

Research on Inertia Force Characteristics and Its Reduction in Extended Expansion Linkage Engine

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ABSTRACT

An extended expansion engine (EXlink engine) using a multiple linkage system was studied as a system that realizes an extended expansion thermal cycle that raises the expansion ratio higher than the compression ratio in order to enhance the thermal efficiency of a general-purpose engine. An initial prototype EXlink engine increased the EPA exhaust emission control operating mode fuel economy by 13% compared to a conventional engine with the same output, but the maximum value of the inertia force produced by the multiple linkage system also increased by 93% compared to the conventional engine. This is because the maximum piston acceleration during the expansion stroke of the initial EXlink engine is 2.7 times that of the conventional engine.

A new linkage layout that controls the maximum piston acceleration was investigated to reduce the inertia force. By using a linkage configuration that allots a longer crank angle to the expansion and exhaust strokes, the piston acceleration is smoothed, enabling reduction of the inertia force. Compared to the initial linkage layout, the newly designed linkage layout reduced the calculated maximum piston acceleration by 42% and the calculated maximum inertia force value by 61%, and the expected vibration reduction effects were also confirmed in actual engine tests.

1. Introduction

The compact general-purpose engines mounted in agricultural and construction equipment have outputs corresponding to the work load, so they are often used at medium to high loads. Amid the growing concern over global warming and depletion of fossil fuels, there are demands to enhance the thermal efficiency of these compact general-purpose engines. However, many compact general-purpose engines are air-cooled and used in the stationary state, so increasing the compression ratio can lead to knocking, and there is a limit to how much the thermal efficiency can be enhanced simply by increasing the compression ratio.

The authors focused on an extended expansion thermal cycle (Atkinson cycle) in which the expansion ratio exceeds the compression ratio as a means of enhancing the thermal efficiency of compact general-purpose gasoline engines. Use of the piston system using a multiple linkage system (EXlink) shown in Fig. 1 was proposed as a means of realizing this extended expansion thermal cycle⁽¹⁾. This system realizes an extended expansion thermal cycle by setting a longer expansion and exhaust stroke length compared to the intake and compression strokes as shown in Fig. 2. Operation tests using an actual prototype engine showed that the EXlink engine has the same friction as the

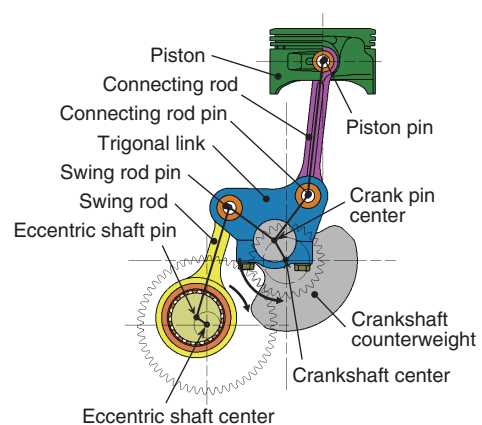


Fig. 1 Extended expansion linkage system (EXlink)

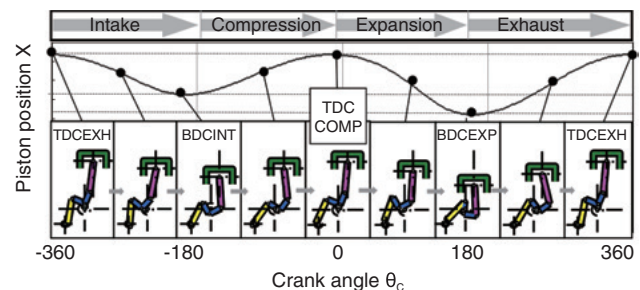


Fig. 2 Piston movement through one cycle

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conventional engine, and enhances fuel economy in the EPA emission control operating mode⁽²⁾ by 13% compared to a conventional engine⁽³⁾. However, operation tests using this initial EXlink engine clarified that vibrations originating in the inertia excitation force are larger than those of a conventional engine. This is because the EXlink engine system differs greatly from a conventional engine.

Recently, user requests to reduce engine vibration are increasing not only for cars and motorcycles, but also for work equipment that uses compact general-purpose engines. As a result, regulations concerning work equipment vibration are being enacted in various countries on a trial basis⁽⁴⁾. This makes the reduction of engine vibration an important technical issue for practical application of the EXlink engine. This paper presents numerical calculations describing an approach to reducing the inertia excitation force using a new linkage layout, and reports on verification of the effects by actual engine operation tests.

2. Inertia Excitation Force Characteristics

2.1. Dynamic Characteristics of Multiple Linkage System

This chapter describes the formulation of equations for the acceleration and inertia force of each component part of the extended expansion engine using multiple linkage system to investigate the dynamic characteristics of the entire system. Figure 3 shows the main dimensional parameters of the multiple linkage system, and Table 1 gives the specifications of the initial EXlink engine proposed in a previous report⁽³⁾ and the conventional engine used for comparison. The stated quantities of the various parts of the multiple linkage system are expressed by the following equations, with the crank angle as $\theta_c^{(1)}$.

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

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

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

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

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

Equation (5) expresses the driving of the eccentric shaft by the crankshaft, and $\eta = -1/2$ is given in the case of gear reduction ratio of 1/2. This results in a phase difference with a cycle of $\theta_c = 720^\circ$ between the crankshaft and the eccentric shaft, which produces a difference in the length of each stroke in piston operation. By applying an appropriate initial phase difference γ , the compression top dead center and the intake/exhaust top dead center can be set to the same piston position. This enables obtaining an extended expansion thermal cycle in which the expansion and exhaust strokes are longer than the intake and compression strokes, and the expansion ratio is larger than the compression ratio.

Table 1 Engine specifications

			EXlink engine	Conventional engine
Engine type			Air cooled, 4 stroke, single cylinder, horizontal shaft	
Bore		[mm]	64	68
Stroke	Intake /Compression	[mm]	42.1	45
	Expansion /Exhaust	[mm]	63.3	
Displacement	Intake /Compression	[cm³]	135.4	163.4
	Expansion /Exhaust	[cm³]	203.6	
Compression ratio			8.5	8.5
Expansion ratio			12.3	
Fuel			Gasoline	
Maximum power		[kW]	3.2	3.1
Valve mechanism			OHC	OHV

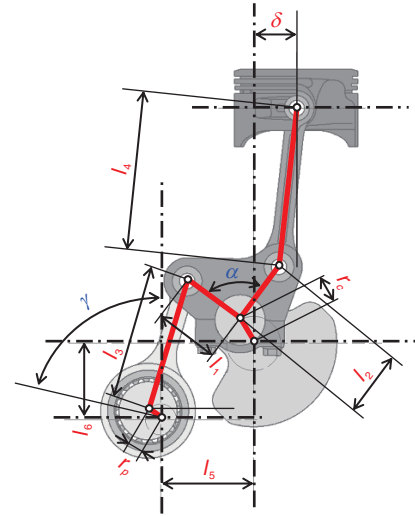


Fig. 3(a) Dimensional parameters of multiple linkage system

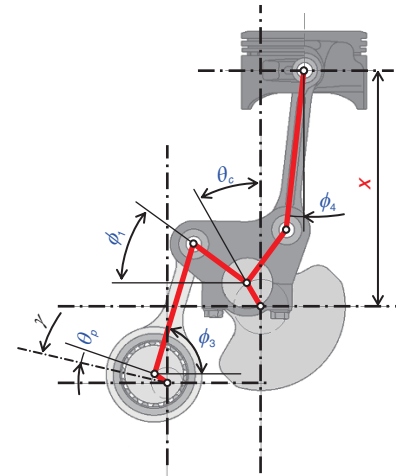


Fig. 3(b) Characteristic state quantities of multiple linkage system

The acceleration of each component part of the multiple linkage system can be obtained by solving the following relational equations, which are derived by the second-order differential of Eq. (1) to Eq. (4) relative to time.

$$\begin{aligned} & \frac{d^2 x}{dt^2} + l_4 \left(\frac{d\phi_4}{dt} \right)^2 \cos \phi_4 + l_4 \frac{d^2 \phi_4}{dt^2} \sin \phi_4 \\ &= -l_2 \left(\frac{d\phi_1}{dt} \right)^2 \cos(\phi_1 + \alpha) - l_2 \frac{d^2 \phi_1}{dt^2} \sin(\phi_1 + \alpha) \\ &+ r_c \frac{d^2 \theta_c}{dt^2} \cos \theta_c - r_c \left(\frac{d\theta_c}{dt} \right)^2 \sin \theta_c \end{aligned} \quad (6)$$

$$\begin{aligned} & r_c \frac{d^2 \theta_c}{dt^2} \sin \theta_c + r_c \left(\frac{d\theta_c}{dt} \right)^2 \cos \theta_c \\ &+ r_p \left(\frac{d\theta_p}{dt} \right)^2 \cos(\theta_p + \gamma) - r_p \frac{d^2 \theta_p}{dt^2} \cos(\theta_p + \gamma) \\ &= -l_1 \left(\frac{d\phi_1}{dt} \right)^2 \cos \phi_1 - l_1 \frac{d^2 \phi_1}{dt^2} \sin \phi_1 \\ &- l_3 \left(\frac{d\phi_3}{dt} \right)^2 \sin \phi_3 - l_3 \frac{d^2 \phi_3}{dt^2} \sin \phi_3 \end{aligned} \quad (7)$$

Next, the inertia force acting on the multiple linkage system was evaluated. Equations describing the inertia force of the entire multiple linkage system were formulated by viewing the entire system as consisting of multiple concentrated masses⁽⁵⁾. The equivalent mass at the node of each part of the multiple linkage system can be calculated from the gravity center and distance to the node of each part, and this equivalent mass was allocated to the six concentrated masses shown in Fig. 4 and Table 2.

The inertia force acting on each concentrated mass is expressed as follows as a function of the crank angle by multiplying the equivalent mass of these concentrated masses with the previously obtained acceleration of each part.

$$\begin{aligned} Fix_k(\theta) &= -m_k \cdot Ax_k \\ Fiy_k(\theta) &= -m_k \cdot Ay_k \end{aligned} \quad (8)$$

In addition, the size of the force is given by the following equation.

$$Fi_k(\theta) = \sqrt{Fix_k(\theta)^2 + Fiy_k(\theta)^2} \quad (9)$$

In this manner, equations describing the dynamic characteristics of the multiple linkage system were obtained.

2.2. Inertia Excitation Force Characteristics

The values of the summation of inertia force and maximum summation of inertia force described below were used as indices to evaluate the inertia excitation force characteristics of the entire system. The summation of

Table 2 Equivalent mass of linkage system

Equivalent mass	Mass of component parts	Corresponding mass
m_1	Piston, Piston ring, Piston pin	Connecting rod (small end)
m_2	Connecting rod pin	Connecting rod (big end) Trigonal link (m_{T2})
m_3	Crank pin	Trigonal link (m_{T3})
m_4	Swing rod pin	Trigonal link (m_{T4}) Swing rod (small end)
m_5	Eccentric shaft pin	Swing rod (big end)
m_6	Crankshaft counter weight	

inertia force is obtained by adding the inertia force of each concentrated mass, given by Eq. (10), for each component of the entire system at an arbitrary crank angle, and is a vector quantity having the components described by the following equations.

$$\begin{aligned} Fix(\theta) &= \sum_k Fix_k(\theta) \\ Fiy(\theta) &= \sum_k Fiy_k(\theta) \end{aligned} \quad (10)$$

In addition, the size of the vector quantity is given by the following equation.

$$Fi(\theta) = \sqrt{Fix(\theta)^2 + Fiy(\theta)^2} \quad (11)$$

The maximum $Fi(\theta)$ value during one cycle (720 degCA) is called the maximum summation of inertia force.

The summation of inertia force is determined according to the engine speed, the system dimensions, and the mass of each part, and the size and characteristics of the inertia excitation force generated in the system can be easily evaluated by analyzing the change in the summation of inertia force relative to the crank angle. This value can also be defined for the piston and crankshaft system of a conventional engine.

The counterweight of the conventional engine used to compare the characteristics was the mass that results in the minimum calculated value for the maximum summation of

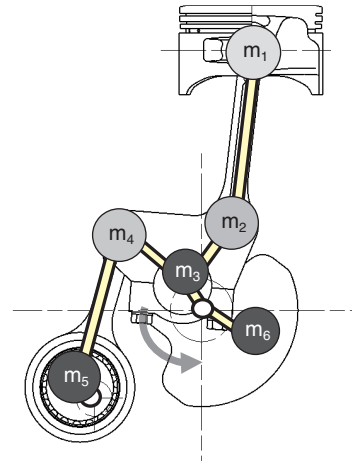


Fig. 4 Equivalent mass system

inertia force noted above. Figure 6 uses contour lines to show the size of the maximum summation of inertia force when the counterweight specifications given in Fig. 5 are varied in the test engine. The mass and the gravity center phase of the test engine counterweight were set to achieve the minimum calculated value for the maximum summation of inertia force.

Figure 7 expresses the summation of inertia force vector relative to the crank angle as a vector with the center of the crank shaft as the origin, and shows the track of the vector tip (Local curve)⁽⁶⁾. The engine speed here is 3000 rpm, which has a high frequency of use for general purpose engines. The results in this figure show that the track for the test engine traces a different shape and the maximum summation of inertia force is 93% higher than that of the conventional engine.

Next, FFT analysis was performed for the inertia force component in the F_x (cylinder axis) direction of the test engine, given by Eq. (10), relative to the crankshaft rotational speed. Figure 8 shows these results. The test engine generated inertia force components of orders in 0.5 increments relative to the crankshaft revolutions, which are not generated in a conventional piston crankshaft

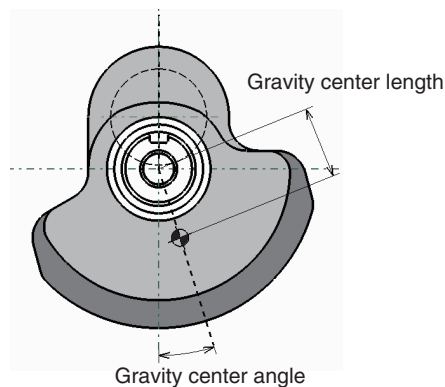


Fig. 5 Crankshaft gravity center position

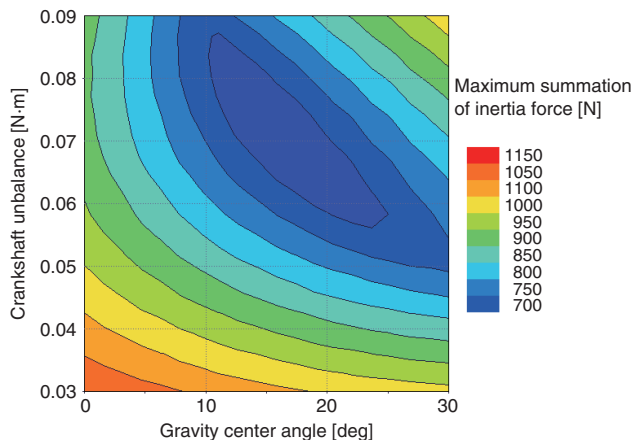


Fig. 6 Specification of counter weight and maximum summation of inertia force

system. This is because the intake and compression stroke lengths differ from the expansion and exhaust stroke lengths, which generates inertia force components with a cycle of two crankshaft revolutions. In addition, these results show that while the first-order component decreases compared to the conventional engine, the maximum summation of inertia force increases due to the second- and higher-order components. These characteristics reduce the effectiveness of rotation balancers and other measures, so an approach that concentrates the components into the main orders is thought to be better.

Figure 9 shows the results of analyzing the timing that increases the higher-order inertia force by breaking down the 2.5th- and higher-order components of the summation of inertia force in the F_x direction by concentrated mass. This shows that there are three high and low peak values from the expansion stroke to the exhaust stroke, and that the reciprocating-part inertial mass m_1 contributes greatly to each of these values. The factors for this are presumed to be that the piston acceleration of the test engine changes

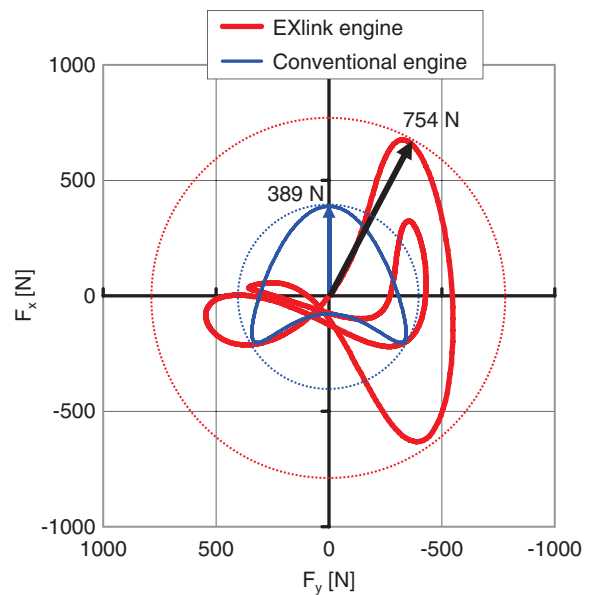


Fig. 7 Local curve

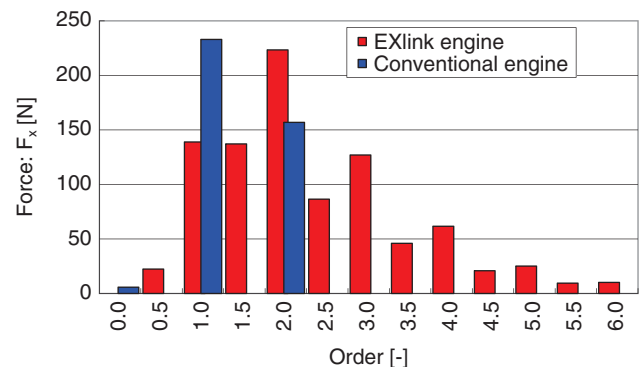


Fig. 8 FFT Analysis result (F_x)

suddenly following 90 degCA ATDC, and the maximum value is 2.7 times that of the conventional engine as shown in Fig. 10. A possible reason for the excessive piston acceleration of the test engine is that the intervals between the top and bottom dead centers of the multiple linkage system are not necessarily 180 degCA, and have unequal lengths. The expansion stroke period of the test engine is 173 degCA, which is 7 degCA shorter than the 180 degCA of the conventional engine. This means that in the test engine, the piston moves over the long expansion stroke in a short time, which is thought to result in a higher maximum acceleration.

In addition, the concentration mass m_2 that corresponds to the big end of the connecting rod also has a comparatively high degree of contribution. As shown

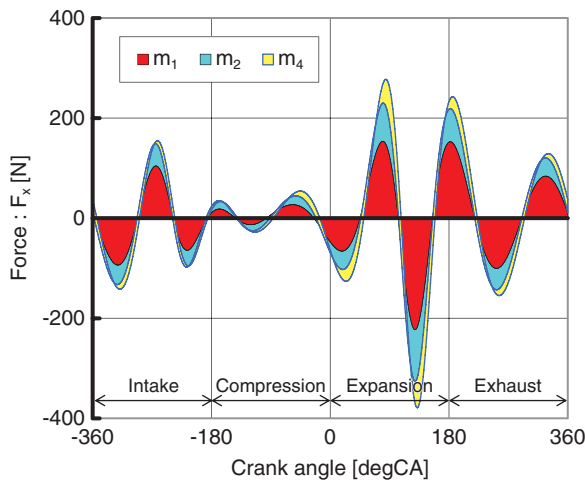


Fig. 9 Summation of inertia force for 2.5 th and higher orders (F_x)

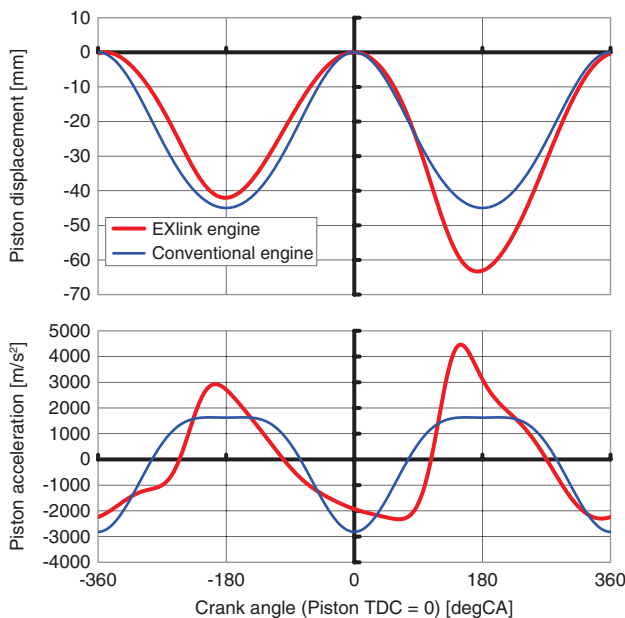


Fig. 10 Piston acceleration and displacement

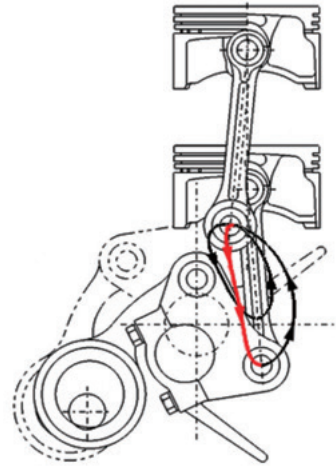


Fig. 11 Connecting rod behavior

in Fig. 11, the big end of the connecting rod traces an elliptical track lengthwise in the cylinder axis direction. For this reason, both the piston and the connecting rod mainly perform reciprocating motion in the cylinder axis direction. Therefore, the inertia force generated from m_1 and m_2 is thought to have a large influence on the piston acceleration.

3. Reduction of Inertia Excitation Force

3.1. Reduction of Piston Acceleration by New Linkage Layout

The authors thought it might be possible to reduce piston acceleration by making use of the above-noted characteristic that the intervals between the top and bottom dead centers of the multiple linkage system have unequal lengths. This was done by applying a longer crank angle to the expansion and exhaust strokes, which have longer stroke lengths.

Figure 12 and Table 3 show the newly designed

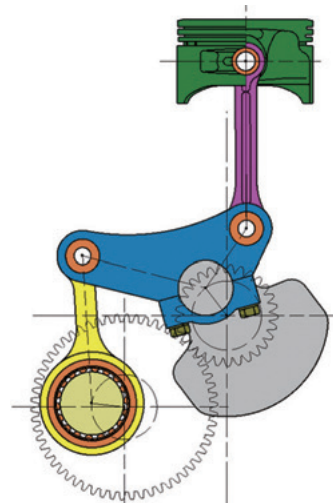


Fig. 12 Newly designed linkage system

Table 3 New and former linkage engine specifications

			Newly designed linkage engine		Former designed linkage engine
Maximum piston acceleration		[m/s²]	2575	-42%	4460
Maximum summation of inertia force		[N]	293	-61%	754
Expansion stroke term		[degCA]	190	10%	173
Bore		[mm]	64		
Stroke	Intake /Compression	[mm]	38.8	-8%	42.1
	Expansion /Exhaust	[mm]	55.7	-12%	63.3
Compression ratio			8.5	-	8.5
Expansion ratio			11.8	-4%	12.3

linkage layout. Compared to the initial linkage layout, this new linkage layout that aims to reduce vibration shortens the intake and compression strokes by -8% and the expansion and exhaust strokes by -12%, and extends the crank angle of the expansion stroke, which has a long stroke length, by 17 degCA to 190 degCA. Figure 13 shows the calculated piston acceleration at 3000 rpm. The maximum acceleration is 2575 m/s², which is a reduction of 42% compared to the initial linkage layout. In addition, the difference in the peak piston acceleration values of each stroke is also controlled, indicating that the reciprocating-part acceleration of each stroke has been smoothed.

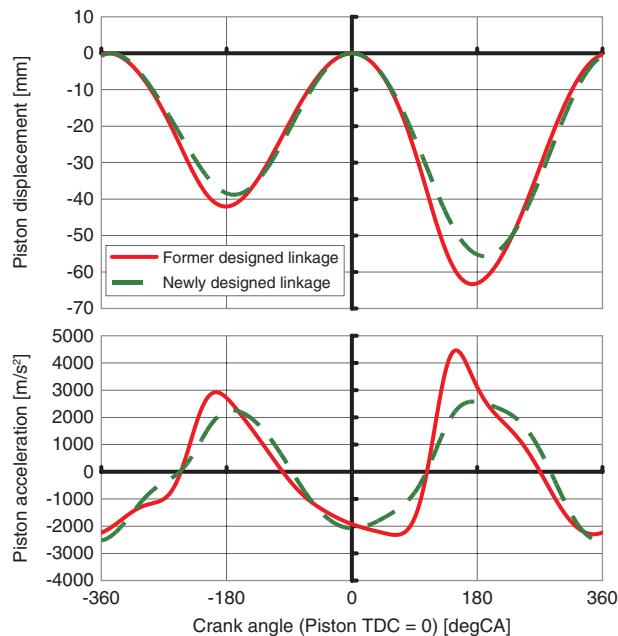


Fig. 13 Piston acceleration and displacement (New and former linkage system)

Figure 14 compares the summation of inertia force of the newly designed linkage layout with that of the initial linkage layout. The maximum summation of inertia force is 293 N, which is a reduction of 61% compared to the initial linkage layout. Figure 15 shows the FFT analysis results of component of the summation of inertia force in the cylinder axis direction, F_x . The newly designed linkage layout does not exhibit the rapid change in acceleration during the expansion stroke seen with the initial linkage layout, which indicates that the second- and higher-order components have been reduced.

3.2. Verification of Effects by Actual Engine Operation Test

The vibration of an actual engine was measured to confirm the vibration reduction effects of the newly designed linkage layout. In this test, the engine shown in Fig. 16 was operated no-load under Free-Free conditions hoisted via rubber, at the same 3000 rpm engine speed as used in the calculation conditions. The engine acceleration was measured by an acceleration sensor located on the

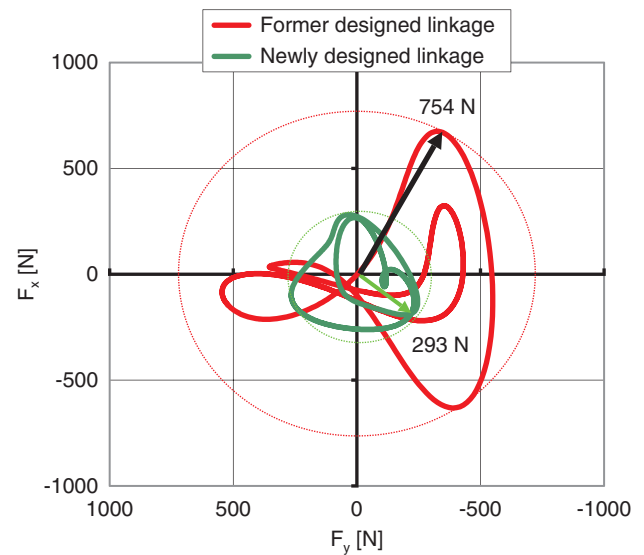


Fig. 14 Local curve (New and former linkage system)

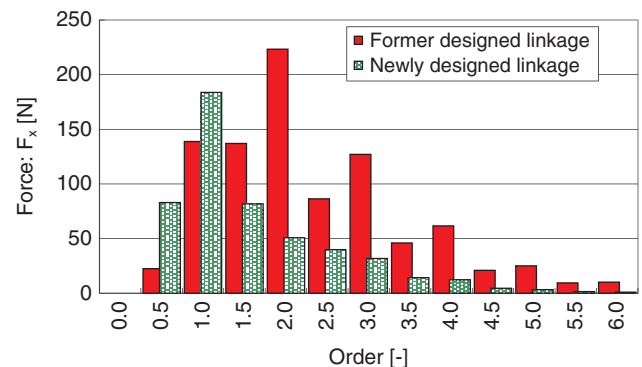


Fig. 15 Frequency characteristics

engine mount position as shown in Fig. 16, and the cylinder axis direction component was used as the typical value. The measurement time was 10 s, and the various order components relative to the crankshaft revolutions were calculated by FFT analysis.

Figure 17 shows the actual engine operation test results. The acceleration of all first- and higher-order components in the newly designed linkage layout decreased, and the overall value decreased by 43.5% compared to the initial linkage layout.

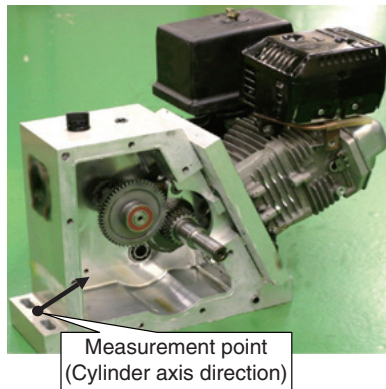


Fig. 16 Acceleration measurement point

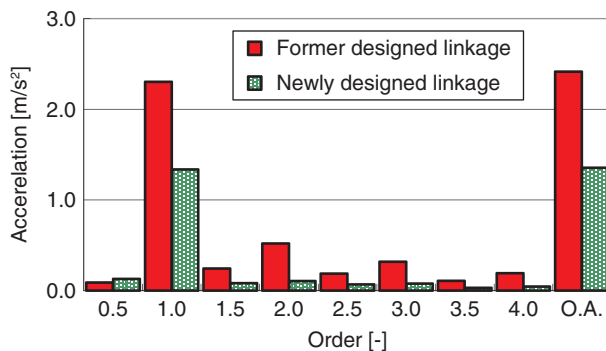


Fig. 17 Measurement results

4. Conclusion

The following conclusions were reached regarding the inertia excitation force characteristics of an extended expansion engine using multiple linkage system and methods for reducing this inertia excitation force.

- (1) The maximum summation of inertia force of the initially designed multiple linkage system was 93% higher than that of a conventional engine with the same output. This is because the crank angle during the expansion stroke, which has a long stroke length, was short at 173 degCA, and the maximum piston acceleration during the expansion stroke was 2.7 times that of a conventional engine.
- (2) The maximum calculated piston acceleration value can

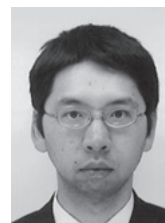
be controlled and the change in acceleration during each stroke can be smoothed by applying a longer crank angle to the expansion and exhaust strokes, which have long stroke lengths.

- (3) A newly designed linkage layout with an expansion stroke period of 190 degCA based on the above approach reduces the calculated maximum piston acceleration by 42% and the calculated maximum summation of inertia force by 61% compared to the initial linkage layout.
- (4) A prototype extended expansion engine was made using the newly designed linkage layout, and actual engine operation tests were performed. The acceleration component in the cylinder axis direction was 43.5% lower than that of the initial linkage layout, proving that linkage design based on the above approach is an effective means of reducing vibration.

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