ISSN 1009-3095 *Journal of Zhejiang University(SCIENCE)* V.2, No.1, P.60-65, Jan. - Mar., 2001
http://www.chinainfo.gov.cn/periodical; http://www.zju.edu.cn;
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EXPERIMENTAL STUDY ON PREMIX COMBUSTION AT ISOSCELES TRIANGLE TYPE RATE OF HEAT RELEASE FOR SQUISH TYPE COMBUSTION CHAMBER

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Received Dec.12, 1999; revision accepted May 15, 2000

Abstract: This paper presents a theory on premix fuel combustion at near isosceles triangle type rate of heat release, describes the measures taken for the combustion system, points out its many theoretical advantages, and that it can solve effectively the problems of rough running, fuel consumption and exhaust emission. Two squish lip type combustion chambers are designed to match separately with multiple holes injector and conical spray type injector in order to achieve premix combustion at near isosceles triangle type rate of heat release. Experimental studies on two single cylinder diesel engines showed that premix combustion at isosceles triangle type rate of heat release resulted in longer ignition delay period, larger amount of fuel injected into cylinder during the ignition delay period, lower maximum pressure, better fuel economy, and better exhaust emission.

Key words: premix combustion, isosceles triangle, oil film, conical spray, squish combustion chamber
Document code: A **CLC number**: TK413.2

INTRODUCTION

The pressuregraph given by Lin in 1960 of four types kinds of rate of heat release showed that the isosceles triangle type rate of heat release was ideal on condition of the same maximum pressure. The conclusion is also reached in thermodynamics.

It is popularly accepted that the large amount of heat release in the first stage before top dead center(TDC) and the low rate of combustion after TDC in traditional diffusion combustion cause lots of problems, such as rough running, black smoke, high NO_x and particulate, and late combustion. Reversely, little amount of heat release before TDC and quick heat release after TDC will decrease rough running, fuel consumption and smoke (Hu et al., 1992; Yoshinaka et al., 1996; Nakagome et al., 1997). A theory on combustion at isosceles triangle type rate of heat release is put forward for application to achieving better combustion performance. The theory is as follows. Before ignition, all fuels are injected into the cylinder with uniform distribution in the air space and a thin layer oil film on the outside wall of the piston combustion chamber, and ignition point is set at TDC or near TDC, so the amount of heat release in the first stage before TDC is reduced. Furthermore, quick combustion of large amount of premix fuel after ignition also reduces late combustion in the third stage, and the rate of heat release in the combustion system is near isosceles triangle in shape.

Regarding measures taken for the premix, we have to think of how to inject all fuels into the combustion chamber before ignition. Measures should be taken to make the ignition delay period longer and to shorten injection duration. We know that a uniform and lean fuel mixture can give longer ignition delay. Oil film and conical spray can achieve this purpose, because the formation of oil film will keep the mixture of air and fuel in the chamber space leaner before ignition, and conical spray can make not only injection duration shorter, but also form fine fuel particles and uniform peripheral distribution. We also know that the increase of the injection pump's plug diameter can shorten fuel injection duration, and that the change of injector structure can also achieve the same purpose, for example, conical spray injector in this case. Although having longer ignition delay, the diesel engines run smoothly, because the isosceles triangle type combustion eliminates heat release before TDC. Longer ignition delay period facilitates mixture of fuel and air, for it is easier to mix fuel with air before ignition, that is, the combustion system is characterized by premix combustion, and therefore, the amount of fuel surrounded by flame with diffusion combustion reduces so that the formation of black smoke is effectively under control, and NO_x emission is also better with delayed ignition.

As soon as combustion starts, the premixed fuel and the vapor evaporated from the thin and homogeneous oil film layer in the combustion chamber wall will be rapidly combusted together. If the evaporation of oil film is rapid, the combustion period will be short so that fuel economy is improved.

EXPERIMENTAL STUDY ON 1 - 110 SINGLE CYLINDER TEST ENGINE

The features of the test engine are given in Table 1.

Table 1 Features of 1 - 110 single cylinder test engine

Stroke	Diameter × stroke	Compression ratio	Injector	Combustion type
4	110 × 120 mm	17	5 × 0.29 × 115°	Diffusion combustion

We conducted some comparative experimental studies to help us design the squish lip type combustion chamber detailed in Fig. 1, where D is the diameter of the cylinder, and

$d_k = (0.35 \sim 0.40)D$,
$d_A = (0.59 \sim 0.64)D$,

$H_A = (0.18 \sim 0.22)D$,
$R_1 = (0.04 \sim 0.045)d_k$,

$R_2 = (0.17 \sim 0.19)d_k$,
$R_3 = (0.85 \sim 0.90)d_k$,

$H_B = (0.28 \sim 0.32)d_k$, and
$\alpha = 115° \sim 130°$.

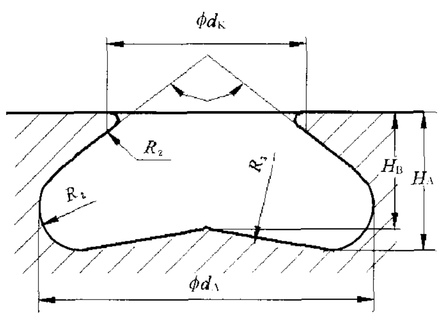

Fig. 1 Squish lip type combustion chamber

Two squish lip type combustion chambers were designed for matching with the multiple holes injector in order to achieve isosceles triangle type rate of heat release. Selection of the above chamber design is based on the following considerations. Compared to the ω type chamber, formation of film is easier in the squish lip type combustion chamber. The swirl flow and squish flow in the chamber are stronger, and will therefore improve the combustion after TDC. The chamber has the advantages of lowering maximum pressure, cutting down exhaust emission, and keeping fuel economy satisfactory (Takeshi et al., 1986; Zhang et al., 1995).

In our experiments, two squish type combustion chambers separately replaced the ω type combustion chamber in the test engine in order to compare the performances among them. In the following parts, ω type chamber scheme is called original scheme, the experiments using squish type chambers are respectively called scheme I, in which the compression ratio is 15.6 and scheme II, in which the compression ratio is 17.5.

An AVL675 digital analyzer and an FP6 emission analyzer were used in the experiments.

Fig. 2 shows the respective pressuregraph of the three schemes, and Fig. 3 shows the rate of heat release, when $n = 2\ 000\ r \cdot min^{-1}$, and $P_e = 11$ kW. Data on the combustion characteristic of the three schemes are given in Table 2. From the diagrams and table, we get the following conclusions.

The longer ignition delay period in both scheme Iand scheme II is considered to be due to the formation of oil film on the combustion chamber wall, the ignition point is almost at TDC but the engine runs smoothly. The maximum rate of pressure increase is 0.24 MPa·°CA^{-1} in scheme I, and 0.26 MPa·°CA^{-1} in scheme II, both less than 0.34 MPa·°CA^{-1} in the original scheme. The maximum pressure decreases in scheme I, does not increase obviously in scheme II with high compression ratio, because the amount of heat release before TDC decreases.

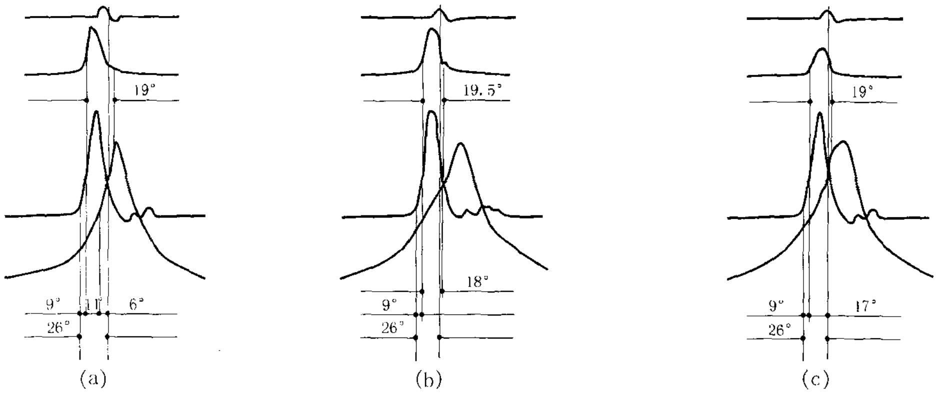

Fig.2 Pressuregraph of three schemes(horizontal ordinate scale mm: °CA)
(a) original scheme (1:3.3); (b) scheme I (1:3); (c) scheme II (1:3.75)

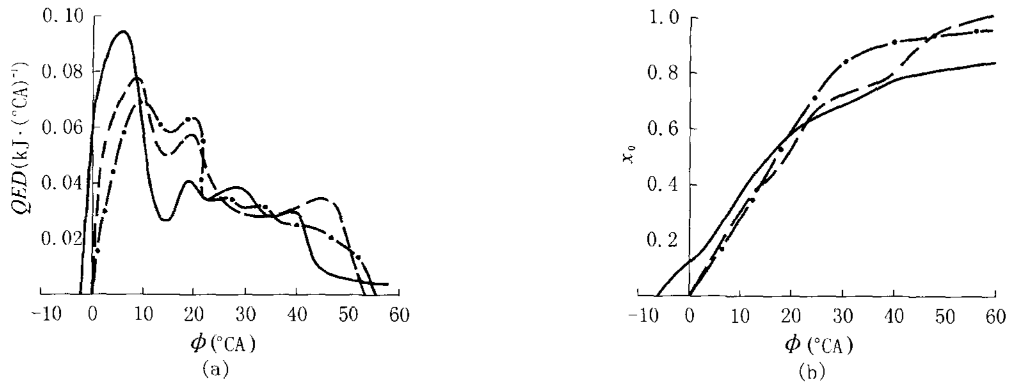

Fig.3 Rates of heat release and combustion percentage
(a) rates of release; (b) combustion percentage
— original scheme; —·— scheme I; - - - scheme II

Table 2 Combustion characteristics data of three schemes using a 1 - 110 test engine

Schemes	Injection start(°CA)	Injection duration(°CA)	Ignition delay period(°CA)	Injection angle after ignition(°CA)
Original	-17	19.0	11	8.0
I	-17	19.5	18	1.5
II	-17	19.0	17	2.0

Ignition start(°CA)	Combustion stage (x_0 exceeds 90%)(°CA)	P_z(MPa)	g_i(g·kW^{-1}·h^{-1})	Smoke R(Bosch)
-6	60	6.1	213	1.3
1	43	5,8	209	0.9
0	42	6.2	203	1.2

All fuels per cycle are injected into the cylinder in the longer period of ignition delay so that there is longer time for the air and fuel to mix perfectly.

Delay of ignition up to TDC accelerates premix combustion. The duration of combustion is shorter in both scheme I and II. Combustion percentage x_0 in scheme I and II is about above

90% at 40° CA after TDC, which contrasts sharply with that of the original scheme (in which x_0 is about 86% at 60° CA after TDC) Fig.3 shows that the rate of heat release in the original scheme is greater than that in scheme I and II in the first stage of combustion, but that soon after TDC, the rates of heat release in scheme I and II exceed that in the original scheme. The figure shows that the rates of heat release in schemes I and II were in the form of a near isosceles triangle.

Fig.4 shows the load characteristic. First, the maximum pressure P_z is higher in the course of load change in the original scheme because of the earlier ignition point and the large amount of combustion in the start stage. In contrast the maximum pressure P_z is the lowest in scheme I with longer ignition delay. Although the compression ratio increases in scheme II, its maximum pressure does not increase obviously. Second, we can notice that the smoke is improved in scheme I because of the longer ignition delay and premix combustion. In scheme II, smoke does not become worse in spite of the higher compression ratio. Lastly, the fuel consumption in scheme I, especially in scheme II in which it is lower than that of the original scheme by 10 g·kW^{-1}·h^{-1}, is significantly improved. The results also clearly show the advantage of premix combustion.

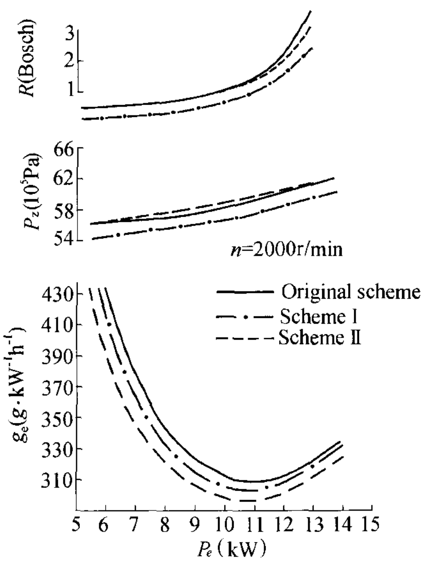

Fig.4 Load characteristics of 1 - 110 test engine

EXPERIMENTAL STUDY ON CONICAL SPRAY TYPE COMBUSTION SYSTEM

The combustion system consists of conical spray type injector and squish lip type combustion chamber.

Fig. 5 shows the structure of the injector. High injection rate, good spray quality, uniform peripheral distribution and short injection duration are its favorable characteristics for premix combustion.

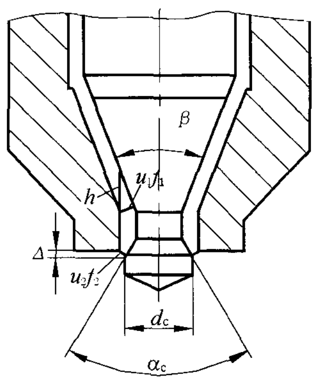

Fig.5 Conical spray type injector

Experiments were performed with a 1 - 135 single cylinder engine. The features of the engine are given in Table 3.

The engine used an ω type combustion chamber and 4 holes injector, we call the experiment with it the original scheme. The two other schemes used conical spray type injector as well as squish lip type chamber, compression ratios of 16.5 and 18.1 respectively, we still call the former scheme I, the latter scheme II. Table 4 shows some experimental results of the three schemes for n = 1500 r·min^{-1}, P_1 = 14.7 kW.

Table 3 Features of 1 - 135 single cylinder test engine

Stroke	Diameter × stroke	Compression ratio	Power	Rotary velocity	Combustion type
4	135 × 140 mm	16.5	14.7 kW	1 500 r·min^{-1}	Diffusion combustion

Table 4 Combustion characteristic data of three schemes in 1 – 135 test engine

Schemes	Combustion chamber	Injector	Injection start(°CA)	Injection duration(°CA)	Ignition delay period(°CA)
Original	*w* type	4 holes	– 21	26	13.7
I	Squish lip type	Conical spray type	– 18	17	14.0
II	Squish lip type	Conical spray type	– 13	15	10.5

Ignition delay period / Injection duration	Injection angle after ignition(°CA)	Ignition start(°CA)	P_z(MPa)	g_e($g\cdot kW^{-1}\cdot h^{-1}$)	Smoke *R*(Bosch)
0.53	12.3	-7.3	7.75	269	2.6
0.82	3.0	-4.0	7.14	263	0.7
0.70	4.5	-2.5	7.34	257	2.8

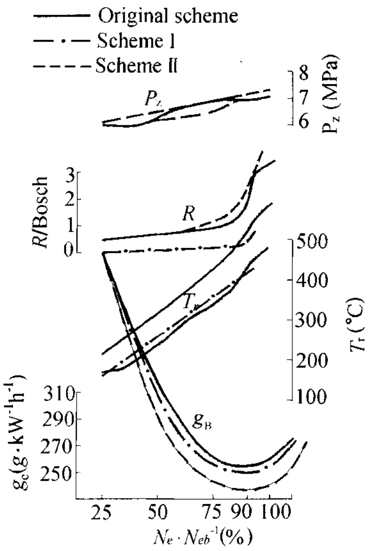

Fig.6 Load characteristics of 1 – 135 test engine

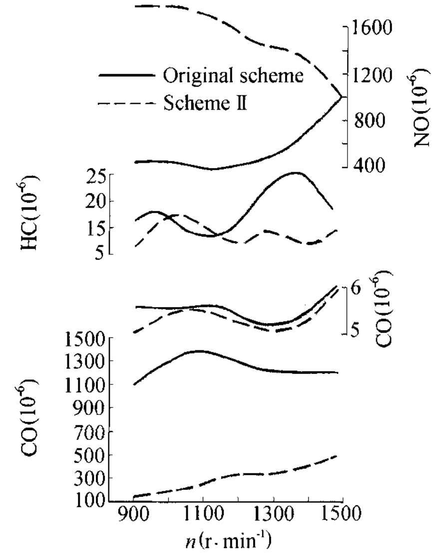

Fig.7 Emission characteristics of 1 – 135 test engine

Fig.6 shows load characteristic for n = 1 500 $r\cdot min^{-1}$, and Fig.7 shows emission characteristics.

From the diagrams and table, we notice that the conical spray type combustion system has shorter injection duration; almost all the fuel oil is injected into the cylinder in the ignition delay period so that the fuel oil mixes with air satisfactorily and the goal of premix combustion is achieved. As a result, the performances of the conical spray type combustion system are much superior to that of the multiple holes spray type combustion system of the original scheme.

Larger amount of fuel injected into the cylinder in the ignition delay period does not necessarily result in higher pressure p_z, because the ignition is delayed towards TDC, so that the amount of heal release before TDC is smaller, especially, maximum pressure p_z in scheme II decreases even with higher compression ratio.

The fuel economy is better in the two conical spray type combustion systems for n = 1 500 $r\cdot min^{-1}$, and P_e = 14.7kW, the fuel consumption in scheme II is less than that in the original scheme by 12 $g\cdot kW^{-1}\cdot h^{-1}$. In addition, smoke is largely decreased in scheme I, smoke does not worsen even with higher compression ratio.

The proportion of premix combustion in the two conical spray type combustion systems is large, the combustion system has high air utilization, its CO emission is less than that of the original scheme. On the other hand, the temperature of the conical spray type combustion system is higher; its NO_x emission is higher than that of the original scheme; but, with increase of rotary velocity, the difference between them becomes less. At lower rotary velocity of the engine, the quality of the spray of the multiple holes injector worsens, and the poor mixture of fuel and air lowers the combustion temperature, and as a result, its NO_x emission is less.

CONCLUSIONS

This paper shows that premix combustion at the isosceles triangle type rate of heat release not only has advantages in theory, but is also feasible in practice.

The squish lip type combustion chamber facilitates formation of a thin oil film helpful for delaying ignition. The conical spray type injector is useful for rapidly injecting fuel into the cylinder.

Experimental studies of two single cylinder diesel engines showed that premix combustion at isosceles triangle type rate of heat release results in longer ignition delay period, larger amount of fuel injected into the cylinder during the ignition delay period, lower maximum pressure, better fuel economy and less exhaust emission. These performances are attributed to efficient mixture of air and fuel during the longer ignition delay period and reasonable delay of ignition toward TDC.

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