Temperature Distribution in Teeth and Blanks of Ultra High-Speed Gears*

(1st Report, Measurement of Temperature Distribution in Teeth and Blanks)

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The purpose of this study are to ensure reliability of the power transmission gears, and to predict operating condition of high speed gears especially with regard to temperature distribution in teeth and blanks. The experiments were carried out using power circulating gear testing machine (1 100 kW) under ultra high pitch line velocity (~ 150 m/s) and temperature distribution in teeth and blanks were measured. By the experiments, the following facts are known. By increasing line velocity, the temperature of teeth increase parabolically and the distribution of bending stress become nonuniform along face width.

Key Words: Gear, Measurement, Temperature Distribution, Bending, Machine Element

1. Introduction

Recently, it is needed to increase the speed and sizes of power transmission gears to deal with a tendency of higher outputs leading to increased capacities and speeds of the high-speed turbo machines like compressors and steam or gas turbines. The ultra high-speed power transmission gear consists of large carburized, tempered and ground gears, supported by ultra high-speed journal bearings. And, its design and fabrication cover a wide variety of subjects to be studied.

Although thermal deformation due to frictional heat of the gear engagement or the bearing contact can occur even in low-speed gears if strictly

examined, it is usually assumed that there would be no such influences to tooth traces or profiles from the practical point of view. However, an accurate prediction of tooth contacts taking thermal deformations into considerations is an important issue for the design of ultra high-speed gears.

In a previous study, we already developed the prediction technique of tooth contacts taking into considerations bending of gear teeth and torsion and bending of bearing shafts⁽¹⁾, the results of which have already been applied to designs of tooth flanks for heavy loaded gear wheels in cement mills, rolling mills and construction machinery, etc., as well as for ultra high speed gears like speed reducers in steam or gas turbines and speed increasers for compressors, etc., and it has been found that a physical deformation as small as $10\text{ }\mu\text{m}$ can greatly affect the load distribution. The setting of the required tooth flank correction considering tooth contacts due to the mechanical deformation has nearly been established. However, ultra high-speed gears would predictably require a further modification of tooth flanks or surfaces added with thermal deformations, and it will be necessary to add a calculational method of thermal deformations

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to the conventional calculational method of tooth contacts taking mechanical deformations into considerations and to enable a comprehensive calculation considering the temperature distribution in teeth and blanks, thermal deformations and tooth contacts, if the operating conditions of the gear are known.

Thermal deformations of gears, especially supported by ultra high-speed bearing supports, must be theoretically verified using a basic test in order to achieve the development goal to the design technique as described above. In the present study, the authors made a high-speed test stand of a gear and its bearing support to measure temperature distribution in teeth and blanks and stress distribution in tooth root under test conditions of the delivered power ; 1 100 kW and the maximum pitch line velocity ; 150 m/s.

2. Test Stand

The following points were considered as the basic concept of the test stand.

(1) Use this test stand commonly for the gear tester and the bearing tester.

(2) Use the power circulating type which can provide a large output with a small motor.

(3) Use the ones assumed to be employed for actual gears regarding the gear configuration, the tooth profile, the material and the bearing type of the test stand.

Table 1 shows the specifications of the test stand newly made.

2.1 Structure and functions

Figure 1 (a) outlines the gear test stand consisting of the power circulating structure. This test stand consists of a test gear, a power circulating gear, a torque load device and a drive unit.

(1) Drive unit

In the driving mechanism, the motor speed is increased through a planetary gear and the motor is connected to the driving shaft through a coupling. Because the drive motor can only be compensated for its mechanical loss in the power circulating system due to the use of the power circulating gear, a motor of 370 kW D.C. was used and controlled at its speeds as required for this test with its torque maintained constant.

Table 1 Specifications of test stand

Center distance	345 mm
Circulating power	~ 1840 kW
Pinion speed	~ 24,000 rpm
Drive machine	370 kW D.C. motor
Test gear unit	Speed increaser

(2) Power circulating gear and test gear

For the gear, tempered material of carburized SNC815 was finished by grinding, and for the bearings, anti-vibration performance was also considered by choosing the tilting pad type for both journal and thrust bearings at the high-speed side, and the two lobe type for the journal bearing and the tapered type for the thrust bearing at the low-speed side. Furthermore, the power circulating gears are coupled to the test gear shafts through the Bendix type diaphragms and one ends of the test gear shafts can be fitted with slip rings designed to take out the lead wires for measurements of the test gear temperatures and strains at the tooth root.

(3) Torque load device

The torque load device was designed to provide mechanical loading during the gear operation. When the power circulating gear is pushed and moved toward the gear shaft using the worm gear and the power output screw, the shaft is turned round for the torsional angle of the power circulating gear as a helical gear and its torque is transmitted to the power circulating system. The load torque was designed to be steplessly changed to 2 000 N·m at maximum by adjusting the strokes of the power output screw.

(4) Lubrication system

Oil pumped by an electrically driven pump is supplied to tooth surfaces through an oil filter, a relief valve, a flow regulator and an oil flow meter. Oil is fed

Fig. 1(a) Schematic diagram of test stand

Fig. 1(b) Lubricating nozzle

Table 2 Specifications of the gear and the bearing

	Pinion	Wheel
Number of teeth	28	83
Module	6	
Normal pressure angle, degree	20	
Helical angle, degree	15.161	
P. C. D., mm	174.054	515.946
Face width, mm	244	246
Material and heat treatment	Carburized Ni · Cr Alloy steel	
Finishing of teeth	Ground	
Maximum speed, rpm	24,158	8,150
Maximum pitch line velocity, m/s	220	
Type of journal bearing	Tilting pad type	Two lobe type
Peripheral velocity of journal, m/s	101	38.4
Bearing pressure, MPa	110	80

Fig. 2 Tooth trace of pinion

through a nozzle diameter of 3.6 mm at an oil pressure of about 3 kgf/cm² from the contacting side. Location of the lubricating nozzles is shown in Fig. 1(b).

2.2 Specifications of the power circulating and test gears

The specifications of both gears were designed to keep them from damage even during their operation at the maximum pitch line velocity of 220 m/s. Principal particulars of the test gear are listed in Table 2. The same specifications also apply to the power circulating gear with the only difference being its face width.

The test gear was provided with a pinion as shown in Fig. 2 in consideration of deflection of the teeth and torsion and deflection of the shaft. To confirm accuracies of the shaft, the tooth profile, the tooth trace and the bearing box, the tooth contact was first provided in unmodified condition; (A) over 20% or more of the tooth depth for over 90% of the tooth trace length and (B) over 40% or more along the direction of the tooth depth for 50% or more of the tooth trace length of (A).

Figure 3 shows bore positions for measuring temperatures at the gear and stresses at the tooth root. Furthermore, stress distribution at the tooth root of the gear wheel was measured so as to check for

Fig. 3 Measuring point of temperature

Fig. 4 The relation between pitch line velocity and temperature

unbalanced tooth contact along the tooth trace, and a strain gauge was fitted at the tooth root of the tooth engagement side opposite to the gear wheel (compression side).

For the temperature measurement of the tooth surface, a sheathed thermocouple was pre-inserted into an inner hole bored at the gear blank and designed to be retained at the position 1.4 mm from the tooth surface. This position was measured by inserting a wire gauge into the bored hole.

3. Test and Its Results

3.1 Test procedure

For the operational method, the speed was increased to its required limit under no-load condition and then, the load torque was gradually increased. The varied test conditions were the peripheral velocity of the test gear, the load torque and the oil feed,

and the temperature distribution under saturation of temperature in certain test conditions was recorded. Test run time in one condition took approximately 15 minutes.

Also, the lubricating oil used for this testing was ISO VG turbine oil of the mineral system, and the lubricating temperature was adjusted to $30^{\circ}\text{C} \pm 2^{\circ}\text{C}$ for the bearings and to $50^{\circ}\text{C} \pm 2^{\circ}\text{C}$ for the gear.

3.2 Measurement of temperature distribution (Temperature changes due to pitch line velocity and load torque)

Trends of temperature changes in sections ①~⑤ in varied pitch line velocities are shown in Fig. 4. Because the oil feed temperature to the bearing was

Fig. 5 Temperature distribution along face width (oil flow rate of gear : 10 l/min, oil flow rate of bearing : 100 l/min)

as low as 30°C through its temperature to the gear was 50°C, the drain oil from the bearing flowed to the gear side, thereby decreasing significantly the temperature at the low-speed side. Increasing the pitch line velocity raises the temperature of each part, nearly following curves of the second order, which shows that the pitch line velocity dependence of the temperature is noticeable.

Of the tooth surface temperatures of the pinion, temperature of ②-B and ③-B near the surface center was highest whereas the temperature of ⑤-B was lowest. When the temperature of the pinion proper (④-F) was measured, it was 10~20°C lower than the tooth surface temperature.

Next, the measured temperature distributions along the face width are shown in Fig. 5. The temperature of the pinion tooth surface was higher than the temperature of the wheel tooth surface and the temperature rose with increasing pitch line velocity. Also, asymmetry in the temperature distribution along the face width became more marked as the operational speed was increased. The asymmetric temperature pattern corresponds to the moving direction of the simultaneous contact line decided by the torsional direction along the face width, which shows that the temperature tends to rise toward the moving direction. Subsequently, peak temperatures along the face width were summarized. Figure 6 shows their relation with the pitch line velocity. The temperature in both the pinion and the gear wheel increases with increasing pitch line velocity whereas there is no difference in the temperature even with smaller loaded torque as shown by the dotted lines.

Subsequently, our examination proceeded to the torque load dependence. Figure 7 shows temperature changes at each part due to an increase of the torque load at the pitch line velocity 70 m/s (oil flow rate; 100 l/min to bearing, 10 l/min to gear). Even when

Fig. 6 The relation between pitch line velocity and subsurface temperature

the loading torque was changed from 200 N·m to its 10 times value of 2 000 N·m, the temperature rises at the relevant parts were as small as 5~6°C, which shows

Fig. 7 The relation between torque of wheel and temperature (pitch line velocity 70 m/s)

Fig. 8 The relation between torque of wheel and temperature (pitch line velocity 150 m/s)

Fig. 9 The relation between torque of wheel and temperature (oil flow rate of gear: 8 l/min, oil flow rate of bearing: 50 l/min)

Fig. 10 (a) Measured temperature distribution and tooth root stress of wheel (pitch line velocity 150 m/s)

Fig. 10 (b) Measured temperature distribution and tooth root stress of wheel (pitch line velocity 90 m/s)

that the effect of the load changes can be small. Figure 8 shows the results at the pitch line velocity 150 m/s (oil flow rate; 100 l/min to bearing, 10 l/min to gear), and the effect of the torque on the temperature of the tooth subsurface is also small as shown in the case of the pitch line velocity 70 m/s. Figure 9

Fig. 11 The relation between temperature and tooth face width

shows a measured example of smaller oil feed rates and increased degrees of the temperature rises. There was hardly a difference in the gradient of the temperature rises to the torque load increase even with an increase of the pitch line velocity.

3.3 Temperature distribution within the gear

A measured example of the temperature distribution within the wheel gear is shown in Fig. 10. This figure refers to the temperature distribution obtained from the measured temperatures. Figure 10(a) shows the measured results at 150 m/s, and it was confirmed that the temperature at the near the tooth surface center tended to rise slightly though the tooth trace crowning was omitted. Although the tooth crest shows 35 mm of a linear relieving, there is no sudden change of the temperature distribution. The measured tooth root stress (calculated from the measured bending stress) in this operational condition is also shown together in the figure, and it is a bending stress distribution approximately corresponding to the measured temperature distribution. Figure 10(b) refers to the case of the pitch line velocity 90 m/s. Because the pitch line velocity was reduced, the value of the temperature rise decreased and the value of the tooth root bending stress at the tooth surface center was smaller than its value in the case of the pitch line velocity 150 m/s. The values with □ in the figure indicate the measured values of the pinion nearly in the same positions, and there was a temperature difference of approximately 20°C at maximum.

3.4 Effect of the tooth face width

Figure 11 shows an example of the gear operations using the tooth face width narrowed to 18 mm and the same torque of the total torque loaded on the face width 244 mm and the gear. Also, the same oil

flow rates were used regardless of the difference in the face width. Because the narrower face width showed smaller temperature rises although the same transmission power was used, this can indicate that there should be a temperature rise caused by some added heating source due to the lengthy tooth trace in addition to the tooth surface friction.

4. Conclusions

In order to investigate the lubricating characteristics of gears and bearings and the temperature (thermal deformation) distribution in teeth and blanks in the ultra high-speed region, the authors made a high-speed test stand of the gear and bearings to measure the temperature distribution and the tooth root stress distribution in the test gear under the test conditions of the delivered power 1100 kW and the pitch line velocity up to 150 m/s and the following results were obtained from this testing.

(1) The pitch line velocity dependence of the

gear temperature is prominent and the temperature increases, nearly following the curve of the second order; however, it was found that the effect of the loaded torque on the gear temperature can be small.

(2) It was verified that the tooth root bending stress at the center of the tooth face width rises with an increase of the pitch line velocity and thus, it was found that the tooth contact changes due to thermal deformation.

(3) Comparison of the difference in the temperature rises due to different tooth face widths showed that there is a possibility of a temperature rise caused by an added heating source due to the lengthy tooth trace in addition to the tooth surface friction.

References

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