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Static Characteristics of Floating Head Mechanism*

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High capacity magnetic storage drums are obtained by making small clearance between head cores and drum surface. We are investigating a head supporting method named an air floating head mechanism, which is an application of self-acting gas bearings. In this report we described static characteristics of the mechanism, namely, influence of clearance ratio, slender ratio of the slider and difference of surface curvatures on pressure distribution or floating force of air film. These are calculated by an electronic digital computer and compared with experimental data.

1. Introduction

In order to make high density and large capacity magnetic storage drums which are used as elements of electronic digital computers and data processing devices, it is necessary to provide small clearances between magnetic head cores and magnetic film on drum surface. When magnetic head cores are fixed, it is difficult to get small clearances because of vibration, expansion owing to revolution and thermal expansion of drums. Therefore, by means of a floating head mechanism, we are going to obtain a small constant clearance without contact with the drum. Floating head mechanisms are classified into a self-acting type and an externally pressurized type. On the former, a slider floats by means of viscous flow induced by revolution of the drum, and on the latter, a slider floats by pushing compressed gas into the clearance. They are respectively applications of self-acting and externally pressurized gas bearings.

A floating head mechanism we are investigating is of an air self-acting type for magnetic storage drums. Air wedge film is made by revolution of the drum between the slider and the drum surface whose radii are nearly equal, as shown in Fig. 1. Meanwhile, the slider can move around only an axle. By balancing self-weight or load-spring against floating force on the slider, we obtain small constant clearance between the drum surface and the core fixed on the slider. Our mechanism is a variation of a partial bearing.

In this report, static characteristics of the

mechanism are described. Namely, effects of clearance ratio, slender ratio of the slider, the relative length of slider to drum and difference between their curvature on floating characteristics are investigated. We compared calculated values with experimental data.

Nomenclature

- b = width of a slider
- h_0 = equivalent height of a slider axle
- h_1 = clearance at leading edge
- h_2 = clearance at trailing edge
- l = arc length of a slider
- l_1 = chord length of a slider
- p = pressure in air film
- p_a = pressure of atmosphere

Fig. 1 Floating head mechanism

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r =radius of a drum

$r+\Delta r$ =radius of curvature of a slider

x, y =coordinates

U =circumferential velocity of a drum

W =static floating force

μ =coefficient of viscosity of air

θ_p =center of pressure

$B=b/l, B_1=b/l_1, H_1=h_1/h_2, H_{20}=h_2/h_0, P=p/p_a,$

$\Delta R=\Delta r/h_2, W'=W/p_a b l, X=x/l, Y=y/l, \theta_0=l/r,$

$\Delta=6\mu U l/p_a h_2^2, \Delta_0=6\mu U l/p_a h_0^2$

2. Pressure distribution of air film

Radius of curvature of the slider surface is nearly equal to radius of the drum, and the slider surface is surrounded by two arcs and two generating lines. Clearance is small enough in comparison to the radii, and constant in direction of generating lines. We put an origin at a corner of the leading edge of the slider and set x - and y -axis respectively in circumferential and generating direction.

The relation between a position in the air film and dimension of its clearance is expressed by

$$H=E\cos(\theta_0 X+\theta_1)+\Delta R \quad \dots\dots\dots(1)$$

where $X=x/l, H=h/h_2, E=e/h_2, \Delta R=\Delta r/h_2$. When $X=0$ and $X=1$, we have $H=H_1=h_1/h_2$ and $H=1$ respectively. Therefore we have

$$\left. \begin{aligned} \theta_1 &= \tan^{-1} \alpha, \quad E=(H_1-\Delta R)\sqrt{1+\alpha^2} \\ \alpha &= \frac{1}{\sin \theta_0} \left(\cos \theta_0 - \frac{1-\Delta R}{H_1-\Delta R} \right) \end{aligned} \right\} \quad \dots\dots\dots(2)$$

We neglect compressibility of air, because the circumferential velocity of the drum is small enough in comparison to sound velocity. And we neglect inertia of air, for instance, centrifugal force owing to curvature of air film. We also neglect air flow in thickness direction and pressure gradient, because the thickness of air film is small enough. Then we have the following Reynolds Equation regarding pressure in air film⁽¹⁾.

$$\frac{\partial}{\partial X} \left(H^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Y} \left(H^3 \frac{\partial P}{\partial Y} \right) = \Delta \frac{\partial H}{\partial X} \quad \dots\dots\dots(3)$$

where

$$P = \frac{p}{p_a}, \quad \Delta = \frac{6\mu U l}{p_a h_2^2} \quad \dots\dots\dots(4)$$

and $Y=y/l, p_a$ =atmospheric pressure, p =pressure at $(X, Y), \mu$ =coefficient of viscosity of air, U =circumferential velocity of the drum. Because Δ is not related with X and Y , we can have

$$P=1+\Delta Q(X, Y) \quad \dots\dots\dots(5)$$

where Q is in proportion to gauge pressure. Then Eq. (3) becomes

$$\frac{\partial}{\partial X} \left(H^3 \frac{\partial Q}{\partial X} \right) + \frac{\partial}{\partial Y} \left(H^3 \frac{\partial Q}{\partial Y} \right) = \frac{\partial H}{\partial X} \quad \dots\dots\dots(6)$$

The boundary conditions are

$$Q(X, Y)=Q(1, Y)=Q(X, 0)=Q(X, B)=0 \quad \dots\dots\dots(7)$$

where $B=b/l$. Clearances in generating direction are constant, therefore $\partial H/\partial Y=0$ and accordingly Eq. (6) is converted to

$$\frac{\partial^2 Q}{\partial X^2} + \frac{\partial^2 Q}{\partial Y^2} = -\frac{3}{H} \frac{\partial H}{\partial X} \frac{\partial Q}{\partial X} + \frac{1}{H^3} \frac{\partial H}{\partial X} \quad \dots\dots\dots(8)$$

We are going to solve this equation by numerical method. First we divide the slider surface into m equal parts by lines parallel to Y -axis and into n equal parts by straight arcs parallel to X -axis, and make the mesh with interval $X=1/m$ and $Y=B/n$. We define mesh points Q_{ij} as follows:

$$Q_{ij}=Q(i\Delta X, j\Delta Y) \quad (i=0, 1, \dots, m, j=0, 1, \dots, n) \quad \dots\dots\dots(9)$$

Then, we convert Eq. (8) to the next difference equations

$$Q_{ij}=C_{0i}+C_{1i}Q_{i+1,j}+C_{2i}Q_{i-1,j}+C_{3i}Q_{i,j+1}+C_{3i}Q_{i,j-1} \quad \dots\dots\dots(10)$$

$$(i=1, 2, \dots, m-1, j=1, 2, \dots, n-1)$$

where

$$\left. \begin{aligned} C_{0i} &= -\frac{1}{\Gamma H^3} \frac{\partial H}{\partial X} \\ C_{1i} &= \frac{1}{\Gamma(\Delta X)^2} + \frac{3}{2\Gamma(\Delta X)H} \frac{\partial H}{\partial X} \\ C_{2i} &= \frac{1}{\Gamma(\Delta X)^2} - \frac{3}{2\Gamma(\Delta X)H} \frac{\partial H}{\partial X} \\ C_{3i} &= \frac{1}{\Gamma(\Delta Y)^2}, \quad \Gamma = \frac{2}{(\Delta X)^2} + \frac{2}{(\Delta Y)^2} \end{aligned} \right\} \quad \dots\dots\dots(11)$$

we solve Eq. (10) numerically by means of Extrapolated Liebmann Method⁽¹⁾⁽²⁾.

Relations between pressure distribution at symmetrical planes of the slider and ratio of clearances H_1 are shown in Fig. 2. Abscissa and ordinate are distance X from the leading edge along circumference of the drum and gauge pressure $Q=(P-1)/\Delta$ respectively. At the boundary

Fig. 2 Effect of clearance ratio on pressure distribution

of the slider, pressure is atmospheric, namely $Q = 0$. $\Delta R = 0$ is the case where radius of the slider and that of the drum are equal. When $H_1 = 1$, pressure in the first and the second half parts are negative and positive respectively. It is considered to be due to the following reason. When curvature of the slider and that of the drum are equal, and clearance in both ends also are equal, clearances in the first half are divergent. Clearance ratio H_1 corresponding to the largest maximum value in Fig. 2 is rather larger than in an infinite width plane slider. Relations between pressure distributions at the symmetrical planes and slender ratio B are shown in Fig. 3. The larger B becomes the larger pressure we obtain. When $B > 4$, pressure distribution is almost invariable. These show the same tendency as plane sliders⁽³⁾. Relations between pressure distributions and radii of curvature of sliders are shown in Fig. 4. When $\Delta R > 0$, radius of the slider is larger than the drum and the slider corresponds to convex plane one. When $\Delta R < 0$, radius of the slider is smaller and the slider corresponds to concave plane one. When $\Delta R > 0$, the pressure in the first half part is larger than when $\Delta R = 0$. When $\Delta R < 0$, these relations are

Fig. 3 Effect of slender ratio on pressure distribution

Fig. 4 Effect of radius of curvature on pressure distribution

reversed. Relations between pressure distributions and relative length of sliders to radius of drums are shown in Fig. 5. When θ_0 is near zero, the characteristics are similar to plane sliders. When θ_0 becomes larger, the difference against plane slider becomes larger.

3. Floating force on slider

We are going to calculate the floating force on sliders and its center of pressure. As shown in Fig. 1, we put an origin at the center of air film and set η - and ξ -axis in tangential and radius direction respectively. We set components of floating force as shown in the figure, and define the chord length of a slider as l_1 .

In order to compare these sliders with plane one, we define non-dimensional floating forces as

$$W'_\xi = \frac{W_\xi}{p_a b l_1} \quad \text{and} \quad W'_\eta = \frac{W_\eta}{p_a b l_1} \quad \dots\dots\dots (12)$$

Considering the condition under which pressure distribution is symmetrical regarding the center line, we have the following floating forces:

$$\left. \begin{aligned} W'_\xi &= \frac{2\Delta\varphi}{B} \int_0^1 \int_0^{B/2} Q \sin\left\{\theta_0\left(X - \frac{1}{2}\right)\right\} dXdY \\ W'_\eta &= \frac{2\Delta\varphi}{B} \int_0^1 \int_0^{B/2} Q \cos\left\{\theta_0\left(X - \frac{1}{2}\right)\right\} dXdY \end{aligned} \right\} \dots\dots\dots (13)$$

where

$$\varphi = \frac{l}{l_1} = \frac{\theta_0}{2 \sin(\theta_0/2)} \quad \dots\dots\dots (14)$$

And $B = b/l$, $B = B_1/\varphi$. Integrations of Eq. (13) are solved by means of Simpson's Method⁽⁴⁾. Accordingly we set number of interval m and $n/2$ even on the calculation of pressure distributions. Then the resultant force is given by

$$W' = \frac{W}{p_a b l_1} = \sqrt{W'^2_\xi + W'^2_\eta} \quad \dots\dots\dots (15)$$

Under the assumption where shearing forces owing

Fig. 5 Effect of slider length of pressure distribution

to fluid are very small, θ_p is defined as angle between acting line of the resultant force of pressure and η -axis. Then we have

$$\frac{\theta_p}{\theta_0} = \frac{1}{\theta_0} \tan^{-1} \frac{W'_x}{W'_y} \dots\dots\dots (16)$$

When there is a divergent part within $0 < X < 1$, negative pressure occurs. When H_1 decreases continuously from large value, we have first $\partial H / \partial X = 0$ and then $\partial H / \partial X > 0$ at the leading or trailing edge. Accordingly, negative pressure occurs first where $\theta_1 = 0$ or $\theta_1 + \theta_0 = 0$. Namely from Eq. (2), boundary lines of occurrence of negative pressure are expressed by equations

$$\left. \begin{aligned} H_1 &= \Delta R \left(1 - \frac{1}{\cos \theta_0} \right) + \frac{1}{\cos \theta_0} \\ H_1 &= \Delta R (1 - \cos \theta_0) + \cos \theta_0 \end{aligned} \right\} \dots\dots\dots (17)$$

But according to calculations, negative resultant floating force occurs where $H_1 \neq 1$ or $H_1 < 1$. Therefore negative floating forces are omitted on the illustrations of the characteristics.

Next, we shall present examples obtained by numerical methods. Effects of slender ratio on floating force are shown in Fig. 6. When we design practical sliders, it is convenient to use chord length. Therefore, B_1 is used as slender ratio. When B_1 increases, floating force per unit area increases. W'/Δ is maximum near $H_1 = 2$ in any case. In practice sliders with several kinds of width can be used. For instance, if only one head core is fixed on a slider, a small width one is used. Meanwhile, if several head cores, they are fixed on a large width slider. Effects of slender ratio on center of pressure are shown in Fig. 7. Equation $\theta_p/\theta_0 = 0$ shows that center of pressure is at

Fig. 6 Effect of slender ratio on floating force

the center of a slider. With θ_p/θ_0 increasing in positive direction, center of pressure approaches the trailing edge. As proved by the figure, increase of B_1 causes decrease of θ_p/θ_0 . The effect is small on small H_1 and large on large H_1 . The minimum value occurs near $H_1 = 1$, but such event doesn't occur in plane sliders.

Effects of radius of curvatures on floating force are shown in Fig. 8. Effect of ΔR is large near $H_1 = 2$ and is small at large H_1 . The larger ΔR becomes, the larger the floating force we obtain. It is inverse on negative ΔR . H_1 corresponding to maximum floating force decreases against increasing ΔR . Abramovitz calculated regarding infinite width convex plane sliders⁽⁵⁾. There Δ s are made regarding clearance on the trailing edge and minimum clearance. When we choose the former, the tendency is similar to our

Fig. 7 Effect of slender ratio on center of pressure

Fig. 8 Effect of radius of curvature on floating force

slider. We can guess that a concave plane slider will have inverse tendency to a convex one. Broken lines shown in the figures present boundary lines of occurrence of negative pressure, which is obtained by Eq. (17). When ΔR is large, it is possible that negative pressure occurs on large H_1 . Effects of radius of curvature of slider on center of pressure are shown in Fig. 9. Effect of ΔR is large near $H_1=1$ and decreases with increasing H_1 . When $\Delta R>0$, center of pressure is near the trailing edge as H_1 is large, and approaches the center of the slider and moreover the leading edge as H_1 decreases. These are the same tendency as infinite width convex plane sliders⁽⁵⁾. When $\Delta R\leq 0$, the center of pressure is near the trailing edge as H_1 is large, and approaches the center of the slider and again the trailing edge as H_1 decreases. This characteristics is considered to be due to the following reason. When $\Delta R<0$, only positive pressure occurs at large H_1 and negative pressure starts to occur at the leading edge as H_1 decreases. Next, we consider resultant forces owing to only positive and only negative pressure at the center of the drum respectively. The direction of resultant force owing to positive pressure approaches positive direction of η -axis from the trailing edge as H_1 decreases. Meanwhile, the direction of resultant force owing to negative pressure approaches negative direction of η -axis from the inverse direction against the leading edge and its amplitude increases as H_1 decreases. Accordingly, characteristics of the center of pressure are considered to be shown as the figure. When $\Delta R>0$, negative pressure starts to occur at the trailing edge and the resultant force due to negative pressure approaches negative direction of η -axis from the inverse direction against the trailing

ing edge, and the resultant force due to positive pressure also approaches the direction of the leading edge from the trailing. Therefore, behaviour of the center of pressure is monotonous. As proved by Fig. 9, as $\Delta R>0$, we have H_1 giving $\theta_p=0$ except for $H_1=1$. Then positive pressure occurs as proved by Fig. 8. On the contrary even if the slider is supported at the center of the slider, we are able to get the floating force by using adequate radius of curvature of the slider. Then the slider can be floated even if the direction of the drum revolution is reversed. Such phenomena also suggested on a convex plane slider⁽⁶⁾.

Effects of relative length of sliders to the drum on floating force are shown in Fig. 10. When $\theta_0=0$, radii of the slider and the drum are infinite and they correspond to plane bearings. Equations we previously deduced are available as $\theta_0\neq 0$,

Fig. 9 Effect of radius of curvature on center of pressure

Fig. 10 Effect of slider length on floating force

Fig. 11 Effect of slider length on center of pressure

therefore we calculated as $\theta_0=0.001$ instead of plane sliders. These calculated values are consistent with calculated values about plane sliders obtained by D.F. Hays⁽³⁾. Floating force per unit area decreases against increasing θ_0 , and the effect is large at large H_1 . Difference of floating force between $\theta_0=0.001$ and $\theta_0=0.1602$ is very little. Effects of relative length of the slider to the drum on the center of pressure are shown in Fig. 11. The center of pressure approaches the trailing edge with increasing θ_0 . Difference owing to θ_0 is small on large H_1 , but is very large near $H_1=1$. Although θ_p/θ_0 approaches zero with H_1 approaching 1 in case of plane sliders, θ_p/θ_0 becomes larger again after the minimum values in case of our sliders. As proved by Fig. 10 and Fig. 11, characteristics of the sliders with small θ_0 are approximately the same as those of the plane sliders.

4. Experimental results

We use the slider as shown in Fig. 12 and investigate the relation between floating force and clearances. When loading weight and floating force act on the slider as shown in the figure, we have the following equation by considering the torque about the slider axle. Moreover, in practice the slider rotates about A_0 , but the variation of the angle is so small that we can have an imaginary axle A_0' and express axle height as h_0 . Then we have

$$\frac{W_g'}{A_0} = \frac{W_g}{A_0} \left(\cos \theta_p + \frac{r_c}{l_c} \sin \theta_p \right) \quad \text{.....(18)}$$

where

$$A_0 = \frac{6\mu U l}{p_a h_0^2} \quad \text{and} \quad W_g' = \frac{W_g}{p_a b l_1} \quad \text{.....(19)}$$

and $\lambda = l_0/l_1$ (l_0 is a distance from A_0' to the trailing edge). Moreover, we have

$$H_{20} = \frac{1}{1 + \lambda(H_1 - 1)} \quad \text{and} \quad A_0 = A H_{20}^2 \quad \text{.....(20)}$$

Fig. 12 Load and floating force in equilibrium.

where $H_{20} = h_2/h_0$. We get W'/A and θ_p/θ_0 against H_1 by using Fig. 6 and Fig. 7, so we can get theoretically the relation between W_g'/A_0 and H_{20} by using Eqs. (18) and (20). In the experiment, we use sliders whose dimensions are similar to practical ones. Accordingly, we shall prove the theoretical results and at the same time investigate the possibility of using the sliders practically. Because both the drum and the slider vibrate, measurement of clearances is done by the following method.

As shown in Fig. 13, we set the axle of the drum horizontal. With diameter $D=250$ mm the drum can be rotated at an arbitrary speed between 1 000 and 9 000 rpm by means of a motor with a continuous speed converter. The contour of the slider is shown in Fig. 14. The slider has an axle supported by pivot bearings on a precisely adjustable base, so that the slider can be given an arbitrary position and inclination to the drum surface. We set a pole at the center of the upper side of the slider surface and vary the load W_g by varying the weight. But torque due to only the slider and the arm is obtained by measuring the load in a horizontal situation. On the contrary, we define these measured results as W_g .

Fig. 13 Experimental device

Fig. 14 Slider for experiment

without loads. In order to measure clearances h_1 and h_2 we fix isolated electrodes on both ends of the slider as shown in the figure and detect the variation of capacitance between the electrodes and the drum surface by means of a capacitance micrometer. Calibration between clearances and the output voltages is done by means of a microscope at rest.

Relations between loads W_g and clearances h_1

Table 1 Experimental results

No.	W_g g	1 000 rpm		3 000 rpm		5 000 rpm	
		h_1 μ	h_2 μ	h_1 μ	h_2 μ	h_1 μ	h_2 μ
1	30.8	71.2	24.7	90.0	45.5	95.5	58.5
2	33.8	70.1	23.0	87.0	44.0	93.0	55.5
3	36.8	69.1	22.5	85.5	42.0	89.0	54.0
4	41.8	67.6	19.5	82.5	39.5	88.5	51.0
5	46.8	66.5	17.7	80.5	37.0	85.5	48.5
6	56.4	64.7	15.5	76.0	33.0	81.5	44.3
7	66.4	63.0	13.5	74.0	30.0	78.5	41.0
8	76.4	62.0	12.2	71.8	27.5	76.0	38.0
9	96.4	60.0	10.2	69.0	23.8	72.5	33.5
10	116.4	59.3	8.8	66.8	21.0	70.0	29.8
11	136.4	58.0	8.0	65.5	18.5	68.5	27.3
12	186.4	—	—	63.0	15.3	65.5	22.0
13	236.4	—	—	61.8	12.2	63.0	18.5
14	286.4	—	—	59.3	10.5	61.5	16.0
15	336.4	—	—	58.3	9.3	60.3	14.2
16	386.4	—	—	57.5	8.5	59.3	12.7
17	451.4	—	—	—	—	58.0	11.2
18	501.4	—	—	—	—	57.5	10.3
19	551.4	—	—	—	—	57.0	9.6
20	22.2	—	—	95.0	57.0	—	—
21	22.2	—	—	75.0	46.5	—	—
22	22.2	—	—	40.8	30.5	—	—

Fig. 15 Floating characteristics

or h_2 at 1 000, 3 000 and 5 000 rpm are obtained as shown in Table 1. If the relative axle height to the drum surface is constant, H_{20} can be obtained by measuring only h_2 after h_0 is once obtained. But h_0 is not constant in practice, because the axle height decreases because of expansion of the drum and deflection of the supporting base by loads. Then we obtain h_0 through calculation from experimental data h_1 and h_2 . Relation between W_g'/A_0 and H_{20} is shown in Fig. 15, where W_g' is obtained by using W_g in Table 1, A_0 by using h_0 and H_{20} by using h_2 and h_0 . Solid line in the figure shows theoretical values calculated by using dimensions of the slider used in our experiment and $\mu = 1.82 \times 10^{-6}$ kg sec/m². The figure shows that if the axle height is constant, clearance at the trailing edge decreases monotonously as the load increases. Because the theoretical values almost agree with the experimental ones, the assumptions for introducing the theory are considered to be adequate and the calculated results can be used in the design of the floating head mechanism.

5. Conclusion

From the numerical analysis and the experiment on static characteristics of the floating head mechanism for magnetic storage drums, we have the following results:

(1) When the slider length is small enough in comparison to the drum radius and their curvatures almost equal each other, the floating characteristics can be approximated by plane sliders with length which equals the chord length of the slider.

(2) The more larger the radius of curvature of the slider is than that of the drum, the larger the floating force is and the closer the center of pressure approaches the leading edge. When radius of curvature of the slider is smaller than that of the drum the floating force decreases and the center of pressure becomes as shown in Fig. 9.

(3) When the slider length is not small enough in comparison to the drum, the floating force is smaller and the center of pressure approaches the trailing edge in comparison to plane bearings.

Moreover, numerical calculations were done by using an digital electronic computer.

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