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Study on the Scoring Resistance of Lubricating Oil in Four Ball Test*

(4th Report, Relation between the Scoring Resistance and the Maximum Contact Surface Temperature)

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In this paper, for two kinds of non-EP mineral oils, the effects of lubricating method, operating time and sliding velocity on initial seizure load were examined by step-load method using a Shell type Four Ball Machine, and the relation between the thermal strength of boundary lubricating film and the maximum contact surface temperature was clarified. The results obtained are summarized as follows:

(1) There is only a negligible difference in maximum contact surface temperature (T_{cmax}), at scoring between the results in 5 seconds test and the results in 300 seconds test.

(2) Without regard to lubricating method and operating time, (T_{cmax}), increases with an increase in sliding velocity, and in the sliding velocity range higher than 35 or 46 cm/s, (T_{cmax}), is nearly constant.

(3) In 300 seconds test, a bulk temperature rise in steel balls has great effect on (T_{cmax}).

1. Introduction

In the third report⁽¹⁾, the effects of bulk oil temperature and sliding velocity on last non-seizure load (*LNSL*) were examined by 60 seconds test according to JIS K 2519. As the result, it was made clear that lubrication state of contact surface at *LNSL* differs much depending on the bulk oil temperature and the sliding velocity, and the 60 seconds test using new steel balls in each load step is not necessarily a proper method to examine the relation between thermal strength of boundary lubricating film and contact surface temperature.

In this paper, in order to easily obtain boundary lubrication state, by a method of increasing the axial load progressively till scoring on the same steel balls, namely, by step-load method the effects of lubricating method, operating time and sliding velocity on initial seizure load (*ISL*) are examined, and the relation between thermal strength of boundary lubricating film and maximum contact surface temperature and further the effect of bulk temperature rise in steel balls on thermal strength of boundary lubricating film are clarified. In view of the former studies about Four Ball test, the equation of contact surface temperature and the relation between scoring resistance of lubricating oil and contact surface temperature have been examined by many researchers^{(4)~(7)}, including H. Blok⁽²⁾ and R. S. Fein⁽³⁾. But every equation of contact surface temperature has been obtained only by the analysis of flash temperature under

the assumptions that a steel ball is approximated by a semi-infinite body in which there is no heat transfer except in heat supply region and the heat source moves in uni-direction with constant velocity. And R. S. Fein⁽³⁾, H. Czichos⁽⁷⁾, and so on made scoring experiments under short operating time such as 5 or 10 seconds in order to satisfy the assumptions used to obtain the equations as much as possible. Therefore, old equations are, strictly speaking, only applicable for the experimental results obtained under limited operating conditions. Then, so as to correct the defect, in the first report⁽⁸⁾ the authors have proposed an original flash temperature and a bulk temperature rise in steel ball in consideration of finiteness of steel ball and heat transfer from steel ball surface to lubricating oil and ball fixtures when heat is supplied repeatedly to steel ball surface, and in the second report⁽⁹⁾ they have offered an equation of maximum contact surface temperature which is also applicable to a long operating time test in which a bulk temperature rise in steel balls occurs in operation. By the way, about maximum contact surface temperature at scoring, if there is a larger difference than experimental error between the values in short operating time test where the bulk temperature rise in steel balls in operation can be ignored and the values in long operating time test where the bulk temperature rise in steel balls in operation can not be ignored, this is a great problem. In this paper, concerning non-EP mineral oils, the above mentioned problem is made clear and the relation between the scoring resistance of lubricating oil and the maximum contact surface temperature is investigated.

2. Experiments

2.1 Testing machine and lubricating oil

In experiments a Shell type Four Ball

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machine with steel balls of 12.7 mm in diameter was used. Steel balls were made of high carbon chromium bearing steel SUJ2, and their surface hardness was 62 to 63 H_{RC} , the retained austenite content at the surface being 13.2 to 15.6 percent and the surface roughness 0.02 to 0.04 μR_{max} . As lubricating oils, #90 base gear oil and #200 turbine oil were used. Table 1 shows their properties. In both lubricating oils EP additives were not contained. #200 turbine oil was the same one used in the third report, while #90 base gear oil differed a little. But as the experimental results obtained in the conditions that bulk oil temperature was 20°C and that sliding velocity was in the range of 13 to 119 cm/s, there was little difference in $LNSL$, coefficient of friction and wear scar diameter from the values of #90 base gear oil used in the third report.

2.2 Lubricating method

Two kinds of lubricating methods were used. One was oil bath lubrication with 12 ml in oil quantity, the other was drop lubrication in which a fresh lubricating oil was supplied continuously around contact regions of three stationary steel balls at the rate of 7.4 ml/min and used lubricating oil was exhausted naturally through a hole opened at the middle of oil pot bottom. In oil bath lubrication, all the surface of stationary steel ball and about 60 percent of the surface of rotating steel ball were submerged in lubricating oil.

2.3 Operating method and judgement of scoring

As clarified in the third report ⁽¹⁾, in examining thermal strength of boundary lubricating film, the 60 seconds test in which steel balls were changed with new ones on each axial load was not necessarily a proper method because at some bulk oil temperature and in some sliding velocity range a scoring occurred suddenly from the state of elastohydrodynamic lubrication without lasting boundary lubrication state. Therefore in this experiment, in order to make a scoring occur after lasting boundary lubrication state the step-load test method was used, that is, on the same steel balls the axial load was increased stepwise till scoring.

Experiments were made at room temperature (from 13 to 23°C), and the machine was operated at constant rotating speed after a fixed axial load was imposed by an oil pressure equipment. The range of sliding velocities was from 13 to 115 cm/s (from 350 to 3000 rpm in axial rotating speed). Steel balls were cleaned with petroleum benzine

every axial load, dried and cooled to room temperature. Dimensions of wear scars in three stationary steel balls were measured in sliding direction by reading the microscope to the limit of 0.01 mm without removing steel balls from oil pot, and their arithmetical average value was defined as wear scar diameter at applied axial load. The difference in wear scar diameters of three stationary steel balls was within the range of 10 percent. Dimensions of wear scars perpendicular to sliding direction were not measured because of difficulty, but wear scars were nearly circular and in the results of measurement using a metallurgical microscope there was little difference in the values in both directions.

In order to clarify the effect of bulk temperature rise in steel balls on maximum contact surface temperature at scoring, two kinds of operating times were selected. One is 300 seconds operating time, the other is 5 seconds operating time in which the bulk temperature rise in steel balls can be neglected. Besides, as shown in the second report ⁽⁹⁾, bulk temperatures of steel balls reach steady state after about 120 seconds from start. Bulk temperatures of stationary and rotating steel balls were measured by the same method of the second report. According to the object of the study, in 300 seconds test it is preferable that a scoring occurs after the bulk temperatures of steel balls nearly reach steady state. On the other hand, in 5 seconds test where a scoring seems to be induced mainly by flash temperature, when a scoring occurs after at least 0.5 in dimensionless time from start, the relation between thermal strength of boundary lubricating film and maximum contact surface temperature can be discussed by authors' defined flash temperature, and for steel balls used in this experiment with 12.7 mm in diameter and 12 mm²/s in thermal diffusivity dimensionless time 0.5 corresponds to 1.68 seconds. In order to calculate maximum contact surface temperature at scoring it is necessary to know the coefficient of friction, wear scar diameter and bulk temperature of steel balls. But it is difficult to know the wear scar diameter at scoring. Therefore, in practice the wear scar diameter at scoring must be estimated from the relation between wear scar diameter and axial load before scoring. Then in this experiment so as to estimate the wear scar diameter at scoring as accurately as possible the initial axial load and the load step were suitably selected according to the sliding velocity.

Occurrence of a scoring is determined by a sudden increase of friction torque in operation. And the thermal strength of boundary lubricating film is examined for the lightest axial load to score, namely, initial seizure load (ISL).

Coefficient of friction was obtained by the deflection of the same thin cantilever plate as used in the third report ⁽¹⁾.

3. Experimental results and discussion

In the case of oil bath lubrication, bulk temperatures of steel balls can be obtained by the equation shown in the second

Table 1 Properties of lubricating oils

Items		#90 base gear oil	#200 turbine oil
Kinematic viscosity cSt	37.8°C	235.9	91.4
	98.9°C	18.69	9.72
Viscosity index		96.0	92.0
Specific weight 15/4°C		0.900 8	0.886 5
Flash point °C		274.0	240.0
Pour point °C		-15.0	-22.5
Sulphur %		0.54	0.1

report⁽⁹⁾ using the experimental values of dimensionless relative coefficient of heat transfer $\bar{h}$ at steel ball surface and the ratio α of heat quantity supplied to steel ball per unit time to heat quantity transferred to ball fixtures per unit time, but in the case of drop lubrication the values of $\bar{h}$ and α are unknown. Therefore, in this experiment bulk temperatures of steel balls were measured without regard to lubricating method. Maximum contact surface temperature T_{cmax} when uniformly distributed heat sources are applied repeatedly to steel balls is given by the following equations⁽⁹⁾.

$$T_{cmax} = T_{fmaxs} + T_{bs} + T_{0s} = T_{fmaxr} + T_{br} + T_{0r} \quad (1)$$

$$T_{fmaxs} = \frac{\alpha_s Q_s}{2\pi\kappa a} \bar{T}_{fmaxs} \quad (2)$$

$$T_{fmaxr} = \frac{\alpha_r Q_r}{2\pi\kappa a} \bar{T}_{fmaxr} \quad (3)$$

$$Q_s = \frac{\mu P v_s}{426.8 \sqrt{6 c \gamma}} \quad (4)$$

$$v_s = \frac{\pi a n}{30 \sqrt{3}} \quad (5)$$

where $c\gamma Q_s$ is the quantity of frictional heat generated per unit time at one of the three contact surfaces cal/s, μ is the coefficient of friction, P is the axial load kg, v_s is the sliding velocity mm/s, γ is the specific weight of steel ball g/mm³, c is the specific heat of steel ball cal/g°C, n is the rotating speed rpm, κ is the thermal diffusivity of steel ball mm²/s, a is the radius of steel ball mm, and T_{bs} and T_{0s} are the bulk temperature rise in steel ball, deg. and the initial temperature of steel ball, °C, respectively. Subscripts s and r designate the values of stationary and rotating steel balls, respectively. α_s is the factor to determine the heat quantity supplied to stationary steel ball, and it is given by the following equation.

$$\alpha_s = \frac{1}{\bar{T}_{fmaxs} + \bar{T}_{fmaxr}} \left[\bar{T}_{fmaxr} + \frac{2\pi\kappa a}{Q_s} \{ (T_{br} + T_{0r}) - (T_{bs} + T_{0s}) \} \right] \quad (6)$$

α_r is obtained by the following equation.

$$\alpha_r + \alpha_s = 1 \quad (7)$$

$\bar{T}_{fmax}$ is a dimensionless maximum flash temperature which is given by the following equation.

$$\bar{T}_{fmaxs} = 106.7 \theta_0^{-0.9888} (0.9^\circ \leq \theta_0 \leq 14^\circ) \quad (8)$$

$$\bar{T}_{fmaxr} = K_{maxr} \times \begin{cases} 108.0 \theta_0^{-0.9174} (0.9^\circ \leq \theta_0 \leq 2^\circ) \\ 104.9 \theta_0^{-0.8888} (2^\circ < \theta_0 \leq 14^\circ) \end{cases} \quad (9)$$

θ_0 is obtained by

$$\theta_0 = \frac{180}{\pi} \sin^{-1} \frac{d_0}{2a} \text{ deg} \quad (10)$$

where d_0 is the wear scar diameter mm. K_{maxr} is the coefficient to consider the effect

of rotating speed on maximum flash temperature, and for wear scar diameters in this experiment it is obtained by Fig.1. Table 2 shows the physical properties of steel ball used in calculation.

3.2 Effects of lubricating method, operating time and sliding velocity on *ISL*

Fig.2 shows the effects of lubricating method, operating time and sliding velocity on *ISL*, (a) being the case of #90 base gear oil and (b) the case of #200 turbine oil. The purpose of this paper is to examine the thermal strength of boundary lubricating film; therefore, the results are only shown for the cases where a scoring occurred after lasting boundary lubrication state. Drop lubrication was employed in expectation of boundary lubrication state being easily obtained, but the result showed that there was not so much difference in lubrication state of contact surface as in the case of oil bath lubrication. In the case of drop lubrication of 300 seconds test with #90 base gear oil, $v_s \geq 92$ cm/s an experiment could not be made because the oil pot was filled with lubricating oil, because oil exhaust did not go well on account of higher oil viscosity and the lubricating oil was bubbled by the rotating axis.

From Fig.2 it is recognized that in both lubricating oils, in a lower sliding velocity range *ISL* is considerably scattered, depending on the lubricating method and operating time. And in the range of $v_s \leq 46$ cm/s the effect of sliding velocity on *ISL* is large, but in the range of $v_s \geq 69$ cm/s it becomes little as in the results of 60 seconds test in the third report⁽¹⁾. Com-

Table 2 Physical properties of steel ball SUJ2

Radius of ball	a	6.35 mm
Thermal diffusivity	κ	12.0 mm ² /s
Thermal conductivity	λ	1.03×10^{-2} cal/mms°C
Specific heat	c	0.11 cal/g°C
Specific weight	γ	7.8×10^{-3} g/mm ³

(In the case of radius of steel ball $a=6.35$ mm and thermal diffusivity of steel ball $\kappa=12$ mm²/s)

Fig.1 Diagram of $K_{maxr}-d_0$

paring with *ISL* obtained by adding 5 kg to *LNSL* in Fig.4 in the third report when bulk oil temperature was 20°C which is nearly equal to the room temperature in this experiment, although there is a slight difference in the properties of #90 base gear oils, *ISL* of both lubricating oils obtained by step-load method is larger.

3.3 Maximum contact surface temperature, coefficient of friction and wear scar diameter

As noted in section 3.2 there was not so much difference in lubrication state of contact surface caused by lubricating method; therefore, in this section the results of oil bath lubrication are mainly shown.

Figs.3 and 4 show the results of #90 base gear oil, and Figs.5 and 6 show the results of #200 turbine oil. Figs.3 and 5 show the relation between maximum contact surface temperature T_{cmax} at the end of each load step and coefficient of friction μ to

(a) #90 base gear oil

(b) #200 turbine oil

Fig.2 Effects of lubricating method, operating time and sliding velocity on initial seizure load

axial load, (a) being the case of 300 seconds test and (b) the case of 5 seconds test. Figs.4 and 6 show the relation between wear scar diameter d_0 and axial load before scoring, (a) being the case of 300 seconds test and (b) the case of 5 seconds test. T_{cmax} at scoring in Figs.3 and 5 was obtained by estimating d_0 at scoring from Figs.4 and 6. In the case of 300 seconds test, the time required to scoring from start of operation at *ISL* was, including the results of drop lubrication, from 35 to 280 seconds in the case of #90 base gear oil, and from 28 to 210 seconds in the case of #200 turbine oil. And further in the range of $v_s \geq 46$ cm/s, in almost all cases a scoring occurred in more than 120 seconds from start of operation and after bulk temperatures of steel balls nearly reached steady state. In the case

(a) 300 seconds test

(b) 5 seconds test

Fig.3 Relation among maximum contact surface temperature, coefficient of friction and axial load (#90 base gear oil, oil bath lubrication)

of 5 seconds test, all scorings occurred in more than 2 seconds from start of operation.

First the results of #90 base gear oil are examined. As clear from Fig.3, the coefficient of friction in 300 seconds test is liable to be smaller than that in the case of 5 seconds test. Wear scar diameter in 5 seconds test is nearly equal to Hertzian elastic contact circle diameter, but in 300 seconds test it deviates greatly from the Hertzian elastic contact circle diameter with an increase in sliding velocity and axial load, and in the range of $v_s \geq 46$ cm/s wear scar diameter at axial load just before scoring is beyond 0.8 mm that is more than twice as large as Hertzian elastic contact circle diameter. There is not so much difference in $T_{c,max}$ caused by operating time, and $T_{c,max}$ increases nearly straight for logarithmic scale of axial load.

Next the results of #200 turbine oil are examined. From Fig.5 it is recognized that the relations of maximum contact surface temperature and coefficient of friction to axial load have the same tendency as in the case of #90 base gear oil. Wear scar diameter in 5 seconds test is not so different from that of #90 base gear oil, but in 300 seconds test in the range of $v_s \leq 35$ cm/s wear scar diameter of #90 base gear oil is about 30 percent larger than that of #200 turbine oil at same axial load. At $v_s = 13, 23$ cm/s wear scar diameter of #90 gear oil at axial load just before scoring is smaller than that of #200 turbine oil, but in the range of $v_s \geq 46$ cm/s wear scar diam-

(a) 300 seconds test

(b) 5 seconds test

Fig.4 Relation between wear scar diameter and axial load (#90 base gear oil, oil bath lubrication)

eter of #200 turbine oil is considerably smaller than that of #90 base gear oil. This means that in a higher sliding velocity range contact surface was more easily worn away in the case of #90 base gear oil than in the case of #200 turbine oil.

3.4 Relation between scoring resistance of lubricating oil and maximum contact surface temperature

Fig.7 shows the effects of lubricating method, operating time and sliding velocity on maximum contact surface temperature at scoring, (a) being the case of #90 base gear oil, and (b) the case of #200 turbine oil. Coefficient of friction and wear scar diameter used in calculation are also shown in the figures. As stated before, wear

(a) 300 seconds test

(b) 5 seconds test

Fig.5 Relation among maximum contact surface temperature, coefficient of friction and axial load (#200 turbine oil, oil bath lubrication)

scar diameter at scoring was estimated from the relation between wear scar diameter and axial load before scoring.

3.4.1 In the case of #90 base gear oil In the range of $v_s \leq 35$ cm/s maximum contact surface temperature (T_{cmax}) at scoring has a tendency to increase with an increase in sliding velocity, but in the range of $v_s \geq 35$ cm/s the effect of sliding velocity on (T_{cmax}) is small and (T_{cmax}) is from 163 to 187 °C. In the range of $v_s \leq 46$ cm/s coefficient of friction is nearly constant without regard to lubricating method, operating time and sliding velocity, but in the range of $v_s \geq 69$ cm/s coefficient of friction in 300 seconds test is a little larger than that in the range of $v_s \leq 46$ cm/s. Wear scar diameter is greatly affected by operating time and sliding velocity, and in the range of $v_s \geq 46$ cm/s in 300 seconds test wear scar diameter is more than twice larger than that in 5 seconds test.

Here the effects of lubricating method and operating time on (T_{cmax}) are examined for each sliding velocity. In view of the effect of lubricating method, (T_{cmax}) of drop lubrication at $v_s = 13$ cm/s in 5 seconds test and that of drop lubrication at $v_s = 23, 46$ cm/s were from 17 to 25 deg. higher than those in oil bath lubrication, but except those cases the difference in (T_{cmax}) was within 10 deg. On the other hand, the effect of operating time on (T_{cmax}) was within the range of 10 to 15 deg. except for 30 deg. at $v_s = 13$ cm/s in drop lubrication. And as a whole, there is no general tendency recognized in the effects of lubricating method and operating time on (T_{cmax}).

3.4.2 In the case of #200 turbine oil In the range of $v_s \leq 35$ cm/s (T_{cmax}) increases with an increase in sliding velocity, but in the range of $v_s \geq 46$ cm/s the effect of sliding velocity on (T_{cmax}) is small

(a) 300 seconds test

(b) 5 seconds test

Fig.6 Relation between wear scar diameter and axial load (#200 turbine oil, oil bath lubrication)

and (T_{cmax}) is from 180 to 205 °C. In the range of $v_s \leq 46$ cm/s, coefficient of friction in drop lubrication is scattered, but in the range of $v_s \geq 69$ cm/s coefficient of friction has almost no connection with lubricating method, operating time and sliding velocity and is a little larger than that in the range of $v_s \leq 46$ cm/s. Unlike the results of #90 base gear oil there is not so much difference in wear scar diameter caused by op-

(a) #90 base gear oil

(b) #200 turbine oil

Fig.7 Effects of lubricating method, operating time and sliding velocity on maximum contact surface temperature at scoring

rating time and sliding velocity. The effect of lubricating method on $(T_{cmax})_s$ is maximum at $v_s = 35$ cm/s, and the difference in $(T_{cmax})_s$ by lubricating method is 27 deg. In another sliding velocity range, the difference is within 10 deg., especially in the range of $v_s \geq 46$ cm/s it is within the range of 10 to 15 deg. The values showing the effect of operating time on $(T_{cmax})_s$ are within 10 deg. in all cases. Like #90 base gear oil there is no general tendency in the effects of lubricating method and operating time on $(T_{cmax})_s$.

As mentioned above, in both lubricating oils without regard to lubricating method and operating time, $(T_{cmax})_s$ increased with an increase in sliding velocity and beyond a certain sliding velocity it was nearly constant. This suggests that in a higher sliding velocity range the boundary lubricating film is broken down at some constant temperature.

3.5 Bulk temperature of steel ball

In section 3.5 it was made clear that $(T_{cmax})_s$ is scarcely affected by operating time. In this section, bulk temperatures of steel balls and the effect of bulk temperature rise in steel balls on $(T_{cmax})_s$ are ex-

amined in the case of 300 seconds test. Like in section 3.3, bulk temperatures of steel balls in oil bath lubrication at the end of each load step only are shown.

Fig.8 shows the relation between bulk temperatures of steel balls and axial load untill scoring, (a) being the case of #90 base gear oil, and (b) the case of #200 turbine oil. In both lubricating oils, in the range of $v_s \leq 35$ cm/s it is noted that bulk temperatures at *ISL* are lower than the values at axial load just before scoring. This is because the time required to scoring was shorter, say, about 40 seconds and bulk temperatures did not reach steady state. In the case of #90 base gear oil, bulk temperatures at scoring in the range of $v_s \geq 46$ cm/s are 52 to 62°C in the case of stationary steel ball and 93 to 107°C in the case of rotating steel ball, and in these cases bulk temperature rise in steel balls from initial states is 28 to 45 deg. and 79 to 90 deg., respectively. In the case of #200 turbine oil, bulk temperatures at scoring are 42 to 54°C in the case of stationary steel ball and 82 to 93°C in the case of rotating steel ball, and bulk temperature rise in steel balls from initial states is 24 to 37 deg. and 63 to 75 deg., respectively. Thus, in long operating time test, in a higher sliding velocity range bulk temperatures at scoring are fairly high, and by comparing Fig.8 (a) with Fig.3 (a) and Fig.8 (b) with Fig. 5 (a) it is recognized that bulk temperature rise in steel balls can not be neglected. And bulk temperature rise in rotating steel ball from initial temperature was more than twice that in stationary steel ball.

Then, like Eq.(1) T_{cmax} is given by the sum of maximum flash temperature T_{fmax} and bulk temperature of steel ball $(T_b + T_0)$. As for the dimensionless value to indicate the effect of bulk temperature rise in steel balls on T_{cmax} , the ratio of T_b to T_{fmax} is to be considered. However, as mentioned above there is great difference between T_{b_s} and T_{b_r} in Eq.(1), and the ratio of stationary steel ball differs from that of rotating steel ball. On the other hand, T_{cmax} of both steel balls are equal and scoring resistance of lubricating oil is examined by T_{cmax} . Therefore, in this paper the effect of bulk temperature rise in steel balls on T_{cmax} was simply examined by the following method. If bulk temperatures of both stationary and rotating steel balls in operation are assumed to be equal, maximum flash temperature T_{fmax}' which is common for both stationary and rotating steel balls is given by the following equation.

$$T_{fmax}' = \frac{\bar{T}_{fmaxs} \bar{T}_{fmaxr} Q_f}{2\pi k a (\bar{T}_{fmaxs} + \bar{T}_{fmaxr})} \dots\dots\dots(11)$$

Therefore, if T_0' is initial temperature of steel ball °C,

$$T_b' = T_{cmax} - (T_{fmax}' + T_0') \dots\dots\dots(12)$$

can be regarded as the average value showing the effect of bulk temperature rise in steel balls on T_{cmax} . The ratio of T_b' to T_{fmax}' is defined as ζ .

$$\zeta = \frac{T_b'}{T_{fmax}'} \dots\dots\dots(13)$$

(a) #90 base gear oil

(b) #200 turbine oil

Fig.8 Relation between bulk temperatures of steel balls and axial load (Oil bath lubrication, 300 seconds test)

Fig.9 Dimensionless value ζ showing the effect of bulk temperature rise in steel balls on T_{cmax}

In the second report⁽⁹⁾, $T_{fmax'}$ at initial time in operation was used, but in this paper $T_{fmax'}$ at any desired time is used because even before scoring there is moderate fluctuation of friction torque. Fig.9 shows the value of ζ at scoring. ζ is scattered, but it is recognized that the effect of T_b' on $T_{fmax'}$ is fairly large. For example in the range of $v_s \geq 46$ cm/s ζ is from 0.60 to 0.95 in the case of #90 base gear oil and from 0.36 to 0.66 in the case of #200 turbine oil. The reason why the value of ζ of #90 base gear oil is larger than that of #200 turbine oil is that as shown in Fig.7 wear scar diameter of #90 base gear oil was larger than that of #200 turbine oil. As clarified in the first report⁽⁸⁾, flash temperature is greatly affected by wear scar diameter, while bulk temperature rise in steel ball is only affected by supplied heat quantity without regard to wear scar diameter.

In the above, bulk temperatures of steel balls in 300 seconds test were examined. In such a long operating time test an oil temperature rise in operation can not be avoided. But for the reason that lubricating oil is heated by steel balls and that there is scarcely any difference between oil temperature measured at the position in Fig.1 in the third report surrounded by four steel balls and bulk temperature of stationary steel ball which is lower than that of rotating steel ball, there is no trouble in long operating time test to examine the relation between thermal strength of boundary lubricating film and maximum contact surface temperature.

4. Conclusions

In this paper, thermal strength of boundary lubricating film of non-EP mineral oils was studied by step-load test using a Shell type Four Ball Machine in the range of sliding velocities v_s from 13 to 115 cm/s. The results obtained are summarized as follows:

(1) There is only a negligible difference in maximum contact surface temperature (T_{cmax}) at scoring between the results in 5 seconds test where a scoring seems to be induced mainly by flash temperature and the results in 300 seconds test where bulk temperature rise in steel balls occurs in operation.

(2) Without regard to lubricating method and operating time, (T_{cmax}) increases with an increase in sliding velocity and beyond a certain sliding velocity it is nearly constant. In the case of #90 base gear oil, (T_{cmax}) was about 175°C in the range of $v_s \geq 35$ cm/s, and in the case of #200 turbine oil it was about 190°C in the range of $v_s \geq 46$ cm/s.

(3) In order to examine the thermal strength of boundary lubricating film a long operating time test such as 300 seconds test is a more preferable method than a short operating time test because in the long operating time test boundary lubrication state is easily obtained and a scoring occurs after bulk temperatures of steel balls nearly reach steady state. And in 300 seconds test, bulk temperature rise in steel balls accounts for large percentage of (T_{cmax}). The ratio ζ of bulk temperature rise in steel ball to maximum flash temperature in the range of $v_s \geq 35, 46$ cm/s where (T_{cmax}) was nearly constant was from 0.60 to 0.95 in the case of #90 base gear oil, and from 0.36 to 0.66 in the case of #200 turbine oil.

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