621.833:539.4

Paper No. 164-14

STUDIES ON SCORING OF SPUR GEARS*

(3RD REPORT, EFFECT OF THE COEFFICIENT OF
ADDENDUM MODIFICATION ON SCORING RESISTANCE)

BY YOSHIO TERAUCHI** AND HIROMASA NADANO***

The scoring resistance of the Vau-Null gear pair whose centre distance is not changed by the variation of the amount of addendum modification was examined from the point of view of the critical surface temperature rise on the meshing faces of the gears, and the following correlations between scoring resistance and the amount of the addendum modification of the gears were clarified :

(1) The position of the maximum surface temperature on the meshing faces of the Vau-Null gears is remarkably affected by the number of teeth of pinion, gear ratio, the coefficient of addendum modification of gears and module.

(2) The effect of the coefficient of addendum modification on the surface temperature on the meshing faces is complicated, and the optimum value of the coefficient of addendum modification against the scoring resistance of gears is decided by gear ratio.

1. Introduction

There are a lot of investigations on the performance of gears in terms of the risk of scoring. Using the power circulating gear test rigs, the authors have revealed the effect of the thermal and mechanical properties of tooth material, and the surface roughness of meshing faces on the scoring resistance of spur gears (1). Further, the experimental results on scoring tests have shown that the scoring resistance of the gears can be assessed to a high accuracy by applying the equation of the surface temperature rise which is modified by the authors to the evaluation of the surface temperature at the instantaneous meshing position of the gear teeth lubricated with non-EP straight mineral oil (2). Therefore, it is possible to determine the dimensions of gear pair such that they will be free from scoring. In this report, the scoring resistance of the Vau-Null gear pair whose centre distance is not changed by the variation of the amount of addendum modification is examined from the point of view of the critical surface temperature rise on the meshing faces of gears, and the permissible capacity against scoring which is necessary to determine the dimensions of gear pair was clarified. Further, from the scoring tests performed on the profile shifted gears by using the power circulating gear test rigs, the effect of the addendum

modification coefficient on the scoring resistance of gears, and the correlation between the amount of addendum modification and the critical temperature rise are revealed.

2. Nomenclature

- B ; Thermal contact coefficient of tooth material, as defined by,

$$B = \sqrt{K\gamma c} \quad \text{kcal}/(\text{cm}^2 \text{ s}^{1/2} \text{ deg})$$
- b ; Effective face width cm
- Ce ; Coefficient of clearance
- c ; Specific heat of tooth material
 $\text{kcal}/(\text{g deg})$
- E' ; Reduced modulus of elasticity, as defined by,

$$\frac{1}{E'} = \frac{1}{2} \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right) \quad \text{kgf}/\text{cm}^2$$
- E ; Young's modulus kgf/cm^2
- i ; Gear ratio
- K ; Thermal conductivity of tooth material
 $\text{kcal}/(\text{cm s deg})$
- m ; Module
- n ; Revolution rpm
- Pn ; Actual tooth load per unit face width in the normal direction of the conjugal zone kgf/cm
- R' ; Radius of relative curvature, defined by,

$$\frac{1}{R'} = \frac{1}{R_1} + \frac{1}{R_2} \quad \text{cm}$$
- R ; Radius of gear teeth in contact cm
- SRMS ; Surface roughness at the incipient stage of scoring μm
- Sn ; Backlash
- Vs ; The relative sliding velocity (perpendicular to the line of action) between two rubbing tooth surfaces, as defined by, $V_s = |V_1 - V_2| \text{ cm/s}$
- x ; Addendum modification coefficient
- z ; Number of teeth
- α ; Pressure angle degree
- β ; Load-sharing

* Received 2nd May, 1977.

** Professor, Faculty of Engineering, Hiroshima University, Hiroshima.

*** Assistant Professor, Kure Technical College, Hiroshima.

γ ; Density of tooth material g/cm^3
 δ ; Deflection of gear tooth mm
 θ ; Temperature $^{\circ}\text{C}$
 θ_0 ; Bulk-temperature $^{\circ}\text{C}$
 μ ; Coefficient of friction
 ν ; Poisson's ratio of tooth material
 where, subscripts 1 and 2 relate to the pinion and the wheel, respectively.

3. Equation of surface temperature rise

For judging the risk of scoring of spur gears, the authors used the following equation considering the effect of the surface roughness on the meshing faces ⁽³⁾

$$\theta = \theta_0 + 0.145 \times 10^{-4} \mu (\beta P_n)^{3/4} E^{1/4} R^{-1/4} \times \frac{|\sqrt{V_1} - \sqrt{V_2}|}{B} \frac{1.27}{1.27 - S_{\text{RMS}}}$$

4. Equation of tooth deflection and the calculating method of tooth deflection

When the flash temperature rise at the instantaneous meshing position is evaluated by using the above-cited equation, the load-sharing β , should be assessed to a high accuracy. The load-sharing can be readily obtained by evaluating the stiffness of the meshing teeth of gears. In this report, by using Ishikawa's formula ⁽⁴⁾ which can be widely applied to the calculation of the deflection of gear tooth, the load-sharing is evaluated by dividing the reciprocal of the deflection of the meshing pair of teeth by the sum of the reciprocal of deflection of each meshing pair of teeth.

The equation of the deflection of gear tooth and the dimensions of the tooth profile shown in Fig. 1 are summarized in Tables 1 and 2, respectively.

Figure 2 shows the flow-chart applied to the calculation of the surface temperature. The surface temperature at the meshing positions of a Vau-Null gear pair of which the pinion is shifted by the positive addendum

Table 1 Equation of the deflection of gear teeth ⁽⁴⁾

Deflection of gear teeth due to bending moment	Part of trapezoid	$\delta_{Bi} = \frac{12 P_n \cos^2 \omega}{E S_f^3} (F_a^2 + \frac{X^2}{3} - F_a \cdot X) \cdot X$
	Part of rectangle	$\delta_{Bii} = \frac{6 P_n \cos^2 \omega}{E} (\frac{W-X}{S_f})^3 \{ \frac{W-F_a}{W-X} (4 - \frac{W-F_a}{W-X}) - 2 \ln \frac{W-F_a}{W-X} - 3 \}$
Deflection of gear teeth due to shearing force	$\delta_s = 2(1+\nu) \frac{P_n}{E} \cos^2 \omega \frac{1}{S_f} \{ X - (W-X) \ln (\frac{W-F_a}{W-X}) \}$	
Deflection due to tooth root inclination	$\delta_G = \frac{24}{\pi} \frac{P_n}{E} \cos^2 \omega (\frac{F_a}{S_f})^2$	
Deformation of tooth material due to Hertzian contact	$\delta_P = \frac{2(1-\nu^2)}{E} \frac{P_n}{\pi} 2(0.5 + 2 \ln 2 + \ln \frac{b}{2a})$ where $a = \sqrt{\frac{8}{\pi} (1-\nu^2) \frac{P_n}{E} \frac{R_1 \cdot R_2}{R_1 + R_2}}$	

Table 2 The dimensions essential to the calculation of the deflection of gear teeth

Figure 1. Schematic diagram of gear tooth

R_k ; Outside radius R_g ; Base circle radius R_r ; Root circle radius R_m ; Effective root circle radius L ; Distance between meshing position and the contacting point on the base circle of wheel along the line of action

Base circle diameter is smaller than effective root circle diameter

$$\omega = \frac{2L}{mz \cos \alpha} - \psi, \text{ where } \psi = \frac{\pi}{2z} + \text{inv } \alpha - \frac{S_n}{mz \cos \alpha} + \frac{2x \tan \alpha}{z}$$

$$S_k = 2 R_k \sin(\psi - \text{inv } \alpha_k), \text{ where } \alpha_k = \text{Arctan} \sqrt{\left(\frac{R_k}{R_g}\right)^2 - 1}$$

$$S_f = 2 R_m \sin(\psi - \text{inv } \alpha_f), \text{ where } \alpha_f = \text{Arctan} \sqrt{\left(\frac{R_m}{R_g}\right)^2 - 1}$$

$$X = \sqrt{R_m^2 - \left(\frac{S_f}{2}\right)^2} - \sqrt{R_r^2 - \left(\frac{S_f}{2}\right)^2}$$

$$A_n = \sqrt{R_k^2 - \left(\frac{S_k}{2}\right)^2} - \sqrt{R_r^2 - \left(\frac{S_k}{2}\right)^2}$$

$$W = \frac{A_n \cdot S_f - X \cdot S_k}{S_f - S_k}$$

Base circle diameter is larger than effective root circle diameter

S_f and R_m in the above equations are replaced by S_g and R_g , respectively.

$$S_g = 2 R_g \sin \psi$$

modification coefficient and the wheel shifted by the negative addendum modification coefficient corresponding to that of the pinion is evaluated under the following conditions; $\alpha = 20^\circ$, $m = 4.0$, $S_n = 0.5$ mm, $C_e = 0.25$, $P_n = 298$ kgf/cm and $n_1 = 6000$ rpm. Further, z_1 is chosen to be 12, 15, 18, 22, 26 and 30, and then the gear ratio corresponding to respective z_1 is varied from 1.0 to 3.0. For the case of the gear shifted by $x = -1.0$, for example, the pitch circle diameter coincides with the outside diameter.

The surface temperature rise at the successive meshing positions along the line of action is evaluated by substituting the load-sharing at the meshing positions which is calculated by the deflection of gear tooth into the above-cited equation. In this report, the terms of θ_0 and S are omitted from the calculation of the surface temperature.

On the other hand, for the case of the gears having $m = 3.0$ and 8.0 ($z_1 = 18$), the effect of module on the surface temperature was examined under the above-mentioned calculating conditions. To prevent the undercut of tooth profile, wheels are shifted by the critical addendum modification coefficient $x_0 = (14 - z_2)/17$ for the case of the gear having $\alpha = 20^\circ$. The deflection of gear tooth and the surface temperature are evaluated under the following pertinent conditions; $E = 2.1 \times 10^6$ kgf/cm², $\nu = 0.3$, $B = 2.231 \times 10^{-4}$ kcal/(cm²s^{1/2}deg) and $\mu = 0.1 \times V_p^{-0.2(5)}$, where V_p stands for the peripheral velocity at the pitch point of gear, m/s.

5. Calculated results

Figure 2 Flow-chart in order to calculate the surface temperature rise on meshing faces

Figures 3 (a) through (e) show the variation of the maximum value of the flash temperature rise $\theta - \theta_0$, at the meshing positions of the gears in which the gear ratio of the gear pair corresponding to $z_1 = 12, 15, 18, 22$ and 30 is varied from 1.0 to 3.0 under the following conditions; $\alpha = 20^\circ$, $m = 4.0$, $S_n = 0.5$ mm, $C_e = 0.25$, $P_n = 298$ kgf/cm and $n_1 = 6000$ rpm. The abscissa in these figures is the gear ratio, i . Figure 3 (a) shows the variation of the maximum value of the flash temperature rise for the case of $z_1 = 12$. The critical addendum modification coefficient x_0 of the gear having $\alpha = 20^\circ$ and $z = 12$ is found to be 0.11, and consequently the flash temperature rise is calculated for the case of gear pair of $x \geq 0.2$. Further, the gear pair of $x \geq 0.7$ is unsuitable for the tooth profile since the tip of tooth gets sharp. In this figure, the region depicted by a solid line indicates that the maximum surface temperature rise occurs when the tip of pinion mates with the root of wheel, and consequently this region is considered to be most

Figure 3 Correlation between surface temperature and gear ratio (Part 1)

Figure 3 Correlation between surface temperature and gear ratio (Part 2)

critical for scoring from the point of view of the flash temperature rise. On the contrary, the region depicted by a dotted line indicates that the maximum temperature rise occurs when the tip of wheel mates with the root of pinion. On the other hand, the region depicted by a one dotted chain line indicates that the maximum temperature rise occurs at the highest position of single tooth contact of pinion. For the cases of

the gear pair shifted by $x_1 = 0.2$ and $x_2 = -0.2$, the maximum surface temperature rise occurs when the tip of wheel mates with the root of pinion. With an increase of i , the maximum value of the flash temperature rise tends to increase, and consequently it may be considered that the scoring resistance of the gears decreases. On the other hand, for the cases of the gear pair shifted by $x_1 = 0.3$ and $x_2 = -0.3$, the maximum temperature rise occurs when the tip of pinion mates with the root of wheel for comparatively small values of i . For the gear pair of $i \geq 1.83$ ($z_2 = 22$), however, the maximum surface temperature rise occurs when the root of pinion mates with the tip of wheel. For the cases of the gear pair shifted by $x_1 = 0.6$ and $x_2 = -0.6$, the maximum temperature rise occurs at the highest position of single tooth contact of pinion.

Similarly, for the case of $z_1 = 15, \dots, 30$, the correlation between gear ratio and maximum surface temperature is shown in Figs. 3 (b) through (e). The maximum surface temperature rise tends to decrease with an increase of z_1 . Though the effect of the addendum modification coefficient of gears on surface temperature is complicated, there is an optimum of addendum modification coefficient against the scoring resistance of the gears. For instance, in the case of $i = 1.0$ shown in Fig. 3 (e), the standard gear pair provides the lowest value of the flash temperature rise in comparison with the other profile shifted gear pair, and the temperature rise remarkably increases with an increase in addendum modification coefficient of gears. Therefore, the standard gear pair will provide higher scoring resistance than the other profile shifted gear pairs, and the scoring resistance of gears will gradually decrease accordingly as the addendum modification coefficient of gears increases.

Figure 4 shows the variation of the flash temperature rise along the line of action of the gear pair having $z_2 = 40$ corre-

Figure 4 The variation of the flash temperature rise in the successive meshing positions along the line of action

sponding to $i \doteq 2.2$ in Fig. 3 (c). From this figure it would appear that for the case of standard gear pair the maximum value of the flash temperature rise at the meshing faces occurs when the root of pinion mates with the tip of wheel. On the contrary, for the cases of the gear pair shifted by $x_1 = 0.4$ and $x_2 = -0.4$, it occurs when the tip of pinion mates with the root of wheel. On the other hand, for the cases of the gear pair shifted by $x_1 = 0.9$ and $x_2 = -0.9$, it occurs at the highest position of single tooth contact of pinion. Thus, the meshing position where the flash temperature rise becomes maximum is considerably affected by addendum modification coefficient, z_1 and i . The above-cited relation is shown in Fig. 5. When the values of z_1 and i of gear pair are decided, the meshing position where the flash temperature rise is at its maximum may be determined by addendum modification coefficient x . In this figure, we denote region I, region II and region III for the meshing positions such that the flash temperature rise is at its maximum at the tip of wheel, at the tip of pinion and at the highest position of single tooth contact of pinion, respectively. The scoring resistance of gears has been conventionally appreciated by evaluating the surface temperature when the tip of one gear mates with the root of the other. However, for the case of the gear pair shifted by a comparatively large addendum modification coefficient, the maximum value of the flash temperature rise occurs in the region III. This region is recognized to be most critical for scoring. Therefore, when the scoring resistance of the gear pair shifted by large addendum modification coefficient is judged by evaluating the critical surface temperature rise, it is necessary to calculate the surface temperature at the highest position of single tooth contact of pinion.

In order to investigate the effect of module on the surface temperature, the surface temperatures on both gear pair having

Fig. 5 The meshing positions where the flash temperature rise becomes maximum by means of the variation of addendum modification coefficient and gear ratio

Fig. 6 Effect of module on surface temperature

$m = 3.0$ and $m = 8.0$, respectively are calculated by varying i from 1.0 to 3.0 under the calculating conditions of $\alpha = 20^\circ$, $z_1 = 18$, $S_n = 0.5$ mm, $C_e = 0.25$, $P_n = 298$ kgf/cm and $n_1 = 6000$ rpm. These calculated results are shown in Figs. 6(a) and (b). From these figures it is clear that the effect of module on the surface temperature is not so large as that of the number of teeth, and the maximum value of the flash temperature rise slightly increases with an increase in module.

6. Scoring tests using power circulating gear test rigs

From the calculated results of the flash temperature rise at the meshing faces of the Vau-Null gear pair it was revealed that the flash temperature rise is remarkably affected by both addendum modification coefficient and gear ratio. In this section, the scoring tests of the profile shifted gear pair were carried out using the power circulating gear test rigs, and the effect of the addendum modification coefficient on scoring resistances of gears and the correlation between the amount of addendum modification coefficient and the critical surface temperature rise were clarified.

6.1 Test gears

The test gears are made of SCM22H. These gears are ground by the Reishauer gear grinding machine after gas-carburizing as shown in Table 3. Pair A, Pair B, Pair C,

Pair D and Pair E indicate the standard gear pair, $x_{1,2} = \pm 0.1, \pm 0.2, \pm 0.3$ and ± 0.5 , respectively.

The accuracy of tooth profile of the test gears corresponds to that of 1st class in JIS except Pair A. In the wheel of Pair A, there is a tooth profile error (having an undercut of approximately 15 μm depth in the vicinity of the meshing limit in the dedendum) at a distance of approximately 2 mm along the line of action from the position of the meshing limit at the root. The accuracy of tooth profile of Pair A corresponds to that of 3rd class in JIS. Wheels of Pair A, Pair B, Pair C and Pair D are chamfered from the tip along the tooth profile to the position of approximately 0.7 mm. Therefore, the effective length of action and the contact ratio of the above-cited gear pair decrease.

6.2 Lubricants

The test gears are lubricated with a # 150-spindle oil. The oil is jetted towards the meshing faces at the rate of 2.0 l/min. The oil temperature is controlled at $70 \pm 2^\circ\text{C}$

Table 3 Testing gears

	Pair A	Pair B	Pair C	Pair D	Pair E
Number of teeth	18	18	18	18	18
z_1	40	40	40	40	40
z_2	116	116	116	116	116
Centre distance mm	116	116	116	116	116
Pressure angle α°	20	20	20	20	20
Module m mm	4	4	4	4	4
Backlash C_e mm	0.5	0.5	0.5	0.5	0.5
Effective face width b mm	9.8	9.8	9.8	9.8	9.8
Addendum modification coefficient	0	0.1	0.2	0.3	0.5
x_1	0	-0.1	-0.2	-0.3	-0.5
x_2	72	72	72	72	72
Pitch circle d_1 mm	160	160	160	160	160
diameter d_2 mm	80.0	80.8	81.6	82.4	84.0
Outside diameter d_{o1} mm	168.0	167.2	166.4	165.6	164.0
diameter d_{o2} mm	19.15	18.98	18.78	18.55	17.97
Length of action mm	1.62	1.61	1.59	1.57	1.52
Contact ratio ϵ	SCM22H, and these gears ground by Reishauer gear grinding machine				
Material and heat-treatment	3rd class	1st class	1st class	1st class	1st class
Accuracy of tooth profile in JIS	17.55	17.36	17.12	16.84	17.97
Actual length of action mm	1.49	1.47	1.45	1.43	1.52
Actual contact ratio ϵ					

by the thermostat during all tests.

6.3 Experimental method

Scoring tests were carried out by the speed stepped up under the constant load per face width $P_n = 27.17 \text{ kgf/mm}$ as shown in Table 4-1. When a surface failure of gears is not observed within the speed range of Table 4-1, the scoring tests are continued to the load range shown in Table 4-2 (the number of revolutions of pinion is held at constant 6070 rpm). Since the methods of scoring tests and the judging method of surface failure are the same as of described in 2nd report (1), the descriptions of these methods are omitted in this report.

6.4 Testing results and observations

6.4.1 The variation of the tooth profile and the surface roughness of meshing faces

Figure 7 depicts the variation of the tooth profile and the surface roughness along the face width at the successive meshing positions in the case of Pair B. A variation of the tooth profile is not observed until the speed range V-11. A surface failure by scoring occurs at both meshing positions of tip and root of pinion at the same time. In this case, a progressive increase of wear occurs in the vicinity of the meshing position where the root of wheel mates with the tip of pinion. This trend coincides with the variation of the surface roughness along the face width.

The right side in Fig. 7 shows photographs of the scored tooth surface at the load range P-II. A destructive surface failure occurs in the vicinity of the root of wheel and of the tip of pinion.

6.4.2 Critical surface temperature rise

Figure 8 shows the variation of the flash temperature rise at the successive meshing positions along the line of action of the test gears which is calculated under the operating condition of $P_n = 27.17 \text{ kgf/mm}$ and $n_1 = 6070 \text{ rpm}$. In this figure, x marks the starting positions of a surface failure owing to scoring. These surface temperature rises are calculated by substitut-

Table 4-1 Speed range

Speed range	V-1	V-2	V-3	V-4	V-5	V-6	V-7	V-8	V-9	V-10	V-11
Number of revolution of pinion n_1 rpm	1750	2060	2400	2770	3170	3580	4030	4500	5000	5520	6070
Total carrying work HP.h		18.8	28.2	37.6	47.1	56.5	65.9	75.3	84.7	93.1	102.5
Peripheral velocity of pitch circle m/s	6.8	8.0	9.3	10.7	12.2	13.8	15.6	17.4	19.3	21.3	23.5

Table 4-2 Load range

Load range	P-I	P-II	P-III	P-IV	P-V	P-VI	P-VII	P-VIII	P-IX	P-X	P-XI	P-XII	P-XIII
Tooth load kgf/mm	27.8	28.4	29.0	29.6	30.3	30.9	31.5	32.2	32.8	33.4	34.1	34.7	35.4
Torque of wheel kg-m	20.9	21.3	21.8	22.3	22.7	23.2	23.7	24.2	24.7	25.1	25.6	26.1	26.6
Hertzian stress kgf/mm ²	102	103	104	105	106	107	108	109	110	111	112	114	115
Total carrying work HP.h	112	122	132	142	153	163	174	185	197	208	220	232	244

Fig.7 Variation of tooth profile and surface roughness along the face width of the Pair B

Fig.8 The variation of the flash temperature rise on the successive meshing positions along the line of action

ing the load-sharing in the successive meshing positions which is obtained by measurement of static strain with the strain gauge at the tooth fillet of wheel into the equation of the surface temperature rise mentioned above. From this figure it would appear that the position of the destructive surface failure caused by incipient scoring well coincides with that of the maximum value of the flash temperature rise.

Figure 9 shows the variation of the critical surface temperature rise at incipient scoring, the total amount of carrying work (the sum of the load carrying work at each speed and load range in Table 4-1 and 4-2 up to the incipient stage of scoring) and the scoring factor calculated by PVT formula. The surface temperature rises, θ_A and θ_B are calculated by substituting the tooth load, the sliding velocity and the coefficient of tooth friction at the incipient stage of scoring, and the measured surface roughness on the meshing faces at incipient scoring

into the equation of the surface temperature rise. θ_A indicates the critical surface temperature rise using the load-sharing β , that is calculated by Ishikawa's formula of the deflection of mating tooth. On the other hand, θ_B indicates the critical surface temperature rise using β that is obtained by the measurement of static strain at the tooth fillet of wheel. The bulk-temperature of gears θ_0 , is assumed to be equal to the oil temperature, 70 °C. Further, the PVT value indicates the larger one of its values calculated at the tip of pinion and wheel by using β that is calculated by Ishikawa's formula. As evident from this figure, the critical surface temperature rise for scoring, θ_B lies between 215 and 240 °C, and the constancy of the critical surface temperature rise given by θ_B is recognized to be somewhat better than that by PVT formula. In the case of Pair A, θ_A at incipient scoring is remarkably larger than θ_B . Such a difference is considered to come from the following fact ;the position of destructive surface failure differs from the meshing position where θ_A is maximum, and its failure occurs at the meshing faces slightly inside from the tip of wheel. Further, there is a tooth profile error having an undercut of approximately 15 μ m depth in the vicinity of the root of wheel, and consequently the load-sharing $\beta = 0.46$ evaluated in the vicinity of the tip of wheel in the case of the gears having no tooth profile error is approximately 1.7 times larger than $\beta = 0.29$ obtained at the meshing position indicated by x mark in Fig.8 by means of the measurement of static strain at the tooth fillet of wheel.

Under the calculating conditions of $P_n = 298$ kgf/cm, $n_1 = 6000$ rpm, $\alpha = 20^\circ$, $m = 4.0$, $C_e = 0.25$ and $S_n = 0.5$ mm, the flash temperature rise (the terms of θ_0 and S are omitted in the calculation of temperature rise) is evaluated in the case of

Fig.9 The variation of the critical temperature rise at incipient scoring, total carrying work and PVT value ($P_n = 38.65$ kgf/mm, Torque of wheel = 29.06 kg-m at the load range P-VIII)

Pair A, Pair B, Pair C and Pair D (wheel of these Pairs is chamfered from the tip along the tooth profile to the position of approximately 0.7 mm), and Pair E having no chamfering. From the calculating results, θ_A of Pair C, Pair D, Pair B, Pair E and Pair A are 73.3, 85.5, 87.0, 94.5 and 108.2 °C, respectively. Where, the load-sharing in the case of Pair A decreases due to the tooth profile error of wheel, and the surface temperature also decreases. Therefore, except Pair A having a large tooth profile error, Pair C provides the highest scoring resistance of all test gears, and consequently the order of the test gears for scoring resistance is : Pair D, Pair B and Pair E. This trend well coincides with the experimental results of the gears.

It is desirable that the designer takes account of the scoring resistance of gears considering the effect of tooth profile error. However, since the troubles owing to scoring are apt to take place in the case of the gear pair having a comparatively high accuracy of tooth profile in practice, it is possible to appreciate the scoring resistance of gears by using the above-cited methods.

7. Conclusions

The calculating results of surface temperature and the experimental results of scoring tests are summarized as follows:

(1) In the case of the Vau-Null gear pair, the meshing positions where the flash temperature rise becomes maximum are remarkab-

ly affected by the variation of z_1 , i , x and m . For instance, in the case of the gear pair shifted by a comparatively large addendum modification coefficient, the maximum of the flash temperature rise occurs at the highest position of single tooth contact of pinion. Therefore, this region is recognized to be most critical for scoring.

(2) Though the effect of the addendum modification coefficient of gears on the flash temperature rise is complicated, the optimum addendum modification coefficient against the scoring resistance of gears may be determined for a given i .

(3) When lubricated with a # 150-spindle oil, the critical surface temperature rise for scoring lies between 215 and 240 °C, and the scoring resistance may be appreciated by means of the critical surface temperature rise to a high accuracy.

References

- (1) Terauchi, Y., et al., Bull. JSME, Vol. 18, No. 125 (1975-11), p.1315.
- (2) Terauchi, Y. and Nadano, H., Jour. Japan Soc. Lub. Engrs. (in Japanese), Vol. 20, No. 11 (1975-11), p.791.
- (3) Terauchi, Y. and Nadano, H., Preprint of Japan Soc. Mech. Engrs., (in Japanese), No. 768-3 (1976-11), p.110.
- (4) Ishikawa, N., Bull. T.I.T., Series-A, No. 3 (1957).
- (5) ISO/TC60/WG6(1975-3), Calculation of scoring resistance of gear drives.