in which the piston is in proximity to the top dead center at a high compression ratio is longer than the timing L2 in which the piston is in proximity to the top dead center at a low compression ratio. In other words, the inequality $L1 > L2$ is true in Figure 14.

[0096] Thus, the multi-link variable compression ratio mechanism engine has a longer timing in which the piston is in proximity to the top dead center than does a normal engine. Furthermore, the timing in which the piston is in proximity to the top dead center is longer than that observed at a high compression ratio. The fact that the piston is in proximity to the top dead center for a long time means that a high compression state is maintained for a long time during combustion. When a high compression state is maintained for a long time, knocking does not readily occur, and combustion is stable because comparatively high combustion energy can be obtained even during ultra-lean combustion.

[0097] Because of such characteristics, the multi-link variable compression ratio mechanism engine has the characteristics shown in Figure 15. Figure 15 is a graph showing the relationship between the air-fuel ratio and combustion stability. The thin line in the diagram denotes a normal engine, and the thick line denotes a multi-link variable compression ratio mechanism engine.

[0098] As can be seen from Figure 15, in a normal engine (compression ratio: about 8 to 12), the air-fuel ratio which can ensure combustion stability is about 22.

[0099] According to the multi-link variable compression ratio mechanism engine, the combustion stability limit is not compromised because the piston remains in proximity to the top dead center for a long time. Increasing the compression ratio (e.g., to about 18) makes it possible to obtain stable combustion even at an air-fuel ratio A/F of about 30. The above are the reasons the mechanical compression ratio is set to a high level in a load range at or below a low load (Figure 10E). The map load range in Figure 10 was set based on this knowledge.

[0100] Next, the reasons for selecting the settings in the extremely low load range in the control map will be described. In the extremely low load range, as described above, the intake valve close timing (IVC) is set nearer to the advance angle than in the bottom dead center (BDC), and the operation proceeds according to the Miller cycle.

[0101] The timing is set nearer to the advance angle at lower loads (Figure 10D). The filling efficiency of intake air is thereby reduced, the effective compression ratio is lowered, and pump loss is reduced. Since the combustion amount decreases with decreased load (the air-fuel ratio is substantially constant because the air intake amount also decreases), the air-fuel mixture loses auto-ignition properties. In view of this, the discharge energy is greatly increased at lower loads (Figure 10C). The map of the extremely low load range in Figure 10 was set based on the above knowledge. As shown, operation is possible even at extremely low load ranges.

[0102] Next, the reasons for the settings in the low-to-moderate load range of the control map will be described. In the low-to-moderate load range, as described above, the barrier discharge start timing lags to a considerably greater extent than in the low load range (Figure 10B). The mechanical compression ratio is set to be much lower than in the extremely low and low load ranges (Figure 10E).

[0103] In cases in which radicals are generated and combustion takes place by compression ignition, the air-fuel mixture has better auto-ignition properties. Therefore, when the load is greater and the amount of combustion increases, there is a possibility that the heat generation rate will suddenly increase to an excessive degree, as shown by line A in Figure 16. When the heat generation rate suddenly increases to an excessive degree in this manner, there is a danger that knocking will occur.

[0104] In view of this, in the present embodiment, when the load increases to within a low-to-moderate load range, the compression ratio is reduced so that the air-fuel mixture does not undergo compression ignition. It is designed so that volumetric ignition is performed by the barrier discharge part during the compression stroke. The fuel in the vicinity of the barrier discharge part thereby undergoes flame propagation. The remaining unburned air-fuel mixture is adiabatically compressed by the burned air-fuel mixture and is made to undergo auto-ignition. As a result, the heat generation rate varies as shown by line B in Figure 16 and does not suddenly increase to an excessive degree, and knocking does not occur. The map of the low-to-moderate load range in Figure 10 is set based on the above. Operation is thereby made possible even in a low-to-moderate load range.

[0105] Spark ignition is performed by the barrier discharge part at a moderate-to-high load or greater, whereby operation is possible even in a moderate-to-high load range.

[0106] Figures 17A and 17B contain graphs showing various effects of the present embodiment. In the present embodiment, it is possible to greatly expand the lean combustion limit because the barrier discharge start timing is appropriately controlled according to the operating state as described above.

[0107] In Figure 17A, plotting the correlation between the air-fuel ratio A/F (horizontal axis) and the fluctuation rate CPi (vertical axis) of the depicted average effective pressure results in line A in normal combustion by compression ignition. The lean combustion limit is the air-fuel ratio AFa.

[0108] Line B depicts cases in which radicals are generated by a sparkplug, and combustion occurs by compression ignition. The lean combustion limit is the air-fuel ratio of AFb, and is somewhat leaner than the air-fuel ratio AFa of the lean combustion limit in normal cases.

[0109] Line C depicts cases in which radicals are generated by the barrier discharge part, and combustion occurs by compression ignition. The lean combustion limit is the air-fuel ratio of AFc. The lean combustion limit can be greatly expanded in comparison with the air-fuel ratio AFa of the lean combustion limit in normal cases and in comparison with the air-fuel ratio AFb of the lean combustion limit in generation of radicals by a sparkplug and combustion by compression ignition. As described above, the operation shown by the dashed lines can be arbitrarily selected because it is possible to control the crank angle MBθ 50% at which the mass combustion ratio is 50% by adjusting the barrier discharge start timing. If the lean combustion limit is expanded, the fuel consumption rate ISFC can be reduced as shown in Figure 17B. The present embodiment makes it possible to reduce the fuel consumption rate regardless of the load, and to improve fuel consumption.

[0110] In the present embodiment, the first electrode composed of a long thin conductive material and the dielectric material for covering the first electrode allow a barrier discharge to be performed in which non-equilibrium plasma is discharged and radicals can be generated within a cylinder. Therefore, the auto-ignition properties of an air-fuel mixture during the compression stroke can be improved, the fuel consumption rate can consequently be reduced regardless of the load, and fuel consumption can also be improved.

[0111] Referring now to Figure 18, an internal combustion engine electric discharge structure in accordance with a second embodiment will now be explained. Basically, in this second embodiment, the internal combustion engine electric discharge structure of the first embodiment is replaced in Figure 1 with a modified structure as discussed below. In view of the similarity between the first and second embodiments, the parts of the second embodiment that are identical to the parts of the first embodiment will be given the same reference numerals as the parts of the first embodiment. Moreover, the descriptions of the parts of the second embodiment that are identical to the parts of the first embodiment can be omitted for the sake of brevity.

[0112] Figure 18 is a simplified schematic cross-sectional view showing the operational configuration of the engine having an electric discharge structure in accordance with a second embodiment. The engine 1 having a barrier discharge function of the first embodiment was a so-called port-injection engine in which the fuel injection valve 65 was provided to the intake port, but the electric discharge structure can also be applied to a direct fuel-injection engine such as the one shown in Figure 18, in which fuel is directly injected into the cylinder.

[0113] In this type of direct fuel-injection engine, the air-fuel mixture is stratified only in the vicinity of the barrier discharge device 70 as shown in Figure 19 to make operation possible even with a lean air-fuel ratio. Generating radicals in this type of lean air-fuel mixture allows the lean combustion limit to be expanded, the fuel consumption rate to be reduced, and fuel consumption to be improved.

[0114] An example of an operation map for the engine having such a barrier discharge function is shown in Figures 20A-20E. An interval in which a barrier discharge is not performed is provided in the vicinity of a comparatively high load within the low load range (Figures 20A and 20B). In the low load range, a high compression ratio is set by the variable compression ratio mechanism, and knocking does not readily occur. Therefore, there is an operation range in which lean combustion is possible even though a barrier discharge is not performed. When a barrier discharge is performed in such an operating range, there is a danger that auto-ignition properties will improve excessively and that knocking will occur. In view of this, a barrier discharge is not performed in the vicinity of comparatively high loads within the low load range.

[0115] In an extremely low load range in which the load is lower than in the low load range, a stratified operation is performed (Figure 20D) and the air-fuel ratio A/F is made leaner (sparser) according to the load (Figure 20A). A barrier discharge is performed because the auto-ignition properties must be improved along with the increase in sparseness. The barrier discharge start timing is set to occur during the intake stroke, wherein the effects of auto-ignition properties improvement are high (Figure 20B). The auto-ignition properties are improved by increasing the discharge energy along with the increase in sparseness (Figure 20C).

[0116] By using the present embodiment, the invention can be carried out even with a direct fuel-injection engine, the fuel consumption rate can be reduced regardless of the load, and fuel consumption can be improved.

[0117] Referring now to Figure 21, an internal combustion engine electric discharge structure in accordance with a third embodiment will now be explained. Basically, in this third embodiment, the internal combustion engine electric discharge structure of the first embodiment is replaced in Figure 1 with a modified structure as discussed below. In view of the similarity between the first and second embodiments, the parts of the third embodiment that are identical to the parts of the first embodiment will be given the same reference numerals as the parts of the first embodiment. Moreover, the descriptions of the parts of the third embodiment that are identical to the parts of the first embodiment can be omitted for the sake of brevity.

[0118] Figure 21 is a simplified schematic cross-sectional view showing the third embodiment of an engine having a barrier discharge function. In the barrier discharge device 70 of the present embodiment, a dielectric layer (insulating layer) 73 is formed on the inner periphery of the tubular electrode 72, and the central electrode 71 is exposed. The distal end of the dielectric layer (insulating layer) 73 protrudes farther toward the combustion chamber than does the distal end of the tubular electrode 72 or the distal end of the central electrode 71. This is because such a configuration makes it possible to suppress the occurrence of a thermal plasma discharge between the distal end of the tubular electrode 72 and the distal end of the central electrode 71, even in cases in which the discharge energy of a non-equilibrium plasma discharge has been increased. The dielectric layer 73 acts as a capacitor in the configuration of the present embodiment as well, and the same effects as in the first embodiment are obtained.

[0119] Referring now to Figures 22A and 22B, an internal combustion engine electric discharge structure in accordance with a fourth embodiment will now be explained. Basically, in this fourth embodiment, the internal combustion engine electric discharge structure of the first embodiment is replaced in Figure 1 with a modified structure as discussed below. In view of the similarity between the first and fourth embodiments, the parts of the fourth embodiment that are identical to the parts of the first embodiment will be given the same reference numerals as the parts of the first embodiment. Moreover, the descriptions of the parts of the fourth embodiment that are identical to the parts of the first embodiment can be omitted for the sake of brevity.

[0120] Figures 22A and 22B contain simplified schematic cross-sectional views showing the fourth embodiment of the engine having a barrier discharge function. In the barrier discharge device 70 of the present embodiment, in contrast to the first embodiment, the central electrode 71 protrudes into the combustion chamber.

[0121] Thus, the barrier discharge device 70 forms a barrier discharge within the combustion chamber as shown in Figure 22A. In the present embodiment, the top surface of the piston 32 or the inside wall surface of the cylinder head functions as an electrode. Specifically, in the present embodiment, a barrier discharge is performed and radicals are generated in the area A between the top surface of the piston 32 and the dielectric layer (insulating layer) 73 of the central electrode 71, or in the area B between the inside wall surface of the cylinder head and the dielectric layer (insulating layer) 73. Whether the barrier discharge is performed in area A or B is determined by the position of the piston 32 when an AC voltage is applied to the barrier discharge device 70. In view of this, the discharge area of barrier discharge can be selected by controlling the application timing of the AC voltage applied to the barrier discharge device 70.

[0122] A concave part can be formed in the top surface of the piston 32 as shown in Figure 22B, and the configuration can be designed so that barrier discharge is performed between the concave part and the distal end of the dielectric material (insulating material) 73 of the central electrode 71.

[0123] Referring now to Figures 23A and 23B, an internal combustion engine electric discharge structure in accordance with a fifth embodiment will now be explained. Basically, in this fifth embodiment, the internal combustion engine electric discharge structure of the first embodiment is replaced in Figure 1 with a modified structure as discussed below. In view of the similarity between the first and fifth embodiments, the parts of the fifth embodiment that are identical to the parts of the first embodiment will be given the same reference numerals as the parts of the first embodiment. Moreover, the descriptions of the parts of the fifth embodiment that are identical to the parts of the first embodiment can be omitted for the sake of brevity.

[0124] Figures 23A and 23B contain simplified schematic cross-sectional views showing the fifth embodiment of the engine having a barrier discharge function. In the barrier discharge device 70 of the present embodiment, the dielectric material (insulating material) 73 is shorter in comparison with the fourth embodiment, and the central electrode 71 is exposed within the combustion chamber. A dielectric layer (insulating layer) 32a is also formed on the top surface of the piston 32.

[0125] Thus, the barrier discharge device 70 performs a barrier discharge within the combustion chamber as shown in Figure 23A. Specifically, a barrier discharge is performed and radicals are generated in the area A between the distal end of the central electrode 71 and the dielectric layer (insulating layer) 32a on the top surface of the piston 32.

[0126] If a concave part is formed in the top surface of the piston 32, and the dielectric layer (insulating layer) 32a is formed in the inner periphery of the concave part as shown in Figure 23B, a barrier discharge is performed between the dielectric layer (insulating layer) 32a and the distal end of the central electrode 71.

[0127] Although alternating current corresponding to the operating state of the engine is applied to the barrier discharge device 70, but the alternating current is not limited to a sine curve (Figure 24A). A bipolar multiple pulse power source can also be used, such as is shown in Figure 24B.

[0128] Also in the above descriptions, a multi-link mechanism was shown as the variable compression ratio mechanism, but other possible examples include, e.g., a mechanism in which a hydraulic device is incorporated into the piston as such to adjust the height of the top surface of the piston, a mechanism in which the distance between the cylinder head and the cylinder block can be adjusted, and a mechanism in which the piston height can be adjusted by offsetting the center of the crankshaft.

[0129] Furthermore, the mechanism for adjusting the valve timing of the intake valve can also be, e.g., an oscillating cam which uses a link (Japanese Laid-Open Patent Application No. 2000-213314), a mechanism in which the cam is twisted in the manner of a vane-type variable valve timing system (Japanese Laid-Open Patent Application No. 9-60508), a system in which a switch is made between two types of cams having different timings in the manner of a direct variable valve timing system (Japanese Laid-Open Patent Application No. 4-17706), or the like.

[0130] In understanding the scope of the present invention, the term "comprising" and its derivatives, as used herein, are intended to be open ended terms that specify the presence of the stated features, elements, components, groups, integers, and/or steps, but do not exclude the presence of other unstated features, elements, components, groups, integers and/or steps. The foregoing also applies to words having similar meanings such as the terms, "including", "having" and their derivatives. Also, the terms "part," "section," "portion," "member" or "element" when used in the singular can have the dual meaning of a single part or a plurality of parts. The terms of degree such as "substantially", "about" and "approximately" as used herein mean a reasonable amount of deviation of the modified term such that the end result is not significantly changed.

[0131] While only selected embodiments have been chosen to illustrate the present invention, it will be apparent to those skilled in the art from this disclosure that various changes and modifications can be made herein without departing from the scope of the invention as defined in the appended claims. For example, the size, shape, location or orientation of the various components can be changed as needed and/or desired. Components that are shown directly connected or contacting each other can have intermediate structures disposed between them. The functions of one element can be performed by two, and vice versa. The structures and functions of one embodiment can be adopted in another embodiment. It is not necessary for all advantages to be present in a particular embodiment at the same time. Every feature which is unique from the prior art, alone or in combination with other features, also should be considered a separate description of further inventions by the applicant, including the structural and/or functional concepts embodied by such features. Thus, the foregoing descriptions of the embodiments according to the present invention are provided for illustration only, and not for the purpose of limiting the invention as defined by the appended claims and their equivalents.

[0132] This application claims priority from Japanese Patent Application No. 2007-298294, filed 16th November 2007, the contents of which are expressly incorporated herein by reference.

Claims

1. An apparatus for an internal combustion engine, comprising:

    a first electrode including a first voltage receiving end and a second engine attachment end with a conductive material that discharges non-equilibrium plasma by a barrier discharge; and
    a dielectric material covering the first electrode.

2. An apparatus as claimed in claim 1, comprising a second electrode facing the first electrode on a periphery of the dielectric material.

3. An apparatus as claimed in claim 2, wherein the second electrode includes a tubular electrode surrounding at least a portion of the first electrode.

4. An apparatus as claimed in claim 2 or claim 3, comprising a cylinder head having the second electrode attached thereto, with the first electrode including a linear central electrode.

5. An apparatus as claimed in any of claims 2 to 4, wherein:

    the first electrode includes a linear central electrode; and
    the second electrode is disposed as at least part of one of a wall surface of a combustion chamber and a top surface of a piston.

6. An apparatus as claimed in any of claims 2 to 5, comprising:

    a fuel injection valve for supplying fuel into a combustion chamber of the internal combustion engine; and
    a voltage application device operatively coupled to the first voltage receiving end of the first electrode for applying a voltage between the first electrode and the second electrode, such that the non-equilibrium plasma generates radicals within the combustion chamber before an air-fuel mixture in the combustion chamber undergoes auto-ignition.

7. An apparatus as claimed in claim 6, comprising a control unit operatively coupled to the voltage application device and arranged to:

    vary a discharge start timing of the non-equilibrium plasma discharge in accordance with a mechanical load of the internal combustion engine;
    set a discharge start timing of the non-equilibrium plasma discharge to occur after an intake valve has opened;
    set a discharge ending timing of the non-equilibrium plasma discharge to occur before an intake valve has closed; and/or
    set a discharge energy of the non-equilibrium plasma discharge such that the discharge energy increases as the mechanical load of the internal combustion engine becomes lower when the mechanical load of the internal combustion engine is in a low load range.

8. An apparatus as claimed in claim 7, wherein the control unit is arranged to set the discharge start timing of the non-equilibrium plasma discharge to occur:

    during an intake stroke when the mechanical load of the internal combustion engine is comparatively low; or
    during a compression stroke when the mechanical load of the internal combustion engine is comparatively high.

9. An apparatus as claimed in claim 7 or claim 8, wherein the control unit is arranged to set the discharge start timing of the non-equilibrium plasma discharge to be:

    increasingly advanced as the mechanical load of the internal combustion engine becomes lower; and/or
    increasingly delayed as the mechanical load of the internal combustion engine increases.

10. An apparatus as claimed in any of claims 7 to 9, wherein the control unit is arranged to increase the discharge energy of non-equilibrium plasma discharge by at least one method selected from:

    increasing a voltage value of an AC voltage applied between the first and second electrodes;
    increasing a frequency of the AC voltage applied between the first and second electrodes; and
    increasing an application duration of the AC voltage applied between the first and second electrodes.

11. An apparatus as claimed in any of claims 6 to 10, comprising:

    a variable compression ratio mechanism arranged to change a mechanical compression ratio of the internal combustion engine; and
    a control unit operatively coupled to the variable

compression ratio mechanism to reduce the mechanical compression ratio so that an air-fuel mixture does not undergo compression ignition when a mechanical load of the internal combustion engine is in a high load range, and volumetric ignition is performed.

12. An apparatus as claimed in any of claims 6 to 11, comprising a fuel injection control unit operatively coupled to the fuel injection valve to control injection of fuel directly into a cylinder of the internal combustion engine such that a stratified air-fuel mixture is formed in the cylinder when a mechanical load of the internal combustion engine is in a low load range.

13. A method for controlling an operating state of an internal combustion engine, comprising:

determining a mechanical load of the internal combustion engine;
injecting fuel into a combustion chamber of the internal combustion engine;
applying a voltage to an electric discharge device having a first electrode and a second electrode to produce a non-equilibrium plasma discharge generating radicals within the combustion chamber before an air-fuel mixture of the fuel undergoes auto-ignition; and
setting a discharge start timing of the non-equilibrium plasma discharge such that the discharge start timing varies in accordance with the mechanical load.

14. An internal combustion engine having an apparatus as claimed in any of claims 1 to 12.

15. A vehicle having an apparatus as claimed in any of claims 1 to 12 or an engine as claimed in claim 14.

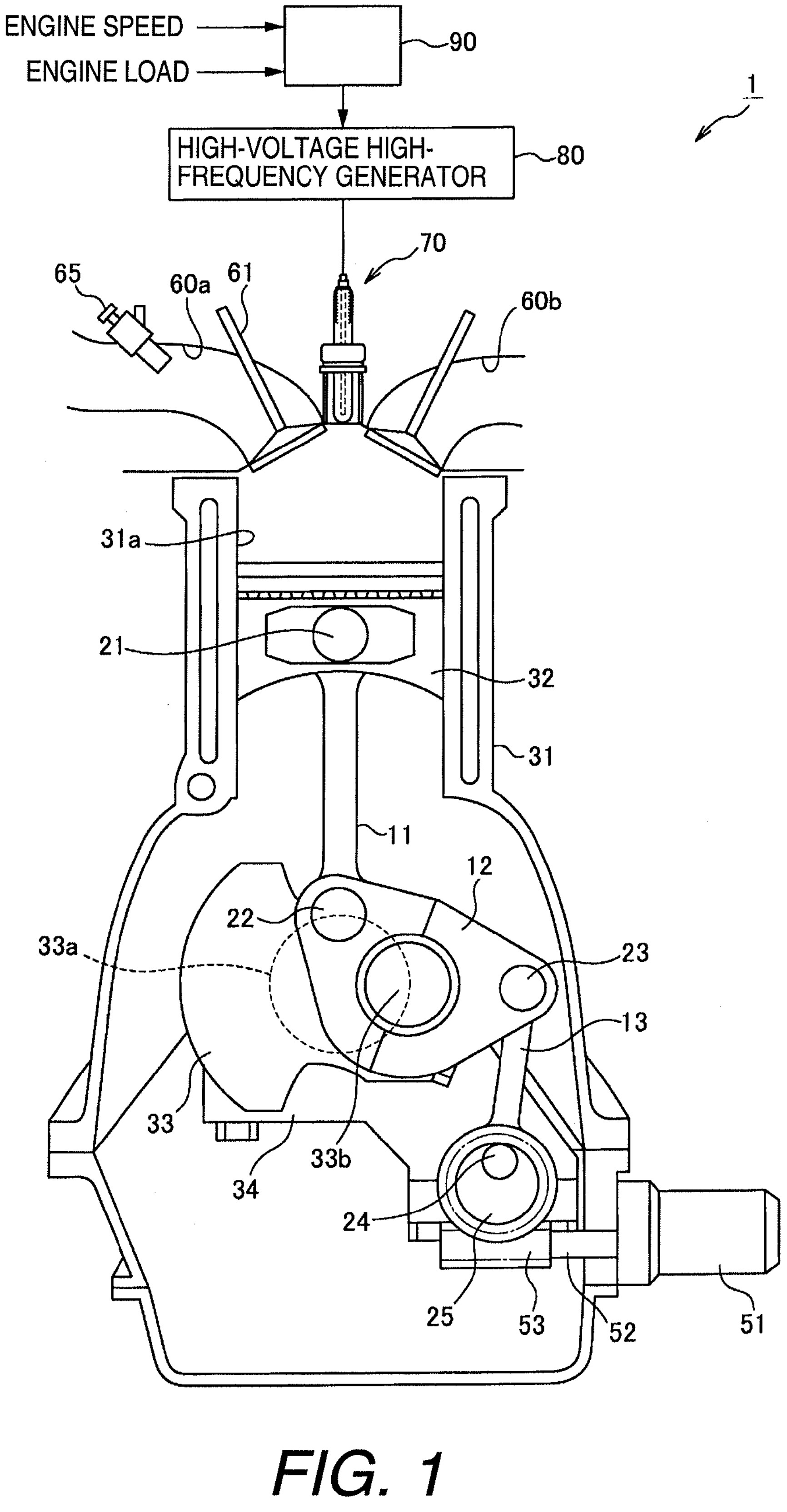

ENGINE SPEED
ENGINE LOAD
90
1
HIGH-VOLTAGE HIGH-FREQUENCY GENERATOR
80
70
65
60a
61
60b
31a
21
32
31
11
12
22
33a
23
13
33
33b
34
24
25
53
52
51

FIG. 1

FIG. 2A

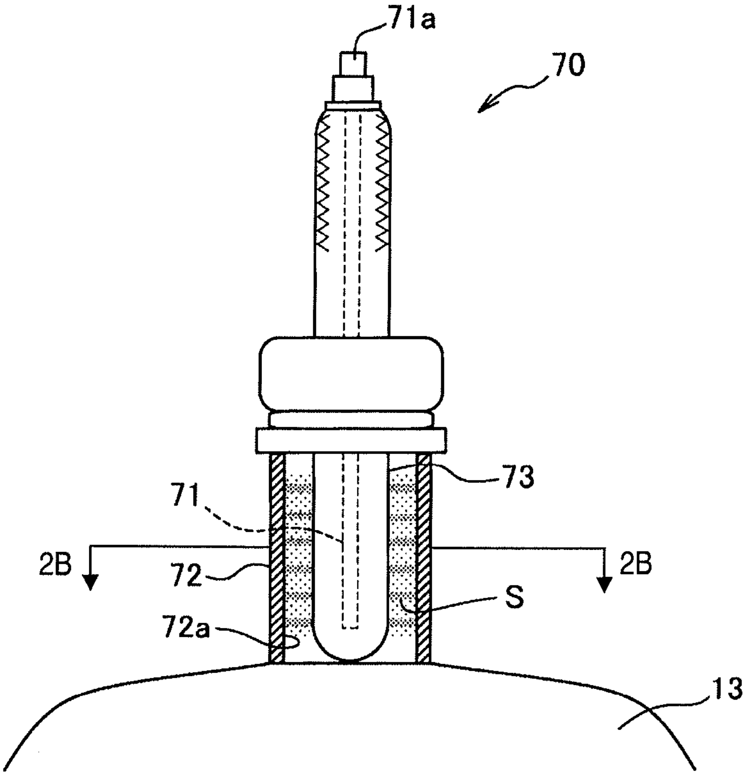
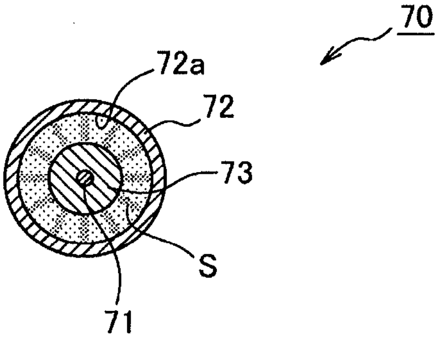
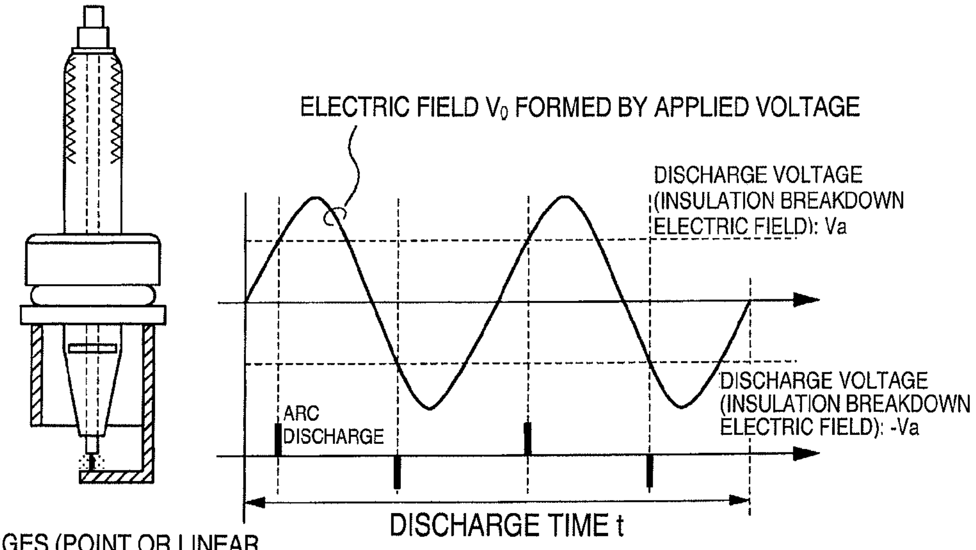


FIG. 2B



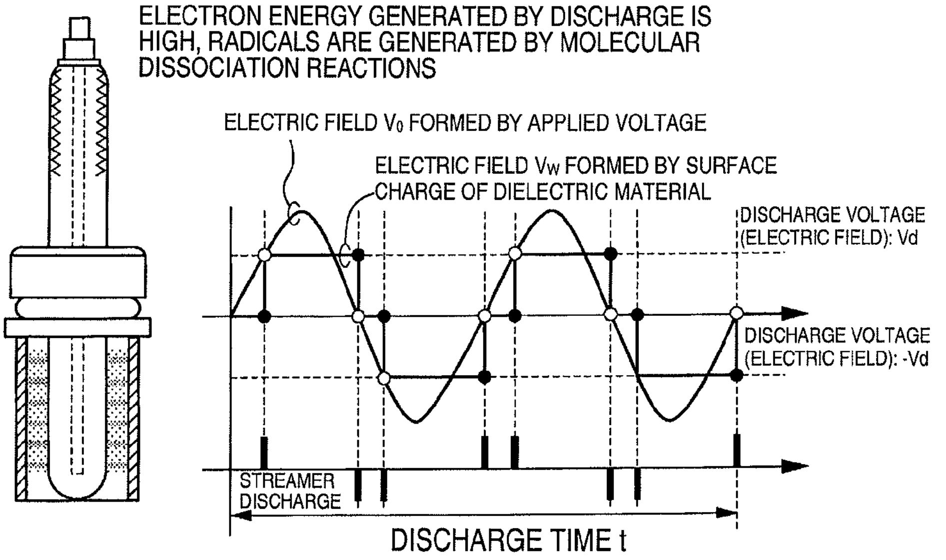


FOUR DISCHARGES (POINT OR LINEAR DISCHARGES) OCCUR ONLY AT THIS LOCATION DURING DISCHARGE TIME t

DISCHARGE BEGINS AT $V_0 = Va$

FIG. 3A

ELECTRON ENERGY GENERATED BY DISCHARGE IS HIGH, RADICALS ARE GENERATED BY MOLECULAR DISSOCIATION REACTIONS



DISCHARGED 8 TIMES WITHIN TIME t

DISCHARGE BEGINS AT $|V_0 - V_W| = Vd$

STREAMERS DISCHARGED 8 TIMES WITHIN TIME t ARE FORMED AT PLURALITY OF LOCATIONS (VOLUMETRIC DISCHARGE) IN WIDE-RANGE SPATIAL AREA BETWEEN DIELECTRIC MATERIAL SURFACE AND GROUNDING ELECTRODE

FIG. 3B

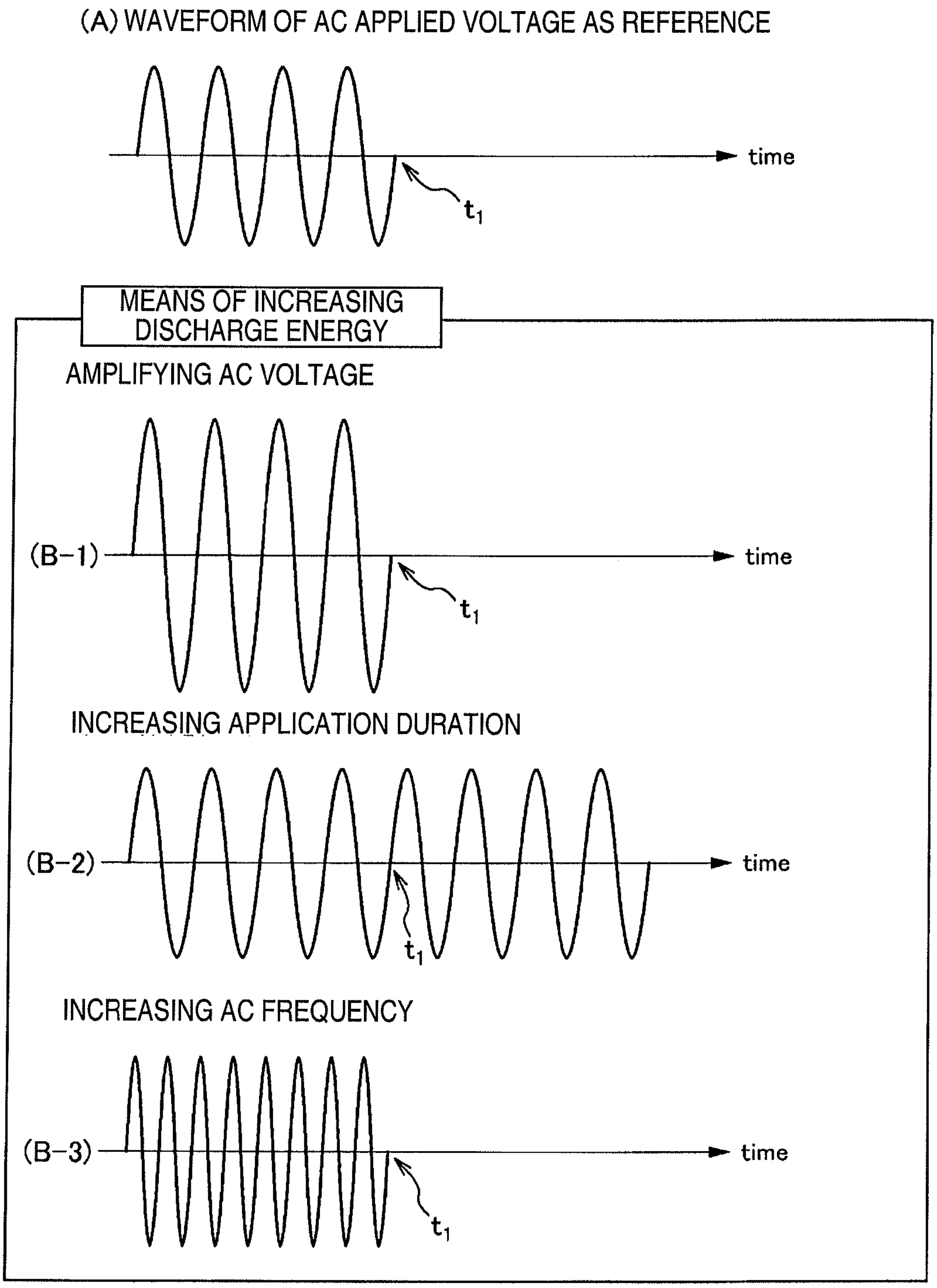

(A) WAVEFORM OF AC APPLIED VOLTAGE AS REFERENCE
time
t_1
MEANS OF INCREASING DISCHARGE ENERGY
AMPLIFYING AC VOLTAGE
(B-1)
time
t_1
INCREASING APPLICATION DURATION
(B-2)
time
t_1
INCREASING AC FREQUENCY
(B-3)
time
t_1

FIG. 4

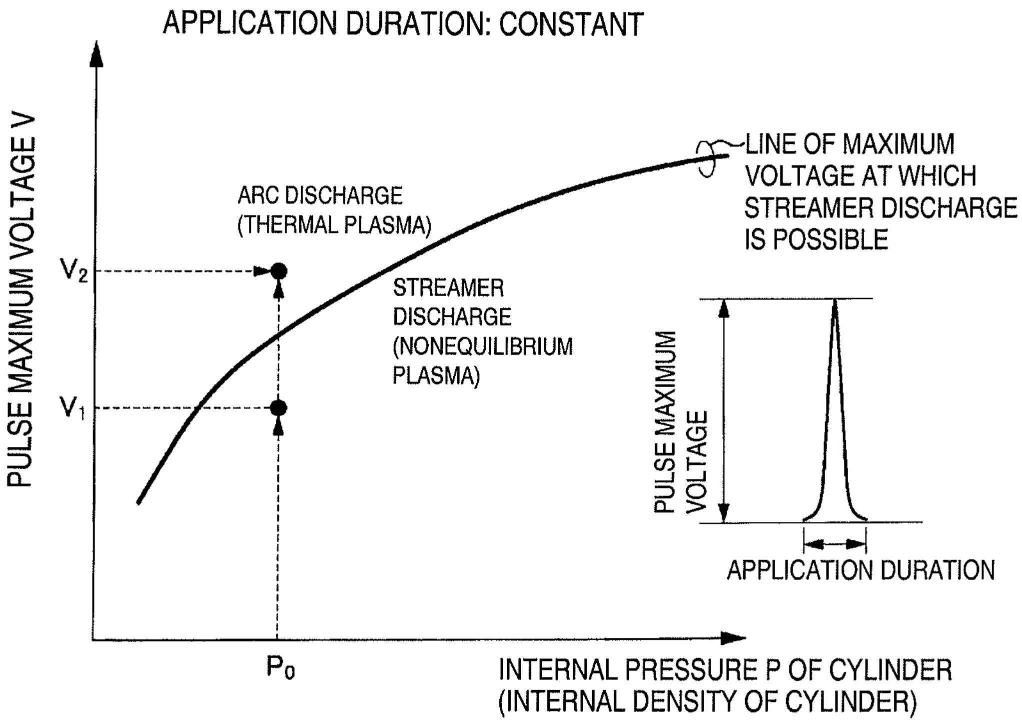

APPLICATION DURATION: CONSTANT
PULSE MAXIMUM VOLTAGE V
V2
V1
P0
INTERNAL PRESSURE P OF CYLINDER
(INTERNAL DENSITY OF CYLINDER)
ARC DISCHARGE
(THERMAL PLASMA)
STREAMER
DISCHARGE
(NONEQUILIBRIUM
PLASMA)
LINE OF MAXIMUM
VOLTAGE AT WHICH
STREAMER DISCHARGE
IS POSSIBLE
PULSE MAXIMUM
VOLTAGE
APPLICATION DURATION

FIG. 5

FIG.6A

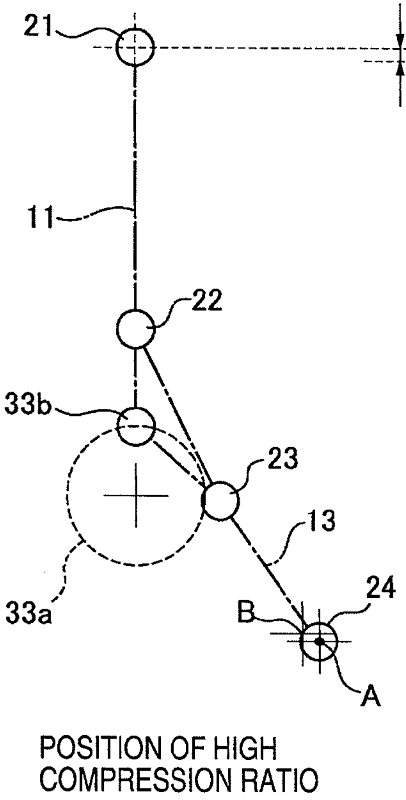


FIG.6B

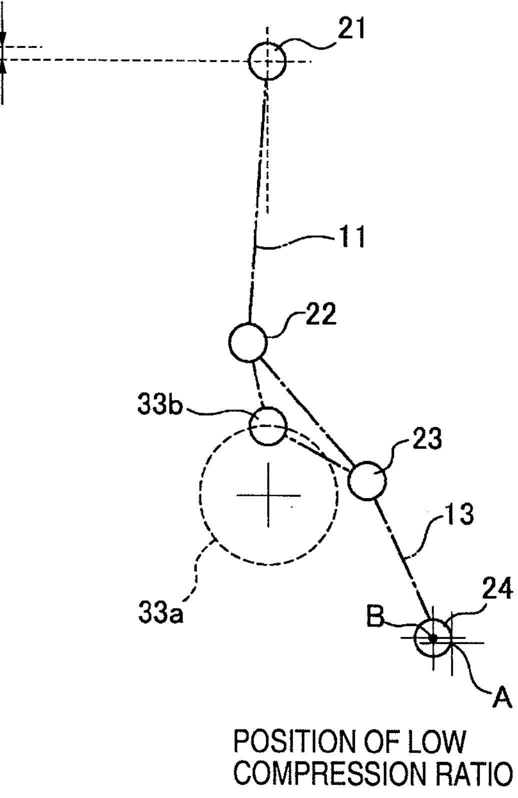
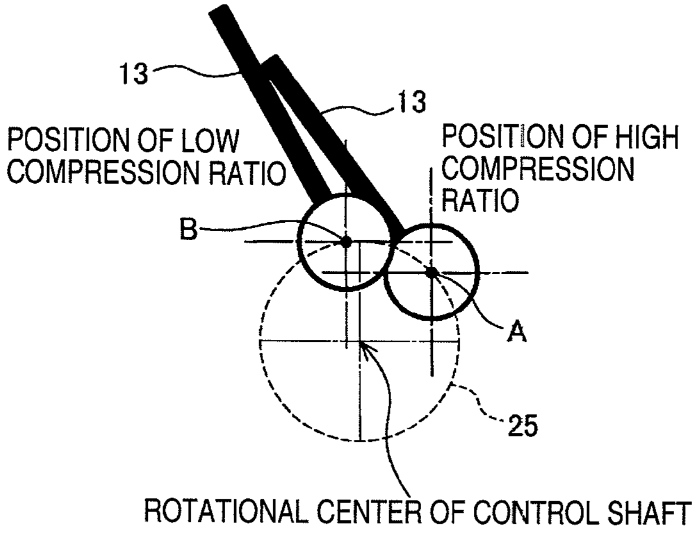


FIG.6C



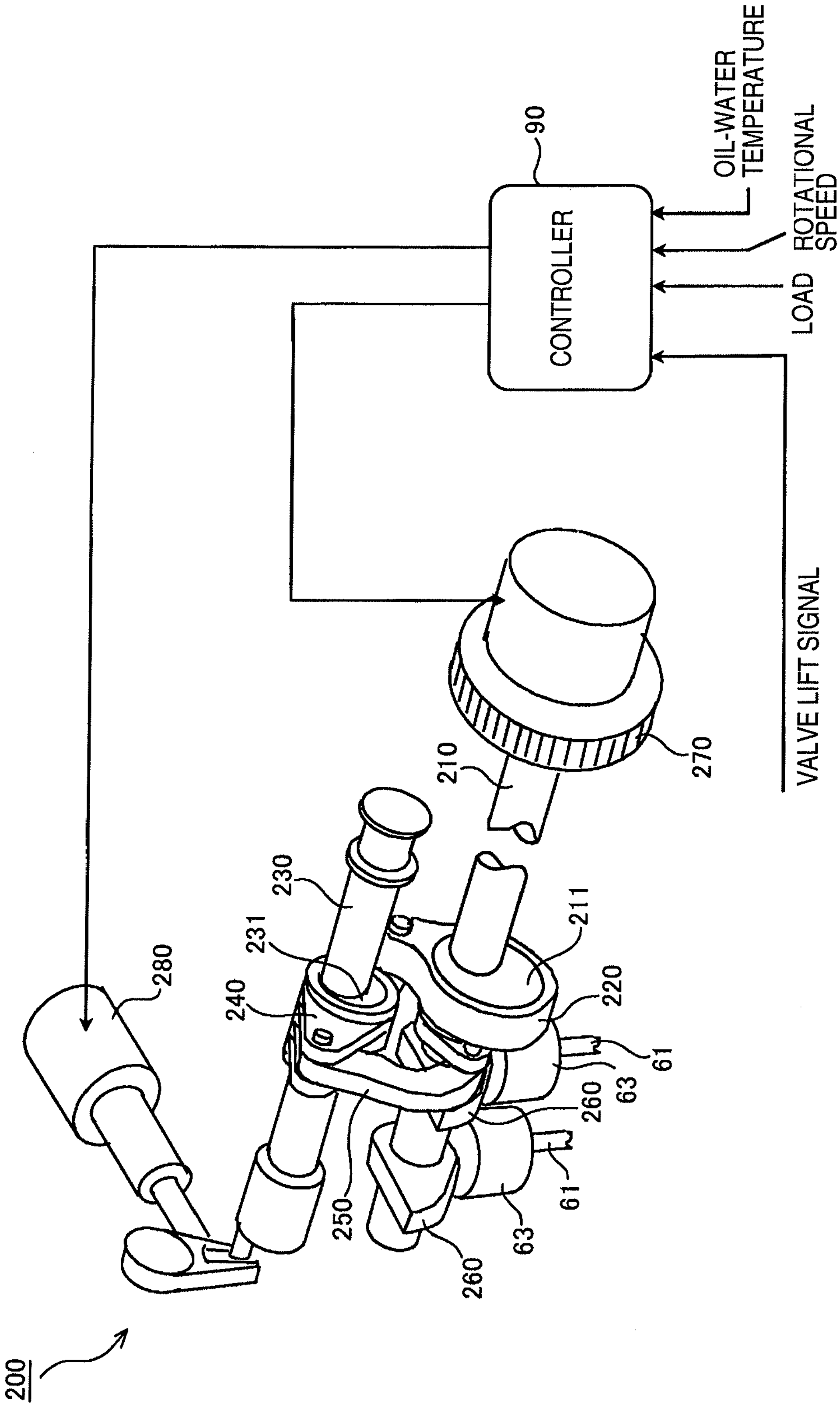

200
280
240
231
230
250
260
63
61
260
63
61
220
211
210
270
90
CONTROLLER
OIL-WATER TEMPERATURE
ROTATIONAL SPEED
LOAD
VALVE LIFT SIGNAL

FIG.7

FIG. 8A

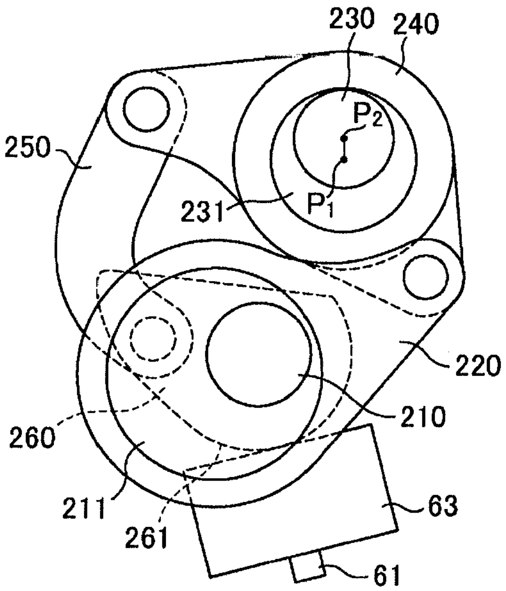


FIG. 8B

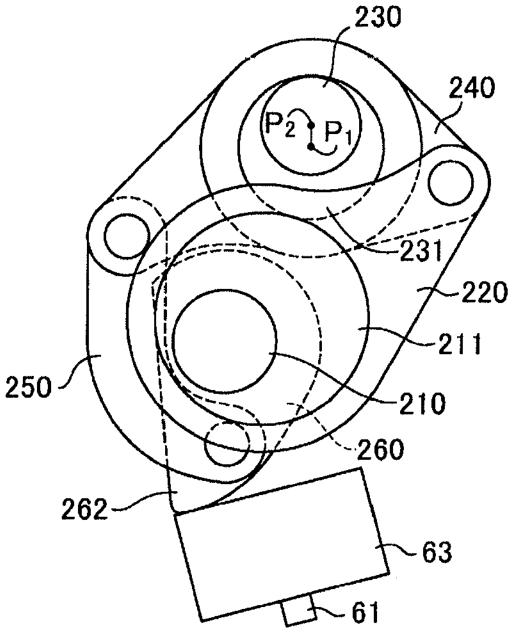


FIG. 8C

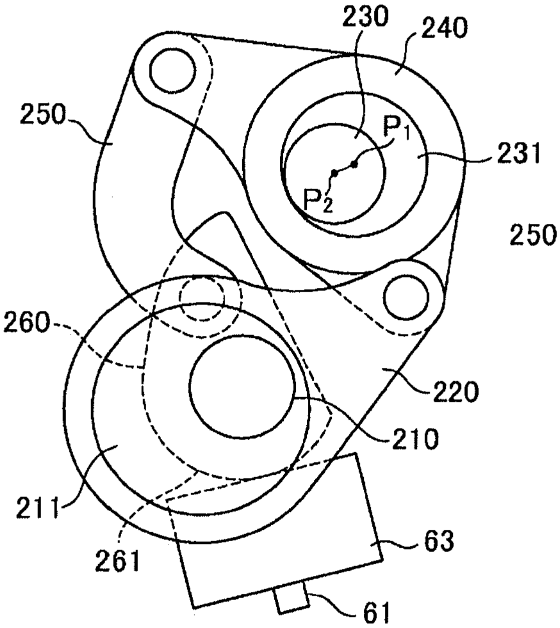
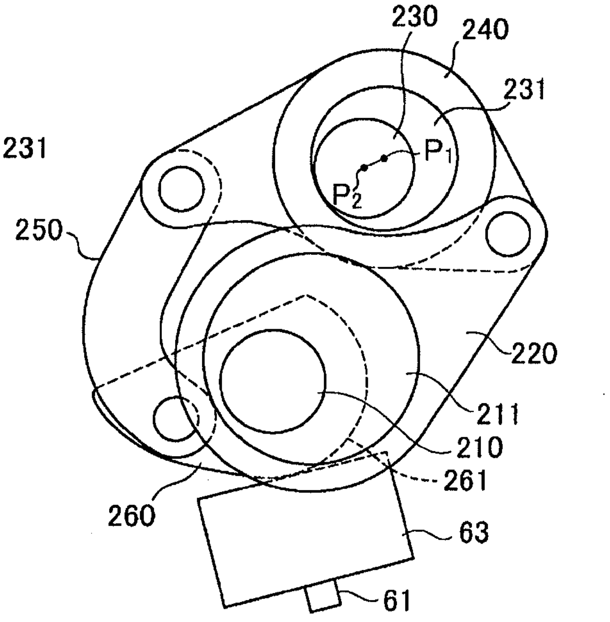


FIG. 8D



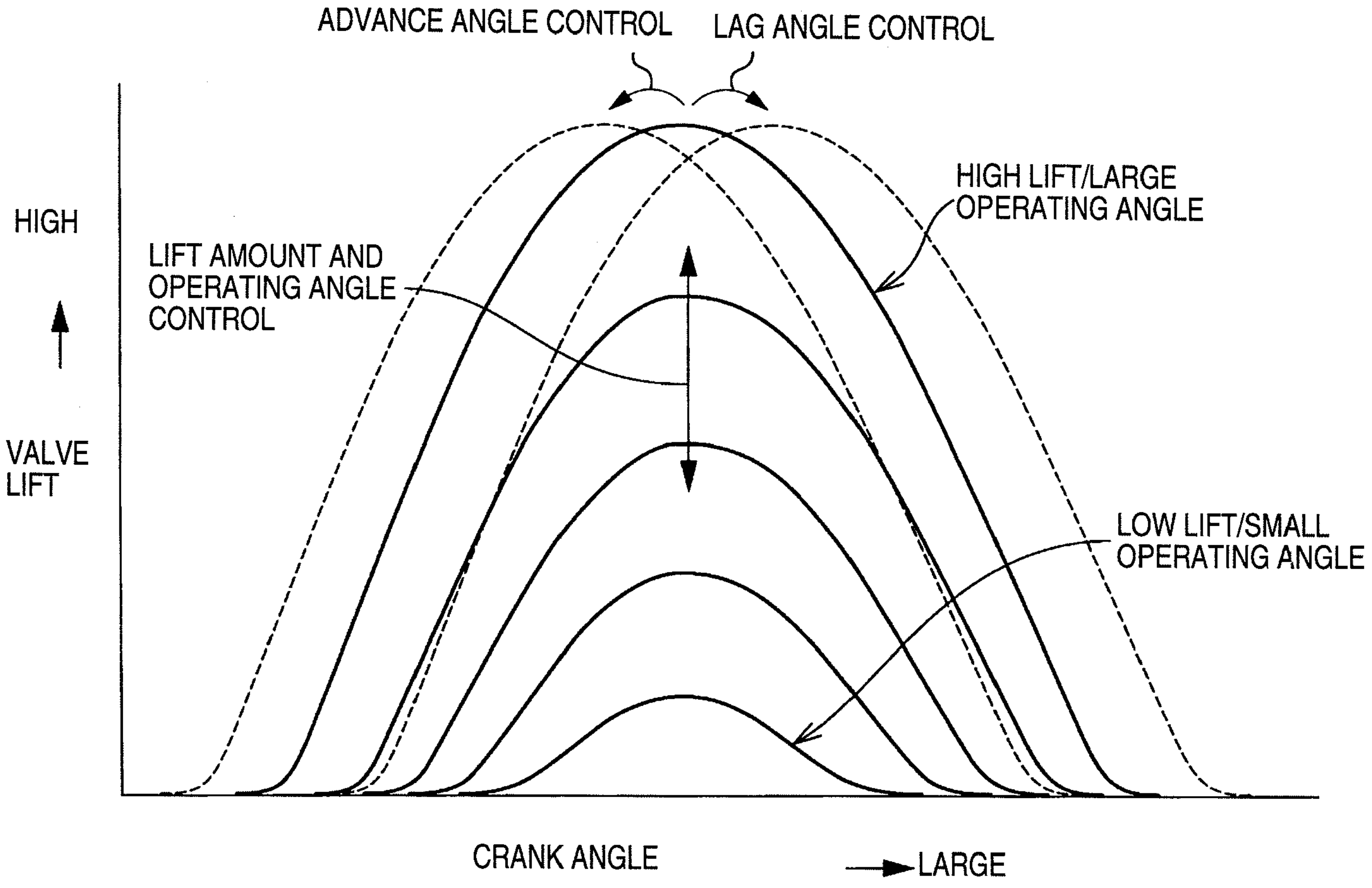

ADVANCE ANGLE CONTROL
LAG ANGLE CONTROL
HIGH LIFT/LARGE OPERATING ANGLE
HIGH
LIFT AMOUNT AND OPERATING ANGLE CONTROL
VALVE LIFT
LOW LIFT/SMALL OPERATING ANGLE
CRANK ANGLE
LARGE

FIG. 9

FIG. 10A

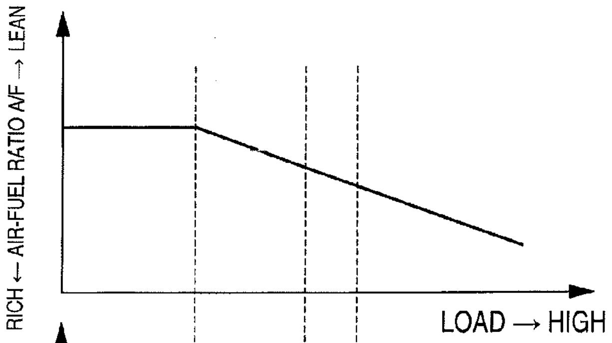


FIG. 10B

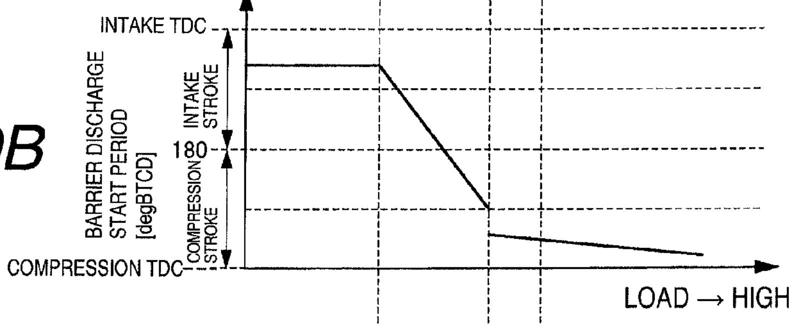


FIG. 10C

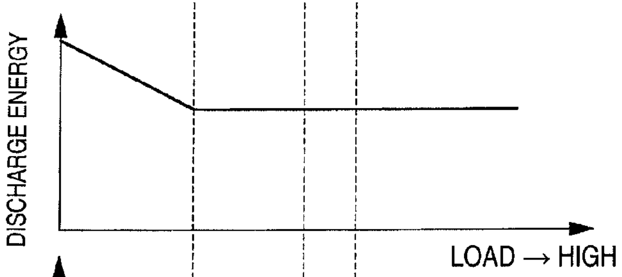


FIG. 10D

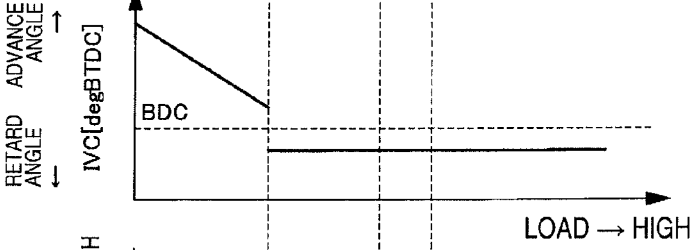
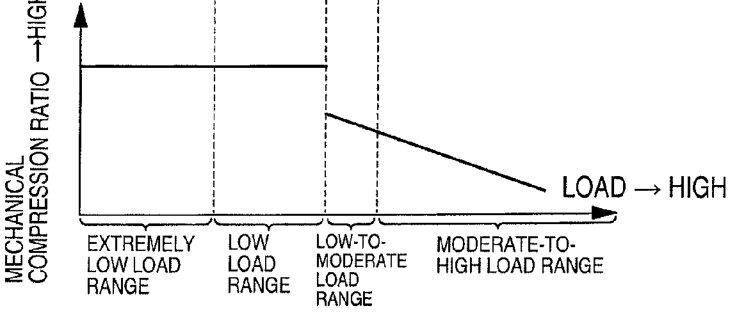


FIG. 10E



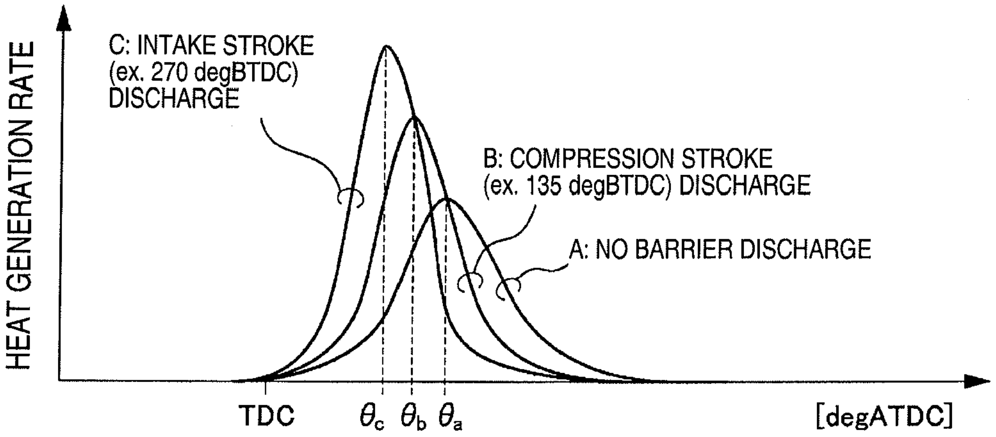

HEAT GENERATION RATE
C: INTAKE STROKE
(ex. 270 degBTDC)
DISCHARGE
B: COMPRESSION STROKE
(ex. 135 degBTDC) DISCHARGE
A: NO BARRIER DISCHARGE
TDC
θ_c
θ_b
θ_a
[degATDC]

FIG. 11

FIG. 12A

NO DISCHARGE

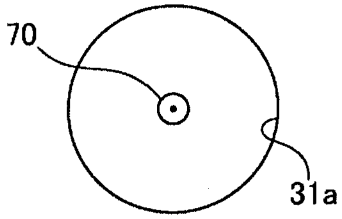


FIG. 12B

DISCHARGE IS INITIATED DURING COMPRESSION STROKE (IN THE VICINITY OF 180-0 deg BTCD)

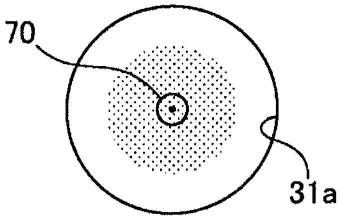
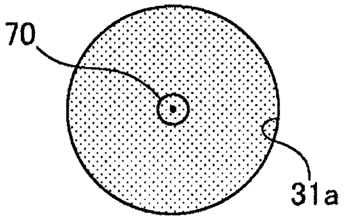


FIG. 12C

DISCHARGE IS INITIATED DURING INTAKE STROKE (IN THE VICINITY OF 180-360 deg BTCD)



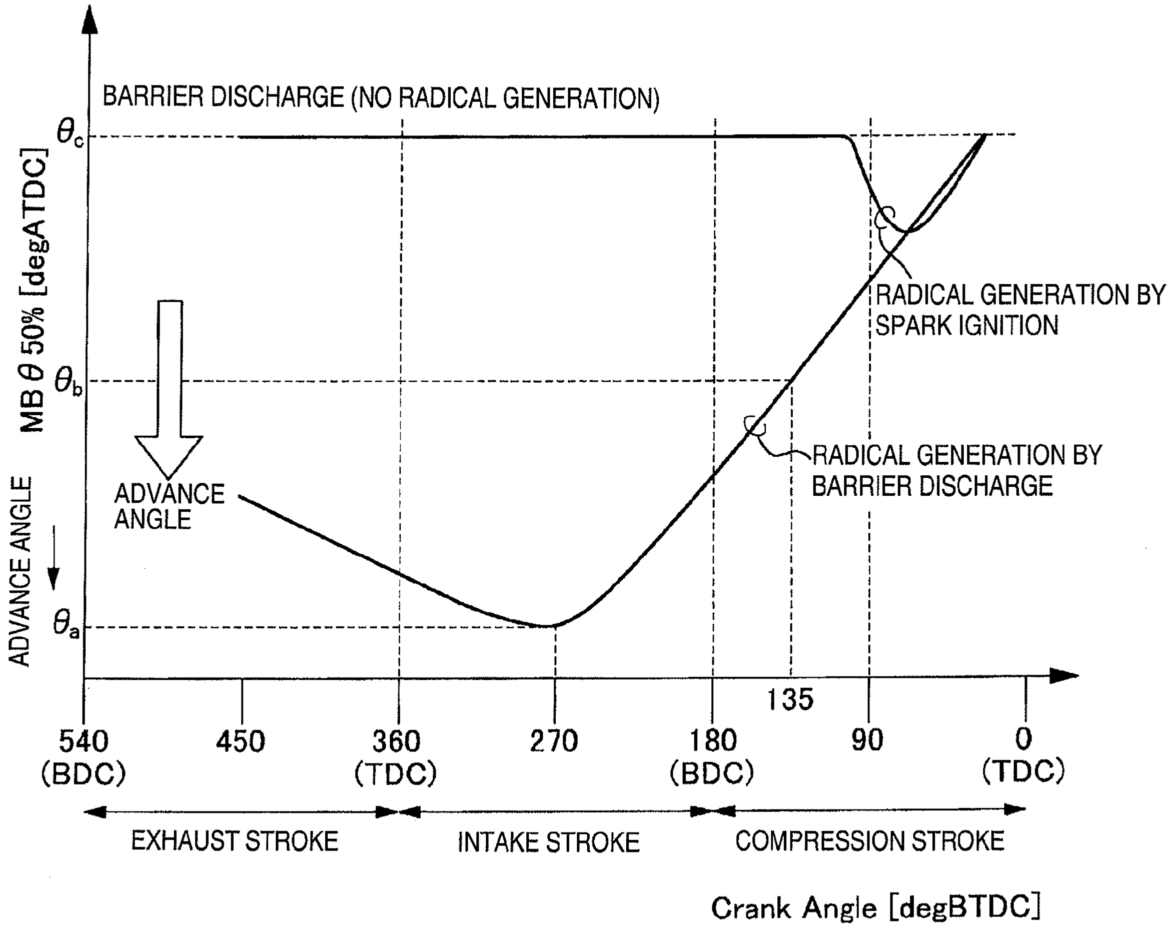

BARRIER DISCHARGE (NO RADICAL GENERATION)
θ_c
θ_b
θ_a
MB θ 50% [degATDC]
ADVANCE ANGLE
ADVANCE ANGLE
RADICAL GENERATION BY SPARK IGNITION
RADICAL GENERATION BY BARRIER DISCHARGE
135
540 (BDC)
450
360 (TDC)
270
180 (BDC)
90
0 (TDC)
EXHAUST STROKE
INTAKE STROKE
COMPRESSION STROKE
Crank Angle [degBTDC]

FIG. 13

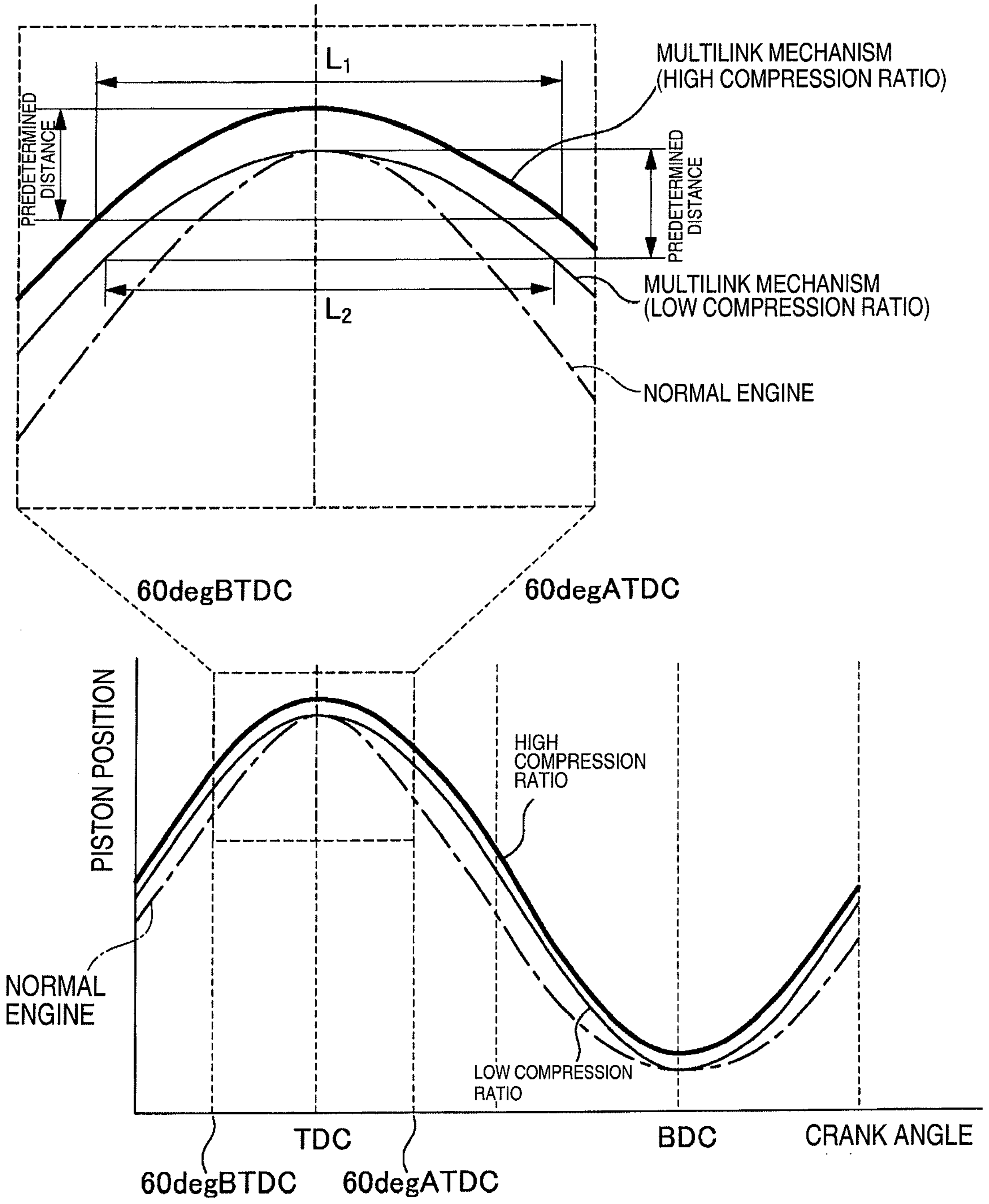

L1
MULTILINK MECHANISM
(HIGH COMPRESSION RATIO)
PREDETERMINED DISTANCE
PREDETERMINED DISTANCE
MULTILINK MECHANISM
(LOW COMPRESSION RATIO)
L2
NORMAL ENGINE
60degBTDC
60degATDC
PISTON POSITION
HIGH COMPRESSION RATIO
NORMAL ENGINE
LOW COMPRESSION RATIO
TDC
BDC
CRANK ANGLE
60degBTDC
60degATDC

FIG. 14

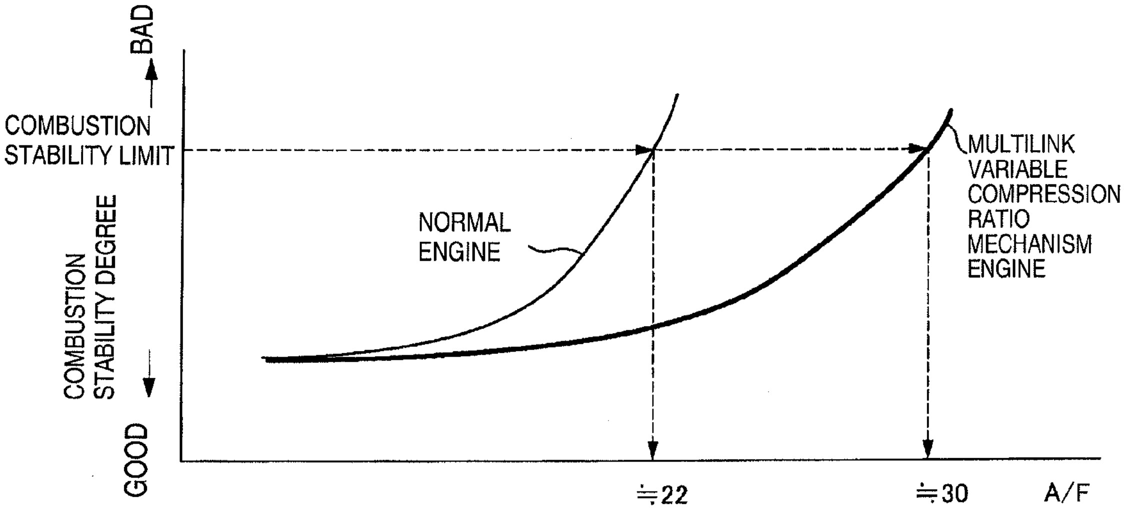
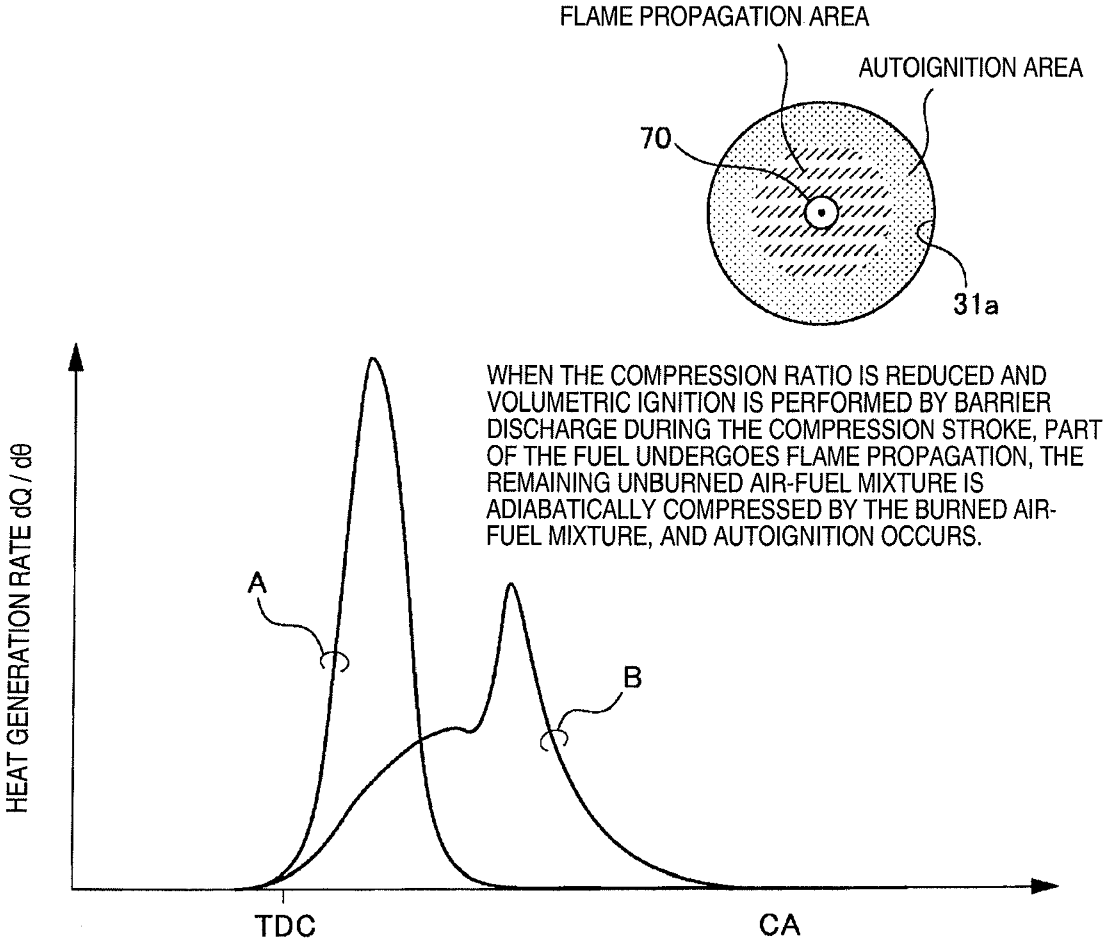

BAD
COMBUSTION STABILITY LIMIT
COMBUSTION STABILITY DEGREE
GOOD
NORMAL ENGINE
MULTILINK VARIABLE COMPRESSION RATIO MECHANISM ENGINE
≒22
≒30
A/F

FIG. 15



WHEN THE COMPRESSION RATIO IS REDUCED AND VOLUMETRIC IGNITION IS PERFORMED BY BARRIER DISCHARGE DURING THE COMPRESSION STROKE, PART OF THE FUEL UNDERGOES FLAME PROPAGATION, THE REMAINING UNBURNED AIR-FUEL MIXTURE IS ADIABATICALLY COMPRESSED BY THE BURNED AIR-FUEL MIXTURE, AND AUTOIGNITION OCCURS.

FIG. 16

FIG 17A

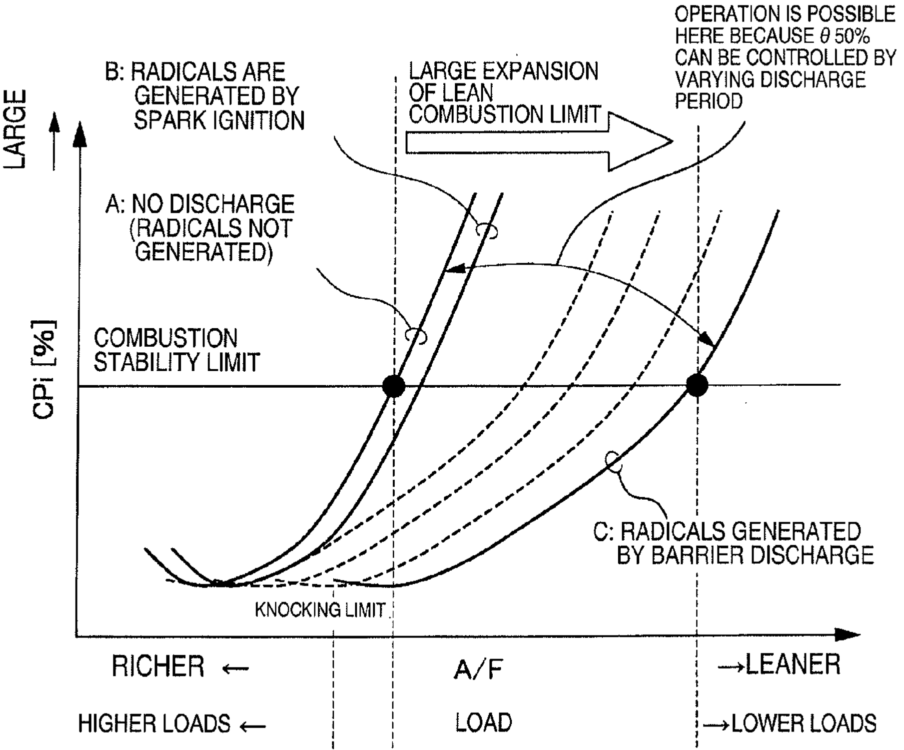
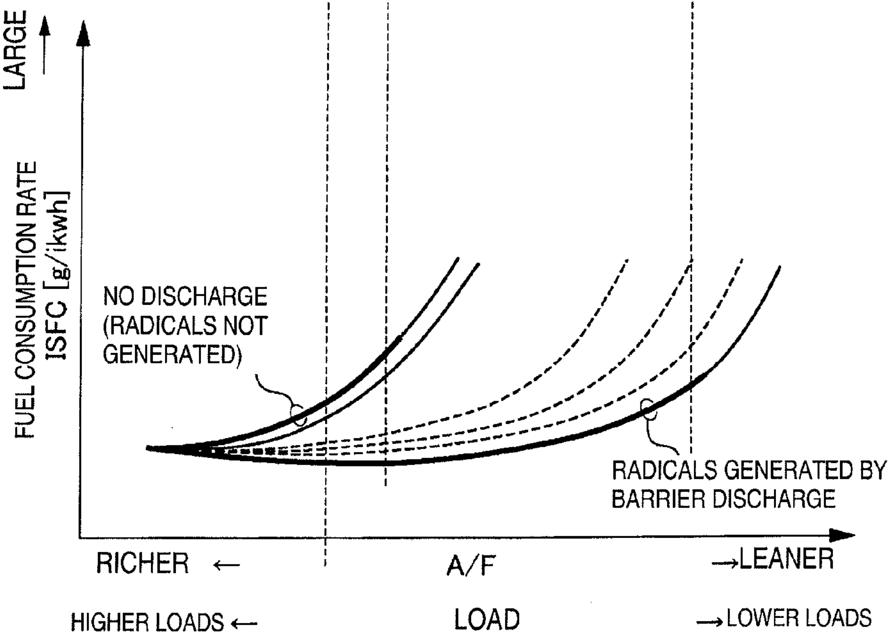


FIG 17B



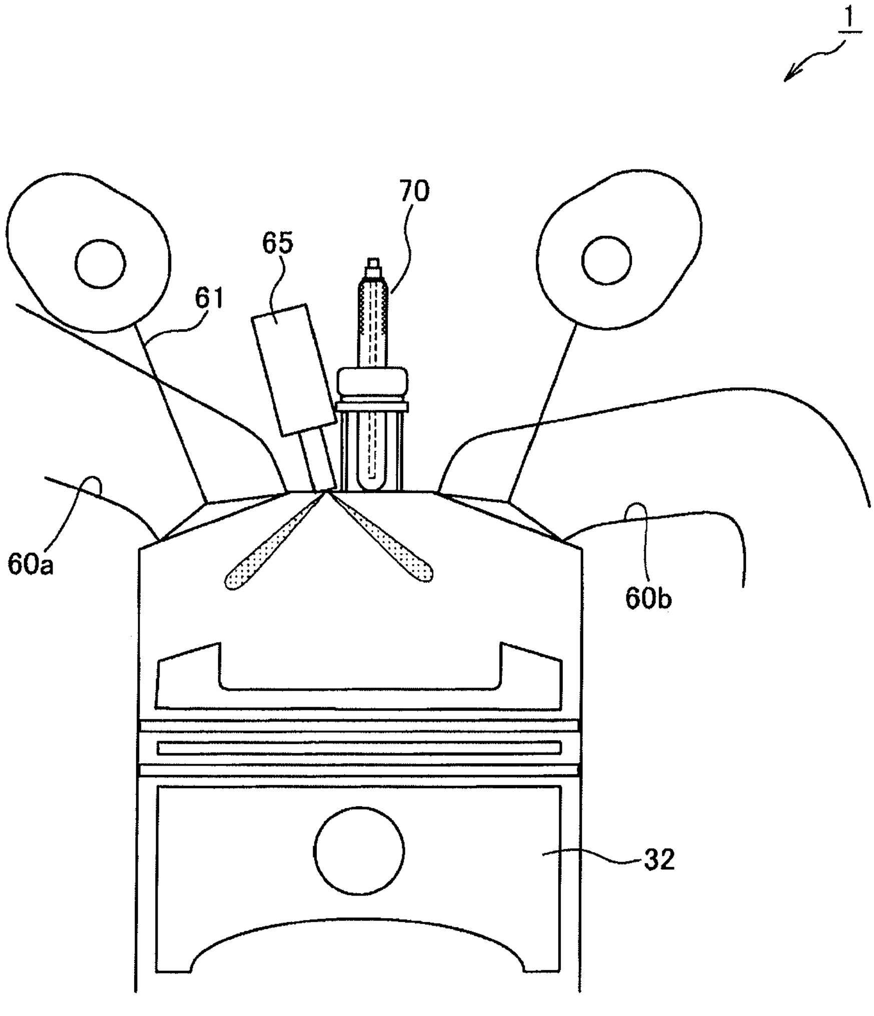


FIG. 18

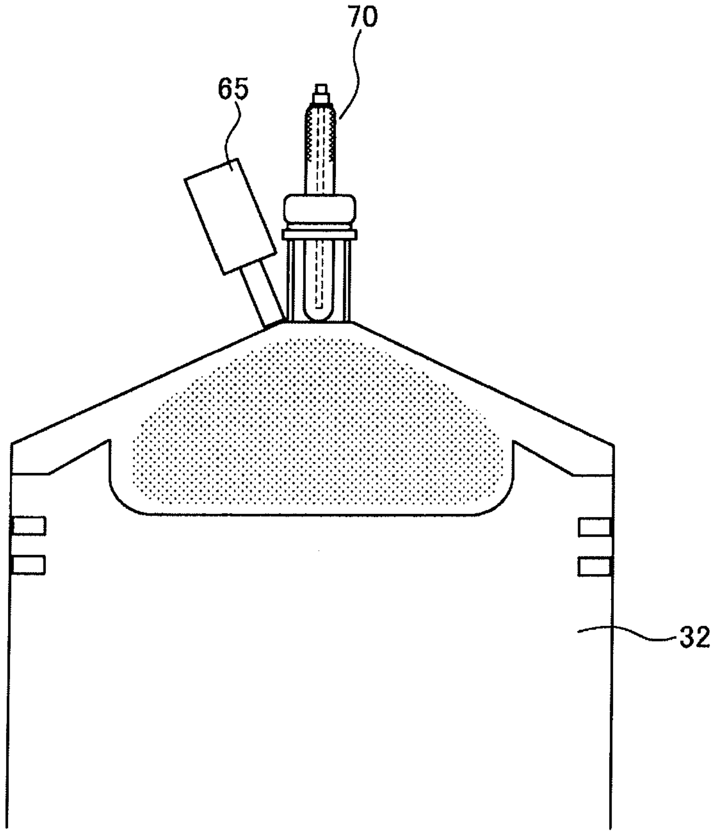


FIG. 19

FIG. 20A

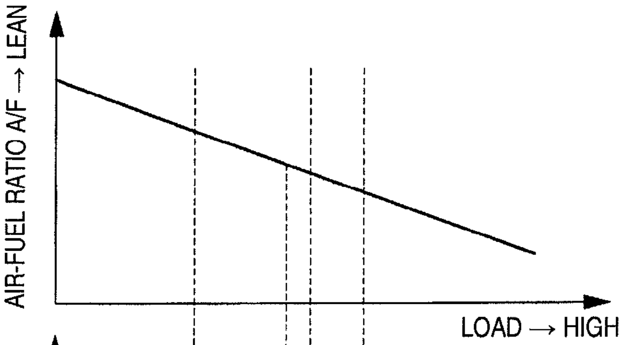


FIG. 20B

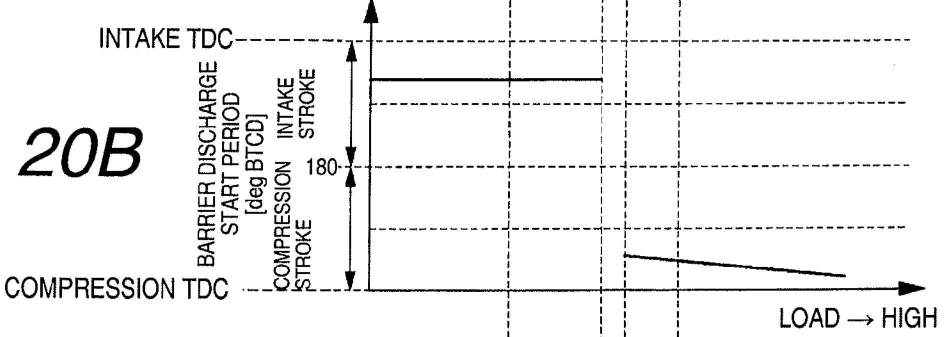


FIG. 20C

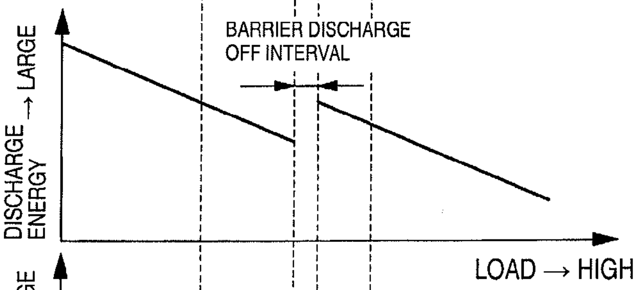


FIG. 20D

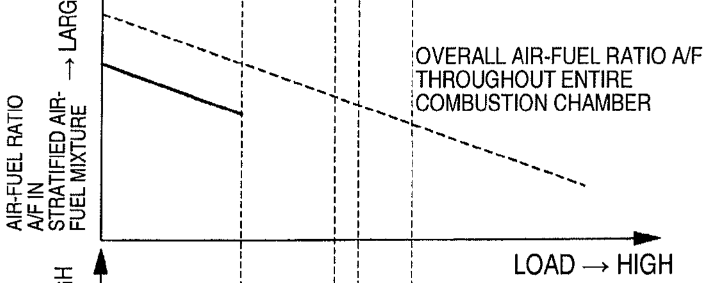
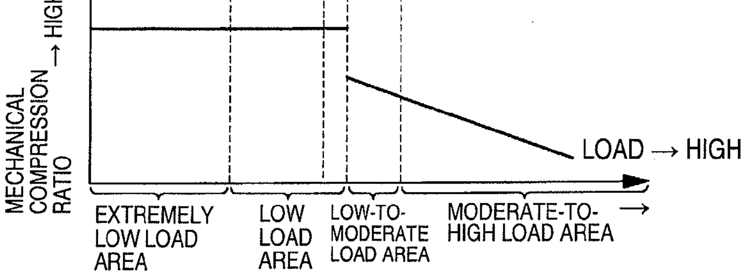


FIG. 20E



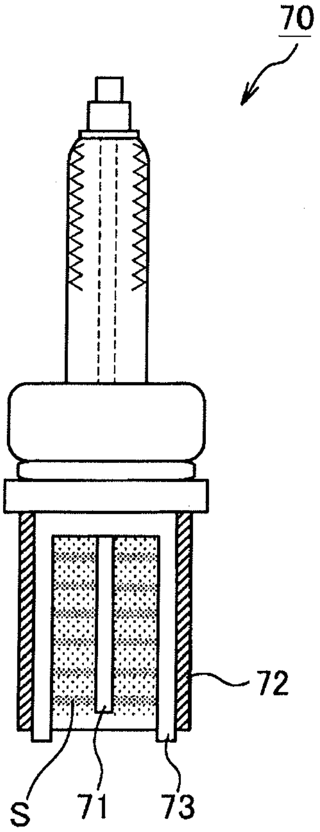


FIG. 21

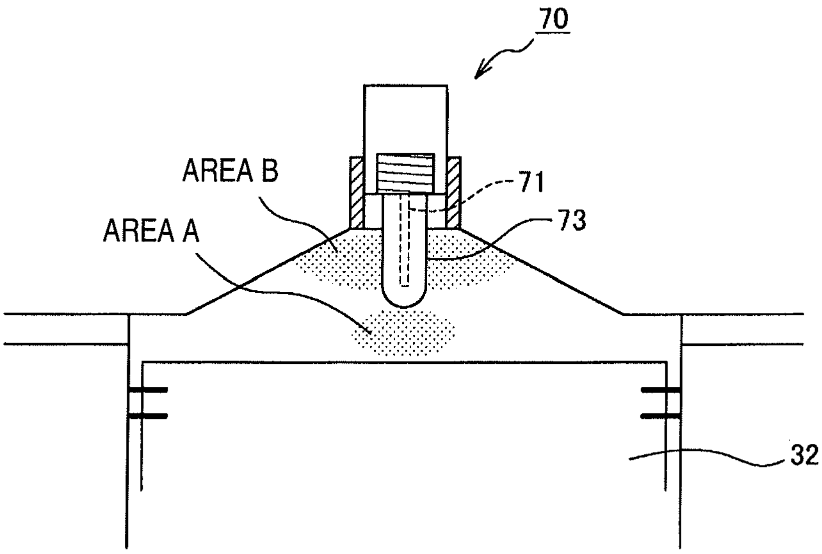


FIG. 22A

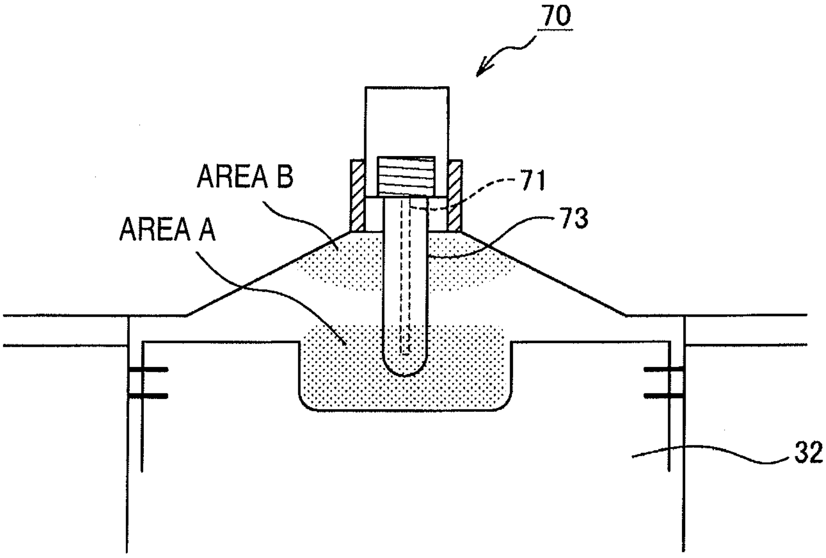


FIG. 22B

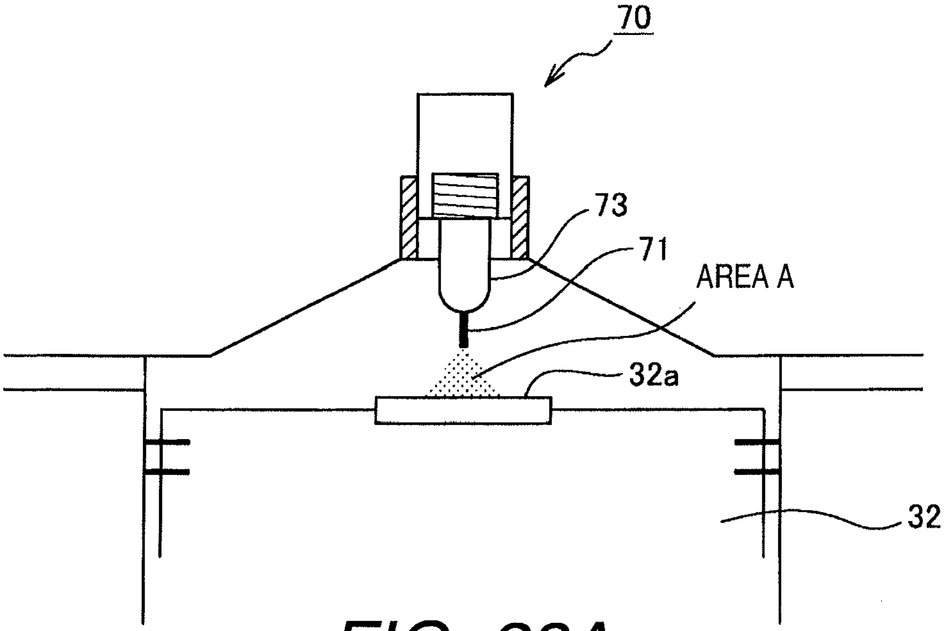


FIG. 23A

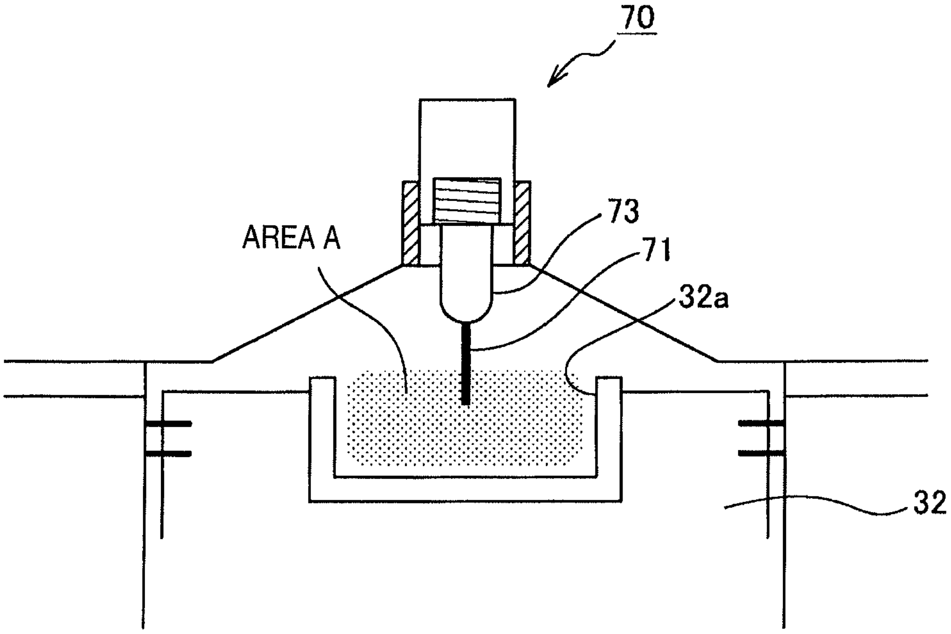
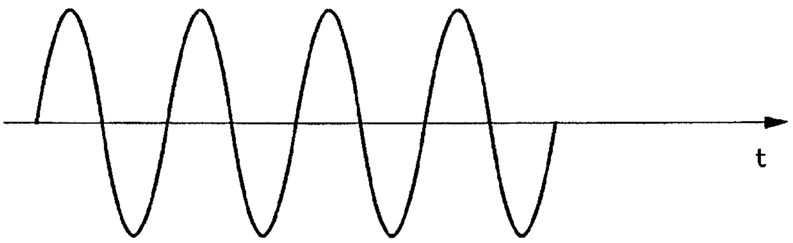


FIG. 23B

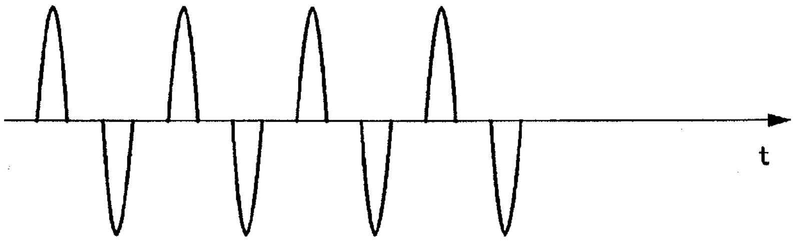
REFERENCE VOLTAGE WAVEFORM

FIG. 24A



BIPOLAR MULTIPLE PULSE VOLTAGE

FIG. 24B



REFERENCES CITED IN THE DESCRIPTION

Patent documents cited in the description

- JP 2001020842 A **[0002]**
- JP 11107725 A **[0056]**
- JP 2000213314 A **[0129]**
- JP 9060508 A **[0129]**
- JP 4017706 A **[0129]**
- JP 2007298294 A **[0132]**