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A Study on Strength and Failure of Induction-Hardened Chromium-Molybdenum Steel Spur Gears*

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In order to determine the strength of induction-hardened chromium-molybdenum steel spur gears, fatigue tests were performed using a power circulating gear testing machine.

As the result, the surface durability was obtained in connection with dynamic loads and it was shown by microscopic observation that there occurred fatigue tooth breakages starting from the middle tooth height which are induced by the spallings on tooth surface or sometimes bending fatigue fractures at the tooth roots depending on the dimension of tooth width.

Moreover, it was made clear that tooth surface strength of induction-hardened gears must be evaluated in terms of stresses beneath the surface.

1. Introduction

The tooth breakage of power transmission gears renders the operation of machine impossible directly and the tooth surface failure puts the operation of machine in danger or induce excessive noise. Therefore the gear teeth must be so designed as not to suffer tooth failures. The failure modes of gears are generally classified into pitting, spalling, scoring and tooth breakage, and the load carrying capacity has been investigated for each failure mode considering the dynamic load. The pitting and spalling are the fatigue failures on tooth surface, but the failure mode of pitting is different from that of spalling and the pitting occurs in different hardness range of tooth surface from the case of spalling; the pitting occurs on soft gears and the spalling does on surface hardened gears. The load carrying capacities of surface hardened gears have so far been often checked only by the bending strength of gear teeth at gear designing, since it has been understood that the load carrying capacities of these gears are generally limited by the tooth breakage. It is known that even on the surface hardened gears, the load carrying capacities are often determined by the one for spalling. However, only a few precise investigations concerning the problems in these fields have been reported⁽¹⁾⁽²⁾.

In this report, the induction-hardened chromium-molybdenum steel gears heat-treated under the actual service conditions were tested using a power circulating gear testing machine, and the strength of these gears was made clear and the mode of surface

failures on these gears was investigated. It was made clear that the load carrying capacities of induction-hardened steel gears are determined by the one for spalling on tooth surface.

2. Experimental procedure

2.1 Test gears

The dimensions of test gears employed in these experiments are shown in Table 1. The materials of gears and their chemical compositions are shown in Table 2. The gears were heat-treated under the conditions shown in Table 3 and were finally ground by a Niles gear grinding machine. The mechanical properties of gear materials thermal-refined under the conditions of Table 3 are shown in Table 4. In Table 4, the mechanical properties of the through hardened chromium-molybdenum SCM3 steel are the ones of a steel which was quenched at 900 °C for 1 hr and tempered at 175 °C for 2 hrs. The hardness distributions in the normal direction of tooth surface at the center of fillet are shown in Fig.1. The teeth were fully

Table 1 Dimensions of test gear

	Pinion	Gear
Module	4	
Number of teeth	18	27
Addendum modification coefficient	0.23	0.0858
Standard pressure angle	20°	
Tip circle diameter mm	82.313	117.159
Whole depth mm	9.236	
Face width mm	12	10
Center distance mm	91.5	
Contact ratio	1.51	

Table 2 Chemical compositions (wt.%) of materials

Material	C	Si	Mn	P	S	Cu	Ni	Cr	Mo
SCM3 Gear	0.38	0.24	0.76	0.018	0.020	0.20	0.13	1.11	0.17
SCM4 Pinion	0.43	0.27	0.76	0.021	0.009	0.35	0.09	0.94	0.25
SCM4 Gear	0.39	0.28	0.71	0.019	0.008	0.13	0.04	1.00	0.22
SNC22 Pinion	0.16	0.25	0.47	0.013	0.007	0.12	3.08	0.78	

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hardened and the hardnesses in all over the region of gear tooth turned out the same hardness as the one on the tooth surface shown in Fig.1.

Table 3 Heat treatments of test gears

Table 4 Mechanical properties of gear materials

Material	Yield strength kg/mm ²	Tensile strength kg/mm ²	Elongation %	Reduction of area %	Brinell hardness HB
Thermal-refined					
SCM3 Gear	95.0	101.2	18.0	61.0	234
SCM4 Pinion	109.2	115.8	16.8	63.3	247
SCM4 Gear	93.0	107.7	18.6	60.0	244
SNC22 Pinion		108.3	14.3	64.3	247
Through-hardened					
SCM3 Gear	194.2	208.1	13.8	42.2	405

Fig.1 Hardness distributions in normal direction at tooth fillet

Fig.2 Schema of gear fatigue testing machine

2.2 Test

The gear testing machine used in these experiments is of power circulating type and is operated by driving a power transmission gear set through a Kopp variable-speed drive with a motor. The center distance between pinion and gear is 91.5 mm. An outline of this machine is shown in Fig.2. The rotating speed of test gear can be selected arbitrarily in the range of 600 to 5400 rpm when belt reduction ratio is one. Lubricating oil was fed to the test gears and transmission gears through separate pressure lubricating systems. The lubricating oil fed to the engaging side of test gears was RIX #200 turbine oil and a rate of its supply was 1.3 l/min. The oil temperature was adjusted to 40 ± 5 °C. The load on test gears was applied by fixing a flange coupling after giving a torque to this system by a weight lever. The gear fatigue tests were performed at 2000 rpm at test gear side.

The tooth bending stress of the test gear with a narrower face width than the test pinion was measured by four wire strain gages which were bonded on the tooth-fillets of four teeth adjoining each other. This measurement was performed when these four teeth with wire strain gages meshed with the fixed mating pinion teeth.

The gear fatigue testing machine was operated under certain torque and constant rotating speed until a tooth failure, and after the predetermined cycles the tooth profile and the dynamic tooth bending stress were measured. The operation of test rig was stopped when a vibration due to tooth breakage was detected. The fractures were observed by optical and scanning electron microscopes in order to elucidate the mode of tooth failures.

3. Experimental results and discussion

3.1 Fatigue life

The results of gear tooth fatigue tests are shown in Fig.3. Fig.3 shows the relationship between Hertzian stress P_{max} at operating pitch point and the number of cycles to tooth breakage N_2 . The actual tooth bending stress $\sigma_t^{(1)}$ by Aida and Terauchi is also shown.

Fig.3 S-N curve

On the induction-hardened test gears made of chromium-molybdenum steels SCM3 and SCM4, the pitting with the percentage of pitted area reaching about 1 % occurred but the fatigue lives of these gears were determined by the tooth breakage in all cases. At Hertzian stress 141.5 kg/mm^2 the induction-hardened SCM4 test gear was not failed until 9.93×10^7 cycles.

3.2 Observation of tooth failure

3.2.1 Appearance of tooth failure

In these experiments all failed test gears suffered tooth breakage. Examples of broken tooth pieces are shown in Fig.4. The originating point of fatigue crack was in the vicinity of the highest point of single tooth contact in any case and the crack developed to a tooth fillet of compression side at about 45° to tooth surface, and on the tooth fillet of tension side there was no starting point of these fatigue failures. These appearances of tooth breakage are quite different from those of usual tooth breakage by fatigue.

Fig.4 Examples of tooth breakage

Fig.6 Cracks at side surface of tooth

Concerning the SCM3 test gear which suffered tooth break by fatigue at 5.28×10^6 cycles under Hertzian stress of 170 kg/mm^2 , photos of the surface of unbroken tooth are shown in Fig.5 at 2.4×10^6 and 5.28×10^6 cycles. The surface appearances of unbroken teeth of the gears which suffered tooth breakage as shown in Fig.4 were the same as those shown in Fig.5 and from Fig.5 it can be said that there was uniform loading through tooth width. However, the originating point of the crack which caused tooth breakage was at right or left end of meshing tooth surface in any case.

(a) $N_2 = 2.4 \times 10^6$

(b) $N_2 = 5.28 \times 10^6$ (At tooth breakage)

Fig.5 Observation of tooth surface

(a) Broken tooth (b) Unbroken tooth

Fig.7 Cracks at tooth fillets

Though more pits were observed on the dedendum flank as shown in Fig.5 that originating point was in the vicinity of the highest position of single tooth contact.

Therefore the teeth of all the failed gears were observed on their side surface especially in the vicinity of the highest single tooth contact point. About all the failed gears the teeth with such cracks as shown in Fig.6 were revealed in these observations. In the case of these induction-hardened chromium-molybdenum steel spur gears it was made clear that the originating point

Fig.8 Microfractographic observations by SEM

Fig.9 Microfractographic observations by SEM

of fatigue tooth break was in the vicinity of the highest point of single tooth contact.

It was investigated whether the cracks were observed or not at the tooth fillet, since the tooth bending stress at fillet was considerably high as shown in Fig.3 though the tooth breakage did not originate from the tooth fillet. Fig.7 shows the examples of the cracks at tooth fillets. About the test gears operated at Hertzian stress or more than 150 kg/mm², such cracks as shown in Fig.7 occurred in many cases. The fatigue limit of gears is influenced by the surface roughness of tooth fillet. The pulsating fatigue limits of S40C⁽²⁾ (3) and SCM4⁽⁴⁾ induction-hardened gears are more than 69 kg/mm² in terms of actual tooth bending stress at fillet. But it can be said that the fatigue limit of these test gears was 34 kg/mm² as shown in Fig.3 for 10⁷ cycles life. These differences between the fatigue limits may be due to the rough surface finishing at fillet, differences in heat treatment and in operating condition.

It can be said from the above mentioned observations that the S-N curves shown in Fig.3 are nearly equivalent to the S-N curves for the tooth bending fatigue, since there were the cracks at the tooth fillets of gears tested under more than 150 kg/mm² of Hertzian stress though the tooth breakage did not originate at the tooth fillet.

3.2.2 Fractographic observation of tooth breakage

Figure 8 shows the scanning electron microphotographs⁽⁵⁾ (6) taken in the vicinity

of crack initiation point in the broken piece shown in Fig.4(a). Fig.8(e) is a schema of Fig.8(a) showing a magnified photo of failed tooth surface, and Fig.8(b), (c) and (d) are scanning electron microscopic photos of the positions ①, ② and ③ in Fig.8(e) respectively. At position ① where the depth beneath surface is about 0.1 mm, "a tongue" which is formed during the crack propagation of spall is observed. At position ② a considerably brittle fracture surface including irregular "striation" which is formed due to the fatigue crack propagation is observed. At position ③ "a dimple" which shows the fracture surface formed by the rapid propagation of cracks is observed.

Figure 9 shows scanning electron microphotographs of another broken tooth. Fig.9(a) is a synthesized photo taken in the vicinity of initiating point of spall and Fig.9(b) is one taken from the bottom of spall shown in Fig.9(a). This portion of Fig.9(b) indicates the initial fracture surface where "a tongue" is seen though the fracture surfaces rub with each other. In Fig.9(c) "a river" pattern which is a comparatively brittle fracture surface is observed and this portion shows the fracture surface formed by the bending fatigue after the initiation of spall. Fig.9(e) shows the tooth surface in the vicinity of the crack initiation point of broken tooth. On the tooth surface, many pin holes and microcracks running along the gear axis are observed, and their appearances are different from those of the surface failure of so-called pitting.

3.2.3 Variations of tooth profile and dynamic tooth bending stress

In these experiments the tooth surfaces were considerably worn before the tooth breakage occurred as shown in Fig.5. It is considered that these worn tooth surfaces might give rise to an increase of dynamic load and accelerate the tooth breakage. Fig.10 shows the variations of tooth profiles at the constant stresses of tooth surfaces which are nearly equal to the fatigue limit. At 8×10^6 cycles of test gear there are indications of wear on the tooth flank of test gear and at 3×10^7 cycles of test gear, i.e. 4.5×10^7 cycles of test pinion the tooth profiles with severe wear are observed for both test gear and pinion.

Fig.10 Number of loading cycles of test pinion and variations of tooth profiles

Figure 11 shows the tooth bending stresses under the conditions of tooth profiles of test gears shown in Fig.10. These tooth bending stresses were measured at a rotating speed of test gear 2000 rpm, that is, at a circumferential operating pitch circle velocity of 11.5 m/s. In Fig.11, as a basis for comparison, the bending stresses measured at 13.3 rpm of test gear are shown by dotted lines.

Fig.11 Number of loading cycles of test pinion and tooth bending stress of test gear

Fig.12 Rotating speed of test pinion and increasing ratio of tooth bending stress of test gear

It is clear from Figs.10 and 11 that in these experiments, which were performed under the contact stresses which were a little higher than the fatigue limit for surface durability, the tooth bending stresses of surface hardened gears increase by 10 to 20 percent up to 8×10^6 cycles of gear and increase by about 65 percent at about 3×10^7 cycles. The natural frequency of gear pair of this test rig is about 3080 Hz and Fig.12 shows the ratio of dynamic tooth bending stress to static one at the rotating speeds which are far lower than the resonance frequency. At the cycles less than 1.1×10^7 of test gear the dynamic load is nearly equal to the static load, but at 3×10^7 cycles the dynamic load becomes considerably higher than the static one. From these considerations, it can be said that the values of the actual tooth bending stresses in Fig.3 showing the results of tooth surface fatigue may be used as a measure of tooth bending fatigue strength.

3.3 Discussion of tooth break mechanism

3.3.1 Mechanism of failure initiation

It is clear from Figs.4, 6, 8 and 9 that the spalling cracks initiate from the vicinity of both ends of tooth and of the highest position of single tooth contact. The crack initiation at both ends of tooth is due to the fact that the value of distributed load at the ends of tooth of test gear having narrower tooth width than the test pinion is higher than the one at the center of tooth width⁽⁷⁾⁽⁸⁾. Crack initiation position was sometimes left end and sometimes right end of tooth, and so it can be said that that result might be due to the lead error but not to the poor parallelness of shafts of test rig. The spalling crack which has occurred beneath tooth surface propagates in the direction of tooth height and parallel to the tooth surface, and simultaneously the secondary microcracks⁽⁵⁾ are formed in the inner direction of tooth as shown in Fig.13. Therefore, when a spalling crack occurs, there may be heavy stress concentration at the tips of these secondary spalling cracks by the tooth bending action due to the meshing of tooth and the failure mode changes from spalling to tooth breakage by bending. This change of failure mode is shown in Fig.8(b),

(c), Fig.9(b), (c) and Fig.13(b). In Fig.6 the crack propagated by bending appears at the side surface of tooth. After the gradual propagation by fatigue, the tooth may not be able to endure the bending load and a tooth break occurs with the rapid propagation of crack.

3.3.2 Discussion of test results

In these experiments performed using the induction-hardened chromium-molybdenum steel gears, the mode of gear tooth failure was spalling fundamentally. The authors have investigated the spalling of surface hardened rollers and made it clear that the rolling fatigue strength and life of surface hardened rollers are related to the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ of the ratio of reversing orthogonal shear stress τ_{yz} to Vickers hardness Hv and that the depths of spalls agree with the depths where the amplitudes $A(\tau_{yz}/Hv)$ become maximum⁽⁵⁾⁽⁶⁾. Thereupon, the circumferential and radial directions at contact point are taken as y and z axes respectively. Fig.14 shows the relationships between the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ and the number of cycles to failure N together with the data from induction-hardened and case-hardened rollers. The $[A(\tau_{yz}/Hv)]_{\max} - N$ curves obtained in these experiments lie in the previous empirical ranges.

Assuming that Hertzian contact stress is $p_{\max}$, half breadth of elastic Hertzian contact is b , radius of relative curvature is R , modulus of longitudinal elasticity is E , Poisson's ratio is ν , maximum amplitude of the ratio of orthogonal shear stress to Vickers hardness Hv is $[A(\tau_{yz}/Hv)]_{\max}$ and depth beneath surface of this $[A(\tau_{yz}/Hv)]_{\max}$ is z , the following equations can be introduced since the amplitude of orthogonal shear stress is not influenced by the frictional force on surface.⁽⁹⁾

$$\left. \begin{aligned} b &= \frac{4(1-\nu^2)}{E} R p_{\max} \\ [A(\tau_{yz})]_{\max} &= 0.25 p_{\max} \\ z &= 0.5b \end{aligned} \right\} \dots\dots\dots (1)$$

In the case of the hardness distributions of test gears shown in Fig.1, since the hardness is uniform from the fillet surface to the depth of about 0.5 mm and the initiation depths of spalls are less than 0.2 mm, the following equation is reasonably introduced.

$$\begin{aligned} [A(\tau_{yz}/Hv)]_{\max} \\ = 0.25 p_{\max} / (Hv)_{z=0} \dots (2) \end{aligned}$$

The values of maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ in Fig.14 were calculated by Eq.(2), considering the increase of dynamic load shown in Fig.12. The results for the spalling of induction-hardened chromium-molybdenum steel gears agree with the results for case-hardened nickel-chromium SNC21 steel rollers in Fig.14.

The initiation depths of spalls shown in Fig.13 are in the range of 0.13 to 0.15 mm while the depth of the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ is 0.13 mm at Hertzian stress of 180 kg/mm². Thus, the calculation result agrees well with the experimental result.

Considering the above mentioned facts, the spalling of the induction-hardened steel gears can be investigated by the method as applied for the surface hardened rollers. The calculation results about the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ in these experiments can be used in the design of induction-hardened chromium-molybdenum steel gears for surface durability.

Fig.14 $[A(\tau_{yz}/Hv)]_{\max} - N$ curves

Fig.13 Spalling of induction-hardened gears
(Subsurface cracks are observed to be parallel to tooth surfaces)

In these experiments the failure mode of gear was the tooth breakage induced by a spall formed in the vicinity of the highest point of single tooth contact. In the case of line contact gear, since the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ or the maximum contact stress $p_{\max}$ is inversely proportional to the square root of tooth width and the bending stress at tooth fillet is inversely proportional to the tooth width, the bending stress condition which causes the tooth break at root fillet can be induced by decreasing the tooth width of test gears. Therefore, it can be said that the failure mode of induction-hardened chromium-molybdenum steel gear used in these experiments is significantly influenced by the selection of tooth width.

4. Conclusions

In order to clarify the strength of induction-hardened chromium-molybdenum steel gears, the gear fatigue running tests were performed using a power circulating gear testing machine. The fracture surface of broken teeth were observed and analyzed, and the experimental results were analyzed and discussed. The obtained results are summarized as follows:

(1) The failure mode of the induction-hardened steel gears employed in these experiments was a tooth breakage originating in the range of the single tooth contact.

(2) It was made clear by the micro-fractographic observations of failed teeth that the spalls initiate fundamentally from the vicinity of the highest contact stress point in the single tooth contact range and then a tooth break is secondarily induced by

these spalls. It can be said that the gear failure is fundamentally a spalling beneath the tooth surface and the mode of gear tooth failure depends upon the values of the contact stress on tooth surface and the bending stress at tooth fillet.

(3) The spalling life and the spall depth are related with the value of the maximum amplitude $[A(\tau_{yz}/Hv)]_{\max}$ and the depth where the amplitude $A(\tau_{yz}/Hv)$ becomes maximum respectively. These relations for the induction-hardened gears were similar to those for the surface hardened rollers.

It was confirmed from the above mentioned results that the amplitude of reversing orthogonal shear stress beneath surface can be reasonably used to investigate the surface durability of surface hardened gears and rollers.

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