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Masatoshi Suzuki, Satoshi Iijima and Hayato Maehara
Honda R&D Co.,Ltd. Motorcycle R&D Center

Yasuo Moriyoshi
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ABSTRACT

In reciprocating internal combustion engines, the Otto cycle indicates the best thermal efficiency under a given compression ratio. To achieve an ideal Otto cycle, combustion must take place instantaneously at top dead center, but in fact, this is impossible. Meanwhile, if we allow slower piston motion around top dead center, combustion will be promoted at that period; then both the in-cylinder pressure and degree of constant volume will increase, leading to higher thermal efficiency. In order to verify this hypothesis, an engine with slower piston motion around top dead center, using an ideal constant volume combustion engine, was built and tested. As anticipated, the degree of constant volume increased. However, thermal efficiency was not improved, due to increased heat loss. Accordingly more experiments, which achieved a slower piston motion around top dead center by adopting a larger ratio between the connecting-rod length and the crank radius, were carried out using direct injection stratified charge combustion, which allows selective reduction of heat loss. High thermal efficiency was attained, as expected. On the other hand, an engine with a faster piston motion around top dead center, created by decreasing the ratio between connecting-rod length and crank radius, attained high thermal efficiency with quick burn pre-mixed spark ignition combustion.

INTRODUCTION

A higher degree of constant volume may be considered as increasing the thermal efficiency of an Otto cycle engine. To that end, attempts have been made to increase combustion speed by enhancing in-cylinder turbulence using swirl and tumble flow[1]-[3], reducing the flame propagation distance by use of a compact combustion chamber, center spark plug, and/or a number of spark plugs per cylinder, etc. Such attempts are aimed at reducing of the combustion duration to bring it close to the ideal Otto cycle, thus increasing the area enclosed by the lines of a P-V diagram. In the mean-time, an attempt to increase the area enclosed by the lines of a P-V diagram by mechanical design has also been reported by Miwa, et al.[4].

Previously the authors attempted to improve thermal efficiency by increasing in-cylinder pressure through slower piston motion around the top dead center (TDC) while promoting combustion during that period[5]. Although the degree of constant volume was increased and a P-V diagram close to an ideal Otto cycle was attained, thermal efficiency was not improved. Through numerical analysis[6], it was determined that the loss of efficiency due to heat loss was greater than the benefit derived from improving the degree of constant volume.

Taking these results into consideration, in the research reported here attention was focused on reducing of heat loss. We decided to try to improve thermal efficiency by two methods. One was to determine how thermal efficiency is affected by improving the degree of constant volume by slower piston speed around the TDC, where the combustion is slow and the degree of constant volume is low. The other was to measure thermal efficiency when the piston speed was conversely faster around the TDC, where the combustion takes place at a degree of constant volume that is comparatively quick. In both cases, an improvement of thermal efficiency was achieved. The detail are reported below.

RELATIONSHIP BETWEEN DEGREE OF CONSTANT VOLUME AND THERMAL EFFICIENCY

IN ORDINARY ENGINES

In a reciprocating internal combustion engine, the piston displaces differently near the TDC and the bottom dead center (BDC). As shown in Fig.1, the piston displacement with crankshaft rotation is greater on the TDC side than the BDC side. This can be clearly understood by comparing it with a sine curve which is symmetrical for both the TDC and the BDC, as shown in the figure.

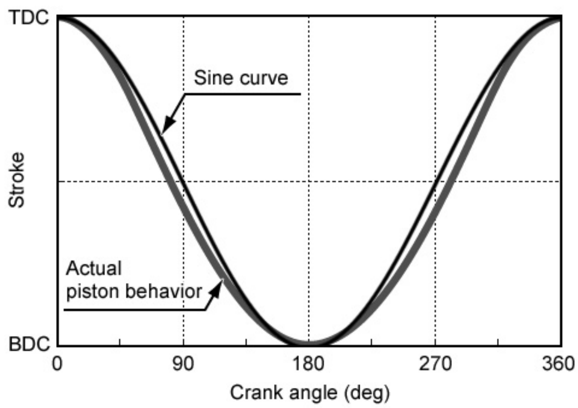

Fig.1 Actual piston behavior against crankangle

Keeping in mind, the authors thought that if the piston could be displaced more moderately than the sine wave at the TDC, just as it is at the BDC, the piston speed would be slower at the TDC. Allowing completion of combustion when the combustion chamber volume is smaller, would increase the degree of constant volume for the improvement of thermal efficiency. Experiments were carried out using the test engine.

AN IDEAL CONSTANT VOLUME COMBUSTION (ICVC) ENGINE

ICVC engine specifications

Using a 4-stroke, air-cooled, single-cylinder, 125 cm^3 gasoline engine for motorcycles as a base, the following modifications were made.

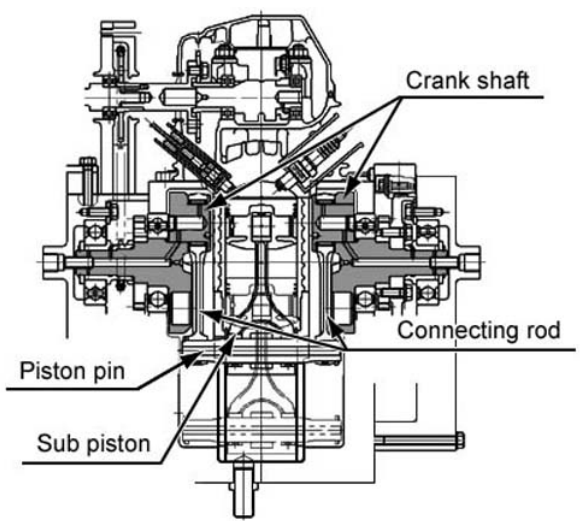

Fig.2 Ideal constant volume combustion engine

As shown in Fig.2, the sub-piston, which slides together with the piston in the same cylinder, is equipped under the ordinary piston. The long piston pin is connected to the sub piston through the cylinder wall. The small ends of a pair of connecting rods pivot on both ends of the long piston pin. Each big end of the connecting rods pulls and rotates the divided two pieces of cantilever-type crankshafts whenever the explosion pressure works on the ordinary piston. Thus, an engine was produced in which the combustion takes place at the rate of combustion chamber volume variations corresponding to the rate at an ordinary engine's BDC. The engine is refered to as an "Ideal Constant Volume Combustion (ICVC) engine". Table 1 shows the major specifications of this engine. Using the same piston, cylinder head, connecting rods, and other parts of the base engine, this engine allows combustion at the rate of combustion chamber volume variations corresponding to the rate at the BDC of the base engine. The piston behavior during a cycle is shown in Fig.3.

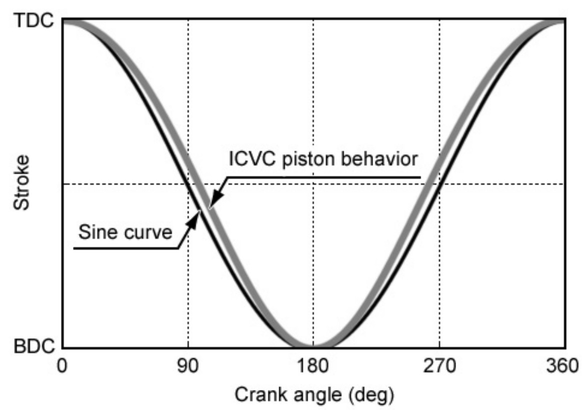

Fig.3 ICVC engine piston behavior

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
Con-rod length (mm)	93.5	

ICVC engine test results

The tests were carried out at an engine speed of 3,000 r/min, volumetric efficiency of 75 %, ignition timing at the minimum spark advance for best torque (MBT), and an air-to-fuel ratio at stoicheometric level when compared with the base engine. Fig.4 shows test results in a P-V diagram. It should be noted that in the ICVC engine the degree of constant volume increases, attaining a P-V diagram close to an ideal Otto cycle. Table2 shows indicated work determined by calculating the area of the P-V diagram in Fig.4. Despite an increase of maximum pressure (Pmax) by approximately 15 %, which means an improvement of degree of constant volume, as indicated in the table the pressure in the expansion stroke became a little lower. As a result, the indicated mean effective pressure (IMEP) declined by 5 % and the indicated specific fuel consumption (ISFC) deteriorated.

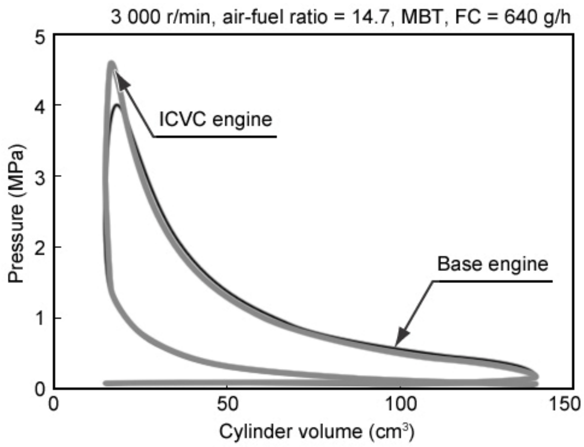

Fig.4 P-V diagram comparison

Table 2 Test results of ICVC engine

3 000 r/min, FC = 640 g/h, air-fuel ratio = 14.7, θ ig = MBT

Engine	Pmax (MPa)	Deg. of C.V.	IMEP (MPa)	ISFC (g/kWh)
ICVC engine	4.64	0.97	0.846	243
Base engine	4.02	0.94	0.893	230

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

The lowering of pressure in the expansion stroke despite the increase of maximum pressure and degree of constant volume could be due to the slowing of piston speed near the TDC, causing heat transfer from the gas to the combustion chamber walls, which in turn results in increased heat loss[6]. To cope with this problem, we considered adopting a direct injection system because the direct injection engine allows for selective reduction of heat losses as stratified charge combustion is attained[7].

In the ICVC engine, the crankshaft is divided into two pieces and linked to the single output shaft via gears in the engine. Due to this complex construction, even if an improvement was seen in the indicated work, an improvement of brake efficiency would not be achieved. Therefore, attention was paid to the ratio between the connecting rod length and the crank radius (hereafter referred to as the "rod ratio") as a way to easily modify piston speed, and we decided to produce a test engine as described below.

ENGINES WITH DIFFERENT ROD RATIOS

COMBINATION OF DIRECT INJECTION AND LARGE ROD RATIO ENGINE

Specifications of large rod ratio engine

It was decided to carry out tests using the rod ratio (the ratio between connecting rod length l and the crank radius r) as a parameter because the modifications can be made relatively easily without extensive modifications to a conventional engine, although the piston speed around the TDC does not vary as much as in the ICVC engine. The rod ratio is indicated by $\lambda(= l / r)$ in this paper. The larger the λ, the lower the piston speed becomes around the TDC, and the displacement curve on the graph becomes close to a sine curve when λ is infinitely large. Thus, for comparisons with the base engine a test engine was built with an extensively elongated connecting rod, 284 mm from the basic length of 119 mm, as shown in Fig.5. The basic specifications of the engine are shown in Table 3. It is a water-cooled, single-cylinder engine with 195 cm^3 displacement.

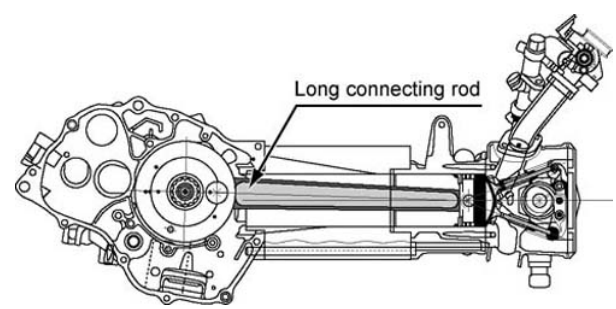

Fig.5 Long connecting rod engine

The compression ratio is 11.5 in premixed operations by port injection (PI). Two spark plugs were installed and a helical type intake port was used to enhance the combustion. The efficiency was high at the minimum ISFC of 210 g/kWh (3,000 r/min, IMEP at 0.5 MPa) in original operation. For this engine, the cylinder head and the piston were prepared to allow for stratified charge operation by direct injection (DI). In DI, the compression ratio was set at 11.8, and a single spark plug was used.

Table 3 Specifications of PI and DI engine

Items	Port injection	Direct injection
Engine type	4-stroke, water cooled	
Bore x Stroke (mm)	ϕ 59 x 71.3 single cylinder	
Displacement (cm^3)	195	
Compression ratio	11.5 : 1	11.8 : 1
Spark plug	Twin plug	Single plug

Fig.6 shows the shape of the combustion chamber of the test engine. In original operations, ISFC was 195 g/kWh (3,000 r/min, IMEP at 0.5 MPa), which means that thermal efficiency was higher than in the premixed PI operation. Table 4 shows the length of the long connecting rod to increase λ as much as possible and also the length of the connecting rod and the crank radius of the base engine.

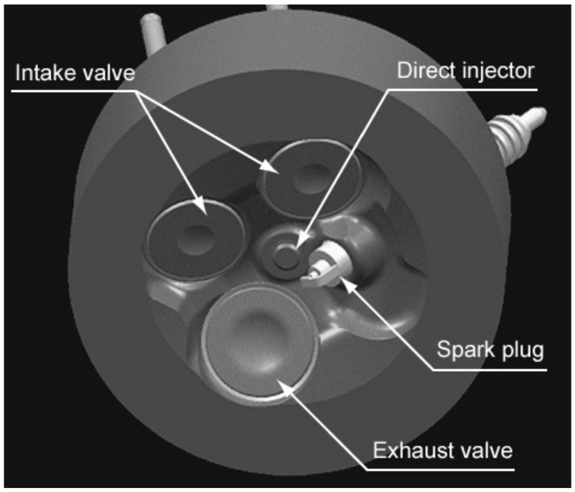

Fig.6 Combustion chamber profile in DI operation

Table 4 Specifications of long connecting rod

Items	Base	Long con-rod
Connecting rod length l (mm)	119	284
Crank radius r (mm)	35.7	
λ ($= l / r$)	3.34	7.97

con-rod: connecting rod

Although the exact λ value is 7.97, hereafter it will be considered as λ=8. Tests were carried out at each combination of the aforementioned PI premixed combustion and the DI stratified charge combustion.

Fig.7 shows the piston behavior around TDC. When λ=8, the displacement curve is nearly intermediate between the basic configuration (λ=3.34) and the sine curve.

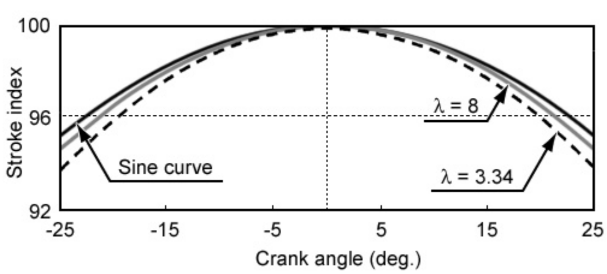

Fig. 7 Expanded piston behavior around TDC of λ=8

Test results at λ=8

The tests were carried out under stratified charge conditions with an engine speed of 3,000 r/min, corresponding load of approximately 0.3 to 0.7 MPa, and the air-to-fuel ratio at stoicheometric level for the PI and at 38 for the DI. The ignition timing was set at MBT in all tests. Fig.8 shows the results. In the basic configuration where λ is 3.34, while ISFC improves from approximately 13 to 7 g/kWh is found in the DI compared to the PI, that improvement includes contributions from an increase of specific heat ratio from the air cycle and the rise of compression ratio, in addition to the reduced heat loss. In the engine at λ=8, ISFC deteriorates by approximately 8 g/kWh in the PI. On the other hand, in DI to DI comparison, an improvement from 4 to 8 g/kWh is seen, attaining 190 g/kWh in the DI with λ=8.

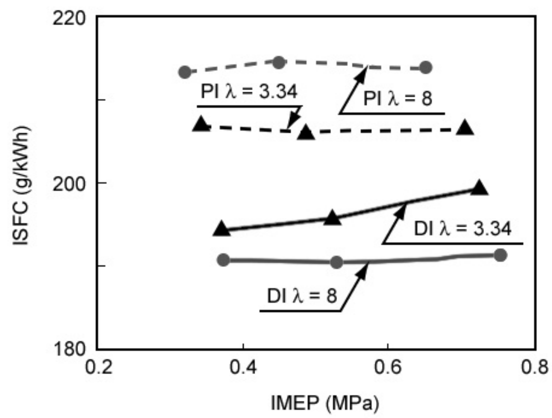

Fig.8 Test results of PI and DI, base and λ=8 engine

Setting λ at 8, lowers thermal efficiency in the PI, but an improvement is seen in the DI. To clarify how the heat loss changed and such results were attained, approximate heat loss was estimated as described below.

Although this engine is a water cooled type, the water pump is externally driven to allow the supply of engine coolant at a constant flow rate and at a constant water temperature. Heat loss is calculated by maintaining the water temperature at the exit at a constant 80 ^{o}C, multiplying the flow rate by the difference in tempertures between the entrance and exit,and converting the value to the work ratio. Also, the exhaust loss has been defined as the balance calculated by subtracting the indicated work and heat loss from the total amount of heat. Fig.9 shows the rate of each loss factor determined by the heat balance, using the above method.

In the engine with a large λ, under PI the increase of heat loss exceeded the reduction of exhaust loss by the improvement of the degree of constant volume, causing a reduction of indicated work. In the mean time, under DI the benefit of reducing heat loss itself has been confirmed, since gas layers with virtually no fuel component can be laid around the combustible mixture, or in other words, on the combustion chamber walls[7].

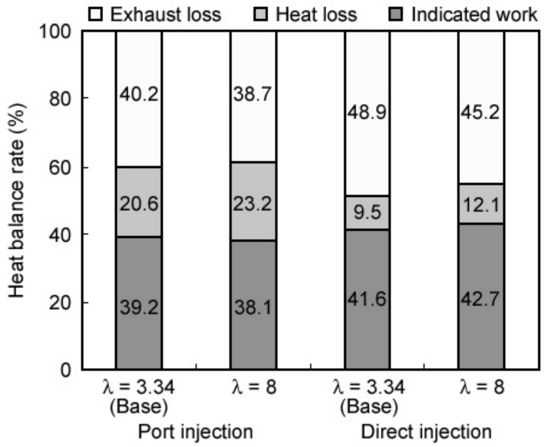

Fig.9 Heat balance rate of PI and DI, λ=3.34 and 8

Although the heat loss also increases when λ is increased, the flame propagation becomes slow in the latter half of the process due to the excessively lean mixture in the boundary zone in the direct gasoline injection. Thus the benefit of an improved degree of constant volume prevails, reducing exhaust loss and enabling the achievement of the best indicated thermal efficiency of 42.7 %.

These findings indicate that there are some benefits to be gained in the improved degree of constant volume by reducing piston speed around TDC, compared to an engine with an originally low degree of constant volume or from startified charge combustion that allows the selective reduction of heat loss. However, it has become obvious that such an attempt has a negative effect on premixed combustion when the degree of constant volume is high.

In this base engine, which had a twin spark plug system and helical intake port configuration for quick burn in the premixed combustion, the benefit from an improvement of degree of constant volume is no longer remarkable. Conversely, it can be considered that it would be more beneficial to quickly convert heat energy to work by increasing piston speed around TDC. In view of that, it was decided to produce an engine with a small λ for another test.

SMALL ROD RATIO ENGINE

Specifications of small rod ratio engine

Unlike the tests described above, it was decided to increase the piston speed around TDC. Due to various limiting factors in the design of an engine with a small rod ratio λ, it was finally set at 3.3 to 3.5. If more considerations were taken from that point and λ was reduced further, the piston speed near TDC could be increased slightly.

In view of these factors, the connecting rod was shortened to 88 mm, which was 31 mm shorter than the basic length of 119 mm as shown in Fig.10, and the test engine was produced.

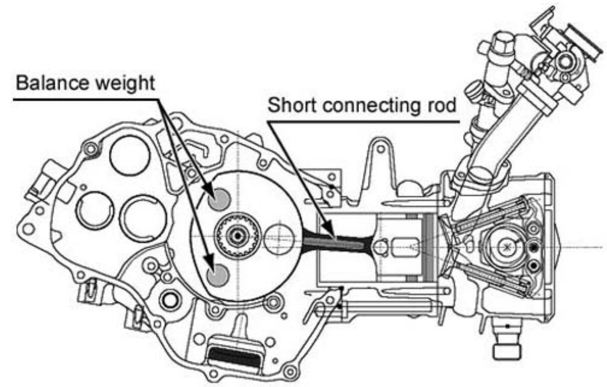

Fig.10 Short connecting rod engine

With this modification, the piston pin boss interferes with the counter weight on the crankarm when the piston reaches BDC. To cope with the problem, the relevant part of the weight was removed. To control vibrations in the horizontal direction due to the lack of a counter weight, two heavy metal pieces were inserted for balance in the part of crankarm opposite the crank pin.

To simplify prototype production, the piston pin boss was created at a place 31 mm lower than in the basic design without modifying the external appearance of the engine, including the shape of the combustion chamber. Consequently, the rod ratio λ became 2.46. This engine is hereafter referred to as the λ=2.5 engine.

The basic specifications such as the bore, stroke, displacement, etc. are the same as those of the long connecting rod engine, shown in Table 5. The piston behavior of this engine near TDC is enlarged and shown in Fig. 11. The difference of stroke length from TDC compared to the base engine was approximately 5 % when the crankshaft passed 25 degrees crank angle from TDC.

Table 5 Specifications of short connecting rod engine

Items		Base	Short con-rod
Engine type		4 stroke, water cooled	
Bore x Stroke	(mm)	ϕ59 x 71.3 single cylinder	
Displacement	(cm^3)	195	
Compression ratio		11.5	
Connecting rod length ℓ	(mm)	119	88
Crank radius r	(mm)	35.7	
λ (= ℓ / r)		3.34	2.46

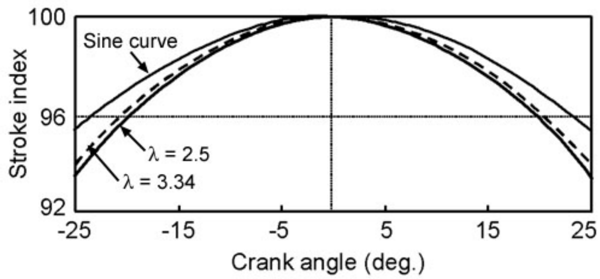

Fig.11 Expanded piston behavior around TDC of λ=2.5

Test results at λ=2.5 engine

As in the case of the large rod ratio, tests were carried out with an engine speed of 3,000 r/min, a load corresponding to 0.3 to 0.65 MPa for three kinds of rod ratio; and λ=2.5, 3.34 and 8 in premixed combustion operations by PI. Fig.12 shows the results. The engine was operated with the air to fuel ratio set at the stoicheometric level, and the ignition timing at MBT.

From Fig.12, although the difference is small, the best thermal efficiency is found in the engine using the shortest connecting rod length of λ=2.5. Fig.13 shows the rate of each loss factor determined by the heat balance in the same manner as in λ=8.

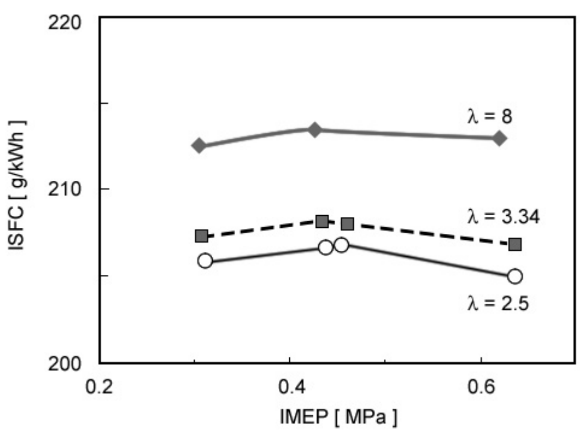

Fig.12 ISFC characteristics of various " λ "

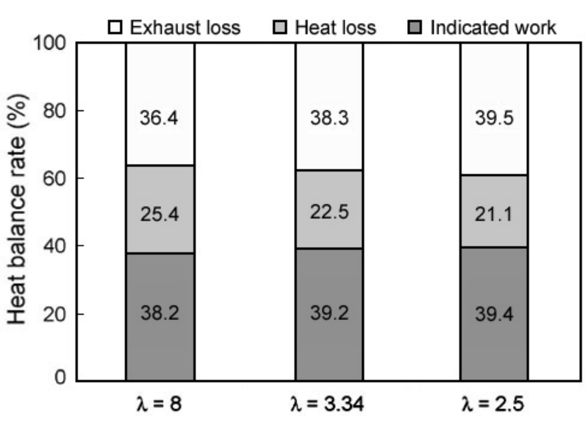

Fig.13 Heat balance rate of λ=2.5 engine

As expected, when λ is reduced from 8, the heat loss gradually decreases and thermal efficiency increases. While the exhaust loss increases at the same time, the decline of degree of constant volume can be considered.

Supplementary benefits of λ=2.5 engine

Increasing the piston speed around TDC means that the piston is located lower than in the base engine when the combustion is completed, increasing effective combustion chamber volume. In view of that, we can assume that the pressure and temperature of unburned end gas near the end of combustion are lower than those in the base engine, which can be considered as beneficial against spark knocking[9]. The anti-knocking characteristics were therefore tested.

Tests were carried out with the engine fully loaded at each engine speed from 2,000 to 5,000 r/min, the air to fuel ratio at 12.5, which provides maximum torque, the coolant temperature and oil temperature at 100 °C. The compression ratio was set at 11.5 in each test. While gradually advancing ignition timing, the times when a combustion pressure waveform indicated knocking were observed and recorded. The results are shown in Fig.14. As expected, the start of knocking was advanced by 1 to 3 degrees in crank angle from ignition timing at the case of the λ=2.5 engine compared to the base engine, thus confirming that increasing the piston's descending speed near TDC is beneficial for improving and engine's anti-knock properties.

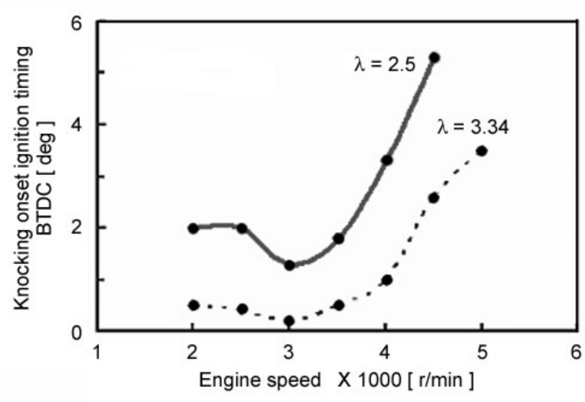

Fig.14 Improving spark knocking of "λ"=2.5 engine

CONCLUSION

An ICVC engine was produced and tested to improve thermal efficiency by enhancing the degree of constant volume; this was accomplished by allowing moderate change of the piston displacement around TDC. Although the degree of constant volume was improved, no improvement of thermal efficiency was attained. To determine the cause of this phenomenon, experiments were carried out by combining various levels of rod ratio λ to alter piston speed during the combustion period, both in port injected (PI) premixed combustion and in direct injected (DI) stratified charge combustion. The following results were obtained.

(1) It has been confirmed that the increase of heat loss does not improve thermal efficiency even when the degree of constant volume is increased by reducing piston speed around TDC.

(2) When the piston speed is reduced around TDC, the increase in degree of constant volume exceeds the increase of heat loss; thermal efficiency is improved when the heat loss is selectively reduced by the DI stratified charge combustion and/or when the combustion speed is relatively low.

(3) In a premixed combustion engine that allows quick burn, the benefit from an improvement of the degree of constant volume is no longer remarkable. Conversely, an increase of the piston speed near TDC allows an improvement in thermal efficiency.

(4) It has been shown that increasing the piston speed near TDC is beneficial for improving anti-knocking properties.

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CONTACT

Masatoshi Suzuki
Chief Engineer, Technology Reseach Division
Honda R&D Co.,Ltd. Motorcycle R&D Center
3-15-1 Senzui, Asaka-shi, Saitama, 351-8555 Japan
Tel +81-48-462 3864
Fax +81-48-462-3973
E-mail masatoshi.suzuki@mail.a.rd.honda.co.jp

Satoshi Iijima
Assistant Chief Engineer, Engineering Development Dept. 1
Honda R&D Co.,Ltd. Motorcycle R&D Center
3-15-1 Senzui, Asaka-shi, Saitama, 351-8555 Japan
E-mail satoshi.01.iijima@mail.a.rd.honda.co.jp

Hayato Maehara
Technology Reseach Division
Honda R&D Co.,Ltd. Motorcycle R&D Center
3-15-1 Senzui, Asaka-shi, Saitama, 351-8555 Japan
E-mail hayato.maehara@mail.a.rd.honda.co.jp

Yasuo Moriyoshi, Dr.
Dep. of Mech. Eng., Chiba Univ.
1-33 Yayoi-Cho, Inage-ku, Chiba 263-8522 Japan
E-mail ymoriyos@faculty.chiba-u.jp