An Evaluation of Effect of Intake Fuel Addition on 
Performances with Various Prechambers 
on a Diesel Engine*

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The present investigation evaluates the effect of the auxiliary fuel on the performances of a
diesel engine by changing the shape of prechamber, and reveals the following points:

(1) The maximum output of the engine with intake fuel addition rises almost to the same
level independently of the prechamber types.

(2) This increase of the output is caused by the improvement of overall thermal efficiency
owing to rapid combustion in the main chamber, though the aspirated mixture gives no effect on
the utilization of the oxygen contained in the cylinder.

(3) The exhaust smoke is beneficially affected by the manifold introduction of auxiliary fuel.
However, various unburned hydrocarbons can be detected in abundance.

1. Introduction

It has been reported that introducing or injecting a portion (10 to 30%) of the fuel charge of a
diesel engine into the cylinder prior to the injection of main fuel is effective on the engine output, the
fuel economy, the exhaust smoke, the combustion noise\(^{11}\), the acceleration\(^{20}\), etc.

Though many researchers in this field have investigated the method of the introduction of auxiliary
fuel and its effects on the engine performances in detail, it seems that degrees of above improvements
by the introduction of auxiliary fuel depend on the test engines. Moreover, there has scarcely
been any debate on the reacting behaviour of lean pre-mixture that may have effects on the perfor-
mancess, or on the combustion mechanism when a lean pre-mixture exists in the cylinder.

An experimental investigation in the present paper is the first step for the coming investigations
to make the above problems clear. The experimental tests were carried out to evaluate the effects of
auxiliary fuel addition to the intake air, namely the effects of "intake spray", on the performances with
various prechambers on a diesel engine. A study on chemical components and compositions of the exhaust
gas from the engine with the intake spray was

carried out with the gas chromatographic technique.

2. Experimental apparatus and method

2-1 Test engine

A four-stroke cycle, four-cylinder, water cooled
diesel engine was used in the present investigation.
Figure 1 shows the arrangement for intake fuel
addition and the positions for pressure pick-up units
and gas sampling valve which operates at every
exhaust stroke. A throttle nozzle of spray-angle 0°
was used for the injections of main and intake fuels.
The optimum injection timing of main fuel may vary with each shape of prechamber, but the timing
was set at a constant angle of 18° BTDC. The

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Fig. 1 Schematic arrangement of experimental apparatus
Evaluation of Intake Fuel Addition to Diesel Engines

intake fuel was injected with one nozzle connected to a four-plunger fuel pump, and the injection was timed to occur at 90° ATDC of the suction stroke of each cylinder, but it has been found that this timing had no influence on the engine performances. Total stroke volume of the engine is 2270 cm³ and the maximum output is 65PS/3200 rpm. Four shapes of prechambers used in the present investigation are shown in Fig. 2, the prechamber (b) being a proper one to this engine and (a) being one with the sleeve detached from (b). In addition, both main and auxiliary fuels were diesel oil, and a series of tests in the present paper were carried out under the intake conditions of -245±15 mmHg and 28±2°C.

2-2 Apparatus and method for analyzing exhaust gas

For inorganic compounds such as O₂, N₂, CO and CO₂, the thermal conductive type gas chromatograph was used, and for organic compounds of C₅H₈, C₆H₁₂, etc. the flame ionization type one was used.

To analyze unburned hydrocarbons in the exhaust gas of 2 cm³ the highest sensitivity of flame ionization detector was used, and to keep the stability of base line on a chromatogram, Thermol-2 which had higher permissible temperature and was leaner in liquid phase (250°C, 5%) was used as a column packing. Column bath temperature was kept at 100°C. Names of peaks in the chromatogram have been determined by the internal marker method and indicated by the representative hydrocarbon.

3. Experimental results

3-1 Effect of main and intake fuel deliveries on brake mean effective pressure (bmep)

Before evaluating the effects of intake spray on different prechamber engines, a preliminary test was carried out on a "prechamber with sleeve" engine [Fig. 2 (b)].

Figure 3 shows the bmep curves for an engine operated with and without intake spray, and six settings of main fuel delivery. The quantity of intake fuel then can be determined by subtracting a setting of main fuel delivery from the total fuel one.

Referring to Fig. 3, it is obvious that the intake fuel addition has a wrong effect on the output when the setting of main fuel delivery is less than 25.8 mg/st (or mg/cycle). When the setting delivery is more than 25.8 mg/st, the intake spray is effective, that is, the bmep loop with intake spray rises in its positions above that without intake spray. If the intake fuel delivery for a setting delivery of 25.8 mg/st is over 10 mg/st, the bmep goes down to the

![Graph showing the effect of auxiliary and main fuel delivery on output](image)

Fig. 3 Effect of auxiliary and main fuel delivery on output ("prechamber with sleeve" engine)
same level as that without intake spray or further down. This may be because of the auto-ignition of aspired fuel and of the increase of unburned hydrocarbons in the exhaust gas. On careful comparison of the loops at the same quantity of total fuel, it can be found that, the less the main fuel or the more the intake spray, the higher the bmep is.

The intake fuel addition in the light load range has no apparent effect on the output, so that in the following performance tests on four prechambers the discussions would be limited to the effects in the heavy load range. Thus, the main fuel delivery setting was about 30.5 or about 33.5 mg/st.

3.2 An evaluation of effect of intake spray on performances with various prechambers in an engine

3.2.1 Comparison of the effects on bmep and exhaust smoke

Figures 4 (a) and (b) show the effects of intake spray on the bmep and the exhaust smoke in regard to each prechamber. Figure 4 (c) shows volumetric percentages of the residual oxygen in the exhaust gas from four types of engines which operate without intake spray. About 40 mg/st of the abscissa is equivalent to 1.1 of the excess air ratio. The intake fuel delivery was increased as far as the combustion noise was permissible, which might be due to the auto-ignition of aspired fuel at the latter end of compression stroke.

Referring to Figs. 4 (a) and (c), the intake fuel addition has little effect on the swirl-chamber and the “prechamber with sleeve”-engine which may effectively utilize the air in the cylinder. Then, the maximum increasing ratios* of these engine outputs are only 3.7% and 4.9%, respectively. As for the pre-combustion- and “prechamber without sleeve”-engine which may be poor in the utilization of the air and is smoky, the intake spray can be very effective. These ratios then are as large as 10.2% and 12.7%, respectively.

* The maximum increasing ratio is given by

\[ \frac{p_{\text{fuel}}}{p_{\text{atm}}} = \frac{p_{\text{bmep}}}{p_{\text{bmep}}^{\text{0}}} \]

where for a given fuel delivery, \( p_{\text{bmep}} \) is the bmep of the engine with intake spray and \( p_{\text{bmep}}^{\text{0}} \) is that without intake spray.
Comparing two dot-dash lines of an engine in equal total fuel delivery, it is found that the small setting of main fuel delivery is more favourable than the large one as for the output. However, the maximum quantity of intake fuel, though it should be varied according to the residual gas temperature, the wall temperature or the droplet size of fuel, is limited to 10 mg/st or less in the present engine. The main fuel charge, therefore, should be a large quantity in order to obtain the maximum bmep. Thus, the maximum bmep attained by each type of the engine rises to about 7.7 kg/cm², almost independently of the shape being swirl, pre-combustion or ill-shaped chamber.

The intake spray also has a similar effect on a poor performance engine, for example, when the main fuel is timed to inject unsuitably late or the main injection nozzle is worn out. Figure 5 illustrates this example on the "prechamber with sleeve" engine. When this engine operates with a pintle nozzle, whose spray-angle is 4°, the bmep generally drops by 0.4 kg/cm² as compared with the operation with a throttle nozzle, and the combustion is very noisy. Retarding the main injection timing by 6 degrees can reduce the combustion noise to silence, but the bmep further drops by 0.4 kg/cm². In spite of these facts, at 40 mg/st with intake spray, the difference of the bmep among three cases is only 0.3 kg/cm², and their combustions are not very noisy.

From the above facts, it can be said that the more defective the combustion in the cylinder is, the larger the increase of bmep with intake spray is. Consequently, the maximum bmep of any engine may be almost equal to that of a perfect engine with intake spray.

It seems that the effect of intake fuel addition on the exhaust smoke is similar to that on the output, that is, the smoke from the pre-combustion chamber engine is most remarkably improved just like the output, whereas the smoke from the swirl-chamber- or "prechamber with sleeve"-engine is little improved. Little effect on the "prechamber without sleeve" engine is caused by the fact that a portion of main fuel charge passes through to the main chamber owing to the large throat area.

3.2.2 Influence of intake spray on compounds in exhaust gas

Figure 6 shows an analytical result for the inorganic compounds in the exhaust gas from the pre-combustion chamber engine.

Referring to O₂ curves, two dot-dash lines are located above a solid line with the exception of the cases that a large quantity of intake fuel is aspirated, and that the intake fuel is aspirated under the conditions of lower excess air ratio (1.1 to 1.2). Then, it may be scarcely said that the intake fuel addition improves the utilization of the oxygen contained in the cylinder.

As for CO in the exhaust gas, it can be remarkably improved. Comparing two dot-dash lines of O₂ and CO₂ in the same total fuel delivery, it is obvious that the exhaust gas with intake spray is rich in O₂ and lean in CO₂. This is attributable to the unburned exhaust of a small portion of fuel. Therefore, the influences of prechamber shapes and of cooling water temperatures on the unburned hydrocarbons in the exhaust gas were investigated. A numerical value of 1.04 for n-octane in Table 1, for example, shows that the volume of n-octane contained in the exhaust gas per one stroke is equivalent to the volume of n-octane contained in the fuel of 1.04 mg. Meanwhile, the mixture consisting of methane, ethane, propane, etc. should be represented by ppm, for these compounds are not contained in this fuel.

Comparison of two cases with and without intake
spray in the "prechamber without sleeve" engine shows that hydrocarbons above the cyclohexane's line are remarkably rich in the case with intake spray, and also hydrocarbons below this line are generally rich. As the mixture consisting of n-heptane and iso-octane in this case is 21.09, the mixture burned may be only 44% of total quantity (30.55+6.95). This is perhaps because this chromatogram peaks and that of an olefin overlap each other owing to the reduction of retention time in the gas chromatography. The olefin is rearranged or decomposed from higher hydrocarbons.

Comparing the influences of intake fuel addition on unburned hydrocarbons among four shapes of prechamber, it is found that lower hydrocarbons abound in an engine having poor performance without intake spray, but among higher hydrocarbons than 4-methylpentane there is little if any difference. On the other hand, compounds above n-octane line in Table 1 have a tendency to decrease with a rise in the cooling water temperature.

Principal causes to increase unburned hydrocarbons with intake spray may be as follows:

(1) If the flame spurring from the main fuel can not completely traverse the whole width of main chamber, the intake fuel which is scattered where the flame does not approach will be exhausted in the state of decomposition or cracking.

(2) The intake fuel attached to the cylinder wall or scattered near the wall may be late in vaporization, thus the fuel cannot burn completely when the flame approaches.

![Graph showing fuel delivery and heat release](image)

**Fig. 7 Effects of auxiliary fuel delivery on heat release and gas flow ("prechamber with sleeve" engine)**

<table>
<thead>
<tr>
<th>2000 rpm</th>
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</table>

<table>
<thead>
<tr>
<th>Items</th>
<th>Influnces of prechamber shapes</th>
<th>Influnces of cooling water temperatures*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Prechamber without sleeve</td>
<td>Prechamber with sleeve</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Main fuel delivery mg/st</td>
<td>37.94</td>
<td>30.55</td>
</tr>
<tr>
<td>Intake fuel delivery mg/st</td>
<td>0</td>
<td>6.95</td>
</tr>
<tr>
<td>Methane, ethane, propane, etc. ppm**</td>
<td>39</td>
<td>144</td>
</tr>
<tr>
<td>n-Heptane and iso-octane etc.***</td>
<td>3.50</td>
<td>21.09</td>
</tr>
<tr>
<td>Cyclohexane</td>
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</tr>
<tr>
<td>4-Methylheptane</td>
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<td>0</td>
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<tr>
<td>n-Octane</td>
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<td>1.06</td>
</tr>
<tr>
<td>2, 3-Dimethylpentane</td>
<td>0.24</td>
<td>0.51</td>
</tr>
<tr>
<td>n-Nonane</td>
<td>0.13</td>
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<tr>
<td>n-Decane</td>
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</table>

* Prechamber with sleeve is used.
** It is calculated in comparison with a household L.P.G.
*** A value (x) of hydrocarbon below this line is given by

\[ x = \frac{\text{Volume of a compound contained in exhaust gas per one stroke}}{\text{Volume of a compound contained in 1 mg of fuel}} \]

(Hydrocarbons are arranged on the basis of retention times.)

**** Not measured.
3.3 Analysis of indicator diagram

If the intake fuel is added to an engine with "prechamber with sleeve" or pre-combustion chamber in heavy load range, the engine output increases above the increasing ratio of 10%. However, an example in Fig. 6 shows that the utilization of oxygen in the cylinder tends to become rather poor.

As a method to make clear why the output increased, the indicator diagrams obtained with the strain gauge type pick-up units were analyzed according to the "Jo's method". An indicator diagram is generally affected by many factors, and the compression ratios are reduced by replacing glow plug with pick-up unit as shown in Fig. 2. Then, the calculations are based on the following simplified assumptions, and the results may be qualitatively rather than quantitatively discussed.

(i) At a point of 18° BTDC the charge temperature in the prechamber is equal to that in the main chamber.

(ii) The fuel injected into the prechamber burns without ignition delay.

(iii) After finishing the main fuel injection,
the mixture flows from the prechamber to the main chamber with an excess air ratio \( \lambda \) of 1, and when it flows conversely \( \lambda \) is equal to 2.

(iv) All the fluctuations on the indicator diagram are caused by combustion.

Referring to Fig. 7, the gas flow to the main chamber in the case of 4.2 mg/st of intake spray occurs near TDC just as in the case without intake spray. In the case of 7.4 mg/st the flow occurs about 3 degrees earlier than in above cases, and at 10 mg/st a combustion, far too early for itself, which may be auto-ignition of the aspirated fuel, occurs at 5\(^o\) BTDC in the main chamber. Such an earlier combustion is interrupted for a while, and normal combustion starts at 5\(^o\) ATDC. Moreover, it can be found from Fig. 7 that the total heat discharged in the prechamber gradually increases with an increasing intake fuel delivery. Auto-ignition of the aspirated fuel brings about the reduction of output and the increase of combustion noise. And the maximum auxiliary fuel-air ratio, which is limited by the occurrence of light auto-ignition, is about 0.017 in the present engine, though it may be slightly affected by the setting of main fuel delivery.

Figures (a) to (d) show indicator diagrams and their results analyzed for each engine with and without intake spray of about 7.3 mg/st, but as for (b) the indicator diagram only is shown.

The compression ratio, hence the temperature at the end of compression stroke, of the engine with the "prechamber without sleeve" is the lowest of all, and its throat area is too large to cause a turbulence in the prechamber. The ignition lag of main fuel, therefore, may be longer in the prechamber. Moreover, the jet-speed to the main chamber may be insufficient to spread out rapidly over the main chamber owing to the larger throat area. These facts are causes of the lowest output of all the engine types without intake spray. However, when adding the intake fuel the ignition lag in the prechamber is shortened by about 5 degrees, the gas flow to the main chamber begins about 3 degrees earlier, and the lag in the main chamber also is shortened by about 2 degrees. Finally, the combustion in the main chamber begins about 5 degrees earlier in appearance, which may result in a more than 10% increase of output.

As to other prechamber engines, it can be similarly said that the flow to the main chamber is hastened by 5 degrees to 6 degrees owing to the shortening of ignition lag in the prechamber, and also in the main chamber the unburned gases from the prechamber burn a little earlier, thus the heat release near TDC may increase, and the improvement of thermal efficiency of the engine may result in an increase of the output.

4. Conclusions

Using a simple equipment shown in Fig. 1 for the intake fuel addition, the effects of the addition of auxiliary fuel on the performances of a high-speed diesel engine were investigated by changing the shapes of prechamber. The results obtained may be summarized as follows:

(1) Although the auxiliary fuel is introduced into the intake air, it has unfavourable effect on the engine output in the range of small quantities of main fuel, and the effect cannot be expected until this quantity is raised over a certain value. This is because a portion of intake fuel escapes with the exhaust gas and leaks out to the crankcase, and cannot take part in the combustion in the cylinder.

(2) Through evaluation of the effects of intake spray on the output by changing only the shapes of prechamber, it can be said that the lower the output of engine without intake spray is, the larger the effect of intake spray is. Therefore, the maximum bmep obtained with the spray is, independently of the shape of prechamber, about the same value.

(3) The exhaust smoke is improved to some extent with intake spray, thus it is possible to operate the engine at lower excess air ratio. However, the exhaust gas considerably abounds in unburned hydrocarbons. This is perhaps because a portion of intake fuel, which is delayed in vaporizing near the cylinder wall and is scattered in the space where it is difficult for the main flame to approach, is exhausted in the state of only decomposition or cracking.

(4) On analyzing the indicator diagram and the exhaust gas, it can be concluded that the increase of engine output is caused by the improvement of overall thermal efficiency, which is brought about by the increase of fuel charge burned near TDC owing to the slight shortening of the ignition lag of unburned gas in the main chamber. Therefore, it may be scarcely said that the increase of the output is caused by the improvement of the utilization of the oxygen in the cylinder.

References

(2) B.S. Murthy: Jour. SAE, Vol. 72, No. 4 (1964), p. 68.