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Limits of Flame Propagation in Two-Stroke Cycle Gasoline Engines*

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In a two-stroke cycle gasoline engine, the limits of flame propagation were measured under various operating conditions, and the composition, temperature and pressure of the cylinder gas at the limits were investigated. The adiabatic flame temperature in a lean or rich limit mixture was calculated on the basis of the cylinder gas state at the time of ignition. It is shown that, in the case where the engine speed and the ignition timing are fixed, the adiabatic flame temperature remains almost constant along both the lean and rich limits, regardless of the residual gas concentration or the mixture temperature.

1. Introduction

In a two-stroke cycle gasoline engine, misfiring begins to occur at a certain fuel-air ratio when the fuel concentration is decreased at the fixed engine speed and delivery ratio. As the fuel concentration is further reduced, misfiring occurs frequently and eventually the engine can no longer operate. On the contrary, with rich mixtures the engine does not fail to operate, even though the fuel-air ratio is further increased beyond the ratio where misfiring begins to occur. The engine can operate under the conditions where firing occurs regularly, for example, every second cycle, or every third cycle, etc. That is, there exists a certain range of fuel-air ratios where "firing on every cycle" or "regular intermittent firing" ("firing on every second cycle", or "firing on every third cycle", etc.) occurs. The mechanism which produces the regular intermittent firing has been studied by Sawa⁽¹⁾ and Kamimoto et al.⁽²⁾. Sawa explained the combustion limits in terms of the relation between the fuel concentration and the oxygen concentration. Kamimoto et al. reported that the combustion limits in the case of low delivery ratios were governed by the oxygen concentration and gas temperature at the time of ignition. However, the various effects of residual gas concentration, mixture temperature and other factors on the limits have not been made fully clear yet.

In this study, a two-stroke cycle gasoline engine was operated at various delivery ratios, fuel-air ratios and engine speeds,

and then the outer limits of the range for firing on every cycle or regular intermittent firing were measured. Further, the scavenging properties of the test engine were made clear by analyzing the gases in the cylinder and exhaust. From these results, the effects of residual gas concentration and mixture temperature on the limits of flame propagation were investigated, and their composite effects were estimated by using the adiabatic flame temperature at the time of ignition as a characteristic value. Further, the limits of flame propagation in the engine were compared with those in quiescent mixtures, and the effects of engine speed and ignition timing were investigated.

A single-cylinder, air-cooled, two-stroke cycle gasoline engine of the crank-case compression and Schnürle scavenging type, having a stroke volume of 123 cm³ (bore=56 mm, stroke=50 mm) and an effective compression ratio of 6.8:1, was used. The engine was equipped with an independent lubricating system. A conventional ignition system with an induction coil (12 V) was used, and the ignition timing was set at 21° before TDC except in the case where the effect of ignition timing was tested. The fuel used was commercially available gasoline, and the fuel flow was adjusted by regulating the pressure in the float chamber of the carburetor. The air flow rate was measured by an orifice flow meter set up before a surge tank. Throughout the experiment, the pressure in the surge tank was kept equal to the atmospheric pressure by supplying air through a blower.

2. Misfire ratio

Success or failure in flame propagation in the combustion chamber was determined by observing the ionization signals and also with reference to the pressure diagrams. As shown in Fig.1, three ionization gaps were set up at a radius of 16 mm

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around the spark plug. The ionization gaps were wired in a parallel circuit, and ionization signals were displayed on an oscilloscope. In the mixtures near the limits of flame propagation, the signals were weak and were superimposed on the spark noise, but it was possible to discriminate the misfiring cycles, where no ionization signals appeared, from the firing cycles. By observing about 200 cycles at random in operating the engine under fixed conditions, a misfire ratio, i.e., the ratio of the number of misfiring cycles to that of all the cycles observed, was determined. Thus, "misfire" was defined as the condition where no flames could grow up to such a size that they were detectable by any of three ionization gaps. By the above definition, the combustion limits were, in effect, the limits of flame propagation in this study.

In investigating the limits of flame propagation in engines, sufficient attention must be paid to the mixture homogeneity and the quenching effect of spark plug electrodes. In the test engine, air and fuel were mixed almost homogeneously in the

Fig. 1 Locations of a spark plug and three ionization gaps

Fig. 2 Effect of spark gap length on the limits of flame propagation

crankcase before being fed into the cylinder. In order to test the quenching effect of the spark plug electrodes, the fuel-air ratio limits where misfiring began to occur were measured by using spark plugs with the gap lengths of 0.2-1.7 mm at an engine speed n of 2850 rpm and delivery ratios l_d of 0.41 and 0.70. The results shown in Fig. 2 revealed that the lean and rich limits hardly changed at both l_d , when a spark plug with a gap length exceeding 0.7 mm was used. Therefore, in the following experiment a conventional spark plug with a gap length of 0.7 mm was used for convenience. The results in Fig. 2 also indicated that a spark plug with a gap length shorter than 0.5 mm slightly narrowed the rich limits, and steeply narrowed the lean limits. It is noticeable that quenching by the electrodes has little effect on the rich limits but has a remarkable effect on the lean limits.

The misfire ratios were measured over the widest ranges of delivery ratios (l_d) and fuel-air ratios (F/A) where the engine speed (n) could be kept constant by an electric brake. The engine speed was set at 2000, 2850, 4000 and 5000 rpm. The misfire ratios at n of 2850 rpm are shown in Fig. 3 as an example. In this figure, each fuel-air ratio range with a constant misfire ratio of 0.5, 0.67 or 0.75 on the rich side corresponds to firing on every second cycle, firing on every third cycle or firing on every fourth cycle, respectively. Such regular intermittent firing occurred stably. However, the engine operation in the region of transition between two different stable conditions was unstable and tended to fall into either stable condition with a slight change in F/A , l_d or n . On the lean side, the regular intermittent firing did not occur, and the engine operation became unstable with a slight increase in the misfire ratio, eventually coming to a stop. The exhaust gas temperature measured by a thermo-couple near the exhaust port is also

Fig. 3 Example of measured misfire ratio

shown in Fig.3. The change in the exhaust gas temperature correlates well with the change in the misfire ratio. Therefore, the occurrence of misfiring may be inferred from the exhaust gas temperature.

By using the test data at each fixed n as shown in Fig.3, the lean and rich limits of flame propagation at each l , were obtained from the fuel-air ratios at the point where misfiring began to occur. In the case of regular intermittent firing, the limits were determined by the fuel-air ratios where, for example, the transition from firing on every second cycle to firing on every third cycle began to occur. As shown in Fig.4, the relation between delivery ratios and the lean and rich limits was established. When l was lowered at any given n , the lean limit F/A increased slightly, and the rich limit F/A for each pattern of firing decreased steeply. In this figure, " $i=1$ ", " $i=2$ " and " $i=3$ " indicate "firing on every cycle", "firing on every second cycle" and "firing on every third cycle", respectively.

3. Scavenging properties

In order to estimate the composition and temperature of the cylinder gas at the time of ignition, the scavenging efficiency and the trapping efficiency of the test engine were determined by the following method. In operating the engine with rich mixtures, the cylinder gases before and

Fig.4 Relation between delivery ratio and fuel-air ratio at the limits of flame propagation

Fig.5 Scavenging efficiency

after combustion (i.e., after and before scavenging, respectively) were extracted by an electro-magnetic gas sampling valve located at the cylinder head. Further, the exhaust gas was extracted by a syringe-type sampler from the furthest rear of five mixing tanks (3 ltr. vol.), which were fitted to the exhaust pipe, so that the representative exhaust gas could be obtained. These sample gases obtained at various l , and n were analyzed by a gas chromatograph, and the O_2 -concentration in each gas was determined. The scavenging efficiency η_s and the trapping efficiency η_{tr} were calculated from the equations⁽⁹⁾:

$$\eta_s = \frac{g_2 \left(\frac{O_2}{N_2} \right) - g_1 \left(\frac{O_2}{N_2} \right)}{g_l \left(\frac{O_2}{N_2} \right) - g_1 \left(\frac{O_2}{N_2} \right)}$$

$$\eta_{tr} = \frac{g_l \left(\frac{O_2}{N_2} \right) - g_a \left(\frac{O_2}{N_2} \right)}{g_l \left(\frac{O_2}{N_2} \right) - g_1 \left(\frac{O_2}{N_2} \right)}$$

where:

$g(O_2/N_2)$ = ratio of O_2 -mass fraction to N_2 -mass fraction

$g_l(O_2/N_2) = 0.3073$

Subscripts:

1 = condition before scavenging (after combustion)

2 = condition after scavenging (before combustion)

a = condition of exhaust gas

l = condition of intake mixture

The values of η_s and η_{tr} are shown in Figs. 5 and 6, respectively. At n of 4000 and 5000 rpm, it was assumed that the cylinder gases after combustion had the same composition as the combustion gas at n of 2850 rpm, since the combustion gases could not be extracted owing to a mechanical restriction in the operation of the sampling valve. Figures 5 and 6 show that η_s increases slightly with n , and that η_{tr} is hardly affected by n .

Next, the scavenging properties of the test engine with regular intermittent firing were investigated. By controlling the operation of the sampling valve by the magnitude of the peak pressure in the cycle followed by the cycle where the gases were sampled,

Fig.6 Trapping efficiency

the cylinder gases after scavenging in the misfiring and firing cycles were extracted. The concentration of the fresh charge in the cycle preceded by a misfiring cycle increases owing to scavenging two times. In the case where firing occurs regularly every i -th cycle ($i=1,2,\dots$) as shown in Fig.7 (in the case $i=3$), the cycle which is scavenged j times ($j=1,2,\dots,i$) after combustion is denoted by $(i-j)$ cycle. The O_2 -concentration in the cylinder after scavenging in the $(i-j)$ cycle was investigated by using the accumulated delivery ratio defined as $j l_0$. Here, $j l_0$ corresponds to the total delivery ratio in the process where scavenging occurs j times. By plotting the values of $G_2(O_2/N_2)$ in the cylinder gas after scavenging against $j l_0$, it was revealed that these values lay generally on one curve regardless of the values of i and j as shown, for example, in Fig.8. Therefore, the concentration of the fresh charge in any $(i-j)$ cycle may be estimated from the value of $j l_0$. The accumulated concentration of the fresh charge in the $(i-j)$ cycle, namely, the accumulated scavenging efficiency $\eta'_{s(ij)}$ was defined as $\eta'_{s(ij)} = G_{Fi-j} / G_{zi-j}$, where G_{zi-j} and G_{Fi-j} were the mass of the total gas in the cylinder and that of the fresh charge, respectively. The value of $\eta'_{s(ij)}$ was obtained from the equation⁽⁴⁾

$$\eta'_{s(ij)} = \frac{G_2 \left(\frac{O_2}{N_2} \right)_{i-j} - G_1 \left(\frac{O_2}{N_2} \right)_{i-i}}{G_l \left(\frac{O_2}{N_2} \right) - G_1 \left(\frac{O_2}{N_2} \right)_{i-i}}$$

In Fig.9, $\eta'_{s(ij)}$ is plotted against $j l_0$. Here,

Fig.7 Example of regular intermittent firing (in the case of $i=3$)—typical oscillogram of cylinder pressure and ionization current

Fig.8 Variation of $G_2(O_2/N_2)$ with accumulated delivery ratio

$\eta'_{s(ij)} = \eta_s$ when $j=1$ or $i=1$, namely, the curves of $\eta'_{s(ij)}$ plotted against $j l_0$ coincide with those of η_s against l_0 for firing on every cycle.

4. Composition and temperature of the cylinder gas at the time of ignition

The temperature of the cylinder gas at the time of ignition (i.e., at the crank angle of ignition timing) at the limits of flame propagation, T_{ig} , was estimated from the equation

$$T_{ig} = P_{ig} V_{ig} / G_z R_z$$

Here, P_{ig} and V_{ig} are the measured values of the cylinder pressure and volume at the time of ignition, respectively. The values of G_z and R_z (i.e., the mass and gas-constant of the cylinder gas, respectively) were obtained by the following procedure.

First of all, the value of G_z for firing on every cycle was obtained, by using the measured values of η_s and η_{tr} , from the relation

$$G_z = l_0 G_h \eta_{tr} / \eta_s$$

Here, l_0 is the delivery ratio in the atmospheric condition, and G_h is the mass of the fresh charge occupying the stroke volume in the atmospheric condition. In the case of firing on every i -th cycle, G_z was replaced by G_{zi-i} (the mass of the cylinder gas in the firing cycle). In the same way as in Chapter 3, the accumulated concentration of the fresh charge in the $(i-i)$ cycle, $\eta'_{s(i)} (= G_{Fi-i} / G_{zi-i})$, was obtained from the equation⁽⁴⁾

$$\eta'_{s(i)} = \frac{G_2 \left(\frac{O_2}{N_2} \right)_{i-i} - G_1 \left(\frac{O_2}{N_2} \right)_{i-i}}{G_l \left(\frac{O_2}{N_2} \right) - G_1 \left(\frac{O_2}{N_2} \right)_{i-i}}$$

On the assumption that l_0 in every cycle is constant, the total mass of the intake mixture delivered during i cycles may be expressed by $i l_0 G_h$. The value $\eta'_{tr(i)}$, which is defined as $\eta'_{tr(i)} = G_{Fi-i} / i l_0 G_h$, corresponds to the trapping efficiency when i revolutions of the engine are treated as one cycle. The value of $\eta'_{tr(i)}$ was obtained by the equation⁽⁴⁾

$$\eta'_{tr(i)} = \frac{G_l \left(\frac{O_2}{N_2} \right) - G_a \left(\frac{O_2}{N_2} \right)}{G_l \left(\frac{O_2}{N_2} \right) - G_1 \left(\frac{O_2}{N_2} \right)_{i-i}}$$

Thus, by using these values of $\eta'_{s(i)}$ and $\eta'_{tr(i)}$,

Fig.9 Accumulated scavenging efficiency

the value of G_{xi-i} was determined from the relation

$$G_{xi-i} = i l_0 G_h \eta'_{tr(i)} / \eta'_{s(i)}$$

Secondly, R_z was obtained from the equation

$$R_z = \sum_{\nu} g_{\nu} R_{\nu}$$

where g_{ν} and R_{ν} were the mass fraction and the gas-constant of the ν -th component of the cylinder gas before combustion, respectively. The value of g_{ν} was calculated from the value of η_s or $\eta'_{s(i)}$ and the combustion gas composition (shown in Fig.10), which was calculated on the following assumptions: the gasoline used has a composition of C_8H_{18} ; the amount of lubricating oil burned is negligible (the consumption of oil was less than 1.5% of that of gasoline); the equilibrium constant for the water-gas reaction is 3.4 (the value at 1700°K) in the rich mixture⁽³⁾.

5. Limits of flame propagation

In Figs.11(a)-(c) are shown the effects of residual gas concentration on the limits of flame propagation at n of 2000, 2850 and 4000 rpm. These figures were obtained from the data in Fig.4. The mass fraction of the residual gas g_R was calculated from the relation $g_R = 1 - \eta_s$ for firing on every cycle and $g_R = 1 - \eta'_{s(i)}$ for regular intermittent firing, by using the value of the scavenging efficiency at each delivery ratio. These figures reveal that at any given n the increase in g_R owing to the decrease in L brings both the lean and rich limits closer to the stoichiometric fuel-air ratio (equivalence ratio of 1.0). The lean limit narrows slightly and the rich limit narrows considerably with an increasing g_R . In those figures, the measured points of the limits for firing on every cycle ($i=1$) and those for firing on every second cycle ($i=2$) lie on a common curve. This fact suggests that the foregoing method of data analysis is effective. The temperature of the cylinder gas at the time of ignition, T_{ig} , is also shown in Figs.11(a)-(c). The temperature T_{ig} was higher than approximately 700°K, and was dependent on g_R and n . At each n , T_{ig}

rose nearly linearly with an increasing g_R , yielding a difference of more than 200°K between the maximum and the minimum. The increase in T_{ig} with an increasing n may be caused by the rise in the temperature of the cylinder wall and by the drop in the heat loss from the gas.

As seen above, the residual gas concentration and the cylinder gas temperature varied markedly along the limits of flame

Fig.10 Combustion gas composition (calculated values)

(a)

(b)

(c)

Fig.11 Effect of residual gas concentration on the limits of flame propagation, and variation of the cylinder gas temperature at the time of ignition (T_{ig}) with residual gas concentration

propagation. Further the residual gas composition varied with F/A (see Fig.10). Therefore, it was necessary to estimate their composite effects on the limits by adopting an appropriate characteristic value. In this study, for this purpose, the adiabatic flame temperature in the lean or rich limit mixture was calculated by using the temperature T_{ig} as the mixture temperature before combustion. The adiabatic flame temperatures (T_f) calculated from the data in Figs.11(a)-(c) were similar values in each case: at n of 2000 rpm, T_f at the lean limit ($T_{f,lean}$) = 1950–2120°K and T_f at the rich limit ($T_{f,rich}$) = 1810–1870°K; at n of 2850 rpm, ($T_{f,lean}$) = 1950–2110°K and ($T_{f,rich}$) = 1790–1900°K; at n of 4000 rpm, ($T_{f,lean}$) = 1980–2160°K and ($T_{f,rich}$) = 1900–2020°K. The cylinder gas compositions yielding an average for the values of ($T_{f,lean}$) or ($T_{f,rich}$) at a given n were calculated and shown by the curve for constant adiabatic flame temperature [indicated by T_f in Figs.11(a)-(c)]. As a result, it was found that at any given n the measured points for the limits of flame propagation were distributed generally near the curves for constant adiabatic flame temperature except in the case of high values of g_R . That is, in spite of wide changes in the composition and temperature of the cylinder gas, the adiabatic flame temperature remained substantially constant along both the lean and rich limits.

6. Discussion of experimental results

In regard to the limits of flame propagation in quiescent mixtures, many experimental studies have revealed the effects of diluent gas concentration, mixture temperature and pressure on the limits. With reference to these results, the limits of flame propagation in the engine will be discussed below.

6-1 Effect of residual gas concentration

In Fig.12 are shown the limits of flame propagation in the gasoline-air mixture diluted with N_2 or CO_2 at the room temperature and atmospheric pressure. This figure was obtained by re-plotting the data from Jones and Gilliland⁽⁵⁾ for the relation between the

Fig.12 Limits of flame propagation in gasoline-air- N_2 (or CO_2) mixture (data from Jones and Gilliland)

gasoline- O_2 equivalence ratio and the mass fraction of diluent N_2 or CO_2 in the mixture. The gasoline was assumed to have the composition of C_8H_{18} . Since the residual gas in the engine consists mainly of N_2 (about 70 % in mass fraction) as shown in Fig.10, the dilution with residual gas is seen to have nearly the same degree of suppression effect on flame propagation as the dilution with N_2 . In comparison between Figs.11(a)-(c) and Fig.12 (N_2 -curve), the lean limit in the engine is slightly leaner than that in the quiescent mixture, while the rich limit in the engine is considerably leaner than that in the quiescent mixture. This is probably due to the differences in the mixture temperature, pressure and mixture motion between these two cases.

Calculating the adiabatic flame temperatures along the lean limits in Fig.12 yielded similar flame temperatures of 2050–2090°K in both the cases of N_2 dilution and CO_2 dilution. The effects of these diluent gases on the lean limits may be explained on the basis of the constancy of the adiabatic flame temperature. On the contrary, the calculated adiabatic flame temperatures along the rich limits in Fig.12 did not remain constant; they increased from several hundreds of degrees K to about 2000°K as the fuel concentration decreased. In the rich mixture, the real flame temperature may differ considerably from the calculated flame temperature under the influence of preferential diffusion⁽⁶⁾. It seems that the real flame temperature may not vary so markedly along the rich limit as the calculated one, and the constancy of the flame temperature may be established. It is conceivable that, in the engine, the influence of preferential diffusion may not be so strong owing to intense turbulence and therefore the adiabatic flame temperature along the rich limit remains constant.

6-2 Effect of mixture pressure

The limits of flame propagation in quiescent mixtures are variously affected by pressure change⁽⁷⁾. Generally, for hydrocarbon fuels, as the pressure increases from 1 atm to several tens of atmospheres, the rich limit broadens considerably, while the lean limit narrows slightly at first and then broadens a little. In the present experiment, the cylinder pressures at the time of ignition varied only about 2 kg/cm² (from 7 to 9 kg/cm²) along the lean and rich limits. Such a small amount of variation is seen to have little effect on the limits.

6-3 Effect of mixture temperature

With regard to the effect of mixture temperature on the limits in quiescent mixtures⁽⁸⁾, it is generally known that the volumetric fuel concentration of the lean limit decreases and that of the rich limit increases nearly linearly with an increasing temperature. That is, the mixture compositions at the limits vary such that their adiabatic flame temperatures may remain constant.

In two-stroke cycle engines, the increase in the residual gas concentration is generally accompanied by the rise in the

cylinder gas temperature. Therefore, the suppression effect of the residual gas in two-stroke cycle engines on flame propagation is reduced in comparison with the effect of the recirculation of the cold exhaust gas in four-stroke cycle engines.

6-4 Effects of engine speed and ignition timing

Figure 13 illustrates the variations in the fuel-air ratios of the limits of flame propagation with the engine speed n in the condition where T_{ig} varies little with n ($g_R=0.2$). The rise in T_{ig} over the n range of 2000–4000 rpm was about 80°K on the lean side and about 50°K on the rich side. This figure reveals that an increasing n slightly narrows the lean limit, while it considerably decreases the rich limit. As obvious from the results in Chapters 5 and 6-1, the adiabatic flame temperatures T_f [indicated in Figs.11(a)–(c)] at the lean limits in the engine (2030–2145°K) resemble closely the mean adiabatic flame temperature at the lean limits in the quiescent mixtures (2070°K). On the contrary, the adiabatic flame temperatures T_f at the rich limits in the engine (1845–1980°K) are much higher than those at the rich limits in the quiescent mixtures. This means that the former rich limits are much narrower than the latter rich limits. These adiabatic flame temperatures at both the lean and rich limits in the engine rise gradually with an increasing engine speed, that is, flames tend to be more difficult to propagate at

Fig.13 Effect of engine speed on the limits of flame propagation

Fig.14 Effect of ignition timing on the limits of flame propagation

higher engine speed.

In Fig.14 is shown the effect of ignition timing on the limits at n of 2850 rpm and l_0 of 0.70. Retarding the ignition timing from 21° to 5° before TDC slightly narrows both the lean and rich limits. As the ignition timing comes closer to TDC, the flames should be easier to propagate because of the rise in T_{ig} . Therefore, the tendency in Fig. 14 can not be explained on the basis of the constancy of the adiabatic flame temperature at the time of ignition. At retarded ignition timing or high engine speed, the limits may be narrowed, probably because of the shortage of the time interval available for flame propagation. From an alternative standpoint, the flame propagation may possibly be inhibited by disturbances which are enhanced by engine speed. The rate of the decrease in the cylinder gas temperature or pressure after TDC, the gas flow velocity, and the turbulence intensity are considered the disturbances. In fact, the limits of flame propagation in the test engine are affected also by the factors which are relevant to time or rate, i.e., ignition timing or engine speed. Taking this fact into consideration, it is conceivable that whether the flames propagate or not in engine combustion chambers may be determined in the process of flame development. However, in the case where the engine speed and the ignition timing are fixed as shown in Fig. 11, it seems reasonable to adopt the adiabatic flame temperature at the time of ignition as a characteristic value in estimating the composite effects of residual gas concentration and mixture temperature on the limits of flame propagation.

7. Conclusions

In a two-stroke cycle gasoline engine, the limits of flame propagation were measured over a wide range of operating conditions. Further, the scavenging efficiency and the trapping efficiency of the engine were determined by the gas sampling method. From these results, the cylinder gas temperature at the time of ignition (T_{ig}) and the residual gas concentration at the limits were estimated. By using T_{ig} as the mixture temperature before combustion, the adiabatic flame temperature in the lean or rich limit mixture was calculated in estimating the composite effects of residual gas concentration and mixture temperature on the limits. The conditions limiting the flame propagation in two-stroke cycle gasoline engines were discussed. The main results are as follows:

- (1) Increasing the residual gas concentration by lowering the delivery ratio slightly narrowed the lean limit, but considerably narrowed the rich limit.
- (2) When the engine speed and the ignition timing were fixed, the adiabatic flame temperature remained almost constant along both the lean and rich limits, in spite of wide changes in the residual gas concentration and the mixture temperature.
- (3) The limits of flame propagation in

the engine were compared with those in the quiescent mixtures (gasoline-air-N₂ or CO₂) by using their adiabatic flame temperatures. It was revealed that the lean limit in the engine did not differ so much from that in the quiescent mixtures but the rich limit in the engine was narrower than in the quiescent mixtures. Both the lean and the rich limits in the engine were made narrower by increasing the engine speed or by retarding the ignition timing.

(4) The quenching effect of the spark plug electrodes was slight in the rich mixture, but remarkable in the lean mixture.

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