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Fundamental Researches on a Perfectly Balanced

Rotation-Reciprocation Mechanism.*

(Report 4, Static Balanced Theory of Rotation-
Reciprocation Mechanism, and Comparison between
Perfectly Balanced Rotation-Reciprocation Device
and Known Piston-Crank Devices)

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The static balanced theory of a rotation-reciprocation mechanism is established, and it is described that this theory is general one which can be applied to the perfectly balanced rotation-reciprocation mechanism and the known piston-crank devices. From the theoretical and experimental comparisons between the perfectly balanced rotation-reciprocation mechanism and the four kinds of known balanced piston-crank devices (containing the static balanced piston-crank device), it is ascertained that the former is much superior to the latter in the vibration, the friction loss between a cylinder and a piston, and the load to a connecting rod. However, the former is a little more complex than the latter in the construction. And it is clear that the static balanced piston-crank device can not be put to practical use without any improvements.

1. Introduction

Inertia force and moment balancing on a known piston-crank device which is provided with a newly devised balanced construction (to be designated as a static balanced device), are theoretically analysed. It is described that the above-mentioned analysis can be applied to both the perfectly balanced rotation-reciprocation device (to be designated as the perfectly balanced device) and the known balanced piston-crank devices. And the comparison between the static balanced device, the perfectly balanced device and three kinds of typical known balanced piston-crank devices^(a) (that is, a rotation balanced device, a half balanced device and an over balanced device), are discussed on the inertia force and moment, the vibration, the load on each portion, etc. The practicality of the static balanced device is examined.

Fig.1 Longitudinal sectional view of static balanced device

1 Cylinder	2 Piston
3 Pistonpin	4 Connecting rod
5 Crankpin	6 Crank arm
7 Crankshaft	8,9,10 Bearing
11 Pressure transducer	
12 Compression chamber	

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2. Construction of Testing Device and its Specification

Fig.1 shows the longitudinal sectional view of the static balanced device at the top dead point. The balancing method of this device is described. Balance weight W_A for oscillation is secured on connecting rod 4 and diametrically opposite to axis O_2 of crankpin 5. Balance weight W_B for rotation is secured on crank arm 6 and diametrically opposite to axis O_1 of crankshaft 7.

The constructions and the balancing methods of the rotation balanced device, the half balanced device and the over balanced device are wellknown, and also those of the perfectly balanced device are presented in Report 3⁽²⁾. The principal specifications which are common to these testing devices are shown in Table 1.

3. Theoretical Analysis

It is assumed throughout paragraphs 3.1~3.3 that frictional forces among members, and the effects of the play between shafts and bearings, gear tooth errors, the backlash of gears and various errors, can be neglected.

3.1 Static Balanced Theory of Known Piston-Crank Device

As shown in Fig.2, the weight of a connecting rod with an oscillation balance weight is replaced by concentrated weights, that is, by concentrating oscillation balance weight W_A at its center of gravity A and dividing weight W_C of the connecting rod except W_A into W_{C2} at crankpin axis O_2 and W_{CP} at piston pin axis P. The relation between these weights are

$$W_C = W_{CP} + W_{C2}, \quad W_{C2}b = W_{CP}a \quad (1)$$

Symbols in Fig.2 are as follows. O_1 = crank axis, θ = crank angle, ϕ = oscillation angle of connecting rod, G = center of gravity of connecting rod, r = crank radius, L = length of connecting rod, a, b = distance, ω = angular velocity of crank.

The concentrated weights system shown in Fig.2 are dynamically equivalent to the connecting rod with weight W_A , when the following moment of inertia J_m is added to the former.

$$J_m = J_A + J_C, \quad J_C = W_C(k^2 - ab)/g \quad (2)$$

where J_A = moment of inertia about A of

Table 1 Specification of testing device

Number of cylinder	1
Type of cylinder	For 2 stroke engine
Stroke of piston	40. mm
Bore of cylinder	40 mm
Compression ratio	7

weight W_A , k = radius of gyration on moment of inertia about G of connecting rod except W_A , g = gravitational acceleration.

Referring to Fig.2, the weight of the static balanced device is replaced by the concentrated weights as shown in Fig.3, and also the inertia force due to each weight is obtained. Symbols in Fig.3 are as follows. W_B = sum of rotation balance weight and crankpin weight, $W_2 = W_{C2}$ in Fig.2, W_P = sum of piston weight and W_{CP} in Fig.2, B = center of gravity of W_B , R_{O_2B} , R_{O_2A} ($|R_{O_2A}| = R_A$), R_{O_2P} , R_{O_2B} , R_{O_2A} , R_{O_2B} ($|R_{O_2B}| = R_B$) = position vector, I_P , I_A , I_2 , I_B = inertia force vector due to $W_P/W_A/W_2/W_B$, x, y, z = cartesian coordinate, $-k$ = unit vector in the z direction. Other symbols are the same as those in Fig.2. From the assumption that angular velocity ω is constant, the concentrated weights in Fig.3 to which the modified moment of inertia J_m in Eq.(1) is added, is considered to be dynamically equivalent to the connecting rod with the oscillation balance weight.

Inertia force vectors obtained as follows:

$$\left. \begin{aligned} I_P &= -(W_P/g)\ddot{R}_{O_2P} = (W_P/g)\{r\omega^2 e^{i\theta} + (i\dot{\phi} + \phi^2)Le^{-i\phi}\} \\ I_2 &= -(W_2/g)\ddot{R}_{O_2A} = (W_2/g)r\omega^2 e^{i\theta} \\ I_A &= -(W_A/g)\ddot{R}_{O_2A} = (W_A/g)\{r\omega^2 e^{i\theta} - (i\dot{\phi} + \phi^2)R_A e^{-i\phi}\} \\ I_B &= -(W_B/g)\ddot{R}_{O_2B} = -(W_B/g)R_B \omega^2 e^{i\theta} \end{aligned} \right\} \quad (3)$$

The vector sum of the above inertia forces is an unbalanced inertia force vector U and is expressed as follows:

$$\begin{aligned} U &= I_P + I_2 + I_A + I_B \\ &= \{(W_P + W_2 + W_A)r - W_B R_B\}\omega^2 e^{i\theta}/g \\ &\quad + (W_P L - W_A R_A)(i\dot{\phi} + \phi^2)e^{-i\phi}/g \end{aligned} \quad (4)$$

From Eq.(4), the complete balancing condition of the inertia forces is

$$(W_P + W_2 + W_A)r = W_B R_B, \quad W_P L = W_A R_A \quad (5)$$

Fig.2 Connecting-rod in static balanced device

Fig.3 Dynamics in static balanced device

Eq. (5) is designated as a static balanced condition.

Moments due to the inertia forces about axis O_2 and O_1 , M_{i2} and M_{i1} are

$$\left. \begin{aligned} M_{i2} &= R_{02P} \times I_P + R_{02A} \times I_A - k J_m \ddot{\phi} \\ &= k M_{i2} \\ &= -k \{ r \omega^2 (W_P L - W_A R_A) \sin(\theta + \phi) \\ &\quad + \ddot{\phi} (W_P L^2 + W_A R_A^2 + g J_m) \} / g \\ M_{i1} &= R_{01O_2} \times (I_P + I_A + M_{i2} / L \cos \phi) \\ &= -k \{ r (W_P L - W_A R_A) \{ \ddot{\phi} \cos(\theta + \phi) - \dot{\phi}^2 \sin(\theta + \phi) \} / g \\ &\quad - P \cos \theta \{ r \omega^2 (W_P L - W_A R_A) \sin(\theta + \phi) \\ &\quad + \ddot{\phi} (W_P L^2 + W_A R_A^2 + g J_m) \} / g \cos \phi \} \end{aligned} \right\} \quad (6)$$

The unbalanced inertia moment M_{i1} in the static balanced device is given by substituting Eq. (5) in Eq. (6).

$$M_{i1} = k (W_P L^2 + W_A R_A^2 + g J_m) \ddot{\phi} \rho \cos \theta / g \cos \phi \quad (7)$$

As it is clear from Eq. (7), the unbalanced inertia moment is due to the oscillation of the connecting rod.

3.2 Application of Static Balanced Theory to Various Kinds of Balanced Rotation-Reciprocation Devices

It is obvious that the static balanced theory can be applied to known balanced piston-crank devices. As it is clear from the comparison between the perfectly balanced device of Report 3 and the static balanced device, the perfectly balanced theory in Report 1⁽³⁾ is considered as a particular case, that is, $\phi = \dot{\phi} = 0$ and $L = r = 2l$ of the static balanced theory. Where the contents of some nomenclature must be modified according to their constructional differences. Then, the static balanced theory coincides with the perfectly balanced theory.

Practical balanced conditions in known piston-crank devices are

$$W_A = 0, \quad R_B W_B = r (W_2 + \lambda W_P), \quad 0 \leq \lambda \leq 1 \quad (8)$$

As some representative examples, a rotation balance ($\lambda = 0$), a half balance ($\lambda = 0.5$) and an over balance ($\lambda = 1$) are taken up here. By substituting the above three kinds of balanced conditions in Eqs. (4) and (6) respectively, the unbalanced inertia forces and moment vectors are obtained as shown in Table 2. In the vector U of Table 2, real part gives the component of the cylinder

Fig. 4 Statics in known piston-crank device

axis direction and imaginary part the component of the direction perpendicular to the cylinder axis. In Table 2, the unbalanced forces are approximate values except the perfect balance and the static balance, and the unbalanced moments are approximate values except the perfect balance.

As shown in Table 2, both inertia forces and moments are completely balanced in the perfectly balanced device. As to unbalanced forces in the rotation balanced device, the half balanced device and the over balanced device, their directions are only controlled.

3.3. Static Force and Bearing Load

Static force analysis on the known piston-crank device is briefly described in order to compare with the perfectly balanced device.

When a pressure force F_P acts upon a piston and a resisting moment M_1 is smoothly loaded on a crankshaft, a free body diagram of each portion in the known piston-crank device is constructed as shown in Fig. 4. Symbols in Fig. 4 are as follows. F_N, F_1, F_2 = reaction force vector acting on axis $P/O_2/O_1$, γ, δ = angle, r = crank radius, L = length of connecting rod. Other symbols are the same as Fig. 3.

Forces acting on each element in Fig. 4 is in static equilibrium. Thus, the following relations are obtained.

$$\left. \begin{aligned} F_P + F_N + F_2 &= 0, & R_{02P} \times F_P &= 0 \\ F_2 &= F_1, & R_{01O_2} \times (-F_2) + M_1 &= 0 \end{aligned} \right\} \quad (9)$$

Table 2 Inertia unbalance of each device

Type	Balanced condition	Unbalanced force U	Unbalanced moment M
Perfect balance	$\theta = \phi, \dot{\phi} = 0$ $L = r = 2l, LW_P = R_A W_A$ $r(W_P + W_2 + W_A) = R_B W_B$	0	0
Static balance	$LW_P = R_A W_A$ $r(W_P + W_2 + W_A) = R_B W_B$	0	$-k J_m \rho^2 \omega^2 \sin 2\theta / 2$
Rotation balance	$W_A = 0$ $r W_2 = R_B W_B$	$W_P r \omega^2 (\cos \theta + \rho \cos 2\theta) / g$	$k M$
Half balance	$W_A = 0$ $r(W_2 + W_P / 2) = R_B W_B$	$W_P r \omega^2 \{ (\cos \theta / 2 + \rho \cos 2\theta) - i \sin \theta / 2 \} / g$	$k M$
Over balance	$W_A = 0$ $r(W_2 + W_P) = R_B W_B$	$W_P r \omega^2 (\rho \cos 2\theta - i \sin \theta) / g$	$k M$
Note, $M = W_P r^2 \omega^2 (\sin 2\theta / 2 - \rho \sin \theta / 4 + 3\rho \sin 3\theta / 4) / g - J_c \rho^2 \omega^2 \sin 2\theta / 2, J_1 = W_P L^2 / g + W_A R_A^2 / g + J$			

The following results are obtained by solving Eq. (9).

$$\left. \begin{aligned} F_N &= F_P \tan \phi \\ F_1 = F_2 &= F_P \sec \phi, \quad \gamma = \delta = \phi \\ M_1 &= F_P r \sec \phi \sin(\theta + \phi) \end{aligned} \right\} \quad (10)$$

where $F_N = F_N e^{i\theta}$, $F_P = F_P e^{i\theta}$, $F_2 = F_2 e^{-i\theta}$, $M_1 = \kappa M_1$
Bearing loads due to inertia, I_{Pb} on axis P, I_{2b} on axis O_2 and I_{1b} on axis O_1 are as follows:

$$\left. \begin{aligned} I_{Pb} &= (1 - W_{CP}/W_P) I_P + I_R \\ I_{2b} &= I_P + I_2 + I_A + I_R \\ I_{1b} &= I_{2b} + I_B = U + I_R \\ I_R &= (M_{12}/L \cos \phi) e^{i(\theta + \phi)} \end{aligned} \right\} \quad (11)$$

where I_R = side force vector from cylinder to piston due to inertia.

From Eqs. (10) and (11), total bearing loads, F_{Pb} on axis P, F_{2b} on axis O_2 and F_{1b} on axis O_1 are obtained as follows:

$$F_{Pb} = F_P + F_N + I_{Pb}, \quad F_{2b} = -F_2 + I_{2b}, \quad F_{1b} = -F_1 + I_{1b} \quad (12)$$

The total side force vector from the cylinder to the piston F_S is

$$F_S = F_N + I_R \quad (13)$$

3.4 Friction Loss

The friction loss due to total bearing loads and the friction force between a cylinder and a piston are analyzed in the known piston-crank device (containing the static balanced device). Then, the influence of temperature, accuracy, lubrication etc. on a friction loss, and the friction losses of gears, pump, oil-seal, oil stirring etc. are neglected.

The friction force F_{Pf} between the cylinder and the piston, the friction torque T_{Pf} on the piston pin bearing, T_{2f} on the crankpin bearing, and T_{1f} on the crankshaft bearing are

$$\left. \begin{aligned} F_{Pf} &= \mu_P |F_S|, \quad T_{Pf} = \mu_P |F_{Pb}| r_P \\ T_{2f} &= \mu_2 |F_{2b}| r_2, \quad T_{1f} = \mu_1 |F_{1b}| r_1 \end{aligned} \right\} \quad (14)$$

where, $\mu_P, \mu_P, \mu_2, \mu_1$ = friction coefficient, r_P, r_2, r_1 = mean radius of bearing, subscript N, P, 2, 1 = between piston and cylinder/piston pin bearing/crankpin bearing/crankshaft bearing, $F_S, F_{Pb}, F_{2b}, F_{1b}$ = load shown in Eqs. (12) and (13).

The displacement x_P and the velocity v_P of the piston are given by referring to Fig. 4.

$$\left. \begin{aligned} x_P &\approx r(1 - \cos \theta) + \rho(1 - \cos 2\theta)/4 \\ v_P &\approx r\omega(\sin \theta + \rho \sin 2\theta/2) \\ \rho &= r/L \end{aligned} \right\} \quad (15)$$

The relative angular velocity ω_{PC} of the piston pin to the connecting rod and ω_{2C} of the crankpin to the connecting rod are

$$\left. \begin{aligned} \omega_{PC} &\approx \rho \omega \cos \theta \\ \omega_{2C} &= (\omega + \phi) \approx \omega(1 + \rho \cos \theta) \end{aligned} \right\} \quad (16)$$

From Eqs. (14)~(16), the total friction loss horsepower P_f and torque T_f are

$$\begin{aligned} P_f &= T_f \omega / 75 \\ &= (F_{Pf}/V_P + T_{Pf}/\omega_{PC} + T_{2f}/\omega_{2C} + T_{1f}/\omega) / 75 \quad (17) \end{aligned}$$

Analyses on the static force, the bearing load and the friction loss of the perfectly balanced device are presented in Report 3⁽²⁾.

4. Numerical Computation and Discussion

4.1 Unbalanced Inertia Force and Moment

The specifications of the testing devices are shown in Table 3 for numerical

Fig. 5 Unbalanced inertia force
(Rotational speed of crank 6000 rpm)

1 Perfect balance	2 Static balance
3 Rotation balance	4 Half balance
5 Over balance	

Fig. 6 Unbalanced inertia moment
(Rotational speed of crank 6000 rpm)

1 Perfect balance
2 Static balance
3 Rotation balance
4 Half balance
5 Over balance

computations. Nomenclature in Table 3 are as follows. l = one fourth of piston stroke, r = crank radius, L = Length of connecting rod, J_m, J_A, J_c = moment of inertia shown in Eq.(2). Other nomenclature are the same as those in Fig.3. Figs.5 and 6 show the unbalanced inertia force and moment at rotational speed of crank 6000 rpm respectively, and they are calculated by substituting the values of Table 3 in the equations of Table 2. As shown in Figs.5 and 6, they are zero in the perfectly balanced device. The rotation balance, the half balance and the over balance of the known piston-crank device, only controls the direction in which unbalanced inertia forces occur. And inertia forces can perfectly be eliminated in the static balanced device, then a large unbalanced inertia moment occurs.

4.2 Load on Each Portion

Fig.7 shows the load on each portion which is calculated by substituting a pressure force acting on the piston in Eq.(10). The pressure force is shown in Fig.9. Bearing loads on the crankpin and the crankshaft are little differences in their magnitudes between the perfectly balanced device and the known piston-crank device. It is remarkably different from the case of the known piston-crank device that the side

Table 3 Specification of each device

Type	Weight, Dimension, Moment of g mm kgcms ²
Perfect balance	$W_p=316, W_p+W_2+W_A=1112, l=10$
Static balance	$W_p=150, W_p+W_2+W_A=366$ $R_A=75, J_A=3.25 \times 10^{-3}$
Rotation balance	$W_p=149, W_2=43.5$
Half balance	$W_p=150, W_2=45$
Over balance	$W_p=150, W_2=45.5$
	$L=84$ $r=20$ $J_c=5.6 \times 10^{-5}$ $W_{cp}=37$

Fig.7 Load on each portion due to piston pressure force

- 1 Perfectly balanced device
2 Known piston-crank device

force to the cylinder doesn't occur in the perfectly balanced device. The load on the big end bearing of the connecting rod is the same as the pressure force acting on the piston in the perfectly balanced device.

The load due to inertia at rotational speed of crank 6000 rpm, are shown in Fig.8 and are calculated by substituting the values of Table 3 in Eq.(11). As shown in Fig.8, the bearing load on the crankshaft and the side force to the cylinder are zero in the perfectly balanced device, and so the perfectly balanced device is much superior to the known piston-crank devices in these points. However, bearing loads on the crankpin and the big end of the connecting rod are large in the perfectly balanced device, and are due to inertia of the rotating weight and the reciprocating weight. Consequently, the rotating weight and the reciprocating weight should be designed to reduce their weight by the unification of the piston and the connecting rod, and by the benefitting selection of a form and the material of a connecting rod, making the best use of the merits of the perfectly balanced device that the piston and the connecting rod receives only a pressure force and a tensile force, and reciprocates on the cylinder axis quite together. Utilizing these designs, bearing loads on the crankpin and the big end of the connecting rod can effectively be decreased. Bearing loads in the static balanced device are large.

Fig.8 Load on each portion due to inertia (Rotational speed of crank 6000 rpm)

- 1 Perfect balance 2 Static balance
3 Rotation balance 4 Half balance
5 Over balance

5. Experimental Result and Discussion

Experiments on vibration, crank torque and friction loss are carried out, according to the measuring methods presented in Report 2⁽⁴⁾ and Report 3.⁽²⁾

The qualities of balance on each portion of the perfectly balanced device are 2nd grade of JIS at rotational speed of crank 1085 rpm, and its balancing method and their details of balance are shown in Report 3.⁽²⁾ The known piston-crank devices are balanced in the wellknown method in accordance with the foresaid balanced conditions, and their errors of balance are within ± 5 gmm.

5.1 Pressure Force and Crank Torque

An example of the pressure in the compression chamber is shown in Fig.9 and the relation between the maximum pressure and the rotational speed of crank in Fig.10. The waveform of the pressure and the magnitude of the maximum pressure are almost the same at various rotational speeds of crank. Fig.11 shows the pressure force acting on the piston (hereinafter, often designated as "load"), and the crank torque calculated by substituting its pressure force in Eq.(10). The above results are almost the same as those of the perfectly balanced device in Report 3.⁽²⁾

5.2 Vibration

Figs.12 and 13 show the vibration amplitudes and their waveforms of testing

Fig.9 Piston pressure force in known piston-crank device
(Rotational speed of crank 2500 rpm)

Fig.10 Maximum value of piston pressure force in known piston-crank device

devices at no load respectively. These experimental results correspond to the theoretical results. It is experimentally ascertained that the rotation balance, the half balance and the over balance only control the direction in which unbalanced inertia forces occur in the known piston-crank device, and that the inertia forces and moment are balanced in the perfectly

Fig.11 Piston pressure force and crank torque

Fig.12 Amplitude of vibration (at no load)

● Perfect balance, ▲ Static balance
△ Half balance, □ Rotation balance
○ over balance

Fig.13. Wave form of vibration amplitude
(At no load, rotational speed of crank 1800 rpm)

Fig.14 Amplitude of torsional vibration of crankshaft

- Perfectly balanced device (at load)
- -○- - Perfectly balanced device (at no load)
- -●- - Known piston-crank device (at load)
- -○- - Known piston-crank device (at no load)

balanced device.

The piston severely strikes on the cylinder wall and loud noises occur in the static balanced device. Because its connecting rod whose moment of inertia is large, oscillates with a big angular acceleration. Vibration experiment is carried out within the rotational speed ranging 1000~3000 rpm and other experiments are discontinued in the static balanced device. From the above results, it seems that the static balanced device is not practical. The static balanced device should be improved to decrease the unbalanced inertia moment, even though the unbalanced inertia force remains in some degree.

Fig.14 shows the amplitude of the torsional vibration of a crankshaft. They are almost the same with each other in three kinds of known balanced piston-crank devices. The amplitudes of the perfectly balanced device are smaller than those of known piston-crank devices, but these amplitudes of both devices can not be compared quantitatively with each other. Because the moment of inertia of the former is larger than that of the latter. The amplitude of known piston-crank devices become larger as the rotational speed of crank increases and that of the perfectly balanced device is almost constant at various rotational speeds. It seems that the inertia moment is not balanced in the former and is balanced in the latter, and that the amplitude due to a friction torque, the play between shafts and bearings, and errors of machining and assembling, etc. mainly occur in the latter.

Fig.15 shows the crank torque and the waveform of the torsional vibration of a crankshaft. These magnitudes correspond to the results in Figs.11 and 14. The crank torque and torsional vibration of known

Fig.15 Wave form of crank torque and torsional vibration (Rotational speed of crank 1800 rpm)

piston-crank devices at no load have second harmonics and these results correspond to the theoretical results of Table 2. The crank torque and torsional vibration at no load occur little in the perfectly balanced device. Consequently, it is ascertained that the inertia moment is balanced in the perfectly balanced device.

5.4 Friction Loss

Fig.16 shows the experimental values of a friction loss torque and their theoretical values calculated on the assumption of $\mu_H=0.05$, $\mu_P=\mu_2=\mu_1=0.02$, $r_P=7$ mm, $r_2=17$ mm, $r_1=10.5$ mm. The coefficients of friction are assumed to be constant in theoretical calculation. Friction losses in three kinds of known balanced devices are almost the same with each other.

As shown in Fig.16, the experimental values agree qualitatively with the theoretical values, and it results from the assumption on the analysis and numerical computation that the former is larger than the latter. The friction loss of the perfectly balanced device is equal to about three over two or two times that in the known piston-crank device. These values show mainly friction losses in bearings. When these devices are used in a reciprocating engine or a compressor, a large friction loss between a cylinder and a piston occurs on account of a big side force to a cylinder due to a big exploding force or a pressure force in the known piston-crank device. However, no side forces occur in the perfectly balanced device. So, it seems that there are little differences between friction losses of both devices in practical. It is ascertained from the relation between an input and an output of the vibrationless reciprocating engines utilizing the perfectly balanced device in some degree, and is discussed still more.

6. Conclusion

The static balanced theory of the piston-crank device are established, and then it is clear that the static balanced theory can be applied to not only the perfectly balanced device but also the known balanced piston-crank devices. The compari-

Fig.16 Friction loss torque

- Experimental value(at load)
- Experimental value(at no load)
- Theoretical value(at load)
- Theoretical value(at no load)

sons among the static balanced device, the rotation balanced device, the half balanced device, the over balanced device and the perfectly balanced device are presented.

This report can be summarized as follows:

- (1) In the perfectly balanced device, the inertia force and moment are perfectly balanced and there are little vibrations.
- (2) The rotation balance, the half balance and the over balance only control the direction in which the unbalanced inertia force occurs in the known piston-crank device.
- (3) The static balanced device is not practical, because the piston severely strikes on the cylinder wall on account of the oscillation of the connecting rod whose moment of inertia is large.
- (4) As the vibration of the perfectly balanced device is mainly caused by the

moment due to the piston pressure, its vibration isolation is easier than that of the known balanced piston-crank device whose vibration occurs in various directions.

(5) A side force from a piston to a cylinder wall, due to inertia and a piston pressure occur in the known piston-crank device, but doesn't occur in the perfectly balanced device theoretically.

(6) The friction loss of the perfectly balanced device is larger than that of the known piston-crank device. A side force to a cylinder wall, due to a big exploding force (or a big pressure force) occurs in the latter, and yet doesn't occur in the former. It seems that there are little differences between friction losses of both devices, when they are used in reciprocating engines and reciprocating compressors.

(7) There are little differences between load on each portion, the crank torque due to the piston pressure in the perfectly balanced device and those in the known piston-crank device. Load on each portion, the crank torque due to inertia in the perfectly balanced device are different from those in the known piston-crank devices.

From these results, it is considered that the perfectly balanced device is effective for reciprocating engines, reciprocating compressors, etc.

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