The Improvement on the Performance of a Pre-combustion Diesel Engine

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Abstract

The influence of the location where ignition starts on gas ejection from the pre-combustion chamber of a Diesel engine has been studied by both theoretical calculation and a simplified model. Possibility of an application of the results obtained was ascertained by a practical engine.

I. Introduction

A pre-combustion Diesel engine has its own advantage at the expense of higher fuel consumption which has been considered as an inherent drawback due to its greater heat loss and throttling loss at its throat. However, the recent researches have revealed that the above mentioned drawback can be improved to a certain extent by suitable designing of a pre-combustion chamber.

On the other hand, we cannot neglect retardation of combustion process in the main chamber upon which the performance of an engine depends to a great extent.

In some engines, when combustion starts in the pre-combustion chamber, ejection of gas into the main chamber begins at almost critical speed. After progress of combustion process in the main chamber has reached a certain stage, the pressure difference between the two chambers becomes considerably small, thereby causing the ejection to stop for an instant. The ejection from the pre-combustion chamber, however, is soon recovered owing to the down stroke of piston movement. Thus, the ejection process actually consists of two stages in succession. The major portion of fuel will be ejected in the second stage which extends over $20° \sim 60°$ atdc.

As combustion at 30° atdc cannot perform more than 80% of effective work compared with at tdc, the above mentioned process is much disadvantageous from the viewpoint of thermal efficiency and power. Furthermore this slow combustion process prevents a pre-combustion engine from running at higher rpm.

Volume ratio of a pre-combustion chamber, area ratio of a throat, compression ratio and their combination have indeed a large influence on combustion process, but we should also give our considerations to the fact that inferiority of thermal efficiency due to retardation of combustion in the main chamber can be improved by bettering combustion state in the pre-combustion chamber and the shape of the pre-combustion chamber itself. As it is essential for the improvement of the performance that the peak in combustion rate diagram should be shifted toward tdc to make combustion complete earlier, the fuel injected in the pre-combustion chamber should be concentrated near the throat and be brought into the main chamber by the first blow of ejection.

In an engine without improvement on the above mentioned process, fuel ejection into a cylinder delays considerably since the fuel injected will be blown up toward the injection nozzle by an air stream flowing from a cylinder during compression stroke, thus rendering it impossible to concentrate the fuel near the throat.

From these facts, the following problems concerning combustion process should be considered:

a) To what extent gas ejection into the main chamber can be influenced by ignition location in the pre-combustion chamber and, moreover, what are the factors that control it?

b) What influence has a gas stream in the pre-combustion chamber on mixture formation and ignition location?

First of all, we tried to analyse theoretically the influence of ignition location on the flame development, pressure rise as well as on kinetic energy of ejecting gas in a simplified model of a pre-combustion chamber, and then we made an attempt to attack both of these problems in a practical engine.

II. Gas ejection from the model of a pre-combustion chamber

1. Theoretical analysis

The purpose of this analysis is to estimate how the location of an ignition spark will influence pressure rise, ejecting period and kinetic energy of ejecting gas. In order to simplify the treatment, we considered a thermally insulated cylindrical model of the pre-combustion chamber having a throat at one end, in which uniform gas is filled and ignited by a spark plug.

The analysis is made on the following assumptions:

1) Both burnt and unburnt gases are of the same ideal ones having constant specific heat and do not change in numbers of molecules by combustion.

2) No heat transfer takes place in burnt or unburnt gas, or normal to the flame front.

3) The pressure is uniform everywhere in the chamber.

4) Ignition starts simultaneously on a whole plane perpendicular to the cylinder axis and the flame front advances with constant burning velocity.

The following nomenclatures are used in this paper, which are illustrated in Fig. 1.

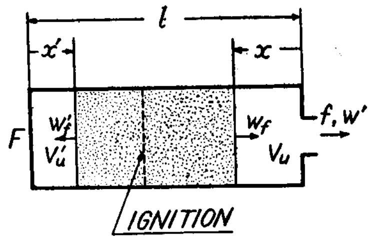


Fig. 1. Schematic diagram of a model chamber.

F : cross-sectional area of the combustion chamber (cm^2)
f : cross-sectional area of the throat (cm^2)
g : acceleration due to gravity ($980\,cm/s^2$)
i : numbers of flame fronts
l : length of the combustion chamber (cm)
R : gas constant (2928 cm kg/kg °K for air)
V_c : volume of the combustion chamber (cm^3)
w_b: normal burning velocity (cm/s)
w_f: flame velocity (cm/s)
w' : isentropic ejecting velocity from the throat (cm/s)
x or x': distance from the throat or from the opposite wall to the flame front (cm)
κ: ratio of specific heats
μ: discharge coefficient of the throat
φ: velocity coefficient of the throat

subscripts

a: in front of the throat
b: burnt gas
o: initial or atomospheric condition
u: unburnt gas

(a) Time rate of pressure rise

Since we are assuming that the change of state in the chamber is adiabatic and the pressure is always uniform, we obtain

$$\frac{dp}{p} = -\kappa\frac{dv}{v} = -\kappa\frac{dV}{V_c}, \tag{1}$$

where v is specific volume of gas at optional position, while dV is the volume change during infinitesimal time dt, which is given by the difference between ejecting volume dV_a and volume increase due to combustion $iFw_b\Delta T\,dt/T_u$.

Accordingly,

$$dV = dV_a - iFw_b\Delta T\,dt/T_u. \tag{2}$$

From the assumption (1), temperature rise ΔT at the flame front becomes always constant. And, isentropic ejecting gas velocity from the throat is expressed by the following equations:

$$w' = \Phi\sqrt{RT_a}\,, \tag{3}$$

where
$$\Phi = \sqrt{2g\frac{\kappa}{\kappa-1}[1-(p_0/p)^{\frac{\kappa-1}{\kappa}}]}\,, \quad p_0/p > p_k/p$$

$$\Phi = \sqrt{2g\frac{\kappa}{\kappa+1}}\,, \quad p_0/p \leqq p_k/p$$

$$\frac{p_k}{p} = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}.$$

Since the ejecting volume in the expanded condition is $\mu f w' dt$, the corresponding volume change in front of the throat can be converted into the following equations: For $p_0/p > p_k/p$,

$$\left.\begin{aligned} dV_a &= \mu f w'\cdot(p_0/p)^{\frac{1}{\kappa}}\,dt = \mu f\cdot\Phi\sqrt{RT_a}\cdot(p_0/p)^{\frac{1}{\kappa}}\,dt \\ &\text{and for } p_0/p \leqq p_k/p, \\ dV_a &= \mu f w'\cdot(p_k/p)^{\frac{1}{\kappa}}\,dt = \mu f\cdot\Phi\sqrt{RT_a}\cdot\left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}} dt, \end{aligned}\right\} \tag{4}$$

while the temperature of ejecting gas T_a varies depending upon whether the gas is burnt one or unburnt.

Now, in this paper, the equations discussed below will be treated only in the case of $p_0/p > p_k/p$, but they can be further simplified in the case of $p_0/p \leqq p_k/p$.

i) The case in which the ejecting gas is an unburnt gas.

Using the adiabatic relation, the temperature of an unburnt gas is determined by the instantaneous pressure in the chamber as follows:

$$T_a = T_u = (p/p_0)^{\frac{\kappa-1}{\kappa}}\cdot T_0. \tag{5}$$

Applying (2), (4), and (5) to (1), the relation between pressure and time can be represented by the following equation:

$$\frac{d(p/p_0)}{dt} = K_1\cdot(p/p_0)^{\frac{1}{\kappa}} - K_2\cdot\Phi\cdot(p/p_0)^{\frac{3}{2}\cdot\frac{\kappa-1}{\kappa}}, \tag{6}$$

where
$$K_1 = \frac{i\kappa F w_b \Delta T}{V_c T_0} = \frac{i\kappa w_b \Delta T}{l T_0}\,, \quad K_2 = \frac{\kappa\mu f\sqrt{RT_0}}{V_c}.$$

ii) The case in which the ejecting gas is a burnt gas.

Taking into consideration the compression caused by combustion, we have the temperature of a burnt gas at the throat expressed by

$$T_a = [(p'/p_0)^{\frac{\kappa-1}{\kappa}}\cdot T_0 + \Delta T](p/p')^{\frac{\kappa-1}{\kappa}}, \tag{7}$$

where p' represents the pressure of the chamber at the moment when the combustion of the same gas has taken place.

The exact calculation is very complex so that we assume $p'=p_0$. Then, T_a corresponds to the temperature of the portion of gas which burnt at the beginning. Hence, the relation between pressure and temperature for the burnt gas can be expressed as before:

$$\frac{d(p/p_0)}{dt} = K_1\cdot(p/p_0)^{\frac{1}{\kappa}} - K_3\cdot\Phi\cdot(p/p_0)^{\frac{3}{2}\cdot\frac{\kappa-1}{\kappa}}, \tag{8}$$

where $$K_3 = \frac{\kappa\mu f\sqrt{R(T_0+\Delta T)}}{V_c}.$$

(b) Flame travel

Applying the law of adiabatic change to an unburnt gas, we always have

$$\frac{dp}{p} = -\kappa\frac{dV_u}{V_u}. \tag{9}$$

And, an unburnt gas is generally divided by a burnt gas into two regions, namely the throat side and the opposite side.

When the flame travels on the throat side, the volume change of the unburnt gas is effected by compression due to combustion and ejection from the throat. Then, we have

$$dV_u = [\mu f\cdot\Phi\sqrt{RT_a}\cdot(p_0/p)^{\frac{1}{\kappa}} - F(w_f - w_b)]\,dt. \tag{10}$$

Now, using the simple relations $V_u=Fx$, $w_f=-dx/dt$ and applying equation (10) to (9), we get the following equation:

$$\frac{dx}{dt} + \frac{x}{\kappa(p/p_0)}\cdot\frac{d(p/p_0)}{dt} + \frac{f}{F}\cdot\mu\cdot\Phi\sqrt{RT_a}\cdot(p_0/p)^{\frac{1}{\kappa}} + w_b = 0. \tag{11}$$

Solving the above, the distance from the throat to the flame front can be obtained as follows:

$$x = \left(\frac{p}{p_0}\right)^{-\frac{1}{\kappa}}\left[C - \int\left\{\frac{f}{F}\cdot\mu\cdot\Phi\sqrt{RT_a} + w_b\left(\frac{p}{p_0}\right)^{\frac{1}{\kappa}}\right\}dt\right], \tag{12}$$

where C is an integration constant determined by the initial conditions.

When the flame travels on the opposite side, the volume change is caused only by compression due to burning. Therefore, putting $\frac{f}{F}\cdot\mu\cdot\Phi\sqrt{RT_a}=0$ in equation (12), we get the distance from the opposite wall to the flame front as follows:

$$x' = \left(\frac{p}{p_0}\right)^{-\frac{1}{\kappa}}\left[C' - \int w_b\left(\frac{p}{p_0}\right)^{\frac{1}{\kappa}}dt\right]. \tag{13}$$

By using equations (6) and (8), the relation between p and t can be graphically calculated for given values of K_1, K_2 and K_3. By introducing the relation between p

and t thus obtained into equation (12) or (13), the position of the flame front x or x' will be obtained by graphical integration.

(c) Kinetic energy of the ejecting gas

Kinetic energy of gas ejecting from the throat dE is expressed by well known equation:

$$dE = \mu\varphi^2 f w'^3 dt/2gv' ,$$

where v' is the specific volume after adiabatic expansion to p_0. Since $w'=\Phi\sqrt{RT_a}$, $v'=v_a(p/p_0)^{\frac{1}{\kappa}}$ and $v_a=RT_a/p$, then the above equation becomes

$$\frac{dE}{dt} = \frac{f}{2g}\cdot\mu\varphi^2\cdot\Phi^3\sqrt{RT_a}\cdot p_0\left(\frac{p}{p_0}\right)^{\frac{\kappa-1}{\kappa}} . \tag{14}$$

2. Results of the calculation and their considerations

In carrying out the numerical calculation by using the theoretical equations derived above, we employed the following values so as to make a comparison of the results obtained with those of experiment.

$\kappa = 1.4$, $f = 1.1^2\times\pi/4\,\mathrm{cm}^2$, $\mu = 0.6$, $\varphi = 0.6$, $R = 2928\,\mathrm{cm\,kg/kg\,°K}$,
$p_0 = 1\,\mathrm{kg/cm}^2$, $T_0 = 288\,\mathrm{°K}$, $V_c = F\times l = 15.8\times 11.3\,\mathrm{cm}^3$, $\varDelta T = 2016\,\mathrm{°K}$,
$w_b = 102\,\mathrm{cm/s}$,

where $\varDelta T$ and w_b were estimated from the experimental results. The former was calculated from the mean temperature corresponding to the pressure rise which had taken place when the same mixture was burnt in a completely closed spherical bomb, and the latter from the pressure rise which had taken place when the ignition started at the center of the bomb by a formula given by A. S. Campbell[1].

Using the above values, constant K_1, K_2 in (6) and K_3 in (8) become

$$K_1 = 75i\,s^{-1}, \quad K_2 = 4\,\mathrm{cm}^{-\frac{1}{2}}, \quad K_3 = 10.6\,\mathrm{cm}^{-\frac{1}{2}} .$$

The ignition points are selected at every decasectional point between the throat and the opposite wall as shown in Fig. 2 with numbers "0"~"10". Fig. 2 illustrates the variation of the pressure in the chamber against time for each ignition point and the corresponding flame development.

The magnitude of the pressure rise increases as the ignition point shifts from "10" to "0", reaching the maximum at a certain point, and then decreases again from there on.

The maximum pressure can be obtained when the ignition starts at such a point that both flame fronts arrive simultaneously at both ends. The time required for the completion of ejection decreases rapidly as the ignition point moves from "10" toward "0", attaining the minimum at the same ignition point where the maximum pressure is obtained, and it is designated by "1.33". In this case the time required is about

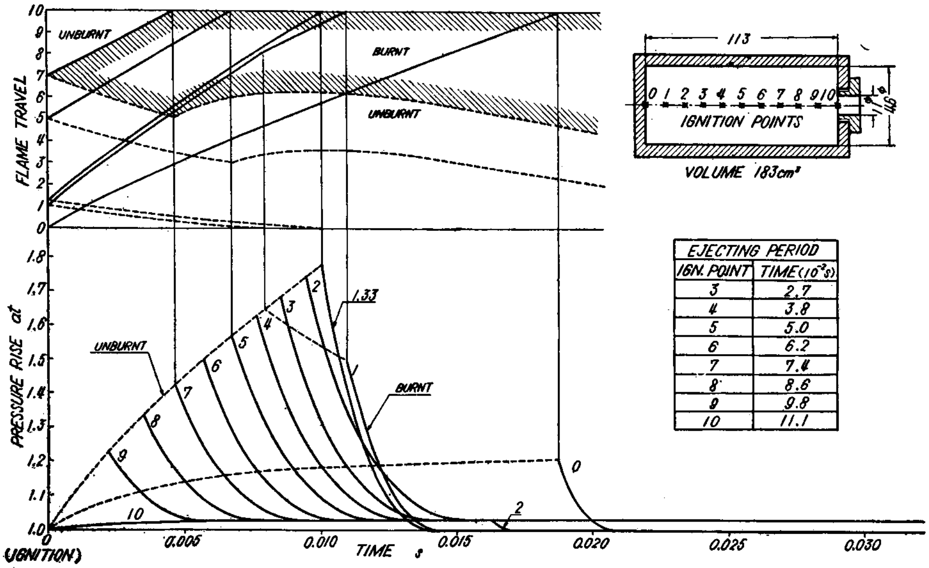


Fig. 2. Pressure variation and flame travel in the chamber calculated with respect to each ignition point.

1/8 of the case "10". When the ignition starts at "0" where only one flame front exists from the beginning, the pressure rise is small, but the ejecting period is comparatively short. From the case "2" to "9", one flame front reaches the throat side earlier than the other and thereafter the pressure decreases rapidly to a certain equilibrium magnitude, because only one flame front is remaining and, further, the velocity of the ejecting gas increases due to conversion of the ejecting gas from the unburnt gas to the burnt, keeping its constant magnitude until the other flame front has reached

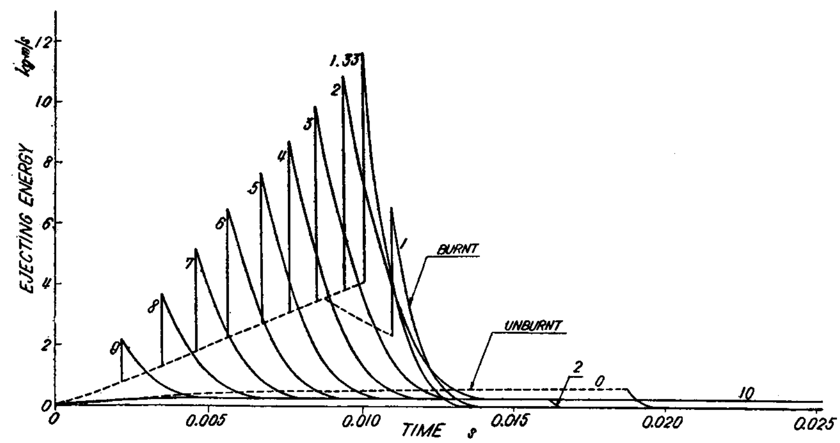
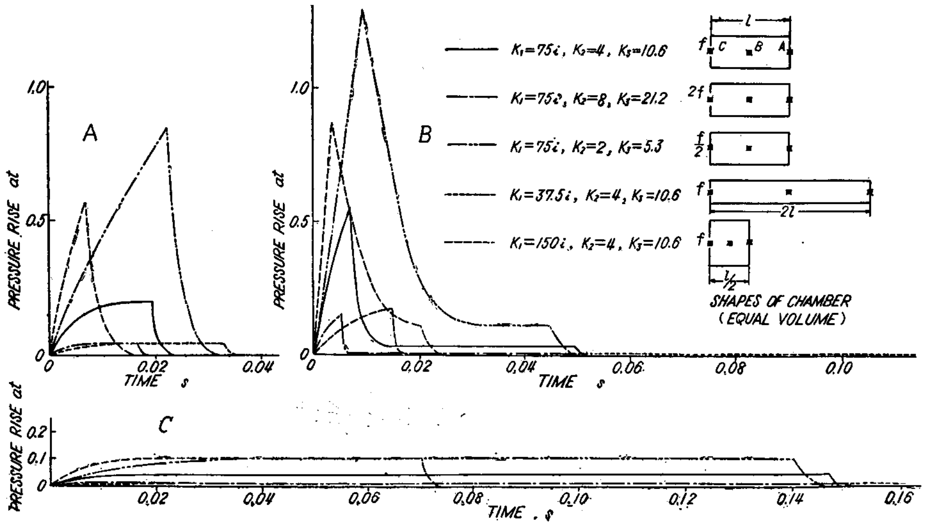


Fig. 3. Ejecting energy from the throat calculated.

the opposite wall. In the case "10", the pressure converges to the same equilibrium magnitude immeadiately after igniting, because the flame front is only one from the beginning and the ejecting gas is always burnt one. In the case "1", on the other hand, the flame front reaches the opposite wall earlier and thereby causes a discontinuous point on the pressure curve in decreasing part.

The relation between time and kinetic energy of the ejecting gas in a unit time corresponding to Fig. 2, is shown in Fig. 3. The total energy is given as the area which is enclosed by the curve and the time axis. At the moment the ejecting gas converts from the unburnt gas to the burnt, the ejecting energy increases discontinuously as clarified by equation (14). Though it has the same tendency as the relation between pressure and time, the ignition point where the total ejecting energy attains the maximum shifts a little toward the throat compared with that of the maximum peak pressure.



Fig, 4. Influence of the shape of the chamber on gas ejection.

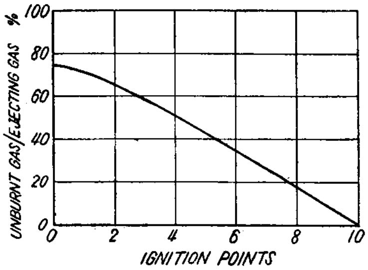


Fig. 5. Mass ratio of unburnt gas to total ejecting gas.

Mass ratio of the ejecting gas to the whole gas in the chamber before ejection, is approximately represented by $\Delta T/(T_0+\Delta T)$ for the ignition points from "2" to "10". However, the ratio of gas ejected in unburnt state to the whole gas ejected depends upon the ignition point as shown in Fig. 4.

Further, the calculations were performed to estimate the influence of shape of the chamber, area of the throat and the burning velocity of gas on gas ejection. Constant K_1 in equation (6) and (8) includes w_b/l (the ratio of burning velocity to length of the chamber) and both constants K_2 in (6) and K_3 in (8) include the cross-sectional area of the throat f. Fig. 5 shows the variation of pressure in five chambers of equal volume having various combination of f and l. In this case the ignition points are chosen at three points, i.e. the throat (A), the center of the chamber (B), and the inner wall (C). It will be noticed that the shape of the combustion chamber exerts a large influence on the pressure rise as well as the ejecting period.

3. Method of Experiment

In order to ascertain the results of our theoretical analysis, we made an attempt to record actually the pressure diagrams in the same model chamber, which is shown in Fig. 6.

After scavenging the chamber completely with mixture, it was ignited by a spark plug immediately upon closing the cock. The combustible mixture was composed of 25% town gas and 75% air by volume.

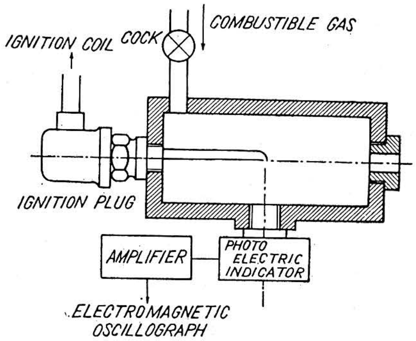


Fig. 6. Experimental arrangement for measuring pressure variation in the model chamber.

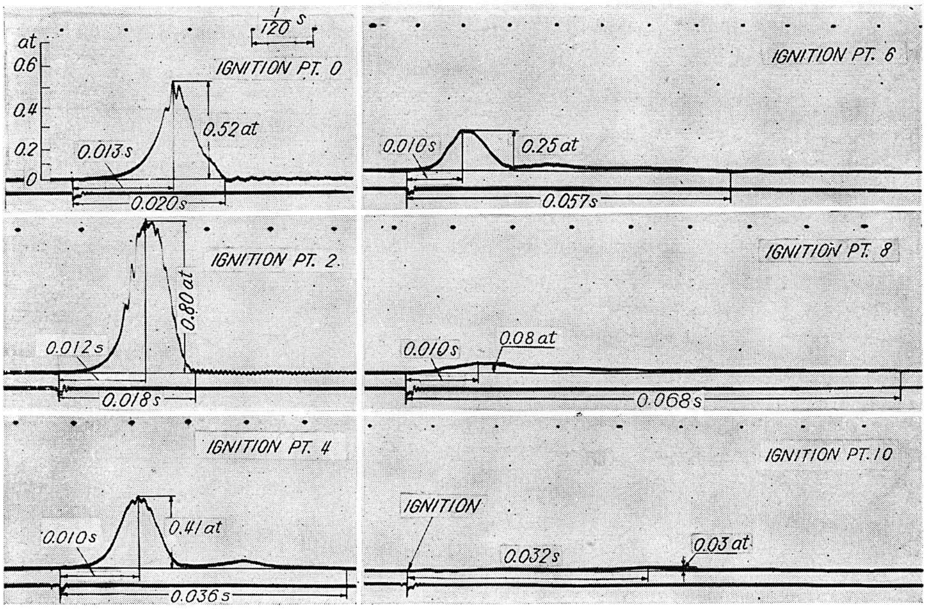


Fig. 7. Pressure variation in a cylindrical chamber recorded.

4. Experimental results

Fig. 7 shows the pressure diagrams recorded by a photo-electric indicator in the same condition as the numerical calculation. The number on oscillograms corresponds to the position of the ignition point. It is natural that the experimental results may differ from the theoretically calculated ones, because we assumed that the ignition starts in a plane in spite of the fact that actual ignition by a spark plug must take place from one point with a considerable ignition lag. However, the results obtained coincide fairly well with the theoretical ones not only in the tendency but also in the magnitude of the peak pressure and the ejecting period.

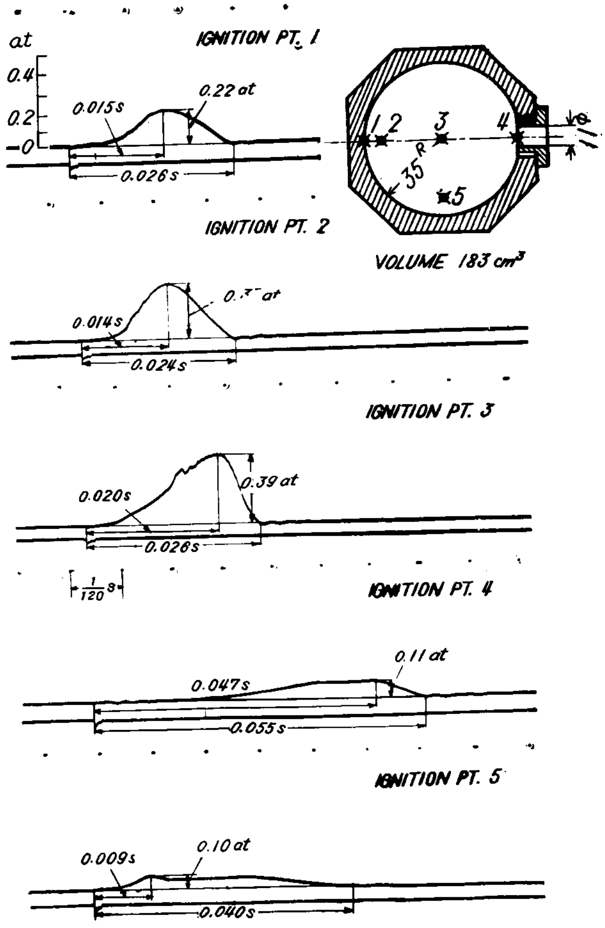


Fig. 8. Pressure variation in a spherical chamber recorded.

The other experiments were carried out to determine the influence of shape of the chamber on gas ejection. A spherical chamber and a stepped cylindrical chamber with the equal volume, as often used in a practical pre-combustion chamber, were employed for the experiments. The results showed that the maximum pressure in a spherical chamber was considerably smaller, and the ejecting period was found to be shorter than in a cylindrical one, both of them being less affected by ignition location as shown in Fig. 8. In regard to a stepped cylindrical chamber, on the other hand, the influence of ignition location became far greater and it was only in the case of ignition in the inner part that larger pressure rise was observed.

III. Experiments with a practical engine

1. Descriptions of an engine used for the experiments

In order to apply the results obtained by the model experiment of a pre-combustion chamber to a practical engine, experiments were carried out with a pre-combustion Diesel engine of the Yanmer-Diesel & Co., of which main dimensions were as follows:

Type	4 cycle single cylinder water-cooled horizontal engine
Cylinder bore/stroke	80 mm/115 mm
Normal output	4 B HP (at 1200 rpm)
Compression ratio	18.2 : 1
Volume ratio of a pre-combustion chamber	32%
Area ratio of the throat	0.45% of piston
Fuel injection pump	needle regulation
Fuel injection nozzle	Bosch DN4S1

The section through the combustion chamber is shown in Fig. 9. The fuel used for the experiment was always heavy oil "A" (specific weight 0.853 at 20°C, Cetan number 45).

2. Experimental results

(a) Influence of ignition location in a pre-combustion chamber

An attempt was made to place a glow element helpful to initiate ignition in a pre-combustion chamber. The experiments were divided into two different glow elements, one of which we call here a hot wire and the other a gas pocket.

Effect of a hot wire: A coil of nicrom wire of 1.8 mm diameter was placed at various locations in the pre-combustion chamber as shown in Fig. 10. Although its temperature during running was not measured, it could be judged from the fact that the surface of the wire had turned to white after running, being heated up to considerably high temperature and showing its own effect. Fig. 10 shows the curve of sfc at 1000 rpm measured at various locations of the wire. From the fact that the best performance is obtained

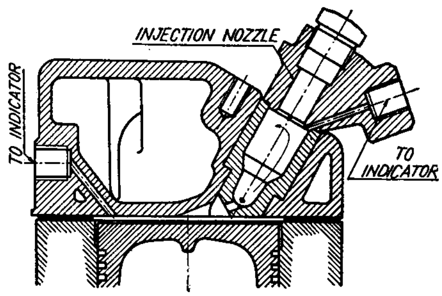


Fig. 9. Section through the combustion chamber of the experimental Diesel engine.

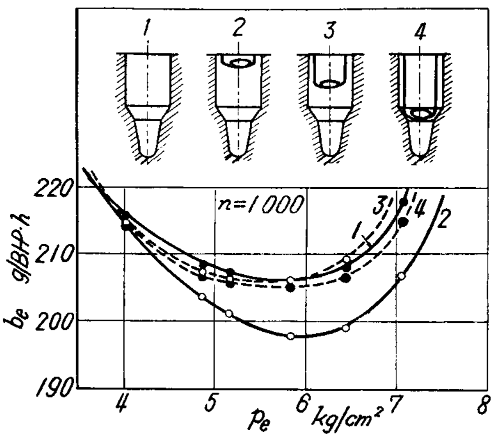


Fig. 10. Effect of a hot wire on the specific fuel consumption.

in the case (2) which had the wire in the upper part and that its effect appeared over 4 kg/cm² bmep, it is quite clear that ignition from the upper part of a pre-combustion chamber is effective in improving combustion process. Furthermore, from the fact that its effect can hardly be observed in the case (3) and (4), it may be concluded that ignition without the wire starts from the lower part of a pre-combustion chamber.

Effect of a gas pocket: Another attempt was made. A gas pocket was placed in the upper part, the function of which was, similar to the hot bulb of a hot bulb engine, to promote ignition by overheating and residual gas in that part of the chamber. The results of this experiment is shown in Fig. 11, in which we can find discontinuous points on both curves of sfc and exhaust gas temperature (t_e). These phenomena show an interesting conclusion that over a certain bmep the ignition location can be transfered to the upper part of a pre-combustion chamber by the influence of a gas pocket and, consequently, an improvement in combustion process is possible. Further, it is quite proper that the discontinuous point in the case of decreasing load appears at different bmep from that in increasing load, moving to lower bmep than in the latter case. Therefore, it can be stated that lower fuel consumption and larger output can be gained by the effect of a gas pocket in the range beyond the discontinuous point.

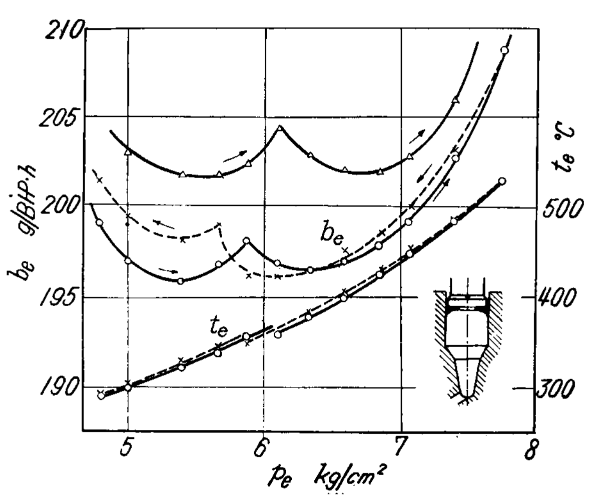


Fig. 11. Effect of a gas pocket.

—△— 900 rpm
—○— 1000 rpm load increasing
--×-- 1000 rpm load decreasing

(b) Influence of a gas stream in a pre-combustion chamber

In this pre-combustion chamber, as shown in Fig. 9, there is a tendency for a gas stream to flow up from a main chamber along the side wall. Therefore, by making grooves of different depths on the chamber wall, it is possible to determine the influence of a gas stream in the pre-combustion chamber on combustion process. The results obtained are given in Fig. 12. The best performance was obtained in the case (7) which had the groove of suitable depth.

It will be understood from the results thus obtained that the gas stream in this case has brought about an effective combustion process by the following two functions:

i) No head-on collision between the gas stream and the injected fuel takes place, thereby keeping the fuel spray from being blown up and, moreover, causing it to stay mainly near the throat.
ii) The swirl arising in the upper part is helpful to initiate ignition.

The latter can be confirmed by the fact that the effect of the wire placed at the upper part was indifferent in the case (7). In regard to the depth of the groove, the performance in the case (8) with excessive depth became somewhat inferior, because the excessive swirl might cause the fuel to adhere to the upper part of the wall.

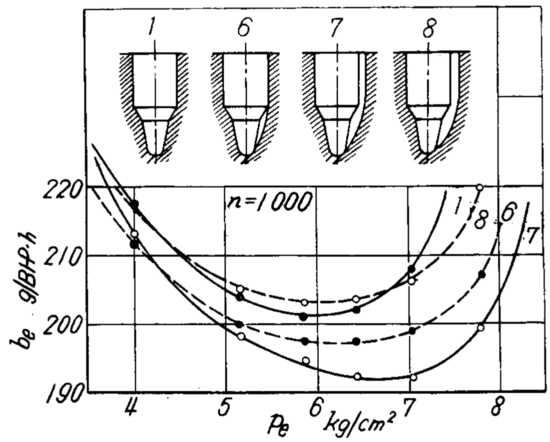


Fig. 12. Influence of gas flow in a pre-combustion chamber.

In the other way, we made an attempt to offset the injection nozzle from the center line of a pre-combustion chamber so as to prevent a collision with the gas stream. The results obtained show a similar tendency as that of the groove, confirming the accuracy of the view we hold.

(c) Comparision of each type of combustion chamber

Selecting three combustion chambers which showed the best performance in each type, we tested the engine over entire speed range. The performance curve of an original pre-combustion chamber is shown in Fig. 13, that of a chamber with a wire in the upper part in Fig. 14 and that of a chamber with a groove in Fig. 15.

A pre-combustion chamber with a groove of suitable depth gave the best performance as

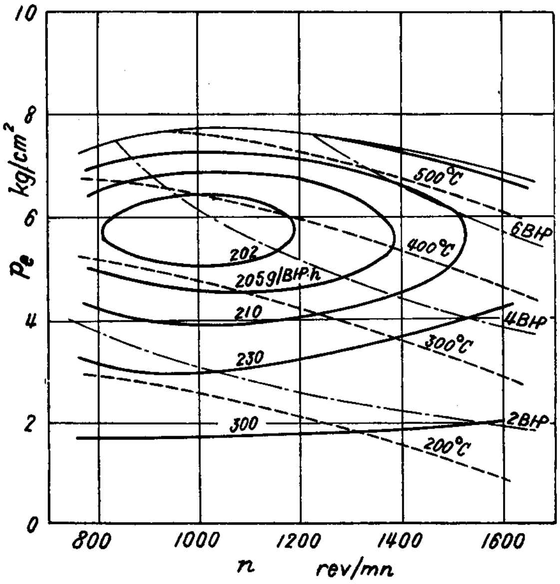


Fig. 13. Performance curve of the chamber (1).

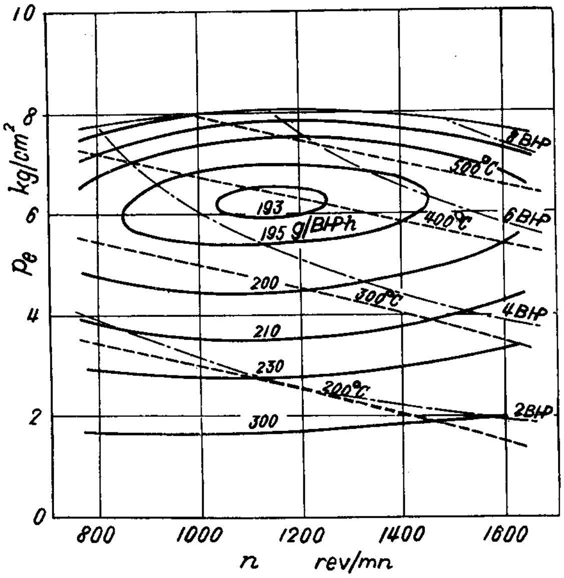


Fig. 14. Performance curve of the chamber (2).

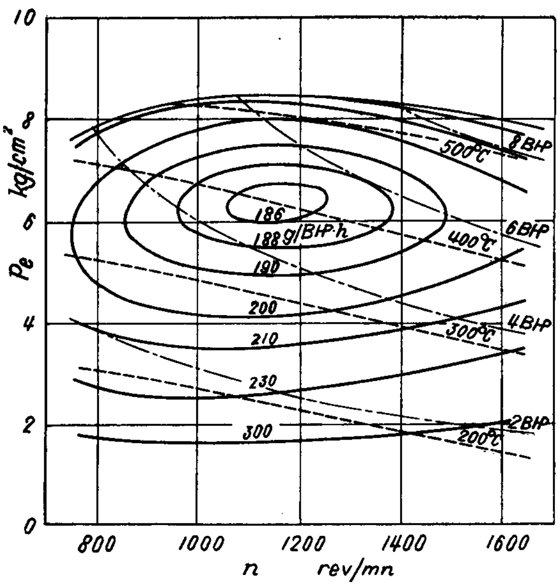


Fig. 15. Performance curve of the chamber (7).

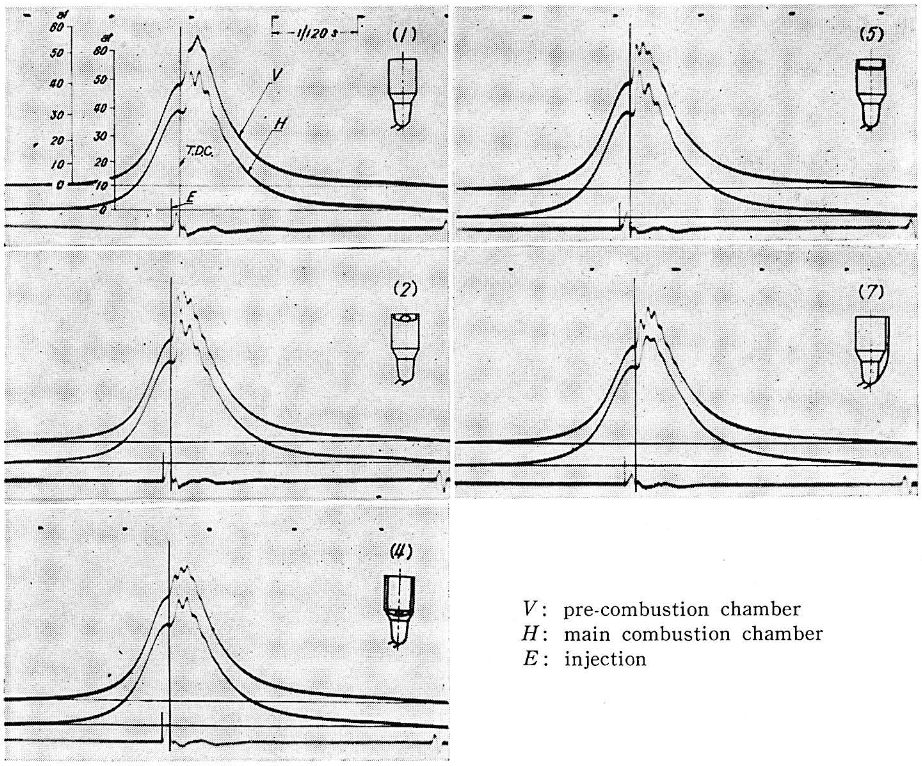
illlustrated in Fig. 15, i.e. the maximum bmep reached 8.5 kg/cm² and the minimum sfc 186 g/B HP-h. These values assure the gain of 0.7 kg/cm² in bmep and 16 g/B HP-h in sfc compared with those before improvement. This is fairly superior performance for a small single cylinder engine with excess air ratio of 1.1 and imep of 10.6 kg/cm².

In the case of a pre-combustion chamber with the hot wire, a large improvement on the maximum bmep and the minimum sfc extends over wide speed range, for instance, at 1600 rpm its gain was 0.8 kg/cm² in bmep and 20 g/B HP-h in sfc.

Though discoloration of exhaust smoke was not measured precisely, it was a noteworthy experience that especially in the case of a pre-combustion chamber with a suitable groove, the engine ran with little soot over entire range of speed and bmep.

3. Comparison of indicator diagrams

In order to make further studies on the above mentioned results, the pressure diagrams in both chambers were simultaneously taken by the strain gage indicators. Some typical diagrams are shown in Fig. 16. Table 1 shows the summary of the mean values obtained from many pressure diagrams.



V: pre-combustion chamber
H: main combustion chamber
E: injection

Fig. 16. Pressure diagrams of the engine recorded with pressure pickups of electric resistance strain gage type (natural frequency 35000 c/s) and electromagnetic oscillograph (natural frequency 6000 c/s).

Table 1

No. of pre-combustion chamber and pressure diagram		1	2	4	5	7
Ignition lag in the pre-combustion chamber	deg.	7	6	5	7	7
Peak pressure in the pre-combustion chamber	ata	53	55	51	54	51
Peak pressure in the main combustion chamber	ata	52	54	49	54	49
Max. pressure difference	at	6	9	4	8	4
Crank rotation from tdc to peak pressure in the main combustion chamber	deg.	8	6	13	6	7

By comparing the pressure pattern (2) with (4), the influence of ignition location in a pre-combustion chamber on combustion process becomes clear. The former indicates steeper and higher pressure rises and greater pressure difference between both chambers, i.e. the maximum peak pressure in a main chamber appears 7 deg. of crank rotation earlier than in the latter. The same effect can clearly be seen in the case of the pre-combustion chamber with a gas pocket as shown in the pattern

(5). In a practical engine, ignition does not always start from one point, however, from the above fact, we can confirm that the results revealed by the model experiment can be applied to a practical engine, i.e. the steeper pressure rise in the pre-combustion chamber, which was caused by ignition in the upper part, contributes to blow out fuel into the main chamber very rapidly with high kinetic energy, and thus promotes combustion process.

Comparing the pressure patterns of (4) and (7), we can find steeper pressure rise in the main chamber in the case of the chamber with a groove, but no significant pressure rise is seen in the pre-combustion chamber. Consequently, we can say that the major portion of fuel blown out into the main chamber at the first ejection promotes combustion process to a great extent. Moreover, in this case, it is clearly seen that the peak pressure and the rate of pressure rise are comparatively low, resulting in the pressure pattern smoother than those of chamber (1), (2) and (5).

IV. Conclusion

From the results of our investigation, the following conclusions can be drawn:

(1) Ignition location in a cylindrical model chamber exerts a large influence upon gas ejection, and only in the case of ignition near the far end of the throat, pressure rise and ejecting energy reach the maximum, hence ejecting period attaining the minimum.

(2) These rules can be translated into practice to improve combustion process of a practical engine, which fact has been ascertained by the indicator diagrams.

(3) In order to improve the performance, fuel ejection from a pre-combustion chamber should be done in a shortest period in order that the major portion of fuel is carried away with high kinetic energy into the main chamber. This state can be attained by concentration of fuel near the throat and also by causing ignition in the upper part of the pre-combustion chamber.

Reference

1) A. S. Campbell: The Time required for Constant-Volume Combustion, A. S. M. E. J. App. Mech. 1952 Vol. 19 No. 12, p. 72.