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On the Measurement of Temperature Flashes on Spur Gear Teeth*

(2nd Report, The Effects of the Amount of Addendum
Modification and the Accuracy of Gears on
the Tooth Surface Temperature)

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In the basic experiment of the study on scoring of gear teeth, it is very important to clarify the flash temperature of tooth surface.

The authors have clarified the effects of the amount of addendum modification on the temperature of tooth surface.

In the present experiment, the gear running testing machine of the power circulation type was used. The contact point of a constantan pinion and a steel gear forms a junction of thermocouple, and the thermoelectric power was measured by oscilloscope.

The following results were obtained.

(1) The wave form of oscillogram is determined in accordance with the amount of addendum modification and it nearly coincides with the value of theoretical calculation.

(2) So long as the amount of addendum modification is too large, the temperature of tooth surface becomes very high. At the most suitable amount of addendum modification, it becomes the lowest.

(3) The flash temperature of hobbled teeth is higher than that of ground teeth.

(4) The flash temperature increases in proportion to $1/2$ to $3/4$ power of circumferential speed, and that value is larger than the value presented by H. Blok.

(5) The flash temperature becomes higher as the carrying load increases, but the rising rate of temperature is less than the rate presented by H. Blok.

1. Introduction

In order to prevent the spur gear scoring, we must find the flash temperature of the tooth surface. Many studies⁽¹⁾⁽²⁾ on the temperature of the tooth surface have been carried out theoretically and experimentally, but there have been a few on the flash temperature. In the first report⁽³⁾, we compared the experimental value with the Blok's theoretical one on the variation of the surface temperature of ground gear teeth, having regard to load and circumferential speed. From these results, we expected that the design and the accuracy in

the cutting of a gear affected the temperature of the tooth surface.

In this paper, we report the effects of addendum modification and the accuracy in the cutting of a gear on the temperature of tooth surface with an aim at clarifying these points.

2. The experimental equipments and the specification of the test gear

The gear running test machine used is of torque circulation type as shown in Fig. 1. As we explained fully in the first report, the mechanism of measuring the temperature is as follows: Both gear shafts of pinion and wheel are electrically insulated, and the thermo-electric force, which is picked up outside by inserting a slider into the center of the shafts, is amplified by the A.C.

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amplifier and measured by the oscilloscope, here the slider is made of the same steel as the shafts.

Signals, which mark the meshing position and make the oscilloscope singly sweep when a couple of specific teeth comes in mesh, are generated by use of the synchronous disks a, b on the gear axes and the electric eyes c, d. The test machine is driven at any speed by the induction motor (capacity : 3HP) m, the PIV speed chain n and the change gear mechanism 1 ; and as described in the first report, we can determine the meshing point of gears by comparing the current of the electric eye at a rotating of the gears by hand slowly with that at a driving of the gears by the motor. In this case, we also measure tooth space to find the meshing point more correctly.

If the right side surfaces of gear teeth mesh at pitch point, the chord of tooth space L is expressed as follows (cf. Fig. 2) :

$$L = r_g \theta_g + r_p \theta_p + \sqrt{a^2 - (r_g - r_p)^2} \dots \dots \dots (1)$$

where

$$\left. \begin{aligned} \theta_g &= \cos^{-1} \left(\frac{r_g - r_p}{a} \right) & \theta_p &= \pi - \theta_g \\ \theta_g + \theta_g &= \frac{\pi}{z_g} (2n+1) + \text{inv } \alpha_0 + \frac{4x_g \tan \alpha_0}{z_g} - \frac{S_g}{r_g} \\ \theta_p + \theta_p &= \frac{2m\pi}{z_p} + \text{inv } \alpha_0 \end{aligned} \right\}$$

$r_g(r_p)$: radius of base circle of gear (pinion)

Fig. 1 Gear running test machine

Fig. 2 Chord of tooth space

a : center distance

α_0 : pressure angle at pitch point

$z_g(z_p)$: number of teeth of gear (pinion)

$n(m)$: tooth number counted from meshing tooth of gear (pinion)

$x_g(x_p)$: addendum modification coefficient of gear (pinion)

$S_g(S_p)$: normal backlash of gear (pinion)

If the left side surfaces of gear teeth mesh at pitch point, the above formulas have to be replaced by the following ones :

$$\left. \begin{aligned} \theta_g + \theta_g &= \frac{2\pi}{z_g} n + \text{inv } \alpha_0 \\ \theta_p + \theta_p &= \frac{\pi}{z_p} (2m+1) + \text{inv } \alpha_0 + \frac{4x_p \tan \alpha_0}{z_p} - \frac{S_p}{r_p} \end{aligned} \right\}$$

The specification of test gears is given in Table 1. The pinion is made of constantan and the gear of high carbon steel so that the pinion and the gear may make a thermocouple at the meshing point. The relation between the temperature of the meshing point and the thermoelectric force shown in Fig. 3 is obtained by the same method as used in the first report. In order to measure correctly the thermoelectric force, we must always keep the tooth surface clean in operation, otherwise welding takes place on the tooth surface and it decreases the thermoelectric force.

Table 1 Test gears

Principal items	No. of gear	x	S
Module : $m=4$	P · 5	1	0.174
Pressure angle : $\alpha_0 = 20^\circ$	P · 6	0.35	0.287
No. of teeth : 40/18	P · 7	0.65	0.258
Face width : 10 mm			
Finishing :	G · 6	-1	0.226
P · 5, P · 6, P · 7,	G · 7	-1	0.326
G · 7, G · 9, G · 11	G · 8	-0.35	0.263
...Hobbing & grinding	G · 9	-0.35	0.313
G · 6, G · 8, G · 10	G · 10	-0.65	0.242
...Hobbing	G · 11	-0.65	0.292

Fig. 3 Relation between temperature and thermoelectric force

In this experiment, welding was hardly observed; for we began the test from low speed and light load at first, and cleaned the tooth surface after every test.

For the purpose of studying the effects of the addendum modification and the accuracy in the cutting of a gear, we used three kinds of the Vau Null gears (the addendum modification coefficient of pinion is $+x$ and that of wheel is $-x$), which were cut with the hob or ground with the Reishauer gear grinding machine. The tooth to tooth variations measured with the single profile rolling checker are shown in Table 2. We selected the values of load as 10.7 kg/cm, 21.3 kg/cm, 32.0 kg/cm and 53.3 kg/cm, and the range of speed as 500 ~ 4000 rpm.

Table 2 Tooth to tooth variations measured with the single profile rolling checker

Test gears	Coefficient of addendum modification	Values of variation
P·6 : G·9	+0.35 : -0.35	15 μ
P·6 : G·8	+0.35 : -0.35	20 μ
P·7 : G·11	+0.65 : -0.65	19 μ
P·7 : G·10	+0.65 : -0.65	20 μ
P·5 : G·7	+1 : -1	12 μ
P·5 : G·6	+1 : -1	23 μ

The point P shows the pitch point.
The thick line shows the zone of single tooth contact.

Fig. 4 Oscillograms of thermoelectric force

3. The variation in the wave form of thermoelectric force due to the addendum modification

Fig. 4 shows the oscillograms of thermoelectric force at different meshing points in a pair of gears. The reappearance of these waves of surface temperature is good. Namely, in each pair of teeth, the waves have a basic form and only a little variation at the high frequency part.

The temperature curve is fairly smooth in the zone of single tooth contact, but has ripples of high frequency in the contact zone of two pairs of teeth; then in the later zone, the mean value of thermoelectric force in addendum and dedendum is measured. Fig. 5 shows the scatter of the thermoelectric force in every tooth.

Fig. 6 shows the oscillograms of the thermoelectric force in gears of various addendum modification; they vary considerably according to the amount of addendum modification. For the Vau Null gear, the point where the mesh of gears begins and one where it finishes depend on the amount of addendum modification.

If we show the position where the mesh of wheel begins with T_g , one where the mesh of pinion finishes with T_p , the intermeshing pitch point with P, and the contact ratio with ϵ , the values of $\overline{PT_g}$, $\overline{PT_p}$ and ϵ for a couple of test gears are calculated as shown in Table 3.

4. On the theoretical value of surface temperature

The flash temperature of tooth surface

Table 3 Values of $\overline{PT_g}$, $\overline{PT_p}$ and ϵ

Test gears	x_p/x_g	$\overline{PT_g}$	$\overline{PT_p}$	ϵ
P·6/G·8, G·9	+0.35/-0.35	6.864	11.553	1.56
P·7/G·10, G·11	+0.65/-0.65	3.857	13.579	1.47
P·5/G·6, G·7	+1.0/-1.0	0	15.840	1.34

$P=21.3 \text{ kg/cm}$, $n_2=500 \text{ rpm}$
Fig. 5 Variation of thermoelectric force

is calculated from the temperature rise due to the sliding friction on the contact surfaces. In the case of non-lubrication, the major part of heat is conducted into the gear body. On flash temperature, H. Blok induced the following equation theoretically.

$$\vartheta = 0.83 \frac{\mu P_n (v_1 - v_2)}{(\sqrt{\lambda_1 \gamma_1 c_1 v_1} + \sqrt{\lambda_2 \gamma_2 c_2 v_2}) \sqrt{b}} \quad \dots\dots\dots (2)$$

where

- ϑ : temperature rise of tooth surface $^{\circ}\text{C}$
- μ : coefficient of sliding friction
- P_n : normal tooth force per unit width kg/cm
- v : tangential speed of meshing point cm/sec
- λ : heat conductivity $\text{cal/cm sec}^{\circ}\text{C}$
- c : specific heat $\text{cal/}^{\circ}\text{C g}$
- γ : specific gravity g/cm^3
- $2b$: width of elastic contact cm
- n : number of revolutions per minute of gear shaft rpm
- suffix 1 : value for gear
- suffix 2 : value for pinion

When the rates of load distribution and normal tooth load are denoted respectively with q and P , we have $P_n = qP$ and $b \propto \sqrt{P_n}$. Then Eq. (2) becomes $\vartheta \propto \mu q^{3/4} P^{3/4} V^{1/2}$ where V is a circumferential speed.

Fig. 7 shows the calculated value of ϑ , where we assume the above constants as follows: $q=1$, $P=16 \text{ kg/cm}$, $n_2=270 \text{ rpm}$, $\gamma_1=7.84 \text{ g/cm}^3$, $c_1=0.113$

$\text{cal/}^{\circ}\text{C g}$, $\lambda_1=0.100 \text{ cal/cm sec }^{\circ}\text{C}$, $E_1=21\,000 \text{ kg/mm}^2$, $\gamma_2=8.9 \text{ g/cm}^3$, $c_2=0.098 \text{ cal/}^{\circ}\text{C g}$, $\lambda_2=0.055 \text{ cal/cm sec }^{\circ}\text{C}$, $E_2=11\,000 \text{ kg/mm}^2$, $\mu=0.5$. Here we use a considerably large coefficient of sliding friction, since the test gears seem to mesh in the mixed friction of both metallic and boundary lubricating one. And μ may be assumed to be constant, as we study the variation of surface temperature during the meshing period. As for the lubrication, we grease the gears only at the beginning of the test. The abscissa of Fig. 7 shows the distance from pitch point on the line of action. The lower part of Fig. 7 shows the model of the mesh of the standard gear and the profile shifted gears in connection with the above abscissa. From this figure we can know that the ratio of the length of the zone of a single tooth contact to that of the contact zone of two pairs of teeth depends on the amount of addendum modification.

Assuming the diagrams of load distribution A and B as shown in Fig. 8, and calculating the tooth surface temperature θ with the equation $\theta = q^{3/4} \vartheta$, we find that θ increases with the coefficient of addendum modification and becomes maximum near the tip of the pinion. As for the temperature measured in this study, we may call it the surface temperature itself in the zone of a single tooth contact, but it is the mean value θ_{mean} of the temperatures at a couple of contact points in the

Fig. 6 Oscillograms for gears of various addendum modifications

contact zone of two pairs of teeth. Calculating θ_{mean} from θ in Fig. 8 under consideration of the meshing region, θ_{mean} proves to be discontinuous at the boundary of two zones. This curve corresponds to the wave form of the oscillogram shown in Fig. 6.

5. Consideration on the wave form of oscillogram

5.1 0.35 m shifted gear

The single tooth contact starts rather at this side of pitch point. In this region [the sliding speed is comparatively low, the value of temperature increases smoothly and linearly with the distance from pitch point. In the contact zone of two pairs of teeth, the surface temperature at the tip of pinion is fairly high and so is even the mean temperature θ_{mean} . In these cases the experimental value is a little higher than the theoretical one.

The measured value for the ground gear lies between the values for A and B, therefore its wave form resembles that of the calculated value. Though the measured value for the hobbed gear is high in the contact zone of the two pairs of teeth, about the wave form the same result as above mentioned is concluded.

Fig. 7 Relation between θ and meshing position

5.2 0.65 m shifted gear

Since the sliding speed is zero at the pitch point, the surface temperature should take the lowest value there. However, as two pairs of teeth mesh at the pitch point of 0.65 m shifted gear, the thermoelectric force in oscillogram does not take the minimum value at the pitch point, but take it at the point where a single teeth contact begins. For the ground gear, we considered that the wave form of Fig. 6 is the middle form of the load distribution rates A and B. In the result for the hobbed gear, the thermoelectric force is higher than that of the ground gear, and does not coincide with the forms of A and B. And in this case, a steep rise of temperature is observed at the point over the pitch point by $1/4$ normal pitch, namely at the point where the single tooth contact begins. This is due to the fact that an impact occurs there, and it corresponds with the fact that there is a sharp peak in the figure of dynamic strain measured at the fillet. Therefore, it seems that a sudden variation of the load distribution occurs there. This phenomenon is specially conspicuous in the low speed drive.

5.3 1.00 m shifted

In this case, the zone of single tooth contact extends, and the tooth surface temperature becomes minimum at the pitch point or the end of the contact zone of two pairs of teeth. The oscillogram

Fig. 8 θ and θ_{mean} calculated theoretically

for the ground gear is similar to the diagram of load distribution rate 1. The oscillogram for the hobbed gear is similar to the diagram of B. In this case, as well as in the case of the $0.65\ m$ shifted hobbed gear, a sharp peak is also observed at the starting point of single tooth contact in the diagram of temperature and that of the dynamic strain.

6. Measurement of the load distribution rate

In order to measure the load distribution rate of the teeth of the test gear, the wire strain gage was affixed on the tooth fillet of compression side of gear and the strain in the fillet was measured statically with the static strain meter through the slip ring. In this case, the meshing point was determined by measuring the tooth space of the gears as mentioned above. On the other hand, inserting the steel plate of thickness $0.2\ \text{mm}$ into meshing teeth, we measured the strain in the full range of the single tooth contact from root to tip, and determined the load distribution rate by comparing this strain with that one. Since the test machine is of torque circulation type, the circulating torque varies a little by inserting of the steel plate, so we must correct the load value. At the tooth load $21.3\ \text{kg/cm}$, the corrected value is 0.89 times as large as the measured value. The load

distribution rate in case of the dynamic strain was measured too.

Fig. 9 shows the oscillogram of the dynamic strain in fillet. The mark to determine the meshing position is shown in the lower part of the figure. The pitch point and the zone of single tooth contact are shown in the figure too. It is recognized that a considerable impact occurs at the starting point of single tooth contact in the hobbed gears shifted $0.65\ m$ and $1.0\ m$.

We can determine the load distribution as shown in Fig. 10 by comparing these three strains. Though the dynamic strain and the static one do not always coincide, they have a good correspondence in general.

Many load distributions are similar to A and B of Fig. 8. However, in the case of the ground gear shifted $1.0\ m$, the tooth contact starts at the tip of gear, continuing within one normal pitch and finishes. Therefore, we consider that the greater part of mesh occurs in the zone of single tooth contact, and hardly occurs in the contact zone of the two pairs of teeth. This is due to the error of pressure angle of gear. The load distribution in this case is called C.

7. Relation between the amount of addendum modification and the surface temperature

Calculating the maximum tooth surface temperature $\theta_{\max}$ of the gears with different profile shifted values and the load distributions of A, B and C by Blok's formula, we get the curves shown in Fig. 11. In the case of C, curve ③ shows the temperature at the point where the tooth of pinion begins to mesh and curve ③' shows that at the addendum end of single tooth contact. These curves ③' and ③ express the maximum temperature over and under $0.51\ m$ of the shifted value respectively. Therefore, in this load distribution, the minimum value of surface temperature $\theta_{\max}$ occurs at the shifted value $0.51\ m$.

The relation between x_p and the scoring factor λ calculated by the Niemann-Winter's formula⁽⁴⁾ is shown in Fig. 12. These curves differ from those of Fig. 11 in form, but the value of λ becomes minimum at the shifted value $0.51\ m$. In the case of B, curve ② shows the temperature at the point where the tooth of pinion begins to mesh and curve ②' shows that at a point of addendum zone of double

Fig. 9 Oscillograms of fillet strain

teeth contact. The maximum temperature is shown with curves ②, ②' and ③' in accordance with the increment of the shifted value. From these curves the minimum value of surface temperature $\theta_{\max}$ is obtained at the amount of addendum modification 0.25 m . In the case of A, curve ① shows the temperature at a point of dedendum zone of double teeth contact and curve ①' shows that at a point of addendum zone of double teeth contact. The maximum temperature is shown with curves ① and ①' according to the augment of the shifted value. The minimum value of surface temperature $\theta_{\max}$ is obtained at the amount of addendum

modification coinciding with the cross point of these curves. Though the amount of addendum modification, which gives the minimum value of surface temperature $\theta_{\max}$, depends on the load distribution, so far as the precise gears of pressure angle 20° and gear ratio 40/18, which were used in our test, are concerned, the Vau Null gear of 0.15 to 0.25 m shifted value is effective against the temperature rise of tooth surface. To compare the calculated value of the surface temperature with the measured value, we examined the peak value of oscillogram. If the peak value shows the mean temperature at two contact points

Fig. 10 Load distribution rate

Fig. 11 Relation between $\theta_{\max}$ and x_p Fig. 12 Relation between λ and x_p Fig. 13 Relation between θ_{mean} and x_p

in the contact zone of two pairs of teeth, the maximum temperature in the range of small amount of addendum modification occurs at a starting point to mesh of pinion teeth for the load distribution B and at a point of dedendum zone of double teeth contact for the load distribution A. When the amount is large and the intermeshing pitch circle enters the dedendum zone of double teeth contact of pinion, the minimum value of θ_{mean} does not become zero as shown Fig. 8, and the peak value in oscillogram is smaller by the difference than the true value. The calculated value of θ_{mean} to x_p is shown in Fig. 13. Fig. 14 shows the tooth temperature θ_{mean} measured in this

experiment to the amount of addendum modification x_p . In this figure, the broken line shows the calculated value for the load distribution B and the full line shows that for the load distribution A, where the same calculating method was used as in Fig. 13. Though the load distribution varies a little according to the tooth profile as mentioned above and these calculated values are not always the values of the true load distribution, they are nearly equal to the experimental values except the case for a very small amount of addendum modification. In the 1.0 m shifted ground gear, the measured value is larger than the calculated one, for its load distribution rate is similar to C. The

measured value at the standard profile, in which the amount of addendum modification is zero, differs a little from the calculated one. This is due to the fact that the test ground gear* was ground with a Maag grinding machine unlike the others and the test hobbed gear had lower accuracy than the others.

8. Factors which affect the tooth surface temperature

8.1 Effects by surface finish

Fig. 5 shows the peak value of the surface temperature for the ground gear and that for the hobbed gear. The latter is greater and more scattered than the former. Furthermore, as shown in Fig. 6 the amplitude in the high frequency part for the hobbed gear is greater than that for the ground gear, therefore the flash temperature on the gear tooth of the hobbed gear is higher than that of the ground one.

8.2 Effects by the circumferential speed

In the case of constant load, the temperature of tooth surface generally increases with the increment of the circumferential speed. Fig. 15 shows the surface temperature on the specific tooth of the ground gear shifted 0.35

Fig. 14 Relation between measured temperature θ_{mean} and x_p

0.35 m shift, Ground gear, $P=53.3$ kg/cm
Fig. 15 Oscillograms for various speeds

* The test gear is one used in the experiment of the first report.

Table 4 Values of α

Coefficient of addendum modification	Finish of gear	Tooth loads *			
		10.7 kg/cm	21.3 kg/cm	32.0 kg/cm	53.3 kg/cm
1.0	Hobbing	0.21 (below 2 000) 0.62 (above 2 000)	0.53	0.67	0.57
	Grinding	0.54	0.31 (below 1 900) 2.0 (above 1 900)	0.77	0.92
0.65	Hobbing	0.44 (below 2 000) 1.00 (above 2 000)	0.91 (below 2 000) 0.40 (above 2 000)	0.68	0.74
	Grinding	0.44 (below 2 000) 1.00 (above 2 000)	0.77	0.78	0.27 (below 1 600) 0.64 (above 1 600)
0.35	Hobbing	0.75	0.57	0.71	0.68
	Grinding	0.66	0.75	0.75	0.62

* The number in parenthesis shows the revolution number per minute of pinion

m , and Fig. 16 shows that of the hobbed gear shifted 0.65 m . The oscillogram in low speed drive coincides with the theoretical wave, but that in high speed drive becomes a triangular wave. This is due to the fact that the load distribution in high speed drive differs from that in low speed drive.

As shown in Fig. 17, the relation between the peak value of thermoelectric force and the circumferential speed is generally expressed as a straight line and partially as a combination of two straight lines, where both ordinate and abscissa are logarithmic.

Putting $\theta \propto n_2^\alpha$, we get experimentally the values of α as shown in Table 4. In this table, the circumferential speed to the changing point of α in the combined straight lines is shown too. In Blok's equation α is 0.5. The value of α in this experiment is occasionally about 0.5, but generally about 0.75. In general, the gradient α of the combined straight lines changes at about 2 000 rpm.

8.3 Effects by the tooth load

In the case of constant circumferential speed, the temperature of tooth surface increases with the increment of the tooth load, but the temperature of tooth surface is not so dependent on the tooth load as on the circumferential speed.

Fig. 18 shows the oscillograms for various loads. As

shown in Fig. 19, the relation between the peak value and the load is expressed as a straight line, where both ordinate and abscissa are logarithmic. Putting $\theta \propto P^\beta$, we get the value of β as shown in Table 5. These values are considerably smaller than Blok's value 0.75.

9. Conclusion

Measuring the surface temperature of contact tooth, we clarified the effect of the addendum modification and the accuracy of gear on the surface temperature.

(i) The oscillogram of surface temperature

0.65 m shift, Hobbed gear, $P=32.0$ kg/cm

Fig. 16 Oscillograms for various speeds

changes with the variation of the amount of addendum modification as the theoretical value, and the peak value becomes larger with the increment of it. Namely, the surface temperature of gear tooth varies with the variation of the amount of addendum modification, and a favourable amount, which decreases the surface temperature, exists.

These facts are important to the design of gear in connection with prevention of gear scoring.

(ii) The profile error of hobbed gear is larger than that of ground gear, so that the hobbed gear shows a greater value than the ground gear on the surface temperature and especially on the peak value of high frequency part.

(iii) The measured value of tooth surface temperature increases in proportion to $1/2$ to $3/4$ power of the circumferential speed, therefore its increment is larger than that presented by Blok.

(iv) The tooth surface temperature gradually increases with the increment of tooth load, but the rising rate of temperature is smaller than that presented by Blok.

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Table 5 Values of β

Coefficient of addendum modification	Finish of gear	β
1.0	Hobbing	0.24
	Grinding	0.34
0.65	Hobbing	0.40
	Grinding	0.42
0.35	Hobbing	0.53
	Grinding	0.22

($n_2 = 500$ rpm)

Fig. 17 Relation between thermoelectric force and speed

Fig. 18 Oscillograms for various loads

Fig. 19 Relation between thermoelectric force and tooth load

experiment, and Mr. S. Umeda, Mr. M. Fujioka for their help in carrying out this experiment.

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Discussion

S. HASHIMOTO: (1) It is explained that, in the case of two contact points, the values of thermal electro-motive force are measured as mean value. In such a condition, is it necessary for equal contact resistance at two points?

(2) μ is very important for comparison between theoretical and experimental values. Is it possible to measure μ by this apparatus?

(3) It is explained that, the test gears are lubricated by grease and the surface must be cleaned. What is the experimental condition? Is the measurement possible under the condition of being lubricated by turbine or mobile oil?

(4) In this study, the measured temperature is compared with the theoretical value assuming the load distribution between teeth. But, the strain at the fillet (the load) is measured during the running. Then, it is considered that the measured temperature is rather better compared with the theoretical value for the measured load distribution between teeth.

(5) The load distribution between teeth can be theoretically calculated by the items of gear, accuracy and running condition. Then, can we theoretically obtain the temperature rise of the surface on the basis of such calculated load distribution?

(6) Is it considered that there is an optimum value of profile shift in case of lubrication by turbine or mobile oil?

Generally, the profile of gear is a little modified at the tip and root or by the pressure angle. In these cases, the load distribution between teeth is changed greatly, so it is considered that the optimum value of profile shift is changed also and not fixed. Then, when the value of the profile shift is given in advance, can we make the good gears for the scoring by the change of the value of the profile or pressure angle modification?

Authors' closure

(1) As described fully in the first report, we assumed that the contact resistances of two contact points were equal. In spur gear the contact

zone of two pairs of teeth is formed by two slender rectangles, in consideration of elastic deformation. Even if the lead error is zero, the tooth bearing is not uniform owing to the roughness of tooth surface but acts at the higher parts of surface. As the result, we thought that the number of contact points increased and the measured value of temperature corresponded to the mean temperature of all contact points.

(2) The coefficient of friction is an essential value to the calculation of tooth surface temperature, so we are preparing for its measurement. In the present case we assumed $\mu=0.5$ in order to calculate only the relative variation of tooth surface temperature.

(3) By the attachment of carbide and the welding of the other material, the thermoelectric force varies considerably. Therefore we must keep the tooth surface clean. The lubrication in this test is grease lubrication, and the load is supported under the mixed condition of dry and viscous friction. In this experiment the thermoelectric force was measured at the part of dry friction, therefore, in the lubrication by turbine oil, mobile oil, etc., we could not measure the thermoelectric force owing to the fact that the meshing surfaces were separated from the oil film at any position except the case of a very low speed or heavy load (c.f. the other report).^(a)

(4) It is possible to compare the measured value of temperature with the calculated value of it by measuring the load distribution in running, but in practice the load distribution in the mixed lubrication differs with every test gear. Therefore, we assumed the typical load distribution and theoretically studied the effect by addendum modification, and compared these theoretical results with the experimental results.

(5) Not only the load distribution measured at the tooth fillet, but also the load distribution along the tooth width, namely each load distribution in mixed lubrication, affects the tooth surface

(a) Y. Terauchi, *Preprint of the 39th Annual Meeting of Japan Soc. Mech. Engrs.*, No. 71 (1962) p. 83.

temperature, so the mechanism of heat generation on tooth surface is complicated. But in consideration of these results we can calculate the approximate value of surface temperature if the approximate value of the load distribution rate and that of the coefficient of friction are known.

(6) In the lubrication with mobile oil or turbine oil, the lubricating condition becomes generally a mixed one of dry and boundary ones. Though the coefficient of friction in such case differs from that in present experiment, we think that the most suitable amount of the addendum modification

exists in that as well as in this. In this case we must consider the thermal diffusion into oil. The value of the tip relief and that of the modification in pressure angle have to be so selected that the load distribution in mesh may vary as smoothly as shown in Fig. 8 (A), where the most suitable amount of addendum modification changes with the variation of the load distribution as described in this paper. If the amount of addendum modification is settled, there is naturally a limit to the decrement of the tooth surface temperature with the tip relief.

621.983.3.054:621.891.2

Pressure Lubricated Deep Drawing*

(Conclusion)

By Yasuo KASUGA** and Shigeaki TSUTSUMI***

Continued from the preceding research, the hydrostatic counter-pressure which actuated upon the material being drawn into the die chamber was investigated in detail and it was made possible to predict the pressure throughout a process. This prediction is necessary for a project of this process to be successful.

Matters of importance as to the control of the pressure during a process were pointed out.

Applicability of this process to an actual press work was ascertained by an equipment having an intermediate scale between a laboratory test and a press-shop practice.

1. Introduction

A few years ago, the "pressure lubricated deep drawing" method, which was a kind of counter-pressure drawing, was developed by the authors to improve the drawing ratio in a single operation. Its principle is as follows. As illustrated in Fig. 1, the back of a drawing die is closed so as to compose a pressure chamber being filled with a lubricating fluid. If the punch plunges into this die chamber together with the material, a counter-pressure which reacts against the punch movement will naturally be generated in the fluid enclosed by the die wall and the deforming material. Under the pressure generated, the fluid escapes from between the surfaces of die and

blank to establish an almost perfect fluid-lubrication over the die surface. Moreover, as the material is squeezed against the punch head by the pressure, the tensile stress in the critical regions of deforming cup where the material is apt to fracture decreases with the aid of friction. This

Fig. 1

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