Energy Saving Type Lubricants and Surface Durabilities of Steel Rollers*

By Komei FUJITA** and Akira YOSHIDA***

In order to elucidate the effect of an energy saving type lubricant on surface durability, steel rollers are tested under four kinds of EP gear oils with additives of V.I. improver and molybdenum disulfide using a sliding rolling contact fatigue tester. The effect of V.I. improver and the usefulness of molybdenum disulfide under ordinary Hertzian stress range could be basically understood. The fatigue life depended on coefficient of friction and surface temperature of the roller. The fatigue life and surface failure could be explained by an amplitude of ratio of stress to hardness considering the coefficient of friction, hardness distribution change due to temperature change of the roller surface and work-hardening.

Key Words: Gear, Steel Roller, Energy Saving Type Lubricant, Rolling Contact Fatigue, Surface Durability, Pitting, Spalling

1. Introduction

Amid movement for saving energy, energy saving type lubricants have come to be introduced for the practical use. Some service results about the energy saving effect of these lubricants employed in power transmission gear pairs have been reported. For getting the energy saving effect, molybdenum disulfide, viscosity index improver and so on are added to oils. By using these energy saving type lubricants power losses of gear pairs and so on are reduced generally. However the effects of the energy saving type lubricants on surface durabilities of gears and so on have not been clarified sufficiently.

In this report, thermally refined and through-hardened steel rollers were tested under four kinds of lubricants using a sliding-rolling contact fatigue testing machine as a fundamental study to elucidate the effects of the energy saving type lubricants on the surface durability. The results about the surface durability and surface failure were qualitatively discussed referring to the measurement results of surface temperature of roller and frictional coefficient between rollers. The experimental results about the surface durability and surface failure were quantitatively discussed in terms of the amplitudes of ratios of shear stress to hardnesses considering the surface temperature of roller and the frictional coefficient between rollers.

2. Test rollers

The roller materials employed in these experiments were 0.45 percent carbon steel S45C and high carbon chromium bearing steel

SUJ2, the chemical compositions of which are shown in Table 1. Fig.1 shows the shapes and dimensions of the roller specimens. S45C rollers were thermally refined and SUJ2 rollers were through-hardened respectively according to the conditions of heat treatments given in Table 2 after being turned to predetermined dimensions of rollers. These rollers were finish-ground to the dimensions shown in Fig.1 after the heat treatments. The surface roughnesses measured in axial direction of the rollers were 3 to 3.5 μm R_{max} on the thermally refined S45C rollers (TR-S45C) and 2 to 2.5 μm R_{max} on the through-hardened SUJ2 rollers (TH-SUJ2).

Table 1 Chemical compositions (wt.%) of materials

Material	C	Si	Mn	P	S	Cr	Cu	Ni	Mo	Al
S45C	0.43	0.27	0.74	0.017	0.019	0.13	0.02	0.02	—	—
SUJ2	1.01	0.29	0.39	0.012	0.011	1.35	0.06	0.06	0.02	0.008

Faster roller

Slower roller

Fig.1 Shapes and dimensions of test rollers

Table 2 Heat treatments

Specimen mark	Material	Heat treatment	Surface hardness Hv
TR-S45C	S45C	1173Kx1h WQ 833Kx1h WC	340
TH-SUJ2	SUJ2	1113Kx40min OQ(343K) 443Kx1.5h AC	850

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** Professor, School of Engineering, Okayama University, Tsushima, Okayama City, 700 Japan.

*** Associate Professor, School of Engg., Okayama University, Tsushima, Okayama City, 700 Japan.

(a)

(b)

Fig.2 Hardness distributions of test rollers at elevated temperature

Figures 2 (a) and (b) show the hardness distributions at elevated temperatures of TR-S45C and TH-SUJ2 rollers respectively. The surface hardnesses at room temperature of TR-S45C and TH-SUJ2 rollers were Hv340 and Hv850 respectively. The hardnesses of TR-S45C and TH-SUJ2 rollers decrease as the temperature increases, and this tendency is remarkable in roller TH-SUJ2. Fig.3 shows the residual stress distributions of rollers. The measuring and modifying methods of the residual stress are the same as described previously⁽¹⁾. In the surface layers of both rollers TR-S45C and TH-SUJ2, the residual stresses $(\sigma_x)_r$ in the axial direction of the roller and $(\sigma_y)_r$ in the circumferential are compressive, and the residual stress $(\sigma_z)_r$ in the radial direction of the roller is slightly tensile. The compressive residual stresses $(\sigma_x)_r$ and $(\sigma_y)_r$ in the surface layer of roller TH-SUJ2 are higher than those of roller TR-S45C.

3. Experimental procedures

The rolling contact fatigue testing machine used is of the spring-loading type⁽¹⁾. Rolling contact fatigue tests were performed under rolling with sliding conditions under the combination of the same kind of rollers. The rollers with widths of 5 mm and 10 mm were employed as the slower and the faster rollers respectively. The rotational speed and circumferential velocity were 1800 rpm

(a)

(b)

Fig.3 Residual stress distributions of test rollers

and 5.65 m/s on the faster roller, and 1432 rpm and 4.50 m/s on the slower roller. The specific sliding was +20.5 % on the faster roller and -25.7 % on the slower roller.

Four kinds of EP gear oils were employed whose characteristics are shown in Table 3. Oils C and D were obtained by adding V.I. improver to oils A and B respectively. Oils B and D were obtained by adding 0.5 % molybdenum disulfide to oils A and C respectively. The viscosities of all these oils at a temperature of 373 K are almost the same. But at a temperature of 313 K the viscosities of oils C and D are about half of those of oils A and B, being affected by the V.I. improver. These four kinds of oils whose temperatures were adjusted to 313 ± 4 K were pressure-fed into the engaging side of roller pair at a rate of about 1500 mL/min.

The temperatures of rollers were measured with the thermocouples at positions (T_1', T_2') being staggered by 90° behind contact position on both rollers and at depths (T_1, T_2) of 1 to 1.5 mm beneath the center surfaces of both rollers. Temperatures T_1' and T_2' were measured directly and temperatures T_1 and T_2 were measured through a slip ring, using a thermorecorder. At measuring temperatures T_1' and T_2' , a temperature rise due to friction between the roller surface and the thermocouple was corrected. The frictional co-

Table 3 Characteristics of test oils

Oil	A	B	C	D
Specific gravity 288/277K	0.8948	0.8978	0.8804	0.8840
Flash point K	489	495	457	455
Pour point K	263.0	265.5	233.0	233.0
Viscosity				
313K	204.9×10^{-6}	207.8×10^{-6}	115.6×10^{-6}	119.0×10^{-6}
373K	18.15×10^{-6}	18.45×10^{-6}	18.54×10^{-6}	19.00×10^{-6}
Viscosity index	97	98	180	181
Total acid number mgKOH/g	0.94	0.98	0.90	0.90
Lubricant type	Single grade		Multi-grade	
MoS ₂ * wt%	—	0.5	—	0.5
Performance level	GL-4			

* Size = 0.3 - 0.5 μ m in diameter

efficient between roller pairs was obtained by measuring the roller driving torque with a strain meter and correcting the torque loss in bearings.

The machine was stopped automatically by a vibration transducer which had previously been adjusted to act at a pitted area of 3 % or more on roller TR-S45C. When the machine was automatically stopped by a spalling formed on roller TH-SUJ2, one or two spalls had been formed. In this experimental range, the D value after P.H. Dawson exceeded unity. Therefore it can be assumed that the asperities on the roller surfaces are in metallic contact. The depths of surface failures were measured using a dialgauge with a special stylus whose tip radius is 0.1 mm.

4. Experimental results and discussion

4.1 Temperature rise of roller and frictional coefficient

Figure 5 shows an example of measuring results of roller temperatures for oil A. Temperatures T_1' and T_2' of roller surfaces and temperatures T_1 and T_2 at the depths of 1 to 1.5 mm beneath roller surfaces increase as the Hertzian stress P_{max} increases. In this experimental range, temperatures T_2 and T_2' of the faster roller are higher by 5 to 10 K than temperatures T_1 and T_1' of the slower roller. At the lower Hertzian stress range the temperatures T_1' and T_2' are slightly higher than the temperatures T_1 and T_2 , but at higher Hertzian stress range the former temperatures tend to be slightly lower than the latter temperatures. However the temperature differences between T_1 and T_1' and between T_2 and T_2' are very small. Therefore, concerning the relation between temperature and Hertzian stress of Fig.5, one curve is drawn for T_1 and T_1' of the slower roller and another curve is drawn for T_2 and T_2' of the faster roller. In this report, the temperatures indicated by these curves as in Fig.5 were defined as the surface temperature of roller. The tendency shown in Fig.5 is similar for all other oils.

Figure 6 shows the surface temperature of the slower roller in each oil. In this experiment the surface failures occurred always on the slower rollers. The surface temperatures are hardly influenced by the kind of oils. But the surface temperature of roller in oil C without molybdenum

Fig.4 Temperature measurement of test rollers

Fig.5 Surface temperatures of test rollers

Fig.6 Surface temperatures of the slower rollers

disulfide whose viscosity is low at a temperature of 313 K is higher than the surface temperatures in other oils except in the higher Hertzian stress range.

Figure 7 shows the frictional coefficient between roller pairs in each oil. The frictional coefficients are also hardly

Fig. 7 Frictional coefficient

influenced by the oils. But the frictional coefficient in oil C is slightly higher than the frictional coefficients in other oils except at the lower Hertzian stress levels at which the frictional coefficient in oil D with molybdenum disulfide is nearly equal to the frictional coefficient in oil C.

These tendencies shown in Figs. 5, 6 and 7 were almost the same for both rollers TR-S45C and TH-SUJ2. Therefore only the results about roller TH-SUJ2 are given here. In this experimental range, an empirical equation $T = 0.16 \cdot f \cdot P \cdot v_s + 316$ was obtained between roller surface temperature T (K),

Fig. 8 p_{max} - N curves Fig. 9 p_{max} - N curves

frictional coefficient f , load P (N) and sliding velocity v_s (m/s). Thereby the coefficient of correlation was 0.99.

4.2 Rolling contact fatigue tests

Figures 8 and 9 illustrate the p_{max} - N curves showing the relation between the Hertzian stress p_{max} and the number N of load cycles to failure for rollers TR-S45C and TH-SUJ2 respectively. In this report, the fatigue life was defined as the number of load cycles when the percentage of pitted area reached 3 % in the case of roller TR-S45C and the number of load cycles when the machine was automatically stopped by the formation of one or two spalls for roller TH-SUJ2. The p_{max} - N curves in Fig. 9 were obtained by the least square method. In these figures, the fatigue lives are not markedly influenced by the oils except in roller TR-S45C in oil C, though the experimental points have some deviations.

(a) Oil A $p_{max}=671\text{MPa}$ $N=4.1 \times 10^7$ (b) Oil A $p_{max}=1700\text{MPa}$ $N=5.8 \times 10^6$ (c) Oil A $p_{max}=1144\text{MPa}$ $N=6.5 \times 10^5$ (d) Oil A $p_{max}=1816\text{MPa}$ $N=4.3 \times 10^6$

Fig. 10 Microscopic observation of failed surfaces

(a) Oil A $p_{max}=1144\text{MPa}$ $N=6.5 \times 10^5$ (b) Oil B $p_{max}=1650\text{MPa}$ $N=6.7 \times 10^6$

Fig. 11 Transverse sectional observation of failed rollers

In Fig.8 the fatigue lives in oil C are shorter than those in the other oils in a Hertzian stress range of 900 to 1400 MPa. This decrease in the fatigue lives may be due to the temperature rise of roller, considering the results in Fig.6 that the surface temperature of roller in oil C is higher than that in the other oils in that Hertzian stress range.

In the case of the oils without molybdenum disulfide, the fatigue lives of the single-type oil A are longer than those of the multi-type oil C. But in the case of the oils B and D with molybdenum disulfide their fatigue lives are almost the same. These facts correspond to the results about the surface temperature of roller and the frictional coefficient shown in Figs.6 and 7. On the other hand, the effect of adding molybdenum disulfide into oil is such that in the shorter life range the fatigue lives of both rollers TR-S45C and TH-SUJ2 lubricated with oils A and C without molybdenum disulfide are longer than those lubricated with oils B and D with molybdenum disulfide except in roller TR-S45C with multi-type oils C and D. This result is in opposition to the result after W.J.Bartz⁽²⁾. However, in the longer life range that is the lower Hertzian stress range under usual operating conditions of machine, the fatigue lives in oils B and D tend to become longer than those in oils A and C.

As above mentioned, The tendencies in

the effects of viscosity difference and of adding molybdenum disulfide into oils on the fatigue lives of the rollers could be fundamentally understood. However these tendencies may be influenced by the operating conditions such as rotating speed, specific sliding and so on. So, more detailed investigations may be required.

In these experiments, failure modes in the rollers TR-S45C and TH-SUJ2 were pitting and spalling respectively. A few examples of these surface failures are shown in Fig. 10. Figs.10 (a) and (c) show the pitting formed on rollers TR-S45C, and (b) and (d) show the spalling on rollers TH-SUJ2. In both rollers TR-S45C and TH-SUJ2, the surface failures occurred on the slower roller having a narrower width whose specific sliding is negative. The pit is fan shaped and symmetrical about the rolling direction as shown in (c). The spall is formed over the full width of the roller and the shape of spall does not depend on the rolling direction unlike the case of pit. Figs.11 (a) and (b) show micrographs in transverse sections of the rollers failed by pitting and spalling respectively. It can be observed that the pitting crack is initiated on the roller surface and propagates inwards at an angle to the surface in the rolling direction, while on the other hand the spalling crack is initiated in the subsurface of the roller and propagates almost parallel to the roller surface. These observation results

Table 4 Results of rolling contact fatigue tests

TR-S45C											
Oil A			Oil B			Oil C			Oil D		
P _{max} MPa	N*	Maximum failure depth μm	P _{max} MPa	N	Maximum failure depth μm	P _{max} MPa	N	Maximum failure depth μm	P _{max} MPa	N	Maximum failure depth μm
671	4.1x10 ⁷ +	410	800	5.0x10 ⁷	***	850	3.0x10 ⁷	***	900	1.7x10 ⁷	360
792	2.0x10 ⁷	400	850	5.5x10 ⁷ ++	420	900	6.5x10 ⁶	700	900	2.1x10 ⁶	450
850	7.4x10 ⁶	600	900	5.1x10 ⁶	470	950	4.8x10 ⁷ f	220	950	4.0x10 ⁶	490
882	1.2x10 ⁷	490	950	8.4x10 ⁵	370	950	9.2x10 ⁵	370	1000	5.0x10 ⁵	430
998	1.8x10 ⁶	430	1050	5.4x10 ⁵	580	1000	1.1x10 ⁷	490	1100	1.2x10 ⁶	610
1100	2.1x10 ⁶	650	1100	1.8x10 ⁶	530	1100	2.3x10 ⁵	610	1200	5.8x10 ⁵	580
1144	5.8x10 ⁵	660	1146	1.5x10 ⁶	500	1150	1.5x10 ⁵	540	1300	6.8x10 ⁵	380
1277	5.6x10 ⁵	430	1200	5.3x10 ⁵	350	1200	2.2x10 ⁵	410	* Number of cycles at a percentage of pitted area of 3 % *** Non-failure		
1350	4.3x10 ⁵	390	1283	7.4x10 ⁵	580	1350	1.1x10 ⁵	360			
1415	1.9x10 ⁵	500	1350	1.1x10 ⁵	370	1350	1.4x10 ⁵	230			

+ Percentage of pitted area (PPA) = 2.3% ++ PPA = 0.4% f PPA = 0.3%

Table 5 Results of rolling contact fatigue tests

TH-SUJ2											
Oil A			Oil B			Oil C			Oil D		
P _{max} MPa	N	Failure depth μm Min-Max	P _{max} MPa	N	Failure depth μm Min-Max	P _{max} MPa	N	Failure depth μm Min-Max	P _{max} MPa	N	Failure depth μm Min-Max
1550	4.2x10 ⁷	***	1600	2.3x10 ⁷	250-630	1500	3.4x10 ⁷	190-430	1550	2.7x10 ⁷	200-450
1636	4.5x10 ⁷	180-440	1650	6.7x10 ⁶	200-430	1550	1.4x10 ⁷	180-400	1600	2.9x10 ⁷	230-460
1700	5.8x10 ⁶	300-580	1700	3.0x10 ⁷	220-430	1600	1.5x10 ⁷	250-410	1700	1.2x10 ⁷	270-530
1728	1.5x10 ⁷	180-410	1750	8.4x10 ⁶	270-510	1650	1.0x10 ⁷	190-450	1800	8.4x10 ⁶	210-540
1816	4.3x10 ⁶	210-350	1800	2.4x10 ⁶	310-440	1650	4.4x10 ⁶	230-920	1900	4.0x10 ⁶	260-520
1912	3.2x10 ⁶	400-710	1900	2.3x10 ⁶	240-420	1700	5.9x10 ⁶	220-670	2000	3.9x10 ⁵	320-610
1996	1.9x10 ⁶	340-770	1950	2.6x10 ⁶	320-630	1800	3.2x10 ⁶	220-520	*** Non-failure		
2100	4.7x10 ⁵	380-800	2000	4.2x10 ⁵	270-720	1900	4.1x10 ⁶	275-450			
						2000	6.4x10 ⁵	295-830			

shown in Figs.10 and 11 were hardly influenced by the oils.

Tables 4 and 5 show the results of the rolling contact fatigue tests of rollers TR-S45C and TH-SUJ2 respectively. The maximum depth of a pit given in Table 4 is the maximum value from all maximum depths of many pits. The minimum and maximum depths of a spall given in Table 5 are the minimum and maximum values from the depths of a spall in almost all cases, and are the smaller value of two minimum depths and the larger value of two maximum depths in the case that two spalls are formed on the roller. From these tables, it can be said that the failure depths of rollers TR-S45C and TH-SUJ2 are hardly influenced by the oils. In the case of roller TR-S45C, there is no distinct tendency in the maximum depths of pits due to the Hertzian stress, and the maximum depths of pits are 300 to 600 μm except in a few examples. In the case of roller TH-SUJ2, it is recognized that the failure depths tend to increase slightly as the Hertzian stress increases, and the minimum depths of spalls are 180 to 400 μm and the maximum depths are 350 to 920 μm .

4.3 Discussion

The depths of the surface failures were hardly influenced by the oils. It can be considered that the depths of the surface failures are related to the shear stresses beneath contact surface of the roller directly or indirectly. Tables 6 and 7 show the maximum value $[A(\tau/Hv)]_{\text{max}}$ of the amplitude $A(\tau/Hv)$ of ratio of shear stress τ to Vickers hardness Hv , the depth where the amplitude of the ratio becomes maximum and the hardness at this depth. In this discussion, it is assumed as described previously⁽³⁾ that the material strength is proportional to the hardness and the effect of mean stress on fatigue is negligible. The shear stresses taken into consideration here are the reversing orthogonal shear stress τ_{yz} and the pulsating maximum shear stress τ_{45° . The amplitude $A(\sigma/\sqrt{3}Hv)$ concerning normal stress σ is one also calculated considering that the amplitude $A(\sigma/\sqrt{3}Hv)$ is macroscopically equivalent to the amplitude $A(\tau/Hv)$. In calculating these amplitudes, the initial residual stress distributions shown in Fig.3, the hardness distributions under each operating condition obtained from Figs. 6 and 2 and the frictional coefficient under each operating condition obtained from Fig.7 are taken into consideration. The results as shown in Tables 6 and 7 are hardly influenced by the oils, so the results are given only for oil A here.

In most cases of spalling, the formation of one spall stops the machine, so the failure depth can be simply determined. In the case of pitting, however, many pits having different sizes and depths are formed on the roller surface as shown in Fig.10 (a). Therefore, concerning the pitting depths, a statistical approach will be required. Fig.12 shows examples of the frequency f_i distribution and the probability density function $f(t)$ concerning the depths of pits. The pitting failure was analysed assuming the Weibull distribution which is widely used in the fields of failure and

Table 6 Maximum amplitudes A_{max} and their depths below surfaces

P_{max} MPa	f	T K	TR-S45C			Oil A		
			$A(\tau_{yz}/Hv)$			$A(\tau_{45^\circ}/Hv)$		
			A_{max}	z mm	Hv	A_{max}	z mm	Hv
600	0.027	320.3	0.048	0.100	314	0.020	0.150	308
700	0.033	321.9	0.056	0.100	314	0.026	0.200	306
800	0.037	323.9	0.065	0.125	310	0.032	0.200	306
900	0.040	326.2	0.074	0.125	310	0.038	0.225	306
1000	0.042	329.0	0.083	0.150	307	0.044	0.300	304
1100	0.043	332.6	0.091	0.150	307	0.049	0.250	305
1200	0.044	336.6	0.100	0.175	305	0.055	0.275	303
1300	0.045	341.9	0.109	0.175	304	0.060	0.300	302
1400	0.045	347.5	0.118	0.175	303	0.066	0.300	301
1500	0.046	353.4	0.126	0.200	303	0.072	0.350	300

$\nu = 0.3$ $E = 206$ GPa

Table 7 Maximum amplitudes A_{max} and their depths below surfaces

P_{max} MPa	f	T K	TH-SUJ2			Oil A		
			$A(\tau_{yz}/Hv)$			$A(\tau_{45^\circ}/Hv)$		
			A_{max}	z mm	Hv	A_{max}	z mm	Hv
1500	0.046	353.4	0.052	0.200	732	0.022	0.100	757
1600	0.046	359.6	0.056	0.225	725	0.023	0.100	752
1700	0.046	367.9	0.060	0.225	720	0.025	0.500	653
1800	0.046	372.3	0.064	0.250	716	0.028	0.500	650
1900	0.046	378.9	0.068	0.275	710	0.031	0.500	646
2000	0.046	385.6	0.072	0.300	702	0.034	0.500	641

$\nu = 0.3$ $E = 206$ GPa

Fig.12 Frequency distribution and probability density function for pitting depth

life, as the pitting is one of failures. Table 8 shows the results of Weibull analyses concerning the failure depth of pitting. In these analyses, location parameter γ is taken as zero, since the pits distribute from near a depth of zero as understood from the frequency distribution of pitting depths shown in Fig.12. Judging from the results of Table 8, it seems that shape parameter m , scale parameter t_0 , mean value μ and standard deviation σ are hardly influenced by the oils and the Hertzian stresses. The mean values of the pitting depths are 70 to 270 μm . The results of Weibull analyses concerning the pitting diameters show that the mean pitting diameters are 0.6 to 1.6 mm.

Concerning the pitting depths, it can

Table 8 Results of Weibull analysis on failure depth of pitting

Oil	Hertzian stress range MPa	Pmax MPa	Percentage of pitted area %	m	t_0	μ μm	σ μm
A	850-950	850	8.4	2.10	9.3×10^4	211	105
	1000-1100	998	5.3	0.93	1.3×10^2	179	189
	1200-1400	1415	2.7	1.11	2.1×10^2	115	103
B	850-950	900 950	2.0 4.9	1.00 1.72	1.4×10^2 6.8×10^3	139 157	151 95
	1000-1100	1000 1100	2.8 1.8	1.38 0.70	9.3×10^2 3.5×10^1	136 211	100 294
	1200-1400	1350	4.3	1.40	4.0×10^2	67	48
C	850-950	900 950	3.2 3.0	1.02 1.65	1.9×10^2 5.2×10^3	166 157	161 98
	1000-1100	1100	5.0	1.65	1.2×10^4	260	162
	1200-1400	1350 1350	1.2 1.9	1.90 2.10	3.9×10^3 9.3×10^3	73 70	40 35
D	850-950	900 950	1.1 1.2	0.90 1.18	8.7×10^1 4.9×10^2	162 219	177 149
	1000-1100	1000 1100	2.3 2.3	1.12 1.37	2.6×10^2 1.9×10^3	138 248	121 184
	1200-1400	1200 1300	2.3 1.0	1.48 0.60	4.0×10^3 2.1×10^1	273 243	189 486

$$\gamma = 0$$

be said comparing Table 8 with Table 6 that the mean pitting depths are close to the depths of the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ rather than those of the maximum amplitude $[A(\tau_{45^\circ}/Hv)]_{\max}$ on the whole. Thus the pitting may have some relation to the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$, though this relation may not be primary since the pitting cracks originate on the roller surface as observed in Figs.10 and 11. On the other hand, concerning the spalling depths, it can be said comparing Table 5 with Table 7 that the spalling depths are primarily related to the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$, since the minimum depths of spalls agree well with the depths of the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ and these depths increase as the Hertzian stress increases.

Figure 13 shows the relation between the maximum amplitude $A_{\max}$ and the number N of cycles to failure. In the case of the pitting on roller TR-S45C the notation $A_{\max}$ is used for the maximum amplitude $[A(\sigma_y/\sqrt{3}Hv)]_{\max}$ and in the case of spalling on roller TH-SUJ2 $A_{\max}$ means the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ as described previously⁽³⁾. In Fig.13(a) without consideration of the work-hardening of roller during the rolling contact fatigue, the plotted points for roller TH-SUJ2 fit within a band given by two fine solid lines with a small deviation, while the plotted points for roller TR-S45C have a comparatively large deviation and moreover the $A_{\max}$ - N curve of roller TR-S45C is higher than that of roller TH-SUJ2. So the hardness of the rollers after the rolling contact fatigue tests were examined. The results show that the hardnesses of rollers TH-SUJ2 after the tests are almost the same as those before the tests but the hardnesses of rollers TR-S45C after the tests are highly work-hardened. Therefore Fig.13 (b) illustrates the $A_{\max}$ - N curve for roller TR-S45C obtained considering this work-hardening. Namely,

Fig.13 $A_{\max}$ - N curves

in this figure, hardness Hv^* is given by the product of the hardness under each operating condition determined by using Figs.6 and 2 and the ratio of the hardness after test to the hardness before test. In Fig.13 (b) the $A_{\max}$ - N curves of roller TR-S45C and TH-SUJ2 come close to each other and the plotted points about roller TR-S45C have a smaller deviation than in Fig.13 (a).

It can be understood from the above that the rolling contact fatigue life under each lubricating condition can be determined with use of the $A_{\max}$ - N curves shown in Fig.13 considering the changes of roller surface temperature and frictional co-efficient for each oil and the work-hardening.

5. Conclusions

As a fundamental study to elucidate the effect on surface durability of the energy saving type lubricants, thermally refined 0.45 % carbon steel S45C and through-hardened high carbon chromium bearing steel SUJ2 rollers were fatigue-tested under rolling-sliding contact condition using four kinds of gear oils that are single-grade oil, multi-grade oil, these two types of oils being added with 0.5 % molybdenum disulfide. The results were discussed by comparing the measurement results of roller surface temperature and frictional coefficient between roller pairs and the calculating results of stress beneath roller surface. The main results are summarized as follows:

(1) The effect of the viscosity index improver in oil on the fatigue life turned out as the effect of viscosity itself. As the viscosity increased, the temperature

rise of roller surface and the frictional coefficient decreased and the fatigue life tended to be prolonged. Concerning the effect of the molybdenum disulfide dispersed in oil on the fatigue life, the fatigue life in the oil with molybdenum disulfide was shorter in the higher Hertzian stress range and tended to become longer in the lower Hertzian stress range than the fatigue life in the oil without molybdenum disulfide.

(2) The pitting due to surface cracking occurred on thermally refined S45C rollers, while the spalling due to subsurface cracking occurred on through-hardened SUJ2 rollers. These failure modes and failure depths were hardly influenced by the oils. The minimum depths of spalls and the mean pitting depths calculated by the Weibull analysis agreed almost with the depths where the amplitudes $A(\tau_{yz}/H_v)$ became maximum.

(3) The rolling contact fatigue lives under different lubricating conditions can be determined with use of the $A_{max} - N$ curves considering the changes of the hardness distribution due to temperature rise and of the frictional coefficient and besides the work-hardening due to rolling contact fatigue.

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