A Study on Combustion in Direct-Injection Diesel Engines
(2nd Report, In the Case of Deep-Bowl Chamber)

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The mixture formation and combustion in deep-bowl type direct-injection engines have been investigated by means of high-speed photography. This type of combustion chamber is often adopted in compact high-speed engines and may be characterized by squish motions created by piston movement. At first, the effects of the squish on combustion have been evaluated in several chambers. It has been revealed that a flame rapidly penetrates into the clearance space soon after ignition, assisted by the reversed squish motion, thereby the flame lasting for a longer period than in the combustion chamber. This implies that a too early outflow of fuel into the clearance space is to be prevented in order to ensure a favorable combustion. Indeed the swirl well removes such an excessive outflow, but at the swirl of a too high intensity the mixture formation in the periphery of the combustion chamber is hampered, thus inviting poorer air utilization in the clearance space.

1. Introduction

This paper is a sequel to the high-speed photographic study on the direct-injection diesel engines. The first report dealt with the mixture formation and combustion in a shallow-dish chamber which consisted of the space between the dished top on the piston and the underside of the cylinder. Compact high-speed engine, however, is often served by a deep-bowl chamber or a tridoidal chamber having a smaller diameter and a deeper cavity than the shallow-dish chamber, with fewer sprays. Although it does not seem that there is a clear definition between them, there arise some essential changes as the combustion bowl becomes deeper. The first difference would be a shorter spray path that would result in an increased amount of fuel deposited on the wall. Others are the changes in the gas motions created in the combustion chamber prior to combustion. The so-called squish induced by the piston motion will become essential in a deep chamber. Swirl, if employed, would be intensified as it is reduced in a small bowl. These would probably cause some essential differences in the course of mixing and combustion, but the systematic investigation is still lacking. In order to arrive at a better understanding, a high-speed photographic study has been carried out in almost the same manner as in the first report.

2. Experimental apparatus and procedure

The piston crown of a loop-scavenged twocycle engine (cylinder diameter 110 mm and stroke 120 mm), the same one used in the first report, was rebuilt to allow the deep-bowl type chambers to be examined. The combustion chambers employed in the study are shown in Fig. 1 and some variables are given in Table 1 together with those of the shallow-dish type used in the previous report. The brief explanation is as follows: The type I chamber has a diametral ratio of the bowl to the bore of 60% with a promontory in the bottom of the chamber, type II is flat-bottomed with same diametral ratio, and type III has a smaller diametral ratio of 50% with a promontory. They are demountable on the piston as shown in Fig. 2.

In the present report, all the tests were performed at a hole-type nozzle of five 0.22 mm-
diameter holes and 250 kg/cm² opening pressure. A diesel fuel of cetane number 70 and specific gravity 0.826 was injected in an amount of 37.2 milligram per firing cycle (when measured at 1000 rpm), at a fixed injection advance of 20° before top dead center. The double scavenging was also adopted.

The high-speed camera and additional measuring devices were the same as those adopted in the first report, except for an alternate high-pressure mercury lamp (500 W) for intensified illumination.

Table 1 Specifications of the combustion chambers

<table>
<thead>
<tr>
<th>Combustion chamber</th>
<th>Diametral ratio of bowl to piston</th>
<th>Volume ratio</th>
<th>Bowl dia./depth</th>
<th>Compression ratio (Effective ratio)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>60%</td>
<td>89.4%</td>
<td>3.66</td>
<td>21.7 (14.0)</td>
</tr>
<tr>
<td>II</td>
<td>60</td>
<td>89.4</td>
<td>3.88</td>
<td>21.7 (14.0)</td>
</tr>
<tr>
<td>III</td>
<td>50</td>
<td>89.5</td>
<td>2.33</td>
<td>21.0 (13.0)</td>
</tr>
<tr>
<td>Shallow-dish type</td>
<td>82</td>
<td>8.18</td>
<td>19.6 (12.6)</td>
<td></td>
</tr>
</tbody>
</table>

3. Mixture formation and combustion under no swirl condition

3.1 General observation

The spray and the flame developments under no swirl condition in three types of combustion chambers were observed. In Fig. 3 some of them are reproduced and in Table 2 the test conditions are summarized.

Photo 14 presents combustion sequence in the case of 140° spray angle with type I chamber. It can be observed that the spray tip reaches the side wall of the bowl before ignition takes place. This would indicate that a considerable amount of fuel distributes directly on the wall or in its vicinity. After the spray hits the side wall, a flame spreads spherically with a considerable rapidity and violently rushes into the wider space between piston and head, the speed of spreading amounting to as high as 20 m/sec. In this period a flame front is formed which is rimmed by a characteristic luminous zone, beyond which a dark region exists. The flame also spreads tangentially along the side wall of the bowl and finally coalesces with those developed from the adjacent sprays at between 5° and 6° after top dead center. At the same time, luminous irregular lumps of flame begin to outflow from the bowl into the clearance space and gradually turn to show random appearances. In the central part of the bowl, there exist flameless spaces never swept by flame until the end of combustion. Finally, it may be interesting to note that in the last stage of burning the extinction of the flame occurs much earlier in the bowl than in the clearance.

As can be clearly observed, a powerful outflow of the flame into the clearance space in the earlier and the middle stages of combustion may be one of the features characteristic to the deep-

Fig. 1 Cross-section of combustion chamber and the direction of spray

Fig. 2 Cylinder head and combustion chamber

Table 2 Data on photographic runs

<table>
<thead>
<tr>
<th>Photo</th>
<th>Chamber</th>
<th>Spray angle</th>
<th>Swirl ratio n_s/n</th>
<th>Engine speed m</th>
<th>Camera speed pictures/sec</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>I</td>
<td>140°</td>
<td>0</td>
<td>1170</td>
<td>6300</td>
</tr>
<tr>
<td>15</td>
<td>II</td>
<td>140°</td>
<td>1</td>
<td>1120</td>
<td>6500</td>
</tr>
<tr>
<td>16</td>
<td>III</td>
<td>140°</td>
<td>2</td>
<td>1170</td>
<td>6500</td>
</tr>
<tr>
<td>17</td>
<td>III</td>
<td>150°</td>
<td>1</td>
<td>1160</td>
<td>5790</td>
</tr>
<tr>
<td>18</td>
<td>III</td>
<td>130°</td>
<td>2</td>
<td>1110</td>
<td>6000</td>
</tr>
<tr>
<td>19</td>
<td>III</td>
<td>100°</td>
<td>1</td>
<td>1090</td>
<td>5880</td>
</tr>
<tr>
<td>20</td>
<td>I</td>
<td>140°</td>
<td>1.8</td>
<td>1050</td>
<td>5460</td>
</tr>
<tr>
<td>21</td>
<td>II</td>
<td>140°</td>
<td>1.8</td>
<td>1040</td>
<td>5420</td>
</tr>
<tr>
<td>22</td>
<td>III</td>
<td>140°</td>
<td>2.7</td>
<td>1150</td>
<td>6000</td>
</tr>
<tr>
<td>23</td>
<td>III</td>
<td>140°</td>
<td>2.7</td>
<td>1230</td>
<td>5910</td>
</tr>
<tr>
<td>24</td>
<td>III</td>
<td>140°</td>
<td>2.7</td>
<td>1300</td>
<td>6250</td>
</tr>
</tbody>
</table>
bowl type of chamber. This phenomenon is called the "outward or reversed squish" by Alcock and Scott who first pointed out the role of this action from a similar high-speed photographic study. Unfortunately, the proper evaluation of them would further require a closer examination of the gas motions produced within the bowl by the normal squish.

Apart from these, it can be considered that fully developed turbulent flames observed in the middle or later stages are the result of the expansion of gas due to combustion and of the reversed squish mentioned above. In order to illustrate such a turbulent nature of the flame at

Fig. 3 Flame developments without swirl (see Table 2 for data) (Part 1)
this stage, the film was projected on a viewer repeatedly within a limited range of crank angles between $25^\circ$ and $40^\circ$ after top dead center. Figure 4 gives an example of obtained sketches of the motion of flamelets, showing that there is no longer any systematic motion but local eddies produced everywhere in the visual field.

3-2 Effect of chamber form

In the next place, different types of combustion chambers were examined at the same spray angle of $140^\circ$. In the case of type II, Photo 15, the general sequence does not so much differ from that in the case of I. Strictly speaking, however, the initial flame is somewhat closer to wedge shape than to sphere, and the flame locates slightly inside of the chamber, covering the greater part of the bowl in the middle combustion stages. Especially, the flame spreading in the clearance is more uniform than in the case of type I.

Fig. 3 Flame developments without swirl (see Table 2 for data) (Part 2)
In type III chamber which has a smaller spray path, the side wall being hit and the resulting splash can be clearly observed in Photo 16. Presumably because of the reduced air-borne fuel resulting from the shorter spray path, the initial spread of the flame and the subsequent ejection into the clearance space are far weaker than in type I or II. Another difference noticeable from the comparison with the former photos is the fact that the bowl appears nearly full of bright flame lasting much longer within the bowl than in the clearance.

In Fig. 5 are summarized the results of performance tests carried out at a constant fuel quantity for three kinds of chambers, showing the measured mean effective pressures and the Bosch smoke concentrations against the static timing of injection beginning. From this figure it is known that at 140° spray angle the type II chamber gives higher output with less exhaust smoke than the type I, and that the type III chamber records more excellent performance than the type II, especially in the reduced concentration of smoke. These tendencies do remain unchanged at spray angles 150°, 135°, and 100°.

As presented above, better performance is experienced at smaller diametral ratios of the combustion chamber. At the same diametral ratio, the chamber without a promontory gives a better one. The motion pictures in type II chamber showed a more uniform flame covering wider areas of the combustion chamber and an earlier extinction of the flame, compared with type I. In type III chamber, it was revealed that the outflow of the flame into the clearance was far fainter in the earlier stages and that the flame lasted for longer period in the bowl than in the clearance. By taking these facts into consideration, it may be presumed that if the reversed squish conveys too much fuel into the clearance space at the earlier stages the resulting deficiency of air will cause poorer burning. Indeed the reversed squish must be really useful in promoting distribution of the fuel over a wider area and in the subsequent spreading of the flame as far as it is not of an excessive degree, but too early outflow of the fuel into the clearance space must be prevented in order to ensure a satisfactory combustion. In type III chamber, the reversed squish may be strong in reality, but the deposition of the fuel on the wall due to the shorter spray path hinders rapid outflow of the fuel, preventing an excessive deficiency of air there. The same explanation would be applicable for the difference between the chambers I and II, although it is not clear how this difference has materialized. The explanation for this would probably require a more detailed inspection into the flow or eddies produced in the combustion chamber.

3-3 Direction of fuel spray

Photos 17, 18, and 19 show the flames taken in various directions of fuel spray, in the type II chamber. At 150° spray angle, Photo 17, no essential difference can be seen (from 140° direction), except for a slightly earlier spreading of the flame into the clearance.

At 130° spray angle, Photo 18, the flame in the clearance space is of luminous and turbulent nature and has no longer any definite front which was experienced at 150° or 140° spray angle. It is noted that the arrival of the flame at the cylinder liner occurs at later timing. This may be explained by the fact that the gas spilling out from the bowl into the clearance involves much fresh air in the earlier stages, as the result
of downward direction of the spray. According to the performance test in Fig. 5, the best performance can be attained in this spray direction in this combustion chamber. According to the former hypothesis that the excessive outflow of fuel is harmful, such a downward injection may prevent poorer combustion in the clearance. At 100° spray angle, Photo 19, it can be recognized that the flame rapidly outflows into the clearance in spite of so small spray angle. However, the flame lasts for a longer period and has a lumpish nature in the bowl in the later stages. Presumably because of these, the performance is inferior to those at other larger spray angles.

4. Swirl and combustion

4-1 Swirling motion during compression stroke

When the combustion bowl is deep and has a small diameter, the angular velocity of the swirling motion may increase during the compression stroke as the air in the cylinder is carried into the bowl, owing to the fact that each fluid element reduces its radius with the angular momentum unchanged. This process has been successfully formulated by Fitzgeorge and others (3), based on the assumption that the velocity profile does not change anymore during the compression stroke. Strictly speaking, this is not a practical assumption in the actual case, so that the present authors did not adopt this but began an analysis with a different concept that the angular velocity of each fluid element was conserved throughout the compression stroke without any exchange of momentum occurring between fluid layers. Additional assumptions for this analysis were the spatially uniform density of air at a specified instant, the absence of leakage of air through piston rings, and no heat exchange between the fluid and the ironware. For convenience, it was further assumed that the combustion chamber had a cylindrical space having the same diameter and volume. Once the profile of velocity at the beginning of compression was given, the angular velocity at a radial position at each crank angle might be calculated from the rules of conserving angular momentum and total gas weight. The details of the formulation are given in Appendix.

In Fig. 6 the calculated ratio of angular velocities before and after compression, that is, final velocity \( \omega \) divided by that at the same position before compression \( \omega_0 \), is shown against the non-dimensional radius \( r/r_0 \), where \( r_0 \) denotes the radius of cylinder. It is noted that this ratio is constant within a radius smaller than that of the rim of the bowl and slightly increases with the radius between this radius and the rim at which the ratio reaches its maximum. In the clearance space \( \omega/\omega_0 \) decreases monotonously with the radius finally to reach unity at the cylinder liner. Thus it may be expected that if a forced vortex is given prior to compression the final swirl at the compression end is almost close to a forced vortex in the combustion bowl. In the clearance space, the estimated profile is far from the forced vortex and will be such that the tangential velocity may decrease with radius like a free vortex. In addition, velocity ratio \( \omega/\omega_0 \) versus radius relation for a shallow-dish type chamber adopted in the first report is given in the same figure, showing that the velocity ratio is far less than those of deeper toroidal chambers.

In order to verify the prediction of the increase in the swirl velocity during compression, a comparison was made between the predicted and the measured histories. Since the test engine was loop-scavenged, the swirl could be easily produced by closing some of the scavenging ports, as has been described earlier in the previous
paper. In the shallow-dish chamber formerly employed, the cylinder swirl was approximately a forced vortex, and the swirl ratio (rpm of the swirl divided by engine rpm) at the end of compression was 3.8 and 2.5 at delivery-air ratios 1.9 and 1.1 respectively, at an engine speed 1000 rpm (see Fig. 8 in the first report). According to Fig. 6, the velocity ratio of the cylinder swirl to the final swirl is 1.4 in the shallow-dish chamber, so that the ratio of swirl speed prior to compression to the engine speed, \( n_s/n_e \), must be 2.7 and 1.8 at the respective delivery ratios. Based on these numerical values, the swirl speed during the cycle was calculated by the above-stated theory. Figure 7 gives the swirl speed versus crank angle relations estimated at the periphery of the bowl under several conditions, together with the ones measured by means of the ignited particle method\(^{(1)}\). The correlation between the predicted and the measured course is

**Fig. 8** Flame developments in the presence of swirl (see Table 2 for data) (Part 1)
not so good, but it may be safely said that the essential nature of the swirl can be fairly well described by the present theory. The poor correlation would not be attributed to the theory but rather to the poor accuracy in the velocity measurement. As the angular velocity of the swirl varies from time to time in the deep-bowl chambers, we will conveniently use \( \nu_c/\nu \) for indicating the swirl intensity in the following.

**4.2 Flame development at a swirl**

Photos 21 and 22 in Fig. 8 show the pictures obtained with the type II chamber at swirl ratios \( \nu_c/\nu = 1.8 \) and 2.7 respectively. It is noticed from the former photo that the spray is considerably bent at the tip by the crosswind of the swirl, thereby the spray being kept from directly hitting the side wall of the bowl. After ignition, a long, narrow flame spreads first in the downstream of the swirl, soon followed by a rapid development to cover the entire space. The spilling out of the flame into the clearance space occurs radially at first, which soon begins to rotate, and it is considerably retarded compared to the case without swirl. Photo 15. The flame forms a luminous and turbulent front behind which the dark part does not exist. At the middle stages, the flame well covers the center of the chamber in which a rapid rotation like a solid body is occurring. In the clearance space the flame motion is not a solid rotation but is nearer to a free vortex, giving an appearance as if the rotating cylinder forces to rotate a very viscous liquid around it. This is not caused by internal nor by external friction but is better attributable to an increase of the turning radius of fluid element at a constant angular momentum, quite inversely to what happens during the compression period. With the progress of crank angle, this rotation slows down and finally turns into a forced vortex.

As to the final stage of combustion, the flame in the bowl lasts for the same as or longer period than that in the clearance, contrary to the case when the swirl is not employed. Another difference noticeable at this stage is in the appearance of flame. The remaining local flames tangled, under no swirl condition, are substituted by fibrous ones which cover the space more uniformly. This would be beneficial to less smoky combustion.

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**Fig. 8 Flame developments in the presence of swirl (see Table 2 for data) (Part 2)**
In the next place, a swirl intensified to be \( n_s/n = 2.7 \), in Photo 22, makes the spray path so bent that it never hits the side wall up to ignition. Considering from the shape of the flame developed soon after the ignition, it may be stated that the mixture must have been located far inside in the bowl apart from the side wall. Owing to the resulting poorer fuel-distribution in the periphery of the bowl, the spread of the flame to areas remote from the nozzle is exceedingly retarded, and the flame diminishes much earlier in the clearance. Inversely, the combustion in the bowl continues for a longer period with an appearance of some lumpish flame. Summarizing these observations, it may be elucidated that a higher rate of the swirling motion will, depending on its degree, serve to reduce the spray penetration to make the mixture situated inside of the bowl, thereby the flame being suppressed from spreading to cover the outer parts. This might result in an inferior utilization of air in the peripheral zone, similar to what has been experienced at a strong swirl in the shallow-dish chamber, as described in the previous report. In the present case of a deep-bowl chamber having a shorter spray path, a similar state can take place owing to a strong crosswind resulting from the boosted swirl.

4-3 Influence of chamber form

In the next place, the combustion sequences under different chamber forms are investigated at a constant swirl ratio \( n_s/n = 1.8 \). In type I chamber, Photo 20, it is felt that there is no serious difference from that of type II chamber, Photo 21. Similarly to the previous comparisons made between the cases with and without swirl, it is noticed that the outflow of the flame into the clearance is considerably delayed, and that the flame situates inside of the bowl and vanishes at almost the same time from the visual field without causing residual fire in the clearance space.

Next, under no swirl condition with type III chamber, Photo 16, the flame spreading was quite gradual especially at the early stages when the flame penetrated into the clearance space. In contrast with this, the swirled condition (Film 23) causes far earlier and stronger outflow into the clearance. This difference would be attributable to the increased air-borne fuel which may be easily outflowed by the reversed squish, resulting from the fact that the spray is removed from depositing directly on the side wall of the bowl.

At a stronger swirl \( n_s/n = 2.7 \) in the same chamber, Photo 24, the fuel spray is kept far from hitting the wall, and accordingly fuel distributes almost in the air. Owing to the mixture formation inner in the bowl, the first flame no longer adheres to the side wall and the subsequent spread is again suppressed, causing a delay in flaming over the entire clearance. As the result, the flame lasts longer again in the bowl than in the clearance.

4-4 Effect of swirl

In Fig. 9 the results of performance tests carried out at a constant fuel quantity are summarized. In each combustion chamber, it is known that at a swirl ratio \( n_s/n = 1.8 \) the performance is more or less improved over that when the swirl is not employed. In type II chamber there is almost no improvement in the power output, nevertheless a reduction in the exhaust discoloration is attainable. These results may lead to the well-known fact that the swirl at a proper intensity can produce a better state of combustion.

But, at a higher swirl ratio \( n_s/n = 2.7 \) this is no longer true as can be seen in the performance test, indicating the so-called overswirled state. Considered from the observations by motion pictures, this overswirling may be attributed to the reduced penetration of the spray, as has been frequently mentioned. Figure 10 gives a sketch showing the estimated mixture distribution at the instant of ignition at various swirl speeds in the type II chamber. At \( n_s/n = 0 \), the mixture distributes along the wall on both sides of the spray, while at \( n_s/n = 1.8 \) it locates only in the
downstream of the swirl. In the latter case, the clearance space can be kept from being overrich by supplying adequately the air from the bowl. This counterbalances the fuel-air distribution in the bowl with that in the clearance space. At \( n_c/n = 2.7 \), there exists a layer of air between the side wall and the mixture as is indicated at the bottom of Fig. 10. This implies that the gas containing a larger proportion of air is conveyed into the clearance at the earlier stages by the reversed squish motion, resulting in poorer air utilization in the peripheral parts. Moreover, the flaming zone can be exerted by thermal pinch effect, as reported in the previous paper, which prevents the inner hot zone from mixing with the outer denser zone. These facts seem to be well supported by the succeeding progress of the flame, that is, the delayed flame spreading towards outer parts, and the earlier fading out of flame in the clearance space.

5. Conclusions

From the high-speed photographic study in some deep-bowl direct-injection chambers, the following were revealed.

(1) Even if there exists no swirl, strong gas motions are induced during the burning period by a rapid expansion due to the flame and the reversed squish action. If too much fuel is transferred into the clearance space by these motions, the air becomes deficient there, causing poorer burning. Better combustion may be obtained if such an excessive outflow is adequately prevented.

(2) When a swirl is employed, the mixture is formed in inner side of the bowl as the result of reduced spray penetration. At an adequate intensity, the excessive outflow of the fuel may be prevented, affording the counterbalance in the air-fuel distributions both in the bowl and in the outer areas. At a swirl of an excessive degree, the mixture formation in the outer parts is again hampered, thus inviting poorer air utilization.

Appendix Angular velocity of the swirling motion during compression stroke

In the mathematical treatment, we shall use the symbols listed below.

- \( D_c \): cylinder diameter
- \( D_b \): diameter of the bowl
- \( D \): arbitrary diameter
- \( G \): total weight of air in the cylinder
- \( g \): acceleration of gravity
- \( \omega \): angular velocity of the swirl
- \( x \): distance from the cylinder head to the piston surface
- \( \beta \): diametral ratio of the bowl to the cylinder

(\( = D_b/D_c \))

\( \gamma \): specific weight of air at an instantaneous crank angle, and

\( t \): depth of the approximate bowl.

Some of these variables are indicated in Fig. 11.

The total weight of air existing in the cylinder at a position of the piston is

\[
G = \left( \frac{\pi}{4} - D_c^2x + \frac{\pi}{4} - D_b^2t \right) \gamma
\]

Putting to the variables a subscript 0 for indicating the state at the beginning of compression, the conservation of total weight of air can be expressed by \( G = G_0 \), or, by rewriting:

\[
\frac{\gamma}{\gamma_0} = \frac{x_0 + \beta^2t}{x + \beta^2t}
\]

The angular momentum of an infinitesimal fluid element \( dG \) does not change throughout the compression, so that we may have

\[
\frac{dG}{g} \frac{D}{2} \omega = \frac{dG}{g} \frac{D_b}{2} \omega_0
\]

where \( D_b \) denotes the diameter at which the fluid element under consideration has been located before the compression, and \( \omega_0 \) the corresponding angular velocity at \( D_b \). Rewriting this, we have

\[
\frac{\omega}{\omega_0} = \frac{D_b^2}{D^2}
\]

The problem is now reduced to deriving the diametral ratio \( D_b/D \). This may easily be calculated by taking into consideration the relative diameters of a fluid element before and after the piston displaces. Figure 11 illustrates the respective
states when the displacement takes place from $x_0$ to $x$. In deriving the ratio $D_0/D$, three different expressions would be necessitated according to the relative locations of the air element in the two states, indicated as 1, 2, and 3 in the figure.

References

(1) p. 588 of this Bulletin.