Surface Durability and Surface Failure of Rolling Contact Roller Rotated at Non-uniform Velocity*

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In the case of a common rolling contact fatigue testing machine, the roller is rotated at uniform velocity. In this research the roller was rotated at non-uniform velocity by applying an eccentric gear mechanism to the machine in order to bring about a contact condition of rollers close to that of gears. Specific sliding of the roller was so adjusted as to change continuously from 0 % to -125 % during one rotation. Under this condition the rolling contact fatigue tests were performed using thermally refined and induction-hardened steel rollers. The fatigue life of each roller was elucidated experimentally, and the surface roughness change of the roller and the surface failure were discussed in relation to the specific sliding.

Key Words : Gear, Steel Roller, Surface Durability, Surface Failure,
Rolling Contact Fatigue under Non-uniform Rotating Velocity

1. Introduction

About 80 percent of failures in power transmission gears are due to fatigue. The fatigue failures of gears are classified into two types, tooth surface fatigue failure such as pitting and spalling and bending fatigue at tooth fillet. Experiments on the tooth surface fatigue are performed by power circulating gear fatigue testing machines using test gears and by conventional roller testing machines using test rollers whose surfaces are tooth surface models. Roller tests are simple for experiment and analysis and economical, but they have a demerit that strictly speaking, the contact condition of test rollers in roller test is different from the contact condition of gear tooth surfaces.

In this report, in order to bring about the contact condition of test rollers which are under a constant specific sliding close to the contact condition of gear tooth surface which is under a continuously varying specific sliding, a twin eccentric gear pair which was set with zero-backlash by coil springs was applied to a conventional rolling contact fatigue roller testing machine and fundamental studies to elucidate surface failure and surface durability of contact roller rotated at non-uniform velocity were carried out. Thermally refined steel rollers whose dominant failure mode is pitting and induction-hardened steel rollers whose dominant failure mode is spalling were tested using this improved rolling contact fatigue testing machine, and the fundamental test results about fatigue life and surface failure were discussed in relation to the specific sliding.

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2. Test rollers

The test rollers were made of 0.45 % carbon steel S45C. Fig.1 shows the shapes and dimensions of the test rollers. Rolling contact fatigue tests were performed using pairs of roller 1 and roller 2. Roller 1 having a diameter of 144 mm and a width of 20 mm is set on a constant speed axis, and roller 2 having a diameter of 96 mm and a width of 5 mm is set on a variable speed axis. The chemical compositions of these roller materials are given in Table 1. In these experiments thermally refined roller pair TRE and induction-hardened roller pair IHE were employed. These rollers were heat-treated under the conditions given in Table 2 after being machined to predetermined dimensions and were finish ground to the shapes and

Table 1 Chemical compositions (wt.%) of roller materials

Material	Roller diameter	C	Si	Mn	P	S	Cu	Ni	Cr
S45C	φ96	0.44	0.28	0.64	0.025	0.014	0.16	0.08	0.17
	φ144	0.45	0.24	0.67	0.022	0.012	0.08	0.05	0.10

Fig.1 Shapes and dimensions of test rollers

$\cos^{-1}[m(z_1+z_2)\cos\alpha_0/(2a_w)]$. Contact ratio ϵ can be calculated using length of path of contact $g\alpha = \{(d_{a1}/2)^2 - (d_{b1}/2)^2\}^{1/2} + \{(d_{a2}/2)^2 - (d_{b2}/2)^2\}^{1/2} - a_w \sin\alpha_w$. Where a is distance between gear-shaft centerlines, e is eccentricity, m is module, α_0 is standard pressure angle, z is number of teeth, d_a is tip diameter and d_b is base circle diameter. In this pair of twin eccentric gears, $z_1=z_2$, $d_{a1}=d_{a2}$ and $d_{b1}=d_{b2}$. Concerning the eccentric gears used in this experiment, $a=155.25\text{mm}$, $e=15.525\text{mm}$, $m=4$, $z_1=z_2=39$, $\alpha_0=20\text{deg.}$, $d_{a1}=d_{a2}=160.238\text{mm}$, $d_{b1}=d_{b2}=146.592\text{mm}$ and addendum modification coefficient $x_1=x_2=-0.5015$. The changes of lag angle 2ψ , distance between geometric centers a_w , working pressure angle α_w and contact ratio ϵ with the changes of rotational angles θ_1 and θ_2 of twin eccentric gears are shown in Fig.5, which were calculated by using the above mentioned equations and values. In Fig.5 angular acceleration $\ddot{\theta}_2$ of the driven eccentric gear is given, and this is also angular acceleration of roller 2. The driving eccentric gear 1 is rotated at uniform angular velocity, so $\dot{\theta}_1=0$.

3.3 Test conditions

In this test rig, the rotational speed is 600 rpm and distance between roller shafts is 120 mm. In this experiment, radius r_1 of roller 1 is 72 mm and radius r_2 of roller 2 is 48 mm. In this case the change of velocity ratio v_2/v_1 of peripheral velocity v_2 of roller 2 to peripheral velocity v_1 of roller 1 and the change of specific sliding σ_{s2} of roller 2 which is taken as object of failure are shown by broken lines in Fig.6. Solid

Fig.5 Changes of contact ratio ϵ , working pressure angle α_w , lag angle 2ψ , distance a_w between geometric centers and angular acceleration $\ddot{\theta}_2$ of driven eccentric gear with change of rotational angle θ of eccentric gear

lines in Fig.6 show these changes when $r_1=r_2=60\text{mm}$. In this case the ratio v_2/v_1 is also ratio $\dot{\theta}_2/\dot{\theta}_1$ in a pair of twin eccentric gears. In this experiment the specific sliding of roller 2 changes continuously from -125 % to 0 % as shown by broken line.

Lubricant employed is EP gear oil of which characteristics are given in Table 3. This oil was pressure fed to the engaging side of roller pair at a rate of 1750 mL/min and the oil temperature was adjusted to $313 \pm 4\text{K}$. Fig.7 shows the changes of theoretical minimum thickness of oil film⁽³⁾ with the changes of the rotational angles for the Hertzian stresses of 700 MPa and 1600 MPa which are the lowest and the highest stresses in the range of this rolling contact fatigue test respectively. The oil film thickness was calculated under the assumption that the temperature of oil between roller pairs is equal to the supplied oil temperature. The oil film thickness decreases as the specific sliding increases, and the film is thickest at a specific sliding of 0 % and thinnest at a specific sliding of -125 %.

Fig.6 Changes of velocity ratio v_2/v_1 and specific sliding σ_{s2} of roller 2 with rotation of eccentric gear

Table 3 Characteristics of EP gear oil

Specific gravity	288/277K	0.8995
Flash point	K	497
Pour point	K	255.5
Viscosity	310.8K	199.2×10^{-6}
	371.9K	17.03×10^{-6}
Viscosity index		99
Total acid number		mgKOH/g
		2.15

Fig.7 Theoretical minimum oil film thickness

4. Experimental results and discussion

4.1 $p_{\max}$ -N curve

The relationship between the Hertzian stress $p_{\max}$ and the number N of cycles to failure obtained by the sliding-rolling contact fatigue tests is presented in Fig.8. In the case of an induction-hardened roller IHE the spalling occurred and in the case of a thermally refined roller TRE the pitting occurred. In both cases the number of cycles when the test rig was automatically stopped was taken as the fatigue life N . At this fatigue life when the test rig was automatically stopped, in the case of spalling one spall was formed on the roller and in the case of pitting the percentage of pitted area reached about 1 % to 3 %. Taking the Hertzian stress for 10^7 cycles life as the surface durability, the surface durabilities of rollers TRE and IHE are about 650 MPa and 1200 MPa respectively. In the case of a thermally refined roller, the surface durability for 10^7 cycles life obtained by a previous experiment⁽⁴⁾ in which the specific sliding of roller pair was constant, i.e., -25.7 % was about 950 MPa. In the previous experiment⁽⁴⁾, the relative radius of curvature was 15 mm but the material and the hardness of roller were almost the same as those in this experiment. In the case of an induction-hardened roller, the surface durability for 10^7 cycles life obtained by a previous experiment⁽⁵⁾ in which the specific sliding of roller pair was constant, i.e., -25.7 % was about 1500 MPa. In the previous experiment⁽⁵⁾, the roller material was chromium molybdenum steel SCM440 but the relative radius of curvature and the hardened depth were almost the same as those in this experiment. It can be said that the surface durability obtained by this experiment in which the specific sliding is continuously changed from 0 % to -125 % tends to be lower than the surface durability obtained by the experiment in which the specific sliding was constant, i.e., -25.7 %, though this can not be asserted considering the effect⁽⁵⁾ of relative radius of curvature on the surface durability and the difference in material.

4.2 Surface roughness

Figures 9 and 10 show examples of the surface observations at each position of

Fig.8 $p_{\max}$ - N curves

Fig.9 Microscopic observation of thermally refined roller surface

Fig.10 Microscopic observation of induction-hardened roller surface

specific sliding with the rollers TRE and IHE before and after the tests. In the case of roller TRE, comparing the roller surface after the test with that before the test, on the roller surface after the test a running-in due to wear and plastic deformation is observed at any position of specific sliding. The running-in is remarkable at the position showing a large absolute value of negative specific sliding. These tendencies are similar irrespective of the Hertzian stress in this experimental range. In the case of roller IHE, from the surface observation of the rollers before and after the test shown in Fig.10, the running-in recognized by microscopic observation is not so clear as the result with the roller TRE. Under the Hertzian stresses of 1600 MPa and 1400 MPa the adhesive marks were observed at the

Fig.11 Roughness curves in axial direction of thermally refined roller surface

Fig.12 Roughness curves in axial direction of induction-hardened roller surface

Fig.13 Specific sliding and surface roughness

positions where the absolute value of negative specific sliding was as large as shown in Fig.10, but under the Hertzian stresses of 1200 MPa and 1050 MPa these adhesive marks were not recognized at any position irrespective of the specific sliding.

Figures 11 and 12 show the roughness curves measured in the axial direction of the rollers shown in Figs.9 and 10, for the position of each specific sliding. In the case of roller TRE, the surface roughness becomes smaller at the positions where the absolute value of negative specific sliding is large and becomes maximum at the position of specific sliding of 0 %, according to the result of Fig.9. The surface roughness after the test is remarkably small irrespective of the specific sliding compared with the surface roughness before the test. In the case of roller IHE, the surface roughness after the test becomes considerably smaller than that before the test but is not so influenced by the change of specific sliding as in the case of roller TRE. However, judging from the roughness curves and Abbott's bearing curves measured separately, the roughness curves near a specific sliding of 0 % show the characteristics of ground surface before the test and the roughness curves at the positions where the absolute value of negative specific sliding is large show the characteristics of a lapped surface. The surface roughness of the adhesive marks observed at the position where the absolute value of negative specific sliding is large in Fig.10 is not very large as shown in Fig.12.

Figure 13 shows the relationship between the surface roughness R_{max} and the specific sliding for each test roller after the test. In both cases of rollers TRE and IHE the surface roughness under the highest Hertzian stress is lowest in almost all positions of specific sliding. In the case of roller TRE, on the whole, the surface roughness becomes minimum at a specific sliding of -125 %, the roughness increases parabolically as the absolute value of negative specific sliding decreases irrespective of the acceleration side ($\ddot{\theta}_2 \geq 0$) or the deceleration side ($\ddot{\theta}_2 \leq 0$) and the roughness becomes maximum at a specific sliding of 0 %. This tendency agrees with the result shown in Figs.6 and 7 that the oil film becomes thinner as the absolute value of negative specific sliding increases, and that tendency may come from the consideration that the asperities on the roller surface are apt to suffer from plastic deformation and wear as the absolute value of negative specific sliding increases. In the case of roller IHE, the surface roughness is not so influenced by the specific sliding as in the case of roller TRE, but on the whole the surface roughness tends to increase near a specific sliding of 0 % though the roughness data have some scatter. These tendencies may come from the consideration that the oil film in the case of roller IHE is considerably thinner than that in the case of roller TRE since the loading condition of roller IHE is severe, so the change of the surface characteristics such as the plastic deformation and wear of the asperities on roller IHE is not so influenced by the change of specific sliding excepting the position of specific sliding of 0 % as in the case of roller IHE. The surface roughnesses before the tests of

rollers IHE and TRE are almost the same, but the surface roughness after the test of roller IHE does not become so small as in the case of roller TRE in spite of the higher Hertzian stress in roller IHE. It can be considered that this is due to the higher strength in the surface layer of an induction-hardened roller IHE.

4.3 Surface failure

Figure 14 shows a photograph of a developed circumferential surface of the roller TRE on which the pitting occurred under a Hertzian stress of 700 MPa. No pit is observed near the position of specific sliding of 0 % and many pits are observed at the position where the absolute value of negative specific sliding is more than 25 %. Fig.15 shows the distribution of pits formed on roller TRE under each Hertzian stress condition against the specific sliding. The circumferential surface of a non-uniformly rotated roller 2 was sectioned by each specific sliding of 10 % (partially 5 %) and the percentage of the pitted area to the surface area of each section was given for each section of specific sliding. On the whole the pitting tends to occur at a section where the absolute value of negative specific sliding is large. This tendency may be due to the fact that the plastic deformation, wear and fatigue damage on the roller surface increase as the absolute value of negative specific sliding increases, considering the result of Fig.13. From Fig.15, there is recognized a slight tendency

Fig.14 Pitting failure

Fig.15 Distributions of pits

that the pitting is apt to occur on the acceleration side ($\theta_2 \geq 0$) in this experimental range.

Figures 16(a) and (b) show a spalled portion and a cracked portion of the rollers IHE on which the spalling occurred under a Hertzian stress of 1600 MPa. The rollers shown in Figs.16(a) and (b) are not the same. Under the same experimental condition, in the case of (a) a spalling occurs near the position of specific sliding of 0 % and in the case of (b) a severe spalling crack occurs at the position where the absolute value of negative specific sliding is large. Fig.17 shows the spalled regions concerning the specific sliding on the circumferential surface of each roller on which a spalling occurred. The spalling failure tends to occur at the position where the absolute value of negative specific sliding is large, but this tendency is not so remarkable as in the case of the pitting on roller TRE. Therefore it can be said that the spalling on roller IHE is not influenced very much by the specific sliding.

Figures 18(a) and (b) show a transverse and an axial section of a fallen-off piece of a spall formed in Fig.16(a). In Fig.18 a spalling crack which is parallel to the roller surface is observed also in the hardened layer. It can be understood, therefore, that the spalling cracks occurred in the hardened layer and at the boundary between the hardened layer and the steel matrix of an induction-hardened roller. It can be also understood from Fig.18(b) that the spalling crack originates at the roller end and propagates to the center of roller width. Table 4 shows the depths of surface failures which occurred in these experiments. The failure depths were measured by a dialgauge

Fig.16 Spalling failure

Fig.17 Distributions of spalls

IHE $P_{max}=1600\text{MPa}$ $N=9.8 \times 10^4$ $\sigma_s=-125\%$

Fig.18 Spalling crack

Table 4. Failure depths

Specimen	P_{max} MPa	N	Type of failure	Failure depth mm Min.-Max.	A(τ_{yz}/H_v)		
					Apeak	z mm	Hv
TRE	650	3.9×10^7	NF	—	0.062	0.18	268
	700	5.7×10^6	Pitting	0.80	0.067	0.20	266
	750	1.9×10^6		0.91	0.072	0.20	266
	800	7.9×10^5		0.62	0.077	0.23	265
	900	4.8×10^5		0.65	0.087	0.25	264
	1000	1.8×10^5		1.15	0.097	0.28	263
IHE	1050	1.3×10^7	NF	—	0.042 0.021	0.25 3.40	643 261
	1200	6.3×10^6	Spalling	0.31-1.02	0.048 0.027	0.30 3.40	642 261
	1400	1.5×10^6		0.16-0.92	0.055 0.036	0.35 3.40	642 261
	1600	9.8×10^4 (2.8×10^4)		2.99-3.71	0.063 0.047	0.40 3.40	644 261

NF = Not failed

with a special stylus whose tip radius is 0.1 mm. In the case of pitting the maximum depth of a pit is given and in the case of spalling the minimum and maximum depths of a spall are given. The peak amplitude $[A(\tau_{yz}/H_v)]_{peak}^{(6)}$ of the ratio of orthogonal shear stress τ_{yz} to Vickers hardness H_v , its depth z and hardness H_v at this depth are also given in Table 4. The ratio τ_{yz}/H_v was calculated considering the hardness distributions shown in Fig.2 and taking the friction coefficient μ as 0.045, Poisson's ratio ν as 0.3 and Young's modulus E as 206 GPa. Taking the peak amplitude in the hardened layer as A_{peak1} and the peak amplitude at the boundary between the hardened layer and the steel matrix as A_{peak2} , in the case of a Hertzian stress of 1600 MPa the spalling crack depth in the hardened layer and the thickness of the fallen-off piece of a spall agree well

with the depths where A_{peak1} and A_{peak2} occur, respectively. In this case A_{peak2} is larger than $0.6A_{peak1}^{(6)}$, so it can be said that the spalling occurred depending upon A_{peak2} . In the case of a Hertzian stress of 1200 MPa it can be said that the spalling occurred by the formation of a crack in the hardened layer being due to A_{peak1} , since A_{peak2} is smaller than $0.6A_{peak1}$. In the case of a Hertzian stress of 1400 MPa it is likely that the spalling was caused by A_{peak1} judging from the measured result of failure depth, though A_{peak2} is larger than $0.6A_{peak1}$. In this case A_{peak2} is equal to $0.65A_{peak1}$ and this condition is close to the previous critical condition. That expectation may be due to the material defects in the hardened layer.

Thus, the tendency of the spalling failure depth is almost the same as that of the results⁽⁵⁾⁽⁶⁾ of the constant specific sliding tests. Therefore it can be considered that the spalling failure depth is not influenced very much by the change of specific sliding. In the case of roller TRE, the formation of a pit and the number of pits were influenced by the specific sliding as mentioned above, but the sizes and depths of pits which were different even in the same region of specific sliding were not influenced very much by the specific sliding in this experimental range.

5. Conclusions

In order to elucidate fundamentally the surface durability and surface failure of a non-uniformly rotating contact roller, thermally refined and induction-hardened steel rollers were fatigue-tested using an improved rolling contact fatigue test rig to which an eccentric gear mechanism was applied. The results are summarized as follows:

(1) It can be said that the fatigue lives of the thermally refined and induction-hardened rollers obtained by this experiment in which the specific sliding varied from 0 % to -125 % during one rotation of roller were shorter than those obtained by the test under a constant specific sliding of -25.7 %.

(2) In the case of a thermally refined roller, the surface roughness after the test was small at the position where the absolute value of negative specific sliding was large and that was maximum at the position of specific sliding of 0 %. In the case of an induction-hardened roller the surface roughness tended to become maximum at the position of specific sliding of 0 %, but on the whole that was almost constant in spite of the change of specific sliding.

(3) The pitting occurred on the thermally refined rollers and the spalling occurred on the induction-hardened rollers. The pitting tended to occur at the position where the absolute value of negative specific sliding was large. The occurring position of the spalling was not influenced very much by the specific sliding.

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