

Research on Extended Expansion General-Purpose Engine

— Theoretical Analysis of Multiple Linkage System and Improvement of Thermal Efficiency —

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ABSTRACT

Research has been conducted on an extended expansion engine, using a multiple linkage system to increase the thermal efficiency of general-purpose engines. A four-jointed linkage was used between the connecting rod and the crank pin of a standard piston-crank system. The end of the linkage rotates at half the speed of the crankshaft, resulting in piston strokes unequal length in each revolution. The length of the expansion stroke is greater than that of the compression stroke, thereby providing an extended expansion cycle (Atkinson cycle).

Theoretical thermal efficiency and numerical simulations were used to determine the increase in thermal efficiency at a compression ratio of 8.5 and an expansion ratio of 12.3. The results show that this extended expansion increases indicated thermal efficiency 4.0% against a conventional engine, from 27.3 to 31.3%. In addition, the new design displayed increases of 2.0 and 2.7% in indicated thermal efficiency against naturally aspirated Miller cycle engines with early valve closure and late valve closure, respectively.

An air-cooled single-cylinder engine equipped with this mechanism was constructed and its thermal efficiency was compared in tests against a conventional engine in the same power class. The results of these tests matched the simulation results almost exactly, with the new engine displaying 4.1% higher thermal efficiency than in the conventional engine.

1. Introduction

Demands for enhanced fuel economy to prevent global warming and conserve fossil fuels are on the rise, not just for cars and motorcycles, but also in the work equipment sector. The compact general-purpose engines mounted in work equipment are typically air-cooled, often operated continuously at medium to high loads and may also use low-octane fuels in developing countries. For

these and other reasons, increasing the compression ratio can lead to knocking, so there is a limit to how much efficiency can be improved simply by increasing the compression ratio.

This research designed and manufactured a prototype extended expansion engine that uses a piston-crank system with a multiple linkage system. This system increases the expansion ratio without changing the compression ratio, a means of increasing thermal

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efficiency that is more suited to general-purpose engines. Running tests were then conducted using this engine. This paper provides a theoretical analysis and overview of this mechanism, and reports the predicted improvement of efficiency and the running test results.

2. Multiple Linkage Extended Expansion Engines

2.1. Background and History

Multiple linkage internal combustion engines have a long history, with the oldest practical engine considered to be the No. 2 engine manufactured by James Atkinson in 1886⁽¹⁾. This engine used a clever link design to set the expansion stroke longer than the compression stroke, thereby providing an extended expansion cycle (Atkinson cycle) in which the expansion ratio is greater than the compression ratio.

Various studies on multiple linkage internal combustion engines have been conducted since then⁽²⁾⁻⁽⁴⁾, but most of these studies involved variable compression ratio engines that aimed to improve fuel economy, mainly under partial load conditions in automobiles, and few studies concerned applications involving extended expansion engines⁽⁵⁾.

On the other hand, practical applications of extended expansion engines⁽⁶⁾⁻⁽⁸⁾ include the Miller-cycle engine, advocated by Ralph Miller in the 1940s. This engine used a conventional internal combustion engine with a high geometric compression ratio (expansion ratio), and eliminated the need for a complex linkage system by suppressing the amount of intake mixture by adjusting the inlet valve closure timing and lowering the actual compression ratio to less than the geometric expansion ratio. However, in practical engines, high-load applications required a supercharging system, and additional devices such as a variable valve mechanism were needed to help achieve performance over a wide range of engine speeds.

There are also cases where thermal cycles using this method have been called Atkinson cycles in the broadest sense of an extended expansion cycle⁽⁹⁾, but even though both types of thermal cycles include intake and compression strokes, they are different and the thermal efficiency also differs as described below. Therefore, this paper refers to the extended expansion cycle, which depends on the inlet valve open and close timing as the Miller cycle to distinguish it from the Atkinson cycle.

2.2. Mechanism and Principle

Table 1 lists the main specifications of the engine used in this research, and Fig. 1 shows an overview of the multiple linkage system. The specifications were determined through theoretical investigation of various linkage types in consideration of the total size including the linkage, a simple parts configuration, the target power output and the improvement of efficiency. Figure 2 shows the piston displacement relative to the crank angle of the test engine. Here, the expansion and exhaust strokes in the second half of the cycle are approximately 1.5 times the length of the intake and compression strokes in the first half of the cycle, thus indicating that an extended expansion cycle based on the Atkinson cycle can be achieved.

Figure 3 shows the main dimensions of the linkage system. The quantity of state of this multiple linkage system is expressed by the following equations, where θ_c is the crank angle.

$$x = l_4 \cos \phi_4 + l_2 \sin (\phi_1 + \alpha) + r_c \cos \theta_c \quad (1)$$

Table 1 Engine specifications

Engine type		Air-cooled 4 stroke OHC single-cylinder spark-ignition
Bore (mm)		64
Stroke (mm)	Intake and compression	42.1
	Expansion and exhaust	63.3
Displacement (cm ³)	Intake and compression	135
	Expansion and exhaust	204
Compression ratio		8.5
Expansion ratio		12.3
Fuel		Gasoline
Air-fuel ratio		12.5
Gross Maximum power (kW/rpm)		3.4 / 4000

$$l_4 \sin \phi_4 + \delta = l_2 \cos (\phi_1 + \alpha) + r_c \sin \theta_c \quad (2)$$

$$l_5 - r_c \cos \theta_c + r_p \sin (\theta_p + \gamma) = l_1 \cos \phi_1 + l_3 \cos \phi_3 \quad (3)$$

$$l_6 + r_p \cos (\theta_p + \gamma) + l_3 \sin \phi_3 = l_1 \sin \phi_1 + r_c \cos \theta_c \quad (4)$$

Assuming the following relationship,

$$\theta_p = \eta \theta_c + \gamma \quad (5)$$

The linkage system has a degree of freedom of "1," so the piston position relative to the crank angle can be determined freely, assuring its operation as a linkage system. Equation (5) represents the pivot shaft rotation driven by the crankshaft, and yields the value $\eta = -1/2$ for 1/2 gear reduction driving. This provides a phase difference with a period of $\theta_c = 720^\circ$ between the crankshaft and the pivot shaft, which causes the length of each piston stroke to differ.

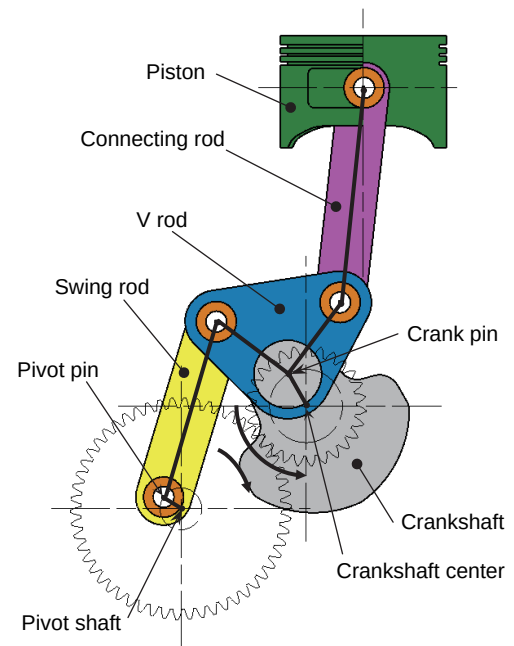


Fig. 1 Configuration of multi-link system

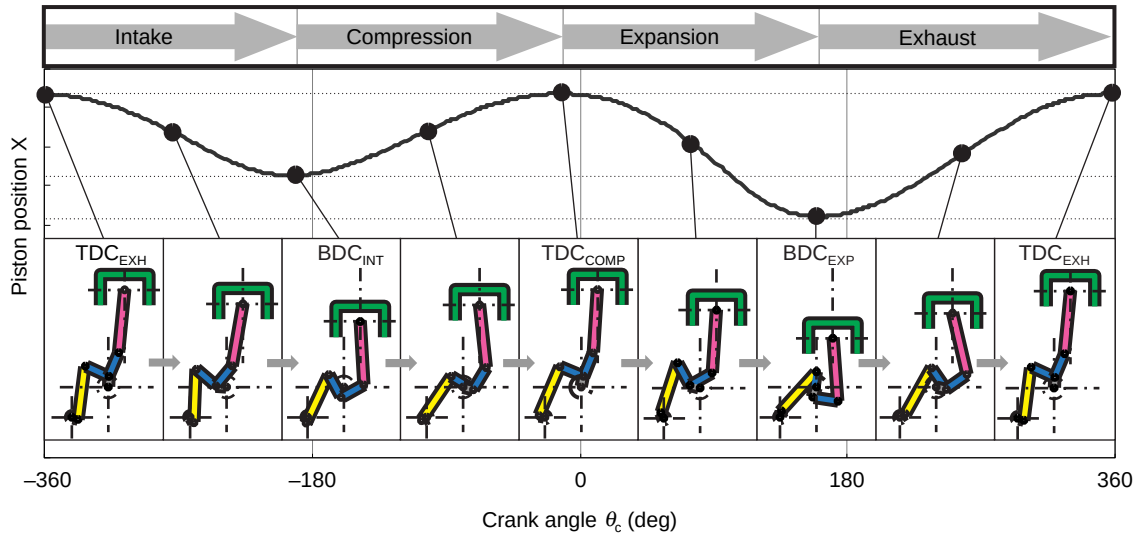


Fig. 2 Piston movement through one cycle

In addition, the initial phase difference γ is related to the top and bottom dead center positions of each stroke (Fig. 4). By applying the appropriate γ , the same piston position can be used for the compression top dead center position and the exhaust top dead center position, making it possible to obtain an extended expansion cycle where the expansion and exhaust strokes are

longer than the intake and compression strokes and the expansion ratio ε_{EXP} is larger than the compression ratio ε_{COMP} .

3. Predicted Performance

The increase in thermal efficiency provided by the extended expansion cycle was predicted by the following method.

3.1. Theoretical Thermal Efficiency

Equation (6) expresses the theoretical thermal efficiency for a general Otto cycle.

$$\eta_{th} = 1 - \frac{1}{\varepsilon_c^{\kappa-1}} \quad (6)$$

ε_c : Compression ratio (= Expansion ratio)

κ : Specific heat ratio (= Cp/Cv)

Expanding this to accommodate an extended expansion cycle with a constant pressure interval in the compression stroke, the theoretical thermal efficiency becomes as shown in Eq. (7) (Fig. 5).

$$\eta_{th} = 1 - \frac{1}{\varepsilon_e^{\kappa-1}} - \left\{ \frac{\varepsilon_e}{\varepsilon_c} (\kappa - 1) - \kappa + \left(\frac{\varepsilon_c}{\varepsilon_e} \right)^{\kappa-1} \right\} \frac{C_v T_0}{Q_{in}} \quad (7)$$

ε_e : Expansion ratio ($> \varepsilon_c$)

C_v : Specific heat at constant volume

T_0 : Compression start temperature

Q_{in} : Additional heat at constant volume

Figure 6 shows the theoretical thermal efficiency when the compression ratio is fixed and only the expansion ratio is increased, in reference to the compression ratio of 8.5 for a typical general-purpose engine. This shows that the increased expansion ratio of the test engine has the potential to improve the thermal efficiency by approximately 10% or more.

3.2. Numerical Simulation

The above-mentioned theoretical thermal cycle does not take into account factors such as the pumping loss involved with intake and exhaust or the heat loss from the cylinder wall during the expansion and compression strokes, so it differs from the actual

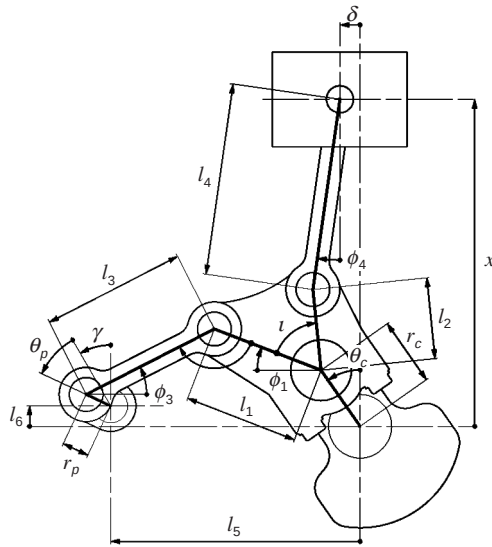


Fig. 3 Characteristic dimensions of linkage system

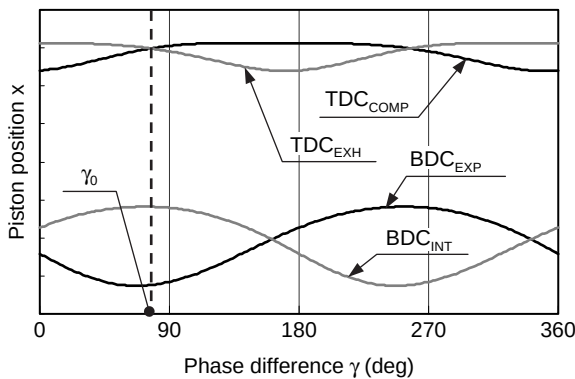


Fig. 4 Relationship between phase difference and each dead center

thermal cycle. To more accurately estimate the increase in efficiency provided by the extended expansion cycle, thermal cycle simulations⁽¹⁰⁾ were conducted using a numerical calculation model that took into account the intake and exhaust air flow as a quasi-one-dimensional flow, and the combustion inside the cylinder and heat transfer to the cylinder walls as a non-dimensional vessel. The functions of existing in-house software were expanded to perform the calculations, allowing the physical levels in each stroke to be calculated based on the operation of a multiple linkage type extended expansion engine.

Table 2 shows the specifications and operating conditions of the conventional engines used for comparison. The Type A conventional engine has the same intake stroke volume as the test engine, and the Type B conventional engine has the same calculated WOT output power as the test engine.

The heat released by combustion was calculated by approximating the time history of the mass fraction burned, based on actual measurement results using a Wiebe function⁽¹¹⁾, and all of the conventional and test engines were assumed to release the same amount of heat using the compression top dead center state as a reference. The total amount of heat supplied to the thermal cycle used the value calculated from the fuel combustion level of the test engine at 100% load and was the same for all specifications. The heat loss was calculated using the heat transfer coefficient

Table 2 Comparison with conventional engine specifications

	Extended expansion	Conventional	
		TYPE A	TYPE B
Bore x stroke (mm)	64x42.1 / 63.3	64x42.1	64x47
Compression ratio	8.5	8.5	
Expansion ratio	12.3	8.5	
Engine revolutions (rpm)		3060	
Air fuel ratio		12.5	
Supplied heat (J/cycle)		456	
Combustion efficiency		0.725	

Table 3 Calculated work and loss of each stroke

		Extended expansion	Conventional	
			TYPE A	TYPE B
Work or loss of each stroke (J/stroke)	Intake	-3.2	-3.9	-5.3
	Compression	-31.4	-32.9	-31.8
	Expansion	178.5	163.3	162.4
	Exhaust	-1.1	-0.7	-0.8
Heat loss from cylinder wall (J/cycle)		91.3	82.1	84.9
Indicated efficiency (net)		31.3%	27.6%	27.3%

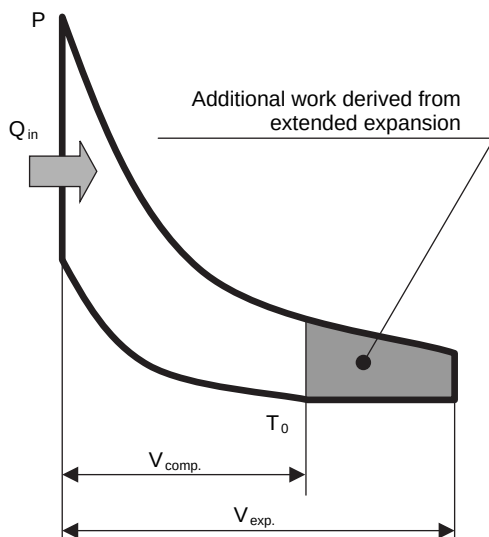


Fig. 5 Extended expansion thermal cycle

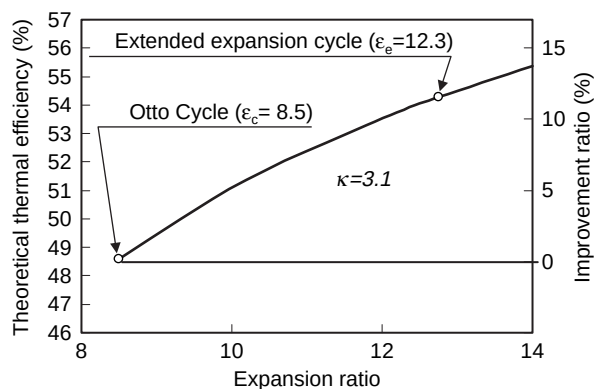


Fig. 6 Theoretical efficiency of extended expansion cycle

formulated by Woschini, et al.⁽¹²⁾, with the same cylinder wall temperature and other boundary conditions used for all engines. The air/fuel ratio, combustion efficiency, valve open and close timing and other operating conditions were also the same. In addition, the intake pressure loss used the value equivalent to when the butterfly valve is completely open to reduce differences in the choking pressure loss when opening the throttle.

The calculation results for each specification are described below. A comparison of the P-V diagrams for the test engine and the Type A engine (Fig. 7) shows that work derived from extended expansion is added at the end of the expansion stroke for the test engine due to the cylinder pressure during the extended expansion stroke. In addition, a slight increase is seen in the maximum cylinder

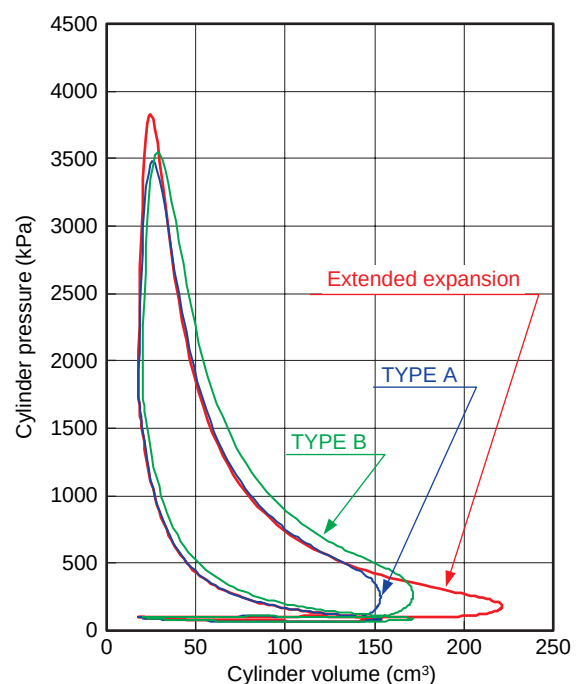


Fig. 7 Calculated P-V diagrams

pressure, but this is because the piston speed of the test engine near the compression top dead center is slower than that of the conventional engine.

To view a breakdown of the increase in efficiency, Table 3 shows the calculated values for the indicated output power of each stroke and the heat loss from the cylinder wall, and Fig. 8 shows the heat balance of each engine relative to the total heat supplied to engine. Note that all subsequent indicated output power values are the indicated output power (net) including the intake and exhaust strokes.

Based on the above results, the test engine was found to have the following characteristics as compared to the Type A and Type B engines.

- (1) The expansion stroke is longer, so the heat loss from the cylinder wall increases.
- (2) The exhaust stroke is longer, so the exhaust pump loss increases, although the absolute value is small.
- (3) The exhaust heat loss decreases due to the increase in the work derived from extended expansion.

These characteristics give the test engine an indicated thermal efficiency of 31.3%, which is 3.7 percentage points higher than the 27.6% for the Type A engine, and 4.0 points higher than the 27.3% for the Type B engine, confirming that the extended expansion cycle of the test engine can increase thermal efficiency, even in a more realistic thermal cycle that includes the intake and exhaust strokes and the heat loss from the cylinder wall.

3.3. Comparison with the Naturally Aspirated Miller Cycle

As mentioned above, most practical extended expansion engines are Miller-cycle engines that increase the expansion ratio through a combination of throttling the intake mixture air flow by varying the inlet valve close timing, and using a combustion chamber that has a heightened geometric compression ratio. This study used the above-mentioned numerical simulations to compare the indicated efficiency of the test engine's Atkinson cycle with those of single cylinder, naturally aspirated Miller cycle engines that use early and late inlet valve closure. Table 4 shows the specifications of the Miller-cycle engines used for the calculations.

The stroke volume of the Miller-cycle engines used for comparison was the same as the expansion stroke volume of the test engine. In addition, the expansion specifications of both engine

Table 4 Comparison with Miller cycle engine specifications

	Extended expansion	Miller cycle	
		TYPE C	TYPE D
Bore x stroke (mm)	64x42.1 /63.3	64x63.3	
Combustion chamber volume (cm³)	18		
Geometrical compression	8.5	12.3	
Expansion ratio	12.3	12.3	
Intake valve close	STD	51deg early closure	78.5deg late closure
Engine revolution (rpm)	3060		
Air fuel ratio	12.5		
Supplied heat (J/cycle)	456		
Combustion efficiency	0.725		

Table 5 Calculated work and loss of each stroke

		Extended expansion	Miller cycle	
			TYPE C	TYPE D
Work or loss of each stroke (J/stroke)	Intake	-3.2	-8.1	-7.8
	Compression	-31.4	-36.5	-37.9
	Expansion	178.5	179.2	177.1
	Exhaust	-1.1	-1.2	-1.2
Heat loss from cylinder wall (J/cycle)		91.3	94.6	95.8
Indicated efficiency (net)		31.3%	29.3%	28.6%

types were aligned by matching the geometric compression ratio of the Miller-cycle engines to the expansion ratio of the test engine. In the early inlet valve closure Miller cycle (Type C), the throttle was set to fully open and the valve open period and lift amount were proportionally reduced with the intake valve open timing fixed. In addition, in the late inlet valve closure Miller cycle (Type D), the throttle was set to fully open and the valve open period at maximum lift was extended with both the inlet valve opening timing and the maximum lift amount fixed. Operation was then compared using the same intake mixture air flow, that is to say the same feed heat amount as that of the test engine (Fig. 9).

Table 5 shows the calculation results for the indicated output power of each stroke, the heat loss from the cylinder wall, and the indicated thermal efficiency obtained by the simulation for each specification. The effects of extended expansion are clearly evident, with the expansion stroke work greatly exceeding that of the previous Type A and Type B conventional engines for all specifications. However, Type C and Type D engines both showed greater loss in the intake and compression strokes compared to that in the test engine. This is caused by the following factors:

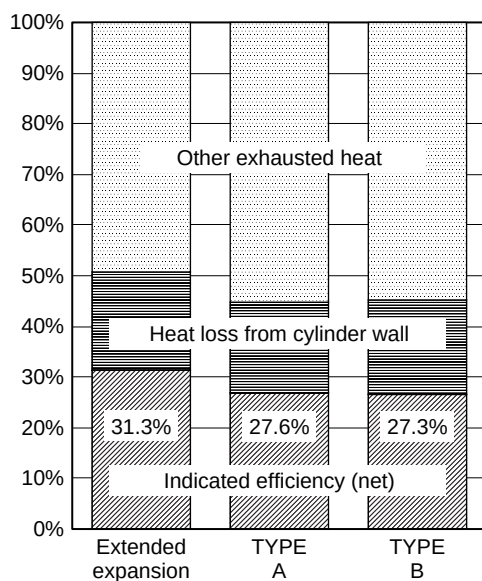


Fig. 8 Heat balance diagram

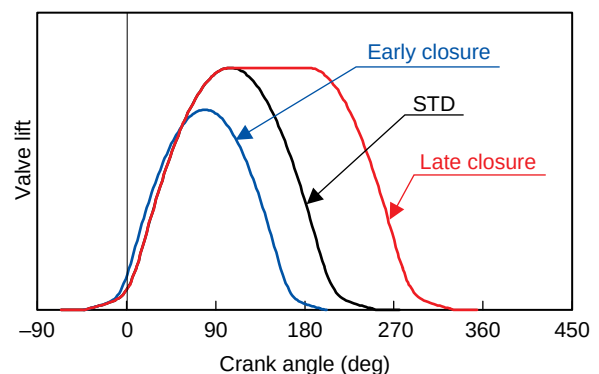


Fig. 9 Inlet valve lift curve

- (1) Type C and Type D engines have longer intake and compression stroke lengths.
- (2) When using a realistic valve lift curve, the inlet valve produces choking pressure loss in the low lift interval near the open and close timing.
- (3) For this reason, the pump loss increases during the intake period in the early closure cycle (Type C).
- (4) Similarly, pump loss occurs during the compression stroke due to intake blow-back in the late closure cycle (Type D).

The single cylinder, naturally aspirated Miller cycle engine increased the indicated thermal efficiency by 1.0 to 1.7 percentage points compared to the conventional engines, but the test engine increased the thermal efficiency even further. These numerical simulation results confirmed the advantage of the ideal extended expansion cycle (Atkinson cycle) achieved by the newly developed mechanism.

4. Test Results

Based on the results of the above investigations, a prototype extended expansion engine using a multiple linkage system was manufactured to verify the linkage system operation and assess performance during actual operation. These test results are described below.

4.1. Verification of Predicted Effects

The stroke work was calculated by measuring the cylinder's internal pressure during operation using a water-cooled piezoelectric pressure sensor mounted on the cylinder head of the prototype engine. Figure 10 compares the P-V diagram at 100% load with the above numerical simulation results. The test results show a slightly lower value for the maximum cylinder pressure, and some pressure change differences in the compression and expansion strokes can also be seen. These differences were determined to be due to the need for increased accuracy in the

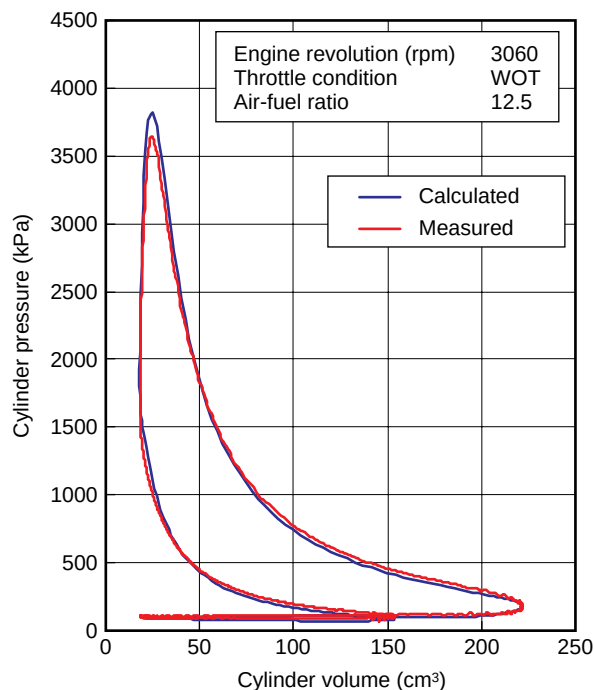


Fig. 10 Measured and calculated P-V diagrams

Table 6 Comparison of calculated and measured work

Work or loss of each stroke (J/stroke)	Calculated		Measured
	Intake	-3.2	
	Compression	-31.4	
	Expansion	178.5	
	Exhaust	-1.1	-1.5
Indicated efficiency (net)		31.3%	31.2%

Table 7 Test engine specifications

	Extended expansion	Baseline Otto cycle
Bore x stroke (mm)	64x42.1 / 63.3	68x45
Compression ratio	8.5	8.5
Expansion ratio	12.3	
Ignition timing (deg BTDC)	24	25
	(MBT on WOT)	
Air fuel ratio	12.5	
Gross maximum power (kW/rpm)	3.2/3060	

heat-generation model used for calculations in the former case, and in the heat-loss model in the latter case. However, the indicated output power and the indicated thermal efficiency calculated from the P-V diagram matched almost exactly (Table 6), thereby confirming the validity of the method for predicting the performance criteria described above.

4.2. Verification of Increased Efficiency

These test results were compared with those of a conventional engine in the same power class to verify the increase in thermal efficiency. A Honda air-cooled, single cylinder general-purpose engine with approximately the same gross maximum power was selected as the engine for comparison. Table 7 shows the engine specifications.

Figure 11 compares the indicated efficiency for each load at the same engine speed. This shows that the efficiency is increased over the entire load range, and the effects are even greater at higher loads. The indicated efficiency at 100% load is 4.1 percentage points

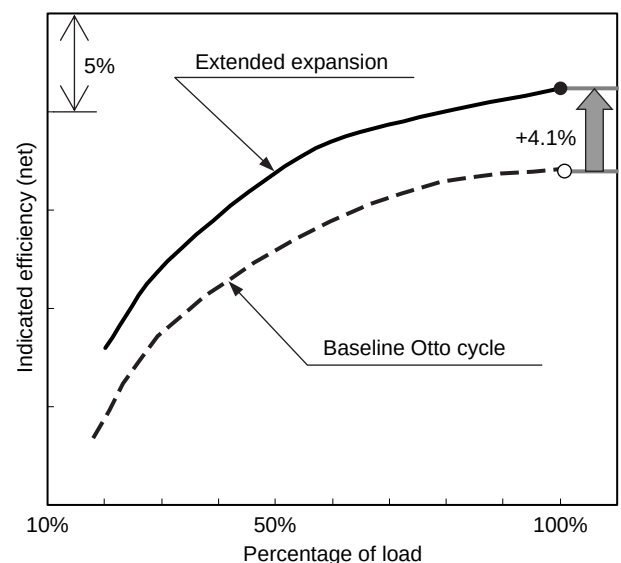


Fig. 11 Improvement of indicated efficiency (net)

higher than the comparison engine, which corresponds to an approximate 15% improvement of fuel economy as compared to the conventional engine. These results confirmed through actual engine operation the effects of applying this technology to general-purpose engines, which require improved efficiency in the medium to high load range.

5. Conclusion

Research was conducted on an extended expansion engine using a multiple linkage system with the aim of increasing general-purpose engine efficiency, and the following results were obtained.

- (1) Numerical simulations showed that an extended expansion engine with a compression ratio of 8.5 and an expansion ratio of 12.3 increases the indicated thermal efficiency by 3.7 percentage points compared to a conventional engine with the same intake volume, and by 4.0 percentage points compared to a conventional engine with the same power.
- (2) This extended expansion engine also increased the indicated thermal efficiency by 2.0 and 2.7 percentage points over naturally aspirated Miller-cycle engines having equivalent expansion specifications with early and late inlet valve closures, respectively.
- (3) The indicated thermal efficiency obtained from operation using a test air-cooled single cylinder engine was almost exactly the same as that obtained by the above simulation. The new engine increased the measured indicated thermal efficiency by 4.1 percentage points over a conventional general-purpose engine in the same power class, which corresponds to improvements in fuel economy of approximately 15%.

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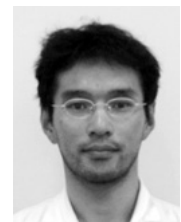
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