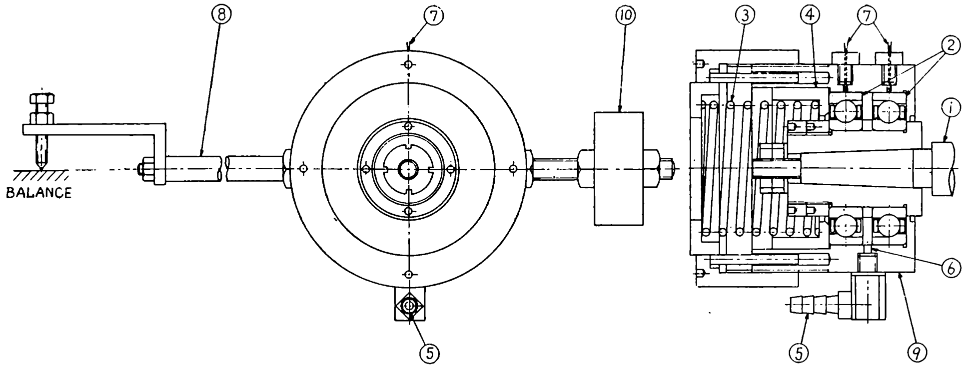

Fig. 1. Bearing testor

the high speed spindle and ② shows a pair of precision single-row angular-contact ball bearings of 40 mm bore with contact angle of 15° having bakelite cages installed facing each other and preloaded by the spring ③, and the preloads are measured by the compression of the spring ③. The outer surface of the sleeve ④ and its guide surface of the bearing housing are finished finely and precisely to give an uniformly distributed preloading in the circumference of outer ring of the bearing. The oil fog enters from the nipple ⑤ through the hole ⑥ and branches off right and left and lubricates the two test bearings. The bearing temperature is measured by the thermocouple ⑦ at the outer surface of the outer ring. The frictional moment of the ball bearing is measured by putting the end of the arm ⑧ attached to the housing ⑨ on the balance as shown in Fig. 1 and reading the force acting on the free end of the arm. For this reason, the housing, which can be freely rotated, is balanced by the counter-weight ⑩ previously.

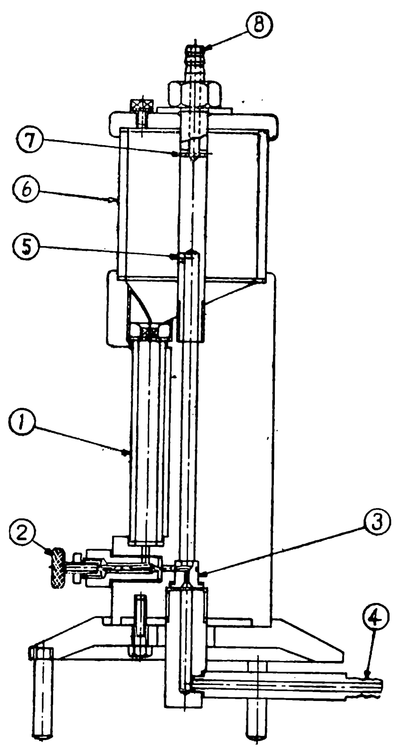

Fig. 2. Oil fog producing unit.

Fig. 2 shows the oil fog producing unit. ① is a level gage glass to measure the consumption of oil and ② is a needle valve to regulate the amount of oil supplied. The oil entering into the top portion of the nozzle ③ is atomized by the air blowing out through the orific of the nozzle ③. The compressed air is supplied from

④. The oil fog, which is the mixture of oil and air, blows out through the radial hole ⑤ into the glass cylinder ⑥. As the large size oil particles adhere to the inner wall of the glass cylinder and flow down to the funnelled bottom and return to the level gage glass ①, the oil fog consisting of very fine particles of oil can be taken out from the nipple ⑧ through the radial hole ⑦. The oil returned to ① flows to the nozzle ③ through the regulating valve ② again, then the descending level of oil in ① indicates the oil consumption of such circulation system. The air consumption is measured by the rotameter before entering the nipple ④.

The variety of oils used in this experiment is tabulated as ABCDEFGHI in Table 1 and the spindle oil B is used in all experiments except the experiment (3) which is for the influence of viscosity and additive of oil.

Table 1. Character of oil.

Sample	Spindle Oil				Turbine Oil		Spindle Oil		
	A	B	C	D	E	F	G	H	I
Sp. Gr. 15/4°C	0.854	0.878	0.871	0.880	0.876	0.878	0.878	0.878	0.879
Viscosity 100°F CS	5.88	9.78	13.6	21.8	33.8	43.5	9.75	9.62	9.86
210°F CS	1.81	2.47	3.04	4.03	5.34	6.24	2.46	2.44	2.47
Additive	None	"	"	"	"	"	Oiliness Agent	Phosphorous Type E.P. Agent	Sulfurchlorine Type E.P. Agent

3. Experimental Result and Consideration

(1) The influence of oil consumption

The Fig. 3 is the diagram showing the influence of oil consumption upon the performance characteristics of the oil fog lubrication under the definite quantity of air consumption and rotating speed of 10,400 rpm with the oil B in Table 1. The temperature rise of the bearing, that is the difference between the outer ring temperature and room temperature, is the average value of the two bearings. The temperature rise of the air means the temperature difference of

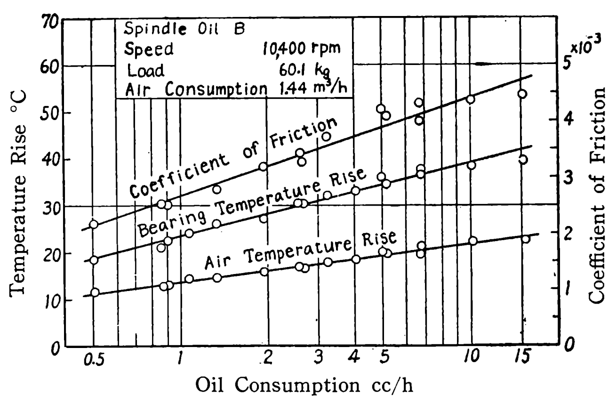

Fig. 3. The influence of oil consumption upon the performance characteristics of oil fog lubrication at 10,400 rpm.

the air temperature flowing out from the bearing and the air temperature flowing into the bearing. To measure the temperature of the air flowing out from the bearing, a thermocouple is placed near the spring-side bearing and this thermocouple is protected from the radiant heat by the leaf aluminium although it is not shown in Fig. 1. The amounts of oil and air consumed, which are shown in the diagram of Fig. 3, are half the amount of total consumption i.e. the amount reduced for a single bearing, and the value of air consumption in this case is the value reduced for atmospheric condition. The coefficient of friction f is calculated by the following formula,

$$f=\frac{M}{Pd/2}$$

where M is the frictional moment reduced for a single bearing, P is the resultant bearing load of single bearing and d is the inner diameter of the ball bearing.

As seen by Fig. 3, the coefficient of friction and temperature rise of the bearing and air increase with oil consumption under a definite quantity of air consumption, and it seems that the less the amount of oil supplied, the better lubricating performance is obtained in this range of oil consumption. However in practice the oil consumption of 0.5cc/h is too little for a stable running condition, and when the oil consumption is so little, the friction of bearing rises intermittently depending upon partial shortage of oil in the bearing in spite of the small average coefficient of friction. Therefore it is desirable to supply oil in the amount of more than 1 cc/h to get a stable running condition. The

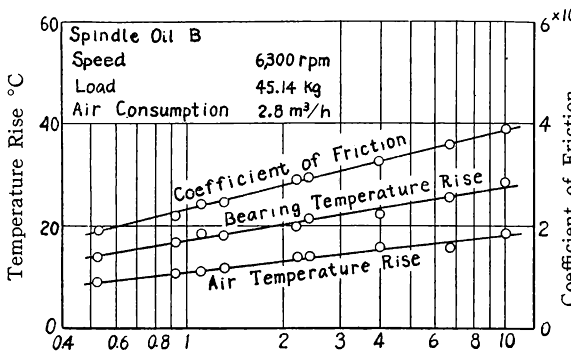

Fig. 4. The influence of oil consumption upon the performance characteristics of oil fog lubrication at 6,300 rpm.

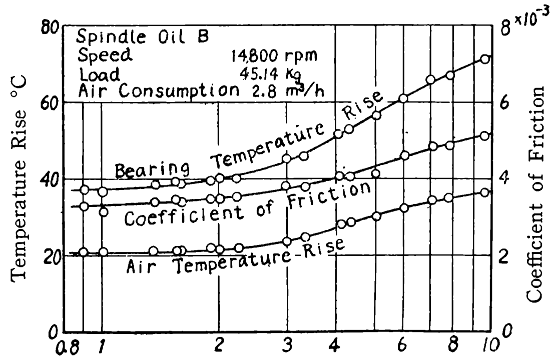

Fig. 5. The influence of oil consumption upon the performance characteristics of oil fog lubrication at 14,800 rpm.

temperature rise of the air is 55~60% of the temperature rise of the bearing. Therefore, the cooling effect of the air seems to be great. Fig. 4 shows the influence of oil consumption under 6,300 rpm and the same tendency as in Fig. 3 can be seen. Fig. 5 and Fig. 6 show also the influence of oil consumption under 14,800 and 20,500 rpm, but different tendencies from those in Fig. 3 and Fig. 4 are shown. Namely, as shown in Fig. 5 and Fig. 6, the coefficient of friction and temperature rise not only do not decrease in proportion to the decrease of oil consumption, but the minimum values appear in Fig. 6. It is considered that this phenomenon occurs owing to the fact that the local boundary friction in the bearing increases in the high speed running and it offsets the decrease of fluid friction which is due to the decrease of oil supply. Accordingly, more amount of oil should be supplied as the running speed rises in order to keep a good lubricating condition.

Fig. 7 shows the experimental results of ordinary oil lubrication. The oil is supplied through the hole ⑥ in the Fig. 1 by the gear pump. Although the coefficient of friction increases as the oil consumption increases, the bearing temperature rise reaches the maximum value at the oil consumption of about 150 cc/h, and it decreases as the oil consumption increases owing to the cooling effect of oil.

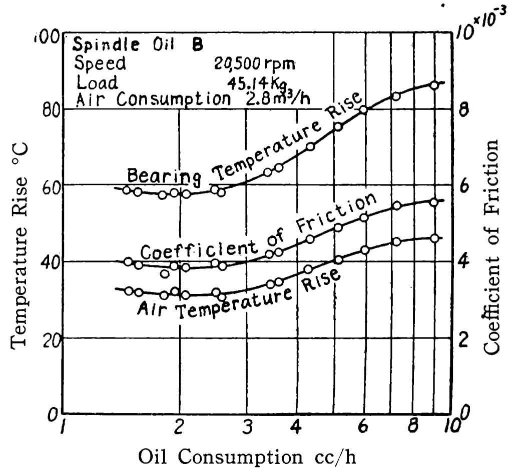

Fig. 6. The influence of oil consumption upon the performance characteristics of oil fog lubrication at 20,500 rpm.

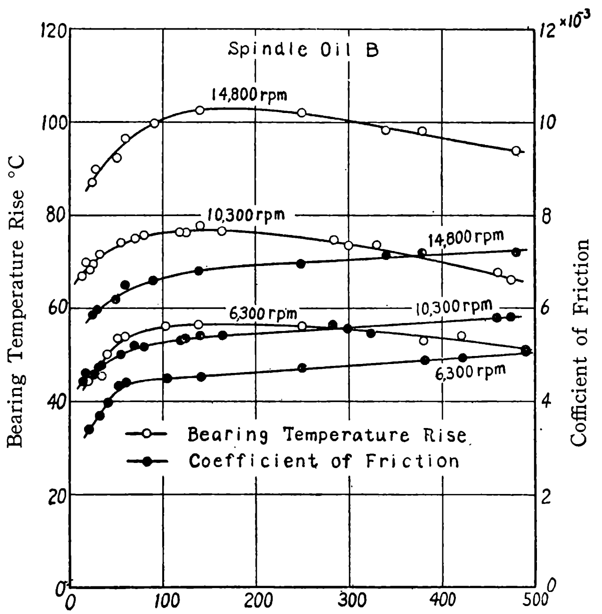

Fig. 7. The influence of oil consumption upon the ordinary oil lubrication.

Fig. 8 shows the comparison of the ordinary oil lubrication with the oil fog lubrication. As seen in the diagram, the coefficient of friction and bearing temperature rise in the oil fog lubrication are considerably less than those in the oil lubrication, in addition, the difference between both lubricating methods becomes greater as the running speed of bearing rises. From this fact, it is recognized that the oil fog lubrication is highly suitable for high speed ball bearing.

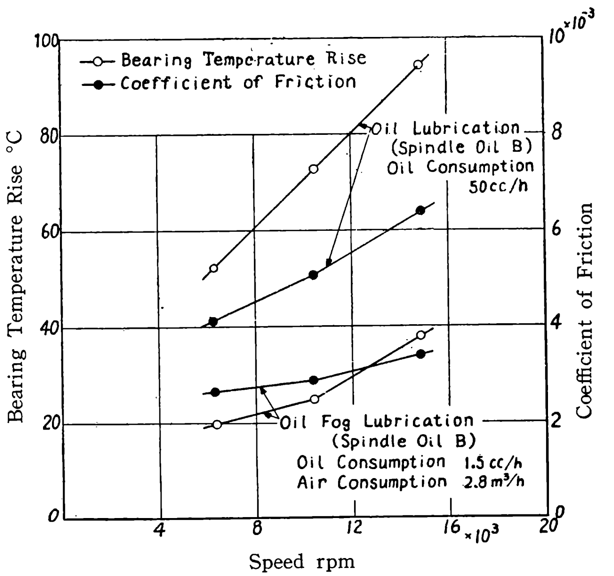

Fig. 8. Comparison of ordinary oil lubrication and oil fog lubrication.

(2) The influence of air consumption

Fig. 9 and Fig. 10 show the influence of air consumption in the cases of rotating speed of 10,300 and 20,400 rpm under a definite quantity of air consumption. The temperature rise decreases with the increase of air consumption and the coefficient of friction increases slightly. This is due to the increase of oil viscosity owing to the decrease of bearing temperature. In Fig. 9 the experimental results of grease lubrication are shown on the line of no air consumption. The grease used in the experiment is a lime-base grease consisting of Ca soap and mineral spindle oil with the consistency of 230 degree. In this case, in spite of the somewhat smaller coefficient of friction than the value of the oil fog lubrication, the bearing temperature rise is greater than that in the oil fog lubrication with a little air consumption. This phenomenon shows the effectiveness of of air cooling.

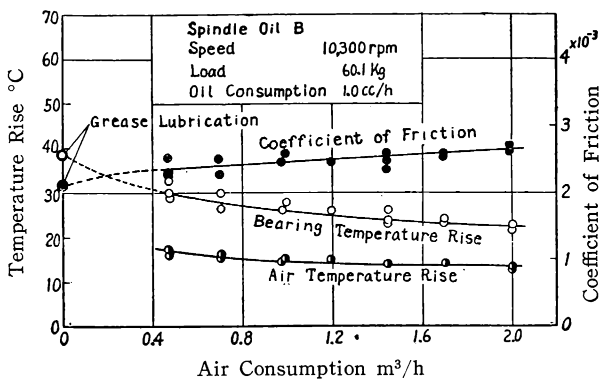

Fig. 9. The influence of air consumption under a definite oil consumption at 10,300 rpm. The results of grease lubrication are shown on the line of no air consumption.

Fig. 11 shows the cooling process of the bearing for the various air consumption

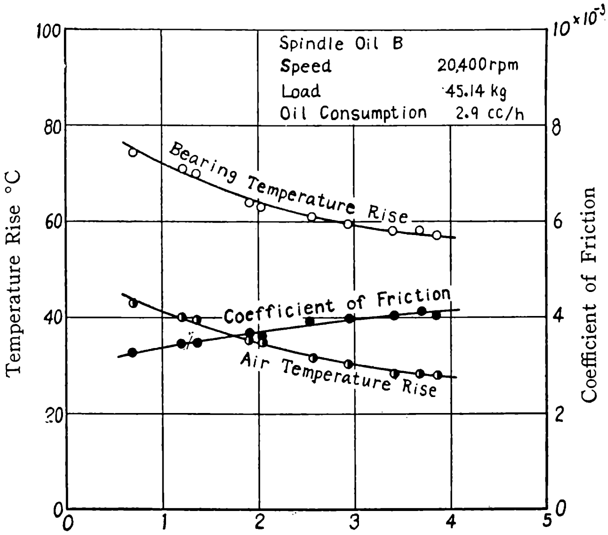

Fig. 10. The influence of air consumption under a definite oil consumption at 20,400 rpm.

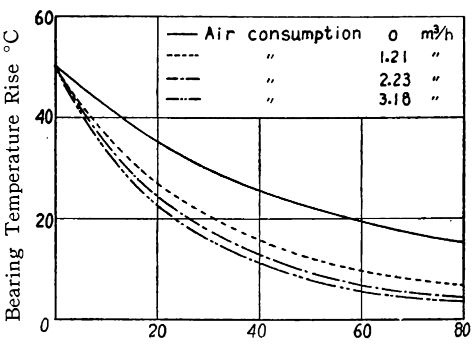

Fig. 11. Curves showing the cooling speed of bearing at rest under the various air consumptions.

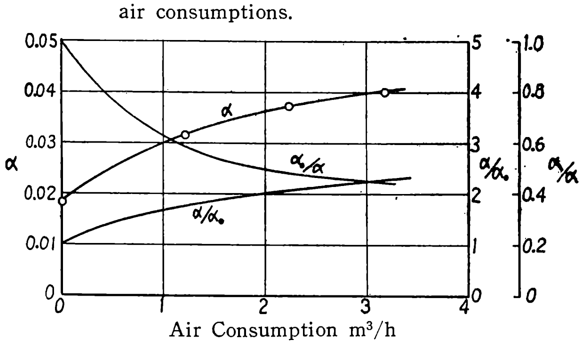

Fig. 12. Curves of α, α_0/α, α/α_0 under various air consumptions.

of 0, 1.21, 2.23 and 3.18 m³/h measured for the bearing, which has been running at 20,400 rpm and has been stopped after the bearing temperature rise has become stational. The greater the air consumption, the higher the cooling speed, and these characteristic curves of cooling speed, and of cooling effect can be formulated as follows

$$T = 50e^{-\alpha t} \qquad (1)$$

where T is the bearing temperature rise measured on the outer surface of the outer ring and t is time in minute and α is an experimental constant. And the relation between α and air consumption is shown in Fig. 12 and it can be formulated as follows:

$$\alpha = 0.018 + 0.012Q^{0.52} \qquad (2)$$

where Q is the air consumption. Differentiating equation (1),

$$\frac{dT}{dt} = -\alpha T \qquad (3)$$

is obtained. As the dissipating amount of the heat carried away with air per unit time is approximately expressed as follows:

$$H_1 = C\alpha T \qquad (4)$$

where C is the heat capacity of the apparatus. Then, denoting α of no air consumption as α_0, the increasing ratio of dissipation heat due to the increase of air consumption is expressed with the ratio α/α_0. As shown in Fig. 12, the increasing ratio of the dissipation heat takes moderately large value under the air consumption of smaller than 1 m³/h, but it does not increase so much for the air consumption. From equation (4), as αT becomes constant in the case the amount of heat disspated is constant and

$$\frac{\alpha_0}{\alpha}=\frac{T}{T_0} \qquad T<T_0$$

where T_0 is T of no air consumption, α_0/α comes to denote the ratio of the bearing temperature rises and α_0/α decreases as the air consumption increases as shown in Fig. 12.

Fig. 11 and Fig. 12 show the cooling characteristics of the bearing which is at rest, and, although the influence of the air which is caused to flow by the rotation of bearing is not considered, the qualitative tendency of influence of air consumption for the cooling characteristics of the running bearing is anticipated to be in a similar condition.

Let us now define the technical term "oil-air ratio" as the ratio of oil volume content to air volume content in the oil fog. Then the oil consumption varies directly proportionally to the air consumption under constant oil-air ratio.

Fig. 13 shows the relation between the coefficient of friction or bearing temperature rise and the air consumption under the oil fog lubrication of the two specified oil-air ratio. As shown in the diagram, the minimum values of the coefficient of friction and bearing temperature rise occur at a certain quantity of air consumption, and the coefficient of friction and temperature rise of bearing rise from their minimum values in the range of less air consumption owing to the shortage of oil supply. When the air consumption increases, the coefficient of friction increases remarkably, and the bearing temperature rise increases in parallel without decreasing as in the case of Fig. 9 and Fig. 10 in which the oil-air ratio decreases as the air

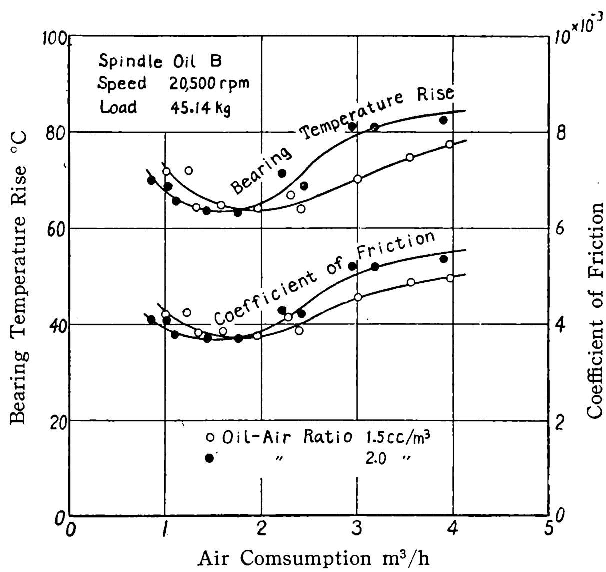

Fig. 13. The influence of air consumption upon the performance characteristics of oil fog lubrication of two specified oil-air ratios.

consumption increases. This phenomenon means that when the air consumption and the cooling effect of air increase, the oil consumption and viscous friction of oil increaese simultaneously under the constant oil-air ratio and the viscous friction effect surpasses the cooling effect.

(3) The influences of viscosity and additive of oil

Fig. 14 shows the results of an experiment on the spindle oil A, and Fig. 15 the results on the turbine oil F. These diagrams show the same tendency as Fig. 6, but owing to the higher viscosity the coefficient of friction and bearing temperature rise in Fig. 15 are much greater than those in Fig. 14. Fig. 16 shows how the coefficient of friction and bearing temperature rise change depending upon the viscosity change of six kinds of lubricating oil. In this diagram, the scale of abscissa on the full line means the viscosity at 100°F, and the scale on the broken line means the viscosity at the running tempearature of the bearing. It is seen that the coefficient of friction and bearing temperature rise remarkably change directly proportionally to the oil viscosity at the running temperature of bearing. This phenomenon means that the friction of ball bearing under the fog lubrication depends principally on the viscous friction of oil, and the oil fog lubrication is chiefly of the oil film lubrication in spite of very little amount of oil.

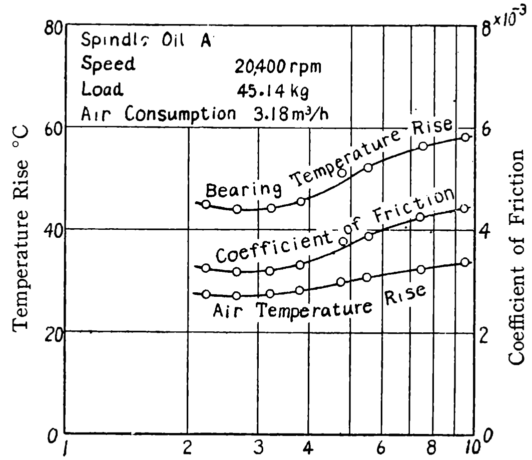

Fig. 14. The influence of oil consumption at 20,400 rpm when spindle oil A is used.

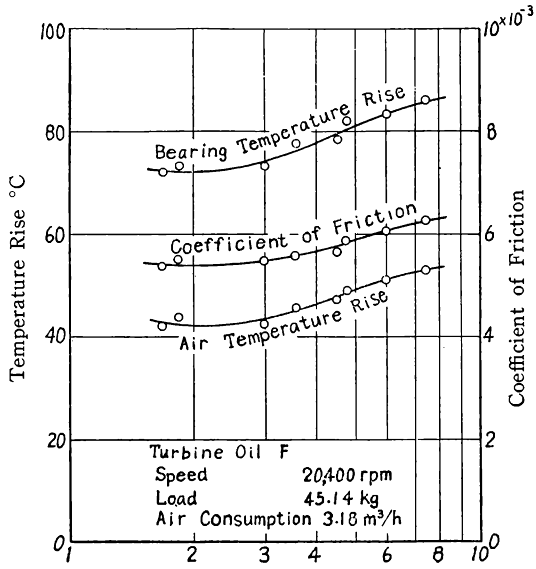

Fig. 15. The influence of oil consumption at 20,400 rpm when turbine oil F is used.

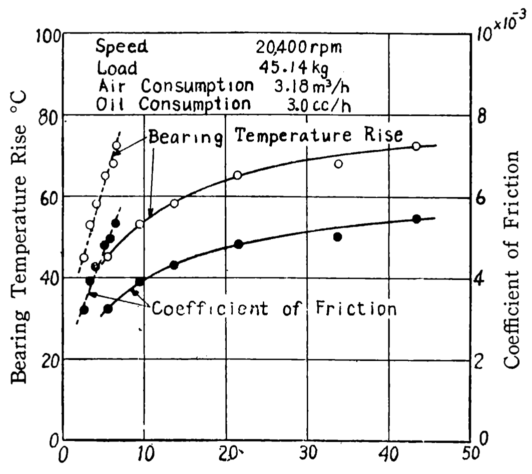

Fig. 16. The influence of viscosity upon the performance characteristics of oil fog lubrication.

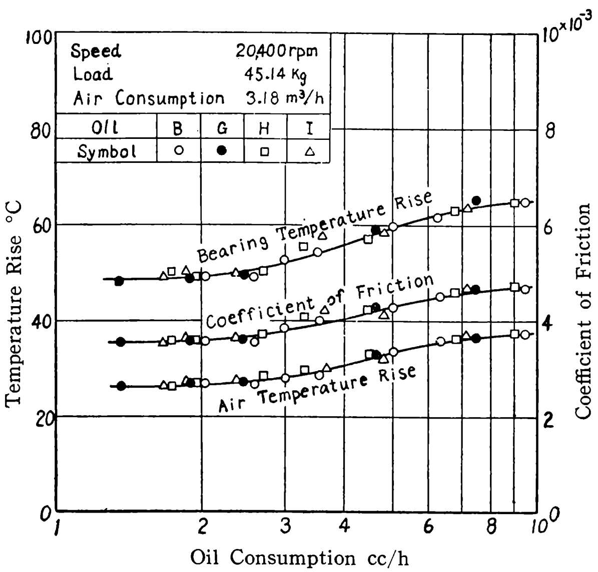

Fig. 17. A diagram showing the effect of additives.

Fig. 17 is the experimental results of the effect of additives. Oil B contains no additive, G contains oiliness additive, H and I contain extreme-pressure (E. P.) additives and these four kinds of oil have the same viscosity character. It is seen as if the additives have no effect on the bearing performance in such an operating condition. But, in the operating condition of very little oil consumption, oil G gives rather stable running of the ball bearing than the other oils.

(4) The influence of preload

The influence of preload is shown in Fig. 18 for five different oil consumptions and a definite quantity of air consumption. The frictional moment and bearing temperature rise increase proportionally to the increase of preload and these curves rise in parallel to the increase of oil consumption. This phenomenon means that in the oil fog lubrication, the viscous friction depending on the amount of oil supply and the friction due to the bearing load are superposed.

(5) The influence of rotating speed of ball bearing

Fig. 19 shows the relations between the rotating speed of bearing and the coefficient of friction or the temperature rise of bearing. As the rotating speed increases the bearing temperature rises very rapidly, on the contrary the coefficient of friction increases gradually, and the increasing rate of temperature and friction decrease as the rotating speed increases.

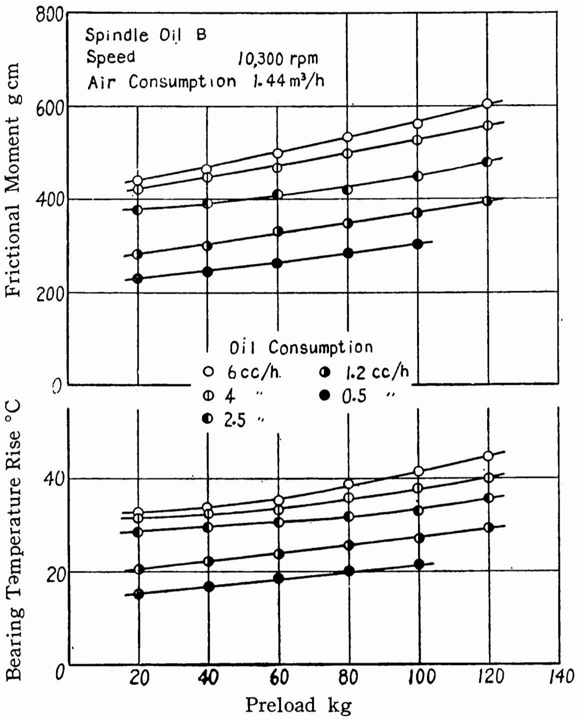

Fig. 18. The influence of preload upon frictional moment and bearing temperature.

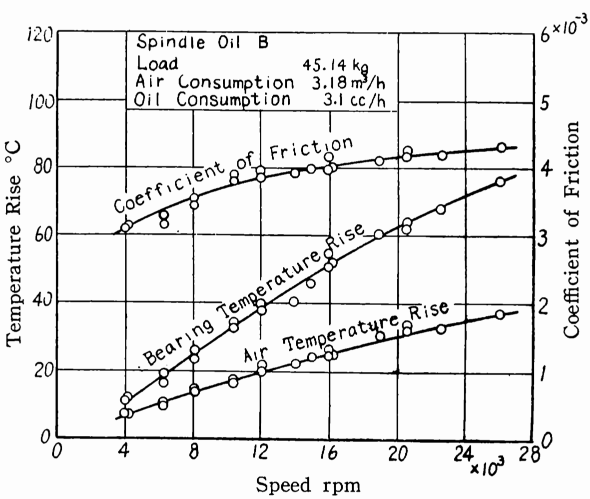

Fig. 19. Relations between rotating speed and coefficient of friction.

The frictional heat of bearing H_2 is expressed in the following formula

$$H_2 = aMN \tag{5}$$

where a is a constant, M is the frictional moment of bearing and N is the rotating speed of bearing. And the frictional moment of ball bearing under an ordinary fog lubrcating condition depends on the fluid friction, therefore the frictional moment is expressed as follows:

$$M = b\mu N \tag{6}$$

where μ is a coefficient of viscosity of oil used and b is a constant. Accordingly,

$$H_2 = k_1\mu N^2 \quad k_1 = ab. \tag{7}$$

At a steady state, H_1 is equal to H_2, consequently from equations (4) and (7)

$$T = \frac{k_1}{C\alpha}\mu N^2. \tag{8}$$

The coefficient of friction is expressed in the following formula

$$f = \frac{k_2}{P}\mu N \tag{9}$$

where f is the coefficient of friction, P is the bearing load and k_2 is a constant.

Thus the bearing temperature rise is proportional to the square of the rotating speed, and the coefficient of friction under constant load is linearly proportional to the rotating speed. Furthermore, the temperature rise owing to the increase of the rotating speed brings about a decrease of the oil viscosity μ in equations (8) and (9).

For that reason, the curves of temperature rise and coefficient of friction bends downward as the rotating speed increases.

4. Conclusion

It is possible to lessen remarkably the friction of ball bearing running at high speed by the aid of the oil fog lubrication due to a small quantity of oil supply. But, when the amount of oil supply exceeds a certain limit, the state of lubrication becomes hydrodynamical. For example, the oil supply of 5 cc/h corresponds to the above limit for a single-row angular-contact ball bearing of 40 mm bore at the *dn* value of 800,000. Although the minimum oil requirement becomes greater as the rotating speed increases, the above mentioned ball bearing can be lubricated safely with the oil supply of 3 cc/h at the *dn* value of 800,000.

The cooling effect of air is very efficient, but when oil and air consumption increase together the viscous friction effect of oil surpasses the cooling effect of air under a definite oil-air ratio.

The coefficient of friction and bearing temperature rise remarkably change directly proportionally to the oil viscosity at the running temperature of bearing. After all, the oil fog lubrication consists very much of the oil film lubrication in spite of a little amount of oil, and the effect of viscosity is predominant.

The frictional moment and bearing temperature rise change proportionally to the preload, and when the oil supply is increased under a definite preload the moment and temperature become greater by the increase of viscous friction.

As the rotating speed of ball bearing increases the bearing temperature rises very rapidly, on the contrary the coefficient of friction increases gradually, and the increasing rates of temperature rise and coefficient of friction decrease as the rotating speed increases.