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References

- (1) Y. Mori: *Trans. Japan Soc. Mech. Engrs.*, Vol. 21, No. 109 (1955), p. 679.
- (2) J. T. Hamrick, A. Ginsburg, & W. M. Osborn: *NACA Report*, No. 1083 (1952).
- (3) J. D. Stanitz: *Trans. ASME*, Vol. 74, No. 4 (1952), p. 1346.
- (4) C. H. Wu: *Trans. ASME*, Vol. 74, No. 8 (1952), p. 1363.
- (5) K. Fujie: *Bulletin of JSME*, Vol. 4, No. 13 (1961), p. 94.
- (6) H. E. Sheets: *Trans. ASME*, Vol. 72, No. 7 (1950), p. 1009.
- (7) M. Shirakura: *Jour. Japan Soc. Mech. Engrs.*, Vol. 60, No. 465 (1957), p. 1092.
- (8) Y. Mori: *Jour. Sci. Res. Inst.*, Vol. 50, No. 1430 (1956), p. 107.

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Influence of Connecting Passage on Diesel Knock in Divided Chamber Type Engines*

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Combustion noise of the Diesel engine of a pre-combustion chamber type will be originated by some mechanical cause involved with sharp pressure rise in the main chamber. This pressure rise is determined by the rate of heat release in it and the quantity of gas issued from the auxiliary chamber. In order to clarify the latter effect, which will be closely related with the pressure rise in the auxiliary chamber and the area of the connecting passage, an air model was proposed. The pressure rise in the main chamber caused by a sudden pressure jump in the auxiliary chamber was measured in various areas of the connecting passage, and was compared with the theoretical calculation in which gas inertia was neglected. As the result, it was found that the rate of pressure rise in the main chamber is proportional to the area ratio of the passage and inversely proportional to the volume ratio of the main chamber. Application of the result to the actual engines shows that transmission of rapid pressure rise in the pre-chamber to the main chamber is strongly suppressed, resulting in a smooth pressure rise with the pre-combustion chamber type engine, while it cannot be with the swirl chamber type.

1. Introduction

The combustion noise of a Diesel engine will be originated by some mechanical cause involved with a sharp pressure rise in the main combustion chamber. As regards the pre-combustion chamber type engine, the pressure rise in the main combustion chamber is the key to determining the intensity of the combustion noise or the Diesel knock as its extreme manifestation, as has been reported by the previous paper⁽¹⁾, rather than the course of pressure in the pre-combustion chamber which, of course, has a close

connection with the pressure in the main chamber. Hence, it may be stated as the general tendency of the noise formation in the divided chamber type engine that the rate of pressure rise in the main chamber is determined by that of heat release in it as well as by the transmission of the sharp pressure rise in the auxiliary chamber into the main chamber. How the latter influences on the pressure rate in the main chamber will be closely concerned with the cross-sectional area of the connecting passage.

In this study, this effect will be theoretically investigated by introducing an idealized model in which there is a step-functional pressure rise in the auxiliary chamber and no combustion takes place in the main chamber. Further, the estimation of the influence of the area of the connecting passage upon

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the pressure rise in the main chamber will be testified by an experiment with an air simulation tank under which the situation of the combustion chambers may be realized. Thus the attained results will finally be applied to the actual engine in order to evaluate the effect of the connecting passage on the intensity of the combustion noise.

2. Idealization of combustion chamber system

For the convenience of analysis, such an idealized case as the following is presumed: The occurrence of the sharp pressure rise due to combustion being usually confined to the neighborhood of the top dead center of crankshaft, the space of the main combustion chamber virtually remains constant during the combustion. The pressure jumps step-functionally in the auxiliary combustion chamber, although its actual course may be too complicated in nature to be calculated. There is no heat release in the main chamber. And, after the pressure jumps in the auxiliary chamber, the pressure in this chamber is held constant.

The state of main chamber under these assumptions is analogous to that of an evacuated tank being filled with air through an entrance from the surrounding of the infinite space. Fig. 1 illustrates the actual system (a) and the analogous system (b). The present problem is then reduced to finding out the pressure response in such a tank where the pressure at the inlet is suddenly changed.

The relationship between the pressure in the auxiliary chamber p_1 , whose space volume V_1 is infinite for the present case, and that in the main chamber p_2 , can be derived from the mass- and the energy-equilibriums of gas flowing in through the connecting passage whose cross-sectional area is denoted as f . Consequently, the quantity of gas transferred dG at an infinitesimal duration dt can be expressed as follows:

$$dG = \mu f \sqrt{2g \frac{p_1}{v_1}} \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_2}{p_1} \right)^{2/\kappa} - \left(\frac{p_2}{p_1} \right)^{(\kappa+1)/\kappa} \right\}} \cdot dt \quad (1)$$

where, g : acceleration of gravity, v_1 : specific volume of gas in the auxiliary chamber, μ : flow coefficient,

κ : ratio of specific heats of gas.

If pressure in the main chamber rises by dp_2 due to the gas inflow dG , the energy balance equation may be represented as follows:

$$dG = \frac{V_2}{\kappa R T_1} \cdot dp_2 \quad (2)$$

where, V_2 : volume of the main chamber, T_1 : temperature in the auxiliary chamber, R : gas constant. Eliminating dG in Eqs. (1) and (2), we obtain the final relation,

$$\frac{dp_2}{dt} = \frac{\sqrt{2\kappa a_1 p_1} \cdot \mu f D}{D V_2} \times \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_2}{p_1} \right)^{2/\kappa} - \left(\frac{p_2}{p_1} \right)^{(\kappa+1)/\kappa} \right\}} \quad (3)$$

where D is the cylinder bore and a_1 is the sound velocity in the auxiliary chamber ($a_1 = \sqrt{\kappa g R T_1}$).

Denoting the pressure and the sound velocity, before combustion, as p_e and a_e respectively, and adopting them as a reference state, we can rearrange Eq. (3) to give

$$\frac{d}{dt} \left(\frac{p_2}{p_e} \right) = \frac{\sqrt{2\kappa K_0} a_1 p_1}{t_0 a_e p_e} \times \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_2}{p_1} \right)^{2/\kappa} - \left(\frac{p_2}{p_1} \right)^{(\kappa+1)/\kappa} \right\}} \quad (4)$$

where,

$$\left. \begin{aligned} K_0 &= \mu f D / V_2 \\ t_0 &= D / a_e \end{aligned} \right\} \quad (5)$$

K_0 is a non-dimensional number characterized by the geometry of the combustion chamber, containing the cross-sectional area of the connecting passage; t_0 represents the unit of time. In this case, cylinder diameter D is not necessary for the present analysis because the uniformity of pressure inside the combustion chamber is considered here and no attention is paid to the phase distortion of pressure due to gas inertia. Eq. (5) has been presented only for giving a room for some further discussion on such a transient process. Consequently, the present treatment will be limited to a study using a ratio of them: namely,

$$K = \frac{\sqrt{2\kappa K_0}}{t_0} \text{ sec}^{-1} \quad (6)$$

Thus the maximum rate of pressure rise, caused by a rapid pressure rise in the auxiliary chamber, is derived by putting in Eq. (6) the relations of $p_1 = p_{\max}$ and $p_2 = p_e$, as follows:

Fig. 1 Schema of combustion chamber

$$\left[\frac{d(p_2/p_0)}{dt} \right]_{\max} = K \left(\frac{a_1}{a_0} \right) \left(\frac{p_{\max}}{p_0} \right) \times \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_0}{p_{\max}} \right)^{2/\kappa} - \left(\frac{p_0}{p_{\max}} \right)^{(\kappa+1)/\kappa} \right\}} \dots (7)$$

This formula yields the maximum rate of pressure rise as a function of K , and means that the maximum rate due to the inflow of gas into the main chamber does not exceed a definite value given by the coefficient K .

As Eq. (4) refers to the case of a quasi-static inflow of gas, so the pressure in the main chamber p_2 will approach asymptotically to $p_{\max}$ at its equilibrium pressure. The actual pressure-time trace, however, will show some pressure vibration due to the inertia effect of gas, which will serve for p_2 to exceed over $p_{\max}$ or to accelerate the pressure more rapidly toward $p_{\max}$ than without it.

3. Experimental apparatus

In order to realize the foregoing idealized system of the combustion chamber arrangement for observation of the pressure rise in such a process, an air simulation tank was employed. Its total arrangement is shown in Fig. 2 and the full sketch of the main chamber with its entrance in Fig. 3. The sudden pressure rise in the auxiliary chamber and the air flowing into the main chamber are simulated by a break-off of the cellophane diaphragm which initially endures the pressure difference of two chambers. The diaphragm is pierced through a needle triggered electrically by an external signal. The volume of the auxiliary chamber is so large as compared with the main chamber that the assumption that the

Fig. 2 Experimental arrangement for realizing a sudden pressure rise in the main chamber

pressure in the auxiliary chamber holds constant after the step-functional jump, may be realized. For this reason, the volumes of the auxiliary chamber and the main chamber were chosen as 1570 cm³ and 57.8 cm³ respectively. Pressure pickups used were of strain gage type designed as cylinder pressure indicator (nominal value of natural frequency is above 35 kc/sec), and their recording was made on a dual beam cathode ray oscillograph.

4. Experimental results and consideration

Fig. 4 shows examples of pressure records obtained by various area ratios of K under the definite pressure ratios of $p_{\max}/p_0 = 1.39$ and 3.03. In Fig. 5 the experimental pressure-time traces are shown together with the theoretical curves yielded from the integration of Eq. (4), for the serial cases of $p_{\max}/p_0 = 1.39$. Numerical values for this computation were as follows:

$$\kappa = 1.40, \quad \mu = 0.65, \quad p_{\max}/p_0 = 1.39, \\ a_1 = a_0 = 340 \text{ m/sec (15°C)}$$

The theoretical curves drawn in the figure virtually coincide with the experimental ones, especially in the earlier stages of pressure rise, although the theoretical ones were obtained on the assumption of a quasi-static process. Therefore, it may be probable that the departure from the quasi-static process is so small except in the later stages of pressure rise that effects of gas inertia in the system are negligible as compared with the throttle effect of the connecting passage. In the later stages of pressure rise, however, a remarkable gas oscillation is observed with a large value of K . This may be due to the inertia effect of gas.

In the actual engine, the knock intensity and the criterion of whether it is knocking or not are

Fig. 3 Details of main-chamber tank

No. of oscillogram	$K \text{ sec}^{-1}$	$p_{\max}/p_c$	$d(p_2/p_c)/dt \text{ sec}^{-1}$		$t_e \text{ sec}$
			Theoretical	Measured	
1-1	19.1×10^2	1.39	11.7×10^2	11.8×10^2	0.30×10^{-3}
1-2		3.03	36.5	45.2	0.29
1-3	8.4×10^2	1.39	5.2	4.1	0.52
1-4		3.03	16.1	9.6	1.4
1-5	3.7×10^2	1.39	2.3	1.5	1.5
1-6		3.03	7.0	4.6	3.2
1-7	1.72×10^2	1.39	1.06	0.89	3.8
1-8		3.03	3.3	2.2	6.7

Fig. 4 Oscillograms of pressure changes

T and E denote the theoretical- and the measured values respectively

Fig. 5 Theoretical- and measured pressure changes with $p_{\max}/p_c = 1.39$

predominantly governed by the maximum rate of pressure rise or by the maximum pressure and the rising time of pressure rise due to combustion, as stated in the Introduction. For this reason, the availability of Eq. (7), in which inertia of gas has been neglected, for the pressure rising rate obtained experimentally with the simulation tank, was examined. In order to avoid the complexity of the pressure rise at its very beginning, as has been seen in the oscillograms, an average rate at a 25% to 50% rise of pressure was conveniently adopted as the rate of pressure rise to be compared with. The measured rate was plotted in Fig. 6 by $d(p_2/p_c)/dt \text{ sec}^{-1}$, as the function of K . In the same figure are shown the corresponding theoretical lines: One is the line calculated by Eq. (7) and the other is the value of pressure rise at 38% which corresponds to the middle pressure at 25% and 50%. In the upper half of the figure are shown the plots and theoretical line, of the rising time t_r , defined as the duration of pressure rise at 10% to 90%. This figure indicates that the average rate of pressure rise and rising time virtually coincide with their theoretical values of a relatively wide range of K . Consequently, the validity of the equation with the inertia effect neglected seems proved when applied to the practical system containing such a transient process.

By the way, in Fig. 6 the effect of the length of the connecting passage l is shown as the plots of dark circular mark. It can be seen that there is almost no influence given on the rate of pressure rise but that a slight change is observed in the rising time t_r for different lengths.

Concerning the vibrations at the end of pressure rise, the pressure was measured at different positions in the test tank, and was compared with that at the original measuring position. Fig. 7 (a) shows a simultaneous record of pressure in the case of $K = 19.1 \times 10^{-2} \text{ sec}^{-1}$: One is of the original position and the other is of the different position in the same

chamber as indicated as Loc. 2 in the figure. No great phase difference can be seen for the fundamental component of vibration, although higher frequency components do not show any remarkable general rule. Fig. 7 (b) shows comparison between pressure at the original measuring position and that at a different position in the auxiliary chamber indicated as Loc. 3. It can be seen from it that two principal frequencies are not identical with each other. Therefore, it is ascertained that the pressure vibration is not of the category of resonance of the whole

Fig. 6 Effect of K on the rate of pressure rise $d(p_2/p_c)/dt$ and on the rising time t_r

Fig. 7 Pressure changes in various measuring positions, in comparison with those of original position

gas system. Judging from these experiments, the vibration system may be considered as a sort of Helmholtz resonator consisting of the space V_2 and the connecting passage. From other experiments with different passage lengths l as summarized in Fig. 8, it can be noticed that a longer passage generally has a lower frequency.

5. Considerations on the case of actual engine

It may be stated that the rate of pressure rise in the main combustion chamber is smaller than the magnitude which is predicted by Eq. (7) if there happens a sharp pressure jump in the auxiliary chamber and if there is no heat release in the main chamber. On the other hand, the intensity and existence of knock in the actual motor may be roughly expressed by the rate of pressure rise on crank-angle basis: 3 kg/cm²/deg is a limit of audibility of knocking while 5- to 6 kg/cm²/deg gives intense knocking⁽¹⁾⁽²⁾, and 4 kg/cm²/deg may be considered as an allowable limit for knock intensity. This knowledge will give an area ratio of connecting passage with which the pressure rise is not influenced by the sharpness of the pressure rise in the auxiliary

chamber.

If p_e and a_e denote the pressure and sound velocity at the end of compression as the reference state and if it is assumed that gas which flows out of the auxiliary chamber just after the pressure jump is simply of the adiabatic compression from the initial state of p_e and a_e , although the actual change will be far from this owing to high temperature product due to combustion, then the following relationship holds between p_1 and a_1 :

$$\frac{a_1}{a_e} = \left(\frac{p_1}{p_e} \right)^{(\kappa-1)/2\kappa}$$

Transforming time t sec to crank angle θ deg by using the relation $d\theta = 6\pi n dt$, where n is the engine speed in rpm, we obtain the following maximum rate of pressure rise on crank-angle basis:

$$\left(\frac{dp_2}{d\theta} \right)_{\max} = \frac{\sqrt{2\kappa K_0}}{6Dn} p_e a_e \left(\frac{p_{\max}}{p_e} \right)^{(\kappa-1)/2\kappa} \times \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_e}{p_{\max}} \right)^{2/\kappa} - \left(\frac{p_e}{p_{\max}} \right)^{(\kappa+1)/\kappa} \right\}} \dots (8)$$

In this formula, K_0 may be represented by the design parameters of engine and of its combustion chamber; namely,

$$K_0 = \frac{\mu f D}{V_2} = \mu(\epsilon-1) \left(\frac{f}{F} \right) \left(\frac{D}{S} \right) \left(\frac{V_e}{V_2} \right) \dots (9)$$

where, ϵ : compression ratio, F : area of piston surface, S : stroke, V_e : clearance volume, V_2/V_e : volume ratio of main chamber to total compression volume. It can be seen from Eq. (8) that the same K_0/Dn gives the same rate of pressure rise if $p_{\max}/p_e$ is constant.

These relations are applied to a high speed engine with typical combustion chambers arranged. Specifications of the engine to be illustrated are as follows:

Stroke S and bore D :

115 mm and 95 mm respectively

Nominal speed: 1400 rpm

Fig. 9 Maximum rate of pressure rise estimated in the actual motor

No. of oscillogram	K sec ⁻¹	l mm	$d(p_2/p_e)/dt$ sec ⁻¹	t_e sec	Angular frequency rad/sec
3-1	19.1×10^2	5	11.8×10^3	0.30×10^{-3}	3.0×10^3
3-2		10	8.7	0.45	2.8
3-3	3.7×10^2	5	1.45	1.46	2.0
3-4		10	1.20	2.32	1.8

Fig. 8 Influence of connecting-passage length on pressure change

p_c , $p_{\max}$ and a_c have been estimated from indicator diagrams, as follows:

$$p_c = 35 \text{ kg/cm}^2, p_{\max} = 50 \text{ kg/cm}^2, a_c = 550 \text{ m/sec.}$$

Combustion chambers are of pre-combustion chamber type and of swirl chamber type as follows:

	Pre-combustion chamber type	Swirl chamber type
f/F	0.52%	2.0%
V_2/V_c	67%	37%
ϵ	19	16
K_0	0.075	0.436

where

$$\kappa = 1.40, p_{\max}/p_c = 1.50 \text{ and } \mu = 0.65$$

Fig. 9 shows the maximum rate of pressure rise calculated from Eq. (8) by using the numerals presented above. A chain line drawn in the figure indicates the upper limit of knock-free running condition, corresponding to $4 \text{ kg/cm}^2/\text{deg}$. With the pre-combustion chamber type, its estimated pressure rising rate exceeds the limit of below 700 rpm for $f/F = 0.52\%$ and, for 2% of f/F with the swirl chamber type, it falls below the limit of scarcely above 4000 rpm. By considering that the nominal engine speed is 1400 rpm, it may be predicted for the pre-combustion chamber type that the sharp pressure rise in the pre-combustion chamber is not transmitted to the main chamber because of its relatively small cross-sectional area, thus leading to a knock-free condition in the regular speed range.

On the other hand, the swirl chamber type combustion chamber has generally a larger area ratio of connecting passage and has a smaller volume of the main chamber for the sake of such a design concept that the greater portion of fuel may be completely burned in the swirl chamber. Therefore, the value K_0 of the swirl chamber type usually reaches three to ten times the value of the pre-combustion chamber type, so that the swirl chamber type engine does not fulfil the knock-free condition even if it runs at the maximum speed as has been seen in the illustration presented above.

Since the greater part of fuel can burn in the auxiliary chamber in the idling run of the pre-combustion chamber type engine, the assumption presented earlier may actually be almost materi-

alized. It should be borne in mind that the idling knock phenomenon of this type engine, which prevails with the decrease of engine speed, corresponds to the fact that the maximum rate of pressure rise $(dp/d\theta)_{\max}$ augments with the decrease of engine speed. However, this is still an unsatisfactory explanation for the idling knock if considerable amount of unburned fuel has been brought into the main chamber to autoignite in it⁽⁸⁾.

6. Conclusion

The present investigation on the effect of the connecting passage area on the course of pressure in the main combustion chamber, has led to the following results:

(1) Under the ideal condition where there happens a step-functional pressure jump in the auxiliary combustion chamber and there is no heat release in the main chamber, the maximum rate of pressure rise in the main chamber is proportional to the area ratio of the passage and inversely proportional to the volume ratio of the main chamber.

(2) For estimation of the rate of pressure rise in the main chamber, Eq. (7) is approximately valid, although it has been derived on the assumption of a quasi-static process.

(3) At the end phase of pressure rise, a large pressure vibration is observed on a large area of the connecting passage. This is considered as a transient vibration of a kind of Helmholtz resonator which consists of inertia of gas and space effect of the main chamber.

(4) Application of these to the actual engine shows that, with the pre-combustion chamber type engine, a rapid combustion in the auxiliary chamber is not transmitted to the main chamber because of the throttle effect of the connecting passage, resulting in a smooth pressure in the main chamber. On the contrary, it will be with the swirl chamber type.

References

- (1) F. Nagao, M. Ikegami and H. Kakimoto: *Trans. JSAE*, Vol. 15, No. 2 (1961), p. 8.
- (2) J. Broeze: *Engineering*, Vol. 169, No. 4396 (1950), p. 464.
- (3) F. Nagao, H. Kakimoto and T. Hiraoka: *Bulletin of JSME*, Vol. 2, No. 5 (1959), p. 175.