621.436.06.

An Experimental Study on the Performance of a Multifuel Engine*

(Part 5, Effect of Exhaust Throttling)

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In order to improve the multifuel capability of the diesel engine, we operated it by means of the exhaust throttling.

By this technique, it was found out that the pressure of the residual gas was increased, and this made it possible to reduce the ignition delay because of a decrease in the quantity of the intake air and the rising temperature in the working gas at the end of compression.

According to the experiment, utilizing a low cetane number fuel as automotive gasoline, the same combustibility as we could obtain with utilising gas oil was obtained, because the residual gas pressure was rose to about 3.0 ata.

But, with a throttling of the exhaust gas, fuel consumption was increased especially in a heavier load range. And yet, the smoke limited output was decreased and the engine noise became louder, because the pumping loss during the suction and exhaust stroke was increased and the charging efficiency was decreased.

However, in getting such a combustibility, these are not considerably important. Because, the difficulties in running with a low cetane number fuel occur mainly at the lighter load range.

1. Introduction

In order to improve the multifuel capability of a diesel engine, test operations were carried out by exhaust throttling.

It was found that the pressure of the residual gas increased, making it possible to reduce the ignition delay because of a decrease in quantity of the intake air and the temperature rise in the working gas at the end of the compression.

As a result of the experiment, in which a low cetane number fuel such as automotive gasoline was used, it was shown that the same combustibility as gas oil was obtained, because the residual gas pressure rose to about 3.0 ata.

But, with a throttling of the exhaust gas, fuel consumption increased markedly, especially in a heavier load range. Then, the smoke limited output decreased and the engine noise became louder, because the pumping loss during the suction and exhaust stroke was increased and the charging efficiency was decreased.

However, in getting such a combustibility, these are not considerably important. Because the difficulty in running on a low cetane number fuel mainly appears at a lighter load range.

2. Apparatus and experimental procedures

2.1 Test engine and fuels

A vertical type, watercooled, single cylinder fourcycle pre-combustion chamber type diesel engine with a compression ratio of 19.95 mm bore and 110 mm stroke was used in this trial. The nominal output was 18 PS at 3200 rpm. The engine was run at an oil temperature of 70~75°C and a coolant temperature of 100°C. Dynamometers

* Received 5th June, 1970.

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of DC and eddy current type (connected in tandem) were used to measure the motoring and brake torque of the tested engine. A handle cock of 2" bore was inserted into the exhaust pipe (90 mm apart from exhaust port flanging part) to control the exhaust pressure, and it had a water cooled construction to prevent scorching.

The types of fuels used in these trials were gas oil, undoped gasoline and automotive regular gasoline. The specifications of the tested engine and fuels are as already reported in detail⁽¹⁾.

2.2 Apparatus and experimental procedures

A strain gauge type pressure indicator (KYOWA PHF-10 V) was provided in the main chamber to measure the gas pressure, and an IC operational amplifier was made and used for trial.

The injection nozzle needle lift and crank angle were detected by a photo transistor and recorded simultaneously by a 4 trace synchroscope. A sound level meter and frequency analyzer (B & K) were used to measure and analyze the engine noise. Then, the measuring position was selected 1 m apart from the engine at the center of piston and at a 1 m height from the floor of the test bed, for the convenience of measurement and because of the relatively small influence of both sound waves from the intake and exhaust system and reflected waves from the room wall and other equipment. A long recording camera was also used to measure the combustion variations, and to observe the starting behaviour.

2.3 The procedures of calculating the progress and the rate of heat release

A calculating program proposed by the authors was used⁽³⁾ for obtaining the progress and the rate of heat release from the indicator diagram under the assumption of a single combustion chamber.

In this case, the quantity of residual gas is estimated using a simplified equation as follows;

$$G_r = \gamma_{\text{air}} V_c \left(\frac{P_r}{P_0} \right)^{\frac{1}{n}}$$

- where G_r : weight of residual gas
 γ_{air} : specific weight of incoming air
 V_c : clearance volume
 P_r : residual gas pressure at the TDC of exhaust end
 P_0 : gas pressure at the suction end
 n : index of the polytropic change

Then, by adding the measured weight of incoming air (by round nozzle) to this quantity, the amount of the working medium was determined. The accuracy of such a calculation was checked stringently by the calculation described later, and sufficient accuracy was secured.

3. Experimental results and discussions

3.1 Improvement of multifuel capability with the aid of exhaust throttling

To clarify the effects of exhaust throttling, at first, performance tests on three kinds of fuels were carried out under a constant engine speed of 2 000

Engine speed 2000 rpm BMEP 1.75 kg / cm²
 Fig. 1 The effect of exhaust throttling

rpm over a varied load range and injection timing.

Typical indicator diagrams obtained in these trials are shown in Fig. 1. In general, the ignition lag reduces in accordance with the exhaust throttling, and significant effects were obtained by the use of lower cetane number fuels. For instance, in burning automotive regular gasoline, if the exhaust is unthrottled, too late ignition and hardly fluctuated combustion result from it and the engine can not be operated smoothly. But, with use of exhaust throttling, the ignition lag is reduced appreciably, and a lowered peak of heat release and an elongated combustion are obtained. Thus, as the result, smooth running on lower cetane number fuels under a decreased rate of pressure rise is achieved.

On the other hand, if the engine is run on gas oil, ideal combustion seems to be almost completely obtained without exhaust throttling, hence there seems to be no room for combustion improvement. Furthermore, if the exhaust throttle is adopted in such a case, the shortest ignition lag results in a closer peak of heat release to TDC, thus, a higher combustion peak pressure and an increased rate of pressure rise are obtained.

Figure 2 shows the engine performance curve

2000 rpm, Injection timing -12°CA

Fig. 2 Engine performance under the varied BMEP

obtained by constant injection timing over a varied load range.

As shown in this figure, the reduction of ignition lag is achieved by the aid of exhaust throttling, especially, in the case of using gasoline. And, a significant reduction in the rate of pressure rise is obtained with gasoline. But, for the reasons described later, some defects i. e. increased fuel consumption, decreased output, the rise of exhaust temperature and the increase of engine noise occur.

Figure 3 shows the engine performance under constant BMEP ($1/4$ load, $P_e=1.75 \text{ kg/cm}^2$) and varied injection timing. When gasoline was used as fuel, the engine performance was markedly affected by the injection timing as compared with gas oil, but, the adoption of exhaust throttle gives insensible characteristics against injection timing. In this case, the minimum fuel consumption seems to be obtained at an earlier timing. The reason for this is considered to be as follows: by the adoption of exhaust throttling, a shortened ignition lag and an improved combustion are achieved, bringing about a lower peak of heat release and an elongated combustion. As a result, it seems that the earlier injection timing becomes necessary to obtain a better thermal efficiency.

Figure 4 also shows the effect of exhaust throttling on the combustion improvement. The reduc-

2000 rpm, BMEP $=1.75 \text{ kg/cm}^2$

Fig. 3 Engine performance under the varied injection timing

tion of ignition lag, the elongated combustion duration and the lowering of the peak of heat release due to the exhaust throttling are shown clearly. Further, the degree of constant volume of heat release is somewhat lowered owing to the exhaust throttling. When gas oil was used as fuel, the degree of constant volume became larger as compared with gasoline, under a lowered peak of heat release and an elongated combustion. In burning gas oil, a closer heat release curve to TDC in comparison with the case of gasoline may be produced because of the shorter ignition lag, under the same injection timing.

3.2 A consideration of the effects of exhaust throttling

By the adoption of the exhaust throttling, a rise of working gas temperature results from reduced incoming air and increased residual gas. According as a result of this, the composition of working gas is varied, and the pumping loss during suction and exhaust strokes is increased, and a part of this work is converted to heat, hence they cause a rise in the combustion chamber wall temperature. On the other hand, the operation with

throttled exhaust brings about an increased fuel consumption, and likewise it seems to constitute a cause for the temperature rise in the combustion chamber wall and residual gas.

At the same time, it is surmised that some chemical effects in addition to the physical effects described above are present. For example, on the operation with a relatively low temperature as in starting, the intermediate products of combustion such as peroxide and aldehyde produced in the previous cycle exists remain in the cylinder and seem to enhance the ignition of the following cycles.

To discuss the effect of exhaust throttling, it is very difficult to take all of these effects into consideration. But, when the discussion is limited to only one effect of mixing of the reduced incoming air and increased residual gas, the gas temperature at the beginning of the compression due to the mixing of incoming air and residual gas may be estimated as follows. i. e.,

$$T = \frac{G_r C_{vr} T_r + G_{air} C_{vair} T_{air}}{C_v (G_r + G_{air})}$$

where T : temperature of working gas at the beginning of compression

C_v : specific heat at constant volume

G : weight

suffix air : incoming air

suffix r : residual gas

Then, the incoming air quantity is measured. In general, if an engine is operated under a throttled exhaust, the gas pressure in the cylinder near the TDC of the exhaust end shows a progress as shown in Fig. 5. Thus, we can obtain the quantity of residual gas as follows:

$$G_r = G_{r1} + G_{r2} + G_{r3}$$

where

G_{r1} : quantity of residual gas flowing into intake in opposition, during A~B duration of Fig. 5

G_{r2} : quantity of residual gas exists in the cylinder at point B (the reverse flow into intake ceases at this point)

Fig. 4 Engine performance under the varied residual gas pressure

Fig. 5 The progress of cylinder pressure near the TDC of the end of exhaust stroke under the throttled exhaust

G_{r3} : quantity of gas through exhaust valve during B~C if exhaust valve is not yet closed at point B

And, each quantity is calculated as follows :

$$G_{r1} = \int \mu_s f_s \psi_1 \sqrt{2g \frac{P_2}{v_2}} dt$$

if

$$\frac{P_1}{P_2} > \left(\frac{2}{\kappa+1} \right)^{\kappa/(\kappa-1)}$$

$$\psi_1 = \sqrt{\frac{\kappa}{\kappa-1} \left[\left(\frac{P_1}{P_2} \right)^{2/m} - \left(\frac{P_1}{P_2} \right)^{(m+1)/m} \right]}$$

$$\frac{P_1}{P_2} \leq \left(\frac{2}{\kappa+1} \right)^{\kappa/(\kappa-1)}$$

$$\psi_1 = \left(\frac{2}{\kappa+1} \right)^{1/(\kappa-1)} \sqrt{\frac{\kappa}{\kappa+1}}$$

$$v_2 = RT_1 \left(\frac{P_2}{P_1} \right)^{(\kappa-1)/\kappa} \frac{1}{P_2}$$

$$G_{r2} = \frac{V_B}{v_{2B}}$$

$$G_{r3} = \int \mu_e f_e \psi_3 \sqrt{2g \frac{P_3}{v_3}} dt$$

if

$$\frac{P_2}{P_3} > \left(\frac{2}{\kappa+1} \right)^{\kappa/(\kappa-1)}$$

$$\psi_3 = \sqrt{\frac{\kappa}{\kappa-1} \left[\left(\frac{P_2}{P_3} \right)^{2/m} - \left(\frac{P_2}{P_3} \right)^{(m+1)/m} \right]}$$

$$\frac{P_2}{P_3} \leq \left(\frac{2}{\kappa+1} \right)^{\kappa/(\kappa-1)}$$

$$\psi_3 = \left(\frac{2}{\kappa-1} \right)^{1/(\kappa-1)} \sqrt{\frac{\kappa}{\kappa+1}}$$

$$v_3 = \frac{RT_3}{P_3}$$

where

μ_s, μ_e : flow coefficients of intake and exhaust valve, respectively

f_s, f_e : opening areas of intake and exhaust valve, respectively

g : acceleration of gravity

P_1, P_2, P_3 : gas pressures of intake pipe, cylinder and exhaust pipe, respectively

T_3 : exhaust temperature

v_2 : specific volume of gas in cylinder

κ : the ratio of specific heat

m : index of efflux

V_B, v_B : volume and specific volume of gas in cylinder at point B

In regard to this calculation, the values studied by Tanaka⁽⁴⁾ were written as μ_s , and μ_e . But the direction of the gas flow is opposite to the usual case, thus μ_s and μ_e given by Tanaka are used as μ_e and μ_s , respectively, in the calculation.

Actually, P_1 and P_3 are fluctuated values, but constant values were assumed for convenience sake,

hence measured values by a manometer were adopted in the calculation. Actually, P_1 is affected by P_3 , but the extent of its influence is negligible, thus the calculation was carried out with $P_1 = 0.98$ ata. Index of efflux m was assumed for $m = \kappa$, and the working gas was considered as a semi-perfect gas. The value of J. Reisacher⁽⁵⁾ was adopted for κ of the working gas. In the decision of flow coefficient, the report of Asanuma and Sawa⁽⁶⁾ was adopted. The unavailable valve lift was taken into consideration in the calculation, but its influence seemed to be negligible.

The results of the above mentioned calculation are plotted in Fig. 6. As shown in this figure, the calculated value of the temperature rise at the compression end based on the temperature rise due to the mixing of incoming air and residual gas shows a good coincidence with the compression end gas temperature calculated from the pressure progress of the indicator diagram by use of the characteristic equation of semi-perfect gas. And, the calculation of the ignition lag by an experimental formula obtained in 4th report⁽¹⁾, using the compression end temperature obtained in the above mentioned manner, shows good coincidence with measured value. Thus, the temperature rise due to the mixing of residual gas with the incoming

Fig. 6 Temperature rise of working gas at the compression end due to the exhaust throttling

air may be mainly attributed to the shortening of ignition lag caused by the exhaust throttling.

Naturally, it may be expected that some other effects of exhaust throttling may exist. For instance, Fig. 7 shows the increase of frictional mean effective pressure and the rise of pre-combustion chamber wall temperature produced by the exhaust throttling in the motoring state. As shown in this figure, a slight rise of wall temperature results due to the increase of pumping loss.

3.3 Combustion improvement under no load operation caused by the exhaust throttling

By means of exhaust throttling, no load operation of the engine becomes stable, especially in low cetane fuels. Figure 8 shows the deviation of the combustion peak pressure obtained in no load operation under a constant engine speed of 2000 rpm. In this case, if automotive regular gasoline were used, a continuous smooth operation would become impossible without exhaust throttling. On the contrary, as shown in this figure, when gas oil was used, a tendency of a somewhat larger deviation of combustion by throttling the exhaust appears. But, in burning gasoline, the combustion deviation decreases owing to the exhaust throttling. Especially, it was found that the combustion deviation under no load operation improves appreciably in regular gasoline.

Fig. 7 The increase of frictional loss and accompanied temperature rise of chamber wall due to the exhaust throttling (motoring)

3.4 Improvement of startability by the adoption of exhaust throttling

It was clarified that the technique of exhaust throttling is very effective as an auxiliary diesel engine starting aid. The improvement of startability caused by the exhaust throttling is shown in Fig. 9. In this figure, on the ordinate the number of cycles from the start of motoring is adopted, and the abscissa shows the ratio of opening area of exhaust throttling valve f to the sectional area of piston F which shows the extent of exhaust throttling, and $f/F=19.5\%$ corresponds to a case of unthrottled exhaust. In this figure, the mark ■ shows a fired cycle, and ■■■ shows continuous firing (for example, the mark ■ appearing on the scale 20 of the abscissa shows the firing of 20th cycle from the start of engine motoring). The starting test was carried out without glow plug and the engine was driven by a starting motor. In principle, the coolant temperature was kept at 10°C, but in burning regular gasoline, a start at 10°C of cooling water was very difficult, hence 30°C of cooling water was applied with special consideration. From this figure, it is obvious that, by throttling the exhaust to some extent, the time from the start of motoring to the initial explosion becomes shorter regardless of the type of fuel. But, if the

Fig. 8 The improvement of combustion deviation under the no load operation

exhaust is kept in a throttled condition, the explosion is repeated at intervals of two cycles, hence it is necessary to open the throttle valve gradually. And, as a result, continuous explosion was readily obtained. When regular gasoline was used in the test engine, no explosion occurred without exhaust throttling even at 30°C of cooling water. But, by means of exhaust throttling, the desired startability was obtained without a glow plug. As mentioned above, the exhaust throttling seems to be a quite effective means as a diesel engine starting aid.

3.5 Performance deterioration caused by exhaust throttling

So far we have discussed about the improvement of multifuel capability of diesel engine by means of exhaust throttling, but needless to say, this technique is accompanied by drawbacks of lowered output and increased fuel consumption. Moreover, it produces an increased smoke under heavy loads and also noisy operations over a wide

Fig. 9 The improvement of startability by means of exhaust throttling

load range. But, the burning of low cetane fuel in diesel engines produces no trouble under heavy loads and the assurance of better combustibility becomes important in a light load range. Thus, in practice, it is necessary to control the extent of throttling in accordance with the load and engine speed; and the performance drop due to the exhaust throttling under heavier loads is not so important.

(a) Power reduction

Figure 10 shows the results of the test under exhaust throttling which leads to power reduction. It is obvious that an increase of frictional loss, which is accompanied by an increase in negative

- a : Net output
- b : Frictional loss without exhaust throttling
- c : Increase of frictional loss due to the increase of residual gas pressure
- d : Decrease of output due to the decrease of charging efficiency
- e : The others (due to η_{gth} and η_w etc.)

Fig. 10 Performance drop due to exhaust throttling 2000 rpm Undoped gasoline

Fig. 11 The increase of smoke density due to exhaust throttling

work during the suction and exhaust strokes, and a decrease in the charging efficiency may be mainly attributed to the power drop.

(b) The increase of fuel consumption

When exhaust gas is throttled, the specific fuel consumption increases especially over a heavier load range. The reasons may be as follows, as reported already in our 4th report⁽¹⁾, i. e. This may be attributed to two main causes, one of which may be the increase in the fraction of the frictional horse power in the indicated horse power, and the other may be the reduction of the excess air factor.

(c) The increase of smoke density

The tendency of exhaust smoke increase under exhaust throttling is shown in Fig. 11. As shown in this figure, the increase of smoke density due to the exhaust throttling is caused by the reduction of the excess air factor.

(d) The increase of engine noise

As stated above, combustion is expected to become moderate by the adoption of exhaust throttling, which suggests that the combustion noise should be reduced. However, on the contrary, as shown in Fig. 3, the measured engine noise increases owing to exhaust throttling. Figure 12 shows the result of a frequency analysis of operating noise and indicator diagram. As shown in this indicator curve, combustion is clearly improved by exhaust throttling, and as a result, the cylinder pressure level (CPL) over a 300~2 000 Hz range becomes lower. And in consequence, the sound pressure level (SPL) over a 1 000~2 000 Hz drops.

But, on the contrary, the sound pressure level over a 100~600 Hz range rises. This may be caused by a reverse flow from the cylinder to the intake owing to an increased residual gas pressure.

3.6 The extent of exhaust throttling necessary to assure a desirable operation

As stated above, the exhaust throttling is accompanied by performance deterioration, especially in stationary operations of the engine, hence the extent of throttling must be kept at a minimum within limitations of smooth running. The extent

Fig. 13 The extent of throttling necessary to keep the rate of pressure rise to within 2 kg/cm²/°CA and accompanying performance deterioration

Fig. 12 The increase of engine noise due to exhaust throttling

of this varies with load and engine speed, and it is difficult to give a general rule of control, but it may be recommended to set a standard at 2 ata. of exhaust pressure for undoped gasoline, and also at 2.5 ata. for regular gasoline. Because, with this throttling, the rate of pressure rise of combustion may be kept to within $2 \text{ kg/cm}^2/^{\circ}\text{CA}$ over a wide load range, as shown in Fig. 2. Figure 13 shows the residual gas pressure necessary to keep the rate of pressure rise to within $2 \text{ kg/cm}^2/^{\circ}\text{CA}$ and the extent of the accompanied increase of fuel consumption as compared with gas oil operation. For example, in order to keep the pressure rise rate within $2 \text{ kg/cm}^2/^{\circ}\text{CA}$ under 5 kg/cm^2 BMEP, a 15 % increase of fuel consumption appeared as compared with gas oil.

Besides, as described above, in engine starting, the exhaust throttling of $f/F=0.2\sim0.3$ may be optimum for undoped gasoline and regular gasoline, but it may be said that after firing, a gradual opening of throttle valve is necessary to assure continuous smooth firing.

4. Conclusions

The results derived in this study may be summarised as follows;

(1) By means of exhaust throttling, the ignition lag of low cetane number fuel can be reduced appreciably, due to the temperature rise of working gas at the compression end caused by the increase in the amount of residual gas.

(2) The combustion deviation under no load

operation was reduced by the adoption of exhaust throttling.

(3) This technique is effective as an aid for starting. The exhaust throttling of $f/F=0.2\sim0.3$ improves the startability of the test engine markedly, and the engine starting on regular gasoline with 30°C of cooling water and without a glow plug was assured.

(4) Exhaust throttling seems to be an effective procedure to improve multifuel capability. But, it causes a deterioration of performances, hence, in practice it must be used together with other auxiliary means, especially for smooth of lighter load operations.

The minimum extent of necessary exhaust throttling differs with the conditions of load and engine speed, hence a general applicable rule could not be given, but we have set up a standard of exhaust pressure of 2 ata. for undoped gasoline, and 2.5 ata. for regular gasoline approximately, and with this throttling, the rate of pressure rise was kept to within $2 \text{ kg/cm}^2/^{\circ}\text{CA}$ over a wide load range.

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