On the Characteristics of Air-Lubricated Bearing (Succeeding Report)

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1. Introduction

It is clarified in the preceeding report, " On the Characteristics of Air-Bearing ",[1] that, by supplying air at the most suitable pressure and distributing its pressure in the bearing clearance, the bearing can be sustained in a state of non-eccentric rotation and of slight friction. Though the actual value of coeffecient of friction of the air-lubricated bearing is very small, it has considerably larger value than the value which is calculated by the assumption that the rotating shaft floats perfectly in the film of air with its frictional resistance depending only on the viscous resistance which is due to shearing of the air film. This fact shows that the air film will be broken intermittently by waviness and roughness of surfaces of the both shaft and bushing, and that coefficient of friction includes the mean summation of these slight metallic contacts which occur when the air film breaks. Therefore, it can be said that the friction of the air-lubricated bearing has a dynamic mechanism, hence, the fundamental problem of the air-lubricated bearing is to develop dynamic stability. Of course, to improve the conditions of the bearing surface and to make the bearing dimension more accurate are important counter-measures, but it is also desirable for the air-lubricated bearing to possess an automatic control mechanism in itself that will, in case the shaft becomes eccentric by some reason, reinstate the former non-eccentric rotation by itself.

In the preceeding report it was also made clear that the air-flow between the air feeder hole and the bearing clearance is equivalent to the air-flow through a nozzle. Then the automatic control operation is obtained by setting a restrictor before the air feeder hole, because the intermediate pressure between this restrictor and the air feeder hole will increase or decrease rapidly with the decrease or increase of the bearing clearance like the gaging pressure in a high pressure type pneumatic gage, and this intermediate pressure will affect the pressure distribution in the bearing clearance and react as a strong restoring force against eccentric rotation. In this report, this operation is investigated by measuring coefficient of friction.

2. Theoretical Consideration

The air supplying arrangement, in case a restrictor is attached in front of the air feeder hole in an air-lubricated bearing, is as shown in Fig. 1, and is quite similar to that of a high pressure type air gage in which air flows through two orifices in series. That is, air is supplied from the air compressor C, shown in the diagram, into the bearing B through the air-reservoir T. The primary restrictor N_1, either an orifice or a nozzle, is located between the reservoir T and the bearing B, and at the same time the air feeder hole N_2 acts as the secondary restrictor.

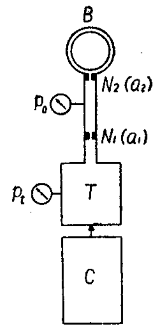

Fig. 1

By passing air through the restrictor N_1, p_t, which is the constant supplying pressure in the reservoir, descends to the intermediate pressure p_0, which is the pressure at the air feeder hole and can be held at any value between the two extremes of p_t and atmospheric p_a. The quantity of flow G through both of the restrictors can be determined at each section by assuming that air flows as an adiabatic flow as follows:

$$G = a_1\left[2g\frac{k}{k-1}\frac{p_t}{v_t}\left\{\left(\frac{p_0}{p_t}\right)^{2/k} - \left(\frac{p_0}{p_t}\right)^{k+1/k}\right\}\right]^{\frac{1}{2}}$$

$$= a_2\left[2g\frac{k}{k-1}\frac{p_0}{v_0}\left\{\left(\frac{p_a}{p_0}\right)^{2/k} - \left(\frac{p_a}{p_0}\right)^{k+1/k}\right\}\right]^{\frac{1}{2}}, \qquad (1)$$

where, a_1, a_2=the section areas of N_1 and N_2, k=the ratio of specific heat of air=1.4, v_t, v_0=the specific volume corresponding with p_t and p_0, p_a=the atomospheric pressure, where p_t, p_0 and p_1 are taken as the absolute pressure.

Substituting:

$$\phi_1 = \left[2g\frac{k}{k-1}\left\{\left(\frac{p_0}{p_t}\right)^{2/k} - \left(\frac{p_0}{p_t}\right)^{k+1/k}\right\}\right]^{\frac{1}{2}}$$

$$\phi_2 = \left[2g\frac{k}{k-1}\left\{\left(\frac{p_a}{p_0}\right)^{2/k} - \left(\frac{p_a}{p_0}\right)^{k+1/k}\right\}\right]^{\frac{1}{2}}, \qquad (2)$$

the formula (1) is reduced to

$$\frac{a_1}{a_2} = \frac{\phi_1}{\phi_2}\left(\frac{p_0}{p_t}\right)^{k+1/2k} \qquad (3)$$

for the relation of adiabatic change is given by

$$p_t v_t^k = p_0 v_0^k.$$

We shall, first of all, consider the air-flow through a nozzle. The quantity of flow G increases rapidly at first in correspondence with the formula (1) as the pressure ratio p_0/p_t or p_a/p_0 is reduced and thereafter gradually levels off to a constant flow, the transition being at the pressure ratio of 0.527 which is critical for air. Then the value of ϕ_1 or ϕ_2 takes the maximum value, ϕ_{max}, of 2.14 accordingly with the critical value of the pressure ratio. Therefore four cases can be considered in relation to the combination of the values of ϕ_1, ϕ_2 and ϕ_{max}. (I) is the case in which air flows non-critically throngh N_1 with $\phi_1<2.14$ and thereafter flows critically through N_2 with $\phi_2=\phi_{max}=2.14$. (II) is the case in which air flows critically through both N_1 and N_2, when $\phi_1=\phi_2=\phi_{max}$. (III) is the case in which air flows critically through N_1 and non-critically through N_2, when $\phi_1=\phi_{max}=2.14$ and $\phi_2<2.14$. (IV) is the case in which air flows non-critically through both N_1 and N_2, when $\phi_1<2.14$ and $\phi_2<2.14$.

By making the pressure p_t constant, the characteristic curves of p_0 are given by the equation (3) as shown by the broken lines in Fig. 2 in relation to a_2/a_1 for the four cases mentioned above. In Fig. 2, the two chain lines of $p_0 \fallingdotseq 2\,\mathrm{kg/cm^2}$ and $a_2/a_1=1.73$ divide the diagram into corresponding sections of the four cases. The real line shows the inflection point of the $p_0 \sim a_2/a_1$ curve. The inflection point always exists in the section (I) or (IV). In the section (I) it takes the constant values of $a_2/a_1=1.56$ and $p_0/p_t=0.63$ and, as it transfers to the section (IV), it gradually attains smaller values of a_2/a_1. It is shown obviously in Fig. 2 that the pressure p_0 increases rapidly as the area of a_2 increases and it decreases with the decrease of a_2 area. The section area a_2 is equal to $s \cdot h$, where s is the circumferential length of the air feeder hole and h is the clearance of bearing at the air feeder hole. Then, when the rotating shaft is displaced by some causes and the clearance h changes, the shaft can be put back in the former state by the intermediate pressure p_0 because this pressure develops the pressure distribution into the bearing clearance with a strong restoring force in proportion to the change of h. This automatic control action must be strongest at the inflection point of the characteristic curve of the intermediate pressure p_0. Therefore, the theoretical values of $a_2/a_1=1.56$ and $p_0/p_t=0.63$ are desirable for the air-lubricated bearing in the range of $p_0>2\,\mathrm{kg/cm^2}$ to maintain a stable rotating state.

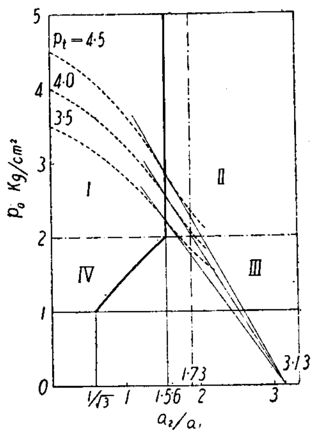

Fig. 2

There is an unnatural flection in the transition part of the theoretical curve of the inflection points from the section (I) to (IV). It is due to the fact that the theoretical curve is obtained by assuming that the air-flow through bearing clearance is equal to the flow through a nozzle under the condition that viscosity of air is negligible.

3. Bearing Testor

In order to verify the effects of restriction before the air feeder hole, the change of friction is measured and inspected by using the balance beam testor shown in Fig. 3 which is reported in our preceeding paper. In Fig. 3, compressed air is supplied through a needle valve V to the air feeder hole H of the bearing bushing

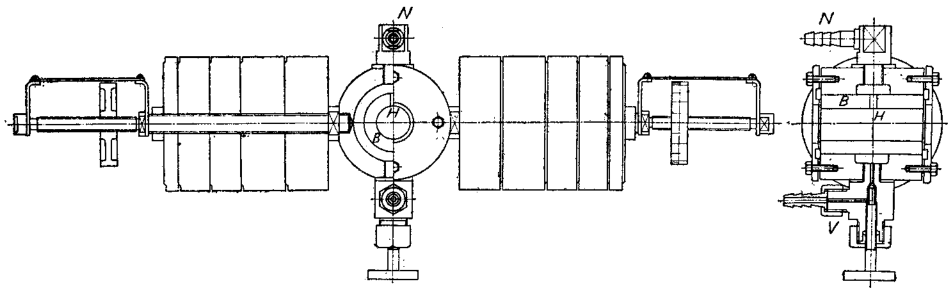

Fig. 3

B. By comparing with Fig. 1, it is obvious that the needle valve corresponds to the primary restrictor N_1, and the air feeder hole H to the secondary restrictor N_2. The intermediate pressure p_0 is measured through a nipple N in Fig. 3. In this testor, it can be considered that p_t is about equal to p_0 when the needle valve is sufficiently opened, and that p_0 can be held at any value between the two extremes of reservoir pressure p_t and atomospheric pressure p_a by controlling the needle valve.

4. Conditions of the Experiment

In this experiment, it is considered that the bearing conditions, such as revolving speed and bearing diameter, have little or no effect. Then, the revolving speed of 3400 rpm is chosen as a constant and the bearing diameter is chosen to be 20 mm. But the bearing clearance 2δ which governs the viscous resistance is taken at various values, and the influences of them are inspected. The bearing clearance of more than 100μ is chosen as a group of large clearances and of about 20μ, as a group of small clearances. The bearing length l is 60, 40, and 30 mm.

Furthermore, the air feeder hole, as the secondary nozzle, may have great influences on restriction effect. It is investigated of two cases: the former is the case in which the length of rectangular hole is changed to 20, 30 and 40 mm where the width is 4 mm, and the latter is the case in which the number of air feeder holes is changed to 1, 2, and 3 where the diameter of hole is 4 mm.

5. Methods and Results of the Experiment

At first coefficient of friction is measured by the balance beam testor under the condition of $p_0 = p_t$ in which the needle valve is fully opened. This primary coefficient of friction, which equals to the coefficient of friction reported in the preceeding paper, is taken as μ_1. The coefficient μ_1 takes the minimum value for each bearing load by adjusting the pressure p_0. In such operating state the shaft rotates non-eccentrically. This minimum value is taken as $\mu_{1\,min}$ and the most suitable pressure for the minimum friction as $p_{0\,min}$. By controlling the needle valve, the secondary coefficient of friction is obtained when the pressure p_t is gradually increased under the condition in which the intermediate pressure p_0 is always kept at the value of $p_{0\,min}$. Then the state of non-eccentric rotation can be maintained by keeping the pressure p_0 to $p_{0\,min}$ not with standing the change of the pressure p_t. Furthermore, restrictions of the needle valve and the air feeder hole, which operate in series, may have an automatic control effect and may reduce the coefficient of friction by increasing the dynamic stability because the coefficient of friction of the air-lubricated bearing has a statistical mechanism in which the coefficient includes the summation of slight metallic contacts.

This deduction has been verified by experimental results as shown in Fig. 4. In Fig. 4, which shows the relation between coefficient of friction and the supply pressure, the secondary friction μ_2 decreases at first with the increase of the pressure p_t and it increases after taking the minimum value $\mu_{2\,min}$ which is much smaller than $\mu_{1\,min}$.

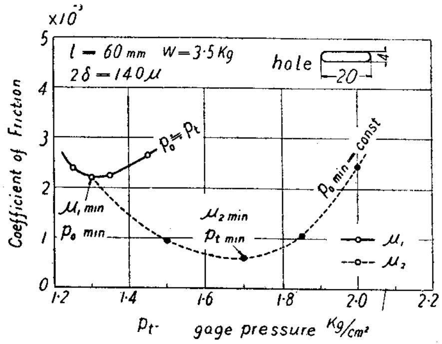

Fig. 4

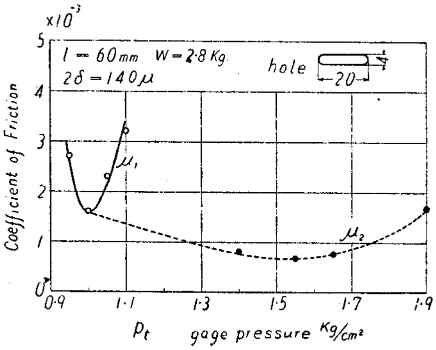

Fig. 5

The most suitable reservoir pressure $p_{t\,min}$ can be considered to depend on the minimum value $\mu_{2\,min}$. Fig. 5 shows the data of the same bushing shown in Fig. 4, but the bearing load W is reduced from 3.5 kg to 2.8 kg. The both secondary minimum values are reduced to such a small value as 0.0006 from the primary minimum value $\mu_{1\,min}$.

Comparing the case of a bushing with a large bearing clearance such as 140μ

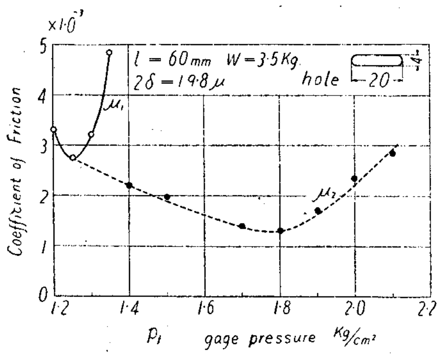

Fig. 6

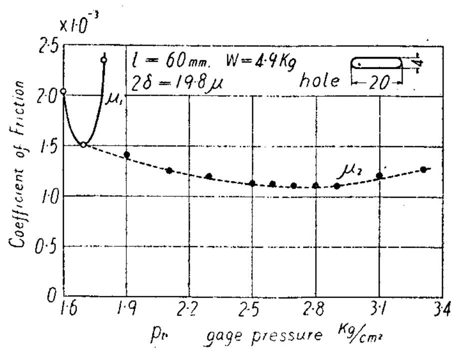

Fig. 7

shown in Fig. 4 and 5, the case of a bushing with a small bearing clearance of 19.8μ is shown in Fig. 6 and 7. These data take the same tendencies in which $\mu_2 \sim p_t$ curves are always concave and $\mu_{2\,min}$ is smaller than $\mu_{1\,min}$. But μ_2 curve becomes flat as shown in Fig. 7 when the load W increases to 4.9 kg from 3.5 kg.

Comparing, however, Fig. 7 with Fig. 6, the value of $\mu_{2\,min}$ is almost the same, but that of $\mu_{1\,min}$ differs considerably. Then it can be considered that the effect of the restrictions in series becomes greater with decrease of bearing load because the range of light load is dynamically unstable. This fact becomes obvious when the relation between the minimum values of μ_1 and μ_2 and bearing load is shown as in Fig. 8, 9 and 10. In Fig. 8, $\mu_{1\,min}$ and $\mu_{2\,min}$ curves come closer to each other and take a certain constant value with the increase of bearing load, and so the effect of a restrictor before the air feeder hole is little in the range of heavy load. Then this

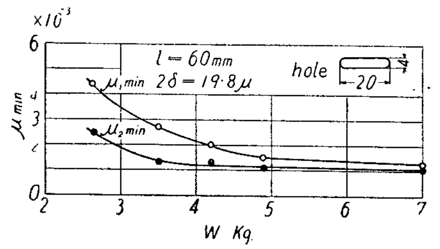

Fig. 8

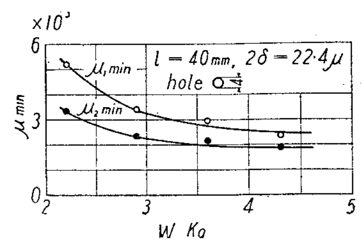

Fig. 9

fact shows that the restriction before the air feeder hole has an effect on increasing dynamic stability only. The cases shown in Fig. 9 and 10, in which the length of bearing has changed from 60 mm to 40 and 30 mm, seem to have the same tendency.

At the same time when measuring coefficient of friction, the quantity of flow is measured by a rota meter which is set between the air compressor and the air reservoir. Fig. 11 shows the quantity of flow Q l/min which is coverted into volumetric quantity at atomospheric pressure. The theoretical line in Fig. 11 is drawn by

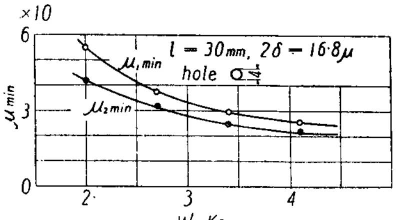

Fig. 10

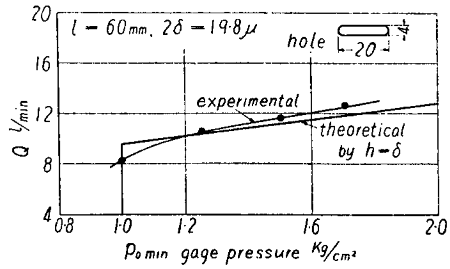

Fig. 11

formula (1) with the assumption of $h=\delta$. The experimental values coincide considerably with the line.

As mentioned above, frictional coefficient can be reduced by setting a restrictor before the air feeder hole as theoretical consideration, which also coincedes with the experimental results for the quantity of flow. Then the pressure $p_{t\,min}$ and the ratio a_2/a_1, which are the conditions to induce the state of $\mu_{2\,min}$, are important in the consideration. Concerning to the pressure $p_{t\,min}$, the experimental values are obtained much by the method as shown in Fig. 4. Fig. 12 shows the values of $p_{t\,min}$ taken as the ratio $p_{0\,min}/p_{t\,min}$. The values of $p_{0\,min}/p_{t\,min}$ in Fig. 12 distribute themselves in a considerably wide range. It is because the value of $p_{t\,min}$ is difficult to determine when the μ_2 curve is flat as shown in Fig. 7. But the plotted values bury the valley of an unnatural flection of the theoretical curve; and no defference caused by the air feedes hole, the bearing length or clearance is recognized. The average value of $p_{0\,min}/p_{t\,min}$, obtained experimentally, is 0.76. It is larger than the theoretical value of 0.63, but may become close to it as the pressure $p_{0\,min}$ increases because the experimental result is obtained in the transition stage of $p_{0\,min}=2.0\sim3.5$ kg/cm. In consideration of the ratio a_2/a_1, a_2 can be calculated for the non-eccentric rotation by $a_2=s\delta$, but a_1 can not be measured exactly in this experiment. But the ratio a_2/a_1 has a function given by equation (3) with the pressure ratio $p_{0\,min}/p_{t\,min}$. Then a_1 can be

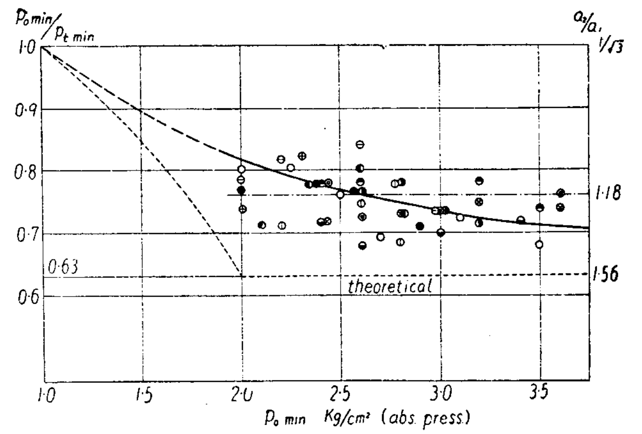

Fig. 12

calculated when $p_{0\,min}/p_{t\,min}$ is given. In the right side of Fig. 12, a_2/a_1 is scaled for the ratio $p_{0\,min}/p_{t\,min}$. The suitable values of $p_{0\,min}/p_{t\,min}$ and a_2/a_1 can be decided by using Fig. 12.

6. Conclusion

It is verified in this report, that, in order to reduce coefficient of friction of air-lubricated bearings, it is effective to attach a restrictor before the air feeder hole on an air supply line because the two restrictions of its primary restrictor and the air feeder hole in series produce an automatic controllability by changing the intermediate pressure accordingly as the bearing clearance changes, and that this effect becomes greater in the range of light load under which the operating state is considered unstable. In order to have a full effect of this on the air-lubricated bearing, $p_{0\,min}$ is determined at first for the bearing conditions, such as bearing length l, diameter d, clearance 2δ, revolving speed n and load W, because $p_{0\,min}/q$, where q is the average bearing pressure W/ld, can be estimated by the results reported in the previous paper. Secondly the reservoir pressure $p_{t\,min}$, which is the line pressure from the air compressor, is determined against $p_{0\,min}$ accordingly with the results of this report in which the suitable value of $p_{t\,min}/p_{0\,min}$ is given as 1.3~1.6. Then a_1, the section area of the primary restrictor, is determined against a_2 which is the preduct of a half of the bearing clearance 2δ and the circumferential length of the air feeder hole s when the ratio a_2/a_1 is suitable in the range of 1.5~1.1.

Then it becomes possible to increase dynamic stability of the air-lubricated bearing and to reduce frictional resistance, when the air supplying pressure from the air compressor and the section area of the primary restrictor are properly disigned and treated in relation to bearing conditions.

Reference

1) T. Sasaki and H. Mori: "On the Characteristics of Air-Bearing", Memoirs of Faculty of Engineering, Kyoto University, Vol. 13, No. 1, pp. 21~28.