The Development of the Floating Head for the Magnetic Drum*

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To develop a floating head mechanism for use with a magnetic drum, investigation of the plane slider and the multi-leaf parallel-spring has been carried out.

A new nondimensionalization has been used in the Reynolds equation to obtain floating characteristics of convergent-divergent film shape plane sliders. Design charts have been made from the numerical solutions.

A multi-leaf parallel-spring has been developed to suspend the slider. This spring applies the load and is simultaneously used as lead wires to the magnetic heads.

The floating head which has been developed has a 2 micron floating space and a 56 bits/mm bit density applied to the MDA-03 magnetic drum (250 mm dia, 3,000 rpm).

The floating head is composed of the multi-leaf parallel-spring, which has small torsional rigidity, and the multi-pad plane slider which has a large torsional moment around the slider length axis. Therefore, no adjustment to the width direction angle is needed to set the floating head.

1. Introduction

When using a magnetic drum memory, magnetic heads must be set close to the drum surface to obtain high bit density. The floating head was contrived to maintain a clearance of a few microns between the magnetic heads and the drum surface by means of the hydrodynamic force of air.

The surface of a floating head slider can be designed in various shapes, but the one applied to a drum is usually cylindrical or plane shaped. The floating head is one of the applications of hydrodynamic gas bearings. However, the difference from ordinary gas bearings involves maintaining a constant clearance instead of sustaining a load. The plane slider has a smaller load capacity than the cylindrical slider, but the plane slider can be manufactured more easily and more accurately.

This paper reports on investigations worked out to develop a floating head for use with a magnetic drum. Floating head mechanisms involving the floating characteristics of the plane slider and a method of sustaining the slider are discussed.

2. Floating characteristics of a plane slider

2.1 Basic characteristics

A plane slider set on the drum as shown in Fig. 1 is considered. In practice, it is advantageous for clearances at the fluid inlet and outlet regions to be sufficiently larger than minimum clearance \( h_0 \), because a discrepancy in the setting point or any little error of flatness at the slider ends scarcely affects the floating characteristics. A gas (usually air) is used for lubricant, therefore floating forces due to compressibility are generated about such a convergent-divergent clearance shape. In this case, the slider length scarcely affects the floating characteristics. Therefore, in discussing the floating characteristics, it is disadvantageous to use a parameter including

![Fig. 1 Plane slider set on the drum](image-url)
the slider length $l$ such as Bearing Number ($\Lambda = 6\mu U l / \rho h_0$). New nondenimensionalizations are used in this paper:

$$\begin{align*}
P &= p / p_a, \quad H = h / h_0 = 1 + X, \quad X = x / R \\
Z &= z / R, \quad L = l / l_0, \quad C = c / l_0, \quad B = b / l \\
G_s &= \sqrt{2} \mu U l^{1/2} / \rho h_0^{3/2} \\
W &= w / (2\mu U l / h_0) \\
\bar{X} &= \bar{x} / R
\end{align*}$$

...(1)

where $p$ is the local pressure, $p_a$ is the ambient pressure, $l$ is the slider length ($l_1 - l_2$), $w$ is the floating force, $\mu$ is the viscosity, $U$ is the velocity of the drum surface, $r$ is the radius of the drum, $\bar{x}$ is the center of pressure measured from the outlet to the inlet direction along the $x$ axis, $b$ is the slider width, $z$ is the coordinates along the width and $R$ is the shaft radius. The isothermal Reynolds equation using Eqs. (1) is

$$\frac{\partial}{\partial X} \left( H^2 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Z} \left( H^2 \mu \frac{\partial P}{\partial Z} \right) = G_s \frac{\partial (H P)}{\partial X}$$

...(2)

$G_s$ in the right hand member is the nondimensional index including no slider length $l$ terms, which corresponds to the Bearing Number and indicates the degree of compressibility effect.

Several methods to solve Eq. (2) may be considered. In this paper, the relaxation method is used. The reasons for using this method are that (i) the convergent solutions can be computed because $G_s$ is not so large in the domain to be solved, (ii) solutions can be computed even in the case of a clearance varying in the width direction and (iii) the necessary memory capacity of a computer is not so large.

When $P$ is computed, $W$ and $\bar{X}$ are integrated according to the following equations:

$$W = \frac{6}{G_s L B} \int \int (P - 1) dX dZ$$

$$\bar{X} = C L \int \int (P - 1) X dZ dX$$

...(3)

Relations between $C$ and $W$ are shown in Fig. 2. At $C = 0$, $W$ decreases as $G_s$ increases, but, at a large value of $C$, $W$ increases because the region of negative pressure decreases as $G_s$ increases, as shown in Fig. 3. It is presumed that the same tendency in the case of $C = 0$ appears when $G_s$ reaches a much larger value. However, solutions cannot be computed in the case of large values of $G_s$. Figure 4 shows

![Fig. 2 Relations between $C$ and $W$](image)

![Fig. 3 Influence of $G_s$ on pressure distribution](image)

![Fig. 4 Relations between $C$ and $G_s$](image)

![Fig. 5 Relations between $B$ and $W$](image)
the relations between C and \( \bar{X} \). As C increases, \( \bar{X} \) varies greatly because the region of negative pressure varies. Especially at \( C > 0.5 \), \( \bar{X} \) changes from \(-\infty\) to \(+\infty\) when \( W \) changes from a negative to a positive value. The relations between B and W are shown in Fig. 5. This indicates that, at a large value of C, it is difficult to presume the floating characteristics by multiplying the solutions of an infinitely wide slider by a coefficient of sideflow leakage. The influences of L upon W are shown in Fig. 6. The reason why W has the largest value at \( C = 0.25 \) is that the positive pressure is best maintained under this condition. The increase of W with the increase of L is marked at \( C = 0.75 \) because the region of the negative pressure decreases with the increase of L and the pressure distribution becomes similar to that at the value of \( C = 0.5 \) as shown in Fig. 7. The change of the slider pressure distribution is shown according to the width of B in Fig. 8. As B becomes small, the maximum value of positive pressure decreases. However, the absolute value of minimum negative pressure does not decrease so much. This indicates that the compressibility effect is not marked at a small value of B.

It can be understood from Fig. 2 and Fig. 6 that the influence of C upon W becomes small as \( G_s \) and L increase. This makes easy the method of setting the floating heads on the drum.

Fig. 6 Relations between floating force and slider length

Fig. 7 Relations between L and the range of negative pressure region

Fig. 8 Relations between B and pressure distribution

Fig. 9 Experimental results for plane slider floating characteristics
Design charts such as Fig. 5 arranged from numerical solutions are used. Figure 9 shows the comparison between the theoretical characteristics evaluated from solutions and the experimental results. They are in good accordance.

2-2 Floating head slider design problems

1) Setting error

Setting errors are (i) discrepancy and inclination in the direction of the slider length, (ii) inclination of the center axis of a slider to the direction of movement of a drum and (iii) inclination against the drum surface in the width direction. There are no serious difficulties in the problem represented by (i) and (ii) in the case of a plane slider from the analysis in the previous section. This is because the change of clearance between the slider and the drum surface due to setting error is smaller for a plane slider than for a cylindrical slider.

The problem represented by (iii) is serious for any slider shape. Figure 10 shows the influence of inclination evaluated by the method mentioned in the previous section. An inclination of $10^{-2}$ radians as a setting error is unavoidable by fixing mechanically and $10^{-4}$ radians by means of adjusting. Therefore, it is very difficult to suppress fluctuation of the floating force when sliders with the inclinations shown in Fig. 10 are used. For the practical floating heads, the inclination self-adjusting function mentioned in the following section becomes necessary.

2) Surface roughness

Several papers report on the influence of the surface roughness of a bearing lubricated by fluid upon the loading characteristics. However, it is fairly difficult to presume it numerically from theory alone. Therefore, a rough estimation was made by the theory of averaged thickness of the lubricant film.

Assuming that the surface of the slider has rugosities which consist of a series of small parallel corrugations as shown in Fig. 11, the local film thickness $h$ is

$$h = f(x) + (1 + \cos \theta)(\delta/2)$$

Then the Reynolds equation becomes

$$\frac{d}{dx} \left( \rho \frac{d\hat{p}}{dx} \right) = 6\mu \frac{d\hat{h}}{dx}$$

where

$$\hat{h} = \frac{1}{2\pi} \int_{0}^{2\pi} \rho d\theta, \quad \hat{h}^{3} = \frac{1}{2\pi} \int_{0}^{2\pi} \rho^{3} d\theta$$

This equation can be easily solved numerically. One of the solutions is shown in Fig. 12. It can be understood that the roughness scarcely affects the floating force if it is smaller than 1/3 or 1/5 of minimum clearance $h_{o}$. It is considered that the theory of averaged thickness of lubricant film can be applied to ordinary surface roughness to a certain extent because the theory deals with the thickness of lubricant film which is averaged. In the case of $\delta << f(x)$, it is supposed that the lubricant scarcely flows in the width direction. Therefore, it is useful to average the lubricant film. However, it is presumed that considerable lubricant flows from the convex region to the concave region of the slider when $\delta$ is a large value. Clarifying the limit of application of averaging the thickness of the lubricant film is a problem. For this problem, the Reynolds equation was solved considering the lubricant flow in the width direction and solutions were compared. Consequently, it

Fig. 11 Surface roughness model

Fig. 12 Roughness effect on floating force
became apparent that the theory of averaged thickness of the lubricant film gave reasonable approximate values when \( \zeta/\lambda < 10^{-1} \) and \( \delta/h_0 < 3 \).

Concerning a floating head, the general condition is that \( \zeta/\lambda < 10^{-3} \), \( \delta/h_0 < 1 \), therefore, using the theory of averaged thickness of lubricant film is tenable.

3) Lubricant inertia force

For the plane slider discussed in this paper, the local thickness of the lubricant film varies from two or three microns to several hundred microns and the velocity of the drum surface reaches several tens of m/sec. Therefore, the influence of the inertia force of the lubricant must be considered. It is usually difficult to solve the Navier-Stokes equation including the inertia force. In this paper, the influence upon the floating force was examined by the averaged inertia method. Consequently, in the range of \( h_0 < 5 \mu \), \( l < 10 \) mm, \( L < 10 \) and \( U < 40 \) m/sec, the inertia force was estimated at under 10% of floating force and it is unnecessary to consider inertia force under an ordinary design condition.

3. Floating head slider suspension

3-1 Slider sustaining method

When the slider is sustained with a pivot and the point to be sustained is suitably settled upon, the slider can be set without adjustment. But such a mechanism is complicated and the mechanism's life and the reliability of parts such as pivots are doubtful. The simplest higher reliability suspension involves use of a spring. Especially, in the case of a plane slider, the discrepancy and inclination in the length direction scarcely affect the floating characteristics. Therefore, it is practical to use a parallel spring. However it is impossible to mechanically correct the setting error of inclination in the width direction as mentioned previously, therefore self-adjusting performance by means of flexibility is necessary.

The single pad slider possesses this self-adjusting function due to the moment which is caused by the transition of the center of pressure. But a spring of fairly low stiffness is required for this purpose.

On the other hand, the suspension spring must sustain the load which balances the floating force. It is necessary for the contrary function that the spring has a proper stiffness and, moreover, small torsional rigidity around the axis of the length because the floating force of a slider which has a large adjusting moment is large. It is difficult to give this function to the suspension spring of the single-pad slider. To eliminate this defect, it is effective to use a multi-pad slider which has two slider surfaces in the direction of the width. In the case of a multi-pad slider, the moment \( M \) is given from the difference of floating force of two surfaces and the distance between them, as shown in Fig. 13. Therefore, the moment of multi-pad slider is larger than that of single slider and can be selected at any value in a certain range. On the other hand, the suspension spring of the slider may have so small a torsional rigidity that the inclination in the direction of the width should be modified to a value which can be practically disregarded by that moment.

A method of decreasing the torsional rigidity of the parallel spring is the use of an opposing-center parallel spring whose leaves oppose each other as shown in Fig. 14 (b) or the multi-leaf parallel spring whose leaves are split, as shown in Fig. 14 (c). The stiffness \( S_f \) of the parallel spring of length \( l_u \), both ends of which are fixed, is

\[
S_f = \frac{12}{l_u^3} \sum \left( EI \right)_i
\]

where \( E \) is Young's modulus. The second moment of area \( I \) is

\[
I = \frac{1}{12} at^3 \left( \frac{a^2}{t^2} \sin^2 \alpha + \cos^2 \alpha \right)
\]

where \( a \) is the width, \( t \) is the thickness and \( \alpha \) is the angle between the width direction and bending. Torsional rigidity \( \phi \) is

\[
\phi = \frac{1}{l_u^3} \sum (12EIa^2 + l_u^3 k_at^4 G)
\]

Fig. 13 Torsional moment of multi-pad slider

Fig. 14 Method to decrease torsional rigidity
where \( I \) is given by Eq. (7), in which \( \alpha \) is the angle shown in Fig. 14 (c), \( d \) is the length of the line combining the center of the section of a leaf and the center of torsion, \( k_t \) is the coefficient concerning the torsion of rectangular section and \( G \) is the shearing modulus. The torsional rigidity of the opposing center parallel spring is smaller, but, the allowable deflection is smaller because of being limited by the maximum stress at the surface of the leaf as shown in Fig. 15. Therefore, the range in which leaves can be used as springs is narrow. Moreover, productivity is not good because the leaves are tilted. For the suspension mechanism of the slider, the torsional rigidity of the multi-leaf parallel spring is so small that the inclination of the slider against the drum surface is sufficiently corrected. Figure 16 shows the experimental results where the difference of the clearance is less than 1 \( \mu \)m, even if the setting error \( \Delta h_s \) is 50 \( \mu \)m.

3.2 Design example

The magnetic head cores should be set at the point of the minimum clearance. On the other hand, the natural frequency should be as high as possible so that the fluctuation of the clearance due to the run-out of the drum surface may be small. The natural frequency depends upon the differential coefficient of floating force \( w \) to minimum clearance \( h_0 \). Figure 17 shows an example of computational results.

| Table 1 Dimension and characteristics of the type 203 Floating Head |
|------------------------|------------------|
| 1. Drum with which the Floating Head is used | 250 mm dia, 3000 rpm |
| 2. Form | Multi-slider sustained by twelve leaf parallel spring (also used as lead wires) with four magnetic cores |
| 3. Slider | Ceramic |
| Material | 5±0.1 mm length |
| Dimension | 1.6±0.05 mm width (two surfaces) |
| 4. Surface roughness | 9 mm total width |
| Material | Less than 0.5 \( \mu \)m |
| Track width | Ferrite |
| Gap position | 0.05±0.05 mm |
| Center of slider | Stainless steel |
| 5. Suspension spring | 0.3 mm |
| Material | 8 mm |
| Thickness | 2 mm |
| Width | 2 mm |
| Effective length | 7.5 mm |
| Spacing between upper and lower leaves | 0.8 mm |
| Total width of suspension spring | 18.75 g/mm |
| Deflection during operation | Less than 120 gcm/\( \mu \) |
| Bending stiffness | Less than 1.3 g |
| Torsional rigidity | 1.8±0.5 \( \mu \) |
| 6. Weight (the floating part) | Higher than 1 KH \( \mu \) |
| 7. Performance | Floating space |
| Natural frequency | 203 Floating Head |

![Fig. 15 Relations between allowable deflection \( d_{\text{max}} \) and angle \( \alpha \)](image)

![Fig. 16 Experimental result of self-adjusting in the width direction](image)

![Fig. 17 Floating force \( w \) and bearing stiffness \( \frac{\partial w}{\partial h_s} \)](image)
of the relation between \( w \) or \( \partial w/\partial h_0 \) and the position of the minimum clearance. When \( C \) increases, \( W \) decreases. On the other hand, \( \partial w/\partial h_0 \) does not vary so much. When \( W \) is large, the risk of failure due to an unexpected touch of the slider to the drum surface becomes large. Therefore, it is advantageous that the minimum clearance be set near the inlet of the lubricant and that floating force be small. However, such unsteadiness occurs as the negative pressure due to the variation of clearance and a sudden change of the center of pressure in the case of \( C > 0.5 \). Therefore, it is practical to adjust the slider so that the minimum clearance may be at the center (\( C = 0.5 \)) also because the effect of flatness error at inlet-outlet region becomes small.

Table 1 shows the dimensions and characteristics of the type 203 Floating Head with which the MDA-03 magnetic drum is equipped. The construction is shown in Fig. 18. The slider is made of ceramic for anti-abrasion and to avoid heat distortion. Four magnetic head cores are built into each floating head between the two slider surfaces. The flat portion of the apex of the head cores is made so small that floating force may not increase. There are 12 suspension spring leaves. They are also used as magnetic head wires. The slider is designed so that the center of the pressure may be between the inlet and the fixed point of the spring. The compression stress, which causes the buckling of spring, may be small. The spring fixed parts are cemented in a sandwich structure of ceramic and aging distortion is negligible. Figure 19 shows the presumption of floating characteristics, taking the manufacturing tolerance into consideration. This floating head, with which MDA-03 magnetic drum is equipped, obtains a bit density of 56 bits/mm.

4. Conclusions

As a result of the research mentioned above, a high performance floating head has been developed. The results of this research are as follows:

i) For a plane slider, the variation of floating force due to the setting error in the length direction is suppressed by applying a long length slider. This fact makes setting easy.

ii) The plane slider which has a marked convergent-divergent lubricant film shape has good floating characteristics and is practically useful.

iii) The surface roughness scarcely affects the floating force if it is smaller than one third or one fifth of the clearance.

iv) For the multi-pad slider, the suspension mechanism is designed more easily than for the single slider, because floating force is small and self-adjusting moment to correct inclination in the direction of width is large.

v) The multi-leaf parallel spring has an excellent function in suspension of a floating head because it has small torsional rigidity and is also used as lead wires.

Moreover, to obtain higher recording density, a smaller clearance is necessary and, to shorten the access time, a higher revolution velocity is necessary. Under these severe conditions, the index \( G_r \) attains to a large value and numerical solutions cannot be computed by the relaxation method. A new numerical method is being investigated to obtain design charts. Problems in manufacture are also being examined.

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References