Evaluation of Surface Fatigue Strength Based on Surface Temperature*
(Surface Temperature Calculation for Rolling-Sliding Contact)

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Surface temperature is considered to be an integrated index that is dependent on not only the load and the dimensions at the contact point but also the sliding velocity, rolling velocity, surface roughness, and lubrication conditions. Therefore, the surface durability of rollers and gears can be evaluated more exactly and simply by the use of surface temperature rather than Hertzian stress. In this research, surface temperatures of rollers under different rolling and sliding conditions are measured using a thermocouple. The effects of load \( P \), mean velocity \( V_m \) and sliding velocity \( V_s \) on surface temperature are clarified. An experimental formula, which expresses the linear relationship between surface temperature and the \( P^{0.86} V_s^{0.83} \) value, is used to determine surface temperature. By comparing calculated and measured temperature on the tooth surface of a gear, this formula is confirmed to be applicable for gear tooth surface temperature calculation.

Key Words: Surface Temperature, Surface Durability, Surface Fatigue, Rolling-sliding Contact, Roller, Gear, Thermocouple, Friction Loss, Lubrication

1. Introduction

Surface fatigue strength of machine elements under a rolling-sliding condition, such as gears and traction drive rollers, is evaluated with so the called \( S-N \) curves which show the relationship between Hertzian stress and the number of load cycles. Because the Hertzian stress is dependent only on the dimensions at the contact point, the applied load and the material properties, and is independent of the surface quality, rolling velocity, sliding velocity and lubricating conditions, all of which are very important factors affecting surface fatigue strength or fatigue life\(^{11} \), many \( S-N \) curves are obtained under different operating conditions for one type of roller or gear. The situation that fatigue strength cannot be evaluated by a single \( S-N \) curve makes it difficult to determine surface fatigue strength exactly in fatigue strength design. In practical use, fatigue strength is simulated by the modification of typical experimental results with many empirical factors which make the simulated fatigue strength different from the actual value. In the design of power transmission devices under severe conditions such as heavy load and high speed in addition to compact size, surface fatigue strength must be evaluated more precisely.

Based on the considerations that surface temperature is an integrated index that is dependent on not only the load and dimensions at the contact point but also the sliding velocity, rolling velocity, surface roughness, lubrication conditions, and material properties such as surface hardness which changes due to temperature rise. Therefore, surface fatigue strength of machine elements such as gears and traction drive rollers can be evaluated more exactly and simply by the use of surface temperature rather than by Hertzian stress. One of the authors has previously shown a general relationship between flash temperature and fatigue life of gears, which indicates the potential of the new surface fatigue strength evaluation method based on surface temperature\(^{20} \).

In this study, we attempt to evaluate surface fatigue strength based on the relationship between
surface temperature and fatigue life. Although there are many studies that deal with the method of surface temperature calculation for rollers and gears, a generally authorized calculation method is not in use yet. In this research, surface temperatures were measured in two-rollers tests, and an experimental formula for the surface temperature calculation of rollers is presented, which is confirmed to be applicable as well for the surface temperature calculation of gears.

2. Surface Temperature Measurement

2.1 Test rollers and experimental conditions

The dimensions of the test rollers are shown in Fig. 1. The contact width is 8 mm. The test rollers were made of low-alloy steel JIS SCM435, quenched and tempered to have about 350HBS hardness. After finishing grinding, the surface roughness of the rollers was \(4.5 - 5.5 \mu m_R_{\text{max}}\). A backup-roller-type two-rollers fatigue test rig was used in the tests. The rotation velocity ratio of the high-speed shaft to the low-speed shaft is 1.21. Two types of high-speed roller of diameters 68.0 and 72.5 mm were used for different sliding velocities in temperature measurements.

2.2 Surface temperature measurement method

In the previous studies of temperature measurement, thermocouples are usually bonded below the surface of a roller or a gear tooth, in which additional machining to set the thermocouples and expensive slipping rings are necessary. In those methods, the measured temperatures are the mean values of the measuring point in one cycle, and periodic changes of surface temperature in and out of contact are not clear. In order to avoid a large temperature rise under heavy load and high-speed conditions, the cooling effect of a lubricant during the period out of contact is as very important as lubrication. To realize a good cooling effect, knowledge of the periodic changes of surface temperature is important. In this study, the method shown in Fig. 2 was used to measure surface temperature. A chromel-alumel (type K) thermocouple with a wire diameter of 0.254 mm was set in such a way that it lightly touched the surface of the low-speed roller. The temperature distribution on the roller circumference was obtained by moving the thermocouple by means of a pulley. Temperature measurements were performed about 50 min after the start of operation when surface temperature become constant. The position of measurement is expressed by angle \(\theta\) shown in Fig. 2.

When the thermocouple strongly touches the roller surface, the friction between the thermocouple and the roller surface will lead to a large temperature rise, and the measured temperature cannot be treated as the surface temperature of the roller. Because the resistance between the thermocouple and the roller surface rose to about 300 ohm from zero after the roller started operating, it can be said that the thermocouple did not touch or very lightly touched on the roller surface during operation. Based on the findings that temperature signals were very stable and the measured temperature without any load was only 2 – 3°C higher than the supplied lubricant temperature, it can be said that an oil film was formed that separated the thermocouple from the roller surface and the friction between the thermocouple and the oil film could be neglected. Due to the presence of the oil film
between the thermocouple and roller surface, the measured temperature is not the exact temperature of the roller surface. However, considering that the oil film is very thin and the response of the thermocouple is not sufficiently fast to detect the flash temperature because of heat capacity, the measured temperature can be approximated as the bulk temperature of the roller surface; this is called surface temperature hereafter.

3. Surface Temperature of Test Roller

The surface temperature distributions of the test roller are shown in Fig. 3. Measurements were performed over the range of 10 deg, the nearest position of the thermocouple to the contact area, to 330 deg, where lubricant dropped to the thermocouple. Because of the presence of lubricant on the roller, temperatures in the range of 120 deg to 270 deg were not measured correctly. On the basis of the surface temperature distribution shown in Fig. 3, it is clarified that after a quick rise over a small range of position, the surface temperature became nearly constant. Owing to the effects of the lubricant at the contact area, the surface temperature distribution in the region of small \( e \) was not stable. The temperature measured at 300 deg, where stable surface temperature distribution was obtained in all tests, was used as the surface temperature of the roller.

Surface temperatures of the low-speed roller at different Hertzian stresses and oil temperatures are shown in Fig. 4. Surface temperature rise with an increase in rotation velocity and Hertzian stress is confirmed, and the change of oil temperature from 40 \( ^\circ C \) to 20 \( ^\circ C \) will reduce surface temperature by about 10 \( ^\circ C \) at rotation velocities ranging from 1000 rpm to 2000 rpm. However, above 2500 rpm, surface temperature increased, the reason for which is considered to be the changes of oil properties. This will be clarified in future research.

4. Surface Temperature Calculation

There are many studies on surface temperature calculation for machine elements under the rolling-sliding condition. The methods presented in those studies can be briefly divided into two types: theoretical and experimental methods. In the former, the heat production mechanism, the movement of heat source and the conduction of heat are considered in temperature calculations\(^{(9)}-(^{11})\). In the latter, empirical formulas are developed based on temperature measurement results\(^{(3),(4),(6)}\). However, the theoretical methods are difficult to use practically because contact and lubrication conditions differ from the actual conditions, and the experimental methods are only applicable to the same conditions under which the empirical formulas are developed. In what follows, a surface temperature calculation method is presented, which is useful for the determination of the surface temperature of rolling-sliding contact elements such as gears and rollers.

4.1 Relationship between surface temperature and friction loss in contact area

The surface temperature of rolling-sliding contact elements is thought to comprise the instantaneous rise in the contact region, which is called flash temperature, and the bulk temperature outside the contact region. Nadano\(^{(10)}\) calculated flash temperature in the contact region based on the assumption that friction loss (heating source) in the contact region is proportional to Hertzian stress. The well known Blok formula is used for the calculation of the maximum temperature in the contact region. However, there is
no general method for bulk temperature calculation. In this study, bulk temperature is considered to be a result of the temperature rise and the cooling effects in and out of the contact region, and the temperature rise is considered to depend on the total friction loss absorbed by contact elements. Thus, surface temperature (bulk temperature) $T$ can be related to friction loss $W_r$ as follows:

$$T \propto W_r, \quad W_r = \mu P |V_s|$$

where $W_r$ is friction loss, $\mu$ is friction coefficient, $P$ is load at unit contact length, $V_s = V_1 - V_2$, $V_1$, $V_2$: tangential speed of high-speed and low-speed roller) is sliding velocity.

4.2 Changes of surface temperature with load and sliding velocity

Based on the assumption that friction coefficient is a constant independent of load and sliding velocity, the total friction loss in the contact region is proportional to $P |V_s|$. Figure 5 shows the results of surface temperature measurement under different sliding ratios, sliding velocities and loads. White circles in Fig. 5 represent the temperatures of one pair of the low-speed and high-speed rollers whose diameters are 68.0 mm and the sliding ratio $S_1 = (V_2 - V_1)/V_2$ is $-21\%$. Solid circles in Fig. 5 represent the temperatures of one pair of the low-speed and high-speed rollers whose diameters are 68.0 and 72.5 mm, respectively, and the sliding ratio $S_2$ is $-29\%$. Although surface temperature increases with an increase in $P |V_s|$, the temperature variation at a constant $P |V_s|$ value is as great as 40°C. The variation is considered to result from disregarding of the change of the friction coefficient under different operating conditions and assuming that the load and sliding velocity have the same contribution to surface temperature. In fact, friction coefficient changes markedly under different operating conditions, and factors such as load, sliding velocity and mean velocity, all of which influence friction coefficient, should be considered in surface temperature calculation. The effects of load $P$ on surface temperature are investigated at constant $P |V_s|$, the results are shown in Fig. 6. If load $P$ and sliding velocity $V_s$ have the same contribution to surface temperature, surface temperature will be a constant value. However, based on the test results that surface temperature rises with an increase in load, it is confirmed that load $P$ and sliding velocity $V_s$ have different contributions to surface temperature. The effects of sliding velocity $V_s$ and mean tangential velocity $V_m = (V_1 + V_2)/2$ on surface temperature were also investigated. Surface temperature rises with an increase in sliding velocity at constant $P |V_s|$, as shown in Fig. 7, and decreases with an increase in mean tangential velocity at constant $P |V_s|$, as shown in Fig. 8.

4.3 Evaluation of the surface temperature

Temperature rise is a very complicated phenomenon that is influenced by friction coefficient, lubrication and operating conditions. O’Donoghue and Cameron(12) clarified the effects of operating conditions on friction coefficient, lubricant film thickness and roller surface temperature instead of formulating a method to calculate surface temperature. Based on the results of surface temperature measurement, the surface temperature calculation method for the roller in rolling-sliding contact is presented as an exponential function referring to the method of film thickness calculation.

$$T = DP^2 V_s^2 V_m^2 + T_0$$
Fig. 8 Effect of mean tangential velocity on surface temperature

\[ V_m = \frac{(V_1 + V_2)}{2} \]  

where \( P \) is load at unit contact length (N/m), \( V_s, V_m, V_t, V_s \) are sliding velocity, mean velocity, tangential velocity of the high-speed roller and tangential velocity of the low-speed roller, respectively (m/sec), and \( a, b, c \) are constants. \( D \) is a gradient factor. \( T_s \) is surface temperature when load \( P \) is zero, which may be substituted by the inlet oil temperature.

Constants \( a, b, c \) and factor \( D \) are determined as follows by the method of least squares applying the temperature measurement results to Eqs. (2) - (4).

\[ a = 0.86, \ b = 1.31, \ c = -0.83, \ D = 1.50 \times 10^{-3} \]  

Figure 9 shows the correlation between measured and calculated temperature using Eqs. (2) - (5). The maximum difference between measured and calculated temperature is only about 10°C. Therefore, surface temperature is said to be proportional to load to a power of 0.86, to sliding velocity to a power of 1.31 and to mean velocity to a power of −0.83. The relationship between surface temperature and load obtained in this study is similar to that mentioned in Refs. (3) and (4) in which surface temperature is proportional to load to a power of 0.75 - 0.90.

4.4 Application of surface temperature calculation formula

For constants \( a, b, c \) and factor \( D \) in Eq. (5) are determined based on the measurements of this study, the applicability of the Eqs. (2) - (5) to other rollers or gears operating under the rolling-sliding condition should be investigated. Some of the published surface temperature measurement results are used to discuss the applicability of the presented formula.

4.4.1 Applicability of the presented formula to other types of rollers

Nakamura et al. measured roller surface temperature by contacting the surface with a thermocouple at sliding ratio of −8~−25%, and Suzuki et al. measured roller surface temperature by setting thermocouples near the roller surface at sliding ratio of −31~+24%. The relationships between their measured temperature and the \( P^{0.86} V_s^{1.31} V_m^{-0.83} \) value are shown in Fig. 10. Solid lines are obtained by the method of least squares. It is clear that the temperatures measured by Nakamura et al. and Suzuki et al. are proportional to the \( P^{0.86} V_s^{1.31} V_m^{-0.83} \) value. Therefore, if the gradient factor \( D \) is determined, the surface temperature can be evaluated correctly and accurately using Eq. (2).

The gradient factor \( D \) is considered to be dependent on the temperature measurement method, the lubrication conditions, the material and thermal capacities as well as the surface roughness of rollers. Therefore, it changes depending on the contact elements. Because of this, it is necessary to determine \( D \) by tests.

4.4.2 Application to evaluation of gear tooth surface temperature

Deng et al. measured the temperature of a gear tooth by setting thermocouples beneath the tooth surface. The relationship...
between the measured temperature of the gear tooth and the \( P^{0.86} V_{c}^{1.31} V_{w}^{-0.83} \) value is shown in Fig. 11. Surface temperature of the gear tooth is also proportional to the \( P^{0.86} V_{c}^{1.31} V_{w}^{-0.83} \) value, but the gradient factor \( D \) changes at different measurement positions because of such differences as the heat diffusion conditions, the thermal capacity as well as the thermocouple setting depth. Based on the results shown in Fig. 11, the formula presented here is applicable to gear tooth surface temperature evaluation, in which the effects of load, sliding velocity, rolling velocity on the surface temperature of gear tooth are clarified.

5. Conclusions

To evaluate the surface durability of rolling-sliding contact elements such as gears and traction drive rollers in terms of surface temperature, surface temperature measurements were performed on rollers, and the effects of load, sliding velocity and mean velocity on surface temperature were evaluated. Based on the measurement results, an experimental formula was presented for surface temperature calculation. The applicability of the formula to other types of rollers and gears was investigated using previously published surface temperature measurement results. The main results are as follows:

(1) The rise of surface temperature with an increase in load and sliding velocity is confirmed. The effect of oil temperature on surface temperature is likewise clarified.

(2) A linear relationship between surface temperature and the \( P^{0.86} V_{c}^{1.31} V_{w}^{-0.83} \) value is obtained in which the effects of load, sliding velocity and mean velocity are evaluated precisely, and a surface temperature calculation formula is presented.

(3) The applicability of the presented formula to other types of rollers and gears is verified using previously published results.

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References