621.833.3

Paper No. 191-14

Studies on Limiting Load for Scoring of Crossed Helical Gears*

(Effects of Kind of Material, Heat treatment and Surface Finishing)

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The limiting loads for scoring of the crossed helical gears expressed with Hertzian pressure were compared with those of the FZG spur gear tests and of the roller tests. The limiting loads for scoring of the crossed helical gears are very low and are changed not so widely according to the kind of oil. As a result it can be seen that the crossed helical gear is under quite unfavourable condition for the formation of a lubricating oil film. In order to increase the limiting load for scoring, various kinds of surface heat treatments and material combinations were tested. The load was stepped up from the first stage to the next higher stage successively according to the load stage.

1. Introduction

Crossed helical gears have been used for a long time, but their usage is limited to the transmitting of a light load, because of a great deal of wear and low load carrying capacity, caused by a high sliding velocity between tooth surfaces and their contact situation (point contact). Because their usage is in practice limited, the load carrying characteristics of the crossed helical gear are not enough researched. Under these present situations it is very important to carry out experiments about the above mentioned load carrying characteristics and to clarify the experimental results. From this point of view we have already published our experimental results about these problems (1) (2). The experiments of the crossed helical gears were carried out here from other standpoints than the previous published one.

The limiting loads for scoring of the crossed helical gears expressed with Hertzian pressure were compared with those of the FZG spur gear tests and of the roller tests. As a result it can be seen that the crossed helical gear is under quite unfavourable condition for the formation of a lubricating oil film. This is the reason why the limiting loads for scoring of crossed helical gears are so low. In order to increase the limiting

load for scoring, suitable surface heat treatments and material combinations must be chosen for the tooth of the crossed helical gear. From this point of view, many kinds of surface heat treatments and material combinations were tried and their limiting loads for scoring of each case were tested. In the previous published report about the limiting loads for scoring of the crossed helical gear (1) the test gears were loaded while the test equipment was not at the state of running but at standstill. In the case of the experiments in this report, the test gears were loaded, while the test gears were in a running state.

- ① Test gears and test gear box
- ② Torquemeter (for output)
- ③ Gears and gear box for power transmission
- ④ Coupling
- ⑤ Gear and gear box for power transmission and loading
- ⑥ Torsion bar
- ⑦ Revolutionary speed changer
- ⑧ Electric motor
- ⑨ Gears and gear box for power transmission
- ⑩ Torquemeter (for input)

Fig.1 Test equipment

* Received 9th July 1979.

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2. Fundamental equation for meshing of crossed helical gear

The fundamental equations for meshing of the crossed helical gears were already introduced(3). The final relationship between the Hertzian pressure (P_H) and a normal force to the gear tooth (P_N) is expressed as follows:

$$P_N = C (P_H^3/E^2)$$

where C is a constant, which slightly changes with the position of meshing of tooth and is determined by geometrical relationship of the crossed helical gears.

3. Test equipment

Fig.1 shows our test equipment of the crossed helical gear. ① is the gear box for our test gear (crossed helical gear) and ③ ⑤ ⑥ are the gear boxes, in which the crossed helical gears are installed for the sake of power transmitting. ⑩ is a torque-meter to measure the input torque. ⑧ is an electric motor (revolutional speed $n = 1500$ rpm). The revolutional speed is changed by means of ⑦. ⑥ is a torsion bar which is twisted according to the value of the load. The twisting action is given through a sliding of one crossed helical gear ⑤ in the gear box in Fig.2. The details of this equipment are shown in Fig.2. The sliding of one gear (crossed helical gear) is done by using the handle ③, which is combined with a flange ④. The power circulating direction is shown in Fig.1.

4. Test condition and method

4.1 Test range

The experiments were carried out in the following range: The maximum input torque is $M_1 = 19.5$ kg m and the corresponding maximum normal force (load) is $P_N = 728$ kg. Seven revolutional speeds were chosen for our tests, i.e. $n = 376, 643, 1060, 1500, 2120, 3500, 4950$ rpm.

The sliding velocity V_g (m/s) of our test gears at the pitch point is expressed as follows:

$$V_g = 5.97 \times 10^{-3} \times n$$

where n is the revolutional speed rpm of driver and follower (the revolutional speeds of driver and follower are same). At the end of path of contact (i.e. at the tip of the test gear), the sliding velocity V_g is about 2 % higher than that at the pitch point.

4.2 Test gears

The drive gear and the follow gear have same dimensions (Table 1). For the final mechanical finishing, two kinds of finishing were done, i.e.

- (1) Hobbing (maximum height of surface roughness R_{max} 4~6 μ m)
- (2) Grinding by means of a Reishauer gear grinding machine (R_{max} 0.7~1.0 μ m)

Fig.3 shows a meshing of the crossed helical gear. V_1 and V_2 are same, i.e.

- ① Load increasing direction
- ② Load decreasing direction
- ③ Handle for loading
- ④ Flange for bearing
- ⑤ Axial slidable gear on shaft
- ⑥ Fixed gear on shaft

Fig.2 Equipment for loading

Table 1 Dimensions of test gears

Diameter of pitch circle d_0	80.61 mm
Diameter of tip circle d_k	86.61 mm
Back lash	0.2 mm
Normal pressure angle α_n	20°
Direction of helix	Right
Helical angle on pitch cylinder β_0	45°
Normal module m_n	3 mm
Number of teeth z	19
Addendum modification coefficient x	0
Face width b	60 mm
Shaft angle δ	90°
Center distance a	80.61 mm*
Tooth surface finishing & gear quality	(a) Ground with Reishauer gear grinding machine, Gear quality grade: JIS 1
	(b) Hobbed, Gear quality grade: JIS 2~4

*Driver and follower have same dimensions. So this center distance is calculated.

Fig.3 Contact of tooth surfaces on pitch point

Table 2 Combinations of test gear

Group		No.	Combination	
			Follower	Driver
A	Case hardened gear group	1	Case hardened gear (ground)	Case hardened gear (ground)
		2	Case hardened gear (hobbed)	Case hardened gear (hobbed)
B	Tempered gear group	3	Tempered gear (ground)	Tempered gear (ground)
		4	Tempered gear (hobbed)	Tempered gear (hobbed)
		5	Normalised gear (hobbed)	Normalised gear (hobbed)
C	Nitrided gear group	6	Case hardened & nitrided gear (ground)	Case hardened & nitrided gear (ground)
		7	Case hardened & nitrided gear (hobbed)	Case hardened & nitrided gear (hobbed)
		8	Tempered & nitrided gear (ground)	Tempered & nitrided gear (ground)
		9	Tempered & nitrided gear (hobbed)	Tempered & nitrided gear (hobbed)
D	Cast iron gear group	10	Cast iron gear (hobbed)	Case hardened gear (ground)
		11	Cast iron gear (hobbed)	Tempered gear (hobbed)
		12	Cast iron gear (hobbed)	Case hardened & nitrided gear (ground)
		13	Cast iron gear (hobbed)	Tempered & nitrided gear (hobbed)

$$|V_1| = |V_2|$$

but the directions of V_1 and V_2 are opposite to each other. The crossed helical gears, which were used for our test, are in practice quite widely used.

Table 2 shows the combinations of material and heat treatment of our test gears. For combination of material and heat treatment, four groups (A,B,C,D) were chosen, i.e. case hardening-, tempering-, nitriding- and cast iron-groups. From each group, two to four kinds of combinations were chosen for the tests. Details are shown in Table 3. In this table the final mechanical finishing is shown in parentheses.

4.3 Lubricating oils

Table 4 lists lubricating oils, which are used for our tests. All base oils of lubricating oils, which were used for our tests, were paraffin type. These lubricating oils (T,G,H) are mainly used as turbine oil, gear oil and hypoid oil respectively. Main additives and viscosity are shown in Table 4. The concentration of the additive ($Zn-P$ anti-wear agent) in gear oil G is higher than that in turbine oil T.

4.4 Test method

The load was stepped up from the first stage to the next higher stage successively according to the load stage shown in Table 5. By using this load stage the limiting loads for scoring were determined. At the maximum load (load at the highest load stage) the torque is $M = 19.5 \text{ kg m}$ (18th load stage). In order to measure the wear of the test

gears, a balance was used, which had $\pm 0.5 \text{ mg}$ accuracy. Before and after the running of each load stage the test gears were weighed on this balance. The total number of revolutions of test gear in each load stage is always constant, namely 22500, and so the running time is variable with the revolutionary speed of test gears. For example, if the revolutionary speed of the gear wheel $n = 1500 \text{ rpm}$, the total running time is 15 minutes in each load stage. The wear at lower stages is comparatively high. Except these lower load stages the specific wear was determined at load stages of the normal wear. This specific wear V_s (mg/HPH) is defined as the wear W (mg) divided by the transmitted work A (HPH) at the load range, where this wear W was measured. In this load range the load stage, where the scoring takes place, is not included.

The limiting load for scoring can be registered by noise, vibration and sudden changes of frictional losses* of test gears. The lubricating oil, which was contained in an oil vessel, was sprayed from a nozzle into the meshing point of gears. The oil temperature at spraying point was $90 \pm 3^\circ \text{C}$ and the quantity of the sprayed oil was about 2.4 l/min.

5. Test results and considerations

In our experiments the combination of material and heat treatment was chosen as widely as possible for the sake of practical usage. The heat treatment for case hardening is a very commonly used one for the machine parts. So the limiting loads for scoring of combination of case hardening group (group A in Table 2) are here compared with the test results of other combinations of materials and heat treatment. Furthermore the tendencies of limiting loads for scoring of crossed helical gears with various combinations of material and heat treatment are described in the following section.

5.1 In the case of "case hardening (ground)" (No.1 in Table 2)

Fig.4 shows the limiting loads for scoring by using this kind of combination

* Frictional losses are the value of difference between input-torque and output-torque measured by torquemeters ⑩ and ⑪ (Fig.1) respectively.

Table 3 Material, heat treatment and finishing of test gears

Description	Material	Heat treatment and finishing method	Vicker's Hardness and effective depth of hardened thickness
Case hardened gear	SCM 21	Gears were already hobbled and after that carburised at 900°C in salt bath for 2 hours and further kept in salt bath at 800°C for 5 min. After that they were quenched in oil. After this quenching treatment they were kept in salt bath at 180°C and cooled in air.	Tooth surface hardness about H_V 730. Core hardness about H_V 420. Effective depth of hardened layer about 0.5 mm.
Tempered gear	S45C	The material was at first normalised.* After that it was kept in salt bath at 850°C for 30 minutes and then quenched in water. After these treatments gear blanks were hobbled and ground by means of the Reishauer gear grinding machine.	Hardness at tooth surface and their neighborhood about H_V 280.
Normalised gear	S45C	After normalising the gears were hobbled.	Hardness at tooth surface and their neighborhood about H_V 280
Case hardened & nitrided gear	SCM 21	Above mentioned "case hardened gear" was soft nitrided (tufftride treatment, using Ti-pot, kept at 570°C for 2 hours) and then cooled in water. In this case of combination of "cast iron gear group" (Table 2) these soft nitrided gears were made smooth by means of sandpaper AA 400 with metal cutting oil.	Hardness at tooth surface (just below the compound layer) H_V 620. Core hardness H_V 200. Effective depth of hardened layer 0.1 mm.
Tempered & nitrided gear	S45C	Above mentioned "tempered gear group" was soft nitrided (tufftride treatment, using Ti-pot, kept at 570°C for 2 hours). In this case of "cast iron group" (Tab. 2) the soft nitrided gears were made smooth by means of sandpaper AA 400 with metal cutting oil.	Hardness at tooth surface (just below the compound layer) H_V 470. Core hardness H_V 300. Effective depth of hardened layer 0.1 mm.
Cast iron gear	FC 20 ~ FC 25	The material was produced by continuous casting method without additional heat treatment. The structure of material is ferrite structure with eutectic graphite all over the cross section. The final mechanical cutting was hobbing.	H_V 180 ~ 250

*The normalising was done as follows : kept at 570°C for 30 minutes in salt bath and after that cooled in air (to room temperature).

Table 4 Lubricating oils

Symbol	Description	Kinematic viscosity, cSt 37.8°C 98.9°C		Specific gravity at 15°C kg/cm ³	Viscosity index	Flashing temp. °C	Additives
T	Turbine oil (No.2 turbine oil)	64.15	8.02	0.880	102	226	Double inhibitor Zn-P anti-wear agent (mild E.P.additive) Rust inhibitor
G	Gear oil (Light duty No.3 gear oil)	195.6	17.05	0.891	101	270	Double inhibitor Zn-P anti-wear agent (mild E.P. additive) Rust inhibitor
H	Hypoid oil (Automotive gear oil No. 90)	199.6	17.67	0.903	104	210	Oxidation inhibitor S-P E.P. additive Rust inhibitor Corrosion inhibitor

Table 5 Load stages

Load stage	Input-torque M kg m	Normal force to tooth surface P_N kg
1	0.24	8.96
2	0.75	28.0
3	1.00	37.3
4	1.33	49.7
5	1.80	67.2
6	2.4	89.6
7	3.2	120
8	4.2	157
9	5.6	209
10	6.5	243
11	7.5	280
12	8.7	325
13	10.0	373
14	11.5	429
15	13.3	497
16	15.4	575
17	17.8	665
18	19.5	728

and three kinds of oil. Generally speaking the limiting loads for scoring in this case are fairly low. Especially in the range of higher revolutional speeds than $n = 1000$ rpm the limiting loads for scoring were very low.

Although oils of high load carrying capacity (high viscosity and high concentration of E.P. additives) were used as lubricating oil, the limiting loads for scoring were not so high. The influences of kinds of oil upon the limiting loads for scoring are not remarkable. Now the test results of the crossed helical gears are compared with those of the spur gears in Fig.5. In this figure the limiting loads for scoring are expressed with the Hertzian pressure (P_H kg/mm²). The dimensions of the spur gear for our tests are: module $m = 4.5$, number of tooth $z_1 = 16$, $z_2 = 24$, addendum modification coefficient $x_1 = 0.8635$, $x_2 = -0.5$, cutter pressure angle $\alpha_c = 20^\circ$, face width $b = 10$ mm*, center distance $a = 91.5$ mm(4)(5).

The material and the heat treatment conditions of the test gears of the FZG spur gear test are just the same as those of the crossed helical gears.

Concerning the FZG spur gear test the Hertzian pressure is the highest at the point of worst loading (at the end of path of single contact), which is situated at the outer side of pinion. In the case of the crossed helical gear the value of the Hertzian pressure at the pitch point is almost the same that at the point of worst loading. The difference between the above mentioned two values (Hertzian Pressure at pitch point and worst loading point) is so small that we can neglect it. In order to simplify our calculation the Hertzian pressure at a pitch

* Original face width of FZG spur gear test is $b = 20$ mm. This face width was reduced considering the bending strength of tooth. In order to realise our purpose specially crowned (corrected) gear teeth in direction of teeth trace were used in our test. So this face width means an effective face width.

point is used for limiting load for scoring. The important difference between spur gear and crossed helical gear is in the contact condition of tooth surface in meshing, i.e., the former (spur gear) is line contact and the latter (crossed helical gear) is point contact. According to Niemann (5)(6) the value of Hertzian pressure in line contact (P_{HL}) corresponds to the value of Hertzian pressure in point contact (P_{HP}) divided by 1.5. So the Hertzian pressure (P_H) of the line contact (spur gear) and the point contact (crossed helical gear) are defined as follows:

In the case of the spur gear at the worst loading

$$P_H \equiv P_{HL}$$

In the case of the crossed helical gear at the pitch point

$$P_H \equiv P_{HP}/1.5$$

The Hertzian pressures in line contact and in point contact, which are calculated by the above mentioned method, are according to our experiences quite reasonable. Fig.5

Fig.4 Limiting load for scoring of case hardened gear (ground)

Fig.5 Limiting load for scoring of crossed helical gear and FZG spur gear

shows the experimental results of the limiting load for scoring expressed with Hertzian pressure, which were calculated by the above mentioned method. As revolutionary speed of both gears, in the case of the spur gear, $n_2 = 1500$ rpm, $n_2 = 3500$ rpm and in the case of the crossed helical gear $n = 1500$ rpm, $n = 3500$ rpm were chosen.

As Fig.5 shows, the limiting loads for scoring in the case of spur gear tests are quite dependent upon the kind of oil, while the limiting loads for scoring in the case of crossed helical gears show only a slight difference with the lubricating oil. Furthermore it may be concluded in the case of crossed helical gears that the limiting loads for scoring expressed with Hertzian pressure are very low. The reason why such a tendency comes out, would be as follows: The velocities of tooth of V_1 and V_2 are at the pitch point just in opposite direction to each other and these two velocities are equal in magnitude at the pitch point. This is very severe condition for lubrication. So theoretically it can be said that a lubricating oil film cannot be built up. Therefore it is very difficult to get a high level of the limiting load for scoring of the crossed helical gear, by using various kinds of oils. Furthermore the results of roller tests were compared with the test results of the crossed helical gear. The test pieces of our roller tests were as follows: The diameter of the smaller test roller $d_1 = 50$ mm and the diameter of the larger test roller $d_2 = 133$ mm. These two test rollers were pressed together under the condition of slide-rolling on a disk-machine. The center distance between them is 91.5 mm. The test rollers are carburized and hardened and the material of the test rollers is steel (SCM 21 according JIS). The conditions were the same as in the case of case hardened gear of the crossed helical gear (table 3). Their circumferential surface, which is used for our test, has been ground to $R_a = 0.2 \mu\text{m}$ (CLA). This roughness is almost the same as that of the crossed helical gear.

In this test the load is stepped up from the first stage to the next higher stage successively. In each load stage the total sliding distance between two test rollers (small and large rollers) is kept constant (i.e. 4953 m) and then the total amount of the running time in each load stage is dependent upon sliding velocity. For example in the case of $n_2 = 1500$ rpm in large roller, specific sliding is $\Sigma_1 = 1.11$ and sliding velocity is $V_g = 5.5$ m/s, the running time is 15 minutes. The results of these experiments are quoted from Ref.(7).

Fig.6 shows the limiting load for scoring expressed with Hertzian pressure as a function of sliding velocity. This sliding velocity of the crossed helical gear changes slightly according to the contact position of gear tooth. The sliding velocity of the abscissa in Fig.6 is that at the pitch point.

In the case of roller tests the lim-

Fig.6 Limiting load for scoring of crossed helical gear tests and roller tests

iting load for scoring changes according to the velocity ratio V_1/V_2 , in which V_1 and V_2 are the circumferential velocities of test rollers 1 and 2 respectively. In our roller test the relation of V_1 and V_2 is always $|V_1| \geq |V_2|$. It is shown that in every case of lubricating oils (oil T, G, H) the limiting load for scoring is dependent upon V_1/V_2 , that is, the limiting load increases with approaching to $V_1/V_2 = 1$ (i.e., approaching to pure rolling). On the contrary in the case of $V_1/V_2 = -1$ the limiting load for scoring

is minimum. The difference between the limiting load for scoring in the case of $V_1/V_2 = -1$ of the roller test and that in the case of the crossed helical gear is shown in Fig. 6. The reason, why the limiting load for scoring of the roller test expressed with the Hertzian pressure does not coincide with that of the crossed helical gear test, can be explained as follows: main reason for this phenomenon is:

- (1) Geometrical difference between the two test methods i.e., difference of contact situations between two kinds of test pieces, namely in the roller test the contact is continuous and in the crossed helical gear test it is intermittent.
- (2) Directions of gear grinding and sliding velocity on the contact position of tooth surfaces are different in each case.

All these factors considered, it is easy to understand that the limiting load for scoring (expressed with Hertzian pressure) of the FZG spur gear test is very high and quite dependent upon the kinds of oils unlike the case of the crossed helical gear. When the hypoid oil is used as a lubricating oil for the FZG spur gear test, the limiting load for scoring is higher than the load of the highest load stage. But in the case of the crossed helical gear, the limiting load for scoring by using this hypoid oil is also comparatively low. So it can be said that a remarkable increase of load carrying capacity in the scoring of the crossed helical gear is not made possible through only the choice of oil. Therefore in order to get a high limiting load for scoring of the crossed helical gear it is necessary to choose a

proper material combination and a proper surface heat treatment. From these points of view our experimental results using many kinds of material combination and surface heat treatment are summarized in the following section.

5.2 Test results with various kinds of material combinations and heat treatment

As mentioned before, case hardened group No.1 (Table 2) is used quite commonly for spur gear and helical gear. But this combination is not suitable for the crossed helical gear. Now in this section the test results using many kinds of material combination and surface heat treatment are compared with each other and discussed from a practical standpoint.

Fig.7 shows the limiting load for scoring of the crossed helical gear using many kinds of material combinations, heat treatments and surface treatments. The revolutional speed was in this case $n = 1500$ rpm. As this figure shows, the limiting load for scoring is quite dependent upon the material combination, heat treatment and surface treatment. The most noteworthy feature is, that among the limiting loads for scoring of various combinations, which are shown in Fig.7. the limiting load for scoring of case hardening (ground)(No.1 in Table 2) is the lowest, and the limiting loads for scoring of case hardening (hobbed)(No.2 in Table 2) and case hardening & nit.(ground)(No.6 in Table 2) and further cast iron (hobbed) / tempering (hobbed)(No.11 in Table 2) by

Fig.7 Limiting loads for scoring of crossed helical gear with various combinations, in the case of revolutional speed $n = 1500$ rpm

using hypoid oil are especially high. These high limiting loads for scoring are explained in detail in the following sections, because these kinds of combinations are for the practical usage very valuable.

5.2.1 In the case of "case hardening (hobbed)"(No.2 in Table 2)

The limiting loads for scoring of this combination are shown in Fig. 8. The limiting loads for scoring using the hypoid oil are extremely high. Although the gear quality grade of accuracy of this combination is lower than that of the combination of No.1 in Table 2, the limiting load for scoring of this combination (i.e.No.2) is higher than that of combination of No.1 using the hypoid oil. The limiting load for scoring in this combination (No.2) using the turbine oil is not so extremely high as that of combination No.1. Generally speaking the higher the viscosity of base oil and concentration of additives (anti-wear additive and E.P. additive), the greater the difference between the limiting loads for scoring of combinations No.1 and No.2. As a general rule, it can be said that the limiting load for scoring is dependent upon the surface roughness and it decreases in accordance with the surface roughness. But in the case of the crossed helical gears the tendency is quite different from this general rule, i.e., the limiting load for scoring does not decrease with the surface roughness and further the specific wear of this combination (No.2) is very low, although the surface roughness is comparatively high (details are shown in section 5.3).

5.2.2 In the case of "case hardening & nit.(ground)"(No.6 in Table 2)

The experimental results of this combination are shown in Fig.9. The limiting load for scoring is in this case extremely high in comparison with that in the case of the combination of No.2. The specific wear is also very low and shows a good tendency of the load carrying characteristics.

5.2.3 In the case of "tempered & nit.(hobbed)"(No.9 in Table 2)

Usually in the case of tempered and nitrided gears tooth surfaces are not ground. So the experiments with test pieces (crossed helical gears), which were only hobbed (not ground), were carried out with various revolutional speeds and oils. These experimental results are shown in Fig.10. As this figure shows, the limiting loads are comparatively high.

5.2.4 In the case of "cast iron (hobbed)/tempered (hobbed)"(No.11 in Tab.2)

Fig.8 Limiting load for scoring of case hardened gears (hobbed)

Fig.9 Limiting load for scoring of case hardened and nitrided gears (ground)

Fig.10 Limiting load for scoring of tempered and nitrided gears (hobbed)

Four kinds of combinations (i.e.No.10, 11,12,13 in Table 2), which belong to the description of cast iron group (Table 2) were tested. The highest limiting load for scoring among these four kinds of combinations is that the combination of No.11. Experimental results with this combination for various revolutional speeds and oils

are shown in Fig.11. Concerning the specific wear, the results are good (low specific wear). But the scattering is in this case comparatively wide, because the cast iron, which was used for these combinations, was not uniform in quality.

5.3 Specific wear

Generally speaking the scattering of the specific wear when using oil T and G in the case of the crossed helical gear varies widely. But if the crossed helical gears are lubricated with oil H, the value of specific wear does not vary very widely. So the specific wear, which are obtained by oil H, are plotted in Fig.12. This figure shows the specific wear in various kinds of combinations and Fig.11 Limiting load for scoring of combination cast iron (hobbed)/tempered gear (hobbed)

The specific wear of combination No.1 is very high and scattered in a wide range. In comparison with this result the specific wear of combination No.2 is the lowest in this figure.

6. Conclusions

The limiting loads for scoring of the crossed helical gears are shown in comparison with those of the FZG spur gear tests and roller tests. The limiting loads for scoring of the crossed helical gears are very low and do not change so widely according to the kind of lubricating oil.

The reason, why the limiting load of the crossed helical gear is so low, lies in the following factors:

- (1) The crossed helical gears are under quite unfavorable condition for the formation of a lubricating oil film.
- (2) The teeth of the crossed helical gears are meshed together at a point (point contact) with high sliding velocity.

But the limiting load for scoring is quite dependent upon the material combination and heat treatment. So it is possible in the case of this gear to obtain a high load carrying capacity by choosing suitable material combination and heat treatment.

The authors wish to thank president and directors of Paker-Netsushori Co.LTD/Tokyo, who were quite helpful for our work. The heat treatment of test pieces was done by goodwill of this company.

Fig.12 Specific wear

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