Studies on Effects of Tip Relief upon Load Carrying
Characteristics of Spur Gears *
(Mainly on Limiting Load for Scoring and Friction Loss)

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The effects of tip relief on the load carrying characteristics of spur gears were studied by means of FZG-Spur gear testing machine. Two kinds of tooth profiles were used for our studies. One was a nearly standard gear with almost no profile shifting, the other was a gear with a large profile shifting. In the case of the standard gear or the gear with profile of a small profile shifting, the load carrying characteristics were very good, if the tip relief was applied to both the driver gear and the follower gear, but it was unfavorable for the load carrying characteristics, if the tip relief was applied only to the driver gear. Furthermore the influences of tip relief in the case of profile shifted gears upon the load carrying characteristics were studied. The results with profile shifted gears were different from the results with standard gears.

Key Words : Gear, Lubrication, Machine Element, Tribology, Dynamics of Machinery

1. Introduction

From olden times it is a known fact that the tip relief of teeth of gears is favorable for the load carrying characteristics. But the concrete results concerning the influences of this tip relief upon the load carrying characteristics were not enough published. In the practice, nowadays therefore the tip relief was applied only by experience. Most of the studies concerning the tip relief are done for the purpose of decreasing the vibration and the noise of gears. To make clear the tendencies about the effects of tip relief upon the damage of tooth especially the limiting load for scoring and friction loss are very important.

Under these conditions of research in this field experiments were carried out in order to make clear the influences of tip relief upon the limiting load for scoring and the friction loss, and furthermore the wear. In our experiments new tendencies

were recognized about the effects of the tip relief upon the load carrying characteristics.

2. Range and purpose of experiments

The explanations of the range and the purpose of our experiments are treated in detail under the other item. Here only an outline about the range and purpose of our experiments is given.

The research in this report consists of two parts, i.e. the part of the limiting load for scoring and the part of the friction loss.

In order to carry out the experiments about the limiting loads for scoring, test gears with a small profile shifting (almost without a profile shifting) and test gears with a great profile shifting were used. The test gears almost without a profile shifting are named here test gears with profile D and the other test gears, i.e. the test gears with a large gears with a large profile shifting are named test gears with profile A. The features of these two profiles of test gears are explained under the other item. Concerning the limiting load for scoring the maximum loads were in the case of profile D, $P_N/b_k = 1974 \text{ N/mm}$, where b_k means the contact breadth of tooth of test gears. In these tests the load was stepped up successively according to the load stage.

In the test of friction loss the circumferential velocity was in comparatively low range, i.e. $V = 0.9 \sim 24.0 \text{ m/s}$, while the circumferential velocity in the case of testing the limiting load for scoring was in comparatively high range, i.e. $V =$

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7.2~39.1 m/s.

The main purpose of this research is to make clear the effects of tip relief upon the load carrying characteristics using the standard gear, i.e. test gear with profile D, and the profile shifted gear with profile A. The experiments were carried out mainly with profile D.

The tip relief was applied in the following way: There are two points, which are defined as "point of worst loading" on the tooth surface. One point is situated outside of the tooth and the other point situated inside of the tooth. The tip relief was here applied on the tooth from the worst loading, which is situated outside of the tooth, to the tip of tooth. The depth of tip relief is increased proportionally to the distance from this point of worst loading to the tip of tooth. The details are given in item 3.4. Many experiments concerning the effects of tip relief upon the load carrying characteristics were carried out. This tip relief was applied here in the following three ways:

- (1) Both to the driver gear and to the follower gear.
 - (2) Only to the driver gear.
 - (3) Only to the follower gear.
- Besides, the same kinds of experiments using test gears with profile A were carried out. The reasons why the spur gear profile A were used here are as follows:
- (1) To make clear the effects of tip relief of the gear with a large profile shifting upon the limiting load for scoring.
 - (2) To make clear the difference between the proposal of ISO standard⁽²⁾ concerning the tip relief and the results of our experiments.

Two kinds of lubricating oils were used. One of them consists of base oil I

and anti-wear agent ZDTP, namely 0.1 wt.% of ZDTP was added to the base oil I, and the other consists of base oil II and anti-wear agent ZDTP, namely 0.2 wt.% of ZDTP was added to the base oil II. The former is named here Oil I and the latter oil II.

Oil I nearly corresponds to the turbine oil with anti-wear agent and oil II nearly corresponds to the common gear oil of Zn-P type. These two kinds of oils were used in the experiments for scoring. Furthermore for measuring the friction loss of gear the other kind of oil was used, namely 1.0 wt.% of anti-wear agent (ZDTP) was added to the base oil II in order to measure the friction loss without surface damage to tooth of test gears. The lubricating oil was splashed from a nozzle on the meshing point of test gears. The flow rate of splashing oil was about 2.4 L/min. The oil temperature at the outlet of nozzle was 90 ± 3 °C. These two kinds of base oils are shown in Table 1.

3. Experimental conditions

3.1 Experimental equipment

The FZG test rig was used as test equipment. This is shown in Fig.1. Two torque-meters were installed in this rig. One torque-meter is installed on the shaft I and the other torque-meter (3) on the shaft II at the outside of the power circulating system. By using the torque-meter (8) the circulating

Table 1 Base oils

Base oil	Kinematic viscosity m^2/s		Specific gravity 15/4 °C	Viscosity index	Flash-ing temp. °C
	40 °C	100 °C			
I	$55.6 \cdot 10^{-6}$	$7.86 \cdot 10^{-6}$	0.880	99.9	226
II	$198 \cdot 10^{-6}$	$18.1 \cdot 10^{-6}$	0.891	100	270

Table 2 Load stage

Load stage	M Nm	Tooth form A		Tooth form D	
		P_N N	P_N / b_K N/mm	P_N N	P_N / b_K N/mm
1	3.3	98.5	12.3	78.8	9.8
2	13.7	405.6	50.7	324.5	40.6
3	34.7	1014	126.7	811.1	101.4
4	58.8	1738	217.3	1391	173.8
5	92.1	2723	340.4	2179	272.3
6	131.3	3882	485.2	3106	388.2
7	170.5	5041	630.1	4032	504.1
8	203.8	6026	753.2	4821	602.6
9	236.2	6982	872.7	5885	698.2
10	269.5	7967	995.8	6373	796.7
11	301.8	8923	1115	7138	892.3
12	344.0	10170	1271	8135	1017
13	385.1	11380	1423	9108	1139
14	417.5	12340	1543	9873	1234
15	451.8	13350	1669	10680	1336
16	490.0	14480	1811	11590	1449
17	534.1	15790	1974	12630	1579
18	581.3			13740	1718
19	632.9			14970	1871

Fig.1 Experimental equipment

power in this rig can be measured and by using the torque-meter ③ the transmitting power from the driving motor can be measured. The direction of the power circulation is shown also in Fig.1.

In order to determine the limiting load for scoring the test gears were installed only in gear box ⑩. In this case the gears in gear box ⑥ are used only for power transmitting.

For the purpose of measuring the friction loss the gears in the both gear box ⑩ and gear box ⑥ are used as test gears. In the gear box ⑥ the driver gear is ④ and the follower gear ⑤, and in the gear box ⑩ the driver gear is ⑫ and the follower gear ⑪. The details of other parts are shown in Fig.1.

3.2 Limiting load for scoring and specific wear

In order to determine the limiting load for scoring and specific wear, the load was stepped up successively according to the load stage shown in Table 2. The total number of revolutions of the test gear in the shaft II each load stage was always constant, namely 22500, and so the running time is variable with the revolutional speed of the shaft II. For example, if the revolutional speed of the test gear $n_{II} = 1500$ rpm, the total running time is 15 minutes in each load stage. But if we determine the running time also in this way, also at higher revolutional speed, that is, in the case of the highest revolutional speed $n_{II} = 8170$ rpm,

the running time becomes so short that this running time in each load stage is not adequate for determining the limiting load for scoring. Therefore we have chosen the same running time as the running time $n_{II} = 4950$ rpm, that is, $t_{run} = 4.5$ minutes. This is shown in Table 3.

The specific wear of the test gear V_s (mg/kW·h) was defined by Niemann & Rettig(1). In the present experiments this method was applied. Except in the range of lower loads (load stage of running-in), in the range of normal wear, the wear in mg per one unit of transmitted work (1kW·h) was defined as specific wear. The wear was measured with a balance to an accuracy of ± 0.5 mg. The wear was determined from the difference in weight of test gear before and after the running of each load stage.

3.3 Frictional loss

Profile D was used for measuring the friction loss. These experiments were carried out completely separate from the experiments

Table 3 Revolutional speed and running time

Revolutional speed of the gear on the shaft II n_{II} (rpm)	Running time per load stage t_{run} (min)
1500	15
2120	10.61
3500	6.43
4950	4.55
8170	4.55

Table 4 Test gears

Tooth form		D	A
Number of teeth	Z_1	20	16
	Z_2		24
Module	m mm	4.5	
Face width	b mm	20 (8)	
Addendum modification coefficient	X_1	0.177	0.8635
	X_2		-0.5
Cutting pressure angle	α_c Degree	20	
Dia. of generating pitch circle	d_{01} mm	90	72
	d_{02} mm		108
Operating pressure angle	α Degree	22.4	
Dia. of Intermeshing pitch circle	d_{b1} mm	91.5	73.2
	d_{b2} mm		109.8
Dia. of tip circle	d_{k1} mm	100.6	88.8
	d_{k2} mm		112.5
Center distance	a mm	91.5	
Length of action of approach	e_1 mm	9.8	3.3
Length of action of recess	e_2 mm	9.8	14.8
Contact ratio	ϵ	1.47	1.36
Surface finishing	Ground		
Gear quality grade	JIS 0-1 Grade		
Material	SCM 415		
Heat treatment	Carburizing in liquid and quenched

Fig. 2 Profiles of test gears (E_1E_2 is the range of engagement)

of scoring. At first the test gears were brought into the state of full running-in. In order to get the state of full running-in, the load was stepped up from the 1st load stage to the next higher load stage successively until the 15th load stage according to Table 2. In this case the revolutional speed was kept constant. The revolutional speeds of shafts I and II were equal to each other, i.e. $n_I = n_{II} = 1500$ rpm and the running time in each load stage was 15 minutes. After this running-in the test gears were run under the load stage of 6th stage, until the total number of revolutions 10^6 . After this running-in the running state of the test gears was very stable and there was no fluctuation of frictional loss. The rate of friction loss at lower stage was dependent upon the load and it decreased with an increasing load but at higher load stage the friction loss was almost constant and independent of the load. The mean value of the friction loss in the range of higher loads was expressed with $\bar{\epsilon}$ and this value was defined as mean rate of friction loss.

3.4 Test gear

Two kinds of test gears, namely profiles D and A, were used here for our experiments. Details are given in Table 4. The profiles of these test gears are shown in Fig.2. Main experiments were carried out with the test gears of profile D. Experiments with the gear of profile A were used only for supplement.

These two kinds of test gears have same center distance ($a = 91.5$ mm), same module,

and same cutting pressure angle, but the profile shifting, i.e. addendum modification coefficient is different in each gear and gear ratio $i = z_2/z_1$ is also different in each gear. In the case of the test gear with profile D the driver gear and the follower gear are of same dimension, i.e. $i = z_2/z_1 = 1$, and in the case of the test gear with profile A, $i = z_2/z_1 = 1.5$.

The specific sliding of the test gears is shown in Fig.3. In our report the specific slidings Σ_1 and Σ_2 are defined as follows:

$$\Sigma_1 = (V_1 - V_2)/V_1 \quad \Sigma_2 = (V_2 - V_1)/V_2$$

where V_1 and V_2 are the tangential velocities at the point of contact of gear 1 and gear 2 respectively. The tip relief was applied

Fig.4 Quantity of tip relief K

(a) Tooth form D

(b) Tooth form A

Fig.3 Specific sliding of tooth profiles D and A (E_1 and E_2 are situated at the end of line of contact, and lie on the tip circle of gear 1 and gear 2 respectively)

Fig.5 Form of tip relief

Fig.6 Relieving on tooth trace

to the tooth surface in the range from the point of worst loading to the tip of teeth. This range is expressed with S in Fig.4. The value of S is different according to the profiles D and A, i.e. in the case of profile D, $S = 3.1$ mm, and in the case of profile A, $S = 3.6$ mm. The quantity of tip relief at the tip of tooth was in the case of profile D, $K = 20, 40, 80, 120, 160$ μm and in the case of profile A, $K = 20, 40, 60$ μm . In the case of profile A, the maximum tip relief (limiting value) at the tip of tooth, the value of K , was 40 μm , because a sharpening at the tip of tooth comes out. These tip reliefs are shown in Fig.5.

Furthermore in order to carry out the experiments with tip relief $K = \infty$, the profile was modified in the following way: The tip of the test gear was shortened (cut away) up to the point of worst loading, which was situated outside of the test gears. This profile (stab profile) is expressed with D_{∞} . For our experiments this kind of profile D_{∞} was also used. The tip relief was applied in the following ways:

- (a) both the driver gear and the follower gear,
- (b) only the driver gear, or
- (c) only the follower gear.

In the case of the test gear with profile A the tip relief was applied only the pinion

Fig.7 Influence of tip relief upon limiting load for scoring (in the case of profile D)

Fig.8 Influence of tip relief upon increasing rate of limiting load for scoring (in the case of profile D)

Fig.9 Influence of tip relief upon rate of friction loss

(driver gear). Profile D_{∞} was used only in the above mentioned cases (b) and (c).

For the experiments of the limiting loads for scoring the face width shortened and set $b_k = 8$ mm. In this case the tooth surface was specially relieved. This is shown in Fig.6. For the experiments of friction loss, this kind of surface relieving was not applied. Therefore gears with whole face width, that is, $b = 20$ mm were used for these experiments.

All the test gears were ground with the OKAMOTO Machine Tool Work LTD/Yokohama. The maximum height of the surface roughness R_{max} was about $1.5 \mu\text{m}$ and their gear quality class range 0~1 according JIS. The material

of test gears was SCM 415 according JIS. The test gears were case-hardened. The surface hardness (Vickers' hardness H_v) was about 750 and the core hardness was about 420. The effective depth of hardness was about 0.5 mm.

4. Experimental results and discussion

4.1 Results with profile D

The effects of tip relief K upon the limiting load for scoring are shown in Fig.7. In this figure three kinds of tip reliefs are shown, i.e. figure (a) shows the tip relief, which is applied both to the driver gear and to the follower gear, and figure (b) shows the tip relief applied only to the follower gear, and figure (c) shows the tip relief applied only to the driver gear. From all these results in our test range ($n = 1500 \sim 8170$ rpm) it can be said that in the case of (a) the limiting load for scoring increases with the tip relief K , but in the case of (b) the limiting load for scoring also increases with the tip relief K , but the increasing degree in this case is lower than that in the case of (a). Finally in the case of (c) the limiting load for scoring did not increase with the tip relief but showed a decreasing tendency.

Fig.8 shows the results of Fig.7 in another way. The limiting load for scoring of the test gears without tip relief is as-

Fig.10 Influence of tip relief upon rate of friction loss

Fig.11 Relationship between specific wear V_s and tip relief K

Fig.12 Relationship between load sharing factor and tip relief

summed as unit 1, and the value of limiting load for scoring of test gear with tip relief is expressed with ξ i.e. $\xi = P_N/P_{NO}$, where P_N is the limiting load for scoring of the test gear with tip relief and P_{NO} is the limiting load for scoring of the test gears without tip relief. P_N and P_{NO} are expressed with the normal force on the tooth surface. By using this kind of expression, we can see the tendency in effects of tip relief upon the limiting load for scoring. In Fig.7(a) and Fig.8(a) the tip reliefs of the driver gear and the follower gear have the same value.

The rate of friction loss $\bar{\zeta}$ as a function of tip relief is shown in Fig.9(a)~(c). The rate of friction loss changes only slightly with an increasing tip relief. But the differences in the friction loss according to the above mentioned conditions (a), (b), and (c) are to be observed in these figures.

In the case of (a), that is, the case where the tip relief is applied to both the driver gear and the follower gear, and in the case (b), i.e. the case where the tip relief is applied only to the follower gear, the rate of friction loss $\bar{\zeta}$ decreases until the tip relief $K \approx 60 \mu\text{m}$, and in the range of higher tip reliefs K , the rate of friction loss $\bar{\zeta}$ is almost constant and independent of the tip relief K . In the case of (c), that is, the case where the tip relief is applied only to the driver, the tendency is not the same as that in above cases (a) and (b). The rate of friction loss $\bar{\zeta}$ in this case (c) is independent of the tip relief. Fig.10 compares the rates of friction loss in these three cases (a), (b), and (c) at rotational speed $n = 180 \text{ rpm}$.

Further the rates of friction loss using the profile D_m are shown in Figs.7~10. From these figures it can be seen that the profile D_m is on an extrapolated line from the line for the results with ordinary tip relief.

The specific wear V , $\text{mg/kW}\cdot\text{h}$ is shown in Fig.11. The upper figure of Fig.11 shows that the specific wear V_s is dependent upon the tip relief in the range $K = 40 \sim 60 \mu\text{m}$ and in the range of greater tip reliefs

than $K = 60 \mu\text{m}$ the specific wear increases again. In the lower figure of Fig.11 the specific wear in above mentioned three cases of tip reliefs is shown. In all cases the specific wear decreases until $K = 60 \sim 80 \mu\text{m}$.

In terms of limiting load for scoring, friction loss, and specific wear, the case (a) shows the best results, and the case of (c) shows the worst results among these three cases. In the case (c) the load carrying characteristics are not improved but worse on account of tip relief.

Now we analyse the above mentioned experimental results. Fig.12 shows the load sharing factor β (3) in the case of tip relief $K = 20 \mu\text{m}$. This load sharing factor is the rate of load in the range of double contact of teeth on the line of force. For example, in the case of $\beta = 1$, the load P_N is applied on the tooth surface. Fig.12(a) shows the load sharing factor in the case of (a), i.e. the tip relief is applied both to the driver gear and to the follower gear. The moving direction of the contact point on the line of force is shown in this figure. In this case the load on the tooth surface is at first very small, but with a moving of the contact point away from E_2 , the value of load on the tooth surface becomes greater, i.e. load sharing factor β becomes greater.

So the engaging condition is comparatively smooth. Therefore the load carrying characteristics are comparatively good. In the case of (b) the engaging condition is almost the same as (a). On the contrary in the case of (c), i.e. if the tip relief is given to the driver gear, the load (or load sharing factor β) on the tooth surface is very high already at the beginning of engagement of teeth. This is the main reason why the load carrying characteristics in this case of (c) are not good. In this Fig.12 point B_1 is the outside of worst loading of the gear 1. The empirical formula of the relationship between the increasing rate of limiting load for scoring ξ and the tip relief K (μm) is, when the tip relief is applied to the driver

Fig.13 Relationship between results of ξ and empirical formula

Fig.14 Comparison of empirical formula with experimental results of limiting load for scoring

gear and to the follower gear, expressed as follows:

$$\xi = P_N/P_{N0} = 1 + (K/\delta_0)^{0.5} (0.77 - V^{3.5} \times 10^{-6}) \quad (1)$$

where δ_0 (μm) is a deflection of tooth at the point of worst loading. This deflection is the sum of the deflections of contact position at the teeth of driver gear and follower gear. P_{N0} (N) is, as above mentioned, the limiting load for scoring without tip relief ($K = 0$) and P_N (N) is the limiting load for scoring with tip relief K . V (m/s) is the circumferential velocity of operating pitch circle. If $V > 50$ m/s the value ξ becomes negative according to Eq.(1). So Eq.(1) is valid only in the range of $V < 50$ m/s. So in the future the research must be carried out in order to clarify this unknown fact.

The whole deflection δ consists of three components. The first component of δ

is a deflection caused by bending moment δ_z , the second part of δ is a deflection or deformation caused by shearing δ_s and the third part of δ is a deflection or deformation caused by Hertzian deformation δ_p . In our study δ_z and δ_s are calculated by means of formula of Ishikawa (3), δ_p is calculated by means of formula of Weber & Banaschek (4). In this case δ_z and δ_s are calculated under the gear breadth $b = 20$ mm and δ_p is calculated under the breadth of gear tooth surface $b_k = 8$ mm.

Fig.13 shows the experimental data and a curve of empirical formula (1). As this figure shows, the empirical formula (1) coincides comparatively well with the experimental values.

The limiting load for scoring P_N increases with the tip relief. This is expressed with the empirical formula (1). If the tooth deflection at the point of the worst loading (expressed with δ_n) is chosen for the tip relief K , this value of the tip relief is quite reasonable. This value of relief is defined here as standard tip relief. From the above mentioned empirical formula we can get

$$\begin{aligned} \delta_n &= (1/k) P_{N0} \{1 + (K/\delta_0)^{0.5} \\ &\quad \times (0.77 - V^{3.5} \times 10^{-6})\} \\ &= \delta_0 \{1 + (K/\delta_0)^{0.5} (0.77 - V^{3.5} \times 10^{-6})\} \end{aligned}$$

where k is spring constant of tooth. In this case $\delta_n = K_N$, so we can get

$$K_N = \delta_0 \{1 + (K/\delta_0)^{0.5} f\}$$

where

$$f = 0.77 - V^{3.5} \times 10^{-6}$$

So the equation is reduced to

$$K_N = \delta_0 \{1 + (f^2/2) + (f/2)\sqrt{f^2 + 4}\} \quad (2)$$

Substituting our experimental value into the above equation in the case of $n = 3500$ rpm, we get $K_N = 67 \mu\text{m}$, and in the case of $n = 4950$ rpm, we get $K_N = 47 \mu\text{m}$, further in the case of $n = 8170$ rpm, we get $K_N = 29 \mu\text{m}$. So the value K_N decreases with an increasing revolutionary speed. These values are plotted in Fig.14.

When the value of K_N is higher than the above mentioned value, the vibration and noise become higher. Therefore the load carrying characteristics are not good, if the tip relief is $K > K_N$. But this limiting value K_N , i.e. standard tip relief must be considered more in detail and so more precise study is required.

4.2 Results with profile A

This profile has a large profile shifting. The details are shown in Table 4 and Fig.2 (lower figure). The tip relief of this profile is also shown in Figs.4 and 5. The experimental results, that is, the increasing rate of limiting load for scoring $\xi = P_N/P_{N0}$ using this profile A, is shown in Fig.15. As this figure shows, the value $\xi = P_N/P_{N0}$ increases with tip relief K .

As mentioned before in the case of profile D, we get a clear tendency about

Fig.15 Relationship between tip relief and increasing rate of limiting load for scoring

Fig.16 Relationship between load sharing factor and tip relief

the relationship between the increasing ratio of limiting load for scoring and the tip relief. But in the case of this profile A, the results concerning the tip relief are different from the above mentioned results using profile D. In the case of profile A the highest sliding velocity and the highest specific sliding come at the tip of driver gear (pinion). When this position of pinion comes into contact with the tooth of gear wheel, surface damage (scoring) to tooth appears under comparatively low load because of high sliding velocity and high specific sliding. Therefore, if the tip relief is applied to the tip of driver gear (pinion), the load sharing factor β of this position (B_2E_1 in Fig.16) is decreased through the tip relief of driver gear. This is the main reason for the increasing the limiting load for scoring using profile A. The specific sliding of this profile A is shown in Fig.3.

So it is clear that the effect of tip relief upon the load carrying characteristics (especially the limiting load for scoring) is different according to the tooth profile, for example between profile D and profile A.

The values calculated according to the formula of DIN (2) are plotted in Fig.8 and Fig.15 in order to compare with each other. Our experimental results did not coincide with the values of DIN formula.

5. Conclusions

Experiments were carried out using test gears with profiles D and A. The profile D is practically a standard gear profile with small profile shifting and the profile A is a gear profile with great profile shifting. These two kinds of gear profiles were used as test gears in order to investigate the effects of tip relief upon the load carrying characteristics. The results obtained here are:

- (1) In the case of profile D (practically standard tooth profile) the effects of tip relief upon the load carrying characteristics are the best, when the tip relief is applied to both driver gear and follower gear. When the tip relief is applied only the driver gear, the limiting load for scoring do not increase, and the decrease in friction loss is quite little.
- (2) In the case of profile D (practically standard gear profile) the effects of tip relief, i.e. increasing rate of the limiting load for scoring ξ , when the tip relief is applied to both the driver gear and the follower gear, are made clear, and the limiting values of K are given.
- (3) As the infinite tip relief (∞) one kind of stab gear was used, i.e. the tip diameter of the tip circle is shortened until the outer point of worst loading. The limiting load for scoring and the rate of friction loss using this profile are on the extrapolated line from the line with ordinary tip relief.
- (4) In the case of profile with large profile shifting (i.e. profile A) the effects of tip relief upon the load carrying characteristics are different from that of the profile

D. That is, the effects of tip relief of the driver upon the limiting load for scoring are evident.

- (5) The experimental results of our studies are compared with the value of DIN-Norm and the reasons for the disagreement between them are explained.

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