Studies on Limiting Load for Scoring and Frictional Loss Of Hypoid Gears of Klingelnberg Type *

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Experiments on the load carrying characteristics of a hypoid gear of Klingelnberg type with various surface treatments were carried out using lubricating oils with various viscosities and concentrations of EP additives. As the result it was found that there are some tendencies about the limiting load for scoring and frictional loss of the hypoid gears. The limiting load for scoring is quite dependent upon the surface treatment and the concentration of EP additives but the frictional loss is mainly dependent upon the oil viscosity. These results were compared with those obtained from the spur gear test, the disk machine test, and the 4 ball test.

Key Words: Gear, Lubrication, Friction, Wear, Lubricant, Machine Element, Tribology

1. Introduction

Since a geometrical and experimental treatment of hypoid gears is very complicated, few systematical studies on hypoid gears were published. This kind of gears plays an important role in the field of industry, because the hypoid gears are used for the power transmission in automobiles. The authors have reported about the load carrying characteristics of hypoid gears of Klingelnberg type with various surface treatments are dealt with. Furthermore the influences of the kinds of lubricating oils upon the load carrying characteristics of the hypoid gear of the Klingelnberg type were studied. As the load carrying characteristics of the hypoid gears are taken the scoring, wear, and frictional loss.

culating gears. The direction of the power circulating is shown in Fig.1. The load is applied to the gears by twisting the torsion bar ④. The torque meter ⑤ indicates the torque in the power circulating circuit. ⑥ and ⑦ are the test hypoid pinion and the hypoid gear wheel respectively. ⑧ is the

Table 1 Base oils

No.	Kinematic viscosity cSt		Specific gravity 15/4°C	Viscosity index	Flash temp. °C
	40°C	100°C			
Base oil I	55.6	7.86	0.880	99.9	226
Base oil II	198	18.1	0.891	100	270
Base oil III	401	28.6	0.903	99.0	308

2. Experimental conditions

2.1 Test equipment and experimental conditions

Fig.1 shows the test equipment used for testing hypoid gears. The mark ① is the driving electric motor (revolutional speed $n = 1500$ rpm). ② is the gear box for speed change. The power is transmitted through the torque meter ③ to the power cir-

Table 2 EP additive

Sulphur content wt. %	27.5
Phosphor content wt. %	1.8
kinematic viscosity 40 °C m^2/s	83.0×10^{-6}
viscosity 100 °C m^2/s	11.0×10^{-6}
Viscosity index	120
Flashing temperature °C	121
Pour temperature °C	-18
Density kg/m^3 (at 16 °C)	1.038×10^3

Table 3 Load stages

Load stage	Load M_1 N·m (Torque on pinion)	Load stage	Load M_1 N·m (Torque on pinion)
1	3.3	9	236
2	13.7	10	270
3	34.3	11	302
4	58.8	12	344
5	92.1	13	385
6	131	14	418
7	171	15	452
8	204	16	490

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Fig.1 Test equipment for hypoid gears

gear box for the test gears, and ⑨ is the gear box for the power circulation gears. ⑩ and ⑪ are the power circulation hypoid gear wheel and hypoid pinion respectively. The power circulation hypoid gears were phosphate coated to prevent the surface damage. ⑫ is the power circulation helical gear box, in which three helical gears are installed. These helical gears have the same diameter and module. The helix angle of these three helical gears are also the same but the helix direction of the middle gear is opposite to those of the others. One of the helical gears is shifted along the shaft by means of apparatus ⑬ for applying the load (Fig.1). In this equipment the power circulation hypoid gears ⑪ and ⑫ engage the coast side surface, when the test hypoid gears ⑥⑦ engage the drive side surfaces. The lubricating system of the test hypoid gears is independent of the lubricating system of other parts. These parts, which are separated from the test hypoid gears, are lubricated with the hypoid gear oil (GL 5) in order to prevent damage. The base oil of the hypoid oil is the base oil II, which is given in Table 1. The EP additive for the hypoid gear oil is shown in Table 2.

Experiments were carried out after adjusting of tooth bearing. The adjustment of tooth bearing was done according to ABmann (2)*.

2.2 Limiting load for scoring and specific wear

The load was stepped up successively according to the load stage shown in Table 3. The limiting load for scoring was defined for showing the tendency of normal wear. This was determined in the following way, except in the range of lower loads (load stages of running-in). In the normal wear range, the wear was indicated by weight loss (mg) per one unit of transmitted work (1 kW·h). This way of expression of the

* The adjustment of tooth bearing of the test hypoid gears (Klingelberg type) is very easy.

Fig.2. Test pinion and test gear wheel

Table 4 Dimensions of test gears

	Pinion	Gear wheel
Number of teeth Z	10	39
Outside dia. mm d_k	(1) 71.49	(7) 171.29
Mean trans. module mm m_{sm}	4.98	3.72
Mean normal module mm m_{nm}	3.20	
Mean spiral angle deg β_m	50.00	30.756
Pressure Drive side deg α_{na}	20	
Coast side deg α_{nb}	22.50	
Face width b mm	(3) 28.76	(11) 27.10
Offset mm E	25.4	
Cutter radius mm r_c	55.0	
Face angle deg δ_k		
Pitch angle " δ_s	(6) 22.547	(14) 66.262
Root angle " δ_r		
Addendum mm h_k	(4) 4.8	(12) 1.6
Height of teeth mm h	(5) 7.2	(13) 7.2

specific wear was tried first by Niemann & Rettig (3). The specific wear in this report was calculated by the method of least squares.

The total number of revolutions of the pinion in each load stage is always constant, namely 22500. For example, when the revolutional speed of pinion is $n = 1500$ rpm, the total running time is 15 min in each load stage.

2.3 Test gears

The hypoid gears used in our tests were of Klingelnberg type, i.e. Cylco-Paloid type with uniform tooth depth along the entire tooth width. The details of these hypoid gears are given in Figs. 2(a) and (b), and their dimensions are shown in Table 4. The test hypoid gears were hobbed, and lapped after case hardening. Backlash C_n was about 0.15 mm. The hardness after case hardening was HRC 60~62. The accuracy was grade 6 according to DIN (Vornorm).

The tooth form in the lengthwise direction is trochoid-form as shown in Fig. 3. The test gears engaged the drive side surface. Four kinds of surface treatments were used for our experiments, i.e.

- (1) case hardening.
 - (2) phosphate coating after case hardening.
 - (3) copper-plating after case hardening.
 - (4) nitriding in salt after case hardening.
- Details of these surface treatments are explained in Table 5.

2.4 Lubricating method and lubricating oils

The lubricating oil was splashed from a nozzle on the meshing point of the test gears. The flow rate of splashed oil was about 2.4 L/min. The oil temperature at the outlet of the nozzle was 90 ± 3 °C. The kind of lubricating oils was decided to meet the following purpose of experiments, i.e. (a) limiting load for scoring and (b) frictional loss. The lubricating oil used for measuring the limiting load for scoring and specific wear, and the lubricating oil used for measuring of frictional loss were separated from each other. For the purpose of measuring the limiting load for scoring, the anti-wear agent (ZDTP) was added to the base oil II with concentration of 0.2 %

Fig. 3 Tooth cutting of Cyclo-Paloid type of hypoid gear using Klingelnberg method

of ZDTP. The oil, which was made in this way, is practically equivalent to the commonly used gear oil of Zn-P type. Furthermore the hypoid gear oils of GL 3, GL 4, GL 5 grade were used. For measuring of frictional loss, the influence of viscosity of oil upon the frictional loss of the hypoid gears was investigated, and the base oils I, II, and III, having great viscosity differences were used. Furthermore the hypoid gear oil (GL 5) was used for measuring the frictional loss.

To all kinds of oils used for our tests, the oxidation inhibitor, rust inhibitor, and defoamant were added in order to secure a good property of lubricating oil. The viscosity of lubricating oil changes according to the concentration of ZDTP, but the change of kinematic viscosity of oils produced by the addition of the ZDTP to the base oil in our test range is quite little. So we have neglected this factor.

3. Running-in

At first the test gears and power circulation gears were brought into the state of full running-in. The running-in was done in following way; the test gears and the power circulation hypoid gears were lubricated with the hypoid gear oil (GL 5) and rotated at a speed of pinion $n_1 = 1500$ rpm. The load was stepped up from the 1st load stage to the next higher stage successively up to the 14th load stage of Table 3. In this case the running time of each load stage was 15 min. After this running-in, these gears were rotated under the 6th load stage for 10 hours at the same revolutional speed using the same oil. After this running-in the running state of the test gears became very stable and there was no fluctuation of power loss. In this state the rate of friction loss was measured.

Table 5 Surface treatments

Kinds of surface treatment	Treatments
Copper-plating	This treatment consists of two steps. The 1st step of this treatment was carried out at room temperature, 60 second long under 2~3 volts. The depth of copper-plating was $1 \mu\text{m}$. The 2nd step of this treatment was carried out at $35 \sim 40^\circ\text{C}$, 5 minutes long under 3~4 volts. The depth of copper-plating was $4 \mu\text{m}$. So the total depth of copper-plating was about $5 \mu\text{m}$.
Phosphate coating	At first the test pieces were washed with trichlorethylene and heated up to 100°C , and after that they were soaked in liquid of phosphate-coating at $95 \sim 99^\circ\text{C}$, 6 minutes. The depth of phosphate coating was about $4 \mu\text{m}$.
Nitriding in liquid	Nitriding in liquid (tufftriding, NS-2 method, at 570°C , 2 hours).

4. Contact pressure and sliding velocity

Because the tooth surfaces of hypoid gears are crowned, the tooth contact is geometrically a point contact under no loading (practically the shape of contact is ellipse). This point contact changes into a line contact because wear occurs on the tooth surfaces. If we take into account these meshing conditions of this line contact, the Hertzian contact pressure p_H (N/mm²) can be calculated using the equivalent helical gear of hypoid gears (see Figure 4) on the drive side surfaces,

$$p_H \approx 74.7 \sqrt{M_1} \quad (1)$$

where M_1 (N·m) is the torque on the pinion. We assume here that the tooth bearing coefficient is $C_T = 1.2$ and the dynamical load can be neglected.

Sliding velocity V_g on the tooth surfaces of the hypoid gear on the gear wheel is shown in Figure 5(a). V_1 and V_2 are the velocity vectors of tooth surfaces of the pinion and the gear wheel respectively. V_g is the difference between V_1 and V_2 , and α_1 and α_2 are the inclination angles of V_1 and V_2 to the tooth trace respectively. w is the inclination angle of the line of contact to the tooth trace. The value of V_g is shown in Figure 5(a). This was calculated by Coleman (7). The equivalent helical gear of the hypoid gears can be used for the calculation of engaging of tooth. The middle point of the path of contact is a critical point, where a scoring may occur. At this point O on the tooth surface, the value of surface temperature is maximum. So we define the point O as the critical point. The sliding velocity at the point O is

$$V_g = k \times n_1$$

where $k = 1.03 \times 10^{-3}$, and n_1 is the rotational speed of the pinion.

For the test hypoid gear, $\alpha_1 = 22.20^\circ$, $\alpha_2 = 30.71^\circ$, $w = 22.04^\circ$.

5. Frictional loss

The frictional loss in this report does not include the frictional loss of no load running. The rotational speed of shaft I and that of shaft II in Fig.1 are the same. This rotational speed is expressed with n_1 . From the construction of experimental equipment the transmitted power from the electric driving motor to the power circulating system is

$$2\pi n_1 M_L / 60 \quad (W) \quad (2)$$

where M_L is the value of torque-meter (3) (N·m). The value of sliding velocity V_g are shown here. The frictional power loss of the power transmitting helical gear is

$$2\pi n_1 T_L / 60 \quad (W) \quad (3)$$

where T_L is the torque of power loss of the power transmitting helical gear system (12). The value of T_L is dependent upon the load

Fig.4 Equivalent crossed helical gear and spur gear for hypoid gear

(a) Absolute velocities and sliding velocities on tooth surface of hypoid gear wheel

(b) Sliding velocities on tooth surface of hypoid gear wheel

Fig.5 Absolute velocities and sliding velocities on tooth surface of hypoid gear wheel

and the revolutional speed. This value T_L was obtained from other experiments using other test equipment. The efficiencies of the test hypoid gears (6) and (7) are expressed with η_{12} and the efficiencies of the power circulation gears (10) and (11) are expressed with η_{21} . So the power loss in these two pairs of the hypoid gears is

$$2\pi(1-\eta_{12}\eta_{21})n_1M_B/60 \text{ W} \dots\dots\dots(4)$$

where M_B (N m) is the value of torque measured by torque meter (5). The total sum of the value from eq(3) and that from eq(4) is equal to the value of eq(2), therefore

$$(1-\eta_{12}\eta_{21})\frac{n_1M_B}{60} + \frac{n_1T_L}{60} = \frac{n_1M_L}{60} \dots\dots\dots(5)$$

The total efficiency of the test hypoid gear and the power circulation hypoid gear is expressed with η , then we can get

$$\eta = \eta_{12}\eta_{21} \dots\dots\dots(6)$$

considering eq(5)

$$\eta = 1 - (M_L - T_L)/M_B \dots\dots\dots(7)$$

The value T_L is already known as described before. So η can be calculated, if M_L and M_B are known values. Furthermore the value η is divided into η_{12} and / or η_{21} is known. Namely we treat here the following two cases:

- (i) η_{21} is a known value and η_{12} is an unknown value.
- (ii) Both η_{12} and η_{21} are unknown values.

In the case of (i) the value η is at first calculated using eq(7) and further by means of eq(6) η_{12} can be calculated. In the case of (ii) the test hypoid gears and the power circulation hypoid gears were brought into the state of full running-in as described before. In this case the hypoid gear oil GL 5 was used as lubricating oil. In this case μ_{12} of the test hypoid gears and μ_{21} of the power circulation hypoid gears are assumed as equal, $\mu_{12} = \mu_{21}$. This assumption is confirmed by means of one experiment. This experiment was carried

out in the following way, i.e. the temperature rises of oil in the bath of the test hypoid gear box and the power circulation gear box were measured during running of the two pairs of hypoid gears.

6. Calculating method for the coefficient of friction of hypoid gears (for $\mu_{12} = \mu_{21} = \mu$)

The efficiency of the test hypoid gears and that of the power circulation hypoid gears are

$$\eta_{12} = (\cos \alpha_{na} + \mu_{12} \tan \beta_{m2}) / (\cos \alpha_{na} + \mu_{12} \tan \beta_{m1}) \dots\dots\dots(8)$$

$$\eta_{21} = (\cos \alpha_{nb} - \mu_{21} \tan \beta_{m1}) / (\cos \alpha_{nb} - \mu_{21} \tan \beta_{m2}) \dots\dots\dots(9)$$

Fig.6 Influence of surface treatment on limiting load for scoring

Table 6 Coding of surface treatment *1 *2

No	Code	Description
1	G.C.	Case hardening in gas
2	G.C. & Phos	Case hardening in gas & phosphate coating
3	G.C. & Cu-pl	Case hardening in gas & copper plating
4	G.C. & Nit	Case hardening in gas & nitriding in liquid

(*1) In all cases, i.e. No.1~No.4 the test pieces were lapped after case hardening.

(*2) In order to express the above mentioned treatments for the pinion and gear wheel, we used an expression of A/B. This means, A and B are the surface treatment of the pinion and the gear wheel respectively.

Fig.7 Influence of kind of lubricating oils (GL grade) upon limiting load for scoring

These formulas are transformed in the following forms:

$$\mu_{12} = (\cos \alpha_{na} - \eta_{12} \cos \alpha_{na}) / (\eta_{12} \tan \beta_{m1} - \tan \beta_{m2}) \quad (10)$$

$$\mu_{21} = (\cos \alpha_{nb} - \eta_{21} \cos \alpha_{nb}) / (-\eta_{21} \tan \beta_{m2} + \tan \beta_{m1}) \quad (11)$$

If we assume $\mu_{12} = \mu_{21}$ and the values of Table 4 are used for these formulas, we can get

$$\eta_{12}^2 + C_1 \eta_{12} + C_2 = 0 \quad (12)$$

where

$$\left. \begin{aligned} C_1 &= -0.7518 - 0.7475\eta \\ C_2 &= 0.4993\eta \end{aligned} \right\} \quad (13)$$

The value of η can be obtained from eq(7), so the coefficients C_1 and C_2 can be obtained from eq(13). Then η_{12} can be obtained from eq(12). Coefficient of friction $\mu (= \mu_{12} = \mu_{21})$ can be calculated using eqs(10) and (11).

7. Experimental results

7.1 Limiting load for scoring and specific wear

Fig.8 Specific wear

Fig.9 Specific wear

Figs.6 and 7 show the limiting load for scoring (expressed with the torque of the pinion M_1) decreases with the revolutionary speed. In Fig.6 the lubricating oil used here was a gear oil of Zn-P type. The influence of various surface treatments of tooth upon the limiting load for scoring are shown in this figure. The hypoid gears with copper plating, and phosphor coating after case hardening have a higher level of the limiting load for scoring than that of the case hardened gears. The symbols for the expression of surface treatment are summarized in Table 6. It can be noticed that the nitrided tooth surface of the hypoid gears have a remarkably high level of the limiting load for scoring. The experiments were carried out using the test hypoid gears with copper plating. This copper plating was made on the surfaces of both the pinion and the gear wheel in some cases and in other cases, on the gear wheel only. The limiting load for scoring in the case of the copper plating only one side, i.e. copper plating only on the tooth surfaces on the gear wheel, is higher than that in the other cases. The range of the unstable state is shown in Figs.6 and 7. It indicates that the scoring has not occurred in this range and only a severe wear was observed. These kinds of damages are marked with W. The reproducibility of experimental results was not good in this range. The Hertzian pressure of Figs.6 and 7 was calculated from eq(1).

Fig. 7 shows the limiting load for scoring using the hypoid gear oil. As a general tendency, the limiting load for scoring using the hypoid gear oils decreases with an increasing revolutionary speed, and the higher the concentration of EP additive in the base oil is, the higher is the limiting load for scoring. In the case of the hypoid gear oil GL 5 the scoring damages were not observed in our experimental range.

Experiments were also carried out using the gear oil of Zn-P type. The specific wear of the hypoid gear with this kind of oil is shown in Fig.8(left). The specific wear increases with an increasing revolutionary speed. The specific wear is almost independent of the kind of surface treatment on the tooth surfaces. This is shown in Fig.8 (right). The specific wear changes greatly according to the concentration of EP additive.

Fig.10 Influence of load and revolutionary speed upon friction loss

Fig.11 Influence of sliding velocity on friction loss

Fig.12 Coefficient of friction in various tests

The specific wear of the hypoid gears (quoted from Fig.8), the specific wear of the spur gears, and the specific wear of the crossed helical gears are shown in Fig.9 for comparison. The abscissa is the circumferential velocity of the driving pinion. The specific wear of the spur gears and that of the crossed helical gears are quoted from the results, which were already published(9). These results show that the specific wear of the spur gears is the lowest and that of

the crossed helical gears is the highest. The specific wear of the hypoid gears lies between that of the spur gears and that of the crossed helical gears. These results are based on following factors: The crossed helical gears used in this research have shaft angle $\delta = 90^\circ$, and the shaft of spur gears are parallel to each other, i.e. the spur gear are a special case of the crossed helical gears with a shaft angle $\delta = 0^\circ$, and the shaft angle of the hypoid gears expressed with the equivalent crossed helical gear is $\delta = \beta_{m1} - \beta_{m2} = 19.244^\circ$. So this phenomenon is ascribed to the fact that the higher the shaft angle is, the higher is the specific wear.

7.2 Frictional loss

The frictional loss in our report does not include the frictional loss under no load running, that is, the frictional loss in this report treats only the frictional loss of engaging of gear tooth. So the churning loss of lubricating oil and the loss of bearings are not included here.

As Fig.10 shows, the coefficient of friction μ_{12} (or rate of friction loss ζ_{12}) decreases with an increasing load in the range of low loads (lower than 3rd load stage), but at the higher loads the value of μ_{12} or ζ_{12} is almost constant. These values are expressed with $\bar{\mu}_{12}$ or $\bar{\zeta}_{12}$. Fig.11(a) shows the coefficient of friction $\bar{\mu}_{12}$ or the rate of friction loss $\bar{\zeta}_{12}$ dependent upon the sliding velocity under various conditions. Figs.11(a),(b), and (c) show the results of surface treatments of G.C./G.C., G.C. & Phos/G.C. & Phos, using various kinds of oils. The lubricating oils for these experiments are shown in these figures.

From Fig.11(a) it may concluded, that the value of $\bar{\mu}_{12}$ (or $\bar{\zeta}_{12}$) is dependent upon the kind of oil. The higher the viscosity of oils is, the lower is $\bar{\mu}_{12}$ (or $\bar{\zeta}_{12}$) in the case of case hardened hypoid gears.

From Fig.11(b) we can see the following tendency, i.e. the value of $\bar{\mu}_{12}$ (or $\bar{\zeta}_{12}$) is also dependent upon the kind of oils, but the value $\bar{\mu}_{12}$ decreases with an increasing viscosity of lubricating oil until a certain viscosity but, in the higher viscosity range of lubricating oil, the value $\bar{\mu}_{12}$ increases with increasing viscosity of lubricating oil in the case of phosphate coated hypoid gears.

From Fig.11(c) it can be said that the value of $\bar{\mu}_{12}$ (or $\bar{\zeta}_{12}$) is dependent upon the sliding velocity V_g using hypoid gear oil GL 5. In this case we cannot see any difference between the surface treatments Nos.1 and 2 shown in Table 6. In this case the teeth of the test pinion and the test gear wheel had been provided with same kind of surface treatments. Further the relationship between $\bar{\mu}_{12}$ (or $\bar{\zeta}_{12}$) (plotted on a logarithmic scale) and V_g is Fig.11 shows, represented with a straight line.

Furthermore the value of $\bar{\mu}_{12}$ (expressed here with $\bar{\mu}$) in the case of case hardened hypoid gear is compared with that of other

tests, i.e. the spur gear tests (10) and 4 ball tests (11). These results are shown in Fig.12. In all tests the lubricating oil was the same, that is, ZDTP was added to the base oil II (weight concentration 1.0 %). From these results it can be said that the coefficient of friction of the hypoid gear tests is higher than that obtained in other tests.

8. Conclusions

Experiments for the load carrying characteristics (limiting load for scoring, specific wear, and friction loss) were carried out using a hypoid gear of the Klingelberg type. The following experimental results have been obtained.

- (1) The limiting load for scoring is quite dependent upon the surface treatments and the kind of lubricating oils.
- (2) The specific wear is affected by the kind of oil.
- (3) The frictional loss is affected by the viscosity and the kinds of lubricating oils, and affected slightly by the surface treatment (in the case of the following two combinations, i.e. case hardening/case hardening and phosphate coating/phosphate coating).

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