

Improvement of Temperature Prediction Method for Traction Contact

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ABSTRACT

This report proposes a method of improving the temperature prediction model for traction drive contact portion in order to improve prediction accuracy of the maximum traction coefficient, and then describes verification of this method.

In our previous report, a method of estimating the maximum traction coefficient by expressing conditions inside the contact ellipse using a simple combination of viscosity and plasticity was proposed. For the rise in oil film temperature, a calculation model is used that considers maximum temperature to be the typical value. Furthermore, a thin film temperature sensor technology was developed to directly measure the temperature of traction contact of a four-roller experimental apparatus and a variator in an actual transmission, and its validity was confirmed. When measured values of the traction coefficient μ were compared with calculated ones, it was learned that μ estimation accuracy decreased under certain low-contact-pressure and high-sliding conditions. The factor that caused μ estimation accuracy to decrease under these load conditions was presumed to be an error in oil film temperature estimation. Therefore, by partially improving the temperature prediction model, that factor was identified and examined.

In the end, the maximum traction coefficient was estimated using the new calculation model and compared with the measured values.

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1. INTRODUCTION

Traction-drive has been developed as, for example, an element of a continuously variable transmission (CVT) for automobiles. An oil film is constantly formed in the contact between rolling elements that transmit power to each other. This oil film is solidified and sheared at a GPa-level contact pressure, and its shearing properties are affected by the heat-generated temperature rise itself. That is, the temperature in the contact area can be considered the most important factor in estimating traction transmission force. With the goal of improving estimation accuracy for the maximum traction coefficient, the authors have been working to estimate and measure the temperature rise in the contact area using a thin film temperature sensor [1, 2]. One of the calculation methods that precisely handles temperature rise in oil film is Thermal- Elasto Hydrodynamic Lubrication (T-EHL) [3], which is a method that couples an energy equation with normal EHL analysis. However, this numerical analysis requires such advanced and large-scale calculation that the calculation method used for convergence could become a research subject in itself. In CVT for automobiles, sequentially estimating the maximum traction coefficient under driving conditions and properly controlling the

normal force at the contact improves power transmission efficiency. However, incorporating large-scale calculation into such sequential control is certainly impossible. Therefore, it is necessary to build a model with the goal of understanding the physical characteristics of μ and expressing μ in as simple calculation as possible while retaining its characteristics.

Through a traction coefficient estimation method using a calculation model suggested in the past [1], estimation accuracy was improved by combining the viscosity and plasticity regions of an oil film inside the contact ellipse in a μ measurement test using a four-roller experimental apparatus. However, under certain conditions, the change slope of μ relative to the slip ratio still differed from measured values (See in [Figure 1](#)). That is, μ estimation accuracy was observed to decrease as the heat generated by shearing inside the oil film increased. It seems that the physical characteristics of μ necessary under certain conditions could not be sufficiently expressed as the model was simplified. The slip ratio in the contact of a typical variator unit is not large, that is under 1%. However, as the slip

increment of the rotating component called “spin velocity” is geometrically added, there was concern that similar decline in μ estimation accuracy might occur as temperature rises.

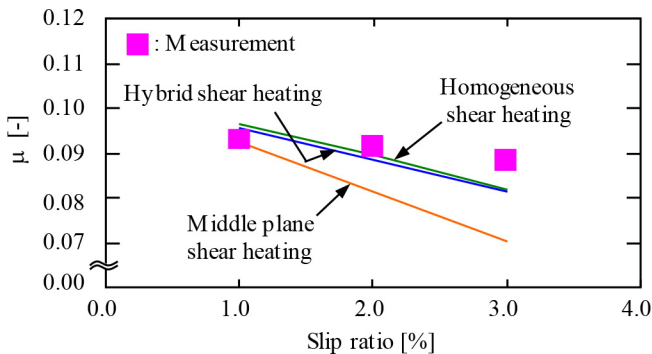


Figure 1. Comparison between measurement and μ calculation with various shear heating models (Four-roller experimental apparatus data, $N_{in}=1,910\text{rpm}$; $U=10\text{ m/sec}$, $P_{hmax}=1.0\text{ GPa}$, and $\theta_{lub}=323\text{ K}$) [1]

The conditions that tend to lower estimation accuracy are relatively low contact pressure and high oil film temperature. The possible cause for the decline is presumed to be error related to the shearing stress generated in the viscous region, which is especially sensitive to temperature. Since the authors' previous temperature estimation model was one-dimensional (calculated only the maximum temperature rise when the heat source passes in the rolling direction of the contact), there were some areas in which the temperature was not accurately expressed compared to actual temperature distribution inside the contact ellipse. Therefore, the objective of this study was to clearly define the physical characteristics of the accuracy-lowering factors, in addition to improving these factors and the method of calculating estimated temperature. A new method for expressing temperature rise is used to derive these factors and verify their effects on μ estimation accuracy.

2. IMPROVEMENT IN TEMPERATURE CALCULATION MODEL

2.1. Rheological Model and Conventional Calculation Methods

Figure 2 shows an overview of the flow for estimating the traction coefficient when the velocity of the rolling elements is controlled. What is especially important in this flow is the part that determines traction force (tangential force) by integrating the shear stress that occurs inside the contact ellipse. Furthermore, since heating and temperature rise caused by shearing of the oil film affect the generated stress value, temperature estimation becomes the key in model calculation. Film shearing, which generates stress under high pressure, is generally categorized as viscous, elastic, or plastic regions. Estimations of maximum traction coefficient require the use of comparatively high shear rate regions, which means that elastic terms can frequently be ignored. Consequently, the previous method proposed attempts to express the rheological model using viscosity and plasticity only [1].

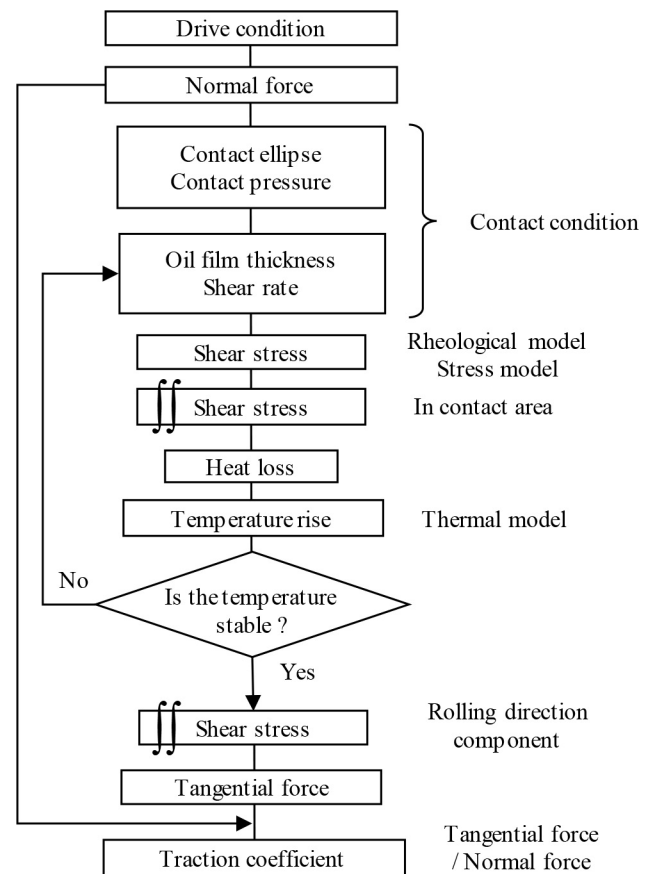


Figure 2. Calculation flowchart for prediction of traction coefficient

As a non-linear model for viscous flow, Eyring's equation (1) was adopted. For the rheological parameters η_p and τ_0 in the equation, the estimation equations [4, 5] proposed by Hata et al. were adopted.

$$\dot{\gamma} = \frac{\tau_0}{\eta_p} \sinh\left(\frac{\tau}{\tau_0}\right) \quad (1)$$

In the plastic region, oil is considered to have solidified through glass transition. Plastic flow is treated as shearing destruction of a solid and generated stress is defined as limiting shear stress τ_L . For τ_L , empirical formula (2), which has been suggested in the past [1], is used. This was derived based on high-speed, four-roller test data obtained using a synthetic naphthene-based traction oil developed by Idemitsu Kosan Co., Ltd.

$$\tau_L = (0.1518 - 0.0001577 \cdot \theta_f) P + 0.01459 \quad (2)$$

The area of a rectangle with sides equivalent to the major and minor diameters of the contact ellipse was discretized into 50×50 meshes. Either viscosity or plasticity stress, whichever was smaller, was selected from the two rheological models (1) and (2) for each mesh in accordance with local conditions (such as the surface contact pressure inside the ellipse, oil film temperature, shear rate, and the like) as

shown in Figure 3. The total integrated rolling direction component of shearing force generated inside the meshes was defined as the traction force. Power that cannot be transmitted in the rolling direction between contact surfaces forms heating loss. The contact pressure has a Hertzian distribution and the EHL oil film thickness uses the central film thickness of Hamrock and Dowson's formulae. This thickness is assumed to be uniform inside the contact ellipse. Shear rate is calculated using this film thickness.

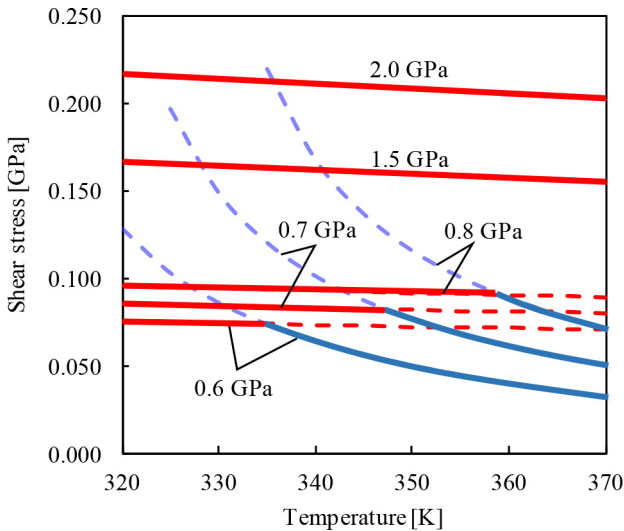


Figure 3. Transition between viscous and plastic regions (Straight line: Limiting shear stress model in plastic region; curved line: Eyring model in viscous region. Shear rate of Eyring model is 1.0×10^5 1/sec)

An estimation of contact area temperature requires a calculation of heating loss and creation of a model to calculate the temperature rise caused by the loss. The oil film heating loss was calculated by the rheological models proposed earlier. Equation (3) shows the component terms of the oil film temperature θ_f .

$$\theta_f = \theta_0 + \Delta\theta_{inlet} + \Delta\theta_s + \Delta\theta_f \quad (3)$$

For details on the individual terms, refer to an earlier report [1]. θ_0 is the rolling element temperature. The temperature rise at the contact film inlet zone $\Delta\theta_{inlet}$ uses Gohar's equation [6]. The maximum temperature rise of the rolling surface inside contact area $\Delta\theta_s$ uses Carslaw and Jaeger's correction equation [7] for the rise in temperature due to a moving heat source. With respect to maximum oil film temperature rise $\Delta\theta_f$, two types of shear heating were studied using the heating and thermal conduction models: middle plane shear heating in the plastic region and homogeneous shear heating in the viscous region. As a typical value of oil film temperature to be used for calculating the traction coefficient, the maximum value of temperature distribution in the film thickness direction was used. This is because the shearing stress that can occur when oil film is being sheared is determined by the weakest area in the film thickness, and is considered in this case to correspond to maximum temperature position.

In the test using the thin-film sensor, the insulation layer formed on only one side of the rolling elements in contact [1, 2] (Figure 4). The thickness of the insulation layer used in this study was around 8 μm ,

which was empirically enough to insulate the electricity. This is a state in which a different material is added to the rolling surface, and was expressed as a correction term for $\Delta\theta_s$ using equation proposed by Rashid and Seireg [8]. The rolling contact surface temperature rise $\Delta\theta_s$ in Eq. (3) calculates only the maximum temperature rise caused by integrated heating loss within the range of contact length of the moving heat source (in this paper, this refers to contact length in the rolling direction). By integrating the heating loss in each of 50 equal meshes over the width of the ellipse, the one-dimensional temperature distribution in the contact width direction can be calculated.

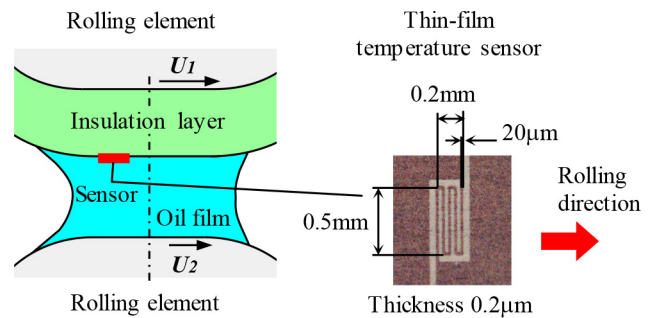


Figure 4. Contact film cross section view and thin-film temperature sensor

On the two rolling surfaces facing each other across the oil film, the amount of heat distributed from the heat generated area of the oil film differs depending on whether an insulation layer is present. This is due to the insulation layer having low thermal conductivity compared to the base metal. The heat distribution is defined by using the rate of heat flow α , and coupling the equations for the two rolling contact surfaces temperature rise $\Delta\theta_s$ with the equation for the oil film temperature rise $\Delta\theta_f$. The temperature calculation convergence value can be determined as the state in which the amount of heat generated by temperature-dependent shear stress comes to equilibrium.

What was described above is the model that matches with the state of temperature measurement. The issue dealt with in this model is the part that corresponds to $\Delta\theta_s$. The results that have been analyzed so far show that because the oil film, which is the heat source, is itself extremely thin and has a small heat capacity, most generated heat is quickly transferred to the rolling surfaces. In other words, oil film temperature is determined primarily by thermal conductivity and thickness of the oil film, and its absolute value is considered to be greatly affected by the temperature rise on the rolling surfaces. Both the correction equation by Carslaw & Jaeger and the correction term by Rashid & Seireg, which was previously used, determine the maximum temperature rise as a constant value in the rolling direction, even though different equations for oil film temperature rise are used in the viscous and plastic regions. Therefore, when the temperature distribution inside the contact ellipse is examined, it can easily be presumed that the temperature is overestimated in some areas. This temperature estimation error is considered to be the cause behind the fall in the μ estimation accuracy that occurred as the amount of heat generated by shearing increased inside the oil film, as shown in Figure 1. Therefore, this will be verified by introducing a new model for analyzing rolling surface temperature rise in each mesh when the heat source is passing through. When the equation for determining $\Delta\theta_s$ is changed to the mesh unit, the equations can no longer be

coupled accurately, and the previous method cannot be used to determine the rate of heat flow α or oil film temperature rise $\Delta\theta_f$. Therefore, the previous calculation convergence result was analyzed and partially adopted. The one-dimensional distribution of the rate of heat flow α does not depend on absolute value of the temperature but is affected by the thermal conductivity of the oil film. Since this thermal conductivity K_f is dependent on pressure [9], it was expanded into two-dimensional distribution while retaining this pressure dependence. Oil film temperature rise $\Delta\theta_f$ depends on the thickness and thermal conductivity of the oil film, but is nearly proportional to rolling surface temperature $\Delta\theta_s$ when film thickness is constant. Therefore, the proportionality coefficient ($\Delta\theta_f/\Delta\theta_s$) of the conventional model under each testing condition was first determined and then new $\Delta\theta_f$ was calculated using $\Delta\theta_s$ of the new model.

2.2. New Temperature Calculation Model

Unlike with a conventional maximum temperature model, to determine temperature rise distribution on the rolling surfaces inside the oil film in the rolling direction, it is necessary to calculate temperature rise on a mesh unit basis. As a methodology, it is also possible to carry out numerical analysis using FEM. However, since a method for analyzing temperature rise in the area where the moving heat source is passing through has been proposed [10, 11], this method is utilized in this study to carry out numerical calculations that are as simple as possible.

Figure 5 shows how a square-pillar-shaped, distributed heat source, whose heat quantity distribution density is uniform at q_0 within the heat source region, moves on the x-y plane (surface) at velocity V . By assigning the two sides of this square pillar to a mesh and combining a variety of small and large square-pillar-shaped heat sources, it would be possible to express any desired heat source distribution (Refer to Figure 6). Furthermore, it is also possible to obtain a more detailed temperature distribution by using analysis solutions.

However, in actual calculations, because temperature rises on a small mesh unit basis were superimposed, the temperature within the mesh was presumed constant.

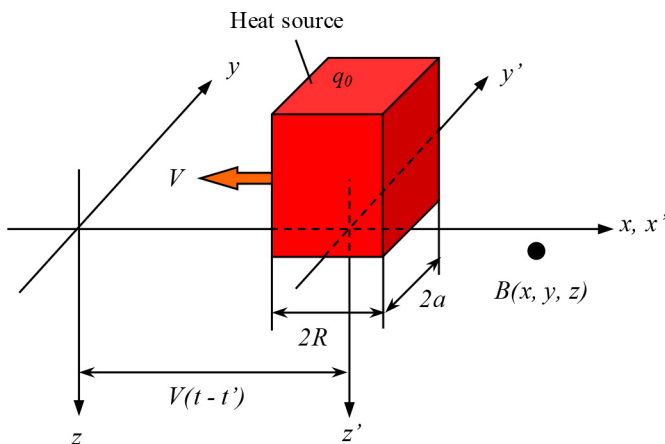


Figure 5. Moving heat source and definition of axes

The following assumptions were made for the initial condition and boundary condition:

1. There is no heat dissipation from the surface.
2. The heat source is steady and its motion continues for a sufficiently long period.
3. Thermal conductivity K , specific heat c , density ρ , and thermal diffusivity κ of the rolling element do not depend on the temperature.

As for assumption 2), the heat flux distribution is always steady if we fix our visual point on the moving heat source (=contact portion) in our case. And the motion continuity of long period will correspond to the state that a temperature distribution shape is in static. As the solution to basic Eq. (4) for thermal conduction that satisfies these conditions, temperature rise θ at a given point $B(x, y, z)$ can be expressed as Eq. (5).

$$\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial z^2} = \frac{1}{\kappa} \frac{\partial \theta}{\partial t} \quad (4)$$

$$\theta = \frac{1}{4(\pi\kappa)^{3/2} \rho c} \int_{-a}^a \int_{-R}^R \int_0^\infty \frac{q_0}{\zeta^{3/2}} \times \dots \exp\left(-\frac{(x-x'-V\zeta)^2 + (y-y')^2 + z^2}{4\kappa\zeta}\right) d\zeta dx' dy' \quad (5)$$

Note that $\zeta = t - t'$.

Applying Laplace transformation to the time-integration part results in the following:

$$\theta = q_0 \frac{\kappa}{\pi K V} \int_{Y-A}^{Y+A} \int_{X-L}^{X+L} \frac{\exp\left(-\sqrt{u^2 + v^2 + Z^2} - u\right)}{\sqrt{u^2 + v^2 + Z^2}} du dv \quad (6)$$

Where,

$$L = \frac{VR}{2\kappa}, \quad X = \frac{L}{R}x, \quad Y = \frac{L}{R}y, \quad Z = \frac{L}{R}z, \quad A = \frac{L}{R}a$$

$$u = \frac{L}{R}(x - x'), \quad v = \frac{L}{R}(y - y')$$

Equation (6) is the three-dimensional temperature estimation equation. In terms of the configuration of this equation, the first term q_0 indicates heat flux in the heat source mesh, and the other terms consist only of rolling velocity and thermal properties at the rolling surface also inside the integration. That is, under the same test conditions, it can be seen that the integral value in Eq. (6) can be applied commonly to all meshes by changing only heat flux volume in the first term.

Figure 6 shows calculation area and a calculation example. Both the length and width of the contact ellipse were divided into 50 equal portions, and an area for checking residual temperature behind the ellipse was also provided. Figure 6 also shows an example of a temperature variation graph in which Eq. (6) was applied to a localized heat source