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Fundamental Researches on a Perfectly Balanced
Rotation-Reciprocation Mechanism*
Report 3 , Structural Analysis on Vibrationless
Geared Devices of a Crankshaft Rotary
Motion System, and their Vibration
and Friction Loss)

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This report relates to two kinds of vibrationless geared devices according to the basic theories in the 1st report, that is, internal geared device and eccentric geared device of crankshaft rotary motion system. Design and construction of the above-mentioned two devices are described and it is ascertained that constructional defects in the eccentric geared device of crankshaft planetary motion system are eliminated by means of these devices, and that these devices are practical. Theoretical analysis of load on each portion, friction loss and equivalent moment of inertia in these devices is made and the results of their numerical computations are shown, and from the experimental results of vibration and torque, it is proved that the inertia forces and torques are perfectly balanced and that their vibration sources are only pressure torque and friction loss torque. Generally speaking, between the two, the internal geared device a more practical one, but in the case of utilizing these devices under special heavy load, the eccentric geared device is more suitable.

1. Introduction

The eccentric geared perfectly balanced rotation-reciprocation device in the previous report⁽¹⁾ has some defects, that is, large size, complex construction and unsuitable motion of crankshaft. So, two kinds of perfectly balanced rotation-reciprocation devices, that is, an internal geared device and an eccentric geared device of crankshaft rotary motion system, are developed to eliminate the above-mentioned defects, and the fundamental researches on these devices are made.

In this report, design and construction of these two devices are described, and also structural analysis of load on each portion, friction loss, equivalent moment of inertia and their numerical computations are made. Then,

experiments on vibration, friction loss, etc. of these devices are carried out and the results are discussed.

2. Construction, design and comparison

Fig.1 shows a longitudinal sectional view of an internal geared device. As shown in Fig.1, the big end of linear reciprocating rod 2 provided with a piston 7 which is slidably engaged in cylinder 1, is rotatably supported by plane bearing 8 on rotating eccentric disc 3 which is rotatably supported by needle bearings 9 on crankpin 10 of crankshaft 4. To make enclosed chamber 11 in cylinder 1, seal 12 is secured in crank casing 13 so that connecting rod 2 is slidably engaged. Pressure transducer 14 is put on cylinder head 15. Planet gear 16 is secured on one end of the rotating eccentric disc 3. Planet gear 16 meshes with internal gear 17 having twice the number of teeth of planet gear 16 and secured in the crank casing 13. The axis of internal gear 17 coincides with axis O_1 of crankshaft 4, and axis of planet gear 16 coincides with axis O_2 of crankpin 10.

Fig.2 shows another embodiment in a longitudinal sectional view of an eccentric geared device. In Fig.2, the same reference numerals are used to represent

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the same parts or portions as those in Fig.1. On one end of rotating eccentric disc 3 rotating spur gear 18 is secured to rotate with disc 3 on axis O_2 of crankpin 10. Revolving spur gear 19 is secured on one end of crankshaft 4. Eccentric gear 24 is secured on eccentric gear shaft 20 and meshes with rotating gear 18. Eccentric gear shaft 20 and reduction gear shaft 21 are rotatably supported by crank casing 13 respectively. Spur gear 22 is secured on eccentric gear shaft 20 and meshes with reduction spur gear 23 which is secured on reduction gear shaft 21 and in turn meshes with revolving spur gear 19. Besides, the number of teeth in rotating gear 18 is equal to that in eccentric gear 24 having eccentricity 1, and the number of teeth in revolving spur gear 19 is equal to that in gear 22 secured on eccentric gear shaft 20.

And also, as shown in Figs.1 and 2, distance between axis O_3 of rotating eccentric disc 3 and axis O_2 of crankpin 10 is equal to that between axis O_2 and axis O_1 of crankshaft 4. They are 1 respectively. Rotating eccentric disc 3 is provided on both sides of rod 2 with left and right rotation balance weights W_4 and W_5 diametrically opposite to the rod end. Crank arm 5 of crankshaft 4 in the left and right sides are respectively provided with revolution balance weights W_6 and W_7 diametrically opposite to crankpin 10. These balance weights are determined such as to satisfy Eqs.(40) and (41) in the 1st report⁽⁶⁾. Then, when crankshaft 4 rotates clockwise at constant angular velocity ω , rotating eccentric disc 3 rotates counter-clockwise on axis O_2 and simultaneously revolves clockwise on axis O_1 at the same angular velocity ω . Consequently, from

Fig.1 Sectional view of internal geared device

paragraph 3.1 in the 1st report, axis O_3 performs linear reciprocating motion of 41 on an axis of cylinder. In these geared devices, crankshaft and rotating eccentric disc are integral respectively. Consequently, it can be considered that these devices are superior to the perfectly balanced rotation-reciprocation devices of crankshaft planetary motion system in the following respects. (1) Decrease in the number of elements and their size. (2) Smooth motion of crankshaft portion. (3) Decrease of rotating weight. And these two devices are superior to the known piston crank device in vibration, strength of connecting rod, abrasion of cylinder and piston, friction loss between cylinder and piston, piston speed, etc. Consequently, devices of crankshaft rotary motion system are more practical than those of crankshaft planetary motion system.

Judging from their constructions, it is necessary to take notice of the following respects in design of these devices. (1) Decrease in rotating weight and reciprocating weight. (2) Forced lubrication of bearings of crankpin and connecting rod big end. In addition to these notices, the strength of planet gear and crankpin must be increased by utilizing the high pressure angle for spur gear, profile shifted gear and selecting materials, because pitch radius of planet gear is set

Fig.2 Sectional view of eccentric geared device

half the piston stroke in internal geared device.

As the internal geared device which is a known piston crank device added with a few elements has the simplest construction of perfectly balanced rotation-reciprocation devices, it is a highly practical one. However, eccentric geared device is of use as the device for heavy load.

3. Structural analysis

3.1 Statics

Load on each portion and crank torque produced by piston pressure force are theoretically analyzed. When a pressure force F_P acts upon piston and a resisting torque M_1 is smoothly loaded on crankshaft, a free body diagram of each portion in the internal geared device is constructed as shown in Fig.3. Nomenclature in Fig.3 are as follows. A: pitch point between internal gear and planet gear, $R_{O_1O_2}$, R_{O_2A} , $R_{O_2O_3}$; position vectors, F_P : pressure force vector, F_A : tooth force vector between internal gear and planet gear, F_3 , F_2 , F_1 : static reaction force vectors acting on axis $O_3/O_2/O_1$, α : pressure angle of gear, θ : crank angle, β : unit vector in z direction, β : angle, O_1 -xyz: cartesian coordinates. Other nomenclature are the same as shown in Fig.1.

Each element in Fig.3 is in static equilibrium. Thus, the following relations are obtained.

$$\left. \begin{aligned} -F_2 &= F_P \\ F_P + F_A + F_2 &= 0, \quad R_{O_2O_3} \times F_P + R_{O_2A} \times F_A = 0 \\ -F_2 + F_1 &= 0, \quad M_1 + R_{O_1O_2} \times (-F_2) &= 0 \end{aligned} \right\} \quad (1)$$

Vectors in Eq.(1) are shown as follows by means of complex polar notation.

$$\begin{aligned} F_P &= F_P e^{i\pi}, \quad F_A = -F_A e^{i(\pi/2 - \theta - \alpha)}, \quad -F_2 = F_2 e^{i\pi} \\ F_1 &= -F_2 = -F_2 e^{i\pi}, \quad R_{O_1O_2} = l e^{i\theta}, \quad R_{O_2O_3} = l e^{i\theta} \\ R_{O_2A} &= l e^{i\theta} \end{aligned}$$

Fig.3 Statics in internal geared device

The following results are obtained by solving Eq.(1).

$$\left. \begin{aligned} F_A &= F_P \sin \theta / \cos \alpha \\ F_1 = F_2 &= \sqrt{F_P^2 + F_A^2} + 2 F_A F_P \sin(\theta \pm \alpha) \\ \sin \theta &= F_A \cos(\theta \pm \alpha) / F_2 \\ M_1 &= -K 2 l F_P \sin \theta \end{aligned} \right\} \quad (2)$$

In Eq.(2), the upper sign of compound sign is chosen when $0 \leq \theta < \pi$, and the lower sign when $\pi < \theta < 2\pi$.

The static force in the eccentric geared device are obtained in the same way as in the internal geared device. As shown in Fig.4, a free body diagram of each portion in the eccentric geared device is constructed. Nomenclature in Fig.4 are as follows. O_4 : axis of eccentric gear shaft, O_5 : axis of reduction gear shaft, O_6 : center of eccentric gear, B, C, D: pitch points of gear, R_1 , R_2 , R_3 : pitch radii of gear, F_1 , F_2 , F_3 , F_4 , F_5 : static reaction force vectors acting on axis $O_1/O_2/O_3/O_4/O_5$, F_B , F_C , F_D : tooth force vectors at point B/C/D, M_1 : resisting torque on axis O_1 , δ , ϵ , η , ζ , φ : angles, α : pressure angle of gear, θ : crank angle, ω : constant angular velocity of reduction gear ($= R_1 \omega / R_3$), O_1 -xyz: cartesian coordinates, K : unit vector in z direction, R_{MN} ($M, N = O_1, O_2, O_3, O_4, O_5, B, C, D$): position vector of N from M. Other nomenclature are the same as shown in Fig.2.

Fig.4 Statics in eccentric geared device

On an example when a pressure force F_P acts upon piston and resisting torque M_i is smoothly loaded on axis O_1 , static force analysis in the eccentric geared device is made. Each element in Fig. 4 is in static equilibrium. Thus, the following relations are obtained.

$$\left. \begin{aligned} -F_3 &= F_P \\ -F_3 + F_2 + F_B &= 0, \quad R_{O_1 O_2} \times (-F_3) + R_{O_2 O_3} \times F_B = 0 \\ F_1 - F_2 - F_D &= 0, \quad M_1 + R_{O_1 O_2} \times (-F_2) + R_{O_2 O_3} \times (-F_D) = 0 \\ F_3 - F_C + F_D &= 0, \quad R_{O_3 O_4} \times F_D + R_{O_4 O_5} \times (-F_C) = 0 \\ -F_B + F_C + F_4 &= 0, \quad R_{O_4 O_5} \times (-F_B) + R_{O_5 O_6} \times F_C = 0 \end{aligned} \right\} \quad (3)$$

where $F_1 = F_1 e^{-i\theta}$, $F_2 = F_2 e^{-i\eta}$, $-F_3 = F_3 e^{i\pi}$
 $F_4 = F_4 e^{i\varphi}$, $F_5 = F_5 e^{i(\pi-\delta)}$, $F_6 = F_6 e^{i(\pi/2-\alpha)}$
 $F_C = F_C e^{i(\pi/2+\delta+\alpha)}$, $F_D = F_D e^{i(\pi/2-\delta-\alpha)}$
 $R_{O_1 O_2} = l e^{-i\theta}$, $R_{O_2 O_3} = l e^{i\theta}$, $R_{O_3 O_4} = R_1 e^{i(\pi-\delta)}$
 $R_{O_4 O_5} = R_2 e^{i\pi}$, $R_{O_5 O_6} = l e^{-i\theta} + R_3$
 $R_{O_6 O_7} = R_1 e^{i\delta}$, $R_{O_7 O_8} = R_3 e^{i(\pi+\delta)}$, $R_{O_8 O_9} = R_3 e^{-i\delta}$

The following results are obtained by solving Eq. (3).

$$\left. \begin{aligned} F_3 &= F_P \\ F_B &= F_3 l \sin \theta / R_2 \cos \alpha \\ F_2 &= \sqrt{F_3^2 + F_B^2 - 2 F_3 F_B \sin \alpha} \\ \sin \eta &= F_B \cos \alpha / F_2 \\ F_D &= F_C = F_B \{ (\cos(\theta \pm \alpha) + R_2 \cos \alpha) / R_1 \cos \alpha \\ F_5 &= 2 F_C \cos(\pm \alpha \pm \delta) \\ \zeta &= \pi / 2 \\ F_1 &= \sqrt{F_2^2 + F_D^2 - 2 F_2 F_D \sin(\alpha \pm \delta - \eta)} \\ \sin \epsilon &= \{ F_2 \sin \eta + F_D \cos(\pm \alpha \pm \delta) \} / F_1 \\ F_4 &= \sqrt{F_1^2 + F_C^2 - 2 F_1 F_C \cos(\pm 2\alpha \pm \delta)} \\ \sin \varphi &= \{ F_1 \cos \alpha - F_C \cos(\pm \alpha \pm \delta) \} / F_4 \\ M_1 &= -K_2 l F_P \sin \theta \end{aligned} \right\} \quad (4)$$

In Eq. (4), the upper sign of compound sign is chosen when $0 \leq \theta < \pi$, and the lower sign when $\pi < \theta < 2\pi$. Besides, Table 1 shows the results of static force analysis when an other shaft is chosen as an output shaft.

Then, the moment vector M_{ii} on axis O_i by inertia force I_i of reciprocating weight W_i , is given as follows by substituting $|I_i|$ in place of F_P in Eqs. (2) or (4).

$$M_{ii} = -K_2 W_i l^2 \omega^2 \sin 2\theta / g \quad (5)$$

3.2 Bearing load

Bearing loads due to inertia in eccentric geared device are given in paragraph 6.2 of 1st report, because balanced structure in the eccentric geared device is the same as that in the internal geared device. And the total bearing load can be obtained as the sum of the static force and the inertia force acting on them.

Consequently, from Eq. (2), total bearing loads F_{3B} , F_{2B} and F_{1B} in the internal geared device are given as follows:

$$F_{3B} = I_{3B} - F_3, \quad F_{2B} = I_{2B} - F_2, \quad F_{1B} = -F_1 \quad (6)$$

Where I_{3B} , I_{2B} : bearing load due to inertia on axis O_3/O_2 .

From Eq. (4), total bearing loads F_{3B} , F_{2B} , F_{1B} , F_{4B} and F_{5B} in the eccentric geared device are given as follows:

$$\left. \begin{aligned} F_{3B} &= I_{3B} - F_3, \quad F_{2B} = I_{2B} - F_2, \quad F_{1B} = -F_1 \\ F_{4B} &= -F_4, \quad F_{5B} = -F_5 \end{aligned} \right\} \quad (7)$$

where I_{3B} , I_{2B} : bearing load due to inertia on axis O_3/O_2 .

3.3 Friction loss

Friction losses due to the above-mentioned bearing loads and the force of piston against cylinder wall are theoretically analyzed. Thus, influences of the temperature, accuracy, lubrication, etc. and friction loss of gears, pump, oil-seal, oil stirring, etc. are not dealt with.

Horsepower P_f and crank torque T_f of friction loss in the eccentric geared device, are obtained by referring to Fig. 3

$$\begin{aligned} T_f &= 75 P_f / \omega \\ &= 2 \mu_p l N |\sin \theta| + \mu_1 F_{3B} r_3 + 2 \mu_2 F_{2B} r_2 + \mu_1 F_{1B} r_1 \end{aligned} \quad (8)$$

Where μ_i ($i=1,2,3,P$): coefficient of friction, r_i ($i=1,2,3$): mean radius of bearing, F_{iB} ($i=1,2,3$): total bearing load, F_{Pf} : friction force between cylinder and piston, N : expanding force of piston ring, subscript $i=1,2,3,P$: crankshaft bearing/ crankpin bearing/ big end bearing of connecting rod/ piston and cylinder. Other nomenclature are the same as shown in Fig. 4. The mean friction horsepower and crank torque, P_{mf} and T_{mf} , are

$$\begin{aligned} T_{mf} &= (1/2\pi) \int_0^{2\pi} T_f d\theta = 75 P_{mf} / \omega \\ &= 4 \mu_p l N / \pi + \mu_1 F_{3B} r_3 + 2 \mu_2 F_{2B} r_2 + \mu_1 F_{1B} r_1 \end{aligned} \quad (9)$$

where F_{iB} ($i=1,2,3$): mean value of F_{iB} ($i=1,2,3$).

Table 1 Results of static force analysis in eccentric geared device

Output Item	Eccentric gear shaft	Reduction gear shaft
F_3	F_P	F_P
F_B	$F_3 l \sin \theta / R_2 \cos \alpha$	$F_3 l \sin \theta / R_2 \cos \alpha$
F_2	$\sqrt{F_3^2 + F_B^2 - 2 F_3 F_B \sin(\pm \alpha)}$	$\sqrt{F_3^2 + F_B^2 - 2 F_3 F_B \sin(\pm \alpha)}$
$\sin \eta$	$F_B \cos \alpha / F_2$	$F_B \cos \alpha / F_2$
F_D	$F_2 l \sin(\theta - \eta) / R_1 \cos \alpha$	$F_2 l \sin(\theta - \eta) / R_1 \cos \alpha$
F_C	F_D	$F_B \{ (\cos(\theta \pm \alpha) + R_2 \cos \alpha) / R_1 \cos \alpha \}$
$\sin \zeta$	$-1, \quad \zeta = -\pi/2$	$\{ F_C \cos(\pm \alpha \pm \delta) - F_D \cos(\pm \alpha \pm \delta) \} / F_5$
F_5	$2 F_C \cos(\pm \alpha \pm \delta)$	$\sqrt{F_C^2 + F_D^2 - 2 F_C F_D \cos 2\delta}$
F_1	$\sqrt{F_2^2 + F_D^2 - 2 F_2 F_D \sin(\pm \alpha \pm \delta \mp \eta)}$	$\sqrt{F_2^2 + F_D^2 - 2 F_2 F_D \sin(\pm \alpha \pm \delta \mp \eta)}$
$\sin \epsilon$	$\{ F_2 \sin \eta - F_D \cos(\pm \alpha \pm \delta) \} / F_1$	$\{ F_2 \sin \eta - F_D \cos(\pm \alpha \pm \delta) \} / F_1$
F_4	$\sqrt{F_B^2 + F_C^2 + 2 F_B F_C \cos \delta}$	$\sqrt{F_B^2 + F_C^2 - 2 F_B F_C \cos(\pm 2\alpha \pm \delta)}$
$\sin \varphi$	$\{ F_B \cos \alpha + F_C \cos(\pm \alpha \pm \delta) \} / F_4$	$\{ F_B \cos \alpha - F_C \cos(\pm \alpha \pm \delta) \} / F_4$
M	$2 F_3 l \sin \theta$	$2 F_3 l \sin \theta \cdot R_3 / R_1$

And, horsepower P_f and crank torque T_f of friction loss, and the mean values of P_f and T_f , P_{mf} and T_{mf} , in the eccentric geared device are given by referring to Fig.4.

$$\left. \begin{aligned} T_f &= 75P_f/\omega \\ &= 2\mu_1 N \sin\theta + \mu_2 F_{10} r_1 + 2\mu_3 F_{20} r_2 + \mu_4 F_{10} r_1 + \mu_4 F_{20} r_2 + \mu_5 F_{30} r_3 R_3/R_2 \\ T_{mf} &= 75P_{mf}/\omega \\ &= 4\mu_1 N/\pi + \mu_2 F_{10} r_1 + 2\mu_3 F_{20} r_2 + \mu_4 F_{10} r_1 + \mu_4 F_{20} r_2 + \mu_5 F_{30} r_3 R_3/R_2 \end{aligned} \right\} (10)$$

where μ_i ($i=1,2,3,4,5,P$):coefficient of friction, r_i ($i=1,2,3,4,5$):mean radius of bearing, F_{i0} ($i=1,2,3,4,5$):total bearing load, F_{pf} :friction force between cylinder and piston, N :expanding force of piston ring, R_1, R_2 :pitch radii of revolving gear of reduction gear, subscript $i=1,2,3,4,5,P$:crankshaft bearing/crankpin bearing/big end bearing of connecting rod/eccentric gear shaft bearing/reduction gear shaft bearing/piston and cylinder, $\bar{F}_{i0}$ ($i=1,2,3,4,5$):mean value of F_{i0} ($i=1,2,3,4,5$).

From the above-mentioned analysis, it is clear how friction losses are affected by the load on each portion in these devices.

3.4 Equivalent moment of inertia on crankshaft

Equivalent moments of inertia of these devices are analyzed by means of the basic theory presented in paragraph 3.3 of 1st report.

Equivalent moment of inertia on crankshaft, J_e in the internal geared device is

$$J_e = J_1 + J_2 + W_r l^2/g + W_{rot} l^2/g \quad (11)$$

where W_r :reciprocating weight, W_{rot} :rotating weight, l :one fourth of piston stroke, J_1, J_2 :moment of inertia on its axis of crankshaft on which all parts are secured/of rotating eccentric disc with rotation balance weights, g :gravitational acceleration.

Equivalent moment of inertia on crankshaft, J_e in the eccentric geared device is

$$J_e = J_1 + J_2 + J_4 + n^2 J_5 + W_r l^2/g + W_{rot} l^2/g \quad (12)$$

where W_r :reciprocating weight, W_{rot} :rotating weight, l :one fourth of piston stroke, J_1, J_2, J_3, J_4, J_5 :moment of inertia on its axis of crankshaft on which all parts are secured/of rotating eccentric disc with rotation balance weights/of eccentric gear shaft on which all parts are secured/of reduction gear shaft on which all parts are secured, g :gravitational acceleration.

4. Experimental result and discussion

4.1 Pressure force in compression chamber

Fig.5 shows a pressure measuring set up in compression chamber. As shown in Fig.5, testing device is driven by a motor and the pressure in compression chamber is detected by pressure transducer.

Fig.6(a) shows the pressure force acting on piston (hereinafter, often

Fig.5 Appearance of pressure measuring set up compression chamber

Fig.6 Pressure force and crank torque in eccentric geared device

designated as "load"), and the crank torque calculated by substituting the pressure force in Eq.(4) at crank speed 2500 rpm in the eccentric geared device and its Fourier analysis. The wave form of the pressure force is almost the same at various crank speeds, but its maximum value becomes large as crank speed increases. So, the relation between the maximum pressure force and the crank speed is shown in Fig.6(b). As shown in Fig.6, resonances between vibrating table and crank torque at load occur at the crank speeds ranging from 1000 to 5000 rpm, because crank torque at load has the higher harmonics to crank speed. Therefore, the results of vibration test should be evaluated carefully. And those in the internal geared device are almost the same as the above.

4.2 Amplitude of vibration

The dynamic balancings for rotation and revolution are made by means of the balancer for one rotor in accordance with the balanced conditions: that is, Eqs.(40)

and (41) in 1st report. And in the eccentric geared device, eccentric gear shaft and reduction gear shaft on which all parts are secured, are also balanced on each axis. These qualities of balance are 2nd grade in JIS. at crank speed 1085 rpm.

Figs.7 and 8 show the vibration amplitudes of these devices measured according to the method explained in 2nd report. In Fig.7, primary amplitude shows a component having the same period as crank speed in actual vibration amplitude: that is, total amplitude. As shown in Fig.7, the primary amplitudes of these devices are below 2μ . Judging from the frequency characteristics of vibrating table and the Fourier analysis of crank torque in Fig.6, the large total amplitudes at load result from the coincidence of natural frequency of the vibrating table with higher harmonics of the crank torque. Besides, it is caused by the difference in accuracy of machining and assembling that the amplitudes of the internal geared device are little larger than those of the eccentric geared device.

Fig.7 Amplitude of vibration (Primary amplitude)

Fig.8 Amplitude of vibration (Total amplitude)

Referring to the characteristics of the vibrating table, the vibration amplitudes in these devices correspond to those of rotor having unbalance, 150 gmm. However, the unbalances in cylinder direction calculated by the equation, $2 \times [\text{reciprocating weight}] \times [\text{eccentricity (one fourth of piston stroke)}]$, are 6600 gmm in the internal geared device and 6320 gmm in the eccentric geared device respectively. The results shown in Figs. 7 and 8, are affected by the play between shaft and bearing, friction force and torque, accuracy of machining and assembling. Consequently, it can be considered that the inertia force due to reciprocating weight is perfectly balanced theoretically in these devices.

4.3 Torsional vibration

Fig.9 shows a torsional vibration measuring set up. As shown in Fig.9, testing device is driven by a motor and torsional vibration of crankshaft is detected by torsional vibration transducer secured to crankshaft.

Fig.9 Appearance of torsional vibration measuring set up for crankshaft

Fig.10 shows the wave forms of torsional vibration at crank speed 3000 rpm and their Fourier analyses. Fourier analyses are made up to 12th harmonics and six terms are shown here in order of its magnitude.

Fig.11 shows the relation between the torsional vibration amplitude of crankshaft and the crank speed. As shown in Fig.11, the amplitudes at no load are very small and almost constant at various crank speeds. And the amplitude at load becomes small as the crank speed increases. The torsional vibrations of crankshaft in these devices seem to be mainly caused by the crank torque due to the pressure. Since equivalent moment of inertia on crankshaft in the internal geared device is smaller than that in the eccentric geared device, the amplitudes in the former are larger than those in the latter.

Fig.11 Amplitude of torsional vibration of crankshaft

Internal geared device				Eccentric geared device			
	Wave form	Fourier analysis		Wave form	Fourier analysis		
		Or-Amplitude	der-tude		Or-Amplitude	der-tude	
Load		1	0.52		1	0.53	
		2	0.27		2	0.10	
		3	0.03		3	0.06	
		4	0.05		4	0.05	
		7	0.05		5	0.05	
		10	0.05		9	0.04	
No load		1	0.04		1	0.04	
		2	0.04		2	0.01	
		3	0.01		3	0.01	
		4	0.01		4	0.01	
		6	0.01				
		7	0.01				
One rotation of crank				One rotation of crank			

Fig.10 Wave form of torsional vibration amplitude of crankshaft and its Fourier analysis (Crank speed 3000 rpm)

Fig.12 Appearance of friction loss measuring set up

4.4 Friction loss and crank torque

Fig. 12 shows a friction loss measuring set up. And Fig.13 shows the experimental results of friction losses in these devices. The friction loss torque becomes larger as the crank speed increases. It is caused by the increases of the inertia load. And it is produced by an increase in the load due to the pressure force that the friction loss at load is larger than that at no load. The friction loss in the internal geared device is about 80 percent of that in the eccentric geared device.

The friction losses of these devices are larger than that of the known piston crank device. However, the friction losses between cylinder and piston in the former

are far smaller than in the latter because of no side force against the cylinder wall in the former. Consequently, the friction losses of these devices are almost the same as that in the known piston crank device when they are used in a reciprocating engine, a reciprocating compressor, etc. in which heavy loads act on piston.

Fig.14 shows a crank torque measuring set up. Trigger marker is set so as to make the wave form of crank torque start at the top dead point in testing device. The wave forms of crank torque and their Fourier analyses are shown in Fig.15. Fourier analyses are made up to 12th harmonics and six terms are shown here in order of its magnitude. The crank torques at load are similar to that in Fig.6 in magnitude and shape. And the crank torques at no load are very small and mainly caused by the friction loss torque.

Fig.13 Experimental value of friction loss

Fig.14 Appearance of crank torque measuring equipment

Fig.15 Wave form of crank torque and its Fourier analysis (Crank speed 1800 rpm)

Table 2 Specification of internal geared device and eccentric geared device

Device	Internal geared device	Eccentric geared device
Item		
1 (mm)	10	10
W_r (g)	330	316
W_{rot} (g)	883	1110
W_{rev} (g)	2393	3250
R_1 (mm)		40
R_2 (mm)		25
R_3 (mm)		50
R_p (mm)	20	
R_1 (mm)	40	
r_1 (mm)	32	32
r_2 (mm)	13	13
r_3 (mm)	28	28
r_4 (mm)		32
r_5 (mm)		32
α (deg)	30	20
γ (deg)		$64^{\circ}28'$
δ (deg)		$57^{\circ}46'$
h (mm)	5.96	3.98

The crank torque by the inertia force due to reciprocating weight at crank speed 1800 rpm, is calculated by Eq.(9) and its value is about $0.23\sin 2\theta$ kgm. However, as shown in Fig.15 the second harmonics of the crank torque in each device is about 0.025 kgm. So, it is ascertained that the crank torques by the inertia force due to reciprocating weight in these devices are perfectly balanced theoretically.

From the results of paragraphs 4.1~4.4, the vibration isolation of these devices seems to be easy, because their vibrations are caused only by the pressure torque and the friction loss torque.

5. Numerical computation and discussion

5.1 Equivalent moment of inertia on crankshaft

Fig.16 Load on each portion due to pressure force

Specifications of these two devices are shown in Table 2. The equivalent moments of inertia of these devices are obtained as shown in Table 3. by substituting the value of Table 2 in Eqs.(11) and (12). The equivalent moments of in-

Table 3 Moment of inertia of each portion (gcm^2)

Device Item	Internal geared device	Eccentric geared device
J_1	11610	17150
J_2	2570	5270
J_4		4900
$n^2 J_5$		1830
J_e	15400	30600

Fig.17 Load on each portion due to inertia

ertia of these devices are large in comparison with that of known piston crank device and a flywheel is unnecessary for these devices in general.

5.2 Load on each portion

Fig.16 shows the loads which are calculated by substituting the pressure force and the values of Table 2 in Eqs.(6) and (8). As shown in Fig.16, the direction and the magnitude of these loads change smoothly and continuously and their magnitudes are below that of the pressure force. Consequently, it is clear that the operation of these devices takes place without difficulty. And a side force to cylinder wall in known piston crank device is turned into tooth force of gears.

Fig.17 shows the loads due to inertia. As shown in Fig.17, the loads on big end bearing of connecting rod and crankpin bearing become larger as the crank speed increases.

5.2 Friction loss

Fig.18 shows the mean friction loss torque calculated on the assumption of $N=0$, $\mu_i(i=1,2,3,4,5)=0.02$, by substituting the values of Table 2 and the total bearing loads in Eqs.(13) and (14). The friction loss at load is almost the same as that at load in crank speed, 5000 rpm or over. It shows that the friction loss torque is mainly caused by the inertia forces at high crank speed. The calculated values in Fig.18 agree qualitatively with the experimental values in Fig.13. And it results from the assumption for analysis in paragraph 3.3 and the assumption of $N=0$, $\mu_i=0.02$ that the former is smaller than the latter.

As it is clear from the results of paragraphs 5.2 and 5.3, it is important to decrease reciprocating weight and to lubricate bearings of crankpin and connecting rod big end forcedly.

Fig.18 Calculated value of friction loss torque

6. Conclusions

Design and construction of two kinds of perfectly balanced rotation-reciprocation devices, that is, the internal geared device and the eccentric geared device of crankshaft rotary motion system, are described. It is made clear that the defects in the eccentric geared device of crankshaft planetary motion system: that is, its complex construction and unsuitable motion of each element, are eliminated in these devices, and that these devices are practical.

Then, from structural analyses, the theoretical equations for load on each portion, friction loss and equivalent moment of inertia are obtained. And, through their numerical computations, it is made clear that there are no side pressure forces to cylinder, and that it is important to decrease reciprocating weight and rotating weight.

Besides, through the vibration tests, it is proved that the inertia force and the torque due to reciprocating weight are perfectly balanced and the vibration isolation of these devices is easy because their vibrations are caused only by the pressure torque and the friction loss torque.

Generally speaking, the construction of the internal geared device is simpler than that of the eccentric geared device, and the internal geared device is a highly practical one. And the eccentric geared device is of use as the device for heavy load.

References

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