Effects of Combustion Chamber Specifications and Swirl Ratio on Transient Heat Transfer and Combustion in a DI Diesel Engine

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ABSTRACT

Experimental investigations are being conducted with a single-cylinder direct injection diesel engine to examine the effects of combustion chamber specifications, swirl ratios and injection timings on the heat release and the transient heat transfer characteristics. Heat rejection was examined on the basis of heat release calculations made from cylinder pressure time histories. Transient surface temperature data obtained from four stationary locations in the piston were used as the basis for determining the transient heat flux rates. Transient surface temperature was measured by the thin film thermocouple constructed on the same material with the piston. The results showed a good agreement with the heat rejection calculated from cylinder pressure data and that transient heat flux in a piston cavity was reduced with a the larger cavity diameter and a higher swirl ratio. On the other hand , a transient heat flux in a piston head was not changed by a cavity diameter and swirl ratio and was significantly influenced by injection timing. Based on these results, the present paper discusses the mechanism of heat rejection in a direct injection diesel engine.

INTRODUCTION

Recently, diesel engines have been persistently required to reduce fuel consumption and emissions. To improve the thermal efficiency, many researchers have been making studies. Research of low heat rejection (LHR) diesel engines is one of them. Much work has been done at many research institutes to examine the potential of LHR diesel engines for reducing heat rejection and achieving high thermal efficiency. While several researchers have reported improvements in thermal efficiency [1~6] with a LHR engine, others have noted that it declined [7~9].

The results of our previous studies[10] on LHR diesel engines showed that heat rejection was reduced by insulating the combustion chamber. However, because combustion deteriorated, it was not possible to obtain an improvement in thermal efficiency corresponding to the reduction in heat rejection. Also our results showed that heat rejection was significantly influenced by the combustion chamber specifications and swirl ratio.

This paper discusses the effects of swirl ratio, cavity diameter and injection timing on the transient heat transfer for the piston cavity and piston head of a diesel engine. It also describes the relationship between the transient heat transfer and heat rejection that was determined by cylinder pressure.

Table1 Engine specifications

Engine Type	Single Cylinder
Combustion System	Direct Injection
Bore	85mm
Stroke	86mm
Displacement	0.488 L
Compression Ratio	18:1
Injection Pump	Bosch VE

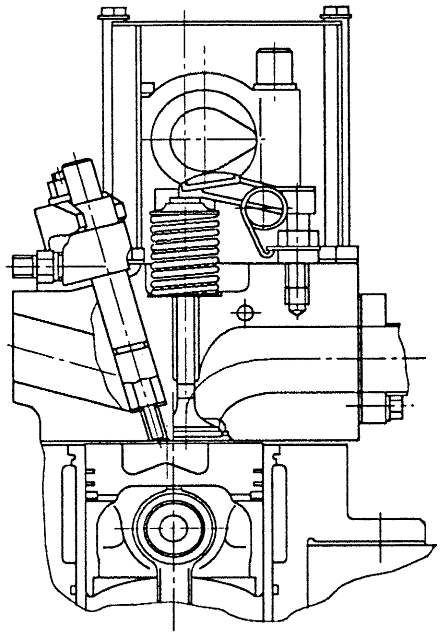

Fig. 1 Cross section of the experimental engine

EXPERIMENTS

Experimental apparatus and procedure

The test engine used in this work was a single-cylinder DI diesel having the specifications shown in Table 1. A schematic of the engine combustion chamber is shown in Fig. 1. Cylinder pressure data was obtained with a water-cooled piezoelectric pressure transducer inserted in the pressure port of the cylinder head. The pressure data was recorded at 0.25 crank-angle degree intervals for 400 consecutive engine cycles.

Engine tests were conducted using four types of aluminum-alloy pistons. The specifications of the pistons are given in Fig. 2. The tests were performed at 1200 rpm with a constant fuel rate and air flow rate. The parameters that were varied included the swirl ratio and injection timing.

For all operating conditions, the lubricating oil inlet temperature and cooling water temperature were maintained at 80±1.0 ℃.

Heat flux measurement

Figure 3 shows the construction of thin film thermocouple used for the transient heat flux measurement. The body was made of the same material as that of the piston. The hot junction was formed by the copper thin film and the constantan wire that had dielectric film and set by pressing into body. Another thermocouple was placed at 3.2mm from the surface to compose the cold junction and record the back side temperature. The locations of four thermocouples on the piston are shown in Fig. 4. The instantaneous temperature data was recorded at 0.25 crank-angle degree intervals for 400 consecutive engine cycles.

DEFINITION OF TWO CHARACTERISTICS

The following definitions of two characteristics were applied in this work in order to make comparisons of heat rejection and combustion conditions.

Fraction of apparent heat release (Qa/Qf)

This characteristic was defined for the purpose of assessing heat rejection. The fraction of apparent heat release was defined as the quotient of dividing the heat release during combustion (Qa) by the input fuel energy (Qf). It is seen from the following relational equations that an increase in the fraction of apparent heat release means that heat rejection (Qc) has been reduced.

$$Qf = Qa + Qc \quad (1)$$

$$1 = Qa/Qf + Qc/Qf \quad (1')$$

Work conversion efficiency (Qw/Qa)

This efficiency shows the proportion of heat release that is converted to indicated work (Qw). It is used in judging the quality of combustion. The relational expressions are given below.

Type	A(mm)	B(mm)	C(mm)	D(mm)	Vol(cm³)
A39	39	—	11.3	17.7	19.3
A42	42	—	8.6	15.6	
A45	45	—	6.3	14.4	
A42L35	42	35	9.3	16.3	19.3

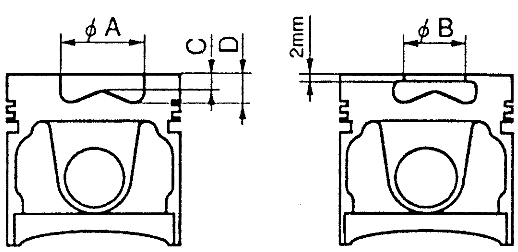

Fig.2 Piston specifications

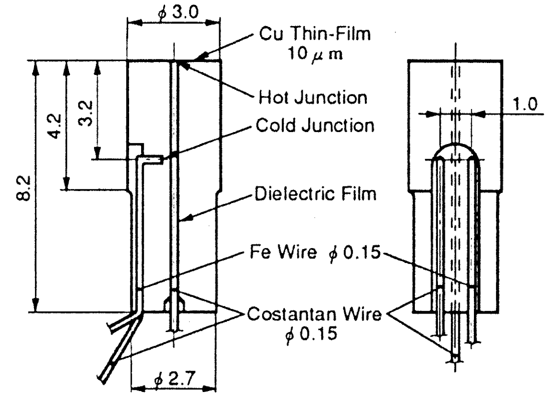

Fig.3 Construction of thin-film thermocouple

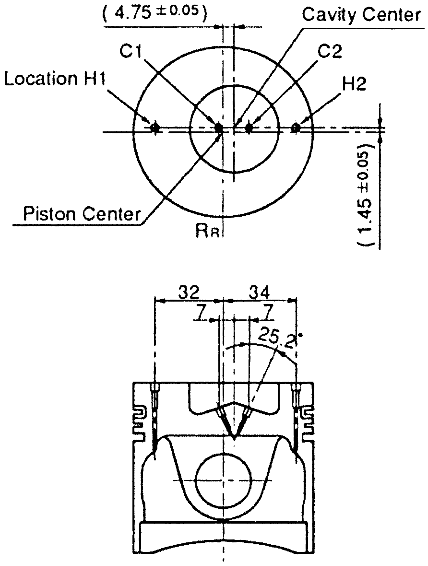

Fig.4 Location of thin-film thermocouples in the piston

$$Qa = Qw + Qe \qquad (2)$$

$$1 = Qw/Qa + Qe/Qa \qquad (2')$$

Using these two equations of efficiency, the indicated thermal efficiency (η i) may be given by

$$\eta\, i = Qw/Qf = Qa/Qf \times Qw/Qa \qquad (3)$$

EXPERIMENTAL RESULTS AND DISCUSSION

Relation between the fraction of apparent heat release and parameters

Figure 5 shows the fraction of apparent heat release (Qa/Qf), work conversion efficiency (Qw/Qa) and indicated efficiency (η i) as the functions of cavity diameter, swirl ratio and injection timing. With the larger cavity diameter, the fraction of apparent heat release significantly improved while the work conversion efficiency deteriorated. As a result of this, the indicated efficiency improved. With a larger swirl ratio, the fraction of apparent heat release and the work conversion efficiency improved, resulting in a significant increase in the indicated efficiency. As the injection timing is retarded, the fraction of apparent heat release improved up to 4° BTDC and slightly deteriorated after then. On the other hand, the work conversion efficiency showed an improvement with an advanced injection timing.

Figure 6 compares the apparent heat release pattern, the maximum cylinder pressure(Pmax) and NOx level in relation to changes in the cavity diameter and the swirl ratio. The heat release pattern results show that with a smaller cavity diameter and a larger swirl ratio, which worked to improve the work conversion efficiency, the initial diffusion combustion became more active, thereby reducing the combustion period. At the same time, Pmax and NOx level increased, indicating a better combustion. Ordinarily, an increase in Pmax results in a larger heat transfer coefficient and an increased heat rejection. The results of a smaller cavity diameter agree with the forgoing phase, however a larger swirl ratio leads to an increase in Pmax and a reduction in heat rejection. To investigate the effects of the combustion chamber specifications and the swirl ratio on the heat rejection, the heat flux in the piston was studied.

Effects of cavity diameter on heat flux

Figure 7 shows the instantaneous heat flux in the piston cavity(C2) and the piston head(H2) as the functions of the cavity diameter and the injection timing. For the 7° BTDC injection timing, the heat flux result at the location C2 (in the piston cavity) with a larger cavity diameter shows the reduction in the heat flux peak and the increase in the heat flux after 30° ATDC. These results agree with the results of heat release rate. At the location H2 (on the piston head), heat flux is not influenced by the cavity diameter. For 2° ATDC injection timing, the heat flux results at the both locations are not changed by the cavity diameter. On the other hand, the heat flux at the compression decreases with the larger cavity diameter under all conditions. The reason for this is estimated to be the effect of the gas flow that is less active with a larger cavity diameter.

Figure 8 compares the mean heat flux during combustion of two locations in relation to changes in the cavity diameter as a function of the injection timing. The mean heat flux of the location C2 decreases with the larger cavity diameter which worked to improve the fraction of apparent heat release calculated from the cylinder pressure data. However, the mean heat flux of the location H2 is not influenced by the cavity diameter.

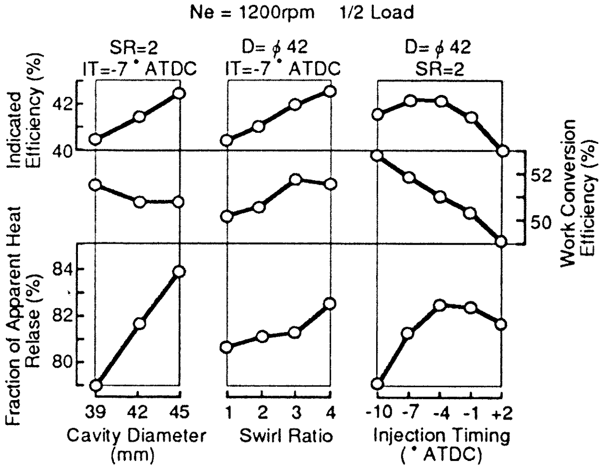

Fig.5 Effect of cavity diameter, swirl ratio and injection timing on thermal efficiency

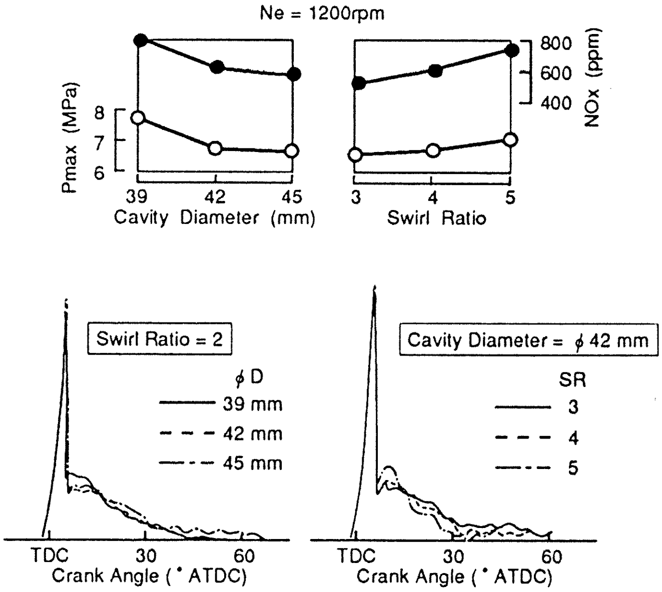

Fig. 6 Comparisons of heat release pattern, Pmax and NOx level

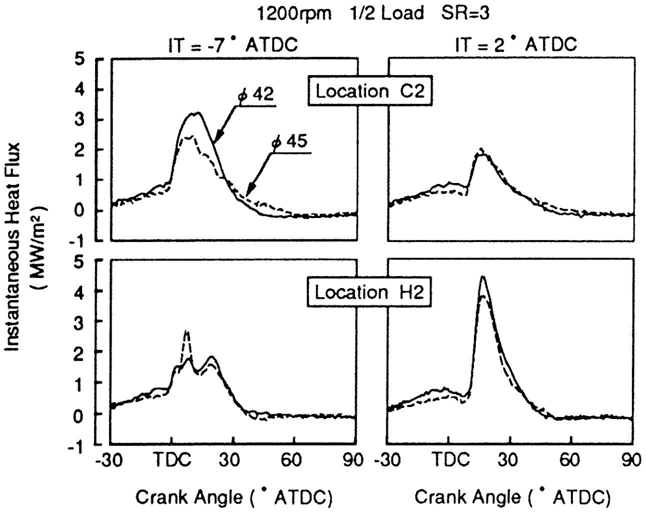

Fig. 7 Effect of cavity diameter on heat flux

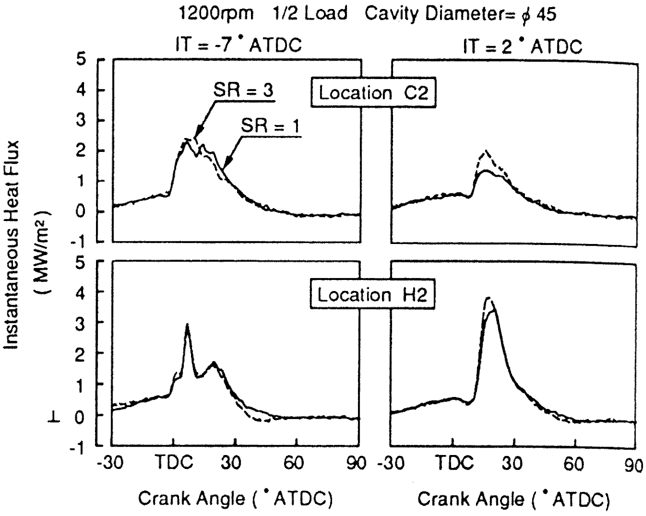

Fig. 9 Effect of swirl ratio on heat flux

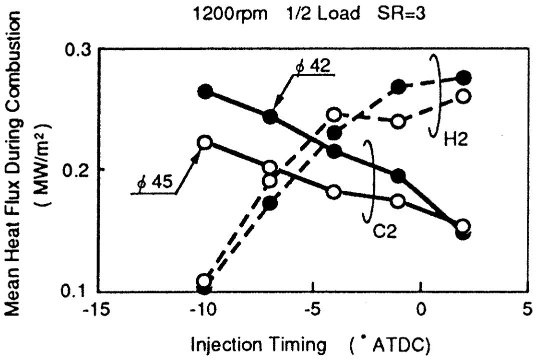

Fig. 8 Mean heat flux during combustion as a function of injection timing in ϕ42 and ϕ45 cavity diameter

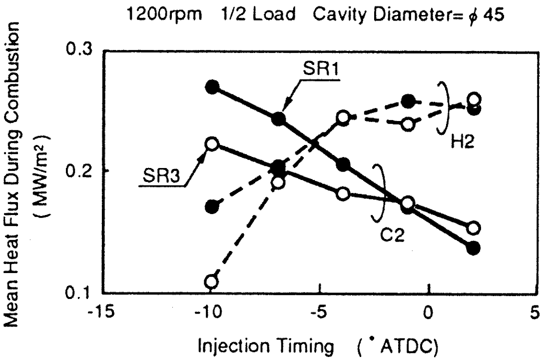

Fig. 10 Mean heat flux during combustion as a function of injection timing in SR1 and SR3

Effects of swirl ratio on heat flux

Figure 9 shows the instantaneous heat flux in the piston cavity(C2) and the piston head(H2) as the functions of the swirl ratio and injection timing. For the 7° BTDC injection timing, the heat flux result at the location C2 (in the piston cavity) with a larger swirl ratio shows the increase in the heat flux peak, however the reduction in the heat flux after 10° ATDC. At the location H2 (on the piston head), heat flux peak is not influenced by the swirl ratio, however the heat flux of the combustion end decreases with a larger swirl ratio. For 2° ATDC injection timing, the heat flux with a larger swirl ratio shows the increase in the heat flux peak at the both locations .

Figure 10 compares the mean heat flux during combustion of two locations in relation to changes in the swirl ratio as a function of the injection timing. It is seen that the mean heat flux of the location C2 decreases with the larger swirl ratio and the mean heat flux of the location H2 is not influenced by the swirl ratio.

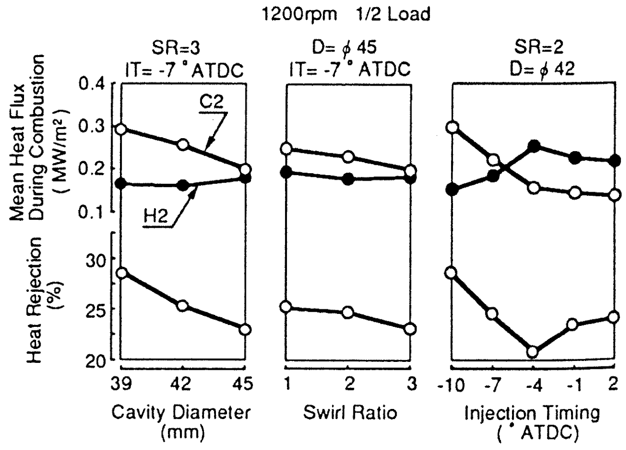

Fig.11 Effect of cavity diameter, swirl ratio and injection timing on heat rejection and mean heat flux of the piston

Figure 11 shows the mean heat flux during combustion of two locations and the heat rejection, which was calculated from the cylinder pressure data, as the functions of the cavity diameter, the swirl ratio and the injection timing. In view of these results, it was concluded that the effects of the cavity diameter and swirl ratio on the heat flux in the piston cavity corresponded with the effects of them on the heat rejection determined by the cylinder pressure data. However, the heat flux on the piston head is not influenced by the cavity diameter and swirl ratio and is influenced by the injection timing. It is estimated that the effects of the injection timing on the heat flux on the piston head is related to the blow-off of the flame from the cavity as a result of reverse-squish due to the piston fall. Consequently, an investigation was made of the effects of the squish lip , which holds the flame in the cavity, on the heat flux of the piston head.

Effect of the squish lip

The fraction of apparent heat release, the work conversion efficiency, indicated efficiency, Pmax and NOx level which are all obtained with the squish-lipped and no squish-lipped pistons that have the same cavity diameter are compared in Fig. 12. The piston with a squish lip shows an improvement in the work conversion efficiency and a reduction of the fraction of the apparent heat release. As these results show, the indicated efficiency with squish lip declines due to the significant increase in heat rejection. In view of the results of Pmax , NOx level and the apparent heat release rate, it is concluded that the combustion with a squish lip becomes active due to the increased turbulence energy. Figure 13 shows the effects of a squish lip on the heat flux. In the piston cavity, a squish lip increases in the heat flux peak and the heat flux at the compression, while reduces the heat flux at the combustion end. The heat flux at the piston head is not influenced by a squish lip.

Figure 14 compares the mean heat flux during combustion between the pistons with and without a squish lip as a function of the injection timing. Irrespective of the measurement location, the mean heat flux with a squish lip increases. On the other hand, the mean heat flux of the piston head becomes larger with later injection timing. In view of these results, the heat flux of the piston head for later injection timing is not influenced by a squish lip and increases due to reverse squish.

Based on these results, the combustion chamber specifications and the swirl ratio changes the total heat rejection calculated by the cylinder pressure data and the heat flux of the piston cavity, however does not influence the heat flux of the piston head. The heat flux of piston head is influenced by the injection timing. The later injection timing reduces the heat flux of the piston cavity and increases the heat flux of piston head. The smaller cavity diameter and a squish lip improve the work combustion efficiency and increase in the heat rejection due to the larger turbulence energy. However, the larger swirl ratio improves the work conversion efficiency and reduces the heat rejection. Thus,

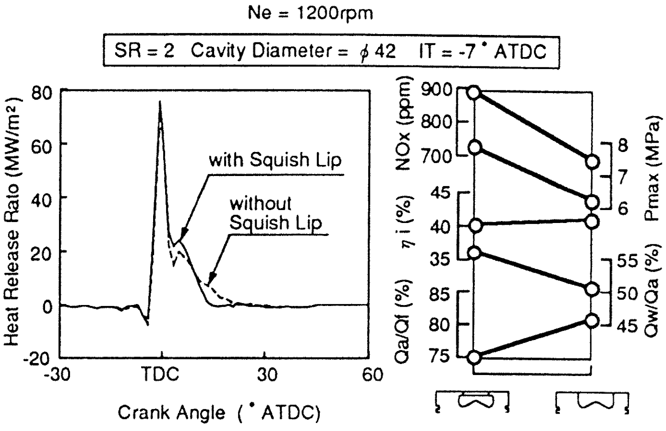

Fig. 12 Comparisons of heat release pattern, Pmax and NOx between with and without squish lip

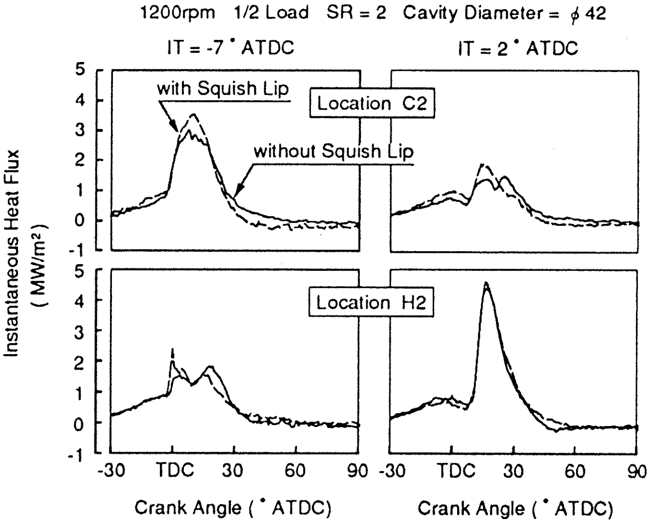

Fig.13 Effect of squish lip on heat flux

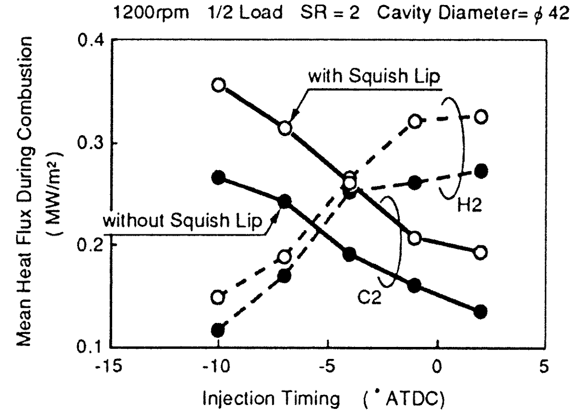

Fig. 14 Mean heat flux during combustion as a function of injection timing in with and without squish lip

the work conversion efficiency has differing effects on the heat rejection depending on whether it varies as a result of changes in the swirl ratio or changes in the combustion chamber specifications. Based on the results that the heat flux of the piston cavity with a larger swirl ratio decreased, it can be conjectured that it is related to the formation of a heat insulating layer of air between the cavity wall and the flame, which is concentrated in the center as result of greater centrifugal force due to the larger swirl ratio.

CONCLUSIONS

The effects of the cavity diameter, the swirl ratio and the injection timing were investigated experimentally with the measurement of transient heat flux and results were compared with the two defined characteristics that were calculated from cylinder pressure data. One characteristic was the fraction of apparent heat release which was defined using heat release values found from indicated pressure diagram. The other characteristic was work conversion efficiency, which showed how much thermal energy was converted to indicated work. The results of the present study are summarized below.

(1) In the range of the experimental conditions conducted in this work, a larger cavity diameter reduces heat rejection and declines work conversion efficiency.

(2) A larger swirl ratio reduces heat rejection and improves work conversion efficiency.

(3) Heat flux in the piston cavity corresponds with the result of heat rejection that is reduced by a larger cavity diameter and a larger swirl ratio.

(4) Heat flux at the piston head does not change with the cavity diameter and the swirl ratio and relates to the combustion timing.

NOMENCLATURE

Qf: Input fuel energy per cycle, kJ/cycle
Qa: apparent heat release per cycle, kJ/cycle
Qc: heat rejection during combustion per cycle, kJ/cycle
Qw: indicated work per cycle, kJ/cycle
Qe: heat loss in the exhaust gas, heat rejection during compression cycle, etc., kJ/cycle
Qa/Qf: fraction of apparent heat release
Qw/Qa: work conversion efficiency

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