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Miyake et al.

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[54] **INTERNAL COMBUSTION ENGINE**

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[21] Appl. No.: **415,789**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 126,771, March 22, 1971, abandoned.

[30] **Foreign Application Priority Data**

Mar. 23, 1970 Japan 45-23577

[52] **U.S. Cl.** ..... **123/32 ST**; 123/32 R; 123/139 BG

[51] **Int. Cl.**[2] **F02B 17/00**; F02M 59/20

[58] **Field of Search** ..... 123/32 R, 32 ST, 139 BG, 123/139 AE, 139 AR, 139 AP

[56] **References Cited**

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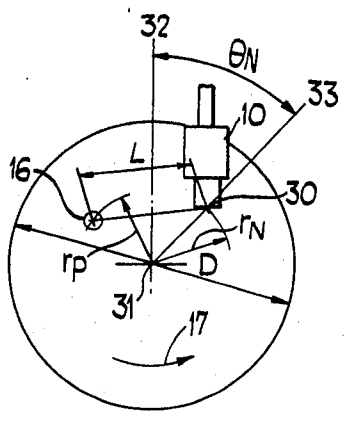
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[57] **ABSTRACT**

The invention is an internal combustion engine which includes a cylinder, means for introducing air into a combustion chamber in the cylinder in a manner to generate vertical flow around the periphery of the combustion chamber, a fuel injection nozzle and a sparking plug, the relative dispositions of the tip of the fuel injection nozzle, the sparking plug and the centre of the combustion chamber, and the bore of the combustion chamber, being defined within certain specified limits, to give advantages in the starting and operation of the engine.

10 Claims, 10 Drawing Figures



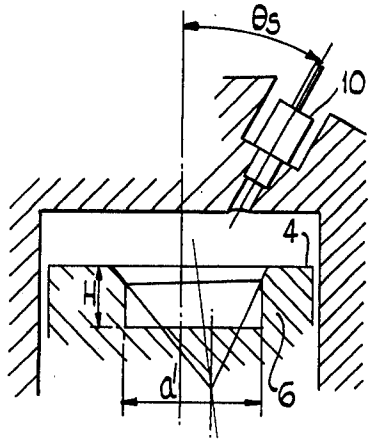
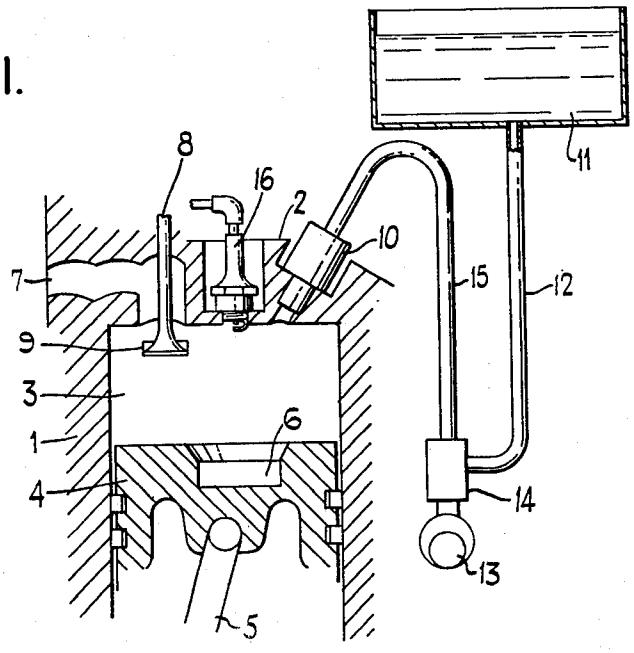


FIG.1.



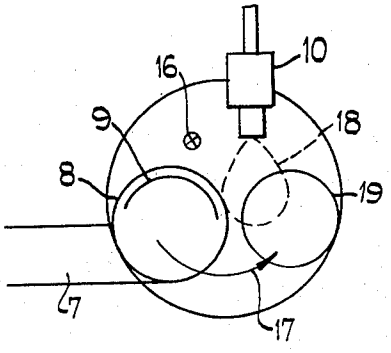
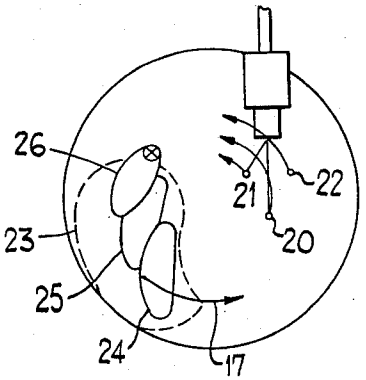


FIG.2.

FIG.3.



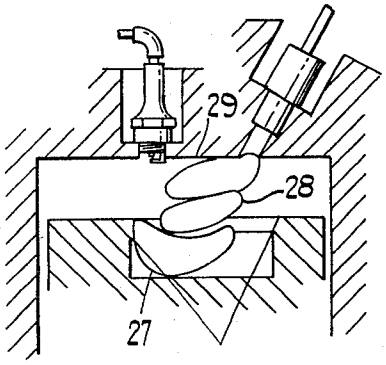


FIG.4.

FIG.5.

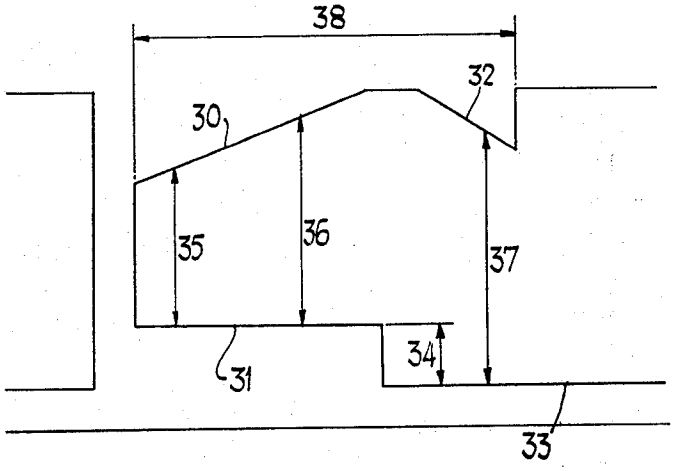


FIG.6.

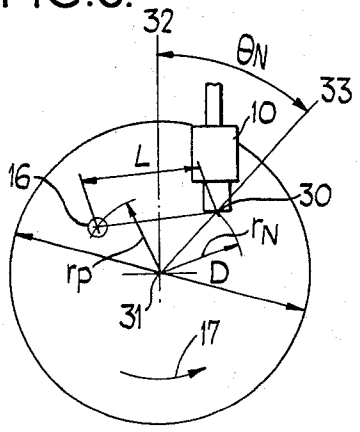


FIG.7.

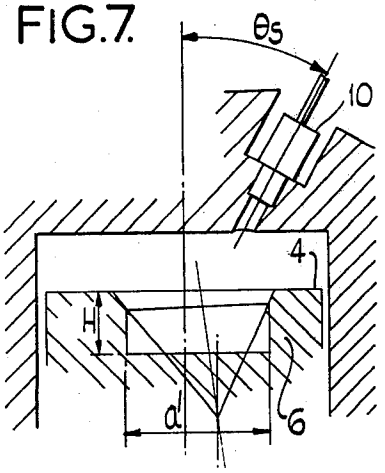
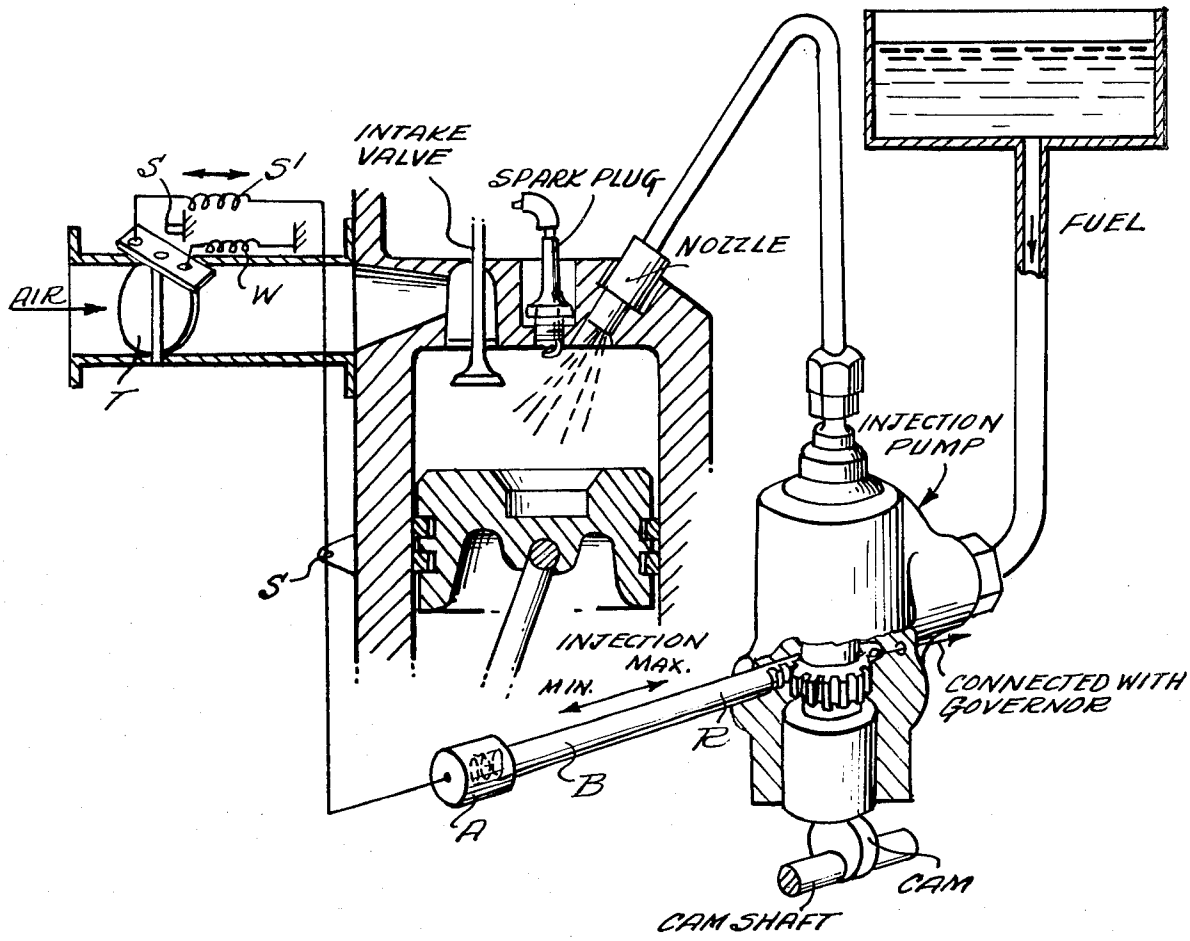


Fig. 8


INTAKE VALVE
SPARK PLUG
NOZZLE
FUEL
AIR
W
T
INJECTION PUMP
INJECTION MAX.
MIN.
CONNECTED WITH GOVERNOR
A
B
R
CAM
CAM SHAFT

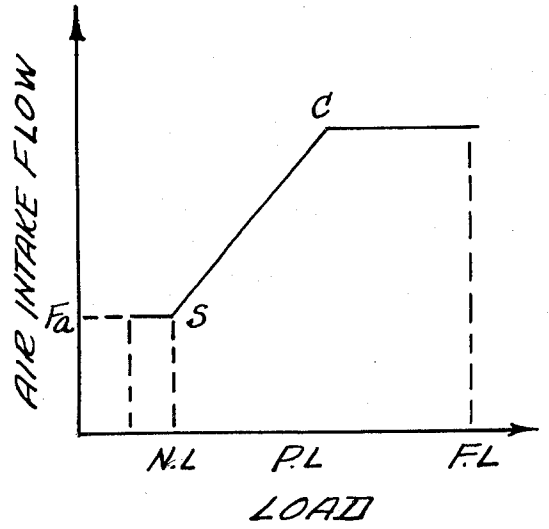
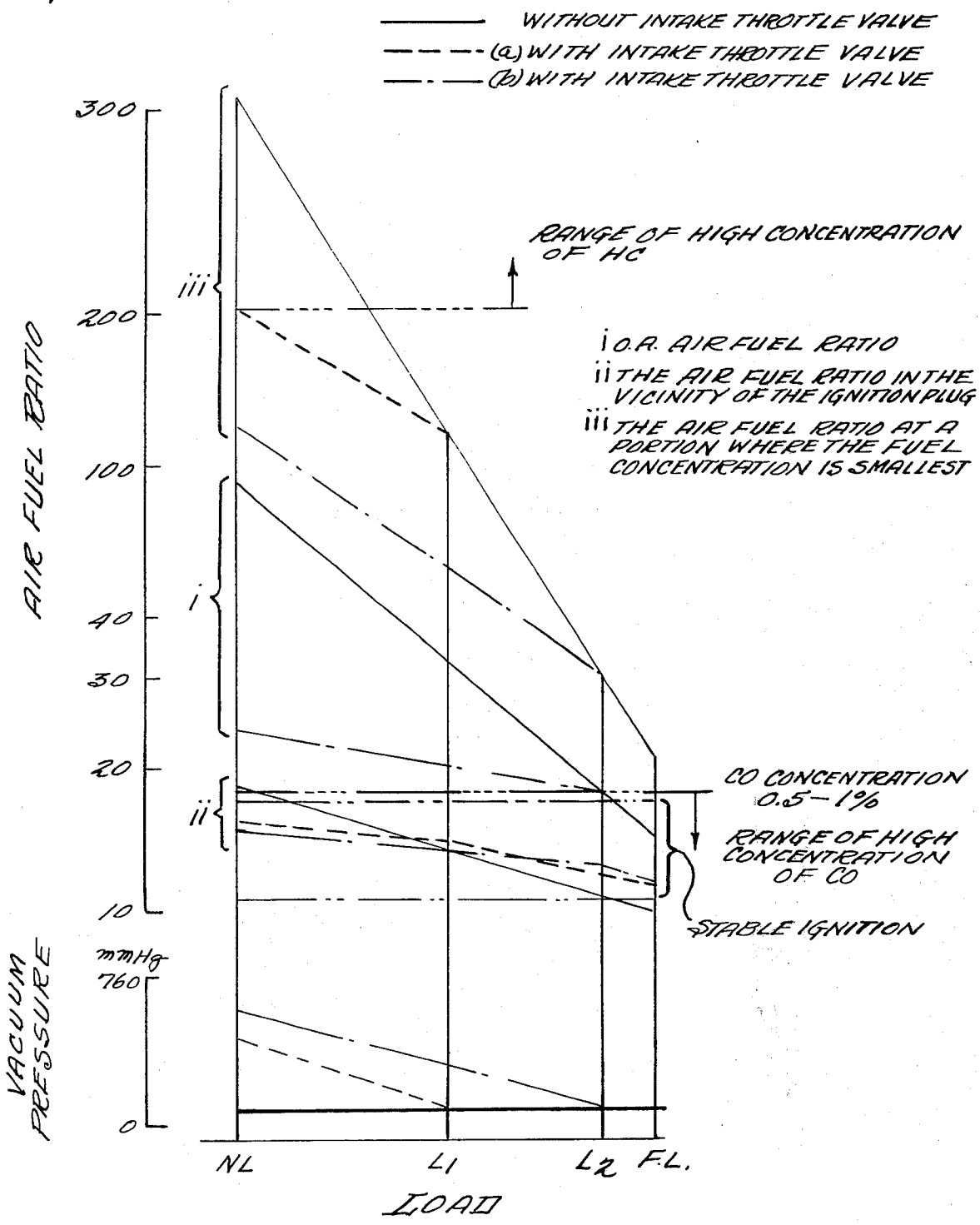

AIR INTAKE FLOW
C
Fa
S
N.L
P.L
F.L
LOAD

Fig. 9

Fig. 10


WITHOUT INTAKE THROTTLE VALVE
(a) WITH INTAKE THROTTLE VALVE
(b) WITH INTAKE THROTTLE VALVE
300
200
100
40
30
20
10
AIR FUEL RATIO
i
ii
iii
RANGE OF HIGH CONCENTRATION OF HC
i O.A. AIR FUEL RATIO
ii THE AIR FUEL RATIO IN THE VICINITY OF THE IGNITION PLUG
iii THE AIR FUEL RATIO AT A PORTION WHERE THE FUEL CONCENTRATION IS SMALLEST
CO CONCENTRATION 0.5–1%
RANGE OF HIGH CONCENTRATION OF CO
STABLE IGNITION
VACUUM PRESSURE
mmHg
760
0
NL
L1
L2
F.L.
LOAD

INTERNAL COMBUSTION ENGINE

REFERENCE TO RELATED APPLICATION

This is a continuation-in-part of application Ser. No. 126,771, filed Mar. 22, 1971, now abandoned in favor hereof.

BACKGROUND OF THE INVENTION

This invention relates to internal combustion engines.

Generally, in a spark ignition engine fuel is sucked into a combustion chamber by means of a vacuum generated of air. As the pressure applied to the fuel is very low, the fuel enters the combustion chamber as droplets together with the air. The fuel sucked into the combustion chamber is vapourized during the period from the intake stroke of the piston to the compression stroke, and it becomes a macroscopically uniform airfuel gas mixture before it is ignited. The time required for the liquid fuel to vapourize and the quantity of fuel which has not vapourized at the time ignition occurs depend upon the volatility of the fuel and the state of the engine, i.e. whether the engine is warm or cold. The volatility of gasoline is high, while those of kerosene and light oil are both low. For this reason, a cold engine cannot be started when kerosene or light oil is used as fuel, so that in a spark ignition kerosene engine, gasoline is separately supplied to start the engine.

When the air-fuel gas mixture is substantially uniformly distributed within the combustion chamber, even if the engine is designed so that the position of the ignition source and the shape of the combustion chamber are optimized and the flame speed is high, knocking is apt to occur, so that if fuel of low octane value is used it is necessary to lower the compression ratio of the engine, resulting in reduction of the thermal efficiency of the engine. On the other hand, if it is desired to increase the thermal efficiency of the engine, the octane value of the fuel must be high. Furthermore, since the range of air to fuel ratios which can be used with spark ignition engines is narrow in the case of uniform air-fuel gas mixtures, it is necessary to adjust the air flow rate by a throttle valve in order to adjust the load, and this throttle loss results in a reduction of the thermal efficiency. A carburetor is provided with various compensating means to enable it to adapt to transient phenomena in addition to the air-fuel ratio required for various operating conditions of the engine, but these means are so adjusted as to adapt the actual quantity of fuel within a carburetor to be more dense than that set by a theoretical air-fuel ratio, taking into account the adaptability to the transient phenomena and the effect of residual gas. For this reason, detrimental components such as unburned fuel or carbon monoxide are discharged as a part of the exhaust gas. If the operation of the engine is carried out with the engine insufficiently warmed up, unvapourized fuel settles on the cylinder wall in the liquid state and is introduced into the crankcase, resulting in the dilution of the lubricating oil present in the crankcase.

According to the present inventon there is provided an internal combustion engine which includes a cylinder means for introducing air into a combustion chamber in the cylinder in a manner to generate vortical flow around the periphery of the combustion chamber, a fuel injection nozzle for injecting fuel into the combustion chamber in a direction so that the fuel has a component of velocity in a direction opposite to theh direction of the vortical air flow in the vicinity of the fuel injection nozzle, and a sparking plug for igniting the combustible mixture formed in mixing of the fuel and the air, in which the fuel injection nozzle is situated so that

1. $2r_N/D$ has a value in the range of from ⅛ to ¾ inclusive, where R_N is the distance between the tip of the injection nozzle and the center of the combustion chamber and D is the bore of the combustion Chamber;
2. θ_N has a value in the range of from 0° to 60° inclusive where θ_N is the angle between a radius of the combustion chamber which is parallel to the axis of the fuel injection nozzle and a radius of the combustion chamber which extends through the tip of the fuel injection nozzle;
3. O_S has a value in the range of from 0° to 45° inclusive, where O_S is the angle between the axis of the fuel injection nozzle and the axis of the cylinder;

the sparking plug is situated so that

4. $2r_p/D$ has a value in the range of from ⅛ to ¾ inclusive, where r_p is the distance between the sparking plug and the center of the combustion chamber and D is as defined above;

and the distance L between the tip of the fuel injection L and the sparking plug lies within the range of from 10 to 60 mm. inclusive.

Fuel can be injected into the combustion chamber at a high pressure so that the fuel is supplied as particles which are extremely small, and simultaneously with the injection the fuel particles are exposed to the air at a high temperature in the compression stroke. Also, vapourization of the fuel is much improved by making the fuel fly at a predetermined relative speed with respect to the air flow. The improvement in the vapourization of the fuel may be so large that even with a cold engine and/or even with fuel of low volatility such as kerosene or light oil, it is possible to start the engine. Since in this case, the liquid fuel does not settle onto the cylinder wall dilution of the lubricating oil does not occur. In other words, the difference in operation due to the temperature condition of the engine and/or the kind of fuel, may be largely or completely eliminated. In addition the rate at which fuel is required can be quickly varied in response with a change in the operating conditions of the engine, so that the response of the engine is improved, and there is no need to supply excessive fuel thereto. Accordingly, the air-fuel ratio of the gas mixture can be adjusted so that the amount of detrimental components in the exhaust gas such as unburned fuel or carbon monoxide is less than a desired value within a restricted limit for the exhaust gas. Owing to the improvement in the vapourization of the fuel and the improvement in the transient response of the engine, the quantity of fuel required for the entire range of operations through partial load and no load is less than that of an engine provided with a carburetor using gasoline, and thus the thermal efficiency is increased. Furthermore, by making use of the system in which the fuel is not supplied to the so-called end gas portion of the combustion chamber, that portion furthest from the sparking plug, it is possible to operate the engine accompanied with little or no knocking, irrespective of the octane value of the fuel.

This state of affairs is realized by making the distribution of the gas mixture in the combustion chamber uneven, by means of the vortical flow of air in the com-

bustion chamber. By making use of the combination of the air vortex and the injection system, it is possible to supply a combustible gas mixture to the ignition source while a gas mixture more dilute than the combustible limit of the fuel is supplied to the end gas portion. Thus knocking may be reduced or eliminated and the compression ratio can be raised, so that the power and the fuel consumption of the engine can be improved.

In general, the combustion pattern of an internal combustion engine is classified into (a) a heat mixing pattern in which the ignition source is disposed in the peripheral portion of the combustion chamber and the combustion proceeds towards the center, and (b) a heat pinch pattern in which the ignition source is disposed at the center of the combustion chamber and the flame propagates outwardly from the center.

In the case of light load and lower fuel feed rate the pattern (b0 is suitable (b) obtaining a stable combustion, while at a heavy load, the pattern (a) in which the combustible gas gathers at the center and the heavy air is fed to the combustion region at the outer periphery, is more effective because the utilization factor of the air becomes maximum.

In a general fuel injection apparatus, as the revolution speed of the engine is increased, the injection period is prolonged, and so the injection rate is lowered. In addition, since the swirl rate tends to decrease as the revolution speed of the engine is increased, the contents ratio of the fuel-air mixture gas at the igniting position is varied in accordance with the revolution speed of the engine.

The invention claimed in this application covers aspects of an important new stratified-charge combustion system termed the Mitsubishi Combustion Process (MCP). The system is further described and explained in the Society of Automotive Engineers paper 720196 entitled *Developing a New Stratified-Charge Combustion System with Fuel Injection for Reducing Exhaust Emissions in Small Farm and Industrial Engines* given at the Automotive Engineering Congress in Detroit, Michigan, Jan. 10–14, 1972 and authored by Masataka Miyahe (one of the applicants in the present case).

The article lends support to applicants' contentions that the design values cited in the claims are (a) critical and (b) not easy to come by, even for those of exceptional skill in the art.

In a stratified charge combustion engine, the concentration of an air-fuel mixture gas (hereinafter referred to simply as gas mixture) is unevenly distributed. Then upon ignition an ignitable gas mixture is formed in the proximity of the ignition plug where said gas mixture is surely ignited by an electric spark at the ignition plug, and with the combustion flame a lean mixture gas existing at other locations is completely burned. In a stratified charge combustion engine employing a direct fuel injection and a combustion chamber swirl but no gas valve, an ignitable gas mixture is formed in the proximity of the ignition plug in a stable manner under every operating condition while at other locations in the combustion chamber there is formed a so lean gas mixture as not to cause severe knocking. The following are the items which determine the mode of the stratified charge combustion engines.

1. Combustion pattern;
2. Method for forming an ignitable gas mixture in the proximity of the ignition plug;
3. Utilization of air at a heavy load; and
4. Effect of piston cavity.

Though these items have an affect upon each other, in order to clarify the respective modes of operation, each item will be explained separately.

1. Combustion pattern: It will be clearly understood from the following description that the position of the ignition source, that is, the ignition plug serves as an important factor for determining the combustion pattern and the function and advantage resulted from said pattern. More particularly, in the case of a spark ignited engine employing a uniform gas mixture without swirl (the so-called gasoline engine, it is a common practice that in order to improve the combustion efficiency and to suppress knocking, the ignition plug is disposed at the center of the combustion chamber where the distance of flame propagation is the shortest. On the other hand, the ignition in a direct injection type of diesel engine making use of a swirl does not occur at the center of the combustion chamber but instead at a middle position between the center and the outer circumference. According to what has been proved as a result of various researches, this is due to the fact that in the combustion system employing a swirl, if the combustion is commenced at the center of the combustion chamber, a combustion pattern of thermal pinch results in which the air about the combustion chamber cannot be utilized and so good combustion cannot be achieved. In addition, while in the spark ignition engine knocking is generated at a heavy load, in the case of the stratified charge combustion engine the concentration of the gas mixture of the end gas within the combustion chamber is too lean to generate knocking, and so there is no need to dispose the ignition plug at the center of the combustion chamber to shorten the flame propagation distance as is the case with the spark ignition engine. Also in the combustion system making use of a swirl, the combustion velocity just after the ignition is varied depending upon the position of the ignition source. More particularly, if the ignition source is positioned at the center of the combustion chamber, substantially no swirl flow occurs, and so the flame diffusion just after the ignition is slow. Then the combustion velocity becomes fastest in the neighborhood of the upper dead center where the pressure is high, and consequently, the combustion temperature is substantially raised resulting in a large amount of NO_x generation. On the other hand, if the ignition source if disposed at the middle point between the center of the combustion chamber and its circumference, then the combustion velocity just after the ignition is fast and thereafter the combustion velocity becomes slow, so that the highest combustion temperature is inversely lowered to perform a slow combustion, and thereby it is made possible to achieve perfect combustion at a low concentration of NO_x emission. As described above, according to the present invention, on the basis of three reasons: (a) thermal pinch is prevented, (b) correlation to knocking is considered, and (c) high combustion velocity just after the ignition can be obtained the location of the ignition plug is selected on the ground of the basic concept that even in the case of a spark ignition engine, for the combustion of a gas mixture having a swirl flow, the method for optimizing the combustion is to dispose the ignition plug at a position displaced from the center of the combustion chamber. And as a result of theoretical investigation (see the SAE paper, referred to above) on the combustion chamber config-

uration and the nozzle position which are adapted to form a ignitable gas mixture at the position of the ignition plug while to form a lean mixture gas at the other locations, the present inventors have found that the operational condition within the numerical limitations as specified in the claims is the only scope where the subject system can be effectively utilized.

SUMMARY OF THE INVENTION

According to the present invention, the ignition plug is disposed in the midway between the center and the outer periphery of the combustion chamber, so that as a combined effect of the swirl air flow and the nozzle position, the fuel-air mixture gas may be formed in the inside region of the combustion chamber with respect to the plug position at a light load, while it may be formed widely in the outside region of the combustion chamber with respect to the plug position towards the outer periphery at a heavy load, and consequently, in view of the above-described theory of combustion a stable combustion can be obtained at a light load while a smoke-free combustion which fully utilizes the air can be obtained at a heavy load.

It is possible to further improve the effect of concentrating the fuel-air mixture gas at a light load as well as the effect of promoting the combustion by enhancing the turbulent flow of the fuel-air mixture gas caused by the squish effect and by utilizing the heat in the wall of the cavity to evaporate the fuel at a heavy load.

Owing to the herein prescribed depth of the cavity as it is possible to further improve the effect of forming fuel-air mixture gas at the ignition plug such that misfire may not occur, through control of the motion of fuel spray upon starting at a light load.

The combustion process of the patterns (a) and (b) in combination of concentrating the fuel spray at a light load and enhancing the utilization factor of the air at a heavy load that is resulted from the positions of the injection nozzle and the ignition plug as well as the configuration of the cavity, the further delay of injection upon starting serves to better prevent the diffusion of the fuel spray to further improve the effect of preparing a fuel-air mixture gas of high concentration upon starting, and thereby stable ignition can be realized.

According to the present invention, the injection nozzle is positioned so as to inject in the direction opposite to the direction of the swirl flow, so that the fuel-air mixture gas is formed in a wide region at a heavy load while in a narrow region at a light load by making the relative velocity of the swirl flow with respect to the injected fuel particles large at a high injection rate but small at a low injection rate. In addition, by mounting the ignition plug at a place on the back of and a little apart from the injection nozzle, it is assured that the combustible mixture gas produced by slow diffusion of a fuel-air mixture gas may reach the proximity of the ignition plug at the ignition timing, and thereby a combustion that is extremely insensitive to the ignition timing and the injection timing can be realized. In other words, even though the timing of fuel injection and/or ignition is varied a little, there never occurs misfire, exhaust smoke, or lowering of fuel efficiency (lowering of output and increase of fuel comsumption). According to test experiments, up to the engine revolution speed of 4000 rmp the injection timing as well as the ignition timing can be fixed. Owing to this fact, a misfire-free, smoke-free and stable combustion can be attained over the entire operation range, and a longer period can be maintained between the injection and the ignition. Therefore, even if a fuel having a poor volatility (kerosene, light oil, heavy oil) is employed, there never occurs the contamination of the ignition plug.

As the fuel is injected against the swirl air flow, when the revolution speed of the engine is low as is the case with the starting, the fuel hardly reaches the ignition plug. In order to avoid this shortcoming, by the provision of a cavity on the top face of the piston it becomes easy for the injected fuel particles to reach the ignition plug after they have struck against the wall of the cavity. The subject invention is extremely effective for improving the stratified charging system into a complete one.

According to the present invention, the position of the ignition plug is located intermediate the center of the combustion chamber and its outer circumference, the injection nozzle is disposed upstream in the swirl flow with respect to the position of the ignition plug, and injection is made in the opposite direction to the swirl flow. Then the injected spray is pushed back by the swirl flow and thereby forms and ignitable gas mixture in the proximity of the ignition plug. The distance from the injection nozzle to the ignition plug is selected in such manner that the finally injected spray may be evaporated to form an ignitable gas mixture and also that excessive diffusion may not occur. Furthermore, the piston-cavity as described infra is effective for further preventing the gas mixture coming to the ignition plug from diffusing.

Utilization of air at a heavy load: At a heavy load, it is necessary to utilize the entire air within the combustion chamber to achieve an efficient combustion. In other words, the injected fuel should contact with the entire air within the combustion chamber to form an ignitable mixture gas at every location.

According to the present invention, since the fuel is injected by the injection nozzle from an intermediate position between the center and the outer circumference of the combustion chamber towards the swirl (with a spray angle of 30°–90°), a part of the fuel spray having a weak penetrating capability is carried away by the swirl to form an ignitable gas mixture in the proximity of the ignition plug, while the remaining spray having a strong penetrating capability shows the pattern as shown in FIG. 3 of the drawings, in which the spray injected towards the center of the combustion chamber is not so much diffused because its relative velocity with respect to the swirl is low, but the spray injected towards the outer circumference of the combustion chamber is largely diffused because its relative velocity with respect to the swirl is high. As described, owing to the composite effect of the gas mixture distribution produced by the fuel spray which is controlled in such manner that the diffusion of the spray is large at the outer circumference of the combustion chamber, but the diffusion is small at the center, and the ignition plug located at the middle point between the center of the combustion chamber and the outer circumference. The mixture gas formed at the outer circumference and showing large diffusion has a faster combustion velocity than the gas mixture formed at the center and showing small diffusion, resulting in thermal mixing in the combustion pattern, and consequently an efficient combustion can be achieved. In addition, the present invention has the following remarkable advantages:

1. A stable ignitable gas mixture can be surely formed in the proximity of the ignition plug;
2. A sufficient time necessary for the evaporation of fuel can be obtained;
3. The fuel spray is well mixed with air at a heavy load operation, so that the entire air within the combustion chamber can be utilized to derive a high output;
4. Since the ignition plug is not located in the direction of injection, wetting and/or damage of the ignition plug is not caused.

The object of providing a piston-cavity exists, as generally known, in the following two points:

1. To obtain an efficient combustion by making use of a squish effect; and
2. To accommodate the injected spray within the cavity so that the spray may not strike against the cylinder wall, and also to prevent an excessive diffusion of the gas mixture.

The piston-cavity results in another effect which was not easily foreseen. More particularly, according to the present invention, owing to the composite effect of the facts that the outlet of the cavity is enlarged and inclined towards the side of the ignition plug and that the ignition plug is located intermediate the center of the combustion chamber and the outer circumference, the gas mixture flowing out of the cavity can flow smoothly towards the ignition plug, and consequently, there is provided an advantage that the ignitable gas mixture, which has been formed by the fuel spray injected and prevented from diffusing, can be surely conveyed to the proximity of the ignition plug.

The engine of the present is superior to prior art engines in that it operates efficiently and with many advantages not only when the intake air is unthrottled but also when the intake air is throttled.

In general, in a gasoline injection engine with spark ignition, the concentration of the mixture becomes uniform, so that intake air throttle means is interlocked accurately with fuel injection control means and complex, expensive means is provided which is capable of maintaining the overall air fuel ratio in the range from 10:1 to 18:1.

In the internal combustion engine of the present invention in which the combustion chambers are occupied by a mixture of non-uniform concentration, if the air fuel ratio in the vicinity of the ignition plug is from 10:1 to 18:1, the overall (hereinafter abbreviated as O.A.) air fuel ratio may be variable to a range in which the fuel is extremely lean, so that the rate of fuel being injected relative to the supply rate of intake air can be controlled in a usual manner and the engine is in need of only such means which is capable of throttling the intake air only to the necessary degree under the necessary conditions to be described later.

The principles of the invention will be further hereinafter discussed with reference to the drawings wherein preferred embodiments are shown. The specifics illustrated in the drawings are intended to exemplify, rather than limit, aspects of the invention as defined in the claims.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a longitudinal sectional view of part of one embodiment of an internal combustion engine according to the present invention;

FIG. **2** is a plan view of the combustion chamber shown in FIG. **1**;

FIGS. **3** and **4** are traces showing the directions of movement of the injected fuel particles;

FIG. **5** is a developed view of the peripheral side wall of a plunger of an injection pump used in the internal combustion engine according to this invention;

FIGS. **6** and **7** are plan and side views, respectively, of a fuel injection nozzle and a sparking plug for the purpose of explaining the installed location thereof according to the present invention.

FIG. **8** is a schematic view showing an example of the intake air throttle means usable with the internal combustion engine of the present invention.

FIG. **9** is a plot of air intake flow versus load, for the engine equipped as in FIG. **8.**

FIG. **10** is a plot of air/fuel ratio versus load (injection rate) for an engine with and without an air throttle valve and for an engine with an air throttle valve the effect of changing load when the engine speed is maintained constant.

DETAILED DESCRIPTION OF THE PRESENTLY PREFERRED EMBODIMENTS

In FIG. **1,** reference numeral **1** designates a cylinder, **2** a cylinder head, **3** a combustion chamber, **4** a piston which has a cavity **6** of an appropriate shape on its top surface and is connected to a crank shaft by means of a connecting rod **5, 7** a spiral inlet passage through which air can be introduced to produce a vortical flow in the combustion chamber **3, 8** an air inlet valve, which may be provided with a shield plate **9** to direct the air in a tangential direction with respect to the combustion chamber **3.** The shield plate **9** can be used to provide the desired vortical flow if the inlet passage **7** is a simple shape, rather than a spiral shape. Alternatively, both the spiral intake passage **7** and the shield plate **9** of the air intake valve **8,** can be provided. Numeral **10** designates a fuel injection nozzle, which is supplied with fuel from a fuel tank **11** through a pipe **12,** an injection pump **14** driven by a cam **13** and an injection tube **15.** Numeral **16** designates a sparking plug, which is connected to an ignition coil type or a magnet type of ignition device (not shown).

Air entering through the inlet passage **7** produces vortical flow in the combustion chamber **3,** in the direction shown by an arrow **17** in FIG. **2** the vortical flow being maintained also in the compression stroke of the engine. The fuel injected from the fuel injection nozzle **10** has a broad injection angle, from 30° to 90° as shown by a dotted line **18,** so that it is injected in a direction towards the vortical flow shown by the arrow **17.** The fuel is injected at an appropriate time prior to top dead center of the piston at an injection pressure of 80 kg/cm^2 to 200 kg/cm^2.

A suitable fuel injection nozzle **10** is a well-known one for use in diesel engines, being of the inwardly flared type so as to impart a spray angle to the injecting fuel which depends upon the taper of the pintle portion thereof.

The fuel injected downwards from the injection nozzle **10** meets the vortical air flow with its initial speed of injection but the direction or motion of the fuel is changed so that it moves in the direction of the vortical flow, fine particles of fuel having little kinetic energy being the first to change direction. Thus, there occurs motion towards the neighborhood of the sparking plug.

On the other hand, larger particles of fuel having greater kinetic energy undergo little or no change in direction of movement until they collide with the top surface of the piston. The spray of fuel is disturbed by the change in direction of the fuel due to the vortical flow of air and by the collision of the fuel with the top surface of the piston, so that the particles of fuel are further refined with the result that the vaporization is further improved, and mixing with air produces a combustible gas mixture most suitable for spark ignition. It is possible to ignite the fuel by providing the sparking plug **16** at the place where this combustible gas mixture is produced, and by adjusting the timing so that when the combustible gas mixture is produced a spark is produced by the sparking plug **16.** In this case, the spray of fuel would not collide directly with the sparking plug **16,** and also larger particles of fuel would not come to the sparking plug **16,** so that the contamination of the sparking plug which occurs when using fuel of low volatility in a conventional engine may be prevented. In addition, since the mixing of the fuel and air is achieved locally, there is no need to restrict the air flow even at a light load, and accordingly, the reduction of the thermal efficiency caused by the throttling of the sucked air may be obviated.

Considering the formation of a gas mixture, there is a predetermined minimum time required for the vapourization of the fuel. If the distance between the fuel injection nozzle and the sparking plug is increased to realize this minimum time, the injected fuel would be scattered, so that only a dilute gas mixture would be present in the neighborhood of the sparking plug upon ignition, with the result that the fuel could not be ignited. In order to provide time for vapourization of fuel while minimizing the distance between the injection nozzle and the sparking plug, it is necessary that the injected fuel particles are caused to move so as to form an ignitable gas mixture in the last part of the spray cloud, and that the timing of the end of injection, the rate of injection and the direction of injection are optimized so that the most suitable gas mixture is formed in the neighborhood of the sparking plug, under every operating condition.

The aforementioned system for igniting the last part of the spray cloud of the fuel by injecting the fuel against the vortical air flow, has the following advantages:

1. The distribution of the spray cloud of the fuel is stable.
2. The time required for vapourization of the fuel may be provided.
3. At a heavy load, the mixing of the fuel with the air is so good that a high output performance may be obtained.
4. The sparking plug may not become wet nor contaminated.

Referring to FIG. **3,** amoung the injected fuel particles the fine particles of the fuel outside the cone of the spray are carried from points **21** and **22,** along the vortical flow path into the region **23.** The larger particles are, after arriving at the neighborhood of the point **20,** successively carried by the vortical flow to the regions **24, 25** and **26,** passing through the neighborhood of the outer periphery of the combustion chamber.

Referring to FIG. **4** the first particles of fuel to be injected reach the lowest region **27,** and those particles which are subsequently injected reach, in turn, the regions **28** and **29.** Accordingly, sufficient fuel to wet the sparking plug may not reach the regions **26** and **29,** until the last spray cloud reaches there. Thus, in order to produce a combustible gas mixture by the termination of the injection of fuel, it is necessary to ensure that the end of the injection of the fuel is constant with respect to the position of the piston over theh whole load range of the engine, or to provide injection characteristics which are suitable for the production of a mixture approximated thereto, especially upon starting the engine, since the temperature in the cylinder is low the rate of vapourization of the fuel is low and the compressb n temperature thereof becomes high. Because the distance between the piston and the injection nozzle becomes short the fuel enters the cavity which prevents dispersal of the fuel. Therefore the timing of the end of the injection of fuel is delayed by an amount in the range of from 10° to 50° (preferably 15° to 30°) of the crank angle, when compared with the normal operating condition, and yet more fuel is injected in order to compensate for the low rate of vapourization of the fuel.

FIG. **5** shows a developed view of a plunger of a fuel injection pump, in which there are provided a lead **30** for adjusting the rate of injection and also for varying the timing of the start of injection and a lead **31** for controlling the timing of the end of injection. A lead **33** is provided for setting the timing of the end of injection, when starting of the engine, at a point slightly later than the end of the injection when the engine is running, to achieve the aforementioned object. Therefore the difference between the timing of the end of injection when the engine is running and when the engine is starting depends upon the distance **34** and the curve for cam lift. Also the rate of injection depends upon the distances **35** and **36** between the leads and the amount of sucking back of the discharge valve of the injection pump, but the stroke **37** of the plunger at the rack position upon starting can be designed to be sufficiently larger than the stroke **36** of the plunger at the position for injecting the maximum quantity of fuel under the operating condition. The rack may operate within the range **38** so as to make the leads **30, 31, 32** and **33** available at the respective conditions. The cam of the injection pump is manufactured by setting its lift curve so that the sustaining time of the rotational motion of the spray of the fuel may become appropriate. The injecting period at the maximum quantity of injection is suitable at 50° – 100° of the crank angle.

Since the gas mixture in the neighborhood of the sparking plug is moving rapidly due to the vortical flow, the mixture of ionized particles between the electrodes of the sparking plug is blown away by the capacitative spark generated as the first part of the electric spark, so that the inductive component subsequent thereto is reduced and consequently the period over which the spark discharges is short. The ignition system can be made to lengthen the period over which the spark discharges to ignite the gas mixture, for example by narrowing the gap between the sparking plug electrodes or by increasing the discharge energy of the sparking plug. Thus, after ignition, the flame is sustained by the gas mixture which is moving along the vortical flow, and when the unburned fuel has burned out, the combustion is completed.

Referring to FIG. **6,** which shows the positions of the injection nozzle and the ignition plug, the position of

the injection nozzle relates to the strength and direction of vortical flow, and also affects the collision of the fuel spray with the piston cavity. The larger the value of r, the distance between the center of the combustion chamber and the tip of the fuel injection nozzle, the higher the speed of the vortical air flow into which the fuel is injected. θ_N is the angle between a radius of the combustion chamber which is parallel to the axis of the fuel injection nozzle and a radius of the combustion chamber which extends through the tip of the fuel injection nozzle. If $\theta_N = 90°$, the fuel is injected opposite to the direction of the vortical flow, while if $\theta_N = 0°$, then the fuel is injected in a direction at a right angle to the vortical air flow. The variation with time of the distribution of the spray cloud can be determined by the combination of $r.\ \theta_N$ and the strength of the generated vortical flow. The strength of the vortical flow depends upon the shape of the inlet port and the size of the inlet valve shroud, but these cannot be enlarged indefinitely, because to do so would result in an increase in the inlet resistance, resulting in a reduction of the inlet efficiency, and also because a high speed of vortical flow will blow out the spark of the spraking plug. The distribution of the spray in the longitudinal direction as shown in FIG. **7**, varies in accordance with the angle θ_s where θ_s is the angle between the axis of the fuel injection nozzle and the axis of the cylinder. If the value of θ_s is increased, the spray of the fuel expands into a broader range along a plane, which has a deleterious effect on the starting characteristics and the light load characteristics of the engine. If the diameter of the cylinder of the engine is small, the fuel spray may collide directly with the cylinder wall, depending upon the position of the injection nozzle, so that when the engine is cool dilution of lubricating oil may occur. On the other hand, if the value of θ_s is too small, the spray of fuel may concentrate within a narrow range, resulting in a local lack of air and consequent reduction in the output power of the engine.

Since the gas mixture in the neighborhood of the sparking plug may be dilute when the quantity of fuel injected is small, at a light engine load and when the amount of fuel vapourized before ignition during the starting period is small, the cavity on the top surface of the piston serves to contain most or all of the injected fuel. As shown in FIG. **7**, since the injection timing is delayed at a light load and during starting of the engine, the piston comes nearer to the top dead center and this effect becomes more remarkable. The configuration of the piston cavity is preferably of a cylindrical shape or other shape approximating to a cylindrical shape having diameter d including the injection nozzle **10** and the sparking plug **16**, and the cavity also has an optimum depth H which is preferably less than or equal to 40 mm. If the shape of the outlet of the cavity is conical, so as to enlarge the outlet, or if it is inclined to the side towards the sparking plug together with the enlargement thereof, then the flow of the gas mixture from the cavity is more smooth, resulting in more advantages. The gap between the top surface of the piston outside of the cavity and the surface of the cylinder head, may be selected so as to obtain the squish effect, whereby the lack of air utilization caused by the lack of time for vapourization and diffusion of the fuel at a heavy engine load is compensated for by the production of as great a squish turbulent flow as possible. For this sort of internal combustion engine, the diameter of the cylinder bore is in the range of 80 mm to 150 mm, and the distance L between the tip of the fuel injection nozzle and the sparking plug is in the range of from 10 mm to 60 mm.

In FIG. **8**, when the rate of injection of fuel has decreased from the maximum point to below a certain level as a result of movement of the fuel injection controlling rack R of the fuel injection pump, i.e. when the slide bar B starts to push the member A slidably supporting said slide bar B, the throttle valve begins to close from its fully open position indicated at the point C in FIG. **9**, and the closing operation of the throttle valve is stopped by the stopper S at the point S in FIG. **9** at which the rate of injection of the fuel is minimum. The supply rate Fa of the intake air in this case can be considerably larger than that during the operation of a carburetor-equipped or manifold injection type gasoline engine with no load (hereinafter abbreviated as N.L.).

As may be understood from the foregoing, the intake air throttle means-equipped injection engine with spark ignition has the three advantages set forth below:

1. A stable ignitable mixture can be formed in the vicinity of the ignition plug.
2. The discharge of hydrocarbons (HC) can be reduced without increasing the concentration of carbon monoxide (CO) in the exhaust gas.
3. The vacuum pressure occurring between the throttle valve and intake valve can be utilized for driving a power brake or other means.

The reason why the foregoing advantages can be obtained will be explained with reference to FIG. **10.** The results shown in FIG. **10** were obtained from an internal combustion engine of the following specifications:

Specifications of the engine submitted for testing (the symbols have the same meanings as defined in the claims):

Bore 80 mm
Stroke 70 mm
Displacement 352 cc
Compression ratio **8.0**
Fuel Kerosene
$2r_n/D = 0.70$
$\theta_N = 25°$
$\theta_S = 30°$
$2r_p/D = 0.57$
$L = 20.00$ mm
$d/D = 0.55$
$H = 30$ mm
Fuel injection ending time during operation with load - **40°** before TDC and constant
Injection rate of fuel - **35** mm^3/cycle at full load, **6.6** mm^3/cycle at N.L.

FIG. **10** shows (i) O.A. air fuel ratio, (ii) the air fuel ratio in the vicinity of the ignition plug and (iii) the air fuel ratio at a portion where the fuel concentration is smallest and the tendency of vacuum pressure, of each of an engine without an intake air throttle valve, (a) an engine with an intake air throttle valve and (b) an engine with an intake air throttle valve, when the load is changed, with the engine speed constant (full load speed and 2,000 rpm).

With reference first to the case of "without an intake air throttle valve", the O.A. air fuel ratio changes greatly from 80:1 at N.L. to 15:1 at the full load (hereinafter abbreviated as F.L.) and in the vicinity of the ignition plug, the air fuel ratio at the time of ignition

fluctuates widely, tending to induce missfire. At a portion where the fuel concentration is smallest, an excessively lean condition from 20:1 (at F.L.) to 500 or larger:1 (at N.L.) occurs, with the result that the flame is extinguished and HCx are discharged from the engine.

On the other hand, in the case of (a) with an intake air throttle valve, the intake air throttle valve begins to close from a load L_1, so that the O.A. air fuel ratio is 40:1 at N.L. Thus, it is very easy to maintain the air fuel ratio in the vicinity of the ignition plug stably in the range from 13:1 to 15:1. The air fuel ratio at the portion where the fuel concentration is smallest, is about 200:1 at N.L., and the discharge of HC from the engine is small. Further, in the case of (b) with an intake air throttle valve, in which the intake air is throttled from a load L_2 corresponding to the intersection of the theCO limit line (the CO concentration in the exhaust gas in this case was 0.5 to 1.0%) the O.A. air fuel ratio is in the range from 25:1 to 15:1. Therefore, the air fuel ratio in the vicinity of the ignition plug can of course be maintained stably in the range from 15:1 to 14:1 and the air fuel ratio even at the portion where the fuel concentration is smallest becomes 130:1, so that the HC concentration in the exhaust gas can be further decreased. As regard the limit of the intake air throttling degree, the fuel concentration high side should be determined by the HC concentration in the exhaust gas and in consideration of stability of ignition, and the fuel concentration low side should be determined by the CO concentration in the exhaust gas in consideration of the fact that the fuel concentration locally becomes larger than that in the vicinity of the ignition plug. However, the ranges for the case (a) with the intake air throttle valve and (b) with the intake air throttle valve, specified above, are generally suitable. The changing tendency of the vacuum pressure in the intake passage between the intake throttle valve and the intake valve is as shown at the lower portion of FIG. **10.** Reference symbols in FIG. **8**:

S : Stopper for the throttle valve in the fully open position (to prevent the opening of the valve from becoming smaller than that at N.L.)
S′ : Spring
S″ : Fulcrum point
T : Throttle valve
W : Spring biasing the throttle valve toward the fully open position
A : Member provided on the throttle valve side of a link interconnecting the throttle valve T and the rack R of the injection pump
B : Member provided on the rack side of the link interconnecting the throttle valve T and the rack R of the injection pump
R : Injection controlling rack of the injection pump
A gap is formed between the members A and B on the maximum injection side.

EFFECTS OF THE PRESENT INVENTION

1. The engine of the present invention can be started whether it is hot or cold and regardless of the vaporability of the fuel used.

BACKGROUND OF THE EFFECT (RELATION WITH THE CLAIMS)

Excessive diffusion of the atomized fuel in a plane including the cylinder axis and the nozzle inclined at 0°–45° to the cylinder axis, is prevented. Excessive diffusion of the atomized fuel can be prevented and the time necessary for other gasification of a poorly vaporizable fuel can be secured.

The cavity prevents excessive diffusion of the atomized fuel by trapping the atomized fuel therein, and since the exit of the cavity is expanded toward the ignition plug, the ignitable mixture formed by the atomized fuel can be positively carried to the ignition plug.

The injection time is retarded at the start of the engine to reduce the time before the fuel is ignited, whereby excessive diffusion of the atomized fuel is prevented and the effect of the cavity if further enhanced. In addition, the amount of the injected fuel is increased to make up the poor vaporability of the fuel. The engine can be started at temperatures below −30°C. even when kerosene is used as fuel. (SAE Paper 720196, page 14, left column, lines 12–14).

2. All of the fuel supplied can be changed to a combustible state, regardless of the engine temperature and the vaporability of the fuel, so that dilution of the lubricating oil can be prevented.

BACKGROUND OF THE EFFECT

The sprayed fuel does not reach the cylinder wall. The cavity prevents the atomized fuel from reaching the cylinder wall.

3. A non-uniform mixture with a minimum concentration distribution variation can be obtained, regardless of the engine temperature, the type of carburetor, the engine speed and the load on the engine. The fuel supplied can be quickly changed to a combustible state. Therefore, the engine follows well the transient phenomena such as abrupt acceleration or deceleration, and the response and flexibility of the engine can be increased.

BACKGROUND OF THE EFFECT

The praticles of fuel injected in the reverse direction slowly change their direction and move towards the ignition plug while being entrained in the swirling flow, so that the mixture at an ignitable concentration stays in the vicinity of the ignition plug for a considerably long time. Such condition is not substantially affected by the engine speed and engine load, and also by the injection time.

4. A high engine output can be obtained and the inclusion of harmful components such as unburned fuel and carbon monoxide in the exhaust gas can be prevented, by supplying the minimum requisite fuel for various oeprating conditions of the engine and making complete use of air for efficient combustion of the fuel.

BACKGROUND OF THE EFFECT

The position of the ignition plug enhances the utility of air (see the SAE Paper, page 7, FIG. **15**).

Unnecessary diffusion of fuel is prevented and complete combustion of fuel is attained.

Since the atomized fuel is ignited at a point close to the end of injection, stable ignition and complet combustion can be achieved.

5. Knocking of the engine is prevented and the combustion ratio can be increased to the economical point allowed by the maximum combustion presure, regardless of the octane value of the fuel, by providing a combustible mixture in the vicinity of the ignition source and an excessively lean mixture at the end gas portion within the combustion chamber.

BACKGROUND OF THE EFFECT

The cavity has a large effect in forming the excessively lean mixture at the end gas portion.

6. A stable mixture distribution can be obtained and the engine output can be adjusted mainly by adjusting the fuel supply rate. The mixture distribution does not change largely with various conditions, so that it is possible to throttle the intake air or to use the throttle valve in its fully open position throughout the operation of the engine without throttling the air. In some engines, it is advantageous to throttle the air during operation of the engines with light loads or no load, for preventing miss-fire or for minimizing HC in the exhaust gas, and in such cases the engines can be operated satisfactorily with throttled air.

It should now be apparent that the internal combustion engine as described hereinabove possesses each of the attributes set forth in the specification under the heading "Summary of the Invention" hereinbefore. Because the internal combustion engine of the invention can be modified to some extend without departing from the principles of the invention as they have been outlined and explained in this specification, the present invention should be understood as encompassing all such modifications as are within the spirit and scope of the following claims.

What is claimed is:

1. An internal combustion engine which includes a cylinder, housing a movable piston means for introducing air into a combustion chamber in the cylinder in a manner to generate vortical flow around the periphery of the combustion chamber, a fuel injection nozzle for injecting fuel into the combustion chamber in a direction so that the fuel has a component of velocity in a direction opposite to the direction of the vortical air flow in the vicinity of the fuel injection nozzle, and a sparking plug for igniting the combustible mixture formed on mixing of the fuel and the air, in which the fuel injection nozzle is situated so that

1. $2r_N/D$ has a value in the range of from ⅓ to ¾ inclusive, where r_N is the distance between the tip of the injection nozzle and the center of the combustion chamber and D is the bore of the combustion chamber;
2. θ_N has a value in the range of from 0° to 60° inclusive where θ_N is the angle between a radius of the combustion chamber which is parallel to the axis of the fuel injection nozzle and a radius of the combustion chamber which extends through the tip of the fuel injection nozzle;
3. θ_S has a value in the range of from 0° to 45° inclusive, where θ_S is the angle between the axis of the fuel injection nozzle and the axis of the cylinder; the sparking plug is situated so that
4. $2r_p/D$ has a value in the range of from ¼ to ¾ inclusive, where r_p is the distance between the sparking plug and the center of the combustion chamber and D is as defined above;

and the distance L between the tip of the fuel injection L and the sparking plug lies within the range of from 10 to 60 mm. inclusive.

2. An interanl combustion engine as claimed in claim **1** which includes a piston which has a cavity in its upper surface, the cavity having a shape of diameter *d*, in which *d*/D a value in the range of from ⅓ to ⅔ inclusive.

3. An internal combustion engine as claimed in claim **2** in which the cavity has a depth H less than or equal to 40 mm.

4. An internal combustion engine as claimed in claim **1** further including:

a fuel injection pump for injecting fuel through said fuel injection nozzle;

means coordinated with the disposition of the piston for controlling the termination of fuel injection upon each achievement of a first disposition of the piston only under normal engine running conditions and upon each achievement of a second, later disposition of the piston only under transient engine starting conditions.

5. The internal combustion of claim **3** wherein the injection nozzle includes surface means providing a fuel spray along the nozzle axis with a cone angle of 30°–90°; the outlet of the piston cavity enlarging upwardly in diameter and being inclined towards the sparking plug.

6. The internal combustion engine of claim **1** wherein the means for introducing air into the combustion chamber includes an intake throttle valve and control means therefor operatively tied to the injection of fuel to reduce the amount of air introduced in response to sensing a decrease in the amount of fuel being injected and to increase the amount of air introduced in response to sensing an increase in the amount of fuel being injected, whereby an ignitable air/fuel mixture is supplied to the vicinity of the ignition plug.

7. The internal combustion engine of claim **6** wherein the control means for the intake throttle valve includes:

a fuel injection pump supplying the fuel injection nozzle;

rack means on the pump, mounted for longitudinal movement corresponding in magnitude and sense to increasing and decreasing rate of fuel injection;

a control member connected to the throttle valve means and operatively connected to the rack;

spring means connected to the throttle valve means and tending to urge the valve toward a fully open condition;

stop means for preventing the throttle valve means from fully closing while fuel is being injected into the combustion chamber;

the control member being operated by movement of the rack to begin to close the throttle valve means against the action of the spring when the amount of injection of fuel has decreased from the maximum to below a selected level;

the stop means being stationed to prevent further closing the throttle valve means beyond a selected extent, even though the amount of fuel being injected may continue to decrease.

8. An internal combustion engine, comprising:

a cylinder providing a combustion chamber having a piston slidably received therein for increasing and

decreasing the volume of the combustion chamber, the piston having a cavity opening internally of the combustion chamber;

a fuel injection device, including an injection pump having means for fixing the timing for termination of fuel injection at a given value during normal running of the engine, and for fixing said timing for termination at another, further delayed value during start-up of engine running;

inlet means for introducing combustion-supporting gas ranging from air to a too-lean-to-be combusted mixture of fuel and air into the combustion chamber from a site and along a path which generates a swirling flow circumferentially of the combustion chamber in a first angular direction;

a fuel injection nozzle for injecting fuel into said combustion chamber from a site and along a path which angularly opposes the swirling flow of combustion-supporting gas;

an ignition device for generating an electric spark within the combustion chamber at a site where the combination of the introduced combustion-supporting gas and the injected fuel has created a combustible gas mixture;

the fuel injection nozzle being disposed in accordance with the following relations:

2. $2r_N/D$ has a value in the range of from ⅓ to ¾ inclusive, where r_N is the distance between the tip of the injection nozzle and the center of the combustion chamber and D is the bore of the combustion chamber;

2. θ_N has a value in the range of from 0° to 60° inclusive, where θ_N is the angle between a radius of the combustion chamber which is parallel to the axis of the fuel injection nozzle and a radius of the combustion chamber which extends through the tip of the fuel injection nozzle; and

3. θ_S has a value in the range of from 0° to 45° inclusive, where θ_S is the angle between the axis of the fuel injection nozzle and the axis of the cylinder; the sparking plug is situated so that

the site of spark generation within the combustion chamber being eccentrically disposed in relation to the longitudinal axis of the cylinder in accordance with the following relations:

4. $2r_p/D$ has a value in the range of from ¼ to ¾ inclusive, where r_p is the distance between the sparking plug and the center of the combustion chamber and D is as defined above;

5. the site of spark generation within the combustion chamber being eccentrically disposed angularly downstream of the fuel injection nozzle with respect to the path of swirling flow of the combustion-supporting gas; and

6. the distance L between the site of the fuel injection nozzle and the site of spark generation lies with the range of $10\text{mm} \leq L \leq 60\text{mm}$; and

the piston cavity being sized in accordance with the following relations:

7. $⅓ \leq d/D \leq ⅔$, where d is the diameter of the cavity; and

8. $H \leq 40\text{mm}$, wherein H is the depth of the piston cavity axially of the cylinder.

9. The internal combustion engine of claim **8**, wherein:

the inlet means for introducing combustion-supporting gas into the combustion chamber includes an intake throttle valve and control means therefor operatively tied to the injection of fuel to reduce the amount of combustion-supporting gas introduced in response to sensing a decrease in the amount of fuel being injected, whereby an ignitable air/fuel mixture is supplied to the vicinity of the ignition plug.

10. The internal combustion engine of claim **9**, wherein the control means for the intake trottle valve includes:

a fuel injection pump supplying the fuel injection nozzle;

rack means on the pump, mounted for longitudinal movement corresponding in magnitude and sense to increasing and decreasing rate of fuel injection;

a control member connected to the throttle valve means and stationed in proximity to the rack;

spring means connected to the throttle valve means and tending to urge the valve toward a fully open condition;

stop means for preventing the throttle valve means from fully closing while fuel is being injected into the combustion chamber;

the control member being operated by movement of the rack to begin to close the throttle valve means against the action of the spring when the amount of injection of fuel has decreased from the maximum to below a selected level;

the stop means being stationed to prevent further closing of the throttle valve means beyond a selected extent, even through the amount of fuel being injected may continue do decrease.

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