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621.833.1.036:536.532.08

On the Measurement of Temperature Flashes on Spur Gear Teeth*

(1st Report, The Case of Meshing with a Pair
of Ground Standard Gears)

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As a basic experiment on gear scoring, the surface temperatures of gear teeth are measured.

So far, we can find the experiment for measuring the mean surface temperatures of gear teeth. But it is significant in considering a countermeasure of scoring that from another point of view we know the change of the instantaneous temperature with the change of the meshing position.

In our experiment the thermocouple consisted of a constantan pinion and a steel wheel, and the thermoelectricity produced in it was observed by oscilloscope.

As a result we reached the following conclusion:

(1) On the relation between the meshing position and the surface temperature of gear teeth, the experimental value is nearly equal to the value calculated theoretically.

(2) As the surface temperature of gear teeth is largely influenced by the load distribution, it is most desirable for designing the gear that we try to change the profile of gear teeth so that the surface temperature may be minimum, and to avoid the local contact of gears.

(3) The surface temperature of gear teeth rises in proportion to the circumferential speed of gears in the range of lower speeds, but in the range of higher speeds its change becomes small. Speaking of the influence of loads, the temperature changes in proportion to $1/2$ the power of the load, but in the range of higher speeds the change becomes small.

1. Introduction

The scoring is one of the gear failures, which occurs on the contacting tooth surface of heavy loading and high speed gears in the initial period of use. It is remarkable that in a slight case tooth surfaces sometimes grow glassy, and in a serious case the tear mark or welding wears occur

on the meshing surface. Such failure comes from the fact that the lubricating oil film is broken down due to the high temperature flashes on the tooth surface, which gets burnt owing to the frictional heat at the contact point.

Many studies on the scoring of tooth surface have been carried out theoretically or experimentally, but there have been few on the flash temperature.

The scoring, however, is not only a lubrication failure as generally believed, but also a failure of the design of the gears and teeth finishing. In the hope of obtaining decisive data on this question,

* Received 12th October, 1962.

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we have measured and studied the temperature flashes of meshing surfaces.

2. Experimental equipments

The specification of testing gear is shown in Table 1. For the purpose of affixing the electric resistance strain gage on the tooth fillet, the gear backlashes are designed to be large. The gear running test machine, shown in Fig. 1, is a torque circulation type. As the pinion and wheel shafts, slender torsion bars are used lest the testing results should be affected by the vibration due to the errors of gears for torque circulation. As the means of measuring the temperature at the contact point of tooth, the thermoelectric method is applied. In this experiment we use the pinions of constantan and wheels of steel. As shown in Fig. 1, both gear shafts of pinion and of wheel are electrically insulated. To measure the thermoelectric voltage, the pointed bars of the same material are contacted at the center of shafts as the electric connector, and A.C. amplifier and oscilloscope are also used.

Now, to mark the meshing position, a circular disk with slits of the same divisions as teeth is set on the shaft. This disk shuts the light to the electric eye intermittently, and variations of output of electric eye are measured by means of D.C. amplifier and oscilloscope. We should study the specified pair of teeth, so that we may know accurately the relations of the surface temperature vs. the circumferential speed, the tooth load and the errors of gear. In this case, another circular plate with a slit is set on the wheel axis. At the

moment that this slit coincides with the slit of circular plate on the pinion axis, the beam of light passes through them and reaches another electric eye. Then the current of electric eye is led to the single sweep starting circuit. This circuit was described in detail in the former report⁽¹⁾.

The power circulating rig may be driven at any speed by using the PIV speed chain, the change gear and the induction motor (capacity : 3 HP). The temperature of gear body and the average temperature of tooth surface were measured with the mV meter, but the temperature increase was scarcely observed in this experiment. The diagram of single profile rolling check is shown in Fig. 2.

3. Determination of the meshing point

The meshing point is determined by oscillogram of photoelectric force. After measuring the current of the electric eye with micro-ammeter rotating the gear slowly, we studied the results indicated by the oscillogram.

As shown in Fig. 3, the meshing point is determined by the measurement of tooth space where the opposed involutes intersect a tangent to both base circles, which is shown with the maximum chord L . If the gears mesh at pitch point, the chord L is expressed as follows:

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

where

$$\theta_g = \cos^{-1} \left(\frac{r_g - r_p}{a} \right), \quad \theta_p = 2\pi - \theta_g \quad \dots \dots \dots (2)$$

Fig. 2 Diagram of single profile rolling check

Fig. 1 Running test apparatus

Fig. 3 Determination of meshing point

$$\left. \begin{aligned} \theta_g + \theta_g &= \frac{\pi}{z_g}(2n+1) + \text{inv} \alpha_0 - \frac{s}{r_g} \\ \theta_p + \theta_p &= \frac{2m\pi}{z_p} + \text{inv} \alpha_0 \end{aligned} \right\} \dots\dots\dots (3)$$

In these equations

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

a : center distance

α_0 : pressure angle at pitch point

z_g (z_p): number of teeth of wheel (pinion)

$\theta_g, \theta_g, \theta_p, \theta_p$: angles as shown in Fig. 3

s : difference between the actual circular thickness and the theoretical circular thickness on pitch circle

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

In the meshing condition shown in Fig. 3, s of the pinion has not an effect on L .

4. Tooth surface temperature at any meshing point

The measured value of surface temperature is shown in Fig. 4 under the following conditions: tooth load 16 kg/cm; speed of pinion 270 rpm; peripheral speed of pitch circle 1.02 m/sec; lubricant, grease. Fig. 4 shows the relationship between the thermoelectric force (mV) and the meshing point, the latter moves from left to right. In this figure, the following notations are used: p, pitch point; a and b, the points where the mesh of reduction gears begins and finishes; ba, zone of single tooth contact. When the MAAG gears are highly accurate, the profile errors are small. In this case, if the tooth load is comparatively small, the tooth deflection is little, and the effect of profile error occurs to some degree in the diagram of thermoelectric force. Considering from the inclination seen in the diagram of each tooth, the same character is observed. Namely the curve of the temperature variation is fairly smooth in the zone of single tooth contact, but the variation is still large in the contact zone of the two pairs of teeth. Moreover, the curve is not smooth and fine ripples are observed on the fundamental wave. The reappearance of these waves is good. If the same mating teeth are made to mesh, the curves coincide with one another. It seems, therefore, that these curves show the tooth surface temperature. In the zone of single tooth contact, the contact point is only one. In the contact zone of the two pairs of teeth, however, there are two contact points, one on the arc of approach and the other on the arc of recess. The vertical coordinate of oscillogram, therefore, shows the surface temperature itself in the zone of single tooth contact, but it shows the

mean value according to the contact resistances of addendum and dedendum in the contact zone of the two pairs of teeth. As it is difficult to separate them, the mean value may be regarded as the mean temperature.

The thermoelectric force of the gear thermocouple is calibrated as follows. After calibrating the thermocouple consisting of the iron and constantan wires (diameter; 0.25 mm), we plotted the solid points in Fig. 5. This thermocouple was welded near the contact point of the mating teeth. Heating

Tooth load: 16 kg/cm
Speed of pinion: 270 rpm
p: pitch point

Bold line shows the zone of single tooth contact

Fig. 4 Relation between thermoelectric force and contact position

around the contact point, we measured the thermoelectric force of gear thermocouple as compared with the value of the above wire thermocouple. These values are plotted with hollow points in Fig. 5. There is some doubt that these values may be adopted under grease lubrication, but in the present experiment we use this calibrating result.

Now by means of Blok's theoretical equation⁽²⁾, that is,

$$\vartheta = 0.83 \frac{\mu P_n (v_1 - v_2)}{(\sqrt{\lambda_1 \gamma_1 c_1 v_1} + \sqrt{\lambda_2 \gamma_2 c_2 v_2}) \sqrt{b}} \quad (4)$$

we calculated the surface temperature under this experimental condition. In Eq. (4), the following notations are used: ϑ , tooth surface temperature °C; μ , coefficient of sliding friction; P_n , normal tooth force per unit width kg/cm; v , tangential speed of gear cm/sec; λ , heat conductivity cal/cm sec °C; c , specific heat cal/°C g; γ , specific gravity g/cm³; $2b$, width of elastic contact cm; suffix 1, value of gear; suffix 2, value of pinion. When the rate of load distribution and normal tooth load are denoted respectively by q and P , we have $P_n = qP$ and $b \propto \sqrt{P_n}$. Then Eq. (4) becomes

$$\vartheta \propto \mu q^{3/4} P^{3/4} v_0^{1/2} \quad (5)$$

where v_0 is a peripheral speed of gear.

The calculated results for ϑ_0 (ϑ_0 represents ϑ in the case of $q=1$) are shown in Fig. 6. The values used in this example are as follows: $q=1$; $P=16$ kg/cm; $n_1=270$ rpm; $\gamma_1=8.9$ g/cm³; $c_1=0.098$ cal/°C g; $\lambda_1=0.055$ cal/cm sec °C; $E_1=11$ 000

kg/mm²; $\gamma_2=7.84$ g/cm³; $c_2=0.113$ cal/°C g; $\lambda_2=0.100$ cal/cm sec °C; $E_2=21$ 000 kg/mm²; $\mu=0.5$.

In this example we have assumed that the coefficient of sliding friction is considerably large as compared with that of hydrodynamic friction, since there is a compound friction of both the metallic and boundary lubrication frictions. Thus the value of μ is not a measured value, but a supposed one. Studying the variation of surface temperature in mesh, however, we may simply assume that μ is a constant. As the sliding speed equals zero at the pitch point, the surface temperature rise becomes zero. As the contact point moves to tip or root, the sliding speed increases. The surface temperature becomes higher in consequence. We assume $q=1$ in this example. This assumption is correct in the zone of single tooth contact, yet q varies in the contact zone of the two pairs of teeth. The state of variation changes according to the various errors of gear and the degree of tooth deflection.

Now the typical diagrams of load distribution are shown in Fig. 7 and calculating the tooth surface temperature results in the diagram of θ . In regard to the load distribution in the cases of (b), (c) and (d) shown in Fig. 7, the surface temperature

Fig. 5 Calibration of thermocouple

Fig. 6 Relation between calculated value of surface temperature and contact position

Fig. 7 Calculated examples of tooth surface temperature

reaches the highest at the tip of pinion and root of gear. Besides, the larger the rate of load distribution becomes, the higher the surface temperature grows. The surface temperature at the tip of pinion is rather lower than at the tip of gear. On the other hand, in the load distribution (a), the surface temperature becomes highest not at the tip of pinion or of wheel, but at the points a little removed from these tips. In the example (a), it is the highest at the distance of 6.7 mm from the pitch point along the line of action. In practical use, the scoring frequently occurs at such position.

The oscillogram indicates the mean temperature value of the two meshing points, the curve of θ_{mean} in Fig. 7, which is discontinuous at both ends of the zone of single tooth contact, even though the curve of θ may be continuous.

We measured the load distribution of gear tooth, rolling the gear under the load $P_n = 16$ kg. The measuring procedure was as follows. First, the wire strain gage was affixed on the tooth fillet of wheel for the purpose of measuring the fillet strain. Next, putting the thickness gage in meshing teeth, we measured both the strains in fully single tooth contact from root to tip, and in ordinary contact. Then, we could determine the load distribution by comparing both values, as shown in Fig. 8. In this figure, the hollow points present the strain in the ordinary contact, and the solid points the strain in fully single tooth contact. In practice, we should measure the dynamic load distribution. The left diagrams in Fig. 9 show the electromagnetic oscillograms of fillet strain in the following conditions: speed of pinion, 270 rpm; reduction gears, tooth load per unit width, 16 kg/cm. The right diagrams in Fig. 9 show the rate of the load distribution determined by correlation of above mentioned strain to the static strain in fully single

tooth contact. The shape of the curve nearly coincides with q in (a) and (c) shown in Fig. 7. Fig. 4 is the photograph of temperature in this case, and we can conclude that the experimental result coincides with the figure θ_{mean} in (a) and (c) calculated theoretically. Consequently, if the rate of load distribution is determined, we can calculate accurately the surface temperature of contact tooth.

Fig. 9 Rate of load distribution

Fig. 8 Measuring value of fillet strain

Reduction gears
Speed of pinion: 270 rpm

Fig. 10 Relation between thermoelectric force and tooth load

5. Relation between the surface temperature and the tooth load

Choosing the specified gear teeth so arranged as to keep a constant speed, we measured the surface temperature. The relation between the surface temperature and the tooth load is shown in Fig. 10. In the figure, the measured values shown by points coincide with the values shown

by the curve that is proportional to $P^{1/2}$. According to Blok's equation, θ is proportional to $P^{3/4}$, so that relation turns out in a straight line in Fig. 10, which differs from the experimental results. The equation has been introduced on the assumption that the tooth surface is fine and the tooth bearing is uniform along the tooth width. The tooth bearing, however, is not always uniform in practice, and the high points on one surface come in contact

Speed of pinion : 270 rpm

470 rpm

715 rpm

980 rpm

1 290 rpm

1 459 rpm

1 520 rpm

Tooth load : 16 kg/cm, Reduction gears
Fig. 11 Oscillogram

with the high points on the other surface. So it is supposed that the power of P decreases, owing to the increase of the true contact area when the applied force exceeds the elastic limit.

6. Relation between the surface temperature and the circumferential speed

When the load is constant, the tooth surface temperature increases according to an increase in circumferential speed. Fig. 11 shows the oscillogram of tooth surface temperature under the following conditions: tooth load per unit width, 16 kg/cm; finishing, MAAG gear grinding; speed change, reduction. Fig. 12 shows the surface temperature at the position where the single tooth contact is just finished. The surface temperature increases in proportion to an increase in circumferential speed, and especially in the contact zone of the two pairs of teeth, the temperature rise shown with the waves of high frequency is remarkable. Since the rate of load distribution of each tooth is variable in the contact zone of the two pairs of teeth, the variation of temperature is frequent and rapid as mentioned above.

Fig. 12 Relation between thermoelectric force and circumferential speed (1)

Fig. 13 Relation between thermoelectric force and circumferential speed (2)

Figs. 13 to 15 show the relation between the maximum thermoelectric force and the circumferential speed under the constant loads. The solid points in the figures represent the thermoelectric

Fig. 14 Relation between thermoelectric force and circumferential speed (3)

Fig. 15 Relation between thermoelectric force and circumferential speed (4)

Fig. 16 Relation between thermoelectric force and circumferential speed (5)

force of specified tooth, and the hollow points, those of any tooth. The latter points scatter somewhat due to tooth errors, e.g., pitch error, profile error and others. The plot of thermoelectric force against circumferential speed shows a linear increase of thermoelectric force according to the increasing speed in low speed range less than about 1 500 rpm, i.e. $\theta \propto v$. In high speed range, however, the increase of thermoelectric force due to the increasing speed is not observed, Blok's report introduces the following equation $\theta \propto v^{1/2}$. This relation holds good in comparatively high speed range; i.e. $bv\gamma c/\lambda \geq 20$, but in low speed range as in the present experiment, we can get the relation of $\theta \propto v$, since the heat conduction comes near the thermal stationary condition. As for the speed up gears we get the result shown in Fig. 16. In comparison with the case of reduction gears shown in Fig. 13, the surface temperature becomes higher. One of the reasons for the difference should be the dynamic load, because, as the present author has already stated, the difference of tooth load between both conditions occurs due to dynamic load⁽¹⁾.

7. Discussion on the flash temperature

In the present study the maximum temperature rise was 150°C and the average temperature rise about 80°C. The measured value, however, shows the mean temperature in the contact zone of the two pairs of teeth and so the surface temperature becomes higher at one of the contact points. In particular cases, the peak value of high frequency ripple varies considerably, and at times the value of temperature flash (see Fig. 7) is more than twice as large as the peak value of basic wave. In this test, the grease lubrication has resulted in serious scratches and some slight scoring has appeared at or near the tip of wheel tooth.

The greater part of this report is concerned with the reduction gears. Generally, the temperature rise in the speed up gears is higher than in the reduction gears.

8. Conclusion

The results obtained in this fundamental test of flash temperature are summarized as follows.

(1) The relation between the surface temperature and the contact position of tooth generally coincides with the calculated value according to Blok's theoretical equation.

(2) As the surface temperature is affected by the load distribution, it is desirable in gear designing to provide a profile modification and the means to avoid the local bearing. The mal-loading of tooth should be also avoided, from the standpoint of surface temperature.

(3) The surface temperature θ_{mean} is proportional to the circumferential speed v in low speed range, but it does not increase so much in high speed range. And the temperature θ_{mean} is proportional to $1/2$ the power of the load P , but it does not increase so much in the region of heavy loading. The designer of the gears should avoid both high dynamic load and high ratio of sliding. This can be performed by raising gear accuracy and choosing proper values of major factors of gear design.

Acknowledgement

The authors are indebted to Mr. T. Kitamura, Mr. T. Hirabayashi for their assistance in carrying out these experiments.

The authors also wish to thank the Mitsubishi Engineering & Shipbuilding Co. and the Osaka Gear Manufacturing Co. for their offering the gears for this experiment.

This investigation was subsidized by the Grant in Aid for Fundamental Scientific Research of the Ministry of Education.

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