On the Lubrication of Crosshead-Pin Bearing with Eccentric Journal*

(Analysis for the Case with both Eccentric Journal and Bearing)

By Yutaro WAKURI**, Shinsuke ONO*** and Mitsuhiro SOEJIMA****

This paper refers to a theoretical study on the lubrication of a crosshead-pin bearing specially equipped with an eccentric journal in a 2-stroke cycle diesel engine. From the analyses of the journal center locus and the oil-film pressure distribution it is clarified that the main and the eccentric bearing surfaces bear the load alternately and facilitate the oil-feed from the circumferential oil-groove to each surface in unloaded period, and that the load carrying capacity is greatly improved. The following optimums for a design become evident.

(i) The offset ratio is preferably large without exceeding the limit determined by the load carrying capacity.

(ii) The ratio of the radial clearance to the radius is as small as possible.

(iii) The width of the main segment is preferably large in the range 65~75% of the whole bearing.

(iv) The effective circumferential angle and the ratio of the width to the diameter are as large as possible.

Key Words : Lubrication, Crosshead-pin Bearing, Eccentric Journal, Eccentric Bearing, Journal Center Locus, Numerical Analysis

1. Introduction

To the crosshead-pin bearing in a large-sized 2-stroke cycle diesel engine, hereinafter termed a cross bearing, the improvement of the load carrying capacity has been an outstanding problem because the bearing metal seizure or fatigue occurs quite often owing to an inevitable difficulty of the lubrication under the peculiar structure and operation condition. Many researches were carried out in the past concerning the lubrication characteristics, (1)-(4) the properties of the bearing alloy, (5)-(7) the mechanism and preventive for the failure (8)-(11) and so on. In the case of the conventional cross bearing with several axial oil-grooves which are necessary for the oil-film exchange on the bearing surface under the conditions of both the unidirectional load and the oscillatory motion, it is quite difficult to improve its load carrying capacity remarkably. In such a situation, a new idea of a model change to an offset type cross bearing shown in Fig.1 was suggested recently to improve the load carrying capacity considerably. (12)-(14) The authors analyzed previously the lubrication characteristics of the cross bearing equipped with an offset type journal. (15) (16) The results of the analysis show

that the load carrying capacity can be improved remarkably by abolition of axial oil-grooves which are indispensable to the conventional bearings, because the bearing surfaces split into main and eccentric segments by a circumferential oil-groove alternately carry the load with the change of sliding direction and the resultant bouncing motion of the journal facilitates the oil-film exchange on the bearing surfaces. Such an offset type bearing structure, however, has the fault that the offset is restricted within the bearing clearance and the structure of fitted bearing advantageous for load carrying capacity can not be realized.

In this paper, the analyses are carried out for the case that both the journal and the bearing are shaped into an offset structure with large offset values. The lubrication characteristics and the influ-

Fig.1 Offset type crosshead-pin bearing

* Received 5th October, 1983.

** Professor, Faculty of Engineering, Kyushu University, Hakozaki, Fukuoka.

*** Professor, Faculty of Engineering, Kyushu University.

**** Assistant, Faculty of Engineering, Kyushu University.

ences of the offset, the radial clearance and others on the oil-film thickness and the pressure are investigated, and the design criteria of them are discussed.

Symbols

- c : radial clearance
 d : bearing alloy thickness
 D : journal diameter
 e : eccentricity
 e_0 : offset of journal
 e'_0 : offset of bearing
 e_1, e_2 : eccentricities of main and eccentric journals
 E : Young's modulus of bearing alloy
 h : oil-film thickness
 H : non-dimensional oil-film thickness, h/c
 l_1, l_2 : axial widths of main and eccentric segments
 L : whole axial width, $l_1 + l_2$
 p : oil-film pressure
 p_1, p_2 : oil-film pressures in main and eccentric segments
 p_{1max}, p_{2max} : maximum values of p_1 and p_2
 p_f : oil-feed pressure
 p_w : mean load pressure
 p_{wmax} : maximum value of p_w
 P : bearing load
 q : reduced pressure
 r : journal radius
 S : representative load carrying capacity, $(r/c)^2 \eta_0 \omega_0 / p_{wmax}$
 t : time
 x, y, z : coordinates
 Z : non-dimensional coordinate, z/L
 α : pressure viscosity coefficient
 β : elastic deflection coefficient
 δ : radial deflection ratio, $\beta p/c$
 ϵ : eccentricity ratio, e/c
 ϵ_0 : offset ratio of journal, e_0/c
 ϵ'_0 : offset ratio of bearing, e'_0/c
 ϵ_1, ϵ_2 : eccentricity ratios of main and eccentric journals
 ϵ_{max} : maximum value of ϵ
 η : viscosity
 η_0 : viscosity at normal pressure
 θ : angular coordinate from maximum gap
 $\bar{\theta}$: boundary in diverging flow region
 θ_c : crank angle
 θ : circumferential angle from bearing top to the leading edge of effective bearing surface
 $\theta_2 - \theta_1$: effective circumferential angle of the bearing
 λ : ratio of connecting rod length to crank radius
 σ : Poisson's ratio
 ϕ : attitude angle from vertical axis
 ϕ_0 : inclination angle of journal center line $\bar{O}_1\bar{O}_2$ from horizontal axis
 ϕ_1, ϕ_2 : attitude angles of main and eccentric journals
 φ : oscillation angle from vertical axis, angle between vertical axis and load line
 Ψ : non-dimensional reduced pressure, $(c/r)^2 q / \eta_0 \omega_0$
 ω : angular velocity of bearing
 ω_0 : angular velocity of crankshaft

Fig.2 Bearing geometry and load direction

2. Fundamental Equations

From the Reynolds equation governing the incompressible fluid lubrication of dynamically loaded finite journal bearings, the following equation relating to the oil-film thickness h and viscosity η is derived by using the oscillating angular velocity of the bearing ω (to which the angular velocity of load line coincides as shown in Fig.2), the eccentricity e and the attitude angle ϕ .

$$\frac{1}{r^2} \frac{\partial}{\partial \theta} \left(\frac{h^3}{\eta} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\eta} \frac{\partial p}{\partial z} \right) = 6\omega \frac{\partial h}{\partial \theta} + 12 \left(\frac{de}{dt} \cos \theta + e \frac{d\phi}{dt} \sin \theta \right) \quad (1)$$

Since the mean load pressure of the cross bearing in a large-sized slow-speed engine usually amounts to 10~20 MPa at maximum and the oil-film pressure becomes considerably high, the change in oil-film viscosity with pressure is taken into account in this analysis using the following relation.

$$\eta = \eta_0 \exp(\alpha p) \quad (2)$$

Further, since the bearing material used in this kind of bearing is usually lined with a soft alloy like a white metal whose Young's modulus E is relatively small, the elastic deflection of the alloy layer under the high pressure seems to have a considerable effect on the bearing characteristics. By assuming that the back metal is rigid the approximate elastic deflection coefficient β is obtained by the following formula. (17)

$$\beta = (d/E) \{ 1 - 2\sigma^2 / (1 - \sigma) \} \quad (3)$$

Accordingly the oil-film thickness h is expressed by a formula including the deflection increment.

$$h = c + e \cos \theta + \beta p \quad (4)$$

3. Solution

Equation (1) can be arranged by the

relation with non-dimensional variables, that is, $\Psi = (c/r)^2 q / \eta_0 \omega_0$ for the reduced pressure $q = \{1 - \exp(-\alpha p)\} / \alpha$ determined by Eq.(2), $H = h/c$ for the oil-film thickness and $Z = z/L$ for the coordinate in the axial direction.

$$\begin{aligned} & \frac{\partial}{\partial \theta} \left(H^3 \frac{\partial \Psi}{\partial \theta} \right) + \left(\frac{r}{L} \right)^2 \frac{\partial}{\partial Z} \left(H^3 \frac{\partial \Psi}{\partial Z} \right) \\ &= 6 \frac{\omega}{\omega_0} \frac{\partial H}{\partial \theta} + 12 \left(\frac{1}{\omega_0} \frac{d\epsilon}{dt} \cos \theta \right. \\ & \quad \left. + \frac{\epsilon}{\omega_0} \frac{d\phi}{dt} \sin \theta \right) \end{aligned} \quad (5)$$

where

$$\begin{aligned} H &= 1 + \epsilon \cos \theta + \delta \\ \delta &= \beta p / c \end{aligned} \quad (6)$$

The pressure boundary conditions around the sliding surface are set as follows.

$$\begin{aligned} \theta &= \theta_1 + \varphi - \phi \text{ (inlet angle)} : \Psi = \Psi_f(Z) \\ \theta &= \theta_2 + \varphi - \phi \text{ (outlet angle)} : \Psi = \Psi_f(Z) \\ Z &= 0 \text{ (circumferential oil-groove)} \\ & : \Psi = \Psi_{f0} \\ Z &= l/L \text{ (environment)} : \Psi = 0 \\ \theta &= \bar{\theta} \text{ (diverging flow region)} \\ & : \Psi = \partial \Psi / \partial \theta = 0 \end{aligned} \quad (7)$$

where $\Psi_f(Z)$ is changed from an oil-film pressure to an atmospheric pressure linearly.

In the bearing structure shown in Fig.1, the main journal segment and the eccentric one are so arranged as to carry the load mainly during the explosion-expansion period ($\theta_c = 0^\circ \sim 180^\circ$) where the mean load pressure is relatively high, and during the suction-compression one ($\theta_c = -180^\circ \sim 0^\circ$) in which the load is low, respectively. The respective symbols of the main journal and the eccentric one are denoted by the subscripts 1 and 2. Supposing, as shown in Fig.3, that the respective offsets of the journal and the bearing are e_0 and e'_0 , and that the journal center line O_1O_2 is inclined with an angle ϕ_0 to the horizontal axis, the following relations hold among the eccentricity ratios ϵ_1, ϵ_2 and the attitude angles ϕ_1, ϕ_2 .

$$\begin{aligned} \epsilon_2 &= [\epsilon_1^2 + \epsilon_0^2 + \epsilon_0'^2 + 2(\epsilon_1 \epsilon_0' \sin(\phi_1 \\ & \quad - \varphi) - \epsilon_0' \epsilon_0 \cos(\varphi - \phi_0) - \epsilon_1 \epsilon_0 \end{aligned}$$

Fig.3 Relative position of bearing centers O, O' and journal centers O_1, O_2

$$\cdot \sin(\phi_1 - \phi_0)]^{1/2} \quad (8)$$

$$\begin{aligned} \phi_2 &= \cos^{-1} \{ (\epsilon_1 \cos \phi_1 + \epsilon_0 \sin \phi_0 \\ & \quad - \epsilon_0' \sin \varphi) / \epsilon_2 \} \end{aligned} \quad (9)$$

where ϵ_0 and ϵ'_0 are the respective ratios of e_0 and e'_0 to the radial clearance c , and φ is the oscillating angle of the bearing.

The resultant force composed of the oil-film pressures p_1 and p_2 on both bearing surfaces has to equilibrate with the mean load pressure p_w under the following relations

$$\begin{aligned} & \int_0^{l/L} dZ \int_{\theta_1 + \varphi - \phi_1}^{\theta_2 + \varphi - \phi_1} p_1 \cos \theta d\theta \\ & + \int_0^{l/L} dZ \int_{\theta_1 + \varphi - \phi_2}^{\theta_2 + \varphi - \phi_2} p_2 \cos(\theta - \phi_1 + \phi_2) d\theta \\ & + 2p_w \cos(\phi_1 - \varphi) = 0 \end{aligned} \quad (10)$$

$$\begin{aligned} & \int_0^{l/L} dZ \int_{\theta_1 + \varphi - \phi_1}^{\theta_2 + \varphi - \phi_1} p_1 \sin \theta d\theta \\ & + \int_0^{l/L} dZ \int_{\theta_1 + \varphi - \phi_2}^{\theta_2 + \varphi - \phi_2} p_2 \sin(\theta - \phi_1 + \phi_2) d\theta \\ & - 2p_w \sin(\phi_1 - \varphi) = 0 \end{aligned} \quad (11)$$

The cyclic behavior of the journal centers is obtained by the numerical calculation shown as follows.

(1) At an arbitrary crank angle, all initial values of the polar coordinates (ϵ, ϕ) and the derivatives ($d\epsilon/dt, d\phi/dt$) are given, and in accordance with them the elastic deflection ratio δ is previously assumed, and then the calculation is started.

(2) As the movement of the eccentric journal is subject to that of the main journal by the relation of Eqs(8),(9), the derivatives of either have only to be gradually modified by the Newton-Raphson method so as to satisfy Eqs (10),(11) with the pressure calculated from Eq.(5) by the finite difference method.

(3) The value of δ is corrected according to the pressure distribution change by Eq.(6) and the boundary $\bar{\theta}$ is reset on the grid where the boundary condition ($\Psi = \partial \Psi / \partial \theta = 0$) is satisfied best.

(4) The foregoing stages (2),(3) are repeated till both δ and $\bar{\theta}$ converge, and after the convergence new polar coordinates of the journal center for the next crank angle are calculated by using the modified derivatives. The computation is continued till the journal center locus is cyclically stabilized.

4. Result and Consideration

For the present analysis the lubrication condition of the cross bearing is set referring to that of a large-sized marine diesel engine as follows.

The specific load pressure $p_w/p_{w\max}$ and the specific angular velocity ω/ω_0 change cyclically with the crank angle θ_c as shown in Fig.4, where $p_{w\max}$ means a maximum value of p_w during the cycle, the value in the vicinity of $\theta_c = 15^\circ$, and ω_0

Fig.4 Changes of mean load pressure and angular velocity with crank angle

means an angular velocity of the crankshaft, $\omega/\omega_0 = \cos\theta_c / \sqrt{\lambda^2 - \sin^2\theta_c}$. Here, the ratio of the connecting rod length to the crank radius λ is 3.7, and the ratio of the oil-feed pressure p_f to p_{wmax} is 0.03. Referring to an engine oil and a whitemetal alloy, the pressure viscosity coefficient of the lubricant α and Young's modulus of the bearing alloy E are assumed to be around $2 \times 10^{-8} \text{ Pa}^{-1}$ and 50 GPa respectively.

4.1 Characteristics of oil-film behavior

As to this bearing composed of an offset journal and an offset bearing, there are so many design parameters affecting the journal center locus or the oil-film pressure distribution that it is difficult to determine all of the optimums regarding the dimension and shape. Therefore, in this paragraph the lubrication characteristics common to all are discussed and later the effects of some parameters on them are examined.

Figure 5 shows the journal center loci in the main and eccentric bearing segments which are calculated under the following conditions. The representative load carrying capacity S is 1.0 and the ratio of width to diameter L/D is 0.75, the width ratio of the main segment to the whole bearing l_1/L is 0.6, and both the offset ratio of journal ϵ_0 and that of bearing ϵ'_0 are 2.0. The effective circumferential angle of bearing $\theta_2 - \theta_1$ is 180° and the inclination angle of journal center line ϕ_0 is 0° . Additionally, the representative radial deflection ratio of alloy layer $\beta p_{wmax}/c$ is 0.01. Figure 6 shows the calculated results of the changes in pressure distribution with crank angle.

From these results the following features of the oil-film behavior are revealed.

Fig.5 Journal center loci of main journal on the left and eccentric journal on the right (Number corresponding to dark point on locus indicates crank angle)

Fig.6 Pressure distribution on bearing surface at each crank angle (Lines $Z = 0$ are circumferential groove boundaries, and $Z(1)$ is main segment side and $Z(2)$ is eccentric segment side.)

(1) During the period ($\theta_c = -60^\circ \sim 60^\circ$) in which the eccentricity of the main journal increases and that of the eccentric journal decreases, a positive pressure distribution (loaded condition) is built up on the main segment surface and the eccentric segment is released from the load.

(2) During the period ($\theta_c = 180^\circ \sim -120^\circ$) the eccentric segment carries the load and the main one is released.

(3) Owing to these behaviors the oil-film on each bearing surface can be exchanged by feeding from the circumferential oil groove.

(4) The respective oil-film pressures built up on the main and eccentric segment surfaces amount to maximum values at $\theta_c = 15^\circ$ and at around $\theta_c = -120^\circ$.

Judging from the fluctuation of the journal center and the situation of the pressure distribution, it is recognized that the pressure generation in the loaded oil-film is largely due to the squeeze action.

Accordingly, with regard to the offset ratios ϵ_0, ϵ'_0 dominating the fluctuation amplitude of the journal center and to the representative load carrying capacity S defined with the bearing clearance and other dimensions which are closely related to the squeeze film action, their influences on the load carrying capacity are investigated and the proper limits as the design criteria are determined.

4.2 Offset ratio and representative load carrying capacity

According to the offset ratios of the journal and the bearing, ϵ_0 and ϵ'_0 , the

journal center loci change as shown in figures (a) and (b) in Fig. 7. From the comparison between these two, it is recognized that the fluctuation amplitude of journal center increases proportionally to the increase in offset ratio, and that the maximum value of the eccentricity ratio during the cycle which corresponds to the minimum oil-film thickness is somewhat reduced.

Figure 8 shows the influence of representative load carrying capacity S on the cyclic movement of journal center loci. From this, it is noteworthy that the maximum value of the eccentricity ratio becomes considerably small as the S value increases, although the fluctuation amplitude of journal is almost unchanged.

Figure 9 shows the change of maximum eccentricity ratio during the cycle ϵ_{max} with both the changes of offset ratio ϵ'_0 (equal to ϵ_0) ranging from 1 to 100 and the S value from 0.25 to 4.0. In this figure the solid line and the broken one indicate the value of the main journal ϵ_{1max} and that of the eccentric one ϵ_{2max} respectively. As the offset ratio ϵ'_0 becomes large, ϵ_{1max} monotonously decreases for all the S values, but the decreasing rate is not always constant. ϵ_{2max} similarly decreases till ϵ'_0 comes to 10~30 but it increases in a region of $\epsilon'_0 > 30$. This tendency of ϵ_{2max} is caused by the following reason. As the offset ratio ϵ'_0 takes a large value, the time in a crank angle at which ϵ_2 amounts to maximum shifts toward a later crank angle near $\theta_c = 0^\circ$, though it is toward an earlier one near $\theta_c = -90^\circ$ in the case with small ϵ'_0 value. Namely, as the time shifts toward $\theta_c = 0^\circ$, the pe-

Fig. 7 Effect of offset ratio in bearing and journal

Fig.8 Effect of representative load carrying capacity

riod in which the eccentric segment carries the load by itself becomes long and the loading intensity to it becomes high. In the bearing design, special attention should be paid not only to the thickness but also to the pressure intensity of oil-film, and it is necessary to find out how the design parameters affect the pressure, especially the maximum one during the cycle. Figure 10 shows the changes of the respective maximum pressures of the main and eccentric segments, p_{1max} and p_{2max} , with the abscissa ϵ'_0 and the parameter S as well. From these results, it is found that the effect of ϵ'_0 or S on p_{max} well corresponds to the one on ϵ_{max} . According as ϵ'_0 becomes large, p_{1max} monotonously diminishes over all the values of ϵ'_0 , and p_{2max} similarly changes till ϵ'_0 reaches a specific value (10 ~ 50), but p_{2max} rapidly rises just after ϵ'_0 exceeds it. This change is caused by the same reason as in the case of ϵ_{2max} . As S becomes large, p_{1max} and p_{2max} both become small. As regards to these pressure changes, it is noteworthy that p_{2max} exceeds p_{1max} in the region of the large value of ϵ'_0 .

Generally, the oil-film pressure in the bearing should be restricted below the plastic flow pressure limit of the bearing alloy. Therefore, it is desirable to design the bearing to lower the maximum pressure. In this case it seems that the upper limit of ϵ'_0 has to be set within the region where p_{2max} is lower than p_{1max} ; around 50 ($S = 0.25$), 100 ($S = 1.0$) or 200 ($S = 4.0$). However, the lower limit of ϵ'_0 remains to be investigated in the future because it cannot be determined only from this analysis. On the other hand, since the oil-film thickness becomes large and

Fig.9 Changes of ϵ_{max} with ϵ'_0 and S

Fig.10 Changes of maximum pressure with ϵ'_0 and S

the maximum pressure becomes low as the representative load carrying capacity $(r/c)^2 \eta_0 \omega_0 / p_{wmax}$ increases, it is desirable to select the ratio of radial clearance to radius c/r as small as possible.

4.3 Width ratio of main segment to whole bearing

To the offset type bearing it is important just as in the case of the above optimums of the offset ratio and the load carrying capacity to find out the proper ratio of the main segment width l_1 to the whole bearing width L for the lubrication of the surfaces split into the main and eccentric segments. Therefore, the changes of ϵ_{max} and p_{max} with the width ratio l_1/L are examined in the same way as in the foregoing paragraph, and shown in Fig.11 and Fig.12 respectively. The curve of ϵ_{2max} in Fig.11 is not smooth in the vicinity of $l_1/L = 0.75$, because respective maximum values are used as ϵ_{2max} for the reason that ϵ_2 reaches maximum at θ_c just before 0° in the case of $l_1/L < 0.75$ or just after -90° in the case of $l_1/L > 0.75$.

From these figures, about the effects of l_1/L , it is recognized that according as l_1/L becomes large, ϵ_{1max} decreases

Fig.11 Changes of ϵ_{max} with l_1/L

Fig.12 Changes of maximum pressure with l_1/L

but ϵ_{2max} increases, and p_{1max} lowers but p_{2max} rises. In considering the optimum of l_1/L , the value around 0.75 is recommendable as the upper limit, because, within this region, p_{2max} does not exceed p_{1max} and ϵ_{2max} does not become extremely high. However, the lower limit is indistinct in these figures. It would be appropriate to make the ratio as large as possible within the range of 0.65 ~ 0.75 if ϵ_{2max} is allowable for the lubrication.

4.4 Effective circumferential angle

Since the dimensions of cross bearing are usually restricted by engine, it is useful in the bearing design to know the influence of effective circumferential angle $\theta_2 - \theta_1$ on the bearing characteristics. Figure 13 shows the influence on the maximum pressures of the main and eccentric segments. As shown in this figure, though the value of $\theta_2 - \theta_1$ is varied only in the range, $90^\circ \sim 180^\circ$, the change in p_{max} is remarkable. Accordingly, it is desirable to design the bearing such as to make the effective circumferential angle as large as possible.

4.5 Ratio of width to diameter

In this kind of engine bearings, the ratio of the width to the diameter L/D often becomes a problem in design. Especially in recent years, in order to make the structure of large-sized engines compact the bearings should be shortened. Therefore the effect of L/D on the bearing characteristics is investigated and the optimum is discussed as well. Figure 14 shows the changes of the maximum pressures with L/D . p_{2max} in this figure corresponds to the pressure at a crank angle $\theta_c \approx 0^\circ$ when L/D is smaller than a value at the fold point of the broken line or the pressure in the vicinity of crank angle $\theta_c = -120^\circ$ when L/D is larger than that. The influence of L/D value on the maximum oil-

Fig.13 Changes of maximum pressure with $\theta_2 - \theta_1$

Fig.14 Changes of maximum pressure with L/D

film pressure is conspicuous in this type of bearing, because the width of this bearing is split into the main and eccentric segments. Accordingly, just like the effective circumferential angle, the ratio has so great an influence on the load carrying capacity that it should not be made extremely small.

5. Conclusion

Concerning a crosshead-pin bearing with an eccentric journal, especially one with the structure having large offsets in journal and bearing, the journal center locus and the oil-film pressure distribution have been analyzed, and the optimums for bearing dimensions have been found with reference to the maximum values of the eccentricity ratio during the cycle and those of the oil-film pressure. The principal conclusions can be summarized as follows.

(1) According to the change of the oscillating direction the main and eccentric segments of the bearing alternately carry the load while approaching their journal surfaces and exchange the oil-film while separating from them.

(2) With regard to the offset, the radial clearance, the axial width and the effective circumferential angle which are principal dimensions of the bearing the following design standards can be recommended.

(i) The ratio of the offset to the ra-

dial clearance is preferably large without exceeding the upper limit determined by the load carrying capacity.

(ii) The ratio of the radial clearance to the radius is as small as possible, that is, the fitted bearing has the best structure.

(iii) The width of the main segment is preferably large in the range of 65~75% of the whole width.

(iv) The effective circumferential angle and the ratio of the width to the diameter are as large as possible.

It should be noted that a part of this research was supported by a scientific research fund appropriated from the Ministry of Education, Japan.

References

- (1) Wakuri, Y., et al., Trans. JSME (in Japanese), Vol.27, No.178 (1961-6), 910.
- (2) Komichi, H. and Asanabe, S., J.JSLE (in Japanese), Vol.13, No.4 (1968-4), 155.
- (3) Wakuri, Y., et al., Prep. JSME (in Japanese), No.700-1 (1970-4), 141.
- (4) Wakuri, Y. et al., Bull.JSME, Vol.25, No.207. C (1982-9), 1457.
- (5) Wakuri, Y. and Akaboshi, M., J.JSLE (in Japanese), Vol.6, No.5 (1961-5), 324.
- (6) Yamada, K., et al., 10th CIMAC Paper, No.30 (1973).
- (7) Asanabe, S. and Taniguchi, S., Sci. Machine (in Japanese), Vol.29, No.5 (1977-5), 603.
- (8) Soda, N. and Miyahara, Y., J.JSME (in Japanese), Vol.62, No.482 (1959-3), 398.
- (9) Wakuri, Y., et al., J.JSLE (in Japanese), Vol.10, No.2 (1965-10), 109.
- (10) Hirano, F. and Goto, S., Proc. Inst. Mech. Eng., Vol.181, Pt.30 (1966-67), 31.
- (11) Edited by JSME, Report of crosshead Bearing Research Subcommittee (in Japanese), (1966-8), 1, JSME.
- (12) Maciotta, R., MTZ, Vol.38, No.2(1977-2), 61.
- (13) Ciliberto, G. and Mariani, L., 12th CIMAC Paper, No.A21 (1977).
- (14) Booker, J. F., et al., Trans. ASME, Ser.F, Vol.104 (1982), 478.
- (15) Wakuri, Y., et al., Bull. JSME, Vol.25, No.206, C (1982-8), 1312.
- (16) Wakuri, Y., et al., Bull. MESJ, Vol.11, No.1 (1983-3), 1.
- (17) Higginson, G.R., Proc. Inst. Mech. Eng., Vol.180, Pt.3B (1965), 31.