

Effect of Piston Speed around Top Dead Center on Thermal Efficiency

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ABSTRACT

In a reciprocating internal combustion engine, the Otto-cycle allows for maximum thermal efficiency with a same compression ratio, which means complete combustion in a moment at top dead center, which is practically impossible. In view of this fact, it can be considered that a slower transition of the piston speed near top dead center would promote combustion during that period. This, in turn, would increase the in-cylinder pressure and degree of constant volume thereby improving thermal efficiency. To prove this, an engine in which the piston can be displaced moderately to achieve ideal constant volume combustion was prepared and experiments were carried out. As anticipated, while the degree of constant volume increased, the thermal efficiency did not improve due to increased heat loss. Subsequently, further experiments were carried out using a direct injection, stratified charge engine which allows selective reduction of heat loss, and a high thermal efficiency was attained.

1. Introduction

Quick burn is an effective method of increasing the degree of constant volume of an internal combustion engine in order to improve the thermal efficiency. Conventional methods of achieving this involve the use of swirling or turbulence in the cylinder to increase the combustion speed⁽¹⁾, the use of a more compact combustion chamber⁽²⁾, and/or multiple spark plugs⁽³⁾ to reduce the flame propagation distance. In recent years, attention has been focused on the fact that homogeneous charge compression ignition (HCCI) systems are also effective in increasing the thermal efficiency using similar methods^{(4)–(7)}. All of these methods attempt to reduce the time required for combustion to occur, in order to approach the ideal Otto cycle and increase the area enclosed by the P-V diagram. Also, attempts to structurally increase the area of the P-V diagrams have been reported by Miwa et al.⁽⁸⁾.

Here, the authors produced two kinds of prototype engines which employed a conventional crankshaft and connecting rod mechanism. Tests were then performed to see whether or not it was possible to increase the thermal efficiency by slowing the piston speed near top dead center, advancing the combustion during this period, and increasing the degree of constant volume. Also, to compensate for the disadvantages of these engines, a third prototype, direct injection-type engine was subjected to the same test. The results of these tests are described here after.

2. Conventional Engine

In a reciprocating internal combustion engine, the piston decelerates and then stops momentarily at both top and bottom dead center. Because the volume of the combustion chamber stops

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changing at these points, this stop of the piston has the important dual role of completing combustion at top dead center and replacing the gas in the cylinder via the intake and exhaust valves at both top and bottom dead center. In the type of internal combustion engine that is most generally used these days, the piston behavior near top dead center differs from that near bottom dead center. As shown in Fig. 1, it is well-known that the piston displacement when the crankshaft turns through a certain angle is greater at top dead center than at bottom dead center.

Accordingly, it was suggested that if it were possible to make the piston displacement at top dead center equal to that at bottom dead center, the piston displacement in the vicinity of new top dead center would decrease and cause combustion to complete before the volume of the combustion chamber had changed significantly. The subsequent increase in the degree of constant volume would improve the thermal efficiency. This point was verified experimentally.

3. Prototype Engine

3.1. ICVC Engine

3.1.1. ICVC engine specifications

To address the above issue, a prototype engine was prepared by modifying a 4-stroke, air-cooled single cylinder 125 cm³ motorcycle gasoline engine as described below. As shown in Fig. 2, at the bottom of the normal piston is a sub-piston that slides together

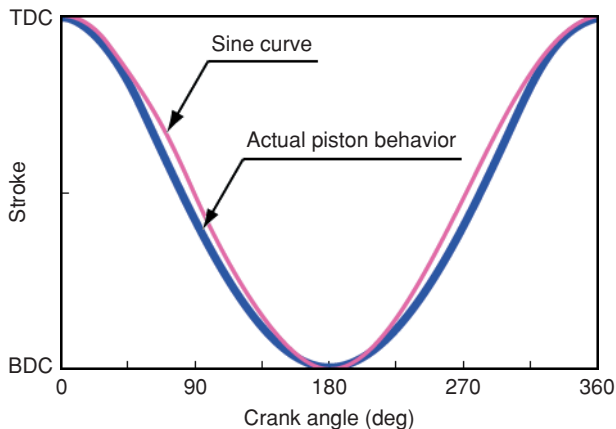


Fig. 1 Actual piston behavior against crank angle

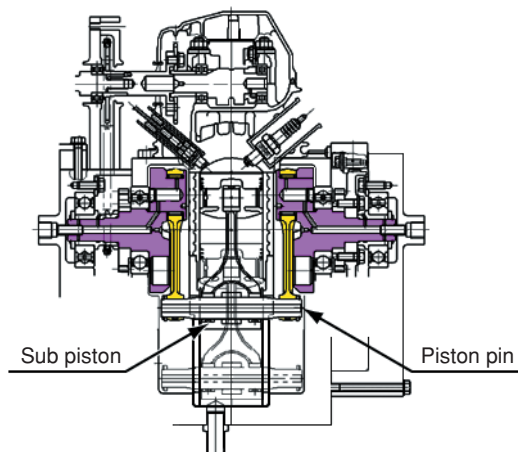


Fig. 2 Ideal constant volume combustion engine

with the normal piston inside the same cylinder. This sub-piston has a long piston pin that passes through the cylinder wall. On both ends of the piston pin are installed the small ends of a pair of connecting rods that can rock freely. The big end of each connecting rod is connected to a crankpin of the divided two pieces of cantilever-type crankshaft. When gas pressure acts on the piston, the connecting rods pull the respective crankpins and rotational force is applied to the crankshaft. The engine was constructed in this way so that combustion would take place at a percentage volume change of the combustion chamber equivalent to that at normal bottom dead center. This type of engine is called an Ideal Constant Volume Combustion (ICVC) engine^{(9), (10)}.

Table 1 shows the main specifications of this engine. The piston, cylinder, cylinder head, connecting rod and some other parts are the same as those used in the base engine. The engine was designed exclusively so that combustion takes place at a position equivalent to bottom dead center of the base engine.

3.1.2. ICVC engine test results

The test was carried out at an engine speed of 3 000 r/min, a partial load resulting in a fuel consumption of 640 g/h, an ignition timing equal to MBT, and stoichiometric air-fuel ratio. The performance of the prototype engine was compared with that of the base engine. Figure 3 shows the results expressed using a P-V diagram. It can be seen that in the case of the ICVC engine, the degree of constant volume has increased, and the P-V diagram is clearly close to the ideal Otto cycle.

Table 1 ICVC engine specifications

Items	Base engine	ICVC engine
Engine type	4-stroke, air cooled, single cylinder	
Bore x Stroke (mm)	52.4 x 57.8	
Displacement (cm ³)	124.6	
Compression ratio	9.35 : 1	9.32 : 1
Connecting rod length (mm)	93.5	

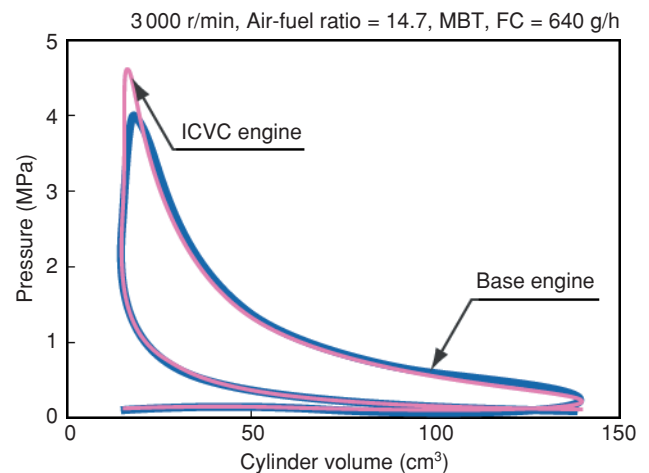


Fig. 3 P-V diagram comparison

Table 2 Test results of ICVC engine

3000 r/min, FC = 640 g/h, Air-fuel ratio = 14.7, θ_{ig} = MBT

Engine	P _{max} (MPa)	Deg. of C.V.	P _{mi} (kPa)	ISFC (g/kWh)
ICVC engine	4.64	0.97	846	243
Base engine	4.02	0.94	893	230

Deg. of C.V. : Degree of constant volume

Table 2 shows the results of calculating the area of the P-V diagram and obtaining the indicated work. It can be seen from the table that the degree of constant volume increased, but despite the fact that the maximum pressure increased by about 15%, the pressure during the expansion stroke fell, and the indicated mean effective pressure fell by 5% causing the thermal efficiency to fall.

3.2. APC Engine

3.2.1. APC engine specifications

To investigate in detail why the thermal efficiency did not improve despite the fact that the degree of constant volume increased, the engine shown in Fig. 4, called an Active Piston Control (APC)^{(9),(10)} engine, with slower piston speed near top dead center was produced.

The crankshaft of the engine has been greatly offset from the center of the cylinder, so that the gas pressure applied to the piston causes the crankshaft to be pulled via the two connecting rods A and B, in the same way as in the ICVC engine, causing the crankshaft to rotate. A control rod is connected to regulate the position where connecting rods A and B are joined together, so that the center of the cylinder does not largely offset from connecting rod A.

As shown in the figure, the engine uses a multiple link mechanism in which the length of the compression stroke differs from that of the expansion stroke. When the crankshaft is rotating counterclockwise, the crank angle required for the intake and expansion stroke is 213 degrees, and that for the compression and exhaust stroke is 147 degrees.

The displacement of the piston of the APC engine is shown in Fig. 5 and the main specifications are shown in Table 3. The engine is a water-cooled, single cylinder gasoline engine that has a 195 cm³ displacement. The compression ratio is 11.5:1, two spark plugs are installed, and a helical intake port is provided, realizing a minimal indicated fuel consumption of 220 g/kWh (at 2 700 r/min, and BMEP300 kPa) during its original operation.

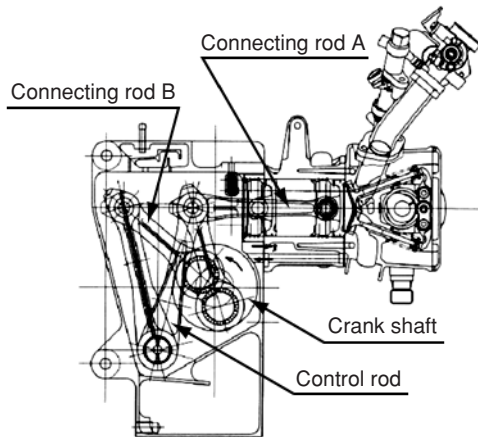


Fig. 4 Active piston control engine

This engine has a camshaft that enables it to be run in reverse. The oil pump and cooling water pump are designed so that the specified quantities of oil and water can be supplied externally at the specified temperature.

3.2.2. APC engine test results

Figure 6 is a P-V diagram which shows a comparison between the results of a test performed on the APC with those of the original engine, for an engine speed of 3 000 r/min, a volumetric efficiency of 34%, a stoichiometric air-fuel ratio, and MBT ignition timing. The crankshaft of the APC engine uses a multiple link mechanism in which the crankshaft is greatly offset from the center of the cylinder. Because an extremely large load acts on each link, the elastic deformation is large and it was found that the crank angle

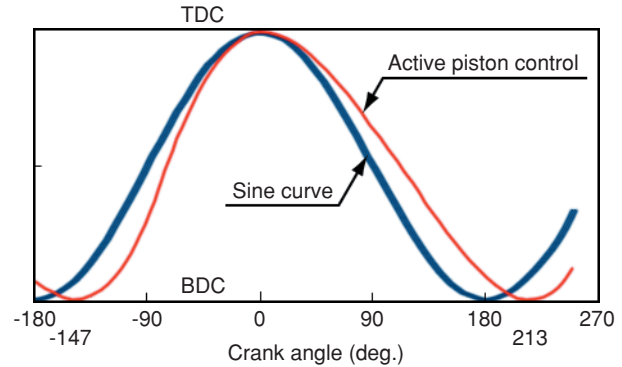


Fig. 5 Piston behavior of APC engine

Table 3 Specifications of APC engine

Engine type	4-stroke, water cooled	
Bore x Stroke (mm)	59 x 71.3	
Displacement (cm ³)	195	
Compression ratio	11.5 : 1	
Spark plug	Twin plug	

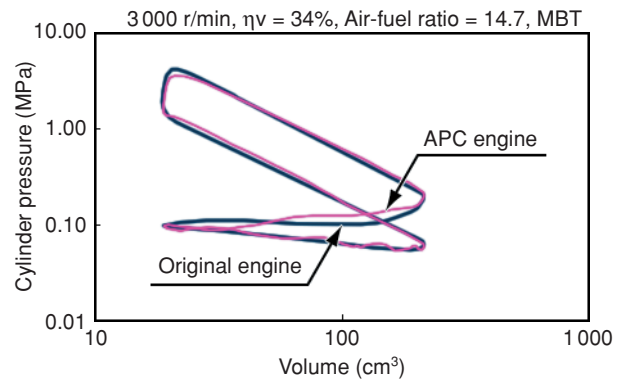


Fig. 6 P-V diagram comparison between APC engine and original engine

Table 4 Test results of APC engine

3 000 r/min, $\eta_v = 34\%$, $\theta_{ig} = \text{MBT}$, Air-fuel ratio = 14.7

Engines	Fuel consumption (g/h)	Exhaust gas		Cooling water (out-in)	
		Temperature (K)	Exhaust loss (kW)	Temperature (K)	Heat loss (kW)
APC engine	490	901	2.40	6.0	1.55
Original engine	494	920	2.47	4.8	1.24

does not determine the piston displacement (combustion chamber volume). For this reason, as shown in Fig. 6, the degree of constant volume of the APC engine did not increase as in the case of the ICVC engine, resulting in a reduction of P_{max} .

Accordingly, it was decided to abandon the idea of obtaining the indicated work from the P-V diagram, and instead to carry out a study from the measured temperature of each part. Table 4 shows the measured results of the fuel consumption, the exhaust gas temperature and the difference in temperature between the cooling water outlet and inlet of the engine at 3 000 r/min and $\eta_v 34\%$. This data is compared with the corresponding figures for the original engine. It can be seen from the table that in the case of the APC engine the exhaust gas temperature has fallen, whereas the water temperature rose, which indicates the amount of heat released to the cooling water is quite large.

On the other hand, it can be considered that because the supplied mixture gas is in the stoichiometric air-fuel ratio, the product of the exhaust gas temperature and the fuel consumption should be proportional to the exhaust loss amount. The results of calculating the heat balance based on the above assumption are shown in Fig. 7.

It is considered that the amount of heat released to the cooling water is equivalent to the heat loss amount, and that the main reason why the thermal efficiency of the ICVC engine did not improve, despite that fact that the degree of constant volume increased, was due to the increased heat loss.

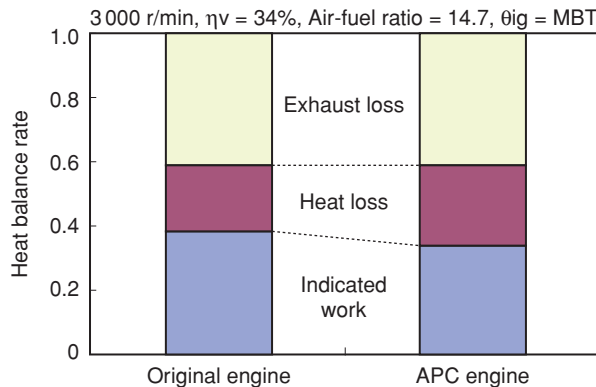


Fig. 7 Heat balance rate of test results

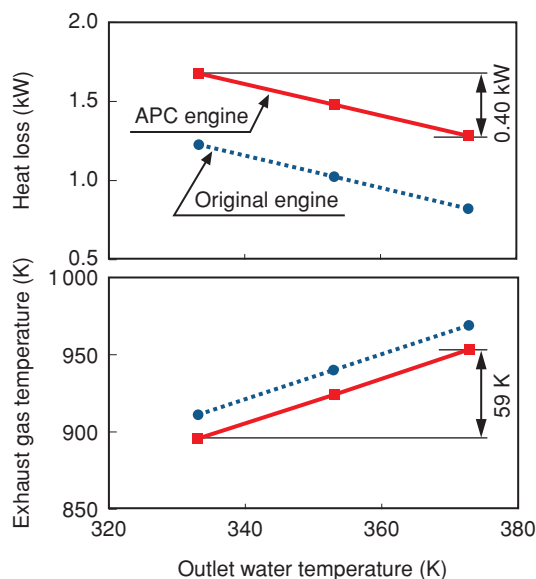


Fig. 8 Heat loss and exhaust gas temperature

To verify this, the cooling water temperature was changed from 60 °C to 100 °C. The exhaust gas temperature at this time and the work ratio obtained by multiplying the difference between the cooling water inlet and outlet temperatures by the cooling water flow rate (heat loss) are shown on the vertical axis of Fig. 8. When the cooling water temperature was raised, the heat loss was reduced by 0.4 kW and the exhaust gas temperature rose by 59 °C. When this value is converted into a work ratio, it corresponds to 0.29 kW, and about 75% of the reduced heat loss is converted into the increase in the exhaust loss. It was found that these figures were the same for both the APC engine and the original engine.

In the final analysis, it was found that reducing the piston speed in the vicinity of top dead center caused the heat loss to increase, and because the sum of the heat loss and the exhaust loss is nearly constant, only the heat loss cannot be selectively reduced.

To further study this relationship, a test was performed with the camshaft reassembled to enable the engine to rotate in reverse. Figure 9 shows the measured values of the temperature difference Δt between the cooling water inlet and outlet temperatures, and the exhaust gas temperature, when the ignition timing was changed through a 10 degree crank angle in the advance and retard directions from MBT to change the heat loss. When the APC engine is running in reverse, the crank angle of the compression stroke is 213 degrees, however the crank angle of the expansion stroke falls to 147 degrees. Consequently, it was predicted that the exhaust gas temperature, which indicates the exhaust loss, would be higher than that of the

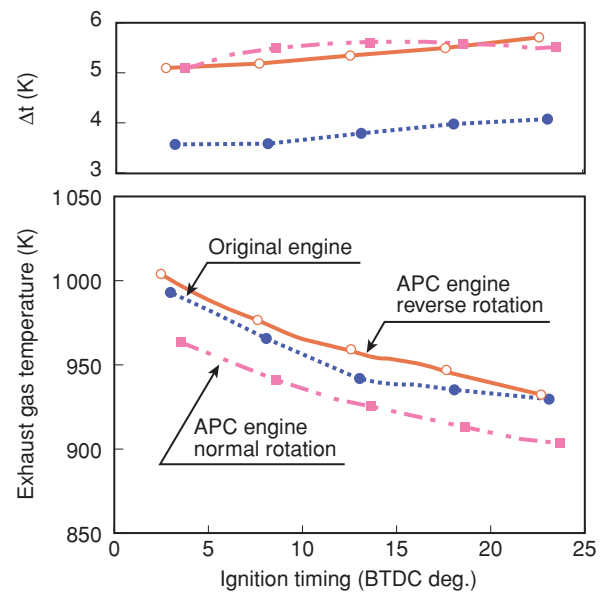


Fig. 9 Heat loss and exhaust gas temperature against ignition timing

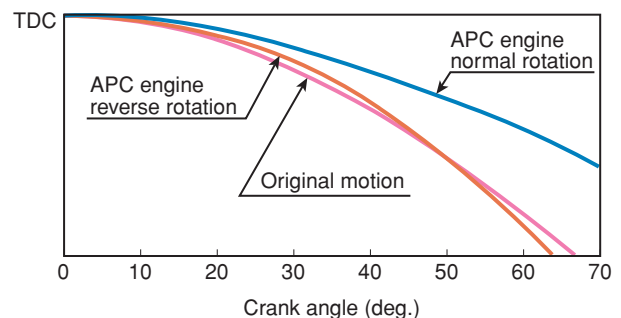


Fig. 10 Detailed figure near TDC including reverse rotation

original engine, and this result was in fact obtained. However, the cooling water inlet and outlet temperature difference was not as small as expected, and the same results were obtained during forward motion as well.

Figure 10 shows an enlarged diagram of the piston displacement in the vicinity of top dead center. In the APC engine, the piston displacement in the vicinity of a 45° crank angle after top dead center is more moderate than that of the original engine, even when the engine is rotating in the reverse direction, so it is considered that the heat lost during this interval probably flowed from the wall of the combustion chamber to the cooling water.

4. Combination with a Direct Injection Engine

4.1. $\lambda = 8$ Engine

4.1.1. $\lambda = 8$ engine specifications

The elastic deformation of each link of an APC engine is large, preventing the position of the piston with respect to the crank angle from being determined. As a result, it was not possible to obtain the indicated work from the P-V diagram. Accordingly, the conventional connecting rod and crank mechanism were used, and the piston speed in the vicinity of top dead center was varied by changing the ratio l between the connecting rod length l and the crankshaft radius r (defined as l/r). By making this value λ as large as possible, it was found that piston displacement versus the crank angle approached a sine curve. Accordingly, a test was carried out on an engine that had a λ of 8, which was considered a realistic value, and carried out an analysis from the P-V diagram. As shown in Fig. 11, the connecting rod of this engine is considerably longer than that of a conventional one.

Figure 12 shows an enlarged diagram of the behavior of the piston in the vicinity of top dead center. The λ of the base engine is 3.34 and the diagram indicates that the displacement characteristics of the test engine are mid-way between a sine curve and the displacement characteristics of the base engine.

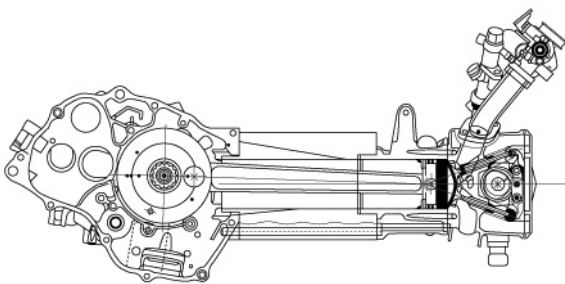


Fig. 11 $\lambda = 8$ engine

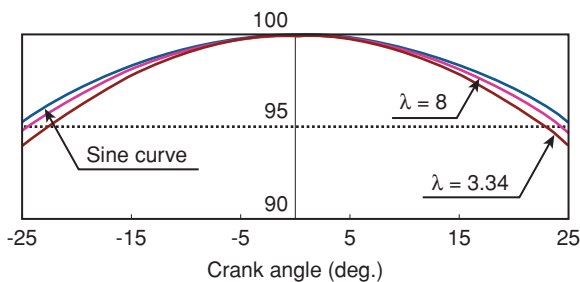


Fig. 12 Piston behavior around top dead center

The bore stroke and specifications of the combustion system are identical to those of the previously mentioned APC engine. This engine was compared with a normal premixed port injection (PI) engine and a direct injection (DI) stratified charge combustion engine. Heat loss can be selectively reduced with the DI stratified charge combustion engine.

4.1.2. $\lambda = 8$ engine test results

The test was carried out at an engine speed of 3 000 r/min and at IMEP of between 0.3 and 0.75 MPa. The results are shown in Fig. 13. Compared to the premixed port injection, the indicated specific fuel consumption of the base DI engine decreased by about 13 to 7 g/kWh. In the case of the engine whose λ is 8, the fuel consumption deteriorated by 8 g/kWh when port injection was used, but decreased by 4 to 8 g/kWh when direct injection was used. This latter case corresponds to the best indicated specific fuel consumption.

4.2. Consideration Concerning Heat Loss and Piston Behavior

The way in which the ratio between the heat loss and the exhaust loss changed was extracted using the respective center values of the data in Fig. 13 and is shown in Fig. 14 as the heat balance rate. The heat loss was obtained as mentioned previously by multiplying the cooling water inlet and outlet temperature difference by the flow rate, and the exhaust loss was obtained by subtracting the sum of the indicated work and heat loss from the input heat amount.

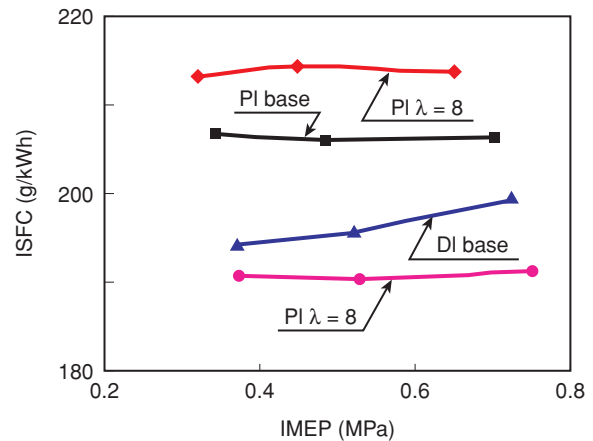


Fig. 13 ISFC comparison between PI and DI engine

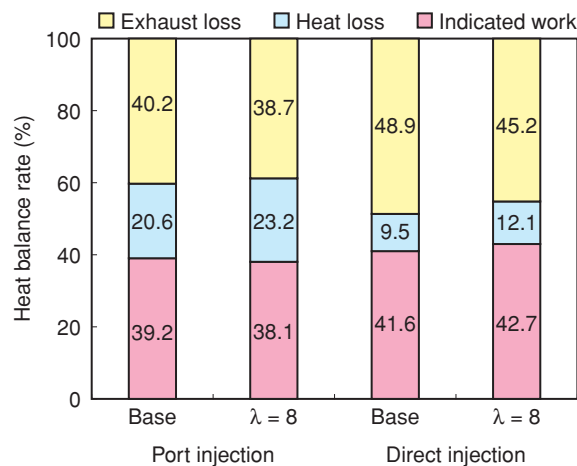


Fig. 14 Heat balance rate

An engine with a large λ had greater heat loss and a reduction in indicated work in the case of port injection. However, when direct injection was used, heat loss was low from the outset, and although the heat loss increased when the λ was raised, the exhaust loss reduction effect resulting from the increased degree of constant volume predominated, resulting in a maximum indicated work of 42.7%.

5. Conclusion

To allow the piston displacement to change moderately around top dead center and increase the degree of constant volume to improve the thermal efficiency, an ICVC engine as well as an APC engine whose compression stroke was of a different length than its expansion stroke were produced and tested. As a result, the following facts were established.

- (1) The heat loss increased and the thermal efficiency did not improve. This was thought to be due to the fact that a high pressure and temperature around top dead center were maintained for a longer period of time.
- (2) By changing the cooling water temperature to reduce heat loss, it was found that the exhaust temperature increased and a mutual trade-off relationship existed between the heat loss and exhaust loss.

Accordingly, to selectively reduce heat loss, the direct injection engine was operated as a stratified charge combustion and a test was carried out using an engine that had a λ of 8. As a result, it was possible to selectively reduce the heat loss alone. Also, the degree of constant volume increased, causing the thermal efficiency to improve by 1.1 point to reach 42.7%, which is the maximum value.

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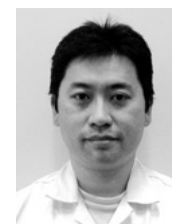
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