

Research on Extended Expansion General-Purpose Engine

– Characteristic of Vibration –

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ABSTRACT

The purpose of this paper is to clarify the vibration properties of the extended expansion engine with a multiple linkage system.

The mass systems were used to obtain a substitution for the center of gravity of each component part in order to formulate the movement of the multiple linkages. Attention was focused on the maximum value for the summation of inertia forces generated by the component parts of the linkages as an indicator to use in evaluating the vibration properties of the multiple linkage system. In order to minimize this value, the optimum weight of the crankshaft counterweight and its position of gravity center were examined. The calculated results of these indicators show that the maximum value for the summation of inertia forces in the previously proposed engine is 35% larger than the value in an existing engine of the same power. In addition, operation of the test engine at 3000 rpm with no load showed that the maximum scalar quantity of biplane vibrational acceleration was 2.4 times larger than the existing engine. When we analyze it about increase of engine vibration, for the characteristics that the expansion stroke gets longer for the high expansion ratio, and the expansion term gets short, we found that it was the main factor that the maximum acceleration of the piston is 2.7 times greater than the existing engine.

INTRODUCTION

With increasing concerns worldwide for such issues as global warming and sharply rising crude oil prices, one important issue involved is the enhancement of thermal efficiency in general purpose engines used in the work equipments. Figure 1 shows an engine previously proposed by the authors that realizes an extended expansion thermal cycle (Atkinson cycle^[1]) by means of a piston-crank system utilizing a multiple linkage system. This demonstrated increased thermal efficiency as shown below^{[2],[3]}.

1. It was shown by actual measurement and numerical simulation that an extended expansion engine with compression ratio 8.5 and expansion ratio 12.3 shows 4.0 percentage points higher indicated thermal efficiency than an existing engine with the same power output.
2. In the multiple linkage engine, there is a smaller change in the minimum spark advance for best torque relative to engine load. Consequently, the brake thermal efficiency of the test engine under the EPA mode^[4] rose to 22.7%, or 2.6 percentage points higher than the 20.1% achieved by the existing engine, when operated with the change of the minimum spark advance for best torque (MBT) timing with the throttle wide open.

Meanwhile, growing demand for reduction of engine vibration has been seen not only from automobiles and motorcycles users, but also from work equipments users. National governments have been responding by instituting vibration regulations for work equipments^[5]. Reduction of engine vibration has consequently become an important technical issue in the development of multiple linkage extended expansion engines. There have been various reports on multiple linkage engines in the past^{[6] [7]}, but few reports concerning vibration^[8-10]. Furthermore, these reports all dealt with variable compression ratio engines in which the actuated cycle of the linking mechanism had 360 degCA, degrees of crank angle. They did not deal with extended expansion ratio engines.

The present report covers an extended expansion engine in which the actuated cycle of the linking mechanism has 720 degCA. The acceleration and inertia force of the component linkages were calculated and their properties were noted. Then the influence of inertia force and combustion force on the brake torque generated by the engine was examined. Finally, the correlation between the calculation results and test engine operational trial results were verified.

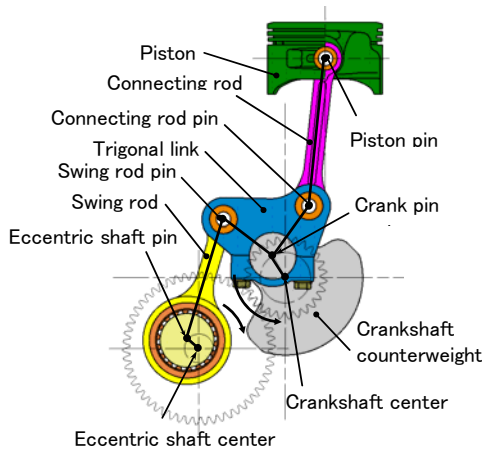


Fig.1 Configuration of multi-linkage system

MAIN SECTION

FORMULATION OF A VIBRATION CHARACTERISTIC FOR MULTI LINKAGE SYSTEM

Calculation of the acceleration and the inertia force

Figure 2 shows the main dimensions of the multiple linkage system, while Table 1 shows the main specifications of the proposed extended expansion test engine, reported previously, together with those of the existing comparison engine. The quantity of state of the various parts of a multiple linkage system is expressed by the following formulas in which the crank angle is θ_c [2].

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

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

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

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

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

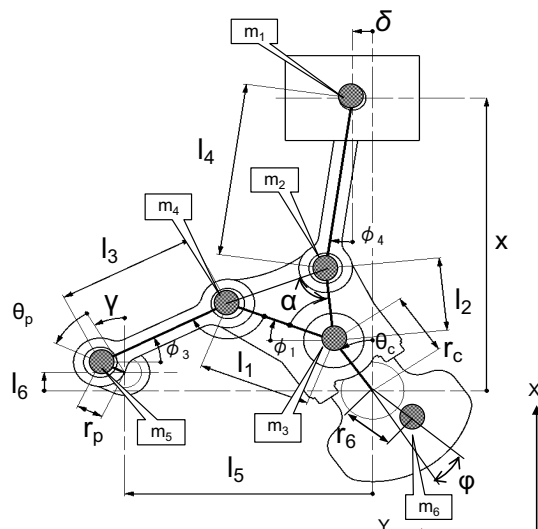


Fig.2 Characteristic dimensions of linkage system

The engine specifications are input into the formulas to calculate the quantities of state of the component parts of the linkages. Figure 3 shows the piston stroke through one cycle.

The properties of the vibration that arises because of the inertia force in this multiple linkage system are established by formulating the acceleration and inertia force of the component parts. The acceleration of the various joints can be obtained by solving the following expression, which is derived from formulas (1) to (5).

$$\frac{d^2x}{dt^2} + l_4 \left(\frac{d\phi_4}{dt} \right)^2 \cos \phi_4 + l_4 \frac{d^2\phi_4}{dt^2} \sin \phi_4 \quad (6)$$

$$= -l_2 \left(\frac{d\phi_1}{dt} \right)^2 \cos(\phi_1 + \alpha) - l_2 \frac{d^2\phi_1}{dt^2} \sin(\phi_1 + \alpha) + r_c \frac{d^2\theta_c}{dt^2} \sin \theta_c - r_c \left(\frac{d\theta_c}{dt} \right)^2 \cos \theta_c - l_4 \left(\frac{d\phi_4}{dt} \right)^2 \sin \phi_4 + l_4 \frac{d^2\phi_4}{dt^2} \cos \phi_4 \quad (7)$$

$$= -l_2 \left(\frac{d\phi_1}{dt} \right)^2 \cos(\phi_1 + \alpha) - l_2 \frac{d^2\phi_1}{dt^2} \sin(\phi_1 + \alpha) + r_c \frac{d^2\theta_c}{dt^2} \cos \theta_c - r_c \left(\frac{d\theta_c}{dt} \right)^2 \sin \theta_c + r_c \frac{d^2\theta_c}{dt^2} \sin \theta_c + r_c \left(\frac{d\theta_c}{dt} \right)^2 \cos \theta_c + r_p \left(\frac{d\theta_p}{dt} \right)^2 \cos(\theta_p + \gamma) - r_p \frac{d^2\theta_p}{dt^2} \cos(\theta_p + \gamma) \quad (8)$$

$$= -l_1 \left(\frac{d\phi_1}{dt} \right)^2 \cos \phi_1 - l_1 \frac{d^2\phi_1}{dt^2} \sin \phi_1 - l_3 \left(\frac{d\phi_3}{dt} \right)^2 \sin \phi_3 - l_3 \frac{d^2\phi_3}{dt^2} \sin \phi_3 - r_p \left(\frac{d\theta_p}{dt} \right)^2 \cos(\theta_p + \gamma) - r_p \frac{d^2\theta_p}{dt^2} \sin(\theta_p + \gamma) - l_3 \left(\frac{d\phi_3}{dt} \right)^2 \sin \phi_3 + l_3 \frac{d^2\phi_3}{dt^2} \cos \phi_3 \quad (9)$$

$$= -l_1 \left(\frac{d\phi_1}{dt} \right)^2 \sin \phi_1 + l_1 \frac{d^2\phi_1}{dt^2} \cos \phi_1 + r_c \frac{d^2\theta_c}{dt^2} \sin \theta_c - r_c \left(\frac{d\theta_c}{dt} \right)^2 \cos \theta_c$$

Next, this multiple linkage system is replaced with a mass system in an attempt to evaluate the inertia forces acting in the multiple linkage system as a whole^[11]. In other words, the mass and center of gravity of each linking part is used to represent every mass as a concentrated mass in the link attachment points. For example, the below equations apply to a trigonal link.

$$m_{T2} = \frac{m_T l_3 l_4}{l_2 l_3 + l_3 l_4 + l_4 l_2} \quad m_{T3} = \frac{m_T l_4 l_2}{l_2 l_3 + l_3 l_4 + l_4 l_2} \quad (10)$$

$$m_{T4} = \frac{m_T l_2 l_3}{l_2 l_3 + l_3 l_4 + l_4 l_2}$$

With mass m_k defined as shown below, and with the acceleration derived above expressed as $A(x,y)_k$, the inertia force $F_i(x,y)_k$ can be expressed as in equation (11).

$$F_i(x,y)_k = \sum_k \{ -m_k \cdot A(x,y)_k \} \quad (11)$$

Table 1 Engine specification

			Multiple linkage engine	Existing 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	[cm ³]	135.4	163.4
	Expansion/Exhaust	[cm ³]	203.6	
Compression ratio			8.5	8.5
Expansion ratio			12.3	
Ignition timing		[deg]	MBT(W.O.T.)	
Fuel			Gasoline	
Maximum power		[kW]	3.2	3.1
Valve mechanism			OHC	OHV



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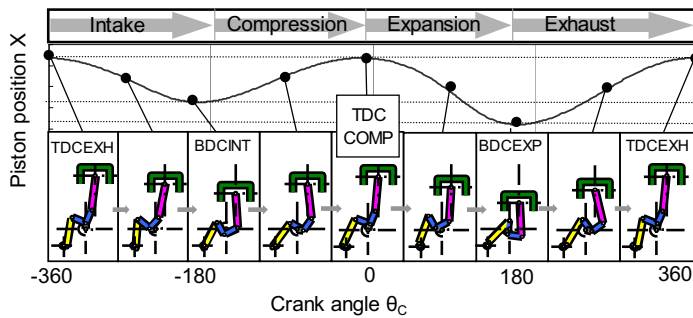


Fig.3 Piston movement through one cycle

The corresponding mass for every mass is shown as in the following.

- m_1 = Piston + Ring + Piston Pin + Connecting Rod; Corresponding Mass
- m_2 = Connecting Rod Pin + (Connecting Rod + Trigonal Link); Corresponding Mass
- m_3 = Crank Pin + Trigonal Link; Corresponding Mass
- m_4 = Swing Rod Pin + (Swing Rod + Trigonal Link); Corresponding Mass
- m_5 = Eccentric Shaft + Swing Rod; Corresponding Mass
- m_6 = Crankshaft Counterweight

Next, the summation of inertia vectors shown below is used as an indicator for evaluating the vibration properties of the multiple linkage system. The summation of inertia vectors is defined as the vector summation of inertia forces generated in every mass of the multiple linkage system at a given moment, which is to say at a given crank angle. The scalar value that represents the magnitude of those vectors is designated the summation of inertia, and its maximum value in a cycle is designated the maximum summation of inertia.

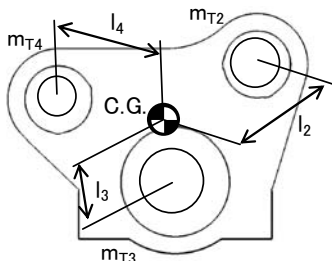


Fig.4 Trigonal link equivalent mass

Counterweight setting

Engines with the conventional piston-crank system commonly have a counterweight attached to the crankshaft to reduce the inertia forces generated by reciprocating parts such as the pistons. The adjustment of balance imposed is expressed by the below balance ratio: R_{CW} . In the case of the test engine, it was decided to use the average of the compression stroke and expansion stroke for the S_{ave} in the below equation.

$$R_{CW} [\%] = \frac{m_6 r_6 - m_3 r_c}{m_1 (S_{ave} / 2)} \quad (12)$$

General single cylinder engines commonly have balance ratios from 50% to 60%^[11]. In order to analyze the relationship between the balance ratio and vibration properties, the following items regarding existing engines were investigated.

The above summation of inertia vectors is calculated from the existing engine specifications shown in Table 1 and that magnitude is shown for every translational component (Fig. 5). Next, the balance ratio is changed to calculate the maximum summation of inertia for the existing engine. As can be seen in Fig. 6, the minimum value is reached in the vicinity of 60%. When the summation of inertia vectors for the crank angle is expressed as a vector with its origin at the center of the crankshaft, its tip plots a curve as shown in Fig. 7^[12]. The trajectory shown in the figure represents the magnitude of the maximum summation of inertia for each balance ratio. It is apparent that the summation of inertia reaches its maximum value at TDC with a balance ratio of 60%.

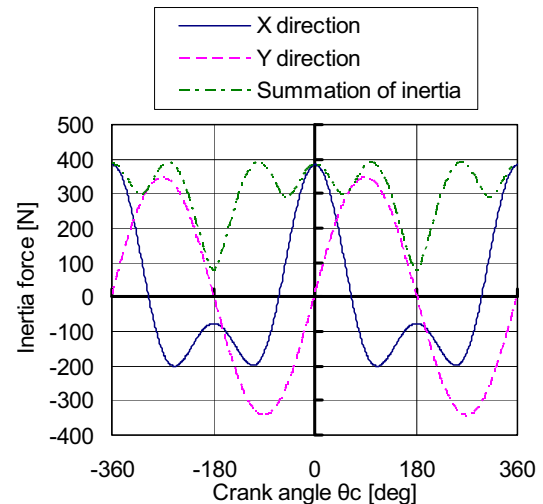


Fig.5 Inertia force through one cycle(Existing engine)

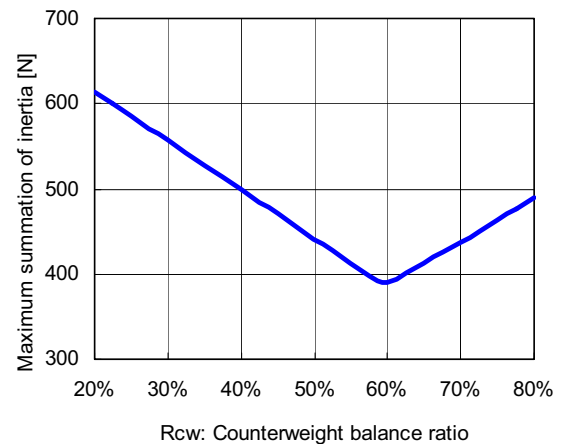


Fig.6 Maximum summation of inertia - R_{CW}