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A Study on Hydrogen-fueled Diesel Combustion*

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A compression ignition of hydrogen fuel has been investigated using a conventional swirl chamber type diesel engine. It has been suggested that the hydrogen-fueled diesel combustion is attainable in spite of the expected difficulty due to a high resistance to autoignition. Smooth running can be obtained if a proper amount of the fuel leakage vitiates the charge in the swirl chamber prior to the injection, otherwise ignition fails to ignite leading to an abnormal combustion with a strong knocking and to misfiring. The conditions required for a smooth burning have been studied by varying the amount and the timing of preinjections and the quantity of the fuel leakage. The test results have shown that the fuel leakage at an adequate rate keeps the hot core in the midst of the swirl chamber from being destroyed and promotes ignition of the main fuel, thus resulting in smooth running.

1. Introduction

In view of pollutant emissions and energy shortage, special attention is drawn nowadays to several alternate fuels for internal combustion engines. They include alcohols, natural gas, synthesized fuels from coal, and hydrogen, some of which are found promising from the standpoint of energy resources. Among others, hydrogen is counted as one of the most promising.⁽¹⁾ It is perhaps the best alternate fuel to achieve the maximum utilization of energy and to fulfill the requirement of low emission. In fact, a number of studies on the hydrogen fueled internal combustion engines have been conducted at home and abroad^{(2) (3)}.

As has been widely accepted, there are several differences in the physical and chemical properties between the hydrogen fuel and other ordinary hydrocarbon fuels. Widely inflammability limits and a higher burning velocity are specific to hydrogen. Another characteristic of it that deserves attention is a greater resistance to autoignition. Owing to these differences, it has been supposed that a hydrogen-fueled spark-ignition engine inherently requires some modifications from the conventional design in the carburetor and the combustion system.⁽⁴⁾ In particular, an appropriate means for assuring moderate combustion is indispensable because of its high burning velocity. The usual method of direct introduction of the fuel into the suction reduces the net air charged in the cylinder, at the expense of the power output. For this either air boosting

or direct injection of the fuel into the cylinder is demanded. An idea of establishing a hydrogen fueled diesel engine seems to be fascinating partly because of an easier combustion control and partly because of no loss in air supply. At least from the viewpoint of combustion, there is little restriction that will hinder the increase in the cylinder diameter. However, such an engine has not been proposed anywhere to date obviously because of an expected difficulty in the compression ignition of the hydrogen fuel.⁽⁵⁾

In the present study, the possibility of achieving a full hydrogen operated diesel engine has been explored experimentally. Although a pure compression ignition of hydrogen is limited owing to the high resistance to autoignition, still it can be suggested from this study that such an engine is feasible provided suitable conditions are satisfied for the combustion system.

2. Experimental apparatus and procedure

Experimental tests were carried out on a conventional water-cooled single-cylinder naturally-aspirated four-stroke cycle engine of a divided-chamber type, having a cylinder diameter of 95 mm and a stroke of 115 mm (Yanmar Diesel Co., ST-95 type reconstructed for test). Fuel injection was performed directly into the pre-chamber in quite the same manner as in the ordinary diesel engine, by using a gas injector devised for the test, Fig. 1. This injector uses a poppet valve which operates in a fully hydraulic manner; the valve stem is linked to a hydraulic actuator D with a push rod. A conventional Bosch type jerk pump for diesels (PE1A70B type) was used to drive this poppet valve, and also a conventional nozzle assembly was employed as the hydraulic piston, in which a single hole nozzle of 0.35 mm drilling with a maximum lift of 0.8 mm was included. A pressurized hydrogen gas was fed to the injector assembly at the inlet B in the figure from hydrogen cylinders after passing through a pressure regulator, a metering cylinder, and finally a pipeline having an inner diameter of 12.7 mm and a length of 1 m. The

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Fig. 1 Hydrogen gas injector

details of the valve tip of the hydrogen injector are given in Fig. 1. In the earlier stages of the study, the type I tip was employed, but a small fuel leakage was detected later, and such a slight leakage turned out to have a large influence upon the quitness of the engine running, as will be described more closely later. For this reason, the original tip was replaced by a type II redesigned tip, with use of which the latter half of the experimental work was conducted. To make sure the valve seated accurately, this new tip adopted a longer guide for the stem. Besides, the stem was nitrided and the seat was finally plated with stellite.

Preliminary tests showed that the operation of the hydrogen injector thus developed was satisfactory; injection was accurately timed and the fuel quantity could be regulated from an idling condition to the full load condition. It turned out in the course of the tests that the spacing between the valve stem and the push rod was very critical to the fuel quantity to be injected as well as to the injection timing. For this reason, a precaution was taken to determine the preset spacing in this valve design. To enable a fine adjustment of the spacing, the simple lock nut in the figure was later replaced by an adjusting screw mounted between the injector body and the actuator assembly. The sealing of the pressurized fuel at the push rod guide was made with a teflon ring S. Usually, the fuel consumption was measured from a pressure drop in the metering cylinder which was inserted in the high pressure fuel line. In the case when a better

accuracy is required, the consumption is determined by measuring the time required for a liquid column in the fuel line to travel a known distance. In the course of every test, the valve motion was monitored by a capacitance type transducer built into the injector body.

In regard to the combustion system, an indirect injection type was preferred to virtue of the expected insensitivity to diesel knock, and a conventional Ricardo Comet III type spherical swirl chamber was tentatively chosen. This type of the combustion chamber was selected not only because it was the most popular one in this class of engines, but also because it had intermediate characteristics between the pure prechamber type and the direct-injection type. As will be described later, the selected swirl chamber type turned out to assure a smooth combustion. In Fig. 2, the details of the combustion chamber are shown. The specifications of it are summarized as follows; the volume ratio of the swirl chamber to total compression space was 59%, the area of connecting passage, 2.0% of the area of the piston surface, the compression ratio, 18.6 : 1, and the piston flat-topped. Two kinds of directions of the fuel injector were included in the tests; if this direction is expressed by the angle of the injector axis from the center of the swirl chamber, α , the selected ones were $\alpha = 15^\circ$ and 0° . To aid the cold start and ignition, a conventional glow plug having a sheath diameter of 5 mm was installed initially, as shown in Fig. 2. All engine tests were made at a constant speed of 1500 rpm, with a fixed temperature of the cooling water exit of 90°C . The hydrogen pressure was regulated to 75 kgf/cm^2 in the metering cylinder.

3. State of burning at steady run

The first experiment was carried out at the injection direction of $\alpha = 15^\circ$ by using the type I injector tip. To keep a high temperature level of the charged air in the engine cylinder at the starting, the engine was well warmed up by motoring, keeping the speed between 500 rpm and 1000 rpm and the coolant temperature at 90°C . It was quite doubtful that starting would be really possible by the pressure injection of hydrogen, but ignition was safely achieved at a proper fuelling rate, though the glow plug was not used. When the correct conditions were met, the engine soon arrived at a normal running steady state. It was later noticed, however, that this smooth burning was attainable only in the case when a small amount of fuel was leaking from the injector.

Fig. 2 Cylinder head

In Fig. 3, typical pressure records obtained for different brake mean effective pressures p_e at a fixed injection timing of 2.5° before tdc are given. Just as in the case of the ordinary fuel diesel engine, the duration of injection and the maximum lift vary with the load. Under the full load condition, under which $p_e = 9.25 \text{ kgf/cm}^2$ is reached, the duration of injection amounts to 20° in crank angle, but this value will still be tolerated judging from our experience with the ordinary liquid injection. From the comparison of the pressure-time relation with the injector lift curve, it may be noted that no sooner than injection starts, the pressure begins to deviate from the compression pressure curve. It should be remembered that such a deviation never results from true ignition but is caused by an inflow of the injected hydrogen. On this account, a difficulty is encountered in determining the true time of ignition. This slow pressure change is soon followed by a rapid pressure rise due to combustion just like in the case of the ordinary fuel. Judging from the usual fuel engines, the velocity of this pressure rise and the peak pressure are quite normal over every loading from idling to the full power output. This is evident from Fig. 4 in which the measured maximum rate of pressure rise $(dp/d\theta)_{\max}$ and the maximum pressure $p_{\max}$ are plotted against the brake mean effective pressure p_e . The value of $(dp/d\theta)_{\max}$ ranges from the idling value of 1.5 kgf/cm^2 per degree of crank angle to the full load value of 5 kgf/cm^2 per degree. The peak pressure varies largely with p_e , but it is well below 70 kgf/cm^2 over the entire range of loadings. These values may be recognized as being most typical of the existing liquid-injection engine of this class.

Fig. 3 Pressure-time records at various loads [$\theta_i = -2.5^\circ$ atdc (or 2.5° before tdc)]

In Fig. 5, the results at various injection timings θ_i are presented. Just as in the usual diesel engine, the retardation of injection decreases the peak cylinder pressure. It is also noted from the figure that the maximum rate of pressure rise is little affected by the injection timing, except for a very late injection at which a higher rate is reached. Under the actual operating condition, there are two factors that determine the maximum advancement of the injection timing. One factor is the limitation due to the increase in the pressure level of the combustion chamber. Since the flow rate of the fuel is controlled by the pressure difference between the injector and the combustion chamber, a very early injection with which a high peak pressure is associated reduces the total amount of fuel delivered, thus limiting finally the maximum power output. The other factor which hinders the advancement of injection is the occurrence of a very rough running which is responsible for an intermittent ignition taking place either in the pumping stroke or early in the compression stroke. A plausible explanation of this phenomenon is that an excessive discharge of the fuel gas from the injector into the combustion chamber during the compression stroke hinders the inflow of fresh air therein, thereby producing not only a fuel overrich state but also a lower temperature. As a result the mixture fails to ignite in the normal phase during the firing stroke, thus inviting a misfiring. This mixture stays in the cylinder until the next compression stroke when it autoignites finally.

Fig. 4 Effect of load on the maximum rate of pressure rise and the maximum pressure [$\theta_i = -2.5^\circ$ atdc]

Fig. 5 Effect of injection timing on the maximum rate of pressure rise and the maximum pressure [$p_e = 6.5 \text{ kgf/cm}^2$]

In the range of operations well remote from those marginal conditions, the engine could operate quite smoothly without trouble. In this context, it is of interest here to look into the engine performance when the hydrogen fuel is injected. Figure 6 gives the test results pertaining to this, obtained for two different injection timings under the same conditions as mentioned before. In this figure, the brake specific energy consumption q_e , fuel-air equivalence ratio ϕ , exhaust gas temperature T_E , and concentration of nitric oxide in the exhaust NO are indicated against the brake mean effective pressure p_e . The specific energy consumption is also indicated as an equivalent value for the ordinary gas oil, b_{eq} . In obtaining both q_e and b_{eq} , the enthalpy influx at injection was ignored for convenience. The NO concentration was measured by using a Horiba AIA-2 type non-dispersive infrared analyzer. It is seen from the figure that in the case of the injection advance of 2.5° before tdc, the maximum attainable value of p_e is as high as 9.25 kgf/cm^2 . The corresponding value for a diesel fuel was about 8 kgf/cm^2 in this single-cylinder test engine. In other words, hydrogen is much superior to the ordinary liquid fuel in regard to the maximum power output. Needless to say, there is another advantage of no limitation due to the exhaust discoloration. It is also observed from the figure that the minimum value of the specific fuel consumption is $2 \times 10^3 \text{ kcal/PSH}$ at 6.5 kgf/cm^2 of p_e . This value is equivalent to 190 grams of an ordinary fuel per metric horsepower hour, the value of which is quite comparable with that when an ordinary fuel is used with the same test engine. From these results, it may safely be said that a hydrogen fuel is never inferior to the ordinary ones regarding the engine performance. It is the authors' belief that there is still room for further improvements in the thermal efficiency as well as in the specific power, since the present combustion system used has not fully been adapted to the hydrogen combustion. Another matter that can be suggested from Fig. 6 is that the exhaust temperature is at relatively high level, probably

due to a higher flame temperature. However, the NO in the exhaust does not differ greatly from the level expected from the normal diesel fuel.

4. Conditions for stable burning

4.1 Effect of pilot injection

In the preceding chapter, it has been demonstrated that a hydrogen-fueled diesel engine is attainable at a conventional swirl-chamber type of indirect-injection engine. However, later when the injector employed for those tests was closely examined, a small leakage of the fuel was detected after each engine run. This was thought to result from the deterioration in the tightness of the valve owing to an attack of a hot gas on the valve head and seat. The measured amount of the fuel leakage was only about 3% of the total fuel delivered under a near full load condition of $p_e = 7.5 \text{ kgf/cm}^2$, but it turned out to have a great influence upon the quietness of the engine. To avoid this leakage, the injector tip was replaced by a redesigned one, type II in Fig. 1 as has been referred to earlier, and a reexamination was made on the same test engine. It was found that the engine suffered from a very rough running at the tightened injector, accompanied with a strong knocking and sometimes misfiring. Figures 7a and 7a' reproduce some cylinder pressure records at no load with no fuel leakage, at which the engine ran by no means smoothly and misfired intermittently. Even though fired, the ignition lag was very long and scattered from cycle to cycle. The subsequent pressure rise was steep enough to result in a strong knock, as may be expected from the pressure records. In other words, in the experiments conducted with a leaking injector using a type I tip fitted, a leaked fuel adequately served to make stable the ignition of the injected main part of the fuel.

For this reason, it was then decided to make clear the effect of the introduction of a preliminary fuelling of hydrogen on the stability of the engine running. To this end, two schemes were employed as the method of preliminary fuelling in the following tests to be made, the one being the so-called pilot injection, and the other being the artificial leaking from the injector. All tests were carried out under a fixed but a slightly different condition from the previous one. Owing to large fluctuations associated with combustion, it was difficult to exert a constant brake torque on the engine, so that every test was made under no load condition, although the engine speed was kept the same as before at 1500 rpm. The measured friction was equivalent to 3.3 kgf/cm^2 in the brake mean effective pressure at this speed. In the cases when misfired cycles were involved, the total quantity of fuel varied from test to test depending on their percentage. Another modification in the test condition was made for the direction of fuel injection in the swirl chamber. From some measurements, it was shown that the central injection ($\alpha = 0^\circ$) gave a larger probability of achieving ignition than the deflected downstream injection ($\alpha = 15^\circ$). Thus the direction of $\alpha = 0^\circ$ was finally adopted.

To perform an investigation of the effect of the pilot injection upon the stability of ignition, double injection was employed in the first series of tests. Two jerk pumps were used to drive the

Fig. 6 Engine performance

hydrogen injector, the one being the main-injection pump of a fixed delivery timing, and the other being a pilot-injection pump with variable timing. The delivery pipelines from these pumps were connected to a single line which communicated to the servo-plunger within the injector assembly. Since the pilot and the main fuels were introduced into the combustion chamber by the same injector, it was difficult to determine directly the pilot quantity. For this reason, the quantity was calculated based on the recorded valve lift curve together with the measured pressure history in the combustion chamber.

In Fig. 7, the recorded cylinder pressure versus crank angle relations together with the valve lift curves are presented for various quantities of pilot fuelling at fixed injection timings, the main injection being $\theta_i = -7.5^\circ$ atdc and the pilot being $\theta_p = -35^\circ$ atdc. In each diagram, the pilot quantity is indicated in terms of both the fuel-air equivalence ratio ϕ_p and the fraction to total fuel quantity r . The former is the ratio of the fuel quantity of

Fig. 7 Pressure-time relations at pilot injection [$\theta_i = -7.5^\circ$ atdc and $\theta_p = -35^\circ$ atdc]

pilot to the theoretical fuel quantity that can be burnt by all the supplied air charge. To evaluate quantitatively the cycle-to-cycle variation in the state of burning, ignition delay for each fuelling condition was measured from a number of the obtained pressure records selected at random. In Fig. 8, the cumulative frequencies x are plotted against the ignition delay crank angle θ_d for those different pilot conditions, including the case when the pilot is not fueled. In the case of no pilot injection, the curves a , θ_d are scattered over a wide range, and ignition is only successful at 53% of the total injection strokes. In the case of pilot injection, b and c in Figs. 7 and 8, shorter and less scattered ignition lags are recorded without exhibiting any misfired cycles. Also, the pressure history is seen to be quite smooth, and in fact, the engine operated satisfactorily. At an excessive amount of the pilot fuel, however, it was experienced that the pilot part itself autoignited, again causing a rough combustion. Diagram d in Fig. 7 is the pressure record at this state of preignition.

Figure 9 gives the measured probability of attaining ignition f and the averaged value of the maximum rate of pressure rise during combustion $(dp/d\theta)_{\max}$ against the pilot equivalence ratio ϕ_p .

Fig. 8 Cumulative frequency x versus ignition delay θ_d

Fig. 9 Effect of ϕ_p on $(dp/d\theta)_{\max}$ and on f

for three different pilot timings θ_p . It may be noted from this figure that at $\theta_p = -35^\circ$ atdc, a pilot ratio greater than 3% is needed to keep all cycles from misfiring. The maximum rate of pressure rise decreases quickly with an increase in ϕ_p , but it reaches a bottom somewhere between 4.5% and 7.2% of ϕ_p . A further increase in ϕ_p causes an increase in the maximum rate of pressure rise owing to the occurrence of preignition. In the case of a later pilot injection, $\theta_p = -20^\circ$ atdc, almost no effect can be expected even at $\phi_p = 6\%$. Contrary to this, in the case of an earlier pilot injection, a relatively large effect on ignition is observed with a small pilot amount. In other words, the earlier the pilot is fueled, the less quantity of the pilot fuel is required to improve ignition. A similar trend holds for the occurrence of preignition; an earlier pilot injection is prone to this state, probably because of a higher temperature level in the combustion chamber due to an incomplete renewal of gas.

4.2 Effect of fuel leakage

A fuel leakage from the injector may be expected to have a similar effect to that of the pilot injection upon the ignition of the main fuel. To reveal this effect, different quantities of the fuel leakage were tentatively produced by adjusting both the degree of the tightness of the injector and the preset spring force acting on the valve head. The quantity of the leakage was determined by measuring the fuel lost during the motored running phase between fired runs. In Fig. 10, the averaged maximum rate of pressure rise $(dp/d\theta)_{\max}$ measured during the tests is plotted against the leakage fuel-air equivalence ratio ϕ_L , with other conditions remaining fixed. It is noted from the figure that $(dp/d\theta)_{\max}$ drops discontinuously at $\phi_L = 0.75\%$, above which the engine can operate quite smoothly. However, in the range of ϕ_L greater than 2.6%, the engine fails to run smoothly owing again to the occurrence of preignition. A satisfactory combustion is therefore gained only in the range of ϕ_L between 0.75% and 2.6%. As far as the fuel leakage was maintained within this range, there was little cycle-to-cycle variation either in the ignition delay nor in the rate of pressure rise. Such a state of almost no fluctuation was distinct from that of the

Fig. 10 Effect of ϕ_L on $(dp/d\theta)_{\max}$ in the case of central injection

pilot injection. It is also interesting to note from the comparison between ϕ_L and ϕ_p that the fuel leakage required to keep a quiet combustion is much less than that required for the pilot fuelling. This is probably consistent with the observed trend that an early pilot requires a smaller amount of fuel.

In Fig. 11, another result for the effect of the fuel leakage, obtained at a different injection direction, is given; the injector axis was deflected downstream of the swirl by $\alpha = 15^\circ$ from the center of the swirl chamber. From this figure, it may be noted that this downstream injection appreciably widens the range of stable burning. It was somewhere in this range that the engine tests described in the previous chapter were carried out. The reasoning behind the expanded range of stable burning would lie in the differences in the position of the mixture formation as well as in the degree of the contact of such a mixture with a hot gas in the chamber.

5. Discussion

Judging from the experimental results presented above, the mechanism to attain a stable burning at the hydrogen injection on the present combustion system may be explained by assuming the presence of a hot core in the middle of the swirl chamber, which may act as the promotor of ignition of the fuel injected during the normal phase. This hot core would be formed by the residual gas from the previous firing stroke, and it would locate in the center of the swirl motion at the compression end. Unless a sufficient amount of the fuel is supplied as a preliminary fuel up to the normal injection, such a hot core will decline owing to the inflow of fresh air during the compression stroke. On the contrary, in the case when a proper quantity of the preliminary fuel is introduced, this core may be preserved until the compression end by releasing a heat, thereby assisting the ignition of the main fuel. In the case when an excessive amount of the preliminary fuel is introduced, however, a noticeable preignition will take place spontaneously.

In order to prove the above supposition, flame was directly detected in the middle of the swirl chamber by measuring an ion current and by

Fig. 11 Effect of ϕ_L on $(dp/d\theta)_{\max}$ in the case of $\alpha = 15^\circ$ injection

recording it over the whole strokes during the fired run. For this purpose, a couple of long electrodes were inserted into the center of the chamber. Owing to the presence of traces of hydrocarbons, flames could easily be detected by this method without using any ionizing substance. Measurements were made for the case when the knocking mode was mixed in the normal combustion mode at some probability, or the case corresponding to $\phi_L = 0.73\%$ in Fig. 10 ($\alpha = 0^\circ$). Under this condition, signals could be observed in two separate phases; one group appeared between 50° and 125° after the top dead center during the suction stroke, and another group did in the main phase of combustion during the compression and expansion strokes. The former signals fluctuated considerably from cycle to cycle, but they well correlated with the ignition timing of the main fuelling. In Fig. 12, the integrated intensities of the ion current during the suction stroke C_i are given against the ignition crank angle of the main fuel θ_f . This figure well indicates that whenever a moderate intensity of ion current is detected, quite a normal combustion follows which is confined between -2.5° atdc and the top dead center. This result well supports the above supposition that a hot core is formed and preserved by the combustion of the leaked fuel in the swirl chamber, thereby assisting ignition of the main part of the fuel at the compression end.

In the proposed mechanism in which the injected hydrogen reaches ignition, it may be expected that the swirling motion produced in the swirl chamber will play a special role in keeping the central core, owing to the centripetal force induced by density gradient. For this reason, the presence of the central core may be considered to be specific to the swirl chamber engine. To see whether or not a similar situation may be attainable in a pure prechamber type engine, another experiment was undertaken with the same test engine using a prechamber type cylinder head. Its main specifications were as follows; the volume ratio of the prechamber was 33%, the area ratio of the connecting passage

Fig. 12 Plots of ignition time against intensity of ion current

to piston surface 0.52%, and compression ratio 18.4:1. The running conditions were the same as those for the previous swirl chamber experiments. At the starting, the engine did not fire for a long time at any fuelling conditions. This state was succeeded sometimes by intermittent explosions in the cylinder and also by violent explosions in the exhaust pipe. Consequently, it was concluded that it was much more difficult to arrive at normal burning state than in the swirl chamber engine. However, once a normal ignition was established in the prechamber, the engine went on in a smooth, knocking-free state. Under this condition, ignition might be aided by hot residual gas in the same manner as in the swirl chamber engine. Probably, the difficulty in arriving at smooth burning state would be attributed to the absence of an orderly swirling motion in the combustion chamber. This matter will give rise to an uncertainty of the location of hot residual gas, thus hindering definite contact of the fuel jet with the hot gas. From these observations, it might be stated at least that the hydrogen combustion in a prechamber engine is possible, provided an efficient means for utilizing thermal interaction between the cycles is established.

6. Conclusions

The experimental study on a hydrogen-fueled diesel engine has shown that in a conventional naturally-aspirated swirl chamber diesel engine, pure compression ignition is limited owing to high resistance of the fuel to autoignition. However, once the swirl chamber is vitiated beforehand either by a small leakage of the fuel or by a pilot fuel, a smooth combustion may be attainable. A plausible explanation for this quiet burning is that ignition is aided by the embers persisting from the previous firing stroke in the form of a hot core within the swirl chamber. By the addition of a preliminary fuel, this hot core can be maintained until the main fuel is introduced. However, an excessive introduction of the preliminary fuel causes autoignition of itself, again resulting in a rough combustion.

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