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Studies on Limiting Load for Scoring of Spur Gears*

(Effects of Zn-P Anti-Wear Agent by Means of FZG Gear Test Rig)

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The limiting load for scoring was investigated by means of the FZG gear testing machine. The influences of coefficient of oil viscosity and concentration of anti-wear agent (Zn-P type) in oil upon the limiting load for scoring were disclosed. Furthermore the results of these gear tests were compared with the results of 4 ball tests. We have found in general that the higher the revolutional speed of gear, the greater the influence of anti-wear agent upon the increasing rate of limiting load for scoring. And further we have found that the lower the viscosity of base oil, the greater the influence of anti-wear agent (Zn-P type) upon the increasing rate of limiting load for scoring.

1. Introduction

Many studies on load carrying capacity of gears have been reported⁽¹⁾ and lubricating oils for those researches were in many cases industrial lubricating oils. It is very important to study the influence of chemical composition of base oil and concentration of anti-wear agent upon the limiting load for scoring but only few experimental results have been reported in this field. From this point of view we have been investigated the limiting load for scoring by using three kinds of base oils (paraffine based) with different viscosities. The anti-wear agent was added to these base oils.

The tests were carried out by means of

the FZG gear test rig.⁽²⁾⁽³⁾ Furthermore the gear test results were compared with the test results obtained using the 4 ball test machine and we found some relationship between them. In practice many kinds of anti-wear agents were used. We have chosen as the most commonly used anti-wear agent a Zn-P type anti-wear agent and as base oil also the most widely used paraffine base oil.

2. Test condition

2.1 Test equipment and test methods

The FZG gear test rig was used for our experiments. The load was stepped up successively according to the load stage shown in Table 1. In order to measure the wear of the test gears, we have used a balance, which has ± 0.5 mg accuracy. Before and after the run-

Table 1 Load stage

Load stage	M_1 kgm	P_N kg	In the case of $b=5$ mm		In the case of $b=10$ mm	
			P_N/b kg/mm	P_H kg/mm ²	P_N/b kg/mm	P_H kg/mm ²
1*	0.34	10.0	2.00	31.4	1.00	22.2
2	1.4	41.4	8.28	63.5	4.14	44.9
3	3.5	104	20.7	100	10.4	71.0
4	6.0	177	35.4	131	17.7	92.9
5	9.4	278	55.6	165	27.8	116
6	13.4	396	79.2	196	39.6	139
7	17.4	514	102.8	224	51.4	158
8*	20.8	615	123.0	245	61.5	173
9	24.1	713	142.6	263	71.3	186
10*	27.5	813	162.6	282	81.3	199
11	30.8	911	182.2	298	91.1	211
12*	35.8	1038	207.6	318	103.8	225
13	39.3	1162	232.4	337	116.2	238
14*	42.6	1260	252.0	350	126.0	248
15	46.1	1363	272.6	364	136.3	258
16*	50.0	1479	295.8	380	147.9	268
17	54.5	1612	322.4	396	162.1	280

The values of P_N and P_N/b are ones at a single tooth contact.

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Fig.1 Engagement of test gears

ning of each load stage (except the load stage with sign * in Table 1) both test gears (pinion and gear) were weighed with this balance. The total number of revolutions of the test gear wheel in each load stage is always constant, namely 22500, and so the running time is variable with the revolutional speed. For example, if the revolutional speed of gear wheel $n_2 = 1500$ rpm, the total running time is 15 minutes in each load stage. The lubricating oil was sprayed from a nozzle into the meshing point of gears. The oil temperature at the spraying point was $90 \pm 3^\circ\text{C}$ and the quantity of spraying oil was about 2.4 l/min. The contact condition and the revolutional direction of test gears are shown in Fig.1 and their dimensions of the test gears in Table 2. The test range of revolutional speeds of gear wheel n_2 was from 367 rpm to 4950 rpm. The tooth profile of test gears is type A of the FZG gear test.

In order to extent our test range, i.e. in order to get the value of limiting load for scoring by using various kinds of oils, i.e. not only usual lubricating oils but also oils, which have high anti-scoring characteristics, the face width (effective face width) of the FZG test gears (original face width $b = 20$ mm) was reduced to $b = 5$ mm and 10 mm considering the bending strength of tooth in the following way:

(a) In the case of $b = 10$ mm

In order to realise effective face width $b = 10$ mm in the test gears we have used a combination of pinion and gear wheel with different face widths. Namely, the face widths of the pinion and gear wheel are 10 mm and 20 mm respectively. The face width of gear wheel is set greater than that of pinion for the purpose of strengthening tooth for bending, because the tooth of gear wheel is undercut.

(b) In the case of $b = 5$ mm

This is quite a severe condition in comparison with above mentioned case (a). In this case we must take into consideration

Table 2 Dimensions of test gears

Effective face width	b	5 and 10 mm
Diameter of tip circle	d_{k1}	88.8 mm
	d_{k2}	112.5 mm
Number of teeth	z_1	16
	z_2	24
Module	m	4.5 mm
Cutter pressure angle	α_c	20°
Diameter of generating pitch circle	d_{o1}	72 mm
	d_{o2}	108 mm
Addendum modification coefficient	x_1	+0.8635
	x_2	-0.5
Operating pressure angle	α_b	22.44°
Diameter of intermeshing pitch circle	d_{b1}	73.2 mm
	d_{b2}	109.8 mm
Length of action of approach	e_1	3.32 mm
Length of action of recess	e_2	14.79 mm
Contact ratio	ϵ	1.36
Center distance	a	91.5 mm

Indexes 1 and 2 represent pinion and gear wheel respectively.

the bending strength of test gears. In our whole test range including the highest load stage, the gear teeth must be strong enough for bending. So specially crowned (corrected) gear teeth in direction of teeth trace, which are shown in Fig.2, were used for our purpose. In this way we could realise an effective face width $b = 5$ mm of test gears.

The test gears were carburized and hardened. The material of the test gears was SCM 21 according to JIS.

The process of heat treatment for the test gears was as follows. At first the material was tempered and after that carburized at 930°C in liquid for 2 hours and quenched in oil. After quenching treatment at 180°C , it was cooled in air and then tempered again at 200°C for 1 hour and cooled in air. The surface hardness was about H_V (Vickers' hardness) 750 and the core hardness was about 420. The effective depth of hardened layer was about 0.5 mm. The test gears were ground by Okamoto gear grinding machine SHG 360 of Okamoto Machine Tool Works LTD/Yokohama. And their maximum height of the surface roughness R_{max} is about $1.5\mu\text{m}$ and the gear quality class is $0 \sim 1$ according JIS.

In addition to the test by the FZG gear test, the 4 ball tests were carried out by means of the 4 ball machine of Soda type according to JIS⁽⁴⁾ and the test results of the gear tests were compared with the results of the 4 ball tests. The test pieces (ball) for the 4 ball machine were steel balls in ball bearing of high class (according to JIS B-1501-1962). The diameter of the test ball was $3/4"$ and its surface hardness was H_V 750~850.

In the 4 ball test the three lower steel balls were fixed and one ball, which was placed on the fixed three steel balls, was rotated under load P.

It is very difficult to execute these two tests (the gear test and 4 ball test) in same condition but we have tried to make these two test conditions as close as possible to each other. So the lubricating oil temperature in the 4 ball tests was kept constant ($90 \pm 3^\circ\text{C}$) as in the gear tests and the loading condition of the 4 ball tests was taken just as in the gear tests, i.e. the load was stepped up from the first load stage to the next 0.5 kg/cm^2 higher load stage (measured by oil pressure in the 4 ball test equipment) successively, until the initial scoring appears. The running time at each load stage was constant (1 minute). The test pieces (steel balls) were not changed during one test.

After running in each load stage, con-

Fig.2 Specially crowned gear teeth

tact points of test balls were inspected with naked eye to see whether a damage had taken place or not. When the initial scoring appears, a sudden change of friction force is observed.

2.2 Wear diagram, specific wear and limiting load for scoring

The accumulated wear of the pinion and gear wheel, which is an accumulated wear from the first load stage to the next higher load stage successively, is expressed here in term of the accumulated transmitted work from the test pinion to the test gear wheel, which is also an accumulated value (transmitted work) from the first stage to the next higher load stage successively. Those accumulated wear and transmitted work are plotted in ordinate and abscissa respectively. This way of expressing the wear was tried first by Niemann and Rettig and is generally known as the FZG-method⁽²⁾⁽³⁾.

Usually we can find one tendency of this wear diagram, i.e. the inclination of this wear diagram is at first very steep and then it flattens to become almost constant (straight line). This range is a so called normal wear range. And at last the scoring occurs at a higher load stage, where the wear jumps to a high level. At this load stage the scoring comes into existence.

An inclination of the wear curve, which is comparatively flat and constant, was defined by Niemann and Rettig⁽²⁾⁽³⁾ as specific wear V_s mg/HPh = $\Sigma W / \Sigma A$ mg/HPh, where ΣW and ΣA are the total accumulated wear (mg) and the total accumulated transmitted work (HPh) at the normal wear range respectively.

Table 3 Base oils

Base oil	Viscosity	
	at 100°F (37.8°C)	cSt at 210°F (98.9°C)
I	9.82	2.47
II	65.52	8.50
III	197.2	17.18

Fig.3 Relationship between concentration of ZnDTP and viscosity changing rate

2.3 Normal load and Hertzian contact stress

Concerning our test gears the Hertzian contact stress is the highest at the point of worst loading (at the end of the path of single tooth contact), which is situated at the outer side of pinion (point B_2 in Fig.1).

Table 1 shows the Hertzian stress at this point (B_2) for effective face width $b = 5$ mm and $b = 10$ mm, and also shows the normal load P_N and the normal load per unit face width P_N/b . The Hertzian contact stress at a higher load stage is so high that its values are far beyond the yielding point of the gear material, and so strictly speaking, its values at the high load stages are not the correct contact stresses but only the calculated values based on the Hertzian formula.

2.4 Lubricating oils

Table 3 shows the three base oils, which were used for this test. The anti-wear agent (Zn-P type) is added to these three base oils in various concentrations. We used as the anti-wear agent Zn-dialkyl-

Fig.4 Limiting load for scoring by base oils I, II and III

dithiophosphate (ZnDTP), because this anti-wear agent is one commonly used.

These three base oils were paraffine type. Those base oils I, II and III are the most conventional ones for spindle oil, turbine oil and gear oil respectively. Oxidation inhibitor, rust inhibitor and deformant were added to these three base oils in order to keep a good condition for tests. All these additives are free from chemical reaction with the Zn-P anti-wear agent, so they have no influences upon the load carrying capacity of the oil.

The concentration of ZnDTP in the base oil was increased gradually. After finishing of a test with the lowest concentration of ZnDTP, the next test with one stage higher concentration of ZnDTP was carried out.

In this way the anti-wear agent ZnDTP is added in steps to the base oil. The tests with these oils (with various concentrations of ZnDTP) were carried out successively. A deterioration of these lubricating oils containing additives was not observed during our experiments.

The relationship between concentration of ZnDTP (weight percentage) and viscosity change is shown in Fig.3. The viscosity changing rate in Fig.3 is defined by the following formula.

$$\{(y-x)/x\} \times 100 \quad [\%]$$

where x is the viscosity in cSt of the base oil with no anti-wear agent and y is the viscosity in cSt of the oil with anti-wear agent.

3. Test results and discussion

3.1 Limiting load in gear test

Two effective face widths of test gear ($b = 5$ mm and $b = 10$ mm) were used in this

Fig.5 Relationship between ξ_{or} and viscosity of base oils

Fig.6 Limiting load for scoring, in the case of base oil I

test mentioned before. The test gears of $b = 10$ mm were used for the base oils I and II with various concentrations of ZnDTP and the test gears of $b = 5$ mm were used for the base oil III with various concentrations of ZnDTP.

Fig.4 shows two kinds of test results, that is, one of them is the difference of the limiting load for scoring P_N/b by $b = 5$ mm and by $b = 10$ mm in the case of the base oil III, and the other is the limiting loads for scoring by $b = 10$ mm in the case of base oils I, II and III. We express here the limiting load for scoring per unit contact length (width) by using face width $b = 5$ mm and $b = 10$ mm with $(P_N/b)_{b=5}$ and $(P_N/b)_{b=10}$ respectively. The ratio of these limiting loads $(P_N/b)_{b=5}/(P_N/b)_{b=10}$ is almost independent of rotational speed and

$$F_b = (P_N/b)_{b=5}/(P_N/b)_{b=10} = 1.32 \dots (1)$$

This ratio by Niemann et al.⁽⁷⁾ is

$$P_N/b \propto 1/\sqrt{b}$$

According to the above mentioned relationship of Niemann, the ratio is $F_b = \sqrt{2}$. It can be supposed that the difference between these two values (i.e. $\sqrt{2}$ and 1.32) is caused by the difference conditions of heat dissipation on the gear flank of Niemann's and our ex-

Fig.7 Limiting load for scoring, in the case of base oil II

Fig.8 Limiting load for scoring, in the case of base oil III

periments.

The test results of the limiting loads for scoring must be discussed on the same basis, i.e. on the same width. For this purpose all the values of $(P_N/b)_{b=5}$ were converted to $(P_N/b)_{b=10}$ by eq.(1).

As Fig.4 shows the limiting loads for scoring by three base oils decrease with an increasing revolutional speed and increase with the viscosity grade of lubricating oil.

Now we define as follows

$$\xi_{or} = P_{NO}/P_{NOI} \dots \dots \dots (2)$$

where P_{NOI} and P_{NO} are respectively the limiting load for scoring lubricated with base oil I and other oils (i.e. oils II and III).

Fig.5 shows the relationship between ξ_{or} and viscosity grade by three revolutional speeds. This ξ_{or} not only increases with the viscosity γ_{100} of lubricating oil (viscosity cSt at 100 °F or 37.8 °C) but also increases extremely with the revolutional speed n_2 . We find here that the higher the revolutional speed of test gear, the greater is the influence of viscosity γ_{100} upon ξ_{or} . The reason lies in the formation of fluid lubrication under a high revolutional speed.

Figs.6,7 and 8 show the relationships between the limiting loads for scoring $(P_N)_{b=10}$ and the revolutional speed of lubrication with base oil I, II and III respectively, in which the concentration (wt.%) of anti-wear agent (ZnDTP) is changed.

In all our test range by base oil I,II and III we have found that the higher the concentration of ZnDTP, the higher the limiting load for scoring.

In the case of base oil I (Fig.6) the

Fig.9 Relationship between ξ_o and concentration of ZnDTP

limiting load for scoring increases with the concentration of ZnDTP but the form of their relation curve is not the same. With lower concentration of ZnDTP the limiting load for scoring is almost independent of revolutional speed. In the case of base oils II and III (Fig.7 and 8) the limiting load for scoring decreases with revolutional speed in our test range (concentration of ZnDTP was 0~1.2%).

Now we define the increasing rate of limiting load for scoring by the following expression

$$\xi_o = P_N/P_{NO} \dots \dots \dots (3)$$

where P_N and P_{NO} are respectively the limiting load for scoring lubricated with base oils with ZnDTP and without ZnDTP.

Fig.9 shows this increasing rate of

Fig.10 Relationship between V_s and concentration of ZnDTP

Fig.11 Experimental results of 4 ball tests

limiting load for scoring as a function of concentration of ZnDTP. From our results it can be supposed that the increasing rate of limiting load for scoring caused by ZnDTP is dependent upon the viscosity of the base oil. This rate in the case of oil I (lowest viscosity) is the highest and on the contrary in the case of oil III (highest viscosity) it is the lowest. Under higher revolutionary speed (i.e. $n_2 = 3500$ rpm) the value ξ_0 becomes extremely high with the concentration of ZnDTP unlike the case of lower revolutionary speed (i.e. $n_2 = 643$ rpm). Under higher concentration of ZnDTP, ξ_0 is almost constant and shows one kind of satulated state.

3.2 Specific wear in the gear test

Specific wear V_s as a function of concentration of ZnDTP is shown in Fig.10. As an example we took up here the value V_s at revolutionary speed $n_2 = 643$ rpm and 3500 rpm in lubrication with base oil I and II having various concentrations of ZnDTP. From these results we can see that the specific wear V_s at a lower concentration range of ZnDTP decreases with the concentration of ZnDTP and at a higher concentration range of ZnDTP it increases with this concentration of ZnDTP.

So there is an optimum concentration of ZnDTP, which causes a minimum specific wear V_s . This optimum concentration in the case of base oil I is $1.0 \sim 1.5\%$ at a higher revolutionary speed ($n_2 = 3500$ rpm) and at a lower

revolutional speed ($n_2 = 643$ rpm) this value is about 0.5% . In the case of base oil II we found the same tendency.

3.3 Limiting load for scoring in 4 ball tests

Fig.11 shows the test results obtained using the 4 ball test machine. The relationship between the revolutionary speed and the limiting load for scoring (expressed with normal load P_N between test balls) by using the base oils I, II and III with concentration of ZnDTP $0 \sim 2\%$ is shown in this figure. The limiting load for scoring decreases with revolutionary speed and increases slightly with viscosity of lubricating oil. Fig.12 shows the limiting load for scoring is dependent upon the concentration of ZnDTP in the base oils I, II and III. Usually the limiting load for scoring increases with concentration of ZnDTP but the form of their relation between limiting load for scoring and concentration of ZnDTP is different according to viscosity of base oil. The area of hatched part represents the range of revolutionary speeds from $n = 500$ to 2000 rpm. The upper limit line of this area shows the limiting load for scoring at $n = 500$ rpm and the lower limiting line does the limiting load for scoring at $n = 2000$ rpm. The limiting load for scoring by using base oil I increases with the concentration of ZnDTP in our test range but in the case of oil II this tendency is not observed, that is, in the case of oil II the increasing rate of the limiting load is dependent upon the revolutionary speed. For example the limiting load at $n = 500$ rpm increases with the concentration of ZnDTP till about 1.0% but above 1.0% concentration of ZnDTP the limiting load is almost constant. In the case of oil III also the limiting load is dependent upon the revolutionary speed. In this case the limiting load at $n = 2000$ rpm is almost independent of the concentration of ZnDTP. But at lower revolutionary speed $n = 500$ rpm the limiting load increases with concentration of ZnDTP till 2% but in the case of higher concentration of ZnDTP (more than 2%) the limiting load decreases with concentration of ZnDTP.

3.4 Comparison of the gear test and the 4 ball test

It is supposed to be very useful to compare the results of the gear tests with the results of the 4 ball tests and to discuss the relationship between them. But the contact conditions (sliding velocity, specific sliding etc.) between two tests are quite different. In addition to that we must consider the difference due to the line contact of gear test and the point contact of the 4 ball test. So it is very difficult to compare these two kinds of test results with each other under the same condition. Therefore we want to treat this problem at a first stage from the view point of practice.

In the 4 ball test also we define the increasing rate of limiting load for scoring ξ_{or} and ξ_0 just as in the gear tests according to eqs.(2) and (3) respectively.

In order to compare ξ_{or} of the 4 ball test with ξ_{or} of the gear test, these two values (i.e. ξ_{or} of the 4 ball test and the gear test) are shown together in Fig.5. The revo-

Fig.12 Experimental results of 4 ball tests

Fig.13 Relationship between gear test and 4 ball test

lutional speeds of the 4 ball test in this figure were $n = 500$ and 2000 rpm. In comparison with the gear test it is quite significant that ξ_{ox} of the 4 ball test is almost independent of the revolutionary speed. ξ_{ox} of the 4 ball test (by $n = 500$ and 2000 rpm) is almost at same level as ξ_{ox} of the gear test for lower revolutionary speed ($n_2 = 643$ rpm). In comparison with the increasing rate ξ_{ox} of the gear test due to oil viscosity (γ_{100}) for higher revolutionary speed ($n = 1500$ and 3500 rpm), the increasing rate ξ_{ox} of the 4 ball test due to oil viscosity (γ_{100}) is very low. It seems that the reason why the increasing rate ξ_{ox} due to oil viscosity (γ_{100}) in the 4 ball test is very low lies in the unfavorable condition for the formation of fluid lubrication film due to low sliding velocity in the 4 test.

Fig.9 shows the relationship between ξ_o and concentration of ZnDTP ($0 \sim 3\%$). In order to compare ξ_o of the 4 ball test with ξ_o of the gear test, these two values of ξ_o are shown together in one figure. The revolutionary speed of the 4 ball test in this figure was $n = 500$ rpm. Due to an increasing concentration of ZnDTP, the value ξ_o under use of base oils I and II increases in all our test range, while the value ξ_o under use of oil III increases till 2 % concentration of ZnDTP and after that (about 2%) it decreases. In comparison with ξ_o of the gear test with higher revolutionary speed ($n_2 = 1500$ and 3500 rpm), ξ_o of the 4 ball test (by $n = 500$ rpm) is very low and almost at the same level as ξ_o of the gear test with lower revolutionary speed ($n_2 = 643$ rpm). It seems that this tendency is due to the lower sliding velocity in the 4 ball test, which is a very unfavorable condition for the activity of ZnDTP.

The correlation between the 4 ball test and the gear test is shown in Fig.13. The limiting loads in the gear test are expressed here with Hertzian pressure (line contact) p_{HB2} at the point of worst loading (at the end of the path of single tooth contact) and the limiting loads in the 4 ball test are expressed with $p (=p_{HB}/1.5)$, which is the Hertzian pressure at a contact point of the test ball (point contact) p_{HB} divided by 1.5.

For the treatment of surface contact strength by point and line contact it is already suggested by G. Niemann⁽⁸⁾ that the Hertzian pressure in line contact p_{HB2} is equivalent to the Hertzian pressure in point contact p_{HB} divided by 1.5. This suggestion was used for our study.

The values, which are plotted in Fig.13, are average of tests repeated more than two times. The relation curve between the 4 ball tests and the gear tests is not same, if the base oil is changed. With use of base oil I the relation between the gear tests and the 4 ball tests is not so clear because of some scattering of plotted points. This tendency is indicated with a hatched area in this figure. With use of base oil II

the relation curve is not the same as that with use of oil I. With use of base oil III the tendency is very complicated. It is therefore very difficult to establish a close relationship between the results of two tests.

4. Conclusions

Our experimental studies on limiting loads for scoring of the spur gear by means of FZG gear test rig are summarized as follows:

- (1) The limiting load for scoring is dependent upon the viscosity of lubricating oil and the concentration of anti-wear agent (ZnDTP).
- (2) It has been shown that the higher the revolutionary speed of gear, the greater the influence of anti-wear agent (ZnDTP) upon the increasing rate of the limiting load for scoring. Furthermore we have found that the lower the viscosity of base oil, the greater the influence of anti-wear agent (ZnDTP) upon the increasing rate of limiting load for scoring.
- (3) The form of relation curve between the limiting load for scoring and the pitch line velocity is changed in accordance with the concentration of anti-wear agent when the viscosity of base oil is very low (i.e. oil I). In the case of base oil with higher viscosity (i.e. oils II and III) this tendency is not so distinct.
- (4) The relationship between the gear test and the 4 ball test is discussed. Certain relationships are recognized but no close relationship in our whole test range is established.

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